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ANALYSIS OF A HEAT PUMP SYSTEM BASED ON BOREHOLE HEAT EXCHANGERS FOR A SWIMMING POOL COMPLEX IN KRYNICA, S-POLAND

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ABSTRACT

The economical possibilities of using heat pumps for space and pool water heating in a swimming pool complex to be built in a spa resort in Krynica, S-Poland are explored. This analysis includes statistical calculations of air temperature data, load, heat demand, heat sources for peak heating loads, and calculations of profitability of selected alternative heating designs based on a heat pump system. Some calculation methods for designing borehole heat exchangers are shown, some data from the experience of utilisation are described and methodology for finding the optimal limits between basic heat load and peak load examined.

1. INTRODUCTION

Krynica is a small town (about 15,000 inhabitants) located in a valley between Beskid Sadecki and Beskid Niski in the Carpathian Mountains, near the Poland-Slovakia border. It is the most famous spa in South Poland.

Because of the health resort character of the town, it is important to keep the air as clean as possible. One of the possibilities considered is the use of geothermal energy. Despite about 200 years of mineral water exploitation and drilling of many wells we have, unfortunately, very little knowledge about the geological conditions in that area. Across the area runs a very deep fault, which probably reaches down to the platform basis of flysch Carpathian. It stretches from SW-Slovakia to NE-Poland. Besides this there are a lot of smaller, younger crosswise running faults. Their connection with volcanic rocks is indicated by the appearance of a large number of CO_2 springs. It is probable that hot water and gases move along the deep fault planes (face of faults) (Sokolowski et al., 1996).

It is not currently possible to use the deep geothermal wells due to insufficient data about the geological conditions. The project design presented here is based on heat pumps, which use borehole heat exchangers as a low-temperature heat source. It is essential to drill deep wells to investigate the geological conditions and the geothermal water. If the results of the investigations show that the water is unsuitable for exploitation, drilled wells will be used for borehole heat exchangers. This solution makes it possible to recognize geological and geothermal conditions and utilise boreholes, irrespective of the results of the exploration.

The use of a heat pump system, based on borehole heat exchangers for space and pool water heating in the swimming pool complex, will make it possible to reduce annual running costs as well as the emission of gases from burning conventional fuels. This project can demonstrate the possibilities of utilising geothermal energy in connection with heat pumps. The use of ground-source heat pumps in connection with drilling for geothermal water will bring profits independent of the results of exploration. The research data will also be very important for future development of geothermal energy for heating in the Krynica region and in other health resorts, towns and places in the Polish Carpathian Mountains where the heating season is long and conventional fuel is difficult to deliver (for example to mountain huts/hotels).

2. ANALYSIS OF HEATING DEMANDS

Economical efficiency of heating systems depends on investment costs and running costs. Investment costs depend on the total capacity of the system. Running costs are dependent on the kind of fuel. Independently, both of them depend on local air temperature conditions.

2.1 Temperature data analysis

Operating a heating system is primarily dependent on climatic conditions. Krynica spa is located in the Carpathian Mountains, about 600 m over sea level. There are big differences between daytime and night temperatures. Snow cover holds from the middle of November to the end of March. The distribution of mean monthly temperatures is shown in Figure 1.

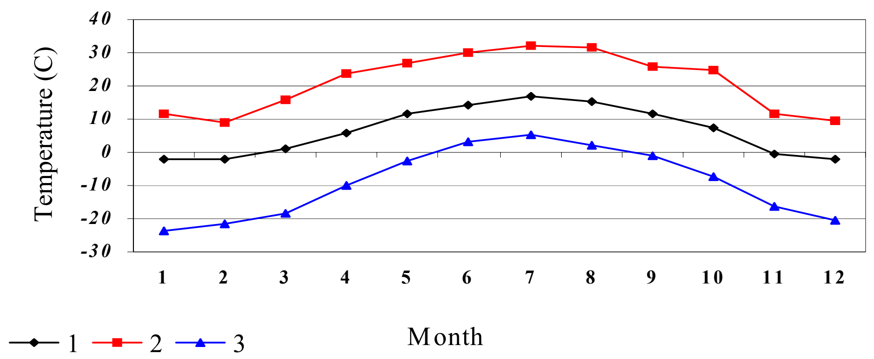

FIGURE 1: Distribution of monthly temperatures; 1) Mean monthly temperature; 2) Maximum monthly temperature observed in study period; 3) Minimum monthly temperature observed in study period

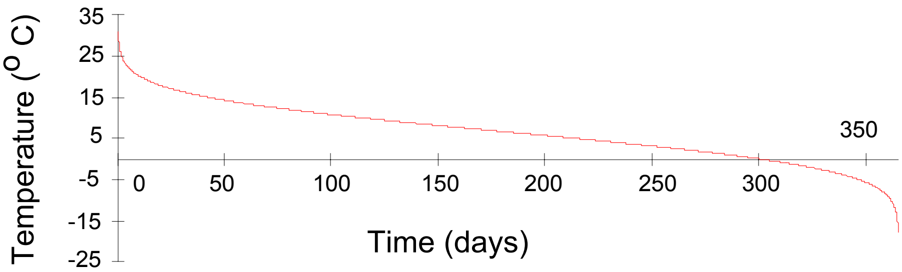

FIGURE 2: Annual temperature duration curve for Krynica

Arranging the days of the year according to the outside temperature results in a diagram like the one shown in Figure 2. This is called a temperature duration curve. An annual temperature duration curve was calculated as normal distribution. The annual mean temperature in the Krynica resort is about 6.5˚C with a standard deviation of about 7˚C. The curve also shows that the utilization time of the peak load is very short.

2.2 Load and heat consumption calculations

The steady-state rate of heat loss of a building is given by Equation 1

$$Q_h = k(T_i - T_o) \tag{1}$$

where: Q_h = Heat loss [W];
k = Overall heat transfer coefficient of building [W/°C];
T_i = Inside temperature [°C];
T_o = Outside air temperature [°C].

Because heat loss is the system load and depends on air temperature, i.e. a function of time, one can write

$$Q_h = \frac{dQ}{dt} \tag{2}$$

So we have the heat loss of buildings to the surroundings

$$dQ = k(T_i - T_o)dt \tag{3}$$

Thus, the annual heat consumption is obtained by integrating Equation 3 over one year

$$Q = k\int_a (T_i - T_o)dt \tag{4}$$

where Q = Annual heat consumption [J];
t = Time [s];
a = One year period.

For continuous, independent-of-air-temperature heat generation (for swimming pool or floors in a swimming pool complex), a constant load can be found using Equation 5

$$Q = Pt \tag{5}$$

where P = Continous power of heating [W];
t = Operating time [s].

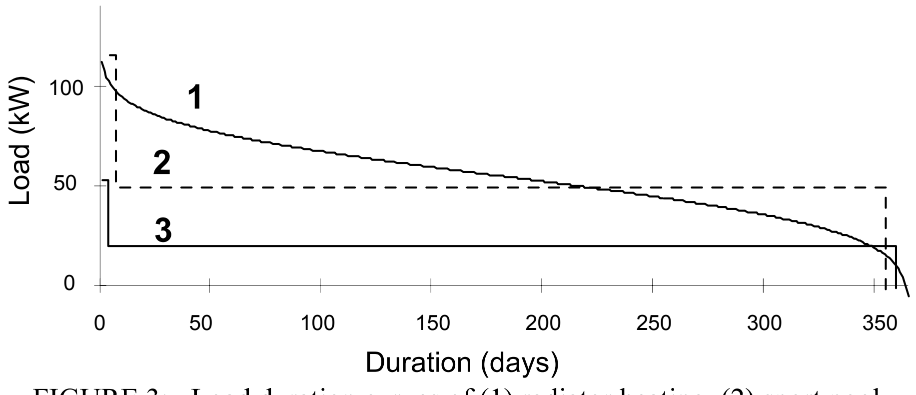

FIGURE 3: Load duration curves of (1) radiator heating, (2) sport pool and (3) small pools, in a swimming pool complex

Radiator heating peak load is calculated as heat loss of the building at the design outdoor temperature (minimum temperature). The value depends on many factors including the overall heat transfer coefficient of the building (k in Equation 1). It depends on the shape of the construction, wall and roof surfaces, thickness and kind of wall materials, window area, shading, mean wind velocity, etc. The peak load value of radiator heating, according to this project, amounts to 138.34 kW. This value was calculated according to Polish Standards computed air temperature for the Krynica zone, and the required inside temperature for swimming pool rooms. Calculated using Equation 1, the value of the overall heat transfer coefficient (k) amounts to 2,943 W/°C. Figure 3 represents the load duration curve of radiator heating in swimming pool rooms. Variations of temperature and load within each day are omitted, since the diagram is drawn by using a normal distribution of annual air temperature according to the temperature duration curve.

The area below the curve represents the heat consumption. Annual heat consumption for radiator heating, calculated according to Equation 4, amounts to 1,724 GJ.

2.3 Basic and peak load value evaluations

Utilization of geothermal energy requires considerable investment, which increases with peak load capacity. Therefore, a less expensive method is often used to obtain the extra power necessary during cold spells. This extra power is often termed a peak load. Due to its short time of use, the price of the power produced during peak loads is less important. Gas-fired boilers can be used as peak power generators because the investment cost per unit output is usually much less than for a geothermal one.

Annual operating costs of a heating system, including debt service, can be calculated by Equation 6.

$$K_a = S_g P_g r + K_g + S_b P_b r + K_b \tag{6}$$

where K_a = Annual cost of utilization [USD];
K_g = Annual cost of geothermal source heat generation [USD];
K_b = Annual cost of boiler heat generation [USD];
S_g = Basic investment cost of geothermal heating [USD/W];
S_b = Basic investment cost of boiler heating [USD/W];
r = Interest rate of bank credit (discount rate, annual investment and operating cost as a percentage of investment).

Basic investment costs S_g and S_b are approximated as the relationship between total cost and total capacity by a nonlinear regression of market costs of projects. The annual cost of heat generation (operating cost) depends on the basic cost of fuel (gas, oil, electricity) and on the amount of heat needed. That cost is given by Equation 7.

$$K = QC \tag{7}$$

where K = Annual operating cost [USD];
C = Basic cost of heat unit [USD/J];
Q = Heat demand [J].

The basic cost of a heat unit depends on the efficiency of the generator. This efficiency is dependent generally on the load of the generator. Efficiency can be estimated in accordance to the load duration curve and data from the manufacturer. Total efficiency of the system is also dependent on heating system's temperature. The value of efficiency grows inversely with temperature. The total air-temperature-dependent heating system demand is computed by Equation 8, or as a sum of heat generated by every heat source, shown in Equation 9.

$$Q_t = k \int_0^{t_{\max}} (T_i - T_o(t))\, dt \tag{8}$$

$$Q_t = Q_g + Q_b \tag{9}$$

where Q_t = Total demand of heat [J];
k = Overall heat transfer coefficient [W/˚C];
t_{max} = Longest of heating season [s];
t = Time [s];
T_i = Required inside temperature [˚C];
$T_o(t)$ = Duration of outside (air) temperature [˚C].

Peak load heat demand (from boiler, Q_b) is given by Equation 10 and the basic heat demand (from a ground-source heat pump, Q_g), according to Equations 8, 9 and 10 is calculated by Equation 11 (Figure 4):

$$Q_b = k\int_0^{t_x}(T_i - T_o(t))\,dt - k(T_i - T_o(t_x))\,t_x \tag{10}$$

$$Q_g = k\int_0^{t_{\max}}(T_i - T_o(t))\,dt - Q_b \tag{11}$$

where t_x = Duration of peak load heating [s].

The factors in Equation 6 are dependent on the variable t_x, shown in Figure 4. Optimum values of peak load and basic load of air-temperature-dependent heating are computed as values which comply with the optimal value of peak load duration t_o. The value of t_o can be numerically estimated by solving Equation 12. Total annual cost factors are functions of t_x according to the above equations and regression functions of basic investment and basic heat unit costs.

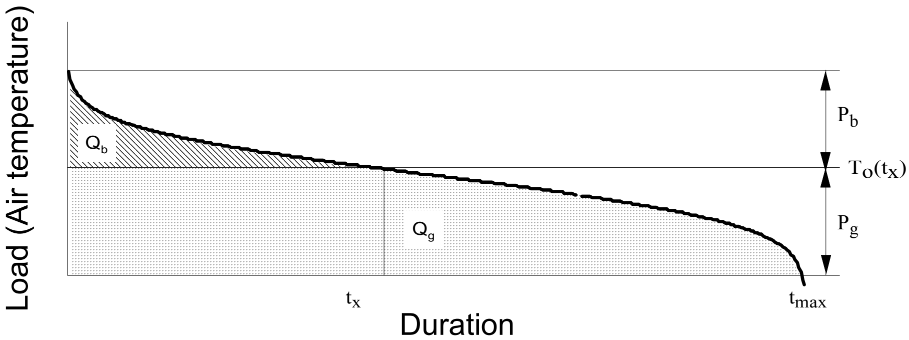

FIGURE 4: Distribution of peak load and basic load as variables of duration of heat generation by peak load heat boiler; symbols are explained in the text

$$\frac{dK_a}{dt_x} = 0 \tag{12}$$

3. GROUND-SOURCE HEAT PUMPS

The ground-source heat pump system (GSHP) is based on the same principle as the home refrigerator, but it can move heat in either direction. In ground-coupled applications, the GSHP system consists of a heat pump connected to a long, closed loop of pipe, which is buried in the earth, either in a borehole or in a trench outside the building. In the winter, heat is removed from the earth and delivered to the building (heating mode). In the summer, heat can be removed from the building and delivered to the earth for storage (air-conditioning mode). In either cycle, tap water can be heated, too.

3.1 Heat-pump mechanics

When a gas is compressed without loss of heat, its temperature and pressure increase because of the work done on the gas by the compressor. Conversely, when a gas expands, its temperature and pressure decrease. By compressing and/or cooling a gas sufficiently, it can be turned into a liquid. Liquids and gases are separately known as states of matter and together known as fluids.

The conversion of a liquid to a gas is called vaporisation or evaporation, and it takes place at constant temperature with absorption of heat from the surrounding environment. The heat absorbed goes into increasing the molecular kinetic energy. The amount of heat required to convert a unit-mass of liquid into vapour is called its heat of vaporisation. In a pan of water boiling on the stove, the liquid water remains at the boiling temperature while the heat input from the stove is taken up as heat of vaporization to convert the water to steam. For water, the amount of heat needed is 2.26 MJ/kg (539 cal/g) . When a gas reverts back into a liquid, a process known as condensation, it releases heat - its so-called heat of condensation – to the surroundings. The heat of condensation and the heat of vaporisation are equal for any given fluid, so the fluid gives up the same amount of heat upon condensation that it took up upon evaporation.

In a heat pump, a working fluid is contained in a closed, sealed circuit. The working fluid in today's ground-source heat pump is most often R-22, a common refrigerant which is gaseous at room temperature and which the EPA considers to be reasonably safe. As the working fluid moves around the circuit, it is repeatedly expanded and evaporated in one part of the system, causing cooling and absorption of heat, and compressed and condensed in another part of the system, causing warming and release of heat. The effect is to move, or pump, heat from the part of the system where the fluid is vaporised to the part of the system where the fluid is condensed. Figure 5 illustrates this idea. The compressor does work on the gas, increasing its temperature and pressure. The hot, high-pressure gas flows into the condenser, where it gives up heat to its surroundings by first cooling as a gas and then condensing, releasing its heat of condensation. The fluid exiting the condenser is a cooled, high-pressure liquid. An expansion device releases the pressure on this liquid, which flows into the evaporator as a low-pressure vapour/liquid mixture. In the evaporator, the heat of vaporisation of the liquid is absorbed and the resulting low-pressure gas flows back to the compressor, completing the circuit. The heat that appears at the condenser is the sum of the heat absorbed at the evaporator and the heat added by the compressor through the work of the motor running the compressor. Because electricity is used only to run the compressor, the GSHP delivers 3 to 4 times more energy than it consumes in the form of electricity.

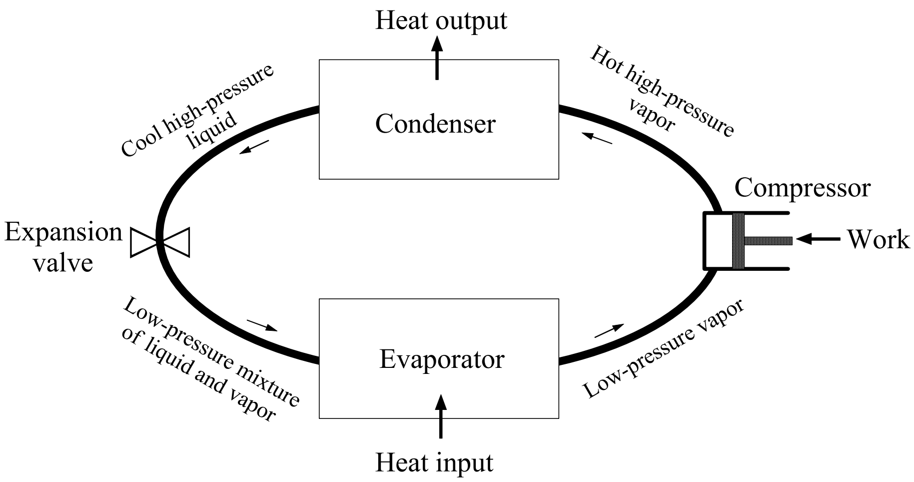

FIGURE 5: Schematic diagram of a heat pump

The heat pump has two heat exchangers. The primary heat exchanger operates between the ground loop and the working fluid, whereas the secondary heat exchanger operates between the working fluid and the building heating circulation fluid or air. It can also have a reversing valve so that the primary heat exchanger can be used as either the evaporator (when the GSHP is operating in its heating mode) or as the condenser (when the GSHP is operating in its cooling mode). In the heating mode, the hot, high-pressure refrigerant vapour exiting the compressor is routed to the secondary heat exchanger, operating as the condenser, and the blower moves fluid/air across this heat exchanger. The moving fluid/air is heated and circulated throughout the home or building. The cooled, high-pressure liquid refrigerant passes through the expansion valve and into the primary heat exchanger, where it evaporates and picks up heat from the water in the ground loop. It then moves back to the compressor to complete the cycle. In the heating mode, the heat available for heating the air/fluid at the secondary heat exchanger is the sum of the heat picked up from the ground loop at the primary heat exchanger and the heat added to the working fluid by work done in the compressor.

In an air-conditioning mode of operation of the GSHP, the reversing valve is placed in its alternate setting, and the hot, high-pressure refrigerant vapour exiting the compressor is routed to the primary heat exchanger, where it gives up heat to the water in the ground loop. It is then expanded into the secondary heat exchanger, where it absorbs heat from the air moving across it, thereby producing cool air for circulation throughout the home or building. The working fluid then returns to the compressor to complete its cycle.

On either the heating or the cooling cycle, heat can be routed to a water heater to furnish hot water for the home or building. All water-heating requirements can typically be furnished during the cooling season, where heat from the home is effectively pumped into the hot water tank. In winter, some supplementary water heating may or may not be necessary, depending on the system design. We should mention here, that GSHP systems can be used solely for heating water for a swimming pool or for tap water when requirements are very large, such as in a laundry (Wright, 1999).

3.2 GSHP properties

Ground-source heat pump systems are increasingly popular in both residential and commercial applications. There are a number of reasons for this, such as energy conservation, lower operating costs and a more balanced demand on the electric utilities.

Heat pumps have been identified as an efficient and economical alternative to the conventional heating and cooling systems. They have attracted a lot of attention and interest in recent years. A heat pump is a device which transfers heat from a low-temperature medium (source) to a high-temperature one (sink). When operated to provide heat (e.g. for space or water heating), the heat pump is said to operate in the heating mode; when operated to remove heat (e.g. for air-conditioning), it is said to operate in the cooling mode. Below are defined two of the most important coefficients that characterise a heat pump's energetic efficiency, Seasonal Performance Coefficient (*SPC*) and Coefficient of Performance (*COP*):

$$SPC = \frac{Heat\ delivered\ per\ season}{Electrical\ energy\ for\ heat\ pump\ and\ fluid\ circulation\ per\ season} \tag{13}$$

$$COP = \frac{Heating\ capacity}{Power\ input} \tag{14}$$

The theoretical value of *COP* can be calculated using Equation 15. Real efficiency is about 50% lower.

$$COP = \frac{T_c}{T_c - T_e} \tag{15}$$

where T_c = Temperature of condensing;
T_e = Temperature of evaporating.

The ground is an attractive heat source and heat sink, since its temperature remains near constant throughout the year, except in Polish conditions for the upper 15-27 m (Plewa, 1994). It is a thermally more stable heat exchange medium than air, essentially unlimited and always available. The high initial costs of the ground-source heat pump system are the major drawbacks, and have moderated the acceptance of these systems. However, as the price of electricity, and concerns over global environmental pollution have increased, more intensive interest has been shown in the ground-source heat pump system. Moreover, advancing technology in drilling and fabrications of ground-loop heat exchanger components make ground-source heat pump systems more and more cost effective. Also, because these systems are protected from harsh outdoor weather conditions, they tend to be more durable with lower maintenance requirements than conventional heat pump systems with exposed compressor units.

The ground-loop heat exchanger used in conjunction with a closed-loop ground-source heat pump system consists of a system of long plastic pipes buried vertically or horizontally in the ground. The heat is extracted from or rejected into the ground along the buried pipes. A fluid, such as water or brine, is circulated through the pipes, transferring thermal energy to or from the ground and the object. The largest cost element in utilising geothermal heat is the cost of financing, that is, interest and depreciation on capital investment. This cost is generally dependent on the maximum load or capacity of the project.

When energy is used at an even load, utilisation time is long and the operation is economically favourable since the cost of financing is distributed over a large number of energy units. Examples are swimming pools. The use of geothermal energy with large variations in load, for example space heating, is a different matter. In that case, the load requirements vary widely since they depend on seasonal weather conditions, and since the utilisation time of maximum load is shorter and investment cost higher than in the first example.

For a space (domestic) heating system, the utilisation time of maximum load can be very short. One of the methods available to increase the number of utilisation hours in heating systems is the use of peak power from other energy sources, requiring much less investment than geothermal heat production. The heat content of the ground can be trapped by groundwater heat pumps, shallow horizontal coils or by vertical earth heat exchangers (borehole heat exchanger, BHE). The latter are small, not central systems, ideally suited to supply heat to smaller objects like single-family or multifamily houses, schoolhouses, etc. They can be installed in nearly all kinds of geologic media. In the depth range 30-150 m, the original temperature field is governed by the thermal conductivity structure of the ground and by the geothermal heat flux. The latter can be influenced by flowing groundwater.

The BHE-heat pump system consists of two main circuits. The primary circuit (BHE probe - evaporator side of heat pump), and the secondary circuit (condenser side of heat pump - heater). Of prime importance is the temperature level in the primary circuit, which depends upon a number of parameters. They are natural temperature fields in the ground, total BHE probe length, probe diameter, fluid circulation rate, ground thermal conductivity, heat transfer between ground and probe and (if present) flowing groundwater.

A heat pump operates most effectively (higher COP) when the temperature difference between the heat source and heat sink (distribution system) is small. This is the reason that the heat distribution temperature for space heating heat pumps should be kept as low as possible during the heating season. Table 1 shows typical *COP*'s for heat pumps operating in various heat distribution systems. Sensible energy economics call for SPC > 3.0. This can be achieved with low-temperature heating systems where the delivery temperature is maximum 45˚C. By optimising BHE and heat pump design and using heating supply temperatures around 35˚C, *SPC*'s of the order of 4.0 can be achieved (Rybach and Hopkirk, 1995).

TABLE 1: Typical COP of heat pumps for systems with different distribution/return temperature

Heat distribution system	Supply/return system temperature	*COP*[1)]
Conventional radiators	60/50˚C	2.5
Floor heating	35/30˚C	4.0
Modern radiators	45/35˚C	3.5

[1)] Water-to-water heat pump with the temperature of the heat source at 5˚C and the heat pump Carnot efficiency 50%

Conventional radiator heating systems require high distribution temperatures, typically 60-90˚C. Today's low-temperature radiators and convectors are designed for a maximum operating temperature of 45-55˚C, while 30-45˚C is typical for floor heating systems.

3.3 GSHP classes

GSHP systems are divided into three classes: ground-coupled systems, groundwater systems and surface water systems. Ground-coupled systems use a closed loop of mostly plastic pipe for coupling to the earth. The groundwater system is an open-loop system in which groundwater from a well is delivered to the heat pump, and then disposed of in a second well or on the surface. The surface water system either uses a closed loop in a body of surface water or uses the surface water directly. The GSHP unit is coupled to the earth by liquid circulating in loops of pipe. A small pump furnishes enough loop circulation for most home installations. Somewhat larger circulation pumps are required for commercial installations, but loop circulation power is never a major factor compared with energy savings from the use of GSHPs. Available pipes are made from polybutylene or high-density polyethylene with joints thermally fused on-site by trained technicians. Most configurations can be classified as one of the following types.

a) **Closed vertical loop (borehole heat exchanger)**: Four inch diameter holes are drilled into the ground to depths of 30-150 m and piping with U-tube at the bottom is inserted into the holes, which are then grouted. This method is somewhat more expensive than other configurations, but has the advantage of being useful where space is limited or where soil moisture is inadequate at shallow depth. The loop temperature is also more uniform year round at deeper levels, and so less pipe length is required and the performance of the heat pump system is improved (Figure 6).
b) **Closed vertical loop**: Configuration like above, but the loop is built as a coaxial tube. Outside pipe often is steel pipe. That kind of borehole heat exchanger can work with a heat pump especially if utilising old abandoned wells or if the borehole is deeper (Figure 6).
c) **Closed horizontal loop**: Narrow trenches are dug to a depth of 1.5-3 m and loops of pipe are placed in the trench, which is then backfilled. A new installation, which requires less trench length, termed the "slinky" after the popular toy, consists of coils of pipe placed in the trench. Trench installations work well where the soil is moist year around and where enough space is available for their construction.
d) **Open vertical loop**: Pumps circulate groundwater from one well through the GSHP heat exchanger and either discharge the water on the surface or inject it back into the ground using a second well. There is no change in the quality of groundwater - only heat has been added or extracted. The open loop method has been used successfully where water is plentiful and injection can be readily obtained.
e) **Closed pond loop**: A closed pipe-loop is submerged in a pond, which is used as a thermal source or sink. This is the least expensive type of installation and works well in areas where the pond does not freeze completely in the winter (Wright, 1999).

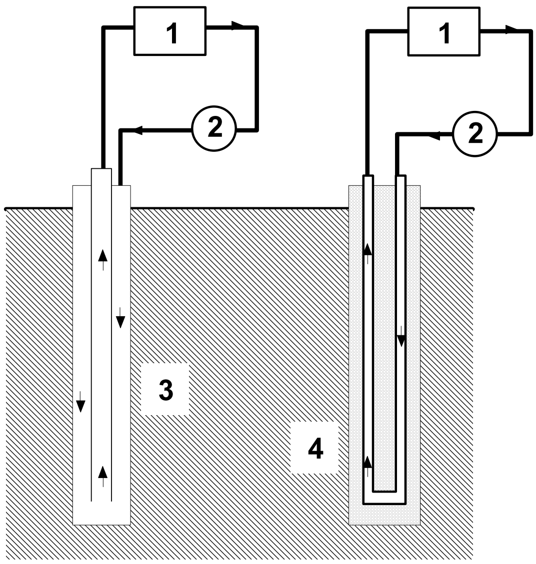

FIGURE 6: Diagram of borehole heat exchangers, 1) Heat pump unit; 2) Circuit pump; 3) Coaxial-tube BHE; 4) U-tube BHE

In the project for a swimming pool complex in Krynica, only the configuration based on borehole heat exchangers (closed vertical loops) will be considered.

4. BOREHOLE HEAT EXCHANGERS

The borehole (vertical) heat exchanger (BHE) is a simple device providing a closed circuit for a fluid to take heat from the first tens/hundreds of metres of ground and to feed the cold side (evaporator) of a heat pump. BHEs can be installed in most geological materials. Different types of BHE have evolved using coaxial tubes or U-tubes inserted into the ground in backfilled drillholes. Probe dimensions up to 10 cm diameter are the most common (Rybach et al., 1990; 1992).

BHEs are useful for exploitation of deep geothermal energy and surface solar heat energy. In Polish geological conditions, atmospheric influences are clearly visible in the depth range 0-15 m. Maximum depth of influences of annual changes of air temperature amount to 27 m (E, NE and NW-Poland) (Plewa, 1994). Generally, below 15 m the geothermal heat fluxes dominate. On average, the BHE supplies a peak thermal power of about 45 W per metre length (Rybach and Hopkirk, 1995).

There are several reasons for the BHE boom. The most important is economic considerations and environmental concerns. At present, installation costs of a BHE system are higher than for a conventional boiler. However, annual operating costs (electricity for heat pump and circulation pump) are considerably less than for a boiler's fuel. So the return time of the relatively high investment for a BHE is definitely shorter than the lifetime of the heating system itself. The increase of gas and oil prices is anticipated for the foreseeable future. This would also call for oil and gas-independent or fewer dependent heating systems like the BHE. The argument most frequently heard for BHEs is that homeowners want to install an environmentally benign heating system. In particular, the emission of combustion products like CO_2 can be avoided. In addition, there is no risk of local groundwater contamination, which could happen with leaking oil heaters. The planned introduction of a CO_2 emission tax is a further argument to reduce CO_2-emissions by selecting a BHE heat pump system.

Deep BHEs can provide whole residential areas with heating energy. These could be installed either in specially drilled holes, or alternately in abandoned deep drillholes. There exist many deep boreholes, otherwise unused, which would apparently lend themselves to development as borehole heat exchangers and consequently as a new type of heat source for heating. These fall into three categories: old research or exploration holes; failed "dry" holes drilled in the search for hydro, gas or oil resources and old production wells in spent oil and gas fields. In such cases, the drilling costs have usually been written off. New investment for their re-use will, therefore, be restricted to eventual cleaning out costs plus the costs of completion as a closed circuit heat exchanger. Provided that a suitable heat consumer is nearby, the viability of this undertaking will depend upon the rate and temperature level at which heat can be extracted over a long period.

4.1 BHE system design

The heat transfer between the ground and the fluid circulated in the BHE probe is influenced by several factors. These are the physical characteristics and velocity of the fluid, the total surface available for heat exchange, the properties and thickness of the probe tube and the quality of contact between probe and ground ("backfill").

A further important design parameter is the size of the heat pump compressor, evaporator and condenser. If these are well matched to cope with the maximum heat demand of the consumer, a relatively wide range of acceptable BHE designs is available. It should be noted that the choice of BHE type and length have effects on the power required for circulating the ground source circuit heat exchange fluid.

A lot of design parameters influence the performance of the borehole heat exchanger. Among these are the length of the BHE, diameter, minimum spacing, borehole heat exchanger tube and backfill materials, fluid circulation rate, as well as site-specific factors like topographic altitude, ground thermal conductivity structure, groundwater characteristics, etc.

Around a shallow BHE, the formation temperature around a borehole heat exchanger at greater depth will sink during heat extraction by the circulating fluid. The intensity of drawdown and, hence, the source temperature of the energy delivered, depend upon the rate of heat extraction from the fluid circuit and the rate of thermal replenishment of the near field. The rate of replenishment depends, in its turn, largely upon the thermal conductivity of the geological formations penetrated by the borehole, but also locally upon the advective replenishment by flowing groundwater. It becomes apparent that a choice range of operational design parameters exists in any given case between a higher source temperature at a lower energy extraction rate and a lower source temperature at a higher energy extraction rate. Here the nature of the heat consumer is decisive in quantifying the range of choice.

The probe performance can be characterised by the specific yield or heat extraction rate per unit probe length (W/m). Specific heat extraction rate coefficients of BHE for the effects of varying the design parameters are specified in Table 2.

TABLE 2: Summary of the parametric runs (Rybach et al., 1990)

Case	BHE probe configuration	Average thermal conductivity of the site	Specific heat extraction rate of BHE	Seasonal performance coefficient
		(W/mK)	(W/m)	-
1	Base case: 4×50 m probe length (double 1” diameter U-tubes)	2.0	39.29	3.060
2	Base case with slightly raised site thermal conductivity	2.5	41.31	3.104
3	Base case with strongly raised site thermal conductivity	3.0	42.44	3.128
4	4×100 m double U-tubes	2.0	25.10	3.271
5	3×50 m double U-tubes	2.0	47.65	2.976
6	4×50 m (8.2 cm diameter) coaxial probes	2.0	39.26	3.060
7	4×100 m coaxial probes	2.0	25.26	3.278
8	2×50 m coaxial probes	2.0	59.54	2.812
9	2×100 m coaxial probes	2.0	42.09	3.120
10	4×50 m coaxial probes with doubled diameter (16.4 cm)	2.0	46.64	3.209

The thermal load of the BHE depends mainly on the thermal conductivity of the surrounding ground (Table 3). These systems operate by conduction, i.e. there are no formation fluids produced. The energy supply for the heat exchanger comes from several sources: the vertical geothermal heat flux itself, the import of heat horizontally by conduction, advective transport with groundwater if present, and the compensating heat exchange between the ground surface and the atmosphere.

TABLE 3: Borehole heat exchanger performance in different rock types
(single BHE, 150 m deep) (Rybach, 1998)

Rock type	Thermal conductivity (W/mK)	Specific extraction rate (W/m)	Energy yield (kWh/ma)
Hard rock	3.0	Max. 70	100-120
Unconsolidated rock, saturated	2	45-50	90
Unconsolidated rock, dry	1.5	Max. 25	50

Some models are shown in Appendix I. Numerical calculations make it possible to calculate seasonal performance coefficient (SPC), which is the proportion of delivered energy to electrical energy demand for circulation pump and heat pump during the entire season. To consider long-term temperature change in a geological formation and for detailed calculation of running costs, the use of simulation based on theoretical models is necessary. In this case long-term operation has not been simulated. There exist simplified methods for design and calculation of BHE and the possibility of receiving a power, but only numerical simulation can show the operating cost as a function of time.

4.2 BHE system properties

Flowing groundwater improves the heat extraction efficiency by advecting heat to the probe in addition to pure conduction. Ground temperatures around a BHE at a site with flowing groundwater are constantly higher than in dry ground. The thermal drawdown is much less pronounced and recovery much more effective.

If several BHE systems are to be installed in the same area, their spacing must be increased correspondingly. This fact also has legal implifications (neighbour rights, minimum distance from property boundaries). In problematic cases the recharge of the extracted heat by a simple solar collector during the summer represents a viable and attractive solution (Hopkirk et al., 1985). Yearly solar recharge reduces the radius of a single annual cycle and reduces correspondingly the minimum spacing between multiple BHE. Such a solar-coupled BHE represents a small but efficient heat storage system. Effects on the biosphere are negligible since the general temperature level in the uppermost 10 metres is buffered by exchanges with sun and atmosphere.

Some open questions remain. First, the best type of material to be used for backfilling the BHE probes. It must ensure good contact between probe and ground under all conditions, during extraction and during storage of heat and during repeated temperature excursions below 0˚C. It is important from a practical point of view that the material can also be pumped for easy and reliable BHE installation. A mixture of bentonite/quartz sand/concrete could be best suited for this purpose (Rybach et al., 1990). Second, BHE groups should be considered in more detail, especially with respect to their possible interaction with groundwater. Finally, combined heat extraction/heat storage schemes should also be investigated.

Typical fluid temperatures within the borehole tubes run between -1 and 35˚C. Heat extraction or rejection between the heat exchanger and the surroundings takes place by pure heat conduction. The heat exchanger as a closed loop formed in a U-shape is the most common and has the advantage that heat extraction may take place even at temperatures below -1˚C if an antifreeze mixture is used. After the exchanger is installed the rest of the space in the borehole is filled again, usually with grout. The grout maintains a good thermal contact between the borehole wall and the pipes.

One problem that designers are faced with is finding a good estimate of the soil parameters such as the thermal resistance, capacity, etc. Geological data provide a large range for each of these parameters. Usually the average value of these parameters is used in simulations. Even then the error might be significant. To get more accurate results using any of the models (Appendix I), experimental methods of computing the site soil properties are needed.

5. DESIGN OF A BHE SYSTEM FOR HEATING A SWIMMING POOL COMPLEX IN KRYNICA

Building of a swimming pool complex in Krynica is planned in the near future. The expected location makes it possible to drill shallow wells and apply a BHE heat pump system as the heat source for the swimming pool complex. Independently it is possible to use groundwater as a heat source for heat pumps. Investigation of the groundwater resource is necessary. Implementation of the project will not disrupt the wealth of Krynica (mineral waters).

5.1 Designing values, assumptions and data

The basis for the study presented in this report is design work already done for the swimming pool complex in Krynica without using heat pumps. The design and economic calculations presented are also based on geological data, prior experience with BHE heat pump systems, and actual prices of energy and drilling of shallow wells.

The swimming pool complex will be built in the Czarny Potok district. It is located southwest of the Krynica dislocation in the Krynica zone. The geological layers belong to the Zarzecze formation. This formation consists of thin-bedded, fine-grained calcareous sandstones. They are separated by mudstones. The formation also consists of a thick bank of sandstones and conglomerate blocks, which belong to the Krynica element. The sandstones and conglomerate are weak-compacted by a mudstone binder (Chrzastowski et al., 1992). The required length of BHE for heat extraction, according to the geological profile and data from Table 3, is assumed to be 15 m/kW. It is assumed that groundwater flow does not exist.

The cost of drilling shallow wells is estimated as 25 USD/m. Costs and properties of heat pumps made in Poland are shown in Table 4 according to producer data. VAT for heat pumps equals 22%.

The above data has been used to statistically estimate the parameters in the following equation, which describes the basic investment cost of a BHE heat pump system; the correlation coefficient equals 0.978:

$$C_{ehp} = a + bP_{hp} + \frac{c}{P_{hp}^{2}} \tag{16}$$

where C_{ehp} = Basic investment cost of BHE's heat pump system [USD/W];
P_{hp} = Heat pump system capacity [W];
Coefficients: $a = 0.56762055$;
$b = -6.1357709 \times 10^{-7}$;
$c = 10724287$.

TABLE 4: Technical parameters and costs of heat pump units made in Poland

Nominal conditions capacity (0˚C/+50˚C)[1] (kW)			Guaranted heat capacity (kW)				Net price (USD)
Heating	Cooling	Power demand	-10/+50˚C	-5/+50˚C	0/+50˚C	+5/50˚C	
7.7	5.9	2.25	5.2	6.4	7.7	9.1	2,485
8.5	6.3	2.6	5.8	7	8.5	10	2,508
11.1	8.3	3.35	7.5	9.3	11.1	13	2,540
12.5	9.5	3.7	8.3	10.3	12.5	15.2	2,700
16.4	12.3	4.9	11	13.5	16.4	19.6	2,745
20.5	15.5	6.1	14	17	20.5	24.3	3,770
24	18.3	7	16	19.9	24	28.7	4,343
31.6	24	9.25	21.7	26.2	31.6	37.7	5,033
32.8	24.6	9.8	22	27	32.8	39.2	5,200
39.6	30.1	11.6	27.2	33.1	39.6	47.3	6,963
41	31	12.2	28	34	41	48.6	7,008
48	36.6	14	32.1	39.8	48.1	57.4	7,288
63.2	48	18.5	43.4	52.4	63.2	75.3	8,325
79.2	60.2	23.2	54.4	66.1	79.2	94.6	9,500

[1] Evaporation/condensing temperatures of the working fluid, respectively

For estimation of extra investment costs of a BHE heat pump system in comparison with gas-fired heating a similar formula for basic boiler costs has been calculated using statistical regression (Equation 17); the correlation coefficient for this equals 0.995:

$$C_{eb} = a - b \times e^{-c P_b^{\,d}} \tag{17}$$

where C_{eb} = Basic investment cost of boiler [USD/W];
P_b = Boiler capacity [W];
Coefficients: $a = 0.51843822$;
$b = 0.47460825$;

c = 114949.94;
d = -1.3417268.

For running cost calculations, assumptions were made regarding the basic cost of heat generated by a heat pump and a gas boiler. Taking into consideration the average efficiency of boiler as 0.9, the cost of gas amounted to about 0.14 USD/m^3, and with a gas heating value according to Polish Standard of 24 MJ/m^3, cost of heat from a boiler amounts to 6.37 USD/GJ. Heat costs for a heat pump have been calculated based on the kind of a consumer according to Table 1. Calculated heating costs for a heat pump with COP equal to 4.0 is 4.01 USD/GJ. A similar value for COP equal to 3.5 is 4.58 USD/GJ. In that calculation, it has been taken into consideration that the price of electricity is different at different times of the day. The weighted average value of energy cost amounts to 0.058 USD/kWh.

5.2 Specification of the investment object

Table 5 shows the elements of the heating system in the swimming pool complex (less tap water) with heating demands. The data comes from the project of a swimming pool complex. For radiator heating, a medium-temperature system has been changed to a low-temperature system. Suppling a low-temperature system with heat generated by heat pump is much more efficient.

TABLE 5: Components of total power (except tap water) in the swimming pool complex

List of consumers	Heating power (kW)	Operation time (days)	Heat consumption (GJ)	Working temperature (˚C)
Sport pool				
- continuous load	50	353	1,525	28
- peak load	136	3	35	
Small pools				32
- continuous load	18	356	554	
- peak load	53	2	9	
Floor heating	50	356	1,534	45/35
Radiator heating	138	Depending on outside temperature	1,724	45/35
Total	**377**	**-**	**5,381**	**-**

5.3 Proposed system design

A simplified diagram of a heating system using heat pumps is shown in Figure 7. Such a solution makes it possible to use heat pumps for base heating and a gas-fired boiler for peak loads (in short periods of time). This system can also use a gas-fired boiler as a spare or emergency heating source.

The capacity of the heat pump for radiator heating was chosen as the optimum value resulting from calculations described in Chapter 2. The main factors influencing the optimum capacity are local air temperature conditions and also investment cost as the basic cost of a BHE heat pump system, basic cost of boilers, electricity and gas costs, experience of heat extraction dependent on geological conditions etc. The optimal capacity of a BHE heat pump is 33 kW (24% of peak load). Average annual heat generation of that load is 997 GJ (58% of total heat demand). Figure 7 shows the designed schedule of heat exchangers and the duration of heat exchanger peak loads.

Pool heating will require a separate heat pump for the pool. Beyond this, the heating capacity of the heat pump will likely be less than that of a typical gas-fired heater in the same application. This is because heat pumps cost about five times what gas-fired pool heaters do per unit of heating capacity. The smaller

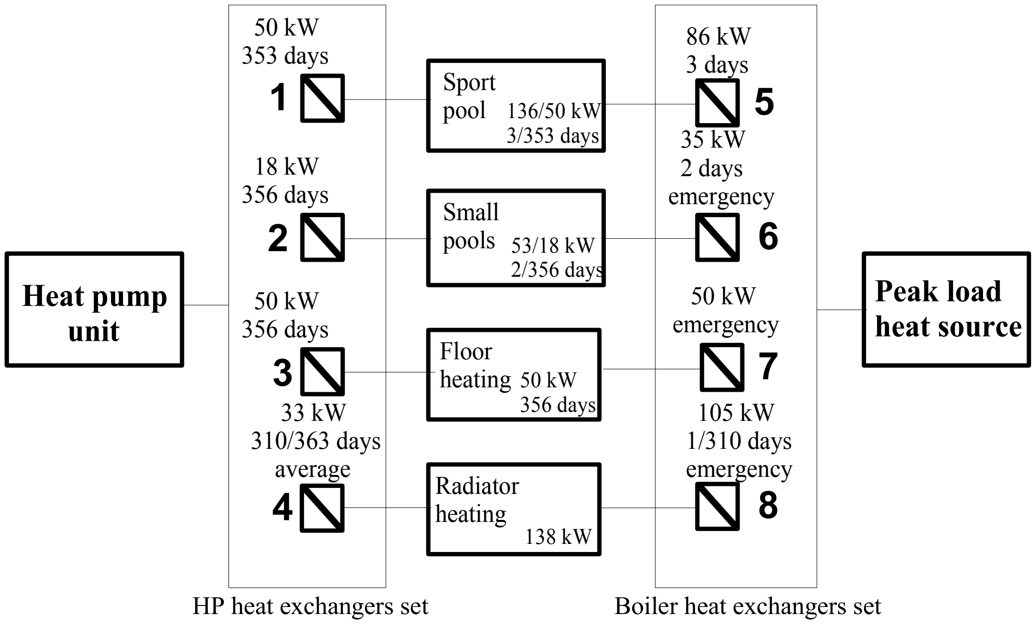

FIGURE 7: Diagram of a heating system for the swimming pool complex

heat pump would not affect the ability to maintain pool temperature, but would result in longer time required to bring the pool temperature from cold up to usable temperatures at the beginning of the season. Thus, the choice of heat pump capacity for pool heating is based on base load only and continuous heating. An additional peak load boiler will be installed for heating the pool water in start-up periods.

5.4 Alternative designs

To find the economically optimal solution, 16 different designs were considered (see Table 6). These are all of the 15 possible combinations of the four heat pump heat exchangers shown in Figure 7 plus one extra alternative with the heat pump for radiator heating designed for peak load (138 kW) instead of optimum capacity (33 kW).

TABLE 6: Sets of heat exchanger power (heat exchanger no. as in Figure 7)

Design no.	**Heat pump capacities (kW)**				**Total (kW)**	**Boiler capacities (kW)**				**Total (kW)**
	1	**2**	**3**	**4**		**5**	**6**	**7**	**8**	
1	50	18	50	138	256	86	35	-	-	121
2	50	18	50	33	151	86	35	-	105	226
3	50	18	50	-	118	86	35	-	138	259
4	50	18	-	-	68	86	35	50	138	309
5	50	-	-	-	50	86	53	50	138	327
6	-	18	-	-	18	136	35	50	138	359
7	-	-	50	-	50	136	53	-	138	327
8	-	-	-	33	33	136	53	50	105	334
9	-	-	50	33	83	136	53	-	105	294
10	-	18	50	33	101	136	35	-	105	276
11	-	18	50	-	68	136	35	-	138	309
12	50	-	50	-	100	86	53	-	138	277
13	50	-	-	33	83	86	53	50	105	294
14	-	18	-	33	51	136	35	50	105	326
15	50	18	-	33	101	86	35	50	105	276
16	50	-	50	33	133	86	53	-	105	244

The heating capacity values for the swimming pool complex (Table 5) have been used to calculate the annual heat demand for each part of the heating system. The results of these calculations are shown in Table 7.

TABLE 7: Balance sheet of heat capacity and annual heat demand for swimming pool elements complex

Element of heating system	Heat capacities (kW)		Annual heat demand (GJ)	
	Base load	Peak load	Base load heating	Peak load heating
Sport pool	50	136	1,538	1,560
Small pools	18	53	557	563
Floor heating	50	50	1,534	1,534
Radiator heating	33	138	997	1,724
Total	151	377	4,626	5,381

Table 8 shows the annual heat generation for each of the designs, divided between different heat sources, that is heat pump heat exchangers and boilers.

TABLE 8: Balance sheet of annual heat generation from every heat exchanger

Design no.	From heat pumps (GJ)				Total (GJ)	From boilers (GJ)				Total (GJ)
	1	2	3	4		5	6	7	8	
1	1,538	557	1,534	1,724	5,353	22.1	5.9	-	-	28
2	1,538	557	1,534	997	4,226	22.1	5.9	-	727	755
3	1,538	557	1,534	-	3,629	22.1	5.9	-	1,724	1,752
4	1,538	557	-	-	2,095	22.1	5.9	1,534	1,724	3,476
5	1,538	-	-	-	1,538	22.1	563	1,534	1,724	4,033
6	-	557	-	-	557	1,560	5.9	1,534	1,724	5,014
7	-	-	1,534	-	1,534	1,560	563	-	1,724	3,847
8	-	-	-	997	997	1,560	563	1,534	727	4,384
9	-	-	1,534	997	2,531	1,560	563	-	727	2,850
10	-	557	1,534	997	3,088	1,560	5.9	-	727	2,293
11	-	557	1,534	-	2,091	1,560	5.9	-	1,724	3,290
12	1,538	-	1,534	-	3,072	22.1	563	-	1,724	2,309
13	1,538	-	-	997	2,535	22.1	563	1,534	727	2,846
14	-	557	-	997	1,554	1,560	5.9	1,534	727	3,827
15	1,538	557	-	997	3,092	22.1	5.9	1,534	727	2,289
16	1,538	-	1,534	997	4,069	22.1	563	-	727	1,312

5.5 Cost-effective and optimal alternate designs

Investment and operating cost calculations are based on up-to-date prices of heat pumps, boilers, drillings, gas and electricity. These are shown in Table 9. As investment costs for the analyses, only heat pump unit, boiler and drilling (BHE) costs were considered. This simplification is possible when comparing systems based on low-temperature heating: A BHE heat pump system with peak load boiler (every alternative design) and a gas-fired boiler heating system. Basic costs of heat pump and boiler power units depend on the capacity of each of them. Functions of basic costs were calculated by nonlinear regression based on actual prices for different power heating units. Operating costs take into consideration the influence of supply water temperature on *COP*-value in different heating systems as shown in Table 1.

5.5.1 Cash flow analysis

Table 9 shows calculated investment and operating costs for each of the different designs considered. It clearly shows that the cost varies considerably between the alternatives, especially the investment cost. One reason for this is different financing of the investments. The cash flow analysis presented in Table 10 are based on comparison with gas-fired heating. They show that high investment costs will, in general, result in lower operating costs and, thus, increased annual savings compared to gas-fired heating systems. Table 10 shows also the simple pay-back time (SPBT), which is the proportion between extra investment costs and annual savings.

TABLE 9: Investment and operating costs for the 16 different designs

Design no.	Capacity of HP unit (kW)	Capacity of boiler (kW)	Total investment cost (USD)	Annual heat generation (GJ)		Annual running cost (USD)		
				From HP	From boiler	From HP	From boiler	Total
1	256	121	111,492	5,353	28	22,448	178	22,627
2	151	226	83,820	4,626	755	19,119	4,809	23,928
3	118	259	70,611	3,629	1,752	14,552	11,160	25,713
4	68	309	50,196	2,095	3,286	8,401	20,932	29,333
5	50	327	42,114	1,538	3,843	6,167	24,481	30,648
6	18	359	27,047	557	4,824	2,234	30,728	32,962
7	50	327	42,059	1,534	3,847	6,151	24,505	30,657
8	33	334	33,837	997	4,384	4,566	27,926	32,492
9	83	294	56,680	2,531	2,850	10,718	18,155	28,872
10	101	276	64,067	3,088	2,293	12,952	14,606	27,557
11	68	309	50,143	2,091	3,290	8,385	20,957	29,342
12	100	277	63,587	3,072	2,309	12,318	14,709	27,027
13	83	294	56,731	2,535	2,846	10,733	18,130	28,863
14	51	326	42,664	1,554	3,827	6,800	24,377	31,178
15	101	276	64,115	3,092	2,289	12,967	14,581	27,548
16	133	244	76,237	4,069	1,312	16,885	8,358	25,243

TABLE 10: Schedule of cash flows and simple pay-back time

Design no.	Extra investment costs (USD)	Annual savings (USD)	Simple pay back time (Years)
1	94,282	11,650	8.1
2	66,610	10,349	6.4
3	53,401	8,564	6.2
4	32,986	4,944	6.8
5	24,904	3,629	6.9
6	9,837	1,315	7.5
7	24,849	3,620	6.9
8	16,627	1,785	9.3
9	39,470	5,405	7.3
10	46,857	6,720	7
11	32,933	4,935	6.7
12	46,377	7,250	6.4
13	39,521	5,414	7.3
14	25,454	3,099	8.2
15	46,905	6,729	7
16	59,027	9,034	6.5

5.5.2 Net present value analysis

Net present value (*NPV*) of each alternative design was calculated basically by using four different ways of financing the project. These are (1) financing total cost with own money, (2) by commercial credit (20% of interest rate) and 3 year repayment time, (3) by commercial credit and 7 year repayment time and (4) by special ecological credit disposed by Polish national foundation for environmental protection. For each of these, two different discount rates have been used, actual (10%) and social discount rate (assumed 4%) (Appendix II). The results after 20 years of operation are shown in Table 11. The Table also shows the pay-back time of different designs for every considered way of financing.

TABLE 11: *NPV* and payback period calculation results by different kind of financing and discount rates after 20 years of operation

Financing	**Commercial bank credit**				**Ecological credit**		**Own money only**	
Time of repayment	**3 years**		**7 years**		**7 years**			
Interest rate	**20%**		**20%**		**8%**			
Part of project costs	**-**		**-**		**50%**			
Maximum amount	**-**		**-**		**50,000 USD**			
Design number	**Discount rate**							
	10%	**4%**	**10%**	**4%**	**10%**	**4%**	**10%**	**4%**
	***NPV* (USD)**							
1	-4,024	43,810	-23,394	11,186	-115	50,857	4,904	64,051
2	15,189	59,737	-147	34,768	15,778	61,741	21,497	74,037
3	14,456	51,528	1,045	30,212	13,222	51,429	19,513	62,993
4	5,983	27,126	-4,452	11,457	1,933	23,777	9,107	34,207
5	3,637	19,075	-5,620	5,641	-1,528	14,439	5,996	24,421
6	425	5,919	-6,636	-3,347	-6,819	-1,115	1,356	8,031
7	3,619	19,017	-5,630	5,598	-1,554	14,372	5,972	24,351
8	-3,008	4,057	-11,059	-7,087	-9,315	-1,896	-1,433	7,627
9	2,807	25,510	-8,573	8,048	-349	23,194	6,545	33,984
10	5,913	34,406	-6,543	14,900	3,777	33,265	10,351	44,465
11	5,963	27,065	-4,465	11,411	1,905	23,708	9,081	34,135
12	10,952	42,192	-1,435	22,819	8,749	40,975	15,344	52,149
13	2,829	25,574	-8,558	8,097	-319	23,265	6,572	34,058
14	-1,478	11,203	-10,815	-2,383	-6,567	6,655	933	16,668
15	5,939	34,472	-6,525	14,953	3,810	33,339	10,381	44,542
16	12,297	51,080	-1,934	28,208	11,840	51,877	17,887	63,752
Pay-back period (years)								
1	-	12	-	17	-	11	17	9
2	12	10	-	13	12	9	10	7
3	12	10	19	13	12	9	10	7
4	13	10	-	15	17	11	11	7
5	14	10	-	16	-	12	12	8
6	17	11	-	-	-	-	14	9
7	14	10	-	16	-	12	12	8
8	-	15	-	-	-	-	-	11
9	16	11	-	16	-	11	13	8
10	15	10	-	15	16	10	12	8
11	13	10	-	15	17	11	11	7
12	12	10	-	13	13	9	10	7
13	16	11	-	16	-	11	13	8
14	-	12	-	-	-	15	18	10
15	15	10	-	15	16	10	12	8
16	13	10	-	13	13	9	11	7

5.5.3 Optimal heating selection

After looking at the results of the *NPV* calculations in Table 11, it can be seen that the best solution is design no. 2. The NPV for this design after 20 years of operation is the highest of all cases for the considered ways of financing except one, that is financing by commercial credit, 7 years repayment time and 10% discount rate. In that case, design 3 has the lowest *NPV*. Compared to design no. 3, the one no. 2 has an additional heat pump capacity of 33 kW, which is used for radiator heating.

Using the simple pay-back time from Table 10 as criteria for optimal solution, one would find design no. 3 to be the best as it has the shortest pay-back time (6.2 years). This is also the only alternative that has a positive NPV after 20 years of operation for all considered ways of financing. It assumes that the BHE heat pump system will generate heat for continuous water heating for all pools and floor heating. Annual heating generation equals 3,629 GJ, which makes up 67% of the total heating demand. The capacity of the optimal alternative equals about 118 kW, that is 31% of the total capacity demand. In second place when looking for profitability is design no. 2. A little lower profitability can be explained by the fact that this solution assumes a higher capacity with lower COP-value, because of the utilisation of a heat pump for base radiator heating. Lower value of COP results from higher temperature demand for the radiator heating circuit. With this solution, 78% (4,227 GJ) of the annual heat demand is met by BHE heat pumps, with capacity equal 151 kW (40% of peak load demand).

The least profitable alternative is using the heat pump as a heat source only for radiator heating (no. 8). The reason is partly explained above and also by the fact that basic investment cost per power unit increases with decreasing capacity of a BHE heat pump system. The same can be seen from the alternative solutions with low heat pump capacity (no. 5, 6, 7 and 14). By comparing designs no. 8, 6 and 14, one can see that cost saving effects are more related to the working conditions (COP) and operating time than to investment costs.

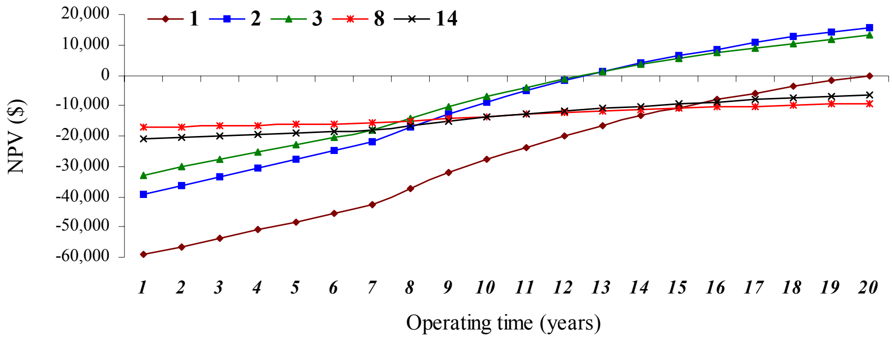

FIGURE 8: *NPV* of 1st, 2nd, 3rd, 8th and 14th designs with discount rate 10% and financing by special ecological credit

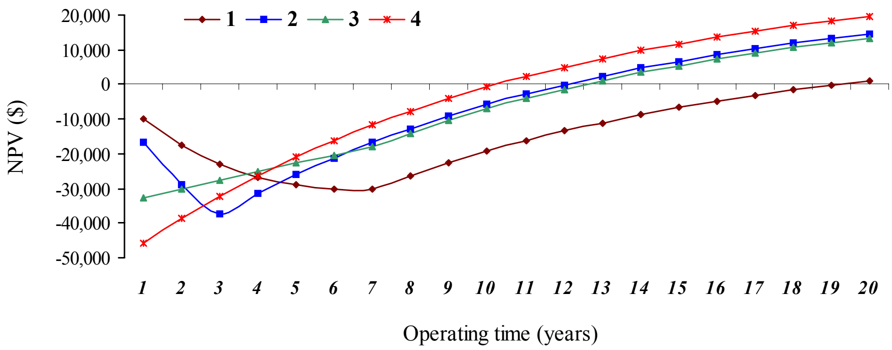

FIGURE 9: *NPV* of 3rd design with 10% discount rate and different financing of investment

Figure 8 shows the *NPV* for 1st, 2nd, 3rd, 8th and 14th designs during the 20 years of operation, financed with special ecological credit. Figures 9 and 10 show similarly *NPV* for the optimal solution (design no. 3) for different ways of financing and 10% and 4% discount rate, respectively.

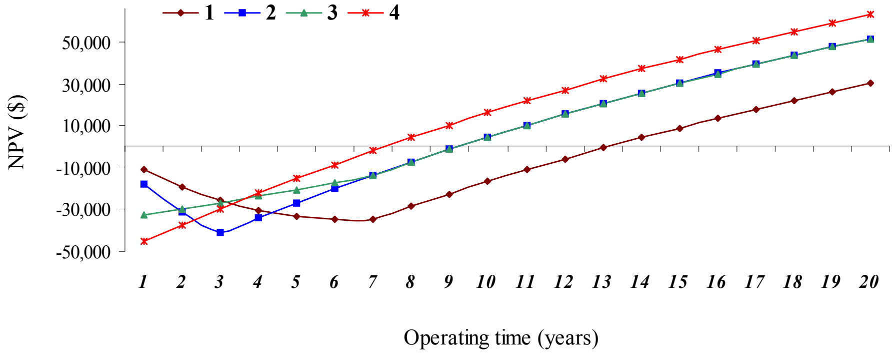

FIGURE 10: *NPV* of 3rd design with civil discount rate (4%) and different financing of investment

6. CONCLUSIONS

To investigate the possibilities of geothermal utilisation, the drilling of a deep well is needed. There are good geological conditions for geothermal water flow along the deep fault planes reaching the Moho-discontinuity (Franko et al., 1995) and other small faults. An already drilled well can be used for geothermal exploitation or reinjection if hot water is found, otherwise it might be possible to use it as a borehole heat exchanger. The drilling site should be close to the planned new swimming pool complex in Czarny Potok district. That location makes it possible to utilise the geothermal water for housing estates, for tourist and holiday camps or as a BHE for a swimming pool. The danger to the mineral water source does not exist in this area.

The use of heat pumps based on a BHE will be a good demonstration of utilisation of geothermal energy. It is important, especially taking into account the spa character of Krynica, where clarity and clean air are necessary for further positive development of the town. The performance of BHE heat pump systems should be calculated very carefully. The heat generation depends on a lot of factors like depth of borehole, diameter, kind of pipes, geological formations, lithology, hydrodynamics, natural regional heat flow, air temperature conditions, etc. The dynamic properties of heat generation should be calculated by numerical models. The use of simple empirical methods can be useful when taking experiences and local conditions into consideration.

Because of high investment costs of BHE heat pump systems, the profitability of such projects should be calculated as precisely as possible. In the calculations carried out, drilling costs are included as an integral part of the project (not use of any abandoned wells). The use of a BHE heat pump system for continuous water in pools and for floor heating as part of the heating system in a swimming pool complex is the most profitable (design no. 3). The second most profitable solution is a heat pump system like above with an additional heat pump for radiator heating (design no. 2).

Utilization of a geothermal source for radiator heating is less profitable than pool heating, because the efficiency values (*COP*) are lower. The reason for this is higher radiator system temperature. Similarly, the time of operation influences the efficiency. Very short times of peak load lead to the use of a gas-fired boiler as a peak load heat source. If the second most effective solution is chosen (design no. 2) a less expensive boiler should be used to obtain the extra power necessary for peak loads for pool water, heating it after water changes and during high demand, for radiator heating. Due to brief peak load usage, the price of the power produced during peak loads is less important.

An optimal design is the use of a 118 kW BHE heat pump as the basic heat source with a 260 kW gas-fired boiler for peak loads. Heat generation by basic geothermal heat source equals 3,629 GJ (67%). Heat from a peak load boiler equals 1,752 GJ. The high value of a peak load source is brought about by its capacity when needed for heating pool water after changing it. Extra costs of that heating system in comparison with a boiler only equals about 53,5 thousand dollars, with annual savings of about 8,5 thousand dollars. *NPV* by 10% of discount rate will be positive after 12 years with ecological credit financing and after 10 years by paying with own money. Adequate *NPV* with a social discount rate equals 9 and 7 years.

For further interpretation it is necessary to consider calculations only for the social discount rate. Utilisation of geothermal heating in a town spa and resort regions will not bring direct economical profits. Profitability will be visible, for example, in the number of visitors and in profits connected with their stay at the resort. The profit means clear air and new technology, which will lead to economical profits for the region.

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APPENDIX I: Borehole heat exchangers models

Size (total depth) of a vertical downhole heat exchanger has the biggest influence on the cost of the heating system based on a heat pump. Total depth depends on a lot of factors. Several models have been developed for the analysis and sizing of the ground-loop heat exchangers. The models are mainly based on the Kelvin's line source theory, cylindrical source theory, or numerical methods. Four major models are reviewed, and briefly summarised.

A. Simple line source model

The Kelvin heat-source theory is based on an infinitely long permanent line source of heat, with a constant rate of heat rejection on an infinite medium at an initial uniform temperature of T_o. Heat transfer between the borehole and soil is carried out by pure radial heat conduction for a perfect soil, borehole contact. Soil properties are considered constant and homogeneous. Groundwater movement is not considered in the model. The temperature at any point in the medium is given by Equation 1 (Ingersoll et al., 1954).

$$T - T_o = \frac{Q'}{2\pi k} \int_x^\infty \frac{e^{-\beta^2}}{\beta} d\beta = \frac{Q'}{2\pi k} I(X) \tag{1}$$

The integral is evaluated between *X* and infinity, where *X* is given as in Equation 2:

$$X = \frac{r}{2\sqrt{\alpha t}} \tag{2}$$

where T = Temperature in soil at any selected distance from the pipe;
T_o = Initial temperature of soil;
Q' = Heat emission of pipe (negative for absorption);
r = Distance from the centre line of pipe;
k = Thermal conductivity of the soil;
α = Thermal diffusivity of the soil;
t = Time since start of operations, hours;
β = Variable of integration.

Equation 1 mathematically defines the earth undisturbed temperature at a given radius. When Q' is non-zero, Equation 1 may be used to determine the change in temperature of the soil contacting the borehole after a given time of operation. This equation is applicable to both single and multiple horizontal and vertical heat exchangers and can be used to determine the thermal interference between boreholes in close proximity. The solution from each borehole is superimposed to get multiple borehole solutions.

One disadvantage of this model is that it does not consider the end effects of the borehole. The heat conduction is assumed to be radial only. For a long loop the assumption produces fairly good results. Another approximation is the modelling of borehole internal structure. It is modelled by an overall heat transfer coefficient, which is the reciprocal of the sum of the soil and pipe heat resistance. Finally notice that the line source model was developed based on a constant rate of heat transfer. For purposes of modelling the boreholes, the heat transfer rate is averaged over each month and the integral in Equation 1 is evaluated as the sum of integrals for each month.

B. Hart and Couvillion's line source model

This model (Hart and Couvillion, 1986) employs Kelvin's line source theory of continuous time-dependent heat transfer between the line source and the ground to derive a time-dependent temperature distribution around the line source, as given by

$$T - T_o = \frac{Q'}{4\pi k} \int_y^\infty \frac{e^{-\lambda}}{\lambda} d\lambda \tag{3}$$

where

$$y = \frac{r^2}{4\alpha t} \quad \textit{and} \quad \lambda = \beta^2$$

The superposition principal is applied to model thermal interference effects (superposition in space). The important aspect of this model is the introduction of the far-field radius, r_∞. Couvillion and Hart state that the ground temperature at a distance from the line source greater than r_∞ is assumed to be the far-field temperature. The value of this far-field temperature mainly depends on the length of time the line source has been operating, and the thermal diffusivity of the soil. The far-field radius is used to check for interference. It is computed by

$$r_\infty = 4(\alpha t)^{0.5} \tag{4}$$

This model has similar limitations to the simple line source model, except it accounts for thermal interference using the concept of far-field radius and far-field temperature.

C. Cylindrical source model

Kavanaugh's model is based on the cylindrical source theory to determine the temperature distribution or the heat transfer rate (Kavanaugh and Deerman, 1991). It is an exact solution for a buried cylindrical pipe of infinite length, and can be applied for either a constant pipe surface temperature or a constant heat transfer rate between the buried pipe and the earth. It assumes that the heat transfer between the borehole and soil with a perfect contact is pure heat conduction. The soil is considered to be an infinite homogeneous solid. The cylindrical solution for a constant heat flux is shown in Equation 5:

$$T - T_o = \frac{Q'}{kL} G(z,p) \tag{5}$$

and

$$z = \frac{\alpha t}{r^2} \quad and \quad p = \frac{r}{r_o}$$

where $G(z,p)$ = Cylindrical source integral;
r_o = Outer pipe radius;
L = Pipe length.

Kavanaugh also provides an equation to calculate the temperature difference across the closely positioned pipes for a non-uniform heat flux.

$$\Delta T_p = \frac{Q'}{CN_i 2\pi r_o L h_{eq}} \tag{6}$$

where C = The correction factor for non-uniform heat flow;
N_i = Number of U-tubes;
h_{eq} = Equivalent heat transfer coefficient per unit area.

Kavanaugh's model accounts for the short circuit heat transfer within the borehole due to the temperature difference between the channels of the U-tube. It also has the ability to calculate the fluid temperature entering and exiting the ground heat exchanger.

D. Eskilson's model

This model (Eskilson, 1987) is intended for sizing the vertical ground-loop heat exchanger. The mathematical formulation of Eskilson's model is governed by the heat conduction equation of the ground temperature in cylindrical co-ordinates:

$$\frac{1}{\alpha}\frac{\partial T}{\partial r^2} = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} \tag{7}$$

The initial and boundary temperatures are assumed to be constant. The heat extraction function, q(t) for N piecewise steps is given by Equation 8.

$$q(t) = \sum \{(q_n - q_{n-1}) \times He(t - t_n)\} \quad (q_o = 0) \tag{8}$$

By superposition, the time-dependent heat extraction step may be regarded as a sum of the basic extraction steps. Superposition of the step responses makes it possible to give a good description of the physical character of the heat extraction process. The simplified temperature functions are detailed below.

Temperature at the borehole wall:

$$T_b = T_{om} - q_1 \times R_q(t) \tag{9}$$

where T_b = Borehole temperature;
T_{om} = Undisturbed ground temperature;

q_1 = Extraction step;
R_q = Thermal resistance due to extraction step.

The thermal resistance R_q is defined as

$$R_q = \frac{1}{2\pi\lambda} g\left(\frac{t}{t_s}, \frac{r_b}{H}\right) \tag{10}$$

where λ = Thermal conductivity of ground;
$g(t/t_s, r_b/H)$ = The g-function;
r_b = Borehole radius;
t = Temperature;
t_s = Steady state temperature.

and the steady state temperature, t_s , as

$$t_s = \frac{H^2}{9\alpha} \tag{11}$$

where H = Borehole depth;
α = Thermal diffusivity of the ground.

A specific g-function represents a specific borehole configuration response to a unit step change in heat extraction or rejection. The g-functions are computed using the finite difference solutions to the finite line source differential equation, which are then superimposed. The term borehole configuration refers to the geometric arrangement of multiple boreholes. For example, nine boreholes in a square layout with a specific spacing between the boreholes is one configuration that has a unique g-function.

Mean fluid temperature:
At each depth H and time t, there is a local steady state process with heat flow between the pipe and the ground around the borehole. Variations of temperature along the pipes depend on the pumping rate. When the pumping rate is high the flow becomes turbulent so that the variation of temperature can be neglected. The correlation among borehole temperature, the mean fluid temperature and borehole resistance can be formulated as in Equation 12:

$$T_b - T_f = q \times R_b \tag{12}$$

where T_b = Average temperature at the borehole wall;
T_f = Mean fluid temperature;
q = Heat extraction step;
R_b = Borehole thermal resistance.

Temperature variation along the borehole:

$$T_{f,in} = T_f - \frac{qH}{2C_f\rho_f V_f} \tag{13}$$

$$T_{f,out} = T_f + \frac{qH}{2C_f\rho_f V_f} \tag{14}$$

where $T_{f,in}$ = Fluid temperature entering ground loop;
$T_{f,out}$ = Fluid temperature exiting ground loop;
q = Heat extraction;
H = Active borehole depth;
C_f = Fluid heat capacity;
ρ_f = Fluid density;
V_f = Pumping rate.

Eskilson's model can account for different load profiles due to different consumers, ground thermal conductivity, grouting, thermal interference effects of the nearby boreholes, and the effects of on/off equipment cycling on the heat transfer between the pipe and the ground. The main drawback of this model is that it does not have the capability to account for the varying conductivity, soil moisture, frozen or unfrozen soil state. Also, this model only offers fixed configurations; using these fixed configurations results in the surface area of the borehole field changing size every time the borehole depth is adjusted.

Ideally, a perfect model for analysis and design of ground-source heat pump systems should be able to predict how soil properties, soil moisture content, ground temperature distribution, piping materials and size, heat carrier fluid properties, and the heat pump model impact the performance and size of the ground heat exchanger. Also, the model should be able to account for the equipment on/off cycling, seasonal earth temperature variation, and heat exchanger surface temperatures below freezing. The above four models have different simplifications. Table 1 is the summary of the features of various models reviewed above.

TABLE 1: Features of various models

Model	Eskilson's model	Simple line source model	Couvillion and Hart's model	Kavanaugh's model
Analytical method	Numerical model using a line-segment source	Line-source	Line-source	Cylindrical-source
On/off cycling	yes	no	yes	yes
Soil moisture freezing	no	no	yes (approximate)	no
Thermal interference effects	yes	no	yes	yes
Thermal effect of grouting	yes	no	no	yes
Modelling of borehole internal structure	Borehole pipes modelled by an equivalent thermal resistance.	Borehole pipes modelled by an equivalent ther-mal resistance.	Borehole pipes modelled by an equivalent thermal resistance.	Borehole pipes approximated by an equivalent pipe diameter.
Edge effects	yes	no	no	no

E. Ground heat exchanger length

Two simplified methods of calculation for ground heat exchanger length L, depending on different factors are shown by Equations 15 (Kavanaugh, 1992) and 18 (Kavanaugh and Rafferty, 1997)

$$L = \frac{P(1-\frac{1}{\eta})\times(R_p+R_gF)}{T_g-T_{wa}} \tag{15}$$

where R_p = Thermal resistance of pipe per unit length of bore;
R_g = Thermal resistance of ground per unit length of bore;
T_g = Normal deep ground temperature;
T_{wa} = Average water temperature entering and leaving ground coil.

and

$$L = \frac{q_a R_{ga} + (q_h - 3.41\,W) \times (R_b + R_{gm} F + R_{gd} F_{sc})}{T_g - \dfrac{T_i + T_o}{2} - T_p} \tag{16}$$

where W = Power input at design heating load;
F_{sc} = Short-circuit heat loss factor;
q_a = Net annual average heat transfer to the ground;
q_h = Building design heating block load;
R_{ga} = Effective thermal resistance of the ground, annual pulse;
R_{gm} = Effective thermal resistance of the ground, monthly pulse;
R_{gd} = Effective thermal resistance of the ground, daily pulse;
R_b = Thermal resistance of bore;
T_g = Undisturbed ground temperature;
T_p = Temperature penalty for interference of adjacent bores;
T_{in} = Liquid temperature at heat pump inlet;
T_{out} = Liquid temperature at heat pump outlet.

These equations can only be used for very rough (estimate) calculation.

APPENDIX II: General theory of cost-effective calculations and optimal design selection

The basic methods of analysis, which rely upon the discounting of payments to adjust for differences in timing, are very simple. However, there are many different organizational contexts within which appraisals are carried out and these affect the way in which cost and earning streams are determined and, hence, their magnitudes.

A. Cash flow (CF)

The financial profile of any project consists of a series of positive and negative payments over time. The basic problem of financial/economic appraisal is to adjust the cash flow to an equivalent basis in time so that they can be compared with each other or with those of other projects. There are a number of ways in which this is done and a variety of indices can be formulated which measure aspects of the economic/financial value of a project.

B. Discount rate

The value of money is not dependent only on its actual worth but also on the time which is needed for consumption of profits. The profits result from ownership of the money in any form.

Present value PV of amount FV (future value) got in year j equals

$$PV_j = \frac{FV_{j'}}{(1+r)^j} \tag{1}$$

where r = Discount rate;
j = Number of years.

Discounting of cash flows is of great importance for economical evaluation of an investment. Investment costs and future profits disclose in different steps of execution of a venture. That is why discounting of the value over a specific time (year)is a necessity. It makes possible the comparison of values of spent/earned money in different years.

Discount rates can be specified in different ways. Good estimates of real discount rates are given by Equation 2:

$$r_a = \frac{r_c - i}{1 + i} \tag{2}$$

More careful approach is the following equation:

$$r_a = \frac{r_k - r_d}{1 + i} \tag{3}$$

where r_a = Real discount rate;
r_c = Interest rate of bank credits;
r_d = Interest rate of bank deposits;
i = Real inflation rate.

For estimating long-term investment ventures, which are important for a community, or for profits not shown directly, often calculations of a social discount rate are applied (2-4%).

C. Net present value (NPV)

NPV calculation is the dynamic method which takes into consideration the change of the money's value in time. The basis for calculation of net present value is the discounted values of annual cash flows. The calculation of net present value is performed in three stages. The individual payments are discounted by the appropriate amounts to determine their present values. Present value of earnings E_j in year j PVE, (USD) is calculated in Equation 4, and present value of costs K_j in year j, PVK_j (USD) in Equation 5. This reduces all costs and earnings to an equivalent basis in time.

$$PVE_j = \frac{E_j}{(1+r)^j} \tag{4}$$

where E_j = Earnings in year j [USD].

$$PVK_j = \frac{K_j}{(1+r)^j} \tag{5}$$

where K_j = Costs in year j [USD].

The total present values of earnings PVE, and of costs, PVK, (USD) are obtained by summation (Equations 6 and 7):

$$PVE = \sum_{j=1}^{j=n} \frac{E_j}{(1+r)^j} \tag{6}$$

and

$$PVK = \sum_{j=1}^{j=n} \frac{K_j}{(1+r)^j} \tag{7}$$

where n = Time [years].

The net present value, NPV (USD), of the whole project is calculated by taking the difference and subtracting any initial investment I (USD), as shown in Equation 8:

$$NPV = PVE - PVK - I = \sum_{j=1}^{j=n} \frac{E_j - K_j}{(1+r)^j} - I \tag{8}$$

If the net present value is positive at this discount rate, the project is viable. If it is negative then the project is non-viable. When comparing the net present value of two projects, it is rational to choose the higher (Harrison et al., 1990).