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GROUND SOURCE HEAT PUMP SIMULATION IN TUV PROVINCE, MONGOLIA

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ABSTRACT

Ground Source Heat Pumps (GSHPs) are a promising technology for decarbonisation of house heating in the cold climate of Mongolia. GSHPs are systems combining a heat pump with a system to exchange heat with the ground. A Natura BWH-280 90 kW heat pump was installed in the 'Ireedui' kindergarten with 1120 square metre heating area in Zuunmod village, Tuv province. It is located near Ulaanbaatar, the capital, some 43 km south at a 1200 metre altitude. This installation is used for reference in this study. The system consists of a ground source heat exchanger, water circulating pumps, solar collector, electrical heater, buffer tanks, vapour compression heat pump and measurement equipment. This paper reports the simulation for integrating the system components of a ground source heat pump system into a house. A thermodynamic model of the ground source heat pump system was constructed with Engineering Equation Solver (EES) software. A model of the well and the buffer tanks was implemented in MATLAB.

1. INTRODUCTION

1.1 Field description

A ground source heat pump was installed in a field located near the Ulaanbaatar capital of Mongolia, 43 km south in Zuunmod village of Tuv province. House heating is as important in the remote areas as it is in the capital, and the government of Mongolia is obliged to provide for her citizens, due to the diversity of responsive environments and the continent´s out and out climate. The Viessmann group of companies is one of the leading international manufacturers of heating equipment worldwide. Viessmann products have a high standard of quality throughout the entire product line that translates into operational reliability, energy savings, environmental friendliness and operational comfort. The Natura BWH-280 90 kW ground source heat pump was installed in the 'Ireedui' kindergarten by National Renewable Energy Centre (NREC) and Mon-Ugsralt Company, and financed by the Government of Mongolia. NREC has been involved in research and designing of different kinds of renewable energy systems, particularly for the rural areas in Mongolia.

The goal of this project was to build a cost effective, customer satisfying and environmentally friendly independent heating system. This was to be an efficient heating system to heat both public and private buildings, using a ground source heat pump and solar collector as its power source.

The heat pumps already installed in the kindergarten will be replicated in a hospital and a school. The project, already implemented at the kindergarten, shows the government and the general public that it is possible to solve heating needs of the houses using heat pumps. The location of Zuunmod village centre in Tuv province is shown in Figure 1.

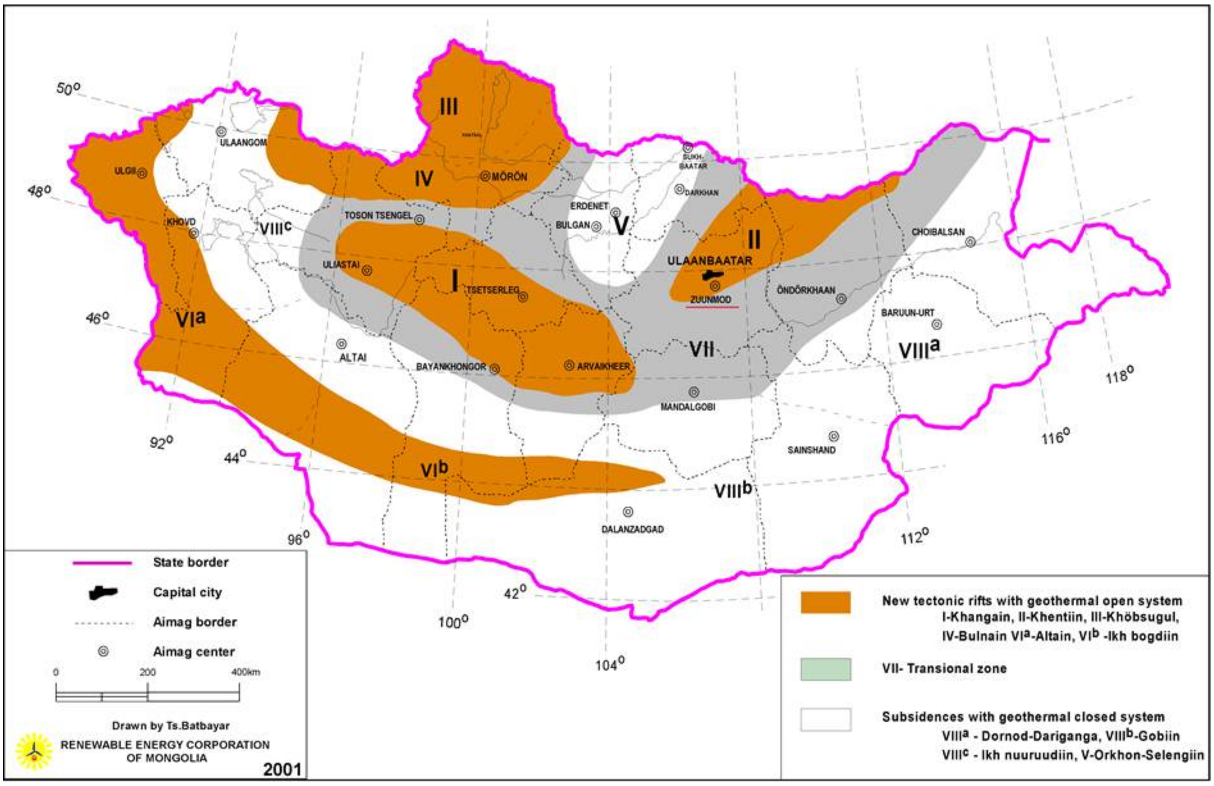

FIGURE 1: The location of Zuunmod village centre in Tuv province and the main geothermal structures of Mongolia (Ministry of Agriculture and Industry of Mongolia, 1999; Geodesy and Cartographical Institute, 2000

It is expected that the heat pumps will offer great solutions to the reduction of air pollution and an increase of houses supplied with heat. This will result in the creation of environmentally friendly living conditions in both urban and rural areas.

2. STUDY DESCRIPTION

2.1 General overview

Heat pumps are generally more expensive to purchase and install than other heating systems, but they save money in the long run in some areas because of lower heating bills. Despite their relatively higher initial costs, the popularity of heat pumps is increasing.

The term "heat pump" refers to a group of technologies that transfers heat from a low temperature to a high temperature. This requires a thermodynamic input in the form of either work or heat. This is made clear in the Clausius statement of the second law of thermodynamics which states that: "*It is* *impossible for any system to operate in such a way that the sole result would be an energy transfer by heat from a cooler to a hotter body."* The main point here is to focus on the energy transfers between the cycle and its surroundings. It is possible to transfer heat from a low to a high temperature while supplying only heat as the driving energy.

The 90 kW system modelled in this study was designed to use two 750 l buffer tanks of hot water, taken from 24 wells with wellhead pressures of about 2.6 bar. Four 100 m long, 32 mm diameter pipes were installed in each well. The boreholes, 150 mm in diameter are staggered at 8 m distance from one another. The system was combined with a solar collector system (Figure 2), but this report does not discuss the solar collector system. The cooled brine is recycled back using a return pipe.

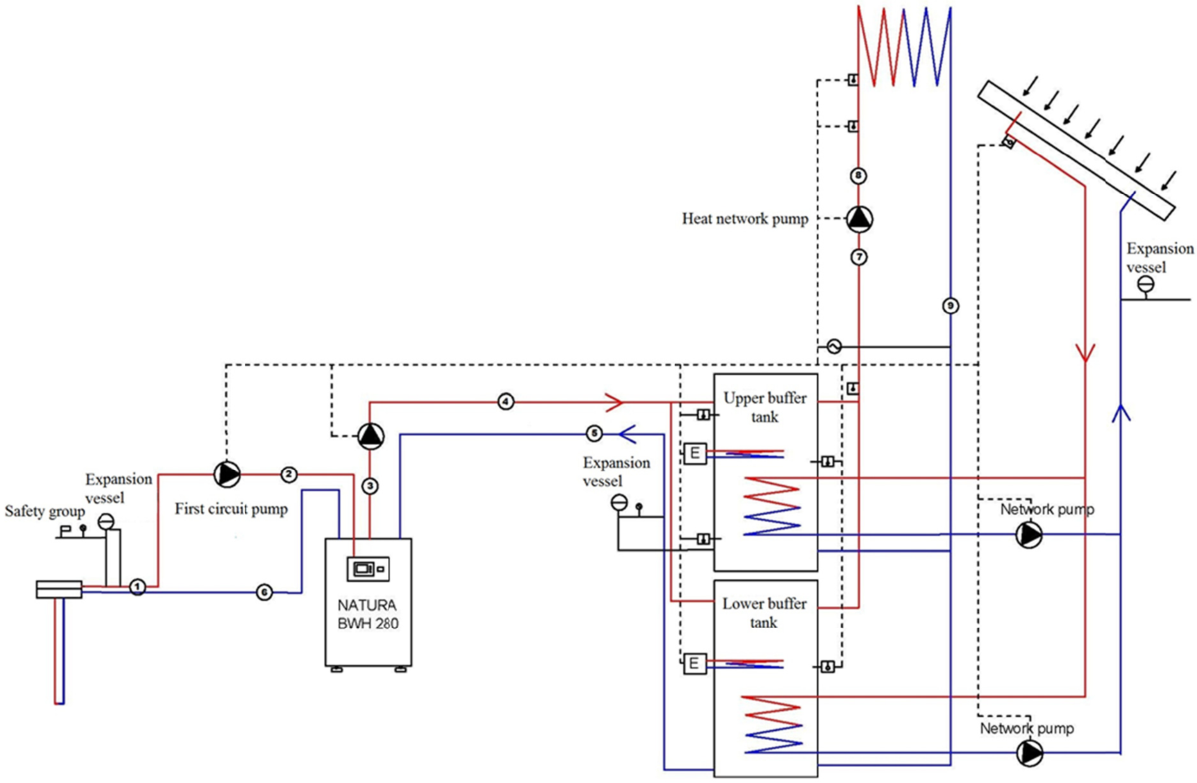

FIGURE 2: Schematic of the ground source heat pump system in the kindergarten, Zuunmod village (National Renewable Energy Centre, 2009)

The building system has been simplified for this report. It is assumed that the system has one buffer tank without a solar collector, as shown in Figure 3.

2.2 Objective of the study

A thermodynamic model of the ground source heat pump system was constructed to analyse house heating using Engineering Equation Solver (EES) software. Buffer tanks and building heat load models were implemented in MATLAB.

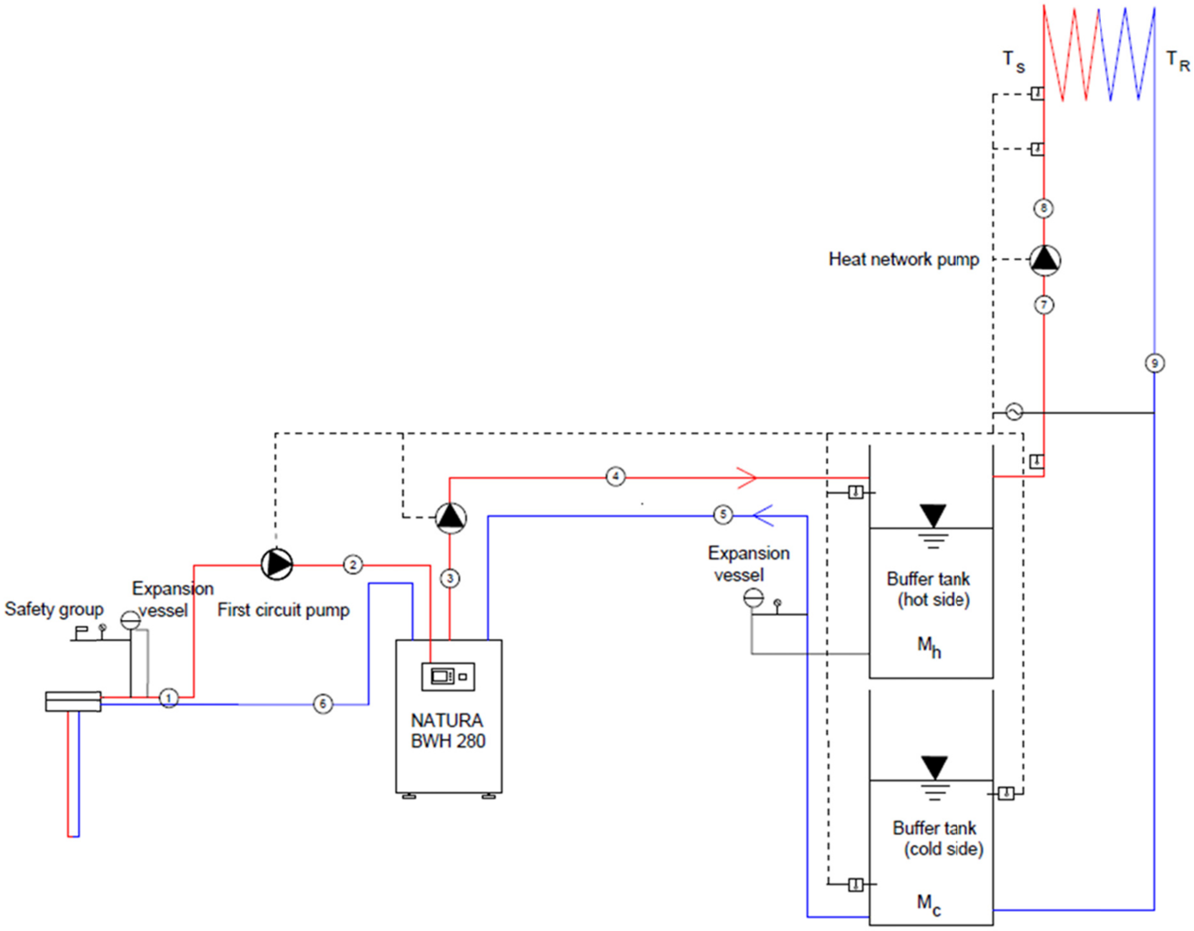

FIGURE 3: Simplified schematic of the ground source heat pump system in the kindergarten

3. METHODOLOGY

3.1 General information and assumptions

EES is an acronym for Engineering Equation Solver. The basic function provided by EES is the numerical solution of non-linear algebraic and differential equations. In addition, EES provides built-in thermodynamic and transport property functions for many fluids, including water, dry and moist air, most CFC and HCFC refrigerants, and others. The software is used to make thermodynamic property calculations and to run the models for each operating condition (Herold et al., 1996).

MATLAB is a high-level technical computing language and interactive environment for algorithm development, data visualization, data analysis, and numeric computation.

Fluid properties
In order to analyse a power cycle based on a specific working fluid, the thermodynamic properties of the fluid must be known. These properties involve (Pálsson, 2009):

- Pressure p, given in Pa, kPa or bar absolute. Note that 1 bar = 10^5 Pa.
- Temperature T, given in °C or K.
- Enthalpy h, which is measurement of the energy contents of a unit mass flow of fluid. A frequently used unit for enthalpy is kJ/kg.

- Entropy s, given in kJ/(kg K). The change in entropy will be zero if a process generates or absorbs work without exchanging heat with the environment.
- Specific volume, v in m^3/kg. Rarely used, but can be used for calculating flow speed based on pipe diameters and mass flow.

3.2 Overview of theory

The heat and mass balances for the system studied are calculated after classifying the variables involved. The inputs are variables which represent factors which cannot be controlled and have to be accepted as they are. The design variables represent factors related to the design of the plant, and can be decided by the designer or the operator. The outputs are as the name implies results of the mathematical model used.

Inputs to the model:
- Outdoor temperature;
- Source fluid inlet temperature;
- Source fluid outlet temperature.

Design variables:
The variables related to the conversion are as follows:
- System water supply temperature;
- Indoor temperature;
- Water quantity in storage.

Output:
- Return water temperature;
- Mass flow;
- System heat load.

The streams in and out of the system have four flow properties: mass, heat capacity, enthalpy and exergy. The mass conservation is obvious, no mixing of the source and cooling streams are assumed. The heat capacity is important for the characteristics of the heat conversion, and will be treated here as a heat capacity flow, the product of fluid heat capacity and flow rate. The product of enthalpy relative to the environmental temperature and the flow rate defines the heat flow in and out of the system. The exergy will give information on the work producing potential of the system, and is calculated in the same way as the enthalpy (Valdimarsson, P., 2011). The heat flow ($\dot{Q}_h$) is given by:

$$\dot{Q}_h = c_h \dot{m}_h (T_h - T_0) \tag{1}$$

where c_h = Source fluid heat capacity [J/(kg°C)];
$\dot{m}_h$ = Flow rate of source fluid [kg/s];
T_h = Source fluid inlet temperature [°C];
T_0 = Cooling fluid inlet temperature (environmental temperature) [°C].

Heat exchanger for geothermal water
Heat exchangers transfer heat Q from one flow stream to another, without mixing the fluids. They are, thus, elements with four connection points, the hot and cold fluids in a counter-flow heat exchange. Note that the hot and cold fluids enter the heat exchanger from opposite ends, and the outlet temperature of the cold fluids in this case may exceed the outlet temperature of the hot fluid. Theoretically, the cold fluid will be heated to the inlet temperature of the hot fluid. However, the outlet temperature of the cold fluid can never exceed the inlet temperature of the hot fluid, since this would be a violation of the second law of thermodynamics.

The related equations (Cengel and Turner, 2001; Lei H., 2004) are:

$$\dot{Q} = \dot{m}_h C_p (T_{h1} - T_{h2}) \tag{2}$$

$$\dot{Q} = \dot{m}_c C_p (T_{c1} - T_{c2}) \tag{3}$$

$$\dot{Q} = UA\Delta T_m \tag{4}$$

$$\Delta T_m = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{ln\frac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}} \tag{5}$$

where $\dot{m}_c$ = Mass flowrate of water required for circulation in the system [kg/s];
$\dot{m}_h$ = Mass flowrate of geothermal water as a heat source [kg/s];
C_p = Specific heat capacity of water [J / (kg°C)];
T_{c1} = Temperature of inlet cold water before heating by heat exchanger [°C];
T_{c2} = Temperature of outlet heated cold water by heat exchanger [°C];
T_{h1} = Temperature of inlet geothermal water at heat exchanger [°C];
T_{h2} = Temperature of outlet geothermal water at heat exchanger [°C];
$\dot{Q}$ = Heat transfer capacity of the heat exchanger [W];
U = Overall heat transfer coefficient [W / m^2 °C];
A = Surface area of the heat exchanger [m^2];
ΔT_m = Log mean temperature difference [°C].

Efficiency of a heat pump
The performance of heat pumps is expressed in terms of the coefficient of performance (COP), which is defined as:

$$COP_{HP} = \frac{desired\ output}{required\ input} = \frac{heating\ effect}{work\ input} = \frac{Q_H}{W_{net,in}} \tag{6}$$

where Q_H = Desired output;
$W_{net,in}$ = Required input.

This relationship can be expressed in rate form by replacing the quantities Q_H and $W_{net,in}$ by $\dot{Q}_H$ and $\dot{W}_{net,in}$, respectively. Note that COP_{HP} can be greater than 1.

Modern heat pumps have a coefficient of performance between 3.5 and 5.5. A coefficient of performance of 4 means that four times more heat is available than the electricity energy consumed. The vast majority of the heating energy originates from the heat source (air, ground and groundwater).

3.3 Calculation steps

In this study, simulations were done in the steps listed below:

- Heat pump;
- Coefficient of performance (COP);
- Supply/return water temperature of buffer tanks;
- Well;
- Building heat load.

4. SYSTEM CONFIGURATION

Ground source heat pump systems consist of borehole heat exchangers, a heat pump, buffer tanks and a building heating system.

4.1 Borehole heat exchangers (BHEs)

Ground source heat pumps can be categorized as having closed or open loops, and those loops can be installed in three ways: horizontally, vertically, or in a pond/lake. The type chosen depends on the available land areas and the soil and rock type at the installation site. These factors will help determine the most economical choice for installation of the ground loop (Gehlin, 2002). In the case of Zuunmod village, closed loops were chosen. A simple schematic of the borehole heat exchanger is shown in Figure 4.

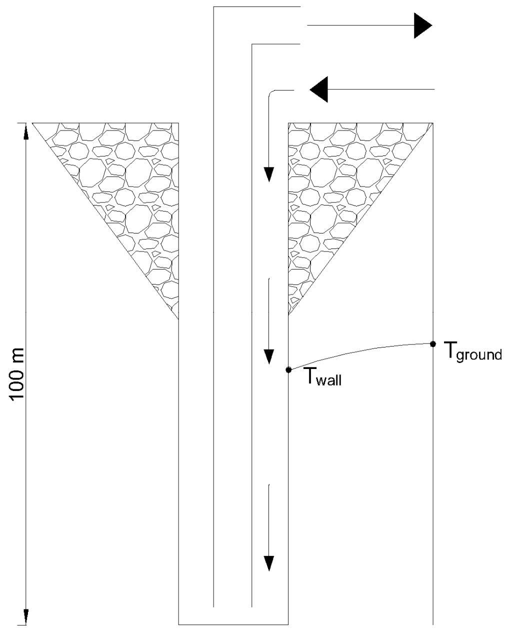

FIGURE 4: A simple schematic of the borehole heat exchanger, Zuunmod village

The collector
Vertical ground heat exchangers are classified based on their cross-sectional geometry and how the heat exchange from the flow channels takes place. The characteristics of the coaxial (also called tube-in-tube) type BHE is that heat exchange occurs from either the upstream or downstream flow channel (the flow direction may also be different during injection or extraction of heat). The inner pipe is often thermally insulated in order to avoid thermal short circuiting between the upward and downward flow channels. Figure 5 shows the coaxial pipe type design.

Analytical models for wells
Analytical models, such as the line source and cylinder source, (same models are valid for line and cylinder sinks) adopt the analytical solution of the heat transfer problem between the borehole and the nearby infinite region. They require several simplifying assumptions regarding the geometry of the borehole and heat exchanger pipes. In the ground outside the borehole it is common practice to assume that the thermal process depends only on the radial distance from the borehole axis. The one or two dimensional heat flow process from the circulating fluid to the borehole wall is assumed to be represented by a thermal resistance that characterises the temperature difference between heat carrier fluid and the borehole wall.

FIGURE 5: Vertical ground heat exchanger

The equation for the temperature field as a function of time (t) and radius (r) around a line source with constant heat injection rate (q) may be used as an approximation of the heat injection from a borehole heat exchanger (BHE) (Gehlin 2002):

$$T(r,t) = \frac{q}{4\pi\lambda}\int_{\frac{r^2}{4at}}^{\infty}\frac{e^{-u}}{u}du = \frac{q}{4\pi\lambda}E_1(r^2/4at) \tag{7}$$

E_1 is the so called exponential integral. For large values of the parameter at/r^2, E_1 can be approximated with the following simple relationship:

$$E_1\left(\frac{r^2}{4at}\right) \approx \ln\left(\frac{4at}{r^2}\right) - \gamma \quad \frac{at}{r^2} \geq 5 \tag{8}$$

where the term γ = 0.5772 is Euler´s constant.

The maximum error is 2.5% for $at/r^2 \geq 20$ and 10% for $at/r^2 \geq 5$. Ground thermal conductivity is denoted with λ and $a = \lambda/c_p$, where c_p is the ground specific heat capacity. The condition means that the accuracy increases as the thermal front reaches farther beyond the borehole wall, and the velocity of the thermal front is dependent on the ratio between thermal conductivity and the heat capacity of the ground, i.e ground thermal diffusivity.

The fluid temperature is evaluated by taking the line source temperature at the borehole radius ($r = r_b$) and adding the effect of the borehole thermal resistance (R_b) between the fluid and the borehole wall. Thus, the fluid temperature as a function of time can be written:

$$T_f(t) = \frac{q}{4\pi\lambda} \times \left(\ln\left(\frac{4at}{r^2}\right) - \gamma\right) + q \times R_b + T_0 \tag{9}$$

where T_0 is the undisturbed ground temperature (°C) and q is positive from the ground into the fluid.

The analytical model assumes constant heat flow from the well. We cannot use this convenient model because the heat taken from the well is not constant.

4.2 Heat pump

A heat pump extracts heat from the environment which is passed to the heating system. The process medium, a liquid which boils even at low temperatures, circulates in a sequence, in which it is first evaporated, then compressed, liquefied and then expanded.

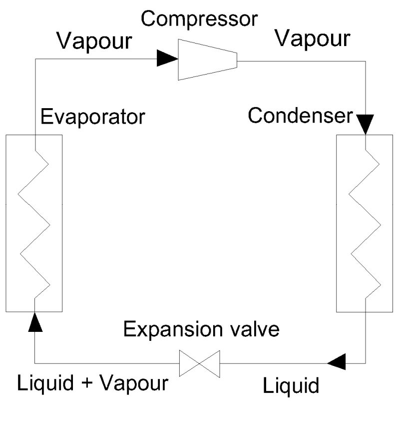

FIGURE 6: Vapour compression heat pump, Zuunmod village

The vapour compression heat pump is widely used for residential and commercial heating and cooling, food refrigeration and automobile air-conditioning, among other uses. The technology is illustrated in Figure 6. A mechanical compressor, typically driven by an electric motor, provides the work input which results in the heat being transferred from a low temperature to a high temperature.

Many of the impracticalities associated with the reversed Carnot cycle can be eliminated by vaporizing the refrigerant completely before it is compressed and by replacing the turbine with a throttling device, such as an expansion valve. The cycle that results is called the ideal vapour-compression refrigerator cycle, and it is shown schematically and on a T-s diagram in Figure 7.

The vapour-compression refrigeration cycle is the most widely used cycle for refrigerators, air conditioning systems, and heat pumps. It consists of four processes (Cengel, Y.A., 1997) as depicted in Figures 7 and 8.

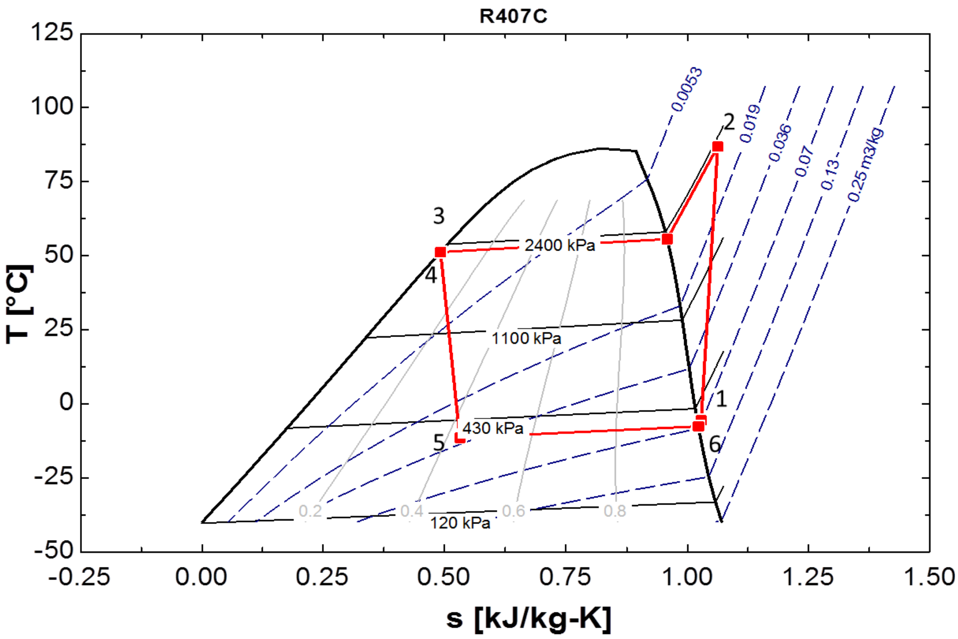

FIGURE 7: T-s diagram of the ground source heat pump; the working fluid is 'R407C'

1-2 Isentropic compression in a compressor;
2-3 Heat rejection in a condenser, P = constant;
3-4 Sub-cooling of the fluid
4-5 Throttling in an expansion device;
5-6 Heat absorption in an evaporator, P = constant.
6-1 Superheating of the fluid before entry into the compressor

In an ideal vapour-compression refrigeration cycle, the refrigerant enters the compressor at state 1 as saturated vapour and is compressed isentropically to the condenser pressure. The temperature of the refrigerant increases during this isentropic compression process, to well above the temperature of the surrounding medium. The refrigerant then enters the condenser as superheated vapour at state 2 and leaves as saturated liquid at state 3 as a result of heat rejection to the surroundings.

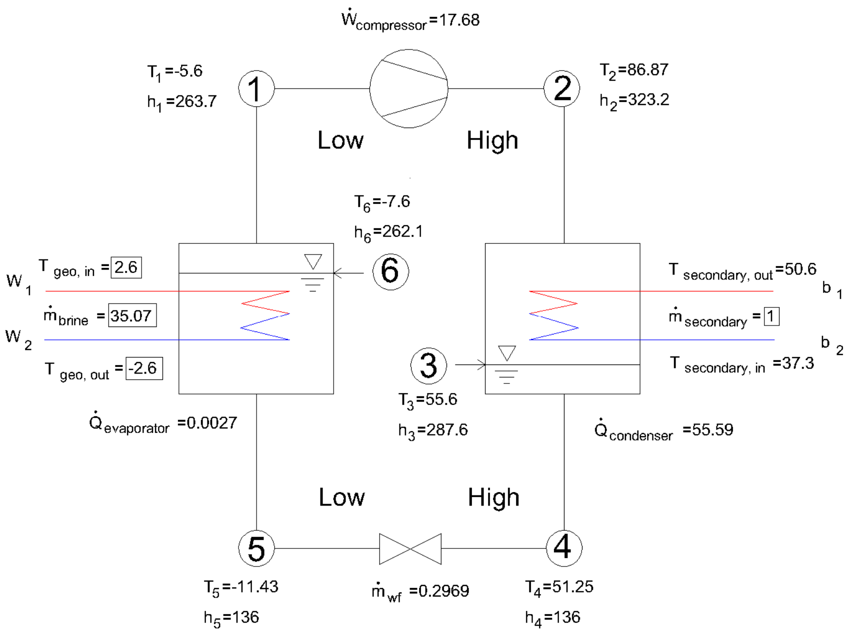

FIGURE 8: The heat pump model (W: wells; b: buffer tanks)

The saturated liquid refrigerant at state 4 is throttled to evaporator pressure by passing it through an expansion valve or capillary tube. The temperature of the refrigerant drops below the temperature of the refrigerated space during this process. The refrigerant enters the evaporator at state 5 as a low-quality saturated mixture, and it completely evaporates by absorbing heat from the refrigerated space. The refrigerant leaves the evaporator as saturated vapour and re-enters the compressor, completing the cycle.

Condenser
The condenser may be either water or air cooled. In this case, the condenser cooling water water goes to a buffer tank after being heated by the condensation of the working fluid vapour. The calculations for the condenser are roughly the same in both cases, as the cooling fluid (air or water) is very linear. State 2 is the working fluid (R407C) coming from the compressor. State 4 is the condensed fluid, normally cooled liquid. State b_2 is the entry of the cooling fluid coming from the buffer tank. State b_1 is the outlet for the fluid going to the buffer tank. The hot fluid temperature is higher than the one of the cold fluid throughout the condenser.

Vaporizer
State W_1 is the entry of the geothermal fluid to the vaporizer, and state W_2 is the outlet. State 1 is the entry of the working fluid vapour from the vaporizer into the compressor, and state 5 the outlet of the working fluid vapour or mixture towards the well.

4.3 Buffer tanks

The addition of a water storage tank as part of a heat pump system can provide operational advantages. The heat pump is used to maintain the water storage tank within a given range of temperatures, and the heat distribution system runs off the water storage tank. The storage tank acts as a heat reservoir to buffer the heat pump from small and frequent space-heating demands, and the heat pump can then operate on longer runtimes with fewer on-off cycles. Water storage tanks can also store heat for use during peak power periods, allowing the heat pump to run during off-peak hours (Meyer et al., 2011).

A combination buffer tank allows the addition of multiple heat sources and some types also produce domestic hot water (DHW) in one tank. Figure 9 shows the connection of the buffer tanks in the system.

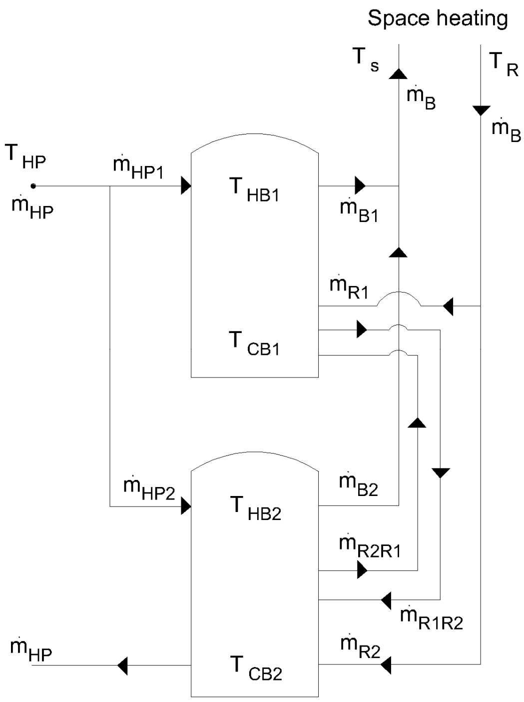

FIGURE 9: Connection of the buffer tanks

Variables used in Figure 9:
T_{HP} = Temperature of heat pump [°C];
T_{HB1} = Temperature at buffer tank 1 (hot side) [°C];
T_{CB1} = Temperature at buffer tank 1 (cold side) [°C];
T_{HB2} = Temperature at buffer tank 2 (hot side) [°C];
T_{CB2} = Temperature at buffer tank 2(cold side) [°C];
T_S = Supply temperature [°C];
T_R = Return temperature [°C];
$\dot{m}_{HP}$ = Mass flowrate of heat pump [kg/s];
$\dot{m}_B$ = Mass flowrate of building [kg/s];
$\dot{m}_R$ = Return mass flowrate of building [kg/s].

Equations used:

$$\dot{m}_{HP} = \dot{m}_{HP1} + \dot{m}_{HP2} \tag{10}$$

$$\dot{m}_{B} = \dot{m}_{B1} + \dot{m}_{B2} \tag{11}$$

$$\dot{m}_{B} = \dot{m}_{R1} + \dot{m}_{R2} \tag{12}$$

$$\frac{d(M_{HB1})}{dt} = \dot{m}_{HP1} - \dot{m}_{B1} \tag{13}$$

$$\frac{d(M_{CB1})}{dt} = -\frac{d(M_{HB1})}{dt} \tag{14}$$

$$M_{HB1} + M_{CB1} = constant = total\ water\ mass \tag{15}$$

$$\frac{d(M_{HB2})}{dt} = \dot{m}_{HP2} - \dot{m}_{B2} \tag{16}$$

$$\frac{d(M_{HB2})}{dt} = -\frac{d(M_{CB2})}{dt} \tag{17}$$

$$M_{HB2} + M_{CB2} = constant = total\ water\ mass \tag{18}$$

$$\frac{d(T_{HB1})}{dt} \times M_{HB1} \times c_p = (\dot{m}_{HP1} \times c_p \times T_{HP} - \dot{m}_{B1} \times c_p \times T_{HB1}) \tag{19}$$

$$\frac{d(T_{HB2})}{dt} \times M_{HB2} \times c_p = (\dot{m}_{HP2} \times c_p \times T_{HP} - \dot{m}_{B2} \times c_p \times T_{HB2}) \tag{20}$$

$$\frac{d(M_{CB1})}{dt} = (\dot{m}_{R1} - \dot{m}_{R1R2} + \dot{m}_{R2R1}) \tag{21}$$

$$\frac{d(M_{CB2})}{dt} = (\dot{m}_{R2} + \dot{m}_{R1R2} - \dot{m}_{R2R1}) \tag{22}$$

$$\frac{d(T_{CB1})}{dt} \times M_{CB1} \times c_p = (\dot{m}_{R1} \times c_p \times T_R - \dot{m}_{R1R2} \times c_p \times T_{CB1} + \dot{m}_{R2R1} \times c_p \times T_{CB2}) \tag{23}$$

$$\frac{d(T_{CB2})}{dt} \times M_{CB2} \times c_p = (\dot{m}_{R2} \times c_p \times T_R + \dot{m}_{R1R2} \times c_p \times T_{CB1} - \dot{m}_{R2R1} \times c_p \times T_{CB2}) \tag{24}$$

The buffer tank system is complicated, requiring assumptions regarding the control method. Basically the buffer tanks store hot fluid from the heat pump in order to enable the pump to have longer running/standstill times (so that the pump is not constantly starting and shutting down). The cold fluid coming from the radiator is stored in the same way. Obviously the sum of the mass in the hot and cold parts of the buffer tank system is constant.

A simplified model for calculation purposes was made having one tank for the hot side and another for the cold side. The simplified system is shown in Figure 10.

Variables used in Figure 10:
T_{HP} = Temperature of heat pump [°C];
T_{HB} = Temperature of hot buffer tank [°C];

T_{CB} = Temperature of cold buffer tank [°C];
T_S = Supply temperature [°C];
T_R = Return temperature [°C];
$\dot{m}_{HP}$ = Mass flowrate of heat pump [kg/s];
$\dot{m}_B$ = Mass flowrate of building [kg/s].

Equations used:

$$\frac{d(M_{HB})}{dt} = \dot{m}_{HP} - \dot{m}_B \tag{25}$$

$$\frac{d(M_{CB})}{dt} = -\frac{d(M_{HB})}{dt} \tag{26}$$

$$M_{HB} + M_{CB} = constant = total\ water\ mass \tag{27}$$

$$\frac{d(T_{HB})}{dt} \times M_{HB} \times c_p = \dot{m}_{HP} \times c_p \times T_{HP} - \dot{m}_B \times c_p \times T_{HB} \tag{28}$$

$$\frac{d(T_{CB})}{dt} \times M_{CB} \times c_p = \dot{m}_B \times c_p \times T_R - \dot{m}_{HP} \times c_p \times T_{CB} \tag{29}$$

4.4 Building heating systems

Mongolia has centralized heating systems for office and apartment buildings in the capital city. The capital and province centres have coal thermal power plants which supply heat during the winter season. The water supply temperature is 70-150°C, and the return temperature is 70-90°C. The winter is long with cold temperatures but summer is hot and not as long. The heating supply is continued between October 1st and May 1st. This is one of the reasons for air pollution during the cold season, especially in the large cities. The district heating system is turned off during the rest of the year. Traditional houses are heated by coal furnaces.

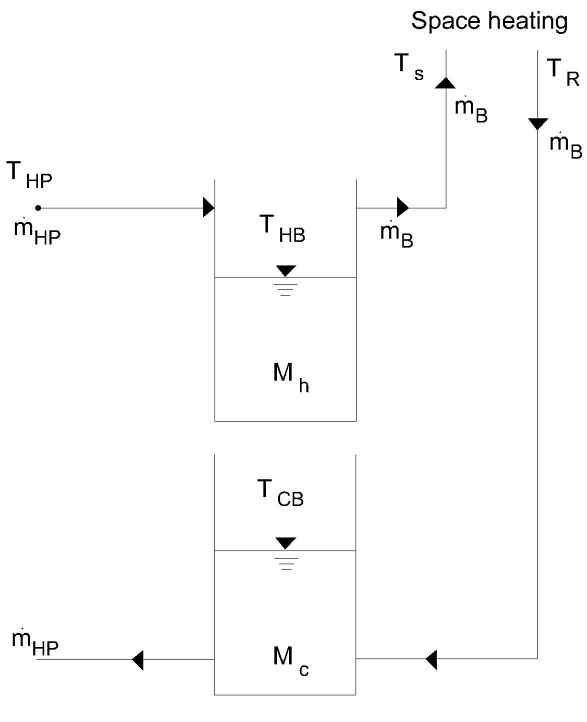

FIGURE 10: Simplified connection of the buffer tanks

Geothermal energy represents a new energy sector in Mongolia that can benefit some rural consumers who have no reliable electricity or heating. Recently Mongolia has started to utilize shallow heat pump systems as renewable energy technologies for space heating. The heating systems consist of buildings, pipes, a pump station and a heat producing station. Heat can be produced from a geothermal field or by combustion.

Common reference values for a building heating network are the following, with all reference values marked with the subscript 0:

1. Supply water temperature T_{s0} = 80°C;
2. Return water temperature T_{r0} = 40°C;

3. Indoor temperature T_{i0} = 20°C.

The reference outside temperature depends on the local climate.

The calculation of temperatures and heat flow in the network is based on the flow solution (Figure 11). Heat is transferred by the fluid; conservation of energy for the network has to be fulfilled, in the same way as for mass in the flow solution.

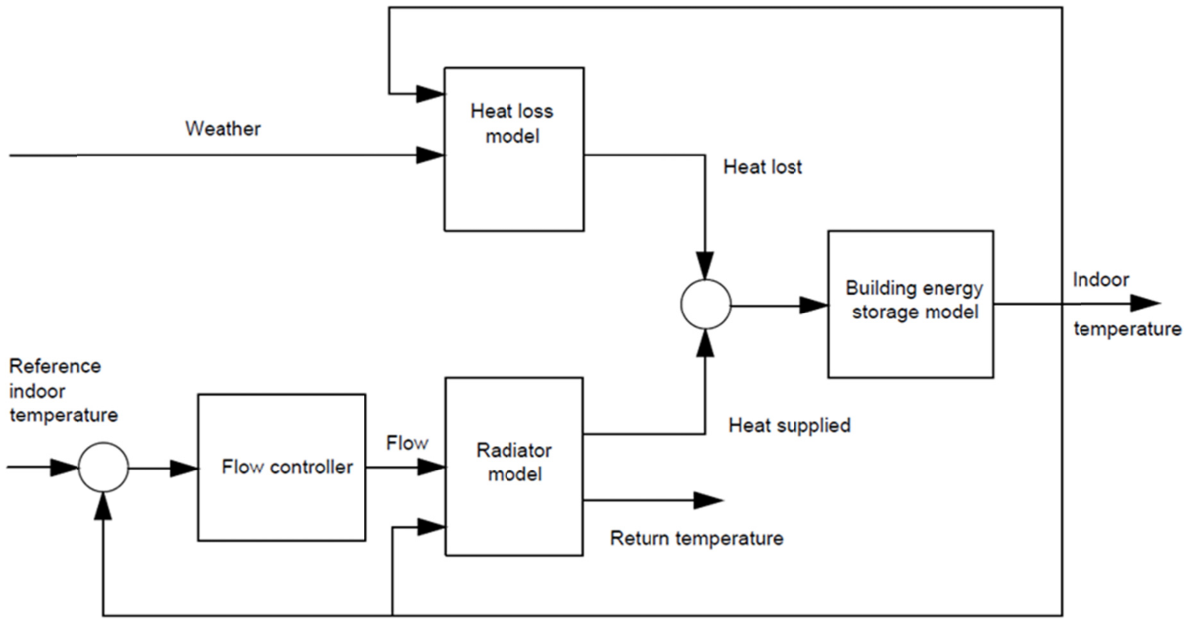

FIGURE 11: Block diagram of a building heating model

For closed loop systems, water or antifreeze solution is circulated through plastic pipes buried beneath the earth's surface. During the winter, the fluid collects heat from the earth and carries it through the system and into the building.

Radiators
A radiator is the heat exchanger that transfers heat from the heating system to the indoors air. According to Anon (1977), the relative heat load of a radiator can be written as:

$$\frac{Q}{Q_0} = \left(\frac{\Delta T_m}{\Delta T_{m0}}\right)^{(4/3)} = \left(\frac{T_s - T_r}{ln\left(\frac{T_s - T_i}{T_r - T_i}\right)} \times \frac{ln\left(\frac{T_{s0} - T_{i0}}{T_{r0} - T_{i0}}\right)}{T_{s0} - T_{r0}}\right)^{(4/3)} \tag{30}$$

where Q/Q_0 = Ratio of actual heat output from the radiator to the heat output at design conditions;
ΔT_m = Radiator logarithmic mean temperature difference [°C];
T_s = Water supply temperature [°C];
T_r = Water return temperature [°C];
T_i = Indoor temperature [°C].

Supply water temperature is assumed to be around 80°C and the return water temperature is 40°C for geothermal systems. For fuel fired systems, similar values are 90/70°C, with the indoor temperature set at 20°C; in order to determine the radiator size, the following equation is used.

The logarithmic mean temperature difference for a radiator, ΔT_m [°C] is defined as:

$$\Delta T_m = \frac{(T_s - T_i) - (T_r - T_i)}{ln\left(\frac{T_s - T_i}{T_r - T_i}\right)} \tag{31}$$

Water heat duty
The heat load, $\dot{Q}$ [W] due to hot water going through the radiator is:

$$\dot{Q} = C_p \dot{m}(T_s - T_r) \tag{32}$$

The relative heat load of the water flow can be written as:

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{\dot{m}(T_s - T_r)}{\dot{m}_0(T_{s0} - T_{r0})} \tag{33}$$

Building heat loss
The heat loss of the building can be defined as:

$$\dot{Q}_{loss} = k_1(T_i - T_0) \tag{34}$$

where k_1 = The building heat loss factor, which is a constant [W/°C].

Relative heat loss is obtained by:

$$\frac{\dot{Q}_{loss}}{\dot{Q}_{loss0}} = \frac{(T_i - T_0)}{(T_{i0} - T_{o0})} \tag{35}$$

Building energy storage
The building will not cool immediately when the heating is stopped because of the heat capacity. The building energy storage model is:

$$\frac{dT_i}{dt} = \frac{1}{C}\dot{Q}_{net} = \frac{1}{C}\left(\dot{Q}_{supp} - \dot{Q}_{loss}\right) = \frac{1}{C}(\dot{m}C_p(T_s - T_r) - k_l(T_i - T_o)) \tag{36}$$

where C = Heat capacity of building [kJ/°C].

In the steady state model, all time derivatives are set to zero, so this equation is only used in the dynamic model (Lei H., 2004).

Relationship between mass flow and indoor temperature
The flow controller in the system is unknown. There is no simple physical relationship between water flow and indoor temperature, and different buildings have different regulation systems. Each consumer has his own preference about the indoor temperature and how to change it. The relationship between the indoor temperature and the water flow has to be presented as some average of all consumers in the system. Here, P-control (proportional) is used. The P-controller is presented by the following equation:

$$\dot{m} = k_p\left(T_{i\_set} - T_i\right) + \dot{m}_{ave} \tag{37}$$

where k_p = P-control parameter [kg/s °C];
T_{i_set} = Desired indoor temperature [°C];
$\dot{m}_{ave}$ = Average mass flow [kg/s].

The average flow constant is used to define the operating point of the controller, which is the same as the flow when the indoor temperature is exactly at the desired value (no error).

Well model
A finite volume dynamic heat flow model was made for a well. It assumes that all the heat is taken from a single well. In reality, the heat was taken from 24 wells in a relatively small region. The actual configuration of the wells is shown in Appendix I.

An axi-symmetric model of this "equivalent" well was made in MATLAB. The elements used for the calculation are toroidal, just like donuts with a square cross-section. The time derivative of the temperature is calculated for each element, and then it is in integrated with a Runge Kutta numerical integrator in MATLAB. The conduction equation in the cylindrical coordinates is:

$$\frac{\rho C_p}{k} \times \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{\partial^2 T}{\partial z^2} \tag{38}$$

This equation is solved with the finite volume method.

Boundary conditions:

Ground surface (towards atmospheric air):

$$\dot{Q}_{surface} = A \times h \times (T_{surface} - T_{air}) \tag{39}$$

where A = Surface area;
h = Convection heat transfer coefficient [W/m^2];
$T_{surface}$ = Average mass flow [kg/s].

Well:

$$\dot{Q}_{well} = \dot{Q}_{heat\ pump} \tag{40}$$

Bottom, one element row deeper than the lowest element:

$$T = \ constant = undisturbed\ ground\ temperature \tag{41}$$

Radial, one element row further away from the well than the farthest element:

$$T = \ constant = undisturbed\ ground\ temperature \tag{42}$$

4.5 Model calculation – results of analysis

1. Heat pump calculations done in EES software and MATLAB.

Figures 12 and 13 show the estimated power required by the heat pump and the estimated mass flow through the heat pump during a year, respectively. There is no heat load in summer. Figure 14 shows that the estimated power of the heat pump depends on the indoor temperature (at 20°C). It can be seen that if the indoor temperature is over 21°C, the heat pump is not operating On the other hand, if the indoor temperature would be less than around 19°C, the heat pump operates at maximum power. It means if the radiator is fully loaded, and maximum flow is reached. The efficiency of the heat pump (COP) is 3.145.

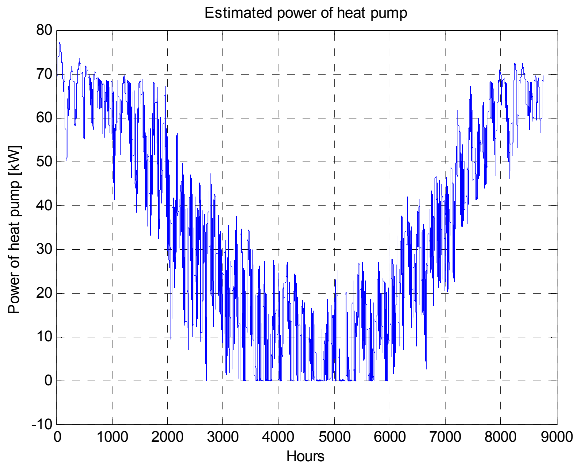

FIGURE 12: Estimated power of heat pump

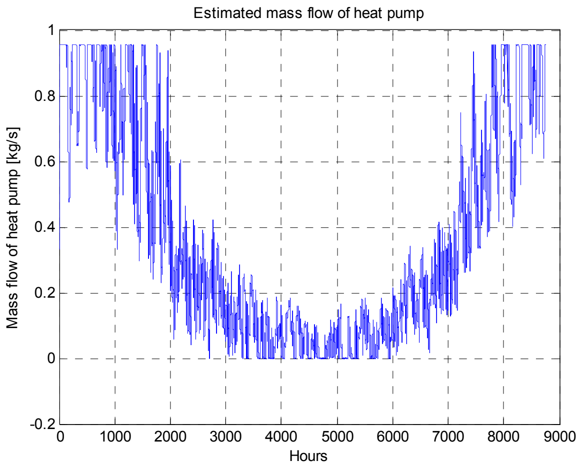

FIGURE 13: Estimated mass flow of heat pump

2. Buffer tanks.

Figure 15 shows the estimated mass in the hot buffer tank.

The maximum heat delivered from the heat pump is 90 kW. The buffer tanks are assumed to have a capacity of 750 kg each.

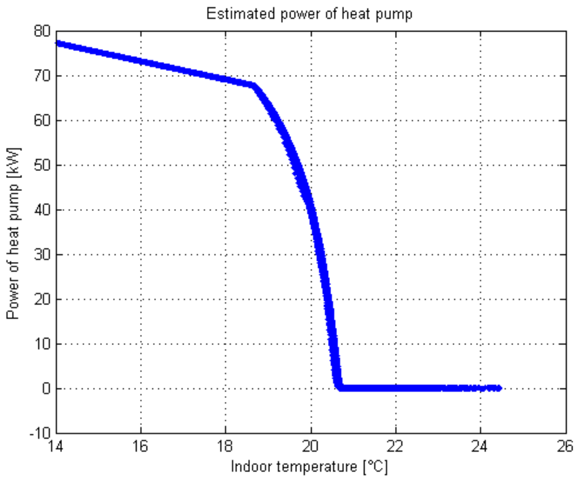

FIGURE 14: Estimated power of heat pump depending on indoor temperature

3. Well

The model assumes the well to have 8 elements with 20°C/km gradient. Figure 16 shows the finite volume rock model for the well, while Figures 17-24 show for elements 1 and 2, at 25 and 75 m depth, respectively, the temperatures at increasing distance from the well, 0.5, 3, 9, and 17 m. Initially the temperature is the undisturbed temperature, but equilibrium has been reached after 6 years.

In Appendix III, the diagrams are shown with a fixed scale.

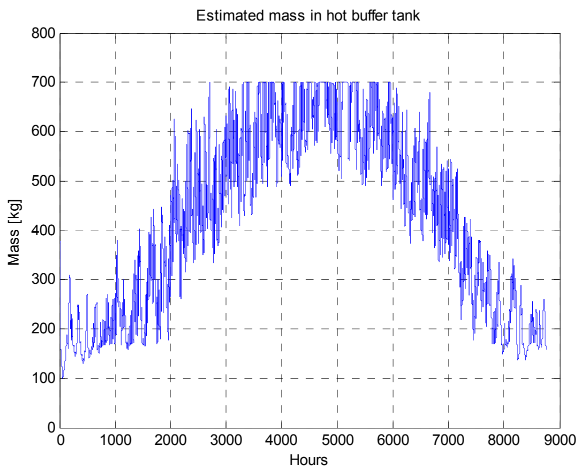

FIGURE 15: Estimated mass in the hot buffer tank

4. Building heat load

Figures 25-29 show the results of the modelling for the building heating system. Figure 25 shows estimated building indoor temperature for a year. The indoor temperature is only 14°C on the coldest days in Ulaanbaatar and if the outdoor temperature is higher than the indoor temperature, then the room temperature is the same as the outdoor temperature.

Figure 26 shows the estimated mass flow to the radiators. Figure 27 presents the estimated mass flow as a function of the indoor temperature. There is no load in summer. The proportionality band is 2°C.

Figure 28 shows the estimated return temperature is dependent on the indoor temperature. If the indoor temperature is less than the indoor temperature requested, the radiator system works on full load, the flow is constant at the maximum value, and thus the return temperature falls as the indoor temperature is further lowered into the radiator overload region.. On the other hand, if the outdoor temperature is higher than the indoor temperature, then the return temperature is the same as the indoor temperature. There is no load and no flow through the radiators.

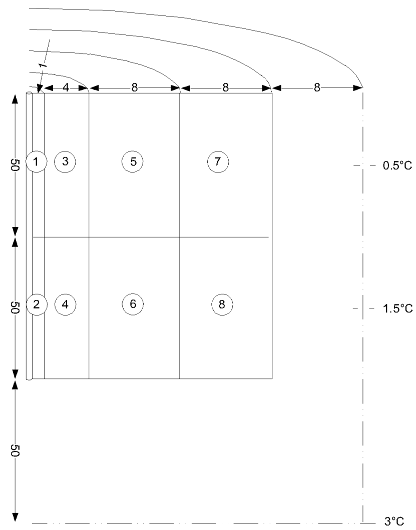

FIGURE 16: The finite volume rock model

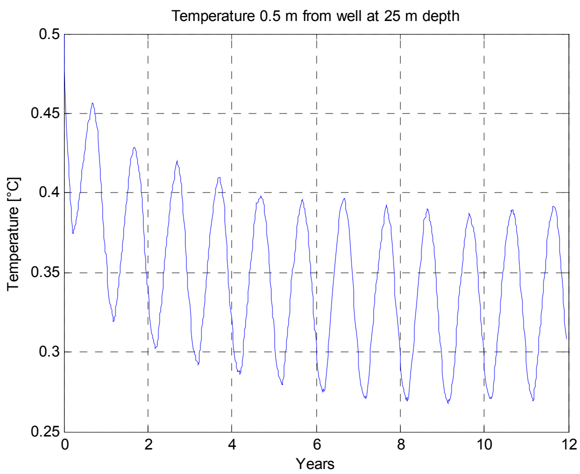

FIGURE 17: Element 1 – Temperature 0.5 m from well at 25 m depth

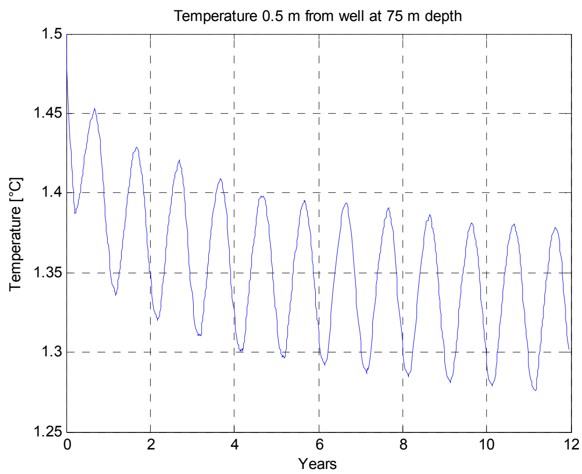

FIGURE 18: Element 2 – Temperature 0.5 m from well at 75 m depth

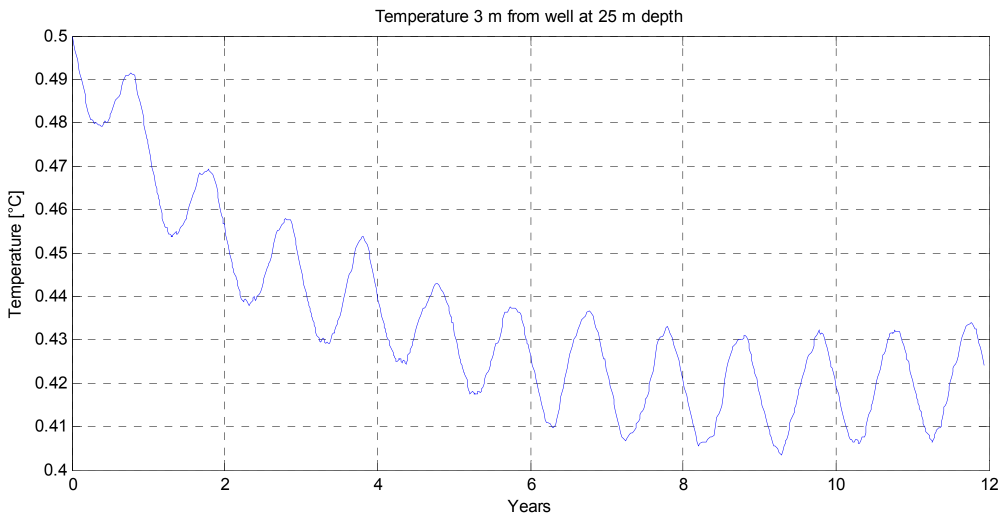

FIGURE 19: Element 3 – Temperature 3 m from well at 25 m depth

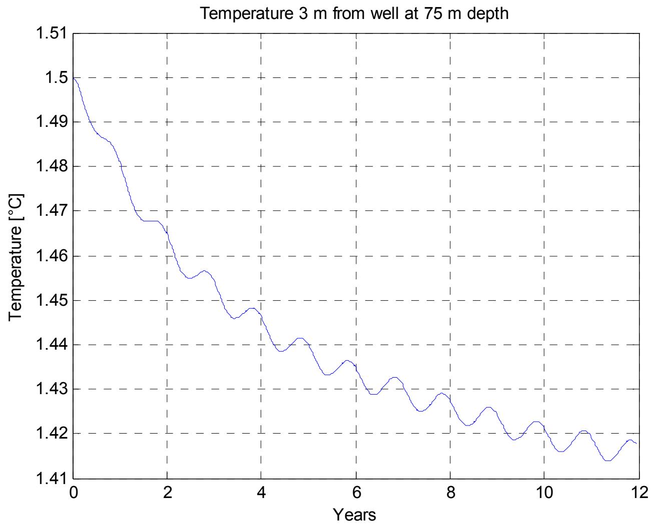

FIGURE 20: Element 4 – Temperature 3 m from well at 75 m depth

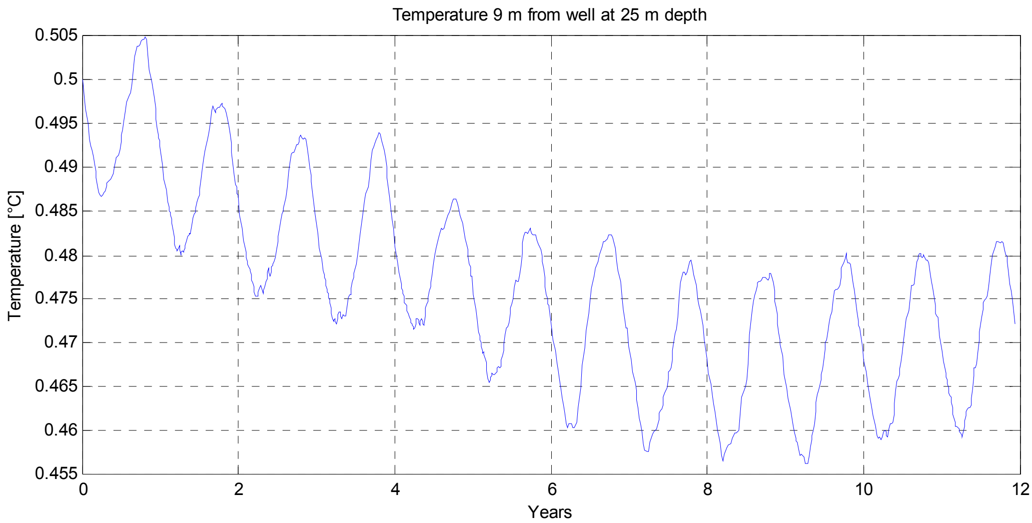

FIGURE 21: Element 5 – Temperature 9 m from well at 25 m depth

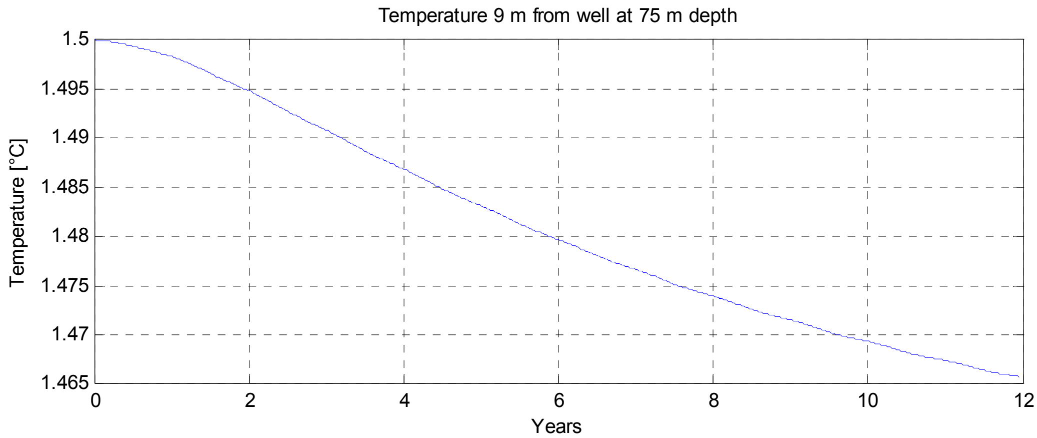

FIGURE 22: Element 6 – Temperature 9 m from well at 75 m depth

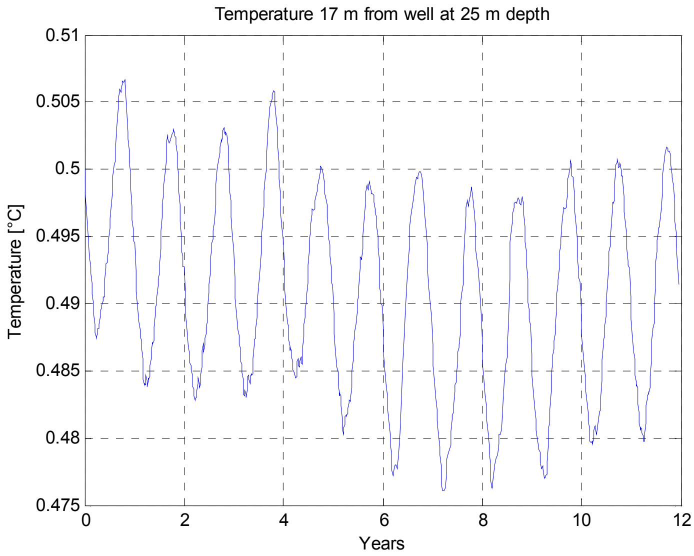

FIGURE 23: Element 7 – Temperature 17 m from well at 55 m depth

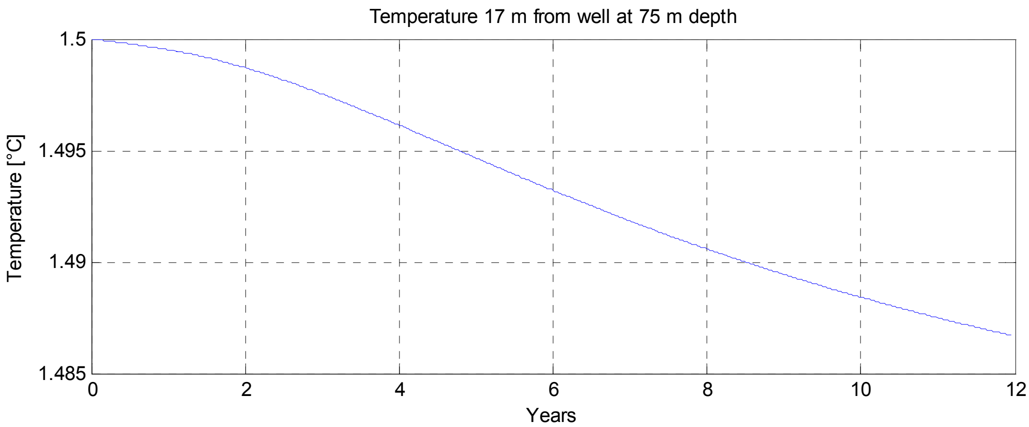

FIGURE 24: Element 8 – Temperature 17 m from well at 75 m depth

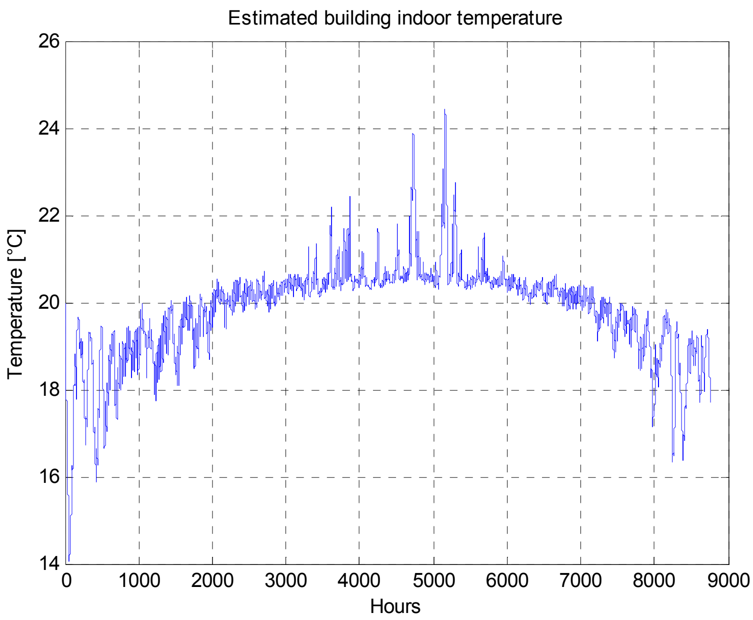

FIGURE 25: Estimated building indoor temperature

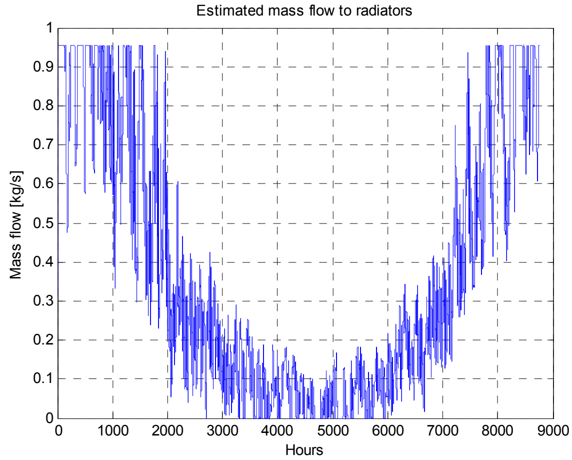

FIGURE 26: Estimated mass flow to radiators

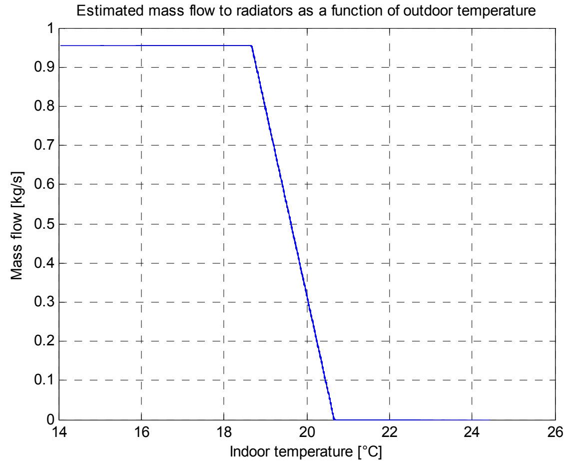

FIGURE 27: Estimated mass flow in a radiator

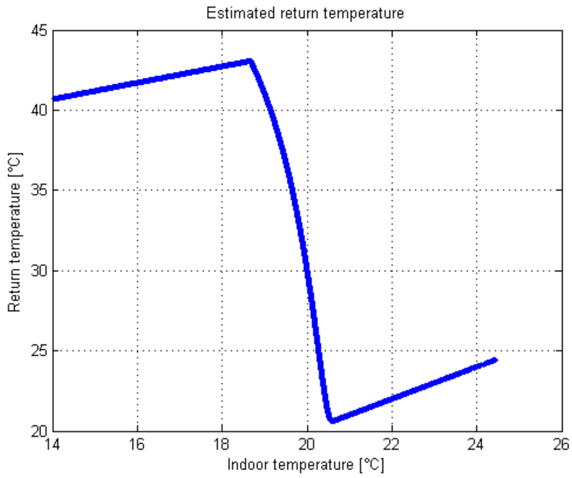

FIGURE 28: Estimated return temperature

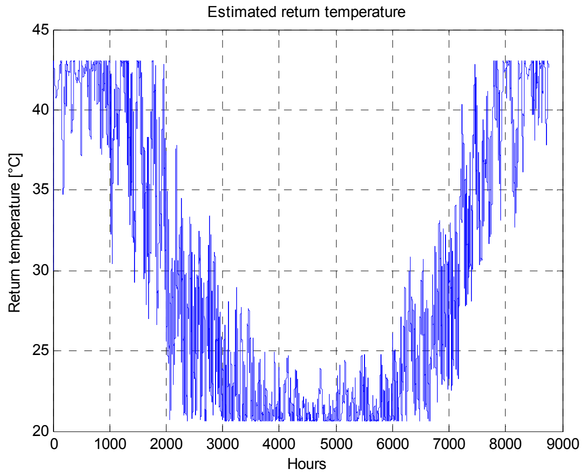

FIGURE 29: Estimated return temperature for a year

Figure 29 shows the estimated return temperature for a year.

5. ENVIRONMENTAL FACTORS OF HEAT RESOURCES IN MONGOLIA

There exists no underground temperature map for Ulaanbaatar or other towns. However, the underground temperature in an outer district of Ulaanbaatar was measured in 2003 by the Institute of Geography, Academy of Science, Mongolia, as shown in Table 1 (Dash and Ulziisuren 2007).

Surface water is usually not abundant in Mongolia. Groundwater boreholes are used since rivers and lakes disappear during the dry seasons. Tables 2-4 show various weather parameters necessary for modelling of groundwater heat pumps.

TABLE 1: Underground temperature in an outer district of Ulaanbaatar (3 January, 2003)

Depth (m)	Gandan [°C]	Tolgoit [°C]	Bayanhoshuu [°C]	Yarmag [°C]	Hailaast [°C]	Dambadarjaa [°C]
0.33	-12.67	-14.64	-16.41	-15.06	-13.85	-15.02
0.66	-9.01	-10.90	-12.92	-9.55	-7.98	-12.27
1	-5.91	-8.35	-9.36	-6.02	-4.90	-6.93
1.33	-3.30	-5.79	-6.52	-3.71	-2.52	-5.31
1.66	-1.11	-3.55	-4.32	-1.98	-0.75	-2.71
2	0.23	-1.75	-2.37	-0.99	0.23	-0.54
2.5	1.56	0.21	-0.09	-0.03	1.06	-0.16
3	2.71	1.45	1.15	0.45	1.69	-0.32
3.5	3.46	2.27	2.06	0.36	2.22	-0.47
4	3.90	2.86	2.67	1.15	2.66	-0.59
4.5	4.17	3.29	3.13	1.36	2.96	-0.68
5	4.35	3.54	3.43	1.49	3.14	-0.77
5.5	4.41	3.68	3.59	1.56	3.25	-0.82
6	4.41	3.82	3.67	1.57	3.27	-0.86

TABLE 2: Standard freezing depth in province centres of Mongolia

Province centre	Freezing depth [m]			
	Clay (loamy)	Sandy (fine)	Sandy (coarse)	Bulky soil
Ulaanbaatar	2.6	3.1	3.3	3.8
Tsetserleg, Arkhangai	2.1	2.5	2.7	3.0
Ulgii, Bayan-Ulgii	2.2	2.7	2.9	3.2
Bayankhongor, Bayankhongor	2.3	2.8	3.0	3.3
Bulgan, Bulgan	2.4	2.9	3.1	3.5
Altai, Gobi-Altai	2.4	2.9	3.1	3.4
Sainshand, Dornogobi	2.2	2.6	2.8	3.1
Choibalsan, Dornod	2.5	3.1	3.2	3.6
Mandalgobi, Dundgobi	2.2	2.7	2.9	3.2
Uliastai, Zavkhan	2.7	3.3	3.5	3.9
Arvaikheer, Uvurkhangai	2.1	2.6	2.7	3.0
Dalanzadgad, Umnugobi	2.0	2.4	2.5	2.8
Baruun-Urt, Sukhbaatar	2.5	2.7	2.9	3.4
Sukhbaatar, Selenge	2.7	2.9	3.2	3.8
Zuunmod, Tuv	2.8	3.1	3.5	4.3
Ulaangom, Uvs	2.9	3.4	3.9	4.5
Khovd, Khovd	2.7	2.9	3.2	3.8
Moron, Khuvsgul	2.8	3.4	3.9	4.5
Undurkhaan, Khentii	2.5	3.2	3.3	3.7

TABLE 3: Average monthly air temperature in Mongolia's capital, Ulaanbaatar

Months	I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII
Air temperature [°C]	-16.0	-11.6	-0.4	8.4	17.4	23.9	25.5	25.8	16.0	12.8	-4.9	-4.2

TABLE 4: Average monthly/yearly weather specifications of Zuunmod village, Tuv province

Location	Latitude: 45°32'; Longitude: 45°32'; Elevation: 1390 m a.s.l.												
Specifications	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**Year**
Outside air temperature, [°C]	-14.8	-12	-2.4	4.3	11.8	18.2	20.6	20.4	13.2	5.6	-4.3	-12.9	4
Average wind speed [m/s]	6	4	9	12	10	8	6	6	6	6	2	4	8
Relative humidity [%]	66	51	44	43	43	47	40	47	46	50	58	71	50
Solar radia-tion (global), [kWh/m^2]	1.64	2.46	3.38	4.66	5.78	6.4	6.48	5.52	4.15	2.84	1.71	1.33	3.8625

6. CONCLUSIONS

The goal of this project was to build a cost effective, customer satisfying and environmentally friendly independent heating system. The main conclusions are the following:

1) In this project, an analysis of a ground source heat pump building heating system of Zuunmod village, Tuv province was undertaken.
2) The system did not heat the building as expected during the coldest weather because of the use of small radiators in the building (Figure 25).
3) The ground temperature drop around the well is minimal, as the heat used was small.
4) Initially the temperature of the ground was the undisturbed temperature. After 6 years it can be assumed to have reached equilibrium.

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APPENDIX I: Pipes and location of borehole heat exchanger

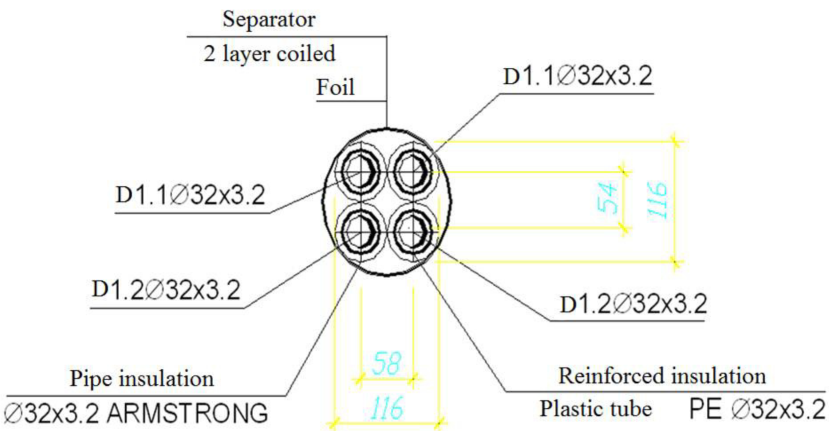

FIGURE 1: The cross-section and diametres of pipes
Note: Four 32 mm diameter pipes of 100 m length were installed in each well. The boreholes have a diameter of 150 mm and are staggered at 8 m distance from one another

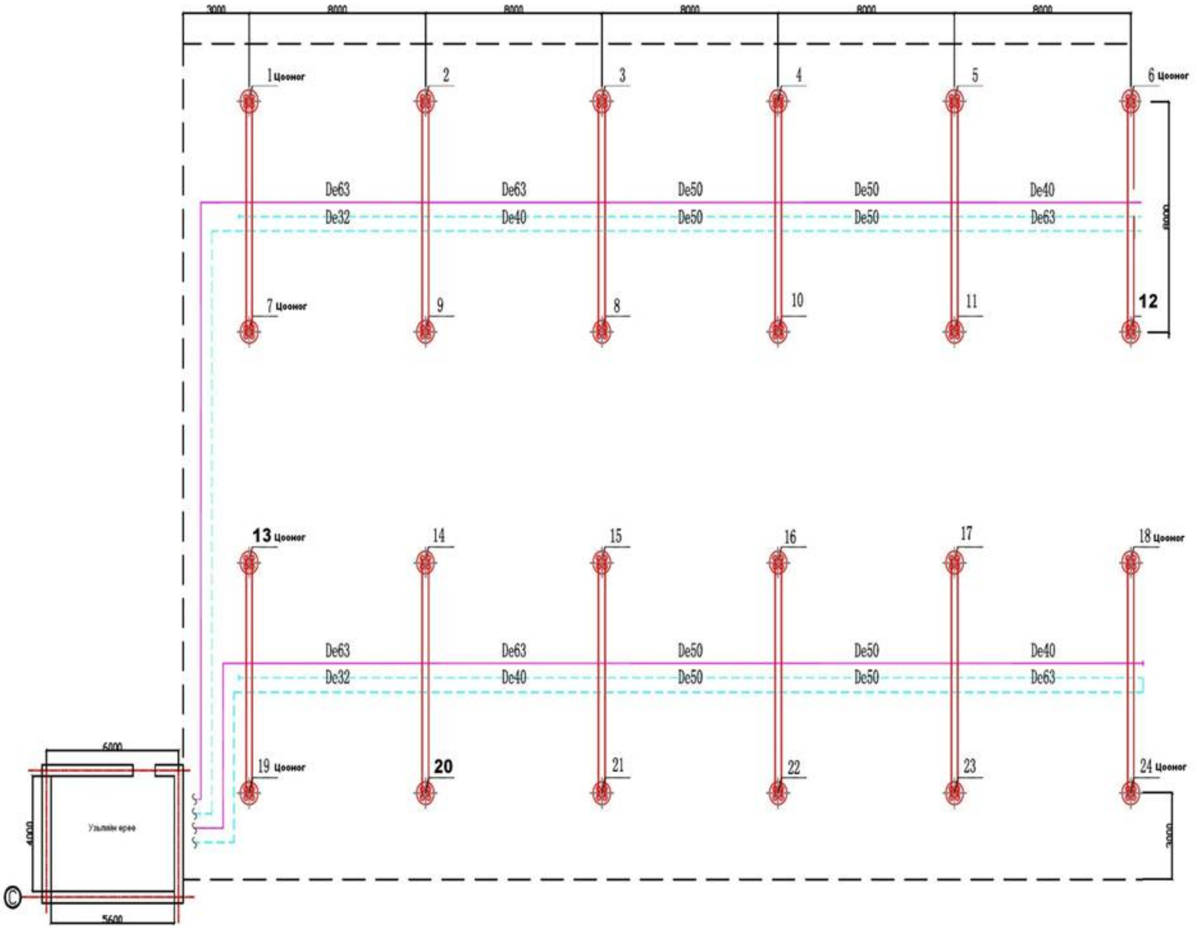

FIGURE 2: The location of borehole heat exchangers
Note: The working fluid absorbs ground heat and transfers to the heat pump

APPENDIX II: Some technical specifications of the heat pump system

TABLE 1: Technical specifications of
Natura BWH 280 (90 kW) heat pump

Operation point	**Unit**	**B2/W65^0 C**
Output power	kW	90
Refrigerant power	kW	61.7
Power consumption	kW	32.2
Efficiency	-	2.91

TABLE 2: Various types of pumps used

Heat pumps	**Type**	**Capacity [kW]**
Heat pump	Natura BWH 28	90
Network pump	Magna 2-100	0.1-0.19
First circuit pump	Grundfos CR10-2	0.75
Heat network pump	UPS 40-60/4F	0.34
Solar heat circuit pump	Grundfos 25-60	0.4-0.7

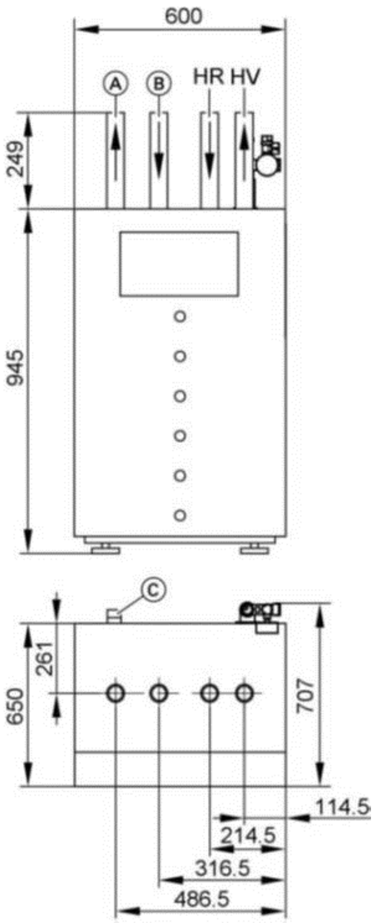

FIGURE 1: The size of the heat pump; HR – Heating return; HV – Heating value;
A – Brine return; B – Brine value; and C – Cable input

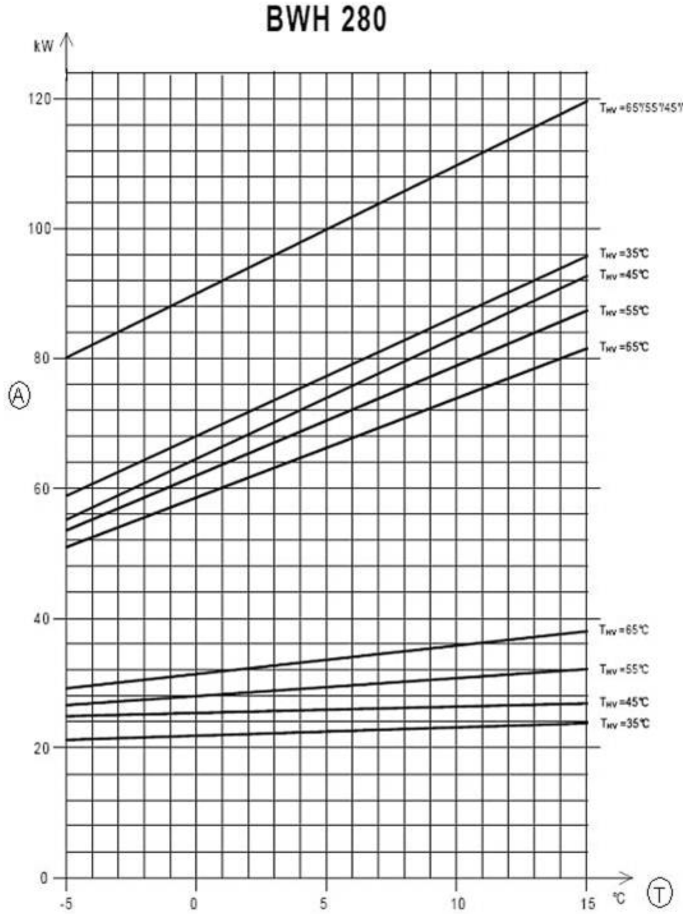

FIGURE 2: Output power of the heat pump

APPENDIX III: Fixed scale temperature graphs for elements 1 and 2

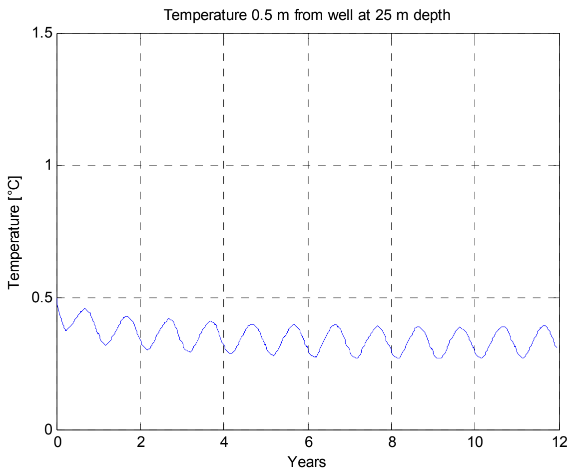

FIGURE 1: Temperature 0.5 m from well at 25 m depth

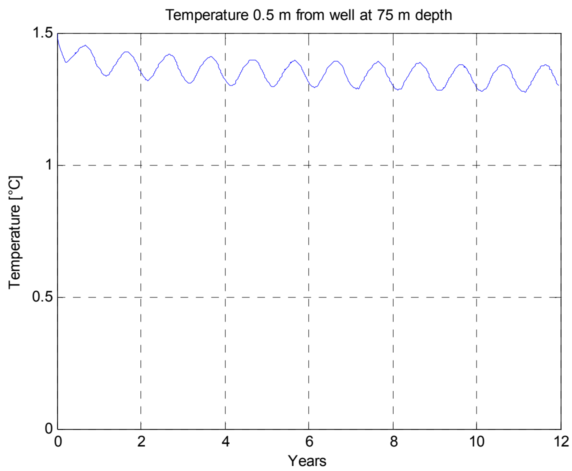

FIGURE 2: Temperature 0.5 m from well at 75 m depth

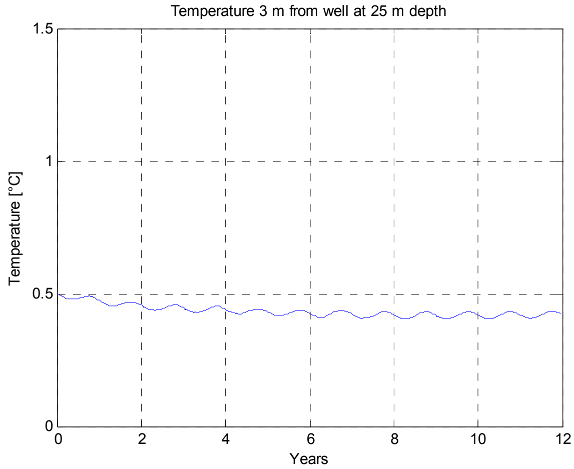

FIGURE 3: Temperature 3 m from well at 25 m depth

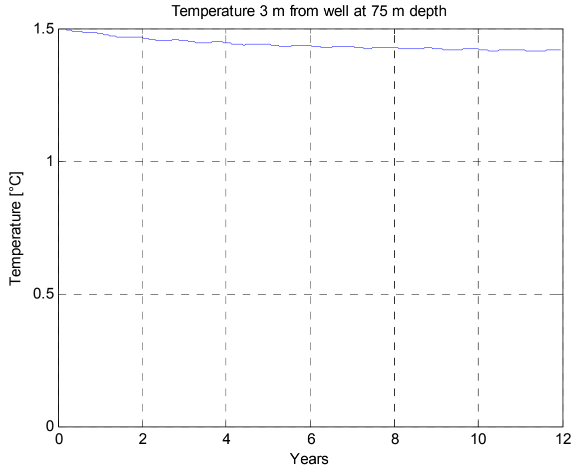

FIGURE 4: Temperature 3 m from well at 75 m depth

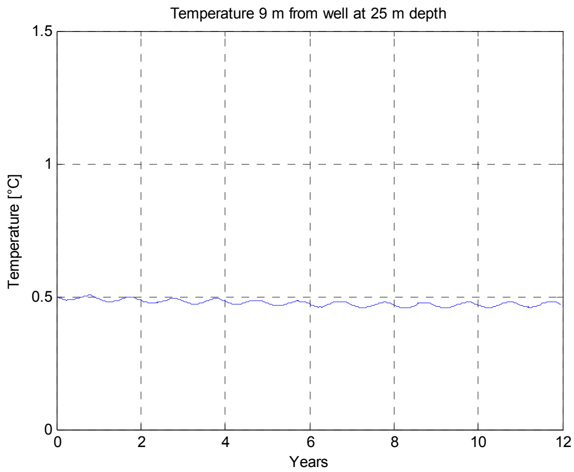

FIGURE 5: Temperature 9 m from well at 25 m depth

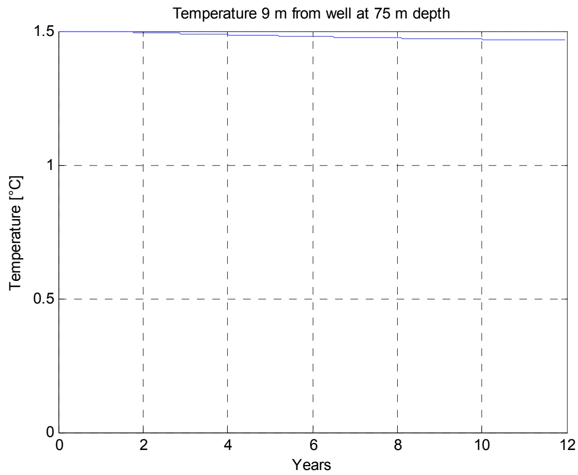

FIGURE 6: Temperature 9 m from well at 75 m depth

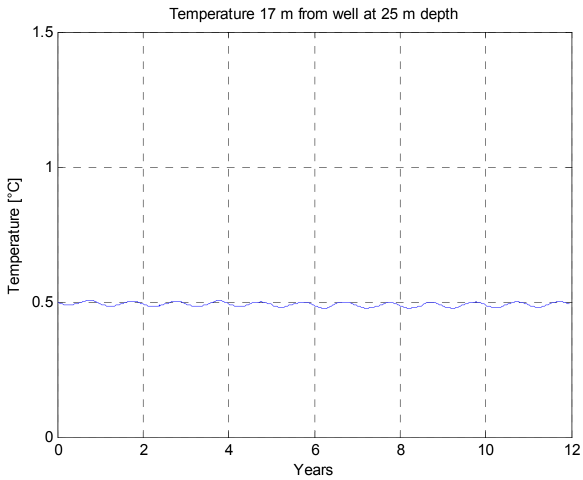

FIGURE 7: Temperature 17 m from well at 25 m depth

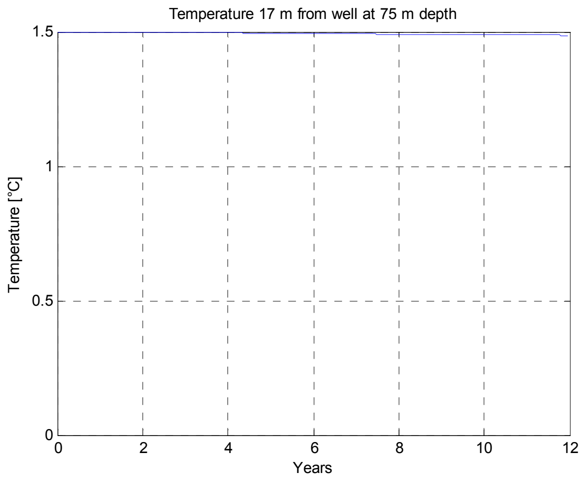

FIGURE 8: Temperature 17 m from well at 75 m depth