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Geothermal heat pump plant performance in a nearly zero-energy building

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Abstract

On the behalf of reaching EU directive 2010/31/EU energy performance targets and fulfilling nearly zero-energy energy buildings (nZEB) requirements by the end of 2020, utilization of renewable energy sources becomes important. Renewable solar and ground energy can be efficiently utilized by a hybrid geothermal heat pump with a solar thermal storage, which is expected to yield high seasonal coefficient of performance (SCOP) making it attractive to consider in nZEB design. This numerical study investigates the impact of various ground heat exchangers and thermal storage options along with their possible combinations on heat pump plant heating/cooling performance in the design of commercial hall-type nZEB located in cold climate of Hämeenlinna, Finland. Components applied in a numerical study were energy piles, vertical boreholes (heat wells), solar collector and/or exhaust air heat as a thermal storage source. A whole year dynamic simulations were performed in IDA-ICE simulation environment, where detailed custom ground-source heat pump (GSHP) plants were modelled. Results revealed GSHP plant to be favorable heat source option in nZEB design. Application of thermal storage enabled to reduce energy piles field by more than two times. Proposed exhaust air thermal storage option performed highly efficient in comparison to solar thermal storage.

Peer-review under responsibility of the organizing committee of the SBE16 Tallinn and Helsinki Conference.

Keywords: heat pump plant; energy piles; boreholes; nZEB; thermal storage; IDA-ICE; whole building simulation

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1. Introduction

European Parliament directive 2010/31/EU [1] raises the level of ambition in buildings energy performance and requires all new buildings built to be nearly zero-energy buildings (nZEB) by the end of 2020. On achieving nZEB requirements consideration of renewable energy sources such as geothermal and solar in the design is expected. Geothermal energy can be efficiently utilized with a ground source heat pump (GSHP) and according to a review on worldwide application of geothermal energy [2] total installed worldwide GSHPs capacity has grown 2.15 times in the period of 2005 to 2010 and application of GSHP is registered in 78 countries around the globe.

High annual GSHP SCOP values up to 4.5 and overall geothermal plant SCOP values up to 3.9 (with control and distribution losses) were obtained by measuring the performance of actual GSHP installations [3-4]. Though, in most cases, operation of a heat pump is accompanied by the unbalanced geothermal energy extraction/injection that leads to a significant loss in long-term operation performance [5]. In order to maintain stable long-term operation of GSHP plant and improve geothermal energy yield along with seasonal coefficient of performance (SCOP), a source of thermal storage should be considered in the plant design. Numerical study conducted by Reda [6] presents the benefits of solar thermal storage in a GSHP plant with a borehole field type ground heat exchanger (GHE), where application of solar thermal storage helped to improve GSHP plant SCOP from 1.6 to 3.0. Allaerts et al. [7] modelled the performance of a GSHP plant with dual borehole field and active air source storage in TRNSYS, where cooling tower i.e. dry cooler was applied as a thermal storage source. As a results of thermal storage application, overall size of borehole field was reduced by 47% compared to the same capacity single borehole field plant without thermal storage.

GSHP plant performance is also depended on the type of GHE considered in the plant design. Typical closed loop GHEs are classified by the position of installation - horizontal and vertical. Horizontal GHE is generally cheaper to install compared to vertical GHE, but requires more land area for the installation. As the horizontal GHE installation depth is generally very shallow (when installed in soil medium) i.e. just below the soil freezing depth, which e.g. in Estonia is ca 2 meter below ground surface, it's performance is very depended on the outdoor air temperature fluctuations, intensity of solar radiation incident on the ground surface and even presence of snow on ground surface. On the other hand, single vertical 100 meter deep borehole would be less impacted by the outdoor air temperature variance and solar radiation, as it is mostly exposed to the temperature of its surrounding medium. In buildings with limited land area, vertical GHE in form of a borehole reaching up to 400 meters in depth might be a solution instead of horizontal GHE installation. However, drilling very deep boreholes might be not only very expensive, but also drilling depth might be limited by the government regulations in the region of interest. In this case, field of multiple shorter boreholes (not exceeding the drilling depth limit) spaced at known distance to each other might be considered as a GHE alternative.

In buildings with pile foundations, installation of heat exchange piping into foundations piles enables the foundation piles to perform as a ground heat exchanger similarly to previously described field of boreholes. Geothermal pile foundations are known also as geothermal energy piles [8]. As the installation of heat exchange piping into foundation pile compared to the drilling of a new borehole is much cheaper, energy piles tend to be a very cost effective GHE solution. As the layout of energy piles is generally defined by the foundation plan, thermal interferences between closely located adjacent piles appear. Thermal interferences may also appear in field of boreholes, depending on the spacing between them. Sizing and assessment of borehole field or energy piles performance is generally carried out with help of numerical modelling regarding which more detailed aspects are described in previously conducted study by Fadejev and Kurnitski [9].

From the perspective of thermal storage application, not all types of GHEs would benefit from a thermal storage due to varying thermal losses intensity, GHE storage capacity and peak heat extraction/rejection rates. To consolidate previous statement, assuming that the same exact amount of heat is stored in a single borehole GHE compared to the same amount of heat stored in a GHE consisting of multiple boreholes and total length of single borehole is equal to the sum of multiple boreholes, field of multiple boreholes would be capable of extracting more heat compared to a single pile due to rejected storage heat of boreholes located in the centre of the field still can be utilized by the boreholes located at the edges of borehole field in the process of storage heat dissipation.

In cold climate regions, where indoor climate conditions are generally ensured with heating, operation of GSHP plant during the heating season cools down the ground surrounding GHE. Installing a "free cooling" heat exchanger between the ground loop and cooling system buffer tank allows to partly cover buildings cooling demand via direct "free cooling".

Considering all abovementioned benefits of GSHP plant, it appears to be very attractive heat source option in nZEB design. Especially in regions, where no heat sources such as district heating with low primary energy conversion factors are available and only electricity energy source is present.

The purpose of this numerical study was to investigate the impact of various ground heat exchangers and thermal storage options along with their possible combinations on the heat pump plant heating/cooling performance in the design of nZEB – a commercial hall-type building HAMK Sheet Metal Center that was built based on the results of this study in cold climate of Hämeenlinna, Finland. Components applied in numerical study are – energy piles, vertical boreholes (heat wells), solar collector and/or exhaust air heat as a thermal storage source.

A whole year dynamic simulations are performed in Equa IDA-ICE simulation environment, where studied custom geothermal plant cases are modeled and coupled with an analytical building model. Most of plant components are modelled as a manufacturer specific products. Boreholes and energy piles are modelled with three-dimensional finite difference IDA-ICE borehole model extension, which accounts for pile/borehole and its inner components detailed geometry, thermal capacitances of components materials, thermal interactions between soil and pile/borehole components. Flat plate solar collector is modelled with IDA-ICE quadratic efficiency matched flow collector model, where thermal capacitance of fluid is taken into account. Ventilation heat recovery thermal storage heat exchanger, air handling unit, pumps, heating/cooling buffer tanks and heat pump with conveyance and control are modelled with standard IDA-ICE library components.

Results of 14 simulated cases with 5 year simulation period are presented in table and graphical form. Energy piles/boreholes, exhaust air, solar collector temperatures and yields of most important cases are presented, studied cases energy performance values (EPV) are calculated and compared to reference district heating case. Suggestions regarding GSHP plant whole year stable operation are presented. Impact of thermal storage on plant heating/cooling performance and possible reduction of ground heat exchanger length are discussed.

2. Methods

The modelling in IDA ICE was performed in advanced level interface, where user can manually edit connections between model components, edit and log model specific parameters, observe models code. An early stage building optimization (ESBO) plant, which is a part of a standard IDA model library, was utilized to generate the plant model. Abovementioned plant was modified to meet specific simulated case design intent. Total of 13 different plant modifications were modelled. A more detailed insight on geothermal plant modelling in IDA-ICE is presented in [9].

2.1. Building model data

Modelled plant modifications were coupled with HAMK Sheet Metal Center IDA-ICE model presented on Fig. 1, which geographical location is Hämeenlinna, Finland.

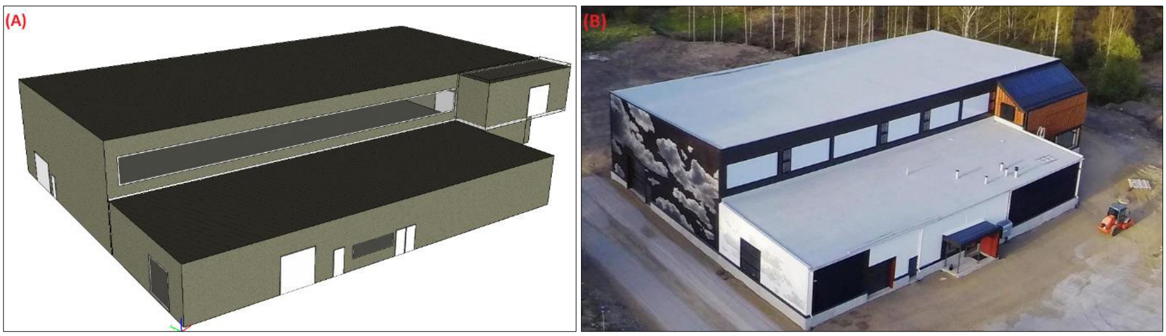

Fig. 1. (a) HAMK Sheet Metal Center model in IDA-ICE. (b) HAMK Sheet Metal Center in Hämeenlinna.

Ambient boundary conditions, regarding local weather data were described in Helsinki test reference year climate file [10] and applied in the simulation. In cold climate conditions of Finland, buildings indoor climate requirements are generally ensured with heating. Building's heating and cooling demand is met with radiant heating/cooling panels, which efficiency is fixed to 0.9. Table 1 presents a detailed overview of general parameters describing the building model.

Table 1. Building model parameters.

Descriptive parameter	Value
Location	Finland
Net floor area, m^2	1496.5
External walls area, U = 0.16 W/(m^2 K), m^2	1199
Roof area, U = 0.12 W/(m2 K), m^2	1475
External floor area, U = 0.14 W/(m2 K), m2	1496.5
Windows area, SHGC = 0.43, U = 0.78 W/(m^2 K), m^2	158
External doors, U = 1.0 W/(m^2 K), m^2	67
Heating set point, ^{o}C	18
Cooling set point, ^{o}C	25
Occupancy/lighting schedule	8:00-17:00(5d)
AHU operation schedule	7:00-18:00(5d)
Occupants, 1.2 met, 0.8 clo, no.	30
Lights load, kW	13
AHU air flow, m^3/s	2.2
AHU heat recovery, %	80
Measured air tightness, m^3/m^2 h	0.76 @50 Pa
Supply air temperature, ^{o}C	18
Heating/cooling room units	radiant panels
Heat load design temperature, ^{o}C	-26
Design heat load, kW	84
Heat pump capacity, kW	30

Building design heat load at design outdoor air temperature of -26 °C is 84 kW, while heat pump was sized to cover ca 35% of design heat load, which is 30 kW. As the peak heat load appears very rarely, it is much feasible to size heat pump within the range of 35…60% of total design load depending on the building function. At peak load operation electric top-up heating is applied to meet the rest 54 kW at design heat load conditions. In this particular building LED lights of main hall are controlled according to indoor illuminance level of 300 watt, which in combination with big windows for increased daylight penetration enabled to decrease lighting consumption by up to 60% compared to conventional ON/OFF 15 W/m^2 lighting solution.

2.2. Numerical study plan

Compiled numerical study plan consists of 14 cases, where case no 1 is a reference district heating case and other 13 are GSHP plant cases. Considered plant components are heat pump, energy pile fields of 16 units and 60 units with length of each pile 11 meters, boreholes (heat wells) either one or two with length of each borehole 200 meters, two different sources of thermal storage such as solar collector and/or in combination of both exhaust ventilation air thermal storage, connection of secondary side cooling system via direct "free cooling" heat exchanger to provide a "free cooling" option.

Numerical study plan with detailed description of each simulated case plant settings is depicted in Table 2.

Table 2. List of study cases plant settings.

Case no.	Case code	District Heating	Geothermal heat pump	Energy piles 11m		Boreholes 200 m		Thermal storage		Free cooling
				16 units	60 units	1 unit	2 units	Solar collector	AHUHX[1]	
1	DH	✓								
2	HP/EP16		✓	✓						✓
3	HP/EP60		✓		✓					✓
4	HP/B1		✓			✓				✓
5	HP/B2		✓				✓			✓
6	HP/EP16/SC		✓	✓				✓		✓
7	HP/EP16/AHU		✓	✓					✓	✓
8	HP/EP16/SC/AHU		✓	✓				✓	✓	✓
9	HP/EP60/SC		✓		✓			✓		✓
10	HP/EP60/AHU		✓		✓				✓	✓
11	HP/EP60/SC/AHU		✓		✓			✓	✓	✓
12	HP/EP16/B1/SC		✓	✓		✓		✓		✓
13	HP/EP16/B2/SC		✓	✓			✓	✓		✓
14	HP/EP60/B2/SC		✓		✓		✓	✓		✓

[1]*Exhaust ventilation air thermal storage*

2.3. Geothermal plant modelling

Fundamental scheme presented on Fig. 2 describes the connections between each individual components in simulated cases, where removing each individual component from a scheme still makes plant possible to operate.

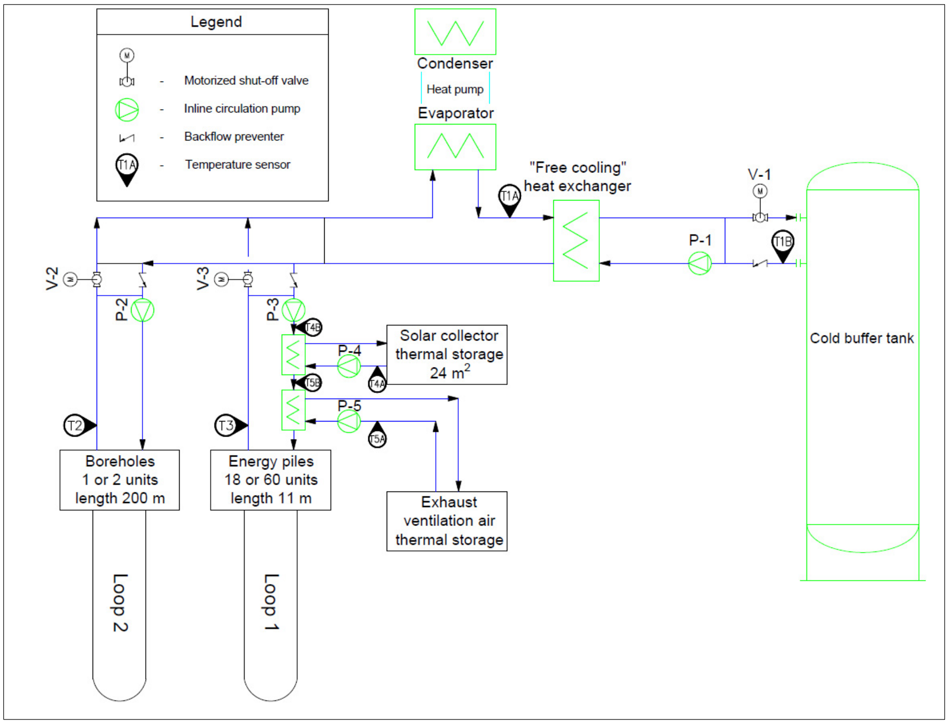

Fig. 2. Fundamental scheme of geothermal plant components.

Plant design considers option to separate energy piles loop via closing motorized valve (V-3) during the summer thermal storage cycle from the boreholes loop, in order to allow boreholes to provide “free cooling” while at the same time energy piles are being loaded with heat from source of thermal storage.

In order to prevent the formation of the ice in the ground and possible frost heave, geothermal loops brine outlet temperature should not drop below 0…-1 °C. Therefore, circulation pumps (V-2 and V-3) in each loop will stop when measured (T2 and T3) brine outlet temperature drops below the set point of 0 °C. Condenser side of the heat pump is connected to a hot buffer tank, in which heat carrier temperature is maintained according to a supply schedule (Fig. 3) temperature that is dependent on outdoor air temperature value with its maximal value of supply side +50 °C at design outdoor air temperature conditions of -26 °C.

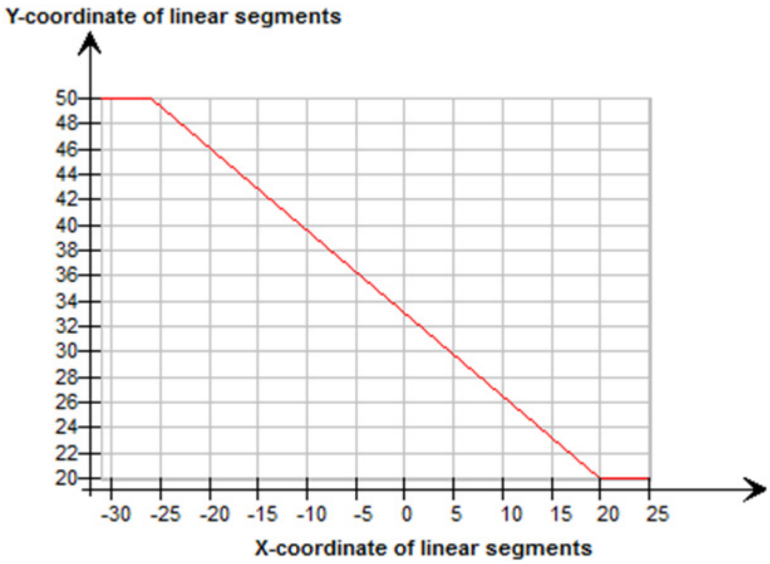

Fig. 3. Secondary side (radiant heating panels, AHU heating coil) supply temperature schedule.

Heat pump is capable of operation whenever the temperature in one of the loops is above the set point of 0 °C. On the contrary, heat pump stops its operation when there is no flow in the system (both loops are below the set point). With this control logics all the available geothermal energy will be absorbed.

Whenever the cooling cycle starts, there is no heat demand in the system and heat pump will not operate. As the heat pump is inactive, energy piles loop should be separated by e.g. three-way valve from evaporator circuit. In this case only boreholes (heat wells) are active and flow in their circuit goes through “free cooling” heat exchanger.

For each of two loops there is a separate thermal storage. In case of boreholes (heat wells), the required amount of heat is supplied during their free cooling operation.

Solar collector (with/without buffer tank) and/or exhaust air heat exchanger is applied as a thermal storage source in energy piles loop. Thermal storage source(s) is (are) connected via heat exchanger to energy piles loop inlet pipe. Whenever the heap pump is inactive, energy piles loop should be separated by e.g. three-way valve and design flow is maintained in energy piles, which are loaded with heat separately from energy wells. More detailed description regarding the plant modelling is described in [9].

Solar and exhaust air thermal storage are controlled according to a temperature difference (ΔT) set point logics, where two temperatures are measured and desired value of ΔT is maintained. In solar thermal storage loop ΔT = 6K and in exhaust air thermal storage loop ΔT = 4K. Measured temperatures in solar thermal storage loops on Fig. 2 are T4A and T4B. Whenever T4A temperature value is higher than 6K of T4B temperature value, pump P-4 starts it operation until the temperature of T4A reaches the desired ΔT = 6K. Same logics is applied in control of exhaust air thermal storage loop with temperature sensors T5A and T5B. Control of “Free cooling” loop operates by the logics “when beneficial” i.e. pump P-1 starts it operation whenever temperature T1B is higher than T1A.

2.3.1. Energy piles

Energy piles field in GSHP plant was modelled with IDA-ICE borehole model extension. Borehole model applies finite difference to calculate a number of temperature fields that combined by superposition generate the three-dimensional field. Model accounts for heat transfer between U-pipe, upward and downward flowing liquid, grout, ground, ground surface and ambient air. The length of each pile is assumed to be equal and ground homogeneous. Model considers the input of parameters (Table 3), which describe thermal and physical properties of ground, pipe, grout and brine. More detailed description of IDA-ICE mathematical model and its parameters is presented in [9].

Table 3. Energy piles field modelling parameters.

Descriptive parameter	Value
Energy piles amount, pcs	18 or 60
Energy piles depth, m	11
Energy pile diameter, mm	115
Distance between energy piles, m	3
Pipes outside walls distance, mm	52.4
U-pipe outer diameter, mm	25
U-pipe inner diameter, mm	20.4
Ground heat conductivity, W/(m K)	1.1
Ground volumetric heat capacity, kJ/(m3 K)	2019
Ground average annual temperature, °C	8
Energy pile grouting heat conductivity, W/(m K)	1.8
Grout volumetric heat capacity, kJ/(m3 K)	2160
Pipe material heat conductivity, W/(m K)	0.3895
Pipe volumetric heat capacity, kJ/(m3 K)	1542
Brine ethanol concentration, %	25
Brine freezing temperature, °C	-15
Brine heat conductivity, W/(m K)	0.43
Brine volumetric heat capacity, kJ/(m3 K)	4023
Brine density, kg/m3	969
Brine viscosity, Pa s	0.006
Borehole thermal resistance, (m K)/W	0.11
Prandtl number	57

2.3.2. Boreholes

Vertical boreholes (heat wells) were modelled with same IDA-ICE borehole model extension as energy piles field. Most important borehole model describing parameters are presented in Table 4.

Table 4. Boreholes modelling parameters.

Descriptive parameter	Value
Borehole maximum amount, pcs	2
Borehole depth, m	200
Borehole diameter, mm	115
Ground heat conductivity, W/(m K)	2.5
Other parameters are exactly same as in energy piles Table 3	-//-

It is worth to note, that boreholes in simulated cases were using same exact construction with single U-pipe loop grouted in 115 mm in diameter volume as energy piles. The difference compared to energy piles is length of the borehole of 200 meters and ground heat conductivity of 2.5 W/(m K) due to being drilled in granite compared to 1.1 W/(m K) heat conductivity of clay for energy piles field.

2.3.3. Solar thermal storage

Flat plate solar collector is modelled with IDA-ICE quadratic efficiency matched flow collector model, where thermal capacitance of fluid is taken into account. Most important parameters describing solar thermal storage are presented in Table 5. Mathematical model of match flow collector model is described in detail in [11].

Table 5. Solar thermal storage modelling parameters.

Descriptive parameter	Value
Solar collector area, m2	24
Orientation from south, °	20
Inclination angle, °	30
Conversion factor n_0	0.823
Loss coefficient a_1, W/m2K	3.44
Loss coefficient a_2, W/m2K	0.021
Longitudinal (50 °) K1	0.93
Transversal (50 °) K2	0.93

2.3.4. Exhaust ventilation air heat thermal storage

To model the exhaust ventilation air thermal storage, a standard IDA-ICE model library component of air-to-water heat exchanger "CCSIM" was connected to air handling unit exhaust side after the exhaust fan prior the flaps.

3. Results

Results of HAMK Sheet Metal Centre numerical study are presented in Table 6 as an average annual data obtained from simulations with duration period of 5 years. Description of each case geothermal plant setup can be found in Table 2. Breakdown by energy usage components is depicted as delivered energy values that account for efficiencies and distribution losses of the heating/cooling system's generation and consumption side. Pumps electricity was calculated based on pumps power consumption according sized according to friction losses method and time of pumps operation obtained as a result of simulation.

To assess the achievement of nZEB requirements for each simulated case an assumption to be made since in Finland nZEB requirements are still yet to be implemented. Nevertheless, in neighbor country Estonia nZEB definitions exist and achievement of nZEB level is accomplished by obtaining energy class "A" in energy performance certificate. Applying Estonian nZEB requirements practice in Finland, commercial hall-type building EPV should be ≤ 90 kWh/m^{2}a, which corresponds to energy class "A". Primary energy factor of electricity is 1.7 and 0.7 of district heating.

Table 6. Annual simulation results (average data based on five year plant operation).

Case no	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Units	Specific annual energy consumption per floor area (kWh/m^{2}a)													
District heating[1]	45.4	-	-	-	-	-	-	-	-	-	-	-	-	-
Top-up heating electricity[1]	-	34.6	16.4	25.2	9.6	27.7	28.5	24.1	9.4	10.1	7.2	12.6	4.6	2.8
Heat pump compressor[1]	-	2.7	6.7	4.6	8.4	4.1	3.9	4.9	8.0	8.0	8.6	7.5	9.1	9.2
Cooling electricity[1]	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Fans electricity	9.2	9.2	9.2	9.2	9.2	9.2	9.2	9.2	9.2	9.2	9.2	9.2	9.2	9.2
Pumps electricity	0.1	0.3	0.9	0.7	0.8	0.4	0.4	0.5	1.6	1.3	1.7	1.2	1.2	2.0
Lighting electricity	13.0	13.0	13.0	13.0	13.0	13.0	13.0	13.0	13.0	13.0	13.0	13.0	13.0	13.0
Domestic hot water[1]	4.0	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Energy performance value (EPV)	**74**	**104**	**81**	**92**	**72**	**95**	**96**	**90**	**73**	**73**	**70**	**77**	**66**	**64**
Units	Seasonal coefficient of performance value (SCOP)													
Heat pump SCOP	**-**	**4.50**	**4.61**	**4.60**	**4.48**	**4.47**	**4.50**	**4.50**	**4.61**	**4.60**	**4.62**	**4.60**	**4.60**	**4.68**
Whole plant heating SCOP	**-**	**1.24**	**1.98**	**1.52**	**2.52**	**1.43**	**1.41**	**1.56**	**2.42**	**2.42**	**2.68**	**2.20**	**3.11**	**3.27**

From total of 14 simulated cases, in 13 GSHP plant was a primary heat source and nZEB requirements were fulfilled in 9 out of 13 GSHP cases which corresponds to ca 70% success rate. In reference case with district heating case no 1 nZEB level was also obtained. Compared to district heating case, only 6 out of 13 GSHP plant setups, which is ca 46%, were able to yield EPV ≤ 74 kWh/m^{2}a (district heating case no 1). Top-up heating electricity value represents the energy consumption at point when ON/OFF heat pump was not able to meet building heat demand (due to evaporator entering temperature reached 0 °C limit) and top-up heating would provide additional energy to keep temperature in hot buffer tank according to desired set point. Cases not meeting nZEB level are single borehole GSHP plant, 16 energy piles GSHP plant without thermal storage and same plant with thermal storage source either solar or exhaust ventilation air heat. It is obvious, that in previous mentioned cases effective length of GHE was too small, which increase would lead to nZEB requirements achievement. SCOP values of GSHP in simulated cases varied within 4.47…4.68 range, while whole plant heating SCOP values with top-up heating and pumping energy consumptions taken into an account varied within 1.24…3.27 range. Energy performance of GSHP plants with exhaust air thermal storage was very similar to solar thermal storage plants performance. Combination of both thermal storage sources (case no 8 and 11) produced a noticeable decrease of EPV compared to single source of thermal storage, in case with 16 energy piles by ca 3 kWh/m^{2}a and in case with 60 energy piles by ca 6 kWh/m^{2}a. When plants without thermal storage performance are compared to plants with thermal storage application, in plant with 16 energy piles EPV decreases by 8…14 kWh/m^{2}a and in plant with 60 energy piles EPV decreases by 8…11 kWh/m^{2}a due to application of some thermal storage solution, which is ca 15% reduction of EPV. As a result of numerical study, case no 14 with two boreholes (heat wells) and 60 energy piles with solar thermal storage was selected for detailed design of HAMK Sheet Metal Center, which is by date built and being monitored. Compared to district heating case no 1, GSHP plant managed to decrease EPV by 10 kWh/m^{2}a and resulting whole plant heating SCOP is 3.27.

Geothermal energy yields plotted by year of GSHP plant operation of simulated cases are presented on Fig. 4. Geothermal yields increase with application of thermal storage and/or increase of effective GHE length. Important trend can be noticed, when borehole (heat wells) cases no 4 and 5 are compared to energy piles cases no 2 and 3 due to the difference in ground surface boundary conditions.

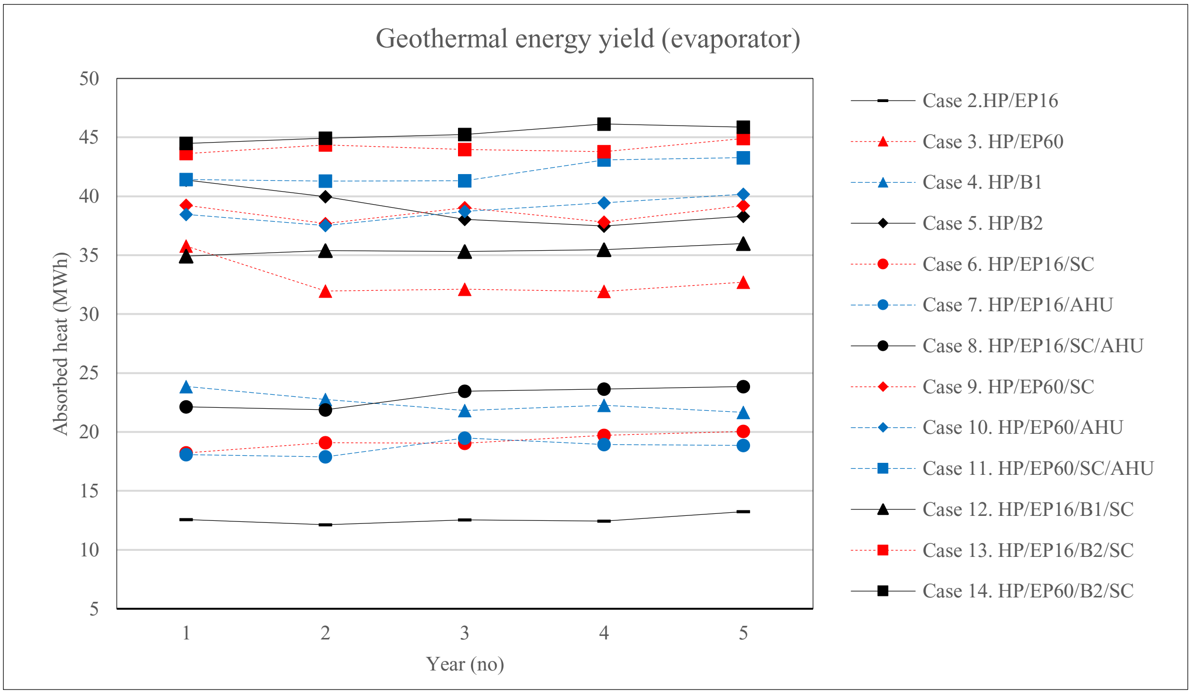

Fig. 4. Geothermal energy yield results.

As the boreholes are exposed to outdoor air, absorbed heat amount decreases over the years of operation, while energy piles ground surface is exposed to floor slab structure and heat losses through it provide a natural thermal storage effect stabilizing the long-term performance.

Table 7 describes simulated cases specific ground energy yields, "free cooling" energy performance and presents breakdown of applied thermal storage energy. To obtain absolute values in kWh/a, effective length of each case should be multiplied by either ground heat yield or storage source specific production value.

Table 7. Specific ground heat exchanger evaporator yield and thermal storage.

Case no.	Case code	Effective HX length	**Ground heat yield**	Thermal storage			
				Free cooling	Solar collector	AHUHX	**Total storage**
Units		m	**kWh/(m a)**	kWh/(m a)	kWh/(m a)	kWh/(m a)	**kWh/(m a)**
1	DH		**-**	-	-	-	-
2	HP/EP16	176	**71**	16	-	-	**16**
3	HP/EP60	660	**50**	5	-	-	**5**
4	HP/B1	200	**112**	15	-	-	**15**
5	HP/B2	400	**98**	8	-	-	**8**
6	HP/EP16/SC	176	**109**	2	92	-	**94**
7	HP/EP16/AHU	176	**106**	10	-	58	**68**
8	HP/EP16/SC/AHU	176	**131**	2	91	26	**120**
9	HP/EP60/SC	660	**58**	4	28	-	**32**
10	HP/EP60/AHU	660	**59**	4	-	26	**30**
11	HP/EP60/SC/AHU	660	**64**	3	27	12	**43**
12	HP/EP16/B1/SC	376	**94**	8	44	-	**52**
13	HP/EP16/B2/SC	576	**77**	6	28	-	**34**
14	HP/EP60/B2/SC	1060	**43**	3	17	-	**20**

Simulated cases ground heat yields vary within 43…131 kWh/(m a) range. Solar thermal storage in case no 6 of 92 kWh/(m a) resulted in ground yield of 109 kWh/(m a), while exhaust air heat of case no 7 of only 58 kWh/(m a) produced a yield of 106 kWh/(m a). Based on the previous, it can be concluded that exhaust air thermal storage utilizes more efficiently, as it is available for longer time period compared to high temperature solar storage, that is available mostly in summer and more of stored heat dissipates by the moment the plant start to utilize stored heat. Increase in pile amount from 16 units of energy piles to 60 units (quadrupling) resulted almost tripling the total amount of ground yield to 33 MWh/a. Application of combined thermal storage in case 8 of plant with 16 energy piles increased the total geothermal yield from 12.5 MWh/a without thermal storage up to 23 MWh/a almost doubling the yield amount and fulfilling nZEB requirements by yielding EPV of 90 kWh/m^{2}a. By utilizing interpolation, thermal storage increase in yield of case 8 can be compared to additional need to install ca 23 energy piles without application of thermal storage, which results in increase of pile field size by factor of 2.6. Borehole (heat well) specific ground heat yield in case no 4 was ca 60% higher when compared to energy piles yield in case no 2 due to higher ground conductivity in borehole case (Table 4) and no thermal interference compared to energy piles case.

In GSHP cases building cooling demand was covered only via "free cooling" and no active cooling was considered. However, in case with lowest free cooling energy production (1.8 MWh in case 6) indoor air temperature exceeded the desired temperature set point, which can be observed on Fig. 5 (a). This particular case has not met nZEB requirements and applied solar thermal storage raised the ground loop temperature in summer up to 40 °C, while on Fig. 5 (b) in case 13 (16.8 MWh of "free cooling" energy) with a combination of two boreholes and 60 energy piles with a solar thermal storage "free cooling" fluid temperature peaked at ca 20 °C, but managed to sustain indoor air temperature at desired set point of 25 °C during the whole cooling season.

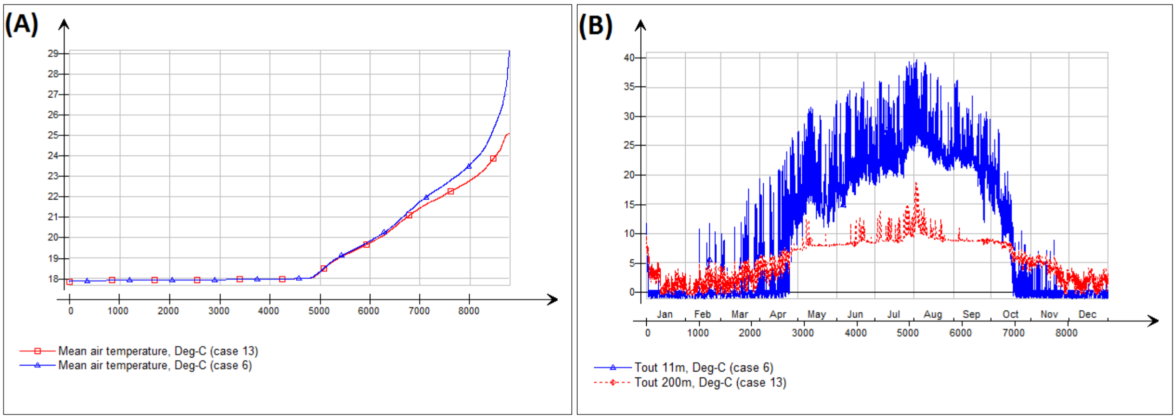

Fig. 5. (a) Indoor air temperature duration curve. (b) Fluid temperature in ground loop.

Efficient "free cooling" in case 13 resulted due to application of two boreholes, that during the cooling season were separated by the plant automatics from the energy piles with solar storage loop. Therefore, in summer period only boreholes were rejecting excess heat of building cooling system, while energy piles were loaded with solar produced heat.

Important aspect to consider in the design of heat pump plant with thermal storage is ground loop fluid temperature at the start of heating season. High temperatures in ground loop can be a problem for a heat pump, as the heat pump has its temperature operation limits, which in evaporator side is generally maximum of +20 °C. To overcome heat pump starting problem due to high evaporator loop temperature, a "free heating" heat exchanger can be an option, which should be connected to a hot tank for direct heating/DHW production until soil temperatures reaches heat pump operation range.

Annual yield of solar and exhaust air thermal storage is presented on Fig. 6. It can be observed, that solar storage on Fig.6 (a) is mostly active starting from mid of April and ending at end of September peaking in end of May. Peak power of thermal storage sources in simulated cases was very similar – 18 kW for solar and 21 kW for exhaust air thermal storage. Exhaust air thermal storage production on Fig. 6 (b) is more even over the year, but amount of thermal storage energy is capped by exhaust air temperature of up to ca 27°C (in case some overheating is present). Therefore, in theory source of solar storage due to higher temperature of operation can heat up the soil to higher temperature values storing more heat than exhaust air thermal storage source.

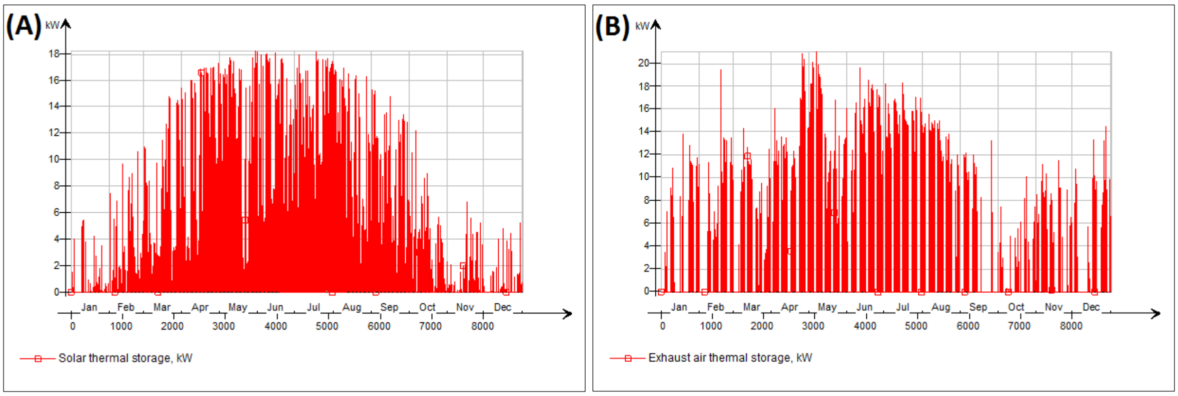

Fig. 6. (a) Case 9 solar collector thermal storage yield. (b) Case 10 exhaust air heat exchanger thermal storage yield.

Amount of thermal storage per square meter of solar collector varied within 675…765 kWh/(m^{2}a) range with average value of ca 710 kWh/(m^{2}a). While amount of exhaust air heat per 1 m^3/s of exhaust air heat with weekly operating schedule 5 days a week 7:00-18:00 in 16 energy piles case was ca 4.6 MWh/(m^3/s) and in 60 energy piles case ca 7.8 MWh/(m^3/s) with huge difference explained by the difference in soil temperature due to storage application (in less piles case temperature rises faster reaching exhaust air temperature limit of ca 25 °C compared to more piles case).

Ground loop evaporator entering brine temperatures are presented on Fig. 7, where temperature levels of plant with 60 energy piles and solar thermal storage on Fig. 7 (a) is compared to temperature levels of plant with exhaust air thermal storage.

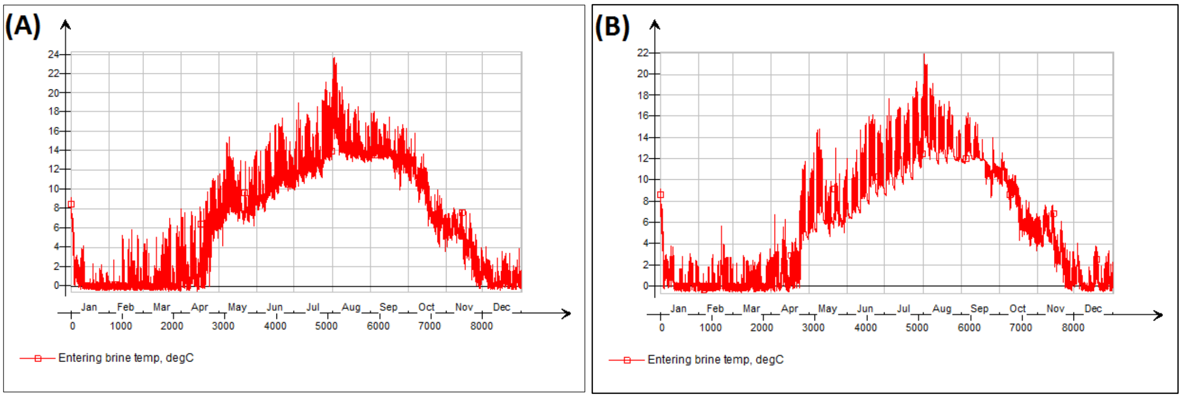

Fig. 7. (a) Case 9 evaporator entering brine temperature. (b) Case 10 evaporator entering brine temperature.

Regardless of thermal storage application, final ground loop temperature levels were most of the time below 18…20 °C and solar storage temperature in the summer was ca 2 °C higher compared to exhaust air thermal storage. Therefore, energy piles loop with thermal storage can still work as a “free cooling” source, which was accounted in simulated cases without boreholes.

During the GHE sizing and design stage it is important to consider extreme minimal temperatures at pile/borehole wall to avoid ice formations and frost heave. Fig. 8 presents piles soil temperature levels of most impacted cases no 2 with 16 energy and case no 4 with 1 borehole.

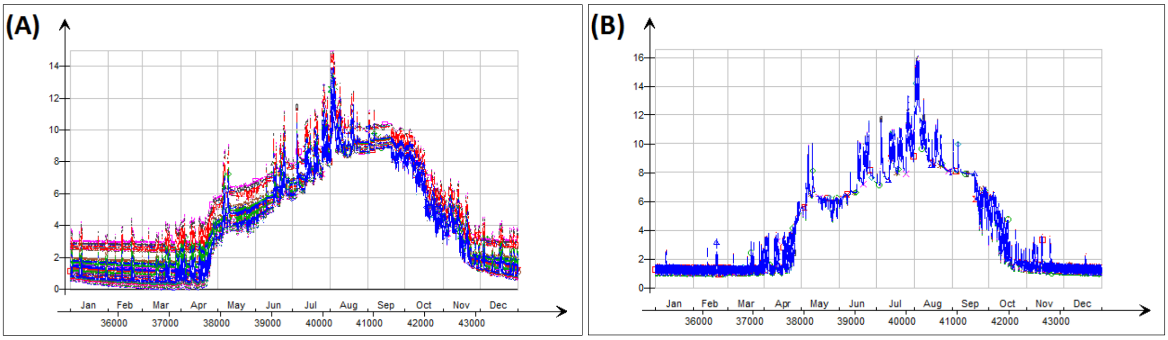

Fig. 8. (a) Case 2 soil temperatures at pile wall (b) Case 4 soil temperatures at borehole wall.

Fig. 8 (a) presents a temperature levels on each energy pile wall in simulation divided by 10 parts in axial direction and lowest temperature was ca 0 °C in the pile located at the center of the field. While, on Fig. 8 (b) minimal registered borehole wall temperature was ca 1 °C. It can be concluded, that in case 2 very high chance of ice formation and frost heave exists, while in case 4 not. Frost heave is the reason, why in geothermal plants energy piles/borehole outlet temperature should not go below °C to avoid freezing.

4. Conclusion

Numerical study revealed that a highly favorable GSHP plant solution was applied in design and construction of nZEB “HAMK Sheet Metal Center” due to high overall heating plant SCOP of 3.27, which was obtained in case with 60 units of 11 meter energy piles combined with 24 m2 of solar thermal storage combined and 2 units of 200 meter boreholes (heat wells). This GSHP plant yielded final EPV of 64 kWh/(m2a), which compared to district heating heat source case is ca 15% more efficient.

From the perspective of difference between boreholes (heat wells) and energy piles field without the application of thermal storage, boreholes specific ground heat yield per meter was ca 60% higher compared to energy piles due to ca 2.5 times better rock heat conductivity and appearance of thermal interaction in energy piles field.

Both studied thermal storage solutions are potential, reduce the initial length of energy piles field significantly and can be recommended for application to stabilize the long-term geothermal plant performance. Exhaust air thermal storage proved itself to be highly efficient solution which performance in simulations was very similar to solar thermal storage and it deserves additional research of its potential with different ventilation operation schedules.

Combination of solar and exhaust air thermal storage sources in studied case produced a reduction of energy piles field length by a factor of 2.6. Limits of exhaust air thermal storage potential are defined by the temperature of exhaust air, which in theory is the limit of possible temperature that soil can be heated up to. Therefore, in buildings with additional thermal storage need solar storage is the additional solution to be applied. Effective “free cooling” operation can be expected even in GSHP plants with energy piles and simultaneous thermal storage. However, it is important first to assess the temperature levels produced by the thermal storage application. Study revealed, that special care should be taken during the GHE sizing and design phase to ensure lack of freezing at the energy pile/borehole wall in order to avoid frost heave.

This study demonstrated the capability of IDA-ICE simulation environment in construction of detailed custom heating/cooling geothermal plants for a detailed numerical analysis which complicated models were successfully simulated. This study will be continued with real measurements and assessment of HAMK Sheet Metal Center geothermal plant energy performance.

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