ENERGY PILE AND HEAT PUMP MODELING IN WHOLE BUILDING SIMULATION MODEL

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ABSTRACT

With growing demand in improving building's energy efficiency, utilization of energy from renewable sources becomes more common. Due to lack of design guidelines, some simplified approaches are applied to solve complex building's geothermal heat/cold supply plants. This paper focuses on modelling of the detailed plant with heat pump and energy piles in IDA ICE. The results of commercial hall-type building case studies showed, that energy piles could be effective, as long as the thermal storage is considered. The comparison of results showed, that detailed simulation results might significantly differ from simplified borehole design software. Simplifications in detailed model computational grid can reduce computational time but affect the results accuracy.

INTRODUCTION

Geothermal energy, as a renewable energy source, is efficiently utilized with an application of ground-source heat pump (GHSP) coupled with ground heat exchanger (GHX). GHSPs are widely used all over the world (Lund et. al., 2010) to meet buildings with high energy performance heating and cooling demand. One of the cost-effective GHX solutions is geothermal pile foundation (Berber et. al., 2010) also called as energy piles. Energy piles has two main functions – building load transfer to ground and ground heat exchange.

Thermal performance of energy piles differ from the common borehole vertical heat exchanger performance. Solar radiation incident on the ground surface above the pile foundation is limited by the building. As the distance between energy piles is limited by the layout of foundation design, thermal interference between adjacent piles appear. The most recent European standard describing the design of heat pump heating systems EN 15450:2007 lacks design guidelines for GHSPs coupled with energy piles.

The design and thermal performance assessment of energy piles field is commonly performed in commercial software such as EED (Wood et. al., 2010), GLHEPRO (Spitler, 2000), Energy Plus and TRNSYS (Javed et. al., 2009). In this software the heat exchange between ground and a single borehole is modeled with Eskilson's g-function (Eskilson, 1987) approach and thermal interference between borehole field is modelled by temperature field superposition (Eskilson, 1986). Depending on the software, the applied heat pump model (Michel et. al. 2012) is either a simple quasi steady state performance map model or more complex parameter estimation based model (Jin et. al., 2002).

The purpose of the study was to model and assess the performance of the detailed heating/cooling plant with heat pump and energy piles in whole building IDA ICE simulation environment. The modelled plant considers application of recently developed three-dimensional model for an arbitrary combination of boreholes, parameter based heat pump model and standard IDA ICE model library components. Above mentioned plant is applied in the whole-year energy performance simulation of commercial hall-type building located in Helsinki, Finland.

The impact of borehole thermal resistance and long-term application of energy piles on absorbed ground heat amount is studied. Results of the whole-year simulation are compared to GLHEPRO simulation results. An overview of borehole and heat pump model input parameters is presented. The Impact of borehole model computational grid decrease on simulation duration and results accuracy is discussed.

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 actual building's plant design intent.

Building model description

The modelled plant was applied in a commercial hall-type one storey building (Figure 1) located in Helsinki, Finland.

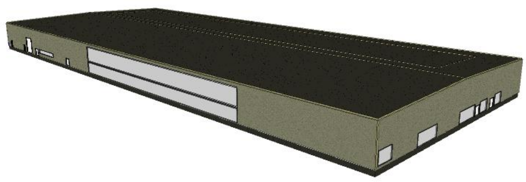

Figure 1 Building mode 3D view in IDA ICE, the dimensions of the building were 149.5 by 61 m.

Thermal and physical envelope properties of building are describe in Table 1. Ambient boundary conditions, regarding local weather data were described in recently updated Helsinki test reference year climate file (Kalamees et. al., 2011) and applied in the simulation. When located in cold climate, indoor climate requirements in commercial hall-type buildings are generally ensured with heating. The heating and cooling energy use ratio in particular building is 1 to 0.056.

Table 1
Building envelope description

CONSTRUCTION	AREA, m^2	U,W/(K m^2)
External walls	2552	0.16
Roof	9123	0.15
External floor	9120	0.09
Windows (SHGC=0.51)	726	0.97
External doors	89	0.96

To meet heating and cooling demand of the building, radiant heating/cooling ceiling panels were considered in the design. Abovementioned room units were modelled with standard IDA ICE component library model, where manufacturer specific performance is described with the input of power law coefficient and exponent values. In Table 2, building model descriptive parameters such as occupational hours, internal heat gains, building operation schedules,

Table 2
Model settings description

INPUT DATA	VALUE	UNITS
Location	Helsinki	-
Net floor area	9119.5	m^2
Heating set point	18	°C
Cooling set point	25	°C
Occupancy/lighting schedule	8:00-21:00	h
AHU operation schedule	7:00-22:00	h
Occupants (1.2 met/0.8 clo)	213	no.
Lights load	72.9	kW
AHU air flow	10.1	m^3/s
AHU heat recovery	80	%
Air tightness	0.3 @50 Pa	ACH
Supply air temperature	18	°C
Radiant heating/cooling panels (b=1.2m)	750	m
Heat load design temperature	-26	°C
Design heat load	463	kW
Heat pump capacity (4 units)	172	kW
Annual heat demand	222	MWh
Annual cooling demand	12.5	MWh

design heat load and other relevant settings are described in more detail.

Heat pump model description

The design intent was to size the heat pump at ca 40% of building's design heat load at design ambient air temperature -26 °C. The rest of the peak load was meant to be covered with electric top up heating.

Model of the brine-to-brine heat pump from the standard IDA ICE component library was applied in the study. The purpose of heat pump modelling was to achieve manufacturer specific heat pump performance (Figure 2) in the simulation. The parameter-based heat pump model consists of a heat exchanger model (Brandemeuhl, 1993) and compressor performance descriptive correlation model. The heat exchanger model, which is based on the NTU-method, describes heat pump condenser and evaporator. The heat pump model input parameters are described in Table 3. It is assumed, that four manufacturer specific (Figure 2) heat pumps will be coupled in parallel.

Table 3
Heat pump model settings description

MODEL PARAMETERS	VALUE	UNITS
Nominal capacity at rating	171.2	kW
COP (rating)	4.6	
dTlog_evaporator (rating)	3	°C
dTlog_condenser (rating)	5	°C
Tin_evaporator (rating)	0	°C
Tout_evaporator (rating)	3	°C
Tin_condenser (rating)	30	°C
Tout_condenser (rating)	35	°C
B (compressor parameter)	4.2	%
C (compressor parameter)	-1.7	%
E (compressor parameter)	0.6	%
F (compressor parameter)	2.1	%

The performance of the heat pump model are described with compressor parameters B, C, E and F presented in the table above.

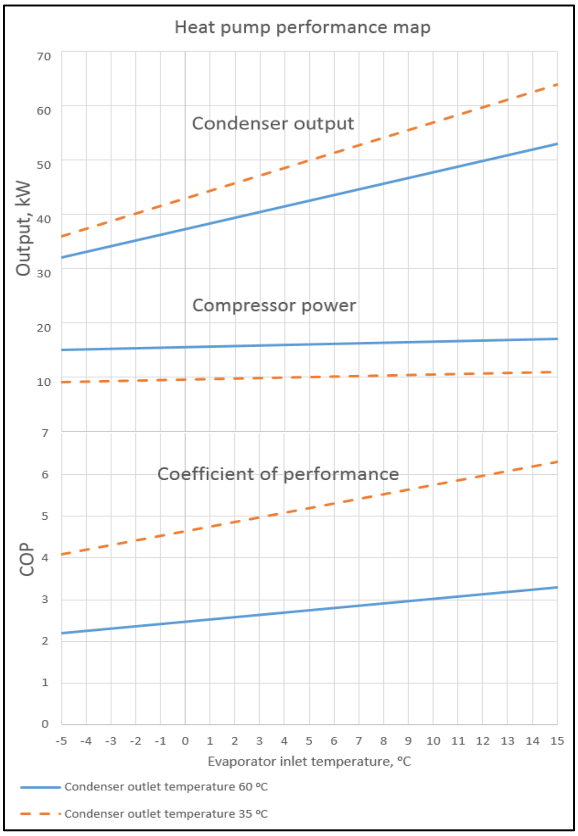

Figure 2 Heat pump performance map

To acquire compressor correlation parameters values, that describe heat pump performance outside of the rating conditions (Table 3) range, an identification model (Figure 3) was developed.

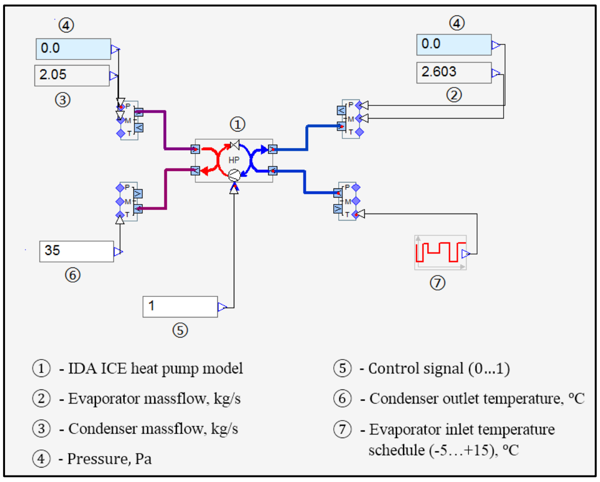

Figure 3 Parameter identification model

The input in parameter identification model considers the application of heat pump performance map data. As the performance map is based on the evaporator inlet temperature range, it is modelled with a schedule (-5…+15 °C). Two cases with condenser outlet temperature +35 °C and +60 °C were simulated and compressor parameters were modified to match the performance map.

Borehole model description

Foundation of 196 energy piles (Figure 5) was modelled with IDA ICE borehole model extension. The model applies finite difference (Giardina, 1995) to calculate a number of temperature fields that combined by superposition generate the three-dimensional field.

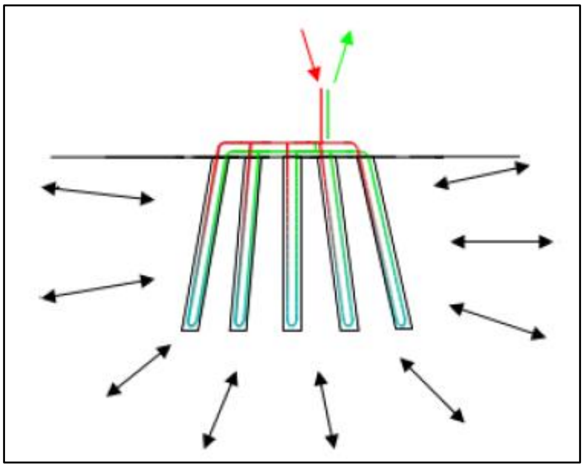

Figure 4 Heat flows in borehole model

Model accounts for heat transfer (Figure 4) 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 4), which describe thermal and physical properties of ground, pipe, grout and brine.

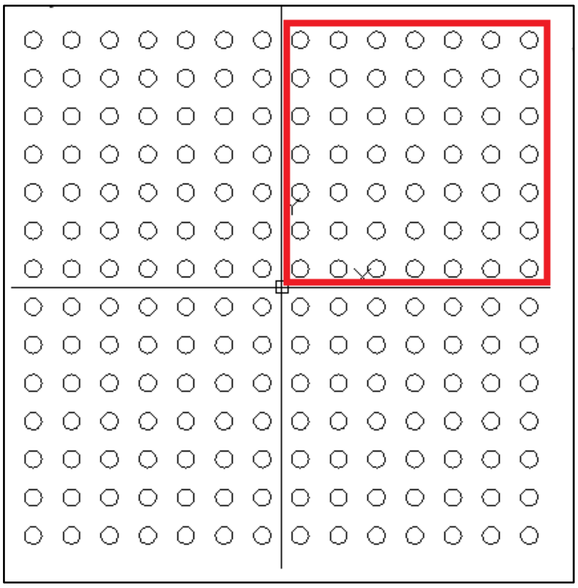

Figure 5 Energy piles layout, the size of the field was 58.5 x 58.5 m and the distance between piles 4.5 m.

In order to decrease the duration of the simulation with high amount of boreholes, the mirror option (Figure 6) was implemented. In this particular study, mirror option 2 (Figure 6) was applied. Therefore, only 49 of

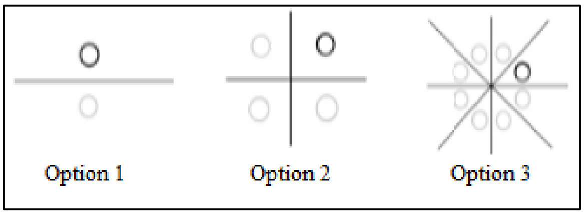

Figure 6 Borehole field mirror

196 energy piles were actually defined in the borehole model. Additionally, the model enables user to modify the computational grid (Figure 7) to reduce the duration of simulation. One test case was studied and

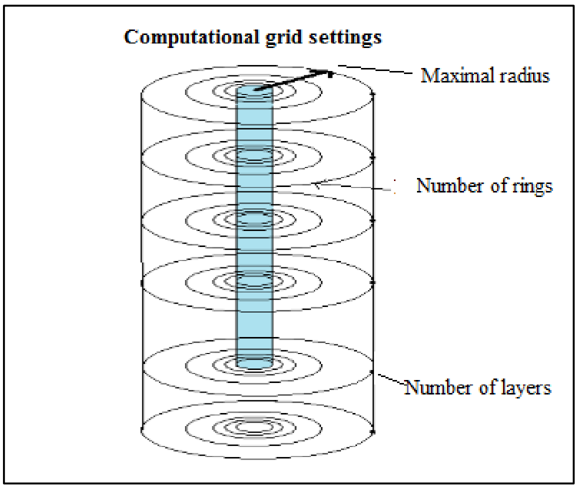

Figure 7 Computational grid

results of the decreased grid simulation are presented in the results section of this paper. Heat transfer between ground surface and floor structure is described with the input of ground surface heat transfer coefficient model parameter.
Table 4 shows more detailed description of model input parameters and values applied in this study in modelled energy piles foundation.

Table 4
Borehole model settings description

MODEL PARAMETERS	VALUE	UNITS
Borehole amount	196	pcs
Borehole depth	15	m
Borehole diameter	115	mm
Distance between boreholes	4.5	m
Pipes outside walls distance	52.4	mm
U-pipe outer diameter	25	mm
U-pipe inner diameter	20.4	mm
Ground heat conductivity	1.1	W/(m K)
Ground volumetric heat capacity	2019	kJ/(K m^3)
Ground average annual temperature	+8	°C
Borehole grouting heat conductivity	1.8	W/(m K)
Grout volumetric heat capacity	2160	kJ/(K m^3)
Pipe material heat conductivity	0.3895	W/(m K)
Pipe volumetric heat capacity	1542	kJ/(K m^3)
Brine ethanol concentration	25	%
Brine freezing temperature	-15	°C
Brine heat conductivity	0.43	W/(m K)
Brine volumetric heat capacity	4023	kJ/(K m^3)
Brine density	969	kg/m3
Brine viscosity	0.006	Pa s
Borehole thermal resistance	0.1	(m K)/W
Heat transfer coefficient at ground surface	0.09	W/(K m^2)
Prandtl number	58	

There are two input values in the model that should be calculated by user. One is the borehole thermal resistance. In this study, it was calculated with following equation (Shonder et. al., 1999):

$$R_b = \frac{1}{2\pi k_g} \ln\left(\frac{d_b}{d_p\sqrt{n}}\right) \quad (1)$$

Another model input value that should be calculated is Prandtl number:

$$Pr = \frac{C_p \mu}{\kappa} \quad (2)$$

Two cases with different borehole resistance were studied and presented in the results section of this paper.

Plant description and operation principles

The modelled detailed plant (Figure 10) heating equipment consists of the brine-to-brine heat pump coupled with energy piles. The condenser side of the heat pump is connected to the stratification tank (hot). An additional electric boiler, the top up heating, is connected to the same tank, to meet building peak heating loads. The heat pump is modelled to meet the performance maps of manufacturer specific product.
Plant cooling equipment consists of the stratification tank (cold) connected to the ground heat exchange loop via heat exchanger to provide free cooling effect, that operates only when needed. No additional cooling equipment such as chiller is considered.

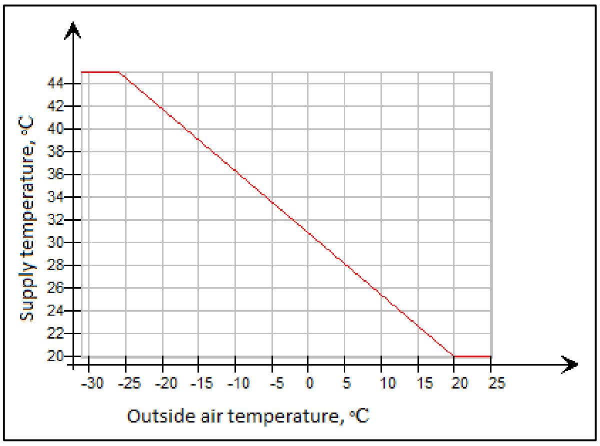

Figure 8 Secondary side supply schedule

The operational principles of the plant were intended to fulfil several conditions. Secondary side supply water temperature, i.e. the heating curve (Figure 8) was controlled according to outdoor air temperature.

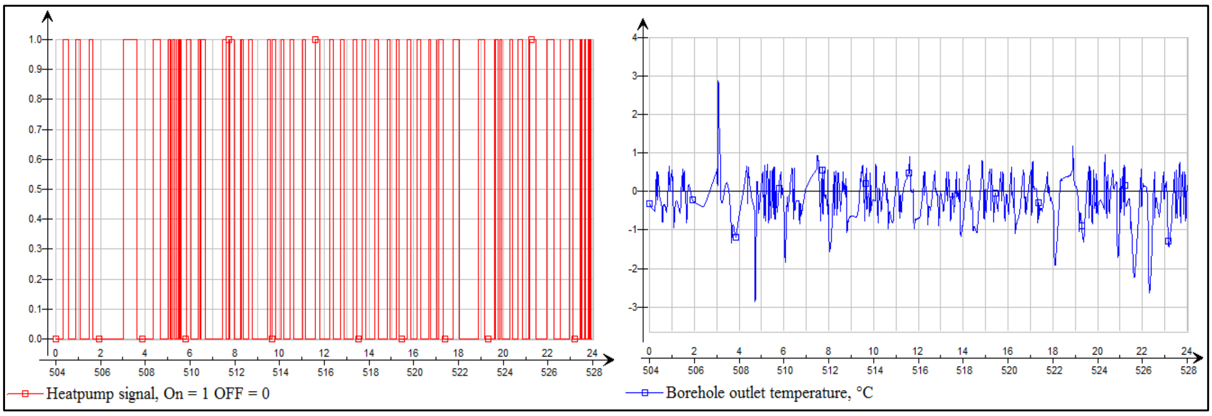

Figure 9 Heat pump operation

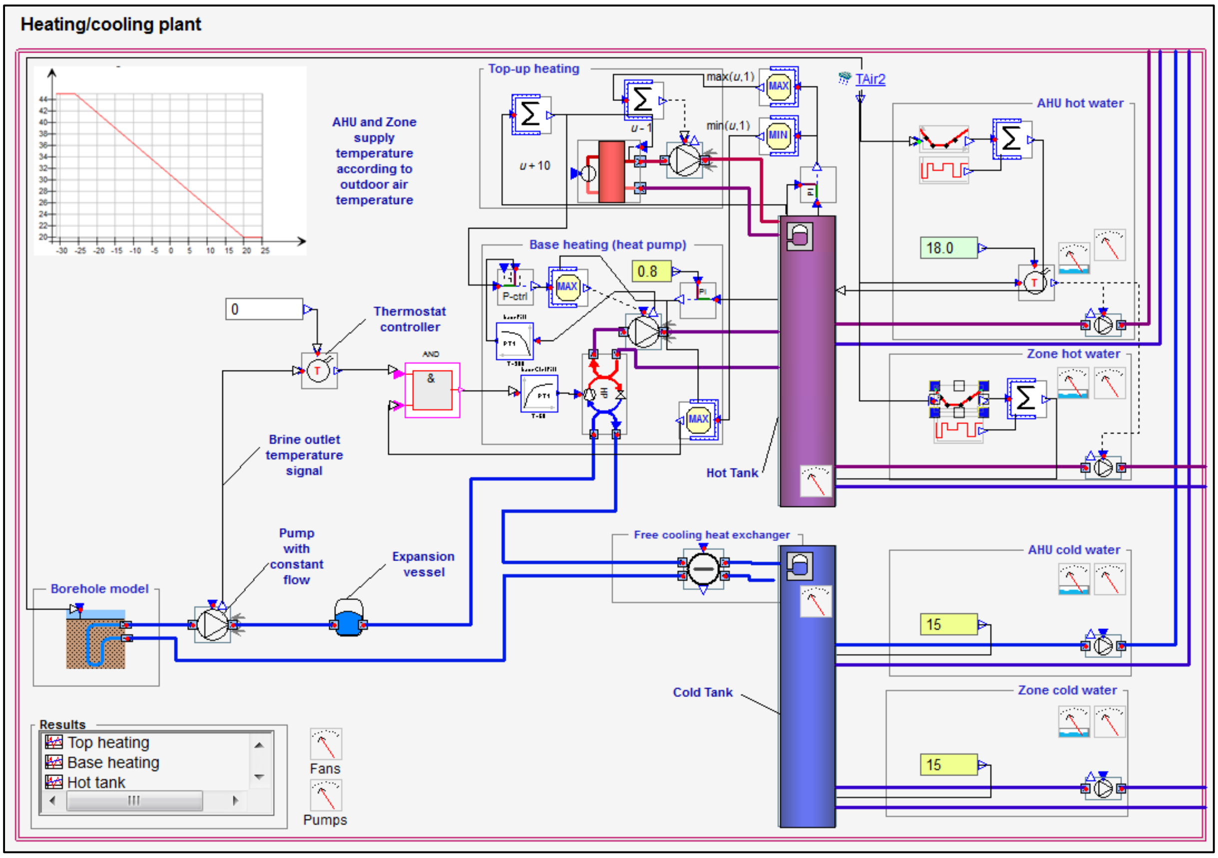

Figure 10 Detailed heating/cooling plant modelled in IDA

In order to avoid ice formation in the underground layers, a thermal controller model was applied to switch off the heat pump, when temperature of the brine supplied from the borehole drops below 0 °C. As the high capacity ground-source heat pumps usually lack compressor inverter, a logical on-off controller model was applied to allow heat pump on-off (no part load) operation.

The operation of the plant with described control logic is presented on the Figure 9. Due to the thermostat dead band of 1 K, delivered brine temperature has dropped below the desired set point. In order to avoid numerical difficulties and errors, that prevent the simulation from running, a signal smoother of control signal was applied. Due to long integration time in signal smoother, the heat pump was still operating, when thermostat controller signal was "off". As a result delivered brine temperature has dropped (Figure 9) sometimes below the dead band of thermostat.

RESULTS

The simulation results of modelled detailed heating/cooling plant in commercial hall-type building are presented in Table 5.

In first case, thermal resistance of energy piles was almost twice smaller, though the reduction of absorbed annual ground heat amount in second case was negligible.

An overall heating system seasonal coefficient of performance (SCOP), that considers top-up heating and circulation pumps energy use, has ranged from 1.92 to 2.02 depending on the test case.

Table 5

Simulation results

DATA	1[1]	2[2]	1[1]	2[2]
	MWh		kWh/m²	
Absorbed heat	110.5	107	12.1	11.7
Absorbed cooling	11.6	11.2	1.3	1.2
HP Compressor	26.7	24.8	2.9	2.7
HP Condenser	138.3	128.9	15.1	14.1
Top-up heating	81.6	90.7	8.9	9.9
Circulation pumps	8.2	8.2	0.9	0.9
Heat pump SCOP	5.18	5.2	-	-
Heating system SCOP	2.02	1.92	-	-

[1] $R_b = 0.1$

[2] $R_b = 0.19$

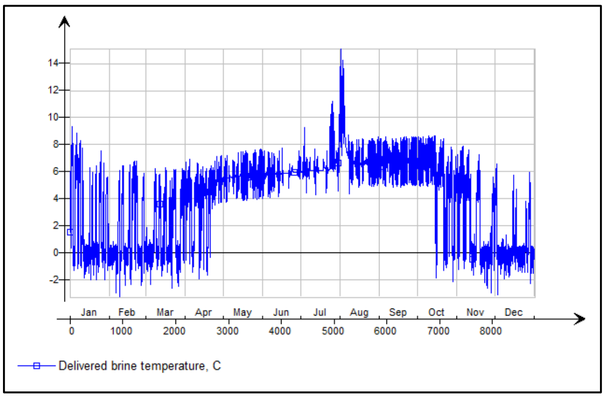

Figure 11 Delivered brine temperature

High value of heat pump seasonal coefficient of performance (SCOP) was achieved due to controlled supply heat carrier temperature schedule and thermostat control, which limits heat pump operation. Based on the simulation results, calculated specific heat extraction rate of simulated energy piles field was ca 30 W/m and annual heating energy yield was ca 47 kWh/m.
Delivered brine temperature (Figure 11) peaked at +14.5 °C during the summer. At that moment radiant cooling ceiling panels, due to their limited cooling capacity, could not maintain air temperature (Figure 12) set point below +25 °C. Most of the time indoor air temperature stayed within set point range.

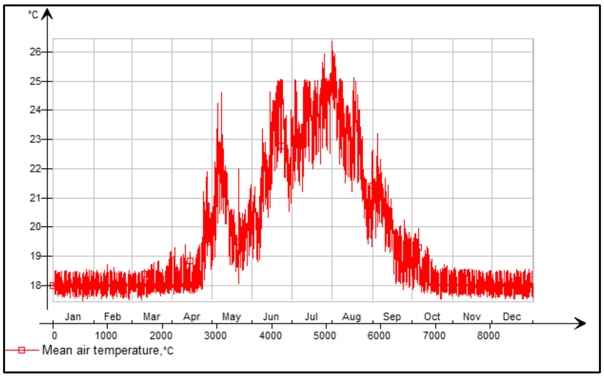

Figure 12 Indoor air temperature

Computational time of one-year period simulation, where 49 energy piles were mirrored to produce 192 energy piles field lasted ca 20 hours. Similar simulation with 252 energy piles lasted ca 30 hours.
Therefore, long-term simulation of 20-year period was first conducted with simplified computational grid. Then, a three-year simulation with detailed grid was conducted to verify the accuracy of simplified grid. The results of detailed grid with plotted trend line and simplified grid simulations are described in Figure 13.

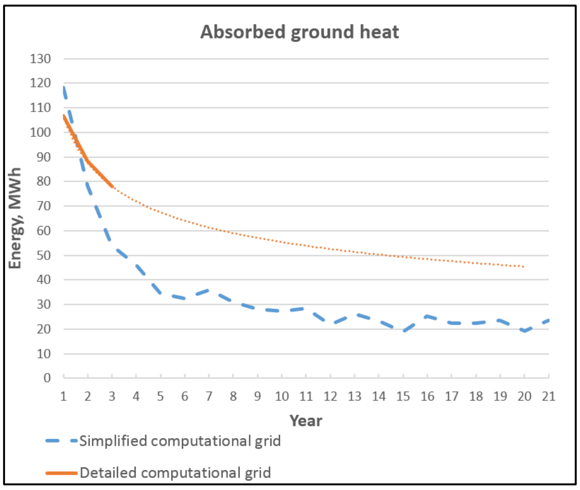

Figure 13 Long-term simulation results

Long-term performance of energy piles field was reducing over the years. There is a significant difference between detailed and simplified grid simulations results. The computational time in simplified grid simulation was 2 days i.e. 14 days reduction in computational time compared to detailed simulation expected duration.
Additional simulations were carried out to compare detailed simulation performance to widely used design tool GLHEPRO. The results of comparison simulations are shown in Table 6.

Table 6
Software results comparison

DATA	GLHEPRO	IDA ICE	UNITS
Energy piles	195	196	pcs
Evaporator	135319	108152	kWh
Compressor	52682	42202	kWh
Condenser	188000	150354	kWh
COP	3.57	3.56	-

At identical settings and boundary conditions, energy piles in GLHEPRO software were able to absorb 135.3 MWh of ground heat, while energy piles in the modelled plant only 108.1 MWh.

DISCUSSION AND CONCLUSIONS

Considering high heat pump SCOP of 5.2 the modelled plant in commercial hall-type building performed well. It can be confirmed (Berber et. al., 2010) that energy piles may become cost-effective even in cold climates.
Low SCOP of total heating system can be explained by the insufficient amount of energy piles. As the specific heat extraction rate resulted in 30 W/m, the appropriate heat pump capacity for 196 energy piles, with overall length of 2940 m, according to VDI 4640 should be ca 112 kW instead of 172 kW. As only a part of buildings foundation performed as energy piles, there is some room for performance increase.
Additional major advantage in energy piles application is the ability to meet building cooling demand via free cooling, in this case without the need of additional cooling equipment.
Long-term simulation results showed a reduction in energy piles heat extraction performance over the years of operation. As the amount of heat extracted from energy piles is that much higher than loaded during cooling season. It can be concluded, that there is a need for a thermal storage in commercial hall-type buildings with energy piles.
The significant difference in the results acquired by GLHEPRO and detailed modelled plant defines the importance of detailed modelling, especially in large buildings.
Despite the fact, that IDA ICE allows the user to perform very detailed simulations with freedom to construct specific plant solutions, it may become very complicated to non-experienced users such as designers, which can lead to erroneous results and design decisions.
The second downside of detailed simulations was in computational times. Based on the results, it can be

concluded that reduction in borehole model computational grid, to conserve time, will lead to significant difference in results compared to detailed grid simulation.
This study will be continued with performance and economic aspects of thermal storage applications, which are needed to provide a stable operation of energy pile field.

NOMENCLATURE

Pr,	Prandtl number;
C_p,	specific heat, W/(m K);
μ,	dynamic viscosity, Pa s;
κ,	thermal conductivity, W/(m K);
R_b,	borehole resistance, (m K)/W;
k_g,	grout thermal conductivity, W/(m K);
d_b,	borehole diameter, m;
d_p,	pipe diameter, m;
n,	number of U-pipes.

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