DESIGN AND OPTIMUM CONTROL OF A SWEDISH DUAL-SOURCE (AIR AND GROUND) HEAT PUMP SYSTEM

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ABSTRACT

The objective of this work is to evaluate if the combination of ambient air and the ground as heat source for heat pumps is technical and economical advantageous for heating only purposes. The simulation results of this study are based upon the actual design of a dual-source heat pump plant for a residential association area in the southwest of Sweden. The main finding is that a dual-source concept is to prefer when compared to just using ambient air or the ground as heat source for heat pumps. The study also show that increasing the number of boreholes or the size of the ambient air heat exchanger of the actual plant could be technical and economical beneficial. The control strategy used regarding ambient air temperature when switching between ambient air and the ground as heat source has only a minor influence of the plant performance. Due to economical reasons a control strategy without any recharging of the ground is also most advantageous in this case.

Key words: ***heat pumps, dual-source, hybrid, ground, ambient air, design, control strategies***

1 INTRODUCTION

In general, a geothermal or ground-source heat pump system means that a geothermal sink/-source is used for the condensing process in cooling mode and the evaporation process in heating mode. When ambient air is used as sink/source in combination with the ground this is often called a dual-source or hybrid geothermal heat pump system. The dual-source concept can of course also refer to other combined heat sinks/sources for heat pumps involving solar heat, ground water, or water from a lake or a river.

Only a few articles found are dealing in detail with dual-source (ambient air and ground) heat pump applications. In general, these articles refer to applications operating in both heating and cooling mode with a dominating cooling load. This means that the “dual-sources” mainly are used as heat sinks for the condensing process in cooling mode. No articles found are dealing with dual-source heating mode only heat pumps for use in cold climates.

However, the combination of ambient air and the ground as heat sources for a heat pump also appears to be advantageous for heating purposes only, due to both technical and economical reasons. The main reasons are that a relatively higher evaporating temperature can be achieved during the warm part of the year and the investment cost for heat capacity is lower for an ambient air heat exchanger compared to drilling more boreholes.

Such a dual-source heat pump plant is designed and built to meet the base heating and hot water demand of an 82 row-house residential association area in the southwest of Sweden. The heat exchanger for ambient air is a regular dry cooler and the ground heat exchanger is 17 boreholes of about 200 meters depth equipped with double U-pipes. Existing oil and electrical boilers provide backup and peak heating.

The ambient air heat exchanger (the dry cooler) is to be used as primary heat source when the ambient air dry-bulb temperature is above +3 °C and the ground heat exchanger is to be used during the colder part of the year. The annual mean ambient temperature on site is +8 °C and the temperature drops below +3 °C during approximately 2200 hours per year.

The dual-source concept aims at providing an economically optimal combination with a reduced first cost and a nearly preserved seasonal performance factor compared to a conventional ground source system.

The reduced first cost in this case is mainly due to the lower cost of heat capacity via the ambient air heat exchanger compared to the drilling of more boreholes. Another reason for using a dual-source configuration could be a lack of available ground area for all boreholes needed for a ground source only heat pump.

In this paper the actual system is evaluated concerning both technical and economical issues. The actual design is compared to several alternative configurations regarding the number of boreholes, the size of the ambient air heat exchanger, and the heat pump size. Special emphasis is also devoted to optimum control strategies, including the most favorable set point for switching between ambient air and the ground as heat source and possible improvement of the system performance by alternative strategies for recharging of the ground by heat from the ambient air.

2 THE CASE STUDY PLANT

In 2004, a dual-source heat pump system was taken into operation to provide for the base heating and hot water need of an 82 row-house residential association area in the southwest of Sweden. A heat pump application was chosen mainly due to good experiences from the old system to be replaced, a heat pump with ambient air only as heat source. As an anecdote, a bio-fuel plant was not considered in this case due to the need of frequent truck transports of fuel, even though a bio-fuel plant might have been more economical favorable. The old oil and electrical boilers in the old system were kept for backup and peak heating demands. The heating capacities of the old oil and hot water boilers are 550 kW and 18 kW.

The two R407C heat pumps (build on site) each consists of two scroll compressors and four welded plate heat exchangers operating as evaporator, hot-gas cooler, condenser and sub-cooler. The sub-cooler and the hot-gas cooler are connected in series to provide hot water of higher temperature. The rated cooling capacity of each compressor is 46 kW.

On the heat source side a pipe circulating system with a water-ethanol solution (32%, freezing point –15 °C) is used to transport heat from the ambient air heat exchanger and/or the ground heat exchanger to the heat pumps. A sketch of the plant configuration is shown in Fig. 1.

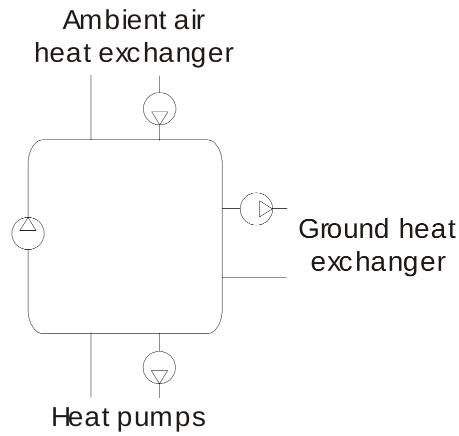

Fig. 1. Sketch of the actual plant configuration

The ambient air heat exchanger is an ordinary dry cooler equipped with 12 fans with a rated capacity of 207 kW. The ground heat exchanger is 17 boreholes of about 200 meters depth with double U-pipes (DN40 PN6). The boreholes are located in two parallel lines, displaced to each other so the mean distance between all adjacent boreholes is 15 m. The ground soil mainly consists of granite with an assumed thermal conductivity of 3.5 W/m/K and a volumetric heat capacity of 2.2 MJ/m^3/K. The designed (short-term) specific heat extraction rate is 50 W/m, which gives a ground heat extraction capacity of about 170 kW.

3 MODELS AND ASSUMPTIONS

For the simulation study several component models are required and in order to simplify the calculation several assumptions where made. The models and associated assumptions are described below. All models except the ground heat exchanger model are directly implemented in the software EES (EES 2005). The ground model source code is written in Fortran and DLL-linked to EES. In general, steady state simulations are performed hour by hour but the time dependent temperature response of past heat exchanges with the ground is taken into account.

The results presented in this paper mainly come from a master of science work accomplished at building services engineering, Chalmers. Therefore, more detailed information regarding actual conditions is found in the master thesis (Hoflund 2004).

3.1 Hot water and heating power demands

The actual heating loads of the case study plant have been logged on a monthly basis during several years so the heating energy demands are quite well known. These monthly heating energy demands have been converted into an average heating power value. Thereby, the monthly mean heating power demand can be expressed as function of the monthly mean ambient temperature. The mean hot water power demand is assumed to be constant 40 kW and the space heating power demand is simulated by a simple polynomial expression found by linear regression.

In the simulations hourly measured weather data (SMHI 1983-1992) for Gothenburg is used as input. Weather data for the years 1985 and 1992 is used, where the former corresponds to a typical cold year and the later to a typical warm year.

3.2 Heat storage

In order to simplify the calculations all heat rejected from the heat pump assumes to be supplied to a single heat storage tank of 6 m^3. In the real system no such a storage tank exist, the condenser is connected to the space heating system, and the sub-cooler and the hot gas cooler are connected in series to the hot water system. The average tank temperature is supposed to be the heat source for the space heating system. Thereby, the actual supply and return temperatures of the space heating system can be taken into account when calculating condenser heat rejection rate possible to utilize due to temperature restrictions. Because of the real systems gas-cooler coupling, the possible hot water temperature is assumed to be 15 K higher than the average tank temperature. Thereby, the heat pump temperature restrictions can be taken into account regarding hot water heating.

In all simulations the mean tank temperature is allowed to vary in between 35 °C and 50 °C and thus the hot water temperatures possible to reach varies in between 50 °C and 65 °C. The incoming tap water temperature is assumed to be constant 10 °C (the annual mean temperature) and the leaving hot water temperature should be 60 °C. The average tank temperature gives the heat pump condensing temperature.

3.3 Heat pump performance

The heat pump performance is characterized by the manufacturer's specification of the actual compressors. The rated cooling capacity of each compressor is 46 kW and the heat rejection rate is 55 kW (refrigerant R407C, evaporating temperature –10 °C, condensing temperature+45 °C, suction gas superheat 5 K and liquid line subcooling 20 K). The general performance is modeled by polynomial expressions for the cooling capacity, motor electrical power input and heat rejection rate as a function of the evaporation and condensing temperature at ISO international standard conditions. The evaporation and the condensing temperature are simply assumed to be 5 K below the mean brine temperature and 5 K above the mean storage tank temperature respectively. Thus, the evaporation temperature is directly dependent on the actual heat source temperature and the actual heat storage tank temperature gives the condensing temperature.

In the simulations the heat pump capacity is varied by on-off regulation of the four compressors, which means in each time step there are three different heat rejection rates possible in between zero and maximum heat rejection rate.

3.4 The ambient air heat exchanger

The ambient air heat exchanger of the real plant is an ordinary dry cooler with a rated capacity of 207 kW (ambient air temperature +3 °C, brine temperature in –5 °C, brine temperature out –3.4 °C). The performance is modeled by a UA-value of 28.8 kW/K, where the temperature on the cold side is assumed to be the mean brine temperature. As an basic assumption the ambient air heat exchanger is not supposed to be in operation when the ambient air dry bulb falls below +3 °C and by that the need of defrosting is avoided.

In the simulations the capacity assumes to be regulated by electronic speed control of the 12 fans. In the real plant the capacity is controlled by on-off regulation of each fan.

3.5 The ground heat exchanger

The ground heat exchanger model is based on the theory presented in (Claesson et al. 1985) and (Eskilson 1987). The time dependent temperature response factors, the G-functions, for the actual borehole configuration is determined based on outputs from the software EED (EED 2000). If the G-functions for the actual borehole configuration are known the time dependent borehole temperature can be calculated for a constant heat extraction or rejection rate:

$$t_{borehole}(\tau) = t_{undisturbed\ ground} - \frac{\dot{q}}{2 \cdot \pi \cdot \lambda} \cdot G(\tau) \qquad (^\circ\mathrm{C}) \tag{1}$$

$t_{borehole}(\tau)$	Time dependent average borehole wall temperature (required)	(°C)
$t_{undisturbed\ ground}$	Undisturbed mean ground temperature	(°C)
$\dot{q}$	Specific heat extraction or rejection rate	(W/m)
λ	Ground average thermal conductivity	(W/m/K)
$G(\tau)$	Time dependent temperature response factor	(-)

Using the software EED, all heat transfer resistances between the borehole wall and the brine can be taken into account. Thereby, modified G-functions can be calculated that applies to the following expression:

$$t_{brine}(\tau) = t_{undisturbed\ ground} - \frac{\dot{q}}{2 \cdot \pi \cdot \lambda} \cdot G_{modified}(\tau) \qquad (^\circ\mathrm{C}) \tag{2}$$

$t_{brine}(\tau)$	Time dependent mean brine temperature (required)	(°C)

The time dependent temperature response of varying heat extraction and heat rejection rates can be calculated by super-positioning:

$$t_{brine} = t_{undisturbed\ ground} - \sum_{i=1}^{n} \left(\frac{(\dot{q}_i - \dot{q}_{i-1})}{2 \cdot \pi \cdot \lambda} \cdot G_{modified}(\tau_i) \right) \qquad (^\circ\mathrm{C}) \tag{3}$$

Where index *i* denotes the actual time step for constant time step lengths.

In order to carry out long-term simulations hour by hour further modifications has to be made due to calculation time. In this case the long-term performance is supposed to be 15 years of operation. In this work the same weather data is used for each year, assuming that 15 cold or 15 warm years will occur after each other. Thereby, the long-term temperature response of the ground can be calculated by using a yearly constant/mean heat extraction rate for the preceding years based on the net heat extraction during the first year of operation:

$$\tilde{\dot{q}} = \frac{q_{net,\ year\ 1}}{8760} \qquad (\mathrm{W/m}) \tag{4}$$

$\tilde{\dot{q}}$	Yearly mean heat extraction rate	(W/m)
$q_{net,\ year\ 1}$	Net energy heat extraction during the first year of operation	(Wh/m)

The long-term degradation can thus be calculated as the difference between the response of the yearly mean heat extraction rate year 15 and year 1. The brine temperature (required) a certain hour during year 15 of operation is calculated by super-positioning of the corresponding value year 1 and the long-term degradation. The long-term degradation is slightly overestimated by this approach.

3.6 Pumps and fans

The electrical power demands for pumps are roughly estimated based on design data for the actual heat pump plant. All pumps are working with constant flow rates. The electrical power demand for the ambient air heat exchanger fans are calculated based on the manufacturer's performance data for the electronic speed control. In the parameter study simple corrections of the electrical power demands are made based on the actual change of the plant configuration.

When changing the number of boreholes in the parameter study the total brine flow rate assumes not to be changed only the flow per borehole. When changing the size of the ambient air heat exchanger, the total pipe length is increased or decreased. The brine flow rates and flow resistance through the evaporators of the heat pumps are assumed not to change due to a change of the heat pump capacity. Thereby, only a changed flow resistance affects the calculated change of electrical power demand for the pumps. The ambient air heat exchanger fan electrical power demand is assumed to increase/decrease in proportion to the change of the heat capacity.

4 SIMULATIONS

In order to evaluate the case study plant a base case scenario is defined aiming to resemble the actual plant configuration. The base case configuration is mainly described in the text above. In all cases both the short- and long-term performance is studied where the long-term performance corresponds to operation during year 15. To study the effect of the ambient climate, weather data for both the cold year 1985 and the warm year 1992 is used. Results from three different studies are presented in this paper and these studies are briefly described below.

4.1 Dual sources vs. single sources

A main question is if the dual source concept is to prefer when compared to just using ambient air or the ground as heat source. In the base case the total heat source heat transfer area consists of the ground and the ambient air heat exchanger. When the base case is to be compared with one of these "extreme" cases the necessary heat transfer area of the single source component has to be increased. As a rough estimate the heat transfer area has to be doubled. In this case the total number of boreholes has to be about 36 (compared to the 17 boreholes of the base case) if using just the ground as heat source. This means the total heat transfer area is increased by 112 %. The same value is also used for the increase of the ambient air heat exchanger; the corresponding UA-value is set to 61.1 kW/K. In both cases the increased heat transfer area assumes to be connected in parallel with the base case heat transfer area. Therefore, in this case, the increased pump and fan electrical power demand calculated is related to the additional section.

For the case when using ambient air only as heat source the ambient air heat exchanger must be in operation when the ambient air temperature is lower than +3 °C. This means that defrosting is required during the colder period of the year. In the simulations, defrosting is just regarded as a decrease of the heat pump net heating energy output by 10 %.

4.2 Alternative control strategies

Control strategies in this context just refer to principal strategies in form of heat source used and set points, no dynamic behaviors are considered. In the base case control strategy the actual heat source used is based on the ambient air temperature. The ground is used when the ambient air temperature is below +3 °C and ambient air when the temperature is higher. When the ambient air temperature is higher than +12 °C, heating energy from the ambient air is used to recharge the ground by means of the brine is circulated trough the ground heat exchanger before entering the heat pump. This base case control strategy is also referring to as control strategy one.

Control strategy two implies that the brine is always circulated trough the ground heat exchanger when the ambient air temperature is higher +3 °C. This means that heating energy from the ambient will be supplied to and/or ground heat will be received from the ground depending on actual ambient air and ground temperatures.

Control strategy three means that no recharging of the ground at all is applied, ambient air or the ground are used as independent heat sources for the heat pump.

Finally, control strategy four is a variant of the base case strategy. The difference is that the ambient air set point for switching between ambient air and the ground as the major heat source is set to be +6 °C.

4.3 Parameter study

The final part of this paper deals with a parameter study of the actual plant configuration using the base case control strategy. The parameters studied are the number of boreholes, the size of the ambient air heat exchanger, and the size of the heat pump. In the base case, the number of boreholes is 17 and here an increase or decrease of 3 boreholes is studied. This means an increase or decrease of the ground heat exchanger area by about ±18 %. The size of the ambient air heat exchanger is also studied for an equal change of the heat exchanger area, an increase, or decrease of the UA-value with 18 %. Also a change of the heat pump size by ±18 % is studied.

The parameter study is performed as a factorial design at two levels (Box et al. 1978). Using this methodology, main effects of each parameter change as well as possibly interactions among the parameters can be evaluated. By studying a low and high level of each studied parameter, the total number of combinations to be calculated is $2^3 = 8$. These 8 combinations are studied for both a warm and a cold year and in a short and long terms perspective (1985 and 1992, 1st year of operation and 15th year of operation). Thus, the total number of simulations made to create input for the parameter study is 32.

4.4 Uncertainty and calculation errors

The simulation results are based on a number of assumptions and simplifications, therefore the uncertainty of the results is hard to evaluate. However, the aim was to make a general study of the dual source heat pump concept and the conclusions in general are supposed to be valid.

Nevertheless, after all the calculations where made a systematic calculation error was detected. The polynomial expressions describing the heat pump performance were underestimating the heating energy taken up by the evaporator by approximately 10 %. The consequence of this is that the resulting mean brine temperature is about 1.0 °C to high. The long-term degradation is by that slightly underestimated. At the same time, the long-term degradation is slightly overestimated by the ground heat exchanger model, which somewhat compensates for the systematic calculation error. Regarding heating energy demand coverage factors and seasonal performance factors the influence of this is negligible because the polynomials describing the heat pump condensing heat rejection rate and compressor work is accurate.

5 RESULTS

To precede the results given below, one can conclude that the dual source concept seems to be competitive compared to the single source concepts. Furthermore, a change of the dual-source base case configuration seems to improve the performance but the choice of control strategy only has a minor influence. All economic figures are given in Swedish crowns (SEK) and the current exchange rates are: 1 SEK ≈ 0.14 USD ≈ 0.11 EUR.

5.1 Dual sources vs. single sources

The main results of the simulations are given in Table 1 and Table 2. In the first table the results are valid for typical warm years and in the second table for typical cold years.

Table 1. The base case compared to just using the ground or ambient air as heat source (warm years)

	Base case		Ground source only		Ambient source only	
Operational year	1	15	1	15	1	15
Heating energy demand (MWh)	1490	1490	1490	1490	1490	1490
Heating energy from the heat pumps (MWh)	1420	1400	1440	1400	1233*	1233*
Compressor electric energy use (MWh)	449	448	429	488	446	446
Pump electric energy use (MWh)	65	65	78	81	76	76
Fan electric energy use (MWh)	20	21	0	0	50	50
Total electric energy use (MWh)	534	534	507	569	572	572
Additional heating energy use (MWh)	70	90	50	90	257	257
Heat pump operational time (h)	7806	8201	8037	8248	7809	7809
$SPF_{heat\ pump}$	3.16	3.13	3.36	2.87	2.76	2.76
SPF_{system}	2.66	2.63	2.84	2.46	2.16	2.16
Heating energy demand coverage factor (%)	95	94	97	94	83	83

* *Reduced by 10% due to the need of defrosting*

Table 2. The base case compared to just using the ground or ambient air as heat source (cold years)

	Base case		Ground source only		Ambient source only	
Operational year	1	15	1	15	1	15
Heating energy demand (MWh)	1800	1800	1800	1800	1800	1800
Heating energy from the heat pumps (MWh)	1470	1400	1540	1450	1233*	1233*
Compressor electric energy use (MWh)	479	479	474	518	482	482
Pump electric energy use (MWh)	65	65	81	83	80	80
Fan electric energy use (MWh)	17	18	0	0	54	54
Total electric energy use (MWh)	561	562	555	601	616	616
Additional heating energy use (MWh)	330	400	260	350	567	567
Heat pump operational time (h)	7823	8224	8366	8506	8193	8193
$SPF_{heat\ pump}$	3.07	2.92	3.25	2.80	2.56	2.56
SPF_{system}	2.62	2.50	2.77	2.41	2.00	2.00
Heating energy demand coverage factor (%)	82	78	86	81	69	69

* *Reduced by 10% due to the need of defrosting*

The heating energy demand coverage factor for the ground source only configuration is high during the first year of operation both for a cold and a warm year. This is due to the great heat capacity of the undisturbed ground. However, the long-term decrease of performance is considerable. The characteristics of an ambient air only configuration is reverse, the heating energy demand coverage factor is low during the first year of operation but in a long-term perspective unaffected. In all cases the heating energy demand coverage factor is of course lower for cold years compared to warm years of operation due to the greater heating energy demand.

In the base case configuration, the short-term advantages connected to the ground source and the long-term advantages connected to the ambient air source can be utilized. Worth mentioning here is that the heating energy demand coverage factor for the base case is just slightly lower compared to the ground source only configuration looking at year 15 of operation.

Looking at the resulting brine temperature for the different configurations, it can be concluded that the lowest brine temperatures will arise using the ambient air source only configuration. Besides the need for defrosting also possibly pumping problems due to the higher viscosity must be taken into account in practice. The brine temperature variations are smallest for the ground source only configuration.

To compare the performance of three different configurations one can also study the seasonal performance factor (SPF). In Table 1 and Table 2, SPF-values are given related to both the compressor electrical energy use ($SPF_{heat\ pump}$) and system total electric energy use (SPF_{system}). An analysis of the SPF-values, in principal, gives the same main result as for the energy demand coverage factor.

A quite simple economic comparison of the three different configurations has also been made. The results of this economic analysis are given in Table 3. The investments for the two single source configurations are based on the actual investment for the case study plant. Only investments related to the heat extraction components are considered. The cost for electricity as well as the cost for additional heating is set to be 1.0 SEK/kWh, which is based on normal Swedish rates. The mean annual net costs are based on an interest rate of 5 % and 20 years of operation. According to this economic estimation, the base case configuration seems to be most profitable.

Table 3. Investments, mean annual operating costs and mean annual net costs

	Investment (SEK)	**Annual cost for electric energy (SEK)**	**Annual cost for additional heating (SEK)**	**Annual net cost (SEK)**
Base case	1 450 000	548 000	230 000	894 000
Ground source only	2 329 000	569 000	201 000	957 000
Ambient air source only	742 000	594 000	412 000	1 066 000

Current exchange rates are: 1 SEK ≈ 0.14 USD ≈ 0.11 EUR.

5.2 Alternative control strategies

Alternative control strategies for the base case configuration have been studied and the results are compiled in Table 4, Table 5 and Table 6.

Table 4. Comparison of different control strategies for operational year 1

	Base case		**Control strategy 2**		**Control strategy 3**		**Control strategy 4**	
	Warm year	**Cold year**	**Warm year**	**Cold year**	**Warm year**	**Cold year**	**Warm year**	**Cold year**
Heating energy demand (MWh)	1490	1800	1490	1800	1490	1800	1490	1800
Heating energy from the heat pumps (MWh)	1420	1470	1400	1460	1420	1470	1400	1460
Compressor electric energy use (MWh)	449	479	430	471	449	479	447	479
Pump electric energy use (MWh)	65	65	76	73	61	62	65	65
Fan electric energy use (MWh)	20	17	21	18	19	15	14	14
Total electric energy use (MWh)	534	561	527	562	529	556	526	558
Additional heating energy use (MWh)	70	330	90	340	70	330	90	340
Heat pump operational time (h)	7806	8201	7810	8203	7799	8192	7809	8204
$SPF_{heat\ pump}$	3.16	3.07	3.26	3.10	3.16	3.07	3.13	3.05
SPF_{system}	2.66	2.62	2.66	2.60	2.70	2.64	2.66	2.62
Heating energy demand coverage factor (%)	95	82	94	81	95	82	94	81

Table 5. Comparison of different control strategies for operational year 15

	Base case		Control strategy 2		Control strategy 3		Control strategy 4	
	Warm year	Cold year	Warm year	Cold year	Warm year	Cold year	Warm year	Cold year
Heating energy demand (MWh)	1490	1800	1490	1800	1490	1800	1490	1800
Heating energy from the heat pumps (MWh)	1400	1400	1380	1390	1390	1390	1380	1390
Compressor electric energy use (MWh)	448	479	441	479	447	478	453	481
Pump electric energy use (MWh)	65	65	76	74	61	62	65	65
Fan electric energy use (MWh)	21	18	23	19	19	15	15	15
Total electric energy use (MWh)	534	562	540	572	527	555	533	561
Additional heating energy use (MWh)	90	400	110	410	100	410	110	410
Heat pump operational time (h)	7823	8224	7840	8233	7799	8194	7828	8235
$SPF_{heat\ pump}$	3.13	2.92	3.13	2.90	3.11	2.91	3.05	2.89
SPF_{system}	2.62	2.49	2.56	2.43	2.64	2.50	2.59	2.48
Heating energy demand coverage factor (%)	94	78	93	77	93	77	93	77

Studying the results one can conclude that the alternative control strategies influence of the plant performance is rather small.

Some general observations found in Table 4 and Table 5 can however be worth pointing out. The long-term expected profit of recharging the ground is not that great looking at for example SPFs or heating energy demand coverage factors. Some of the net profit is lost due to the increased electricity use for pumps and fans when recharging.

Switching between the ground and ambient air as heat source at a higher ambient air temperature influences the net heat extraction from the ground. The net result of a higher set point temperature is that the brine temperature and accordingly the evaporation temperature will be lower during the colder part of the year. Thus, the electricity savings in form of decreased use of the fans is lost when operating the heat pump at a relatively lower evaporation temperature.

To make an economic comparison of the different control strategies one can look at the mean annual operating costs. The results given in Table 6 are based on electricity and additional heating costs of 1.0 SEK/kWh, an interest rate of 5 % and 20 years of operation.

Table 6. Mean annual operational costs

	Base case		Control strategy 2		Control strategy 3		Control strategy 4	
	Warm year	Cold year	Warm year	Cold year	Warm year	Cold year	Warm year	Cold year
Annual operational cost (SEK)	614 000	926 500	633 500	942 000	613 000	925 500	629 500	934 500

Current exchange rates are: 1 SEK ≈ 0.14 USD ≈ 0.11 EUR.

According to Table 6, the base case strategy and strategy three result in almost equal operational costs. Thus, the excluded utilization of recharging in strategy three does not give rise to increased annual operating costs. Comparing with strategy two, where utilization of recharging is increased, shows that the operational cost rises. Therefore, the profit of recharging seems to be negative but the relative differences in annual operational costs are rather small. Therefore, the general conclusion is that the control strategy chosen does not influence the annual operational costs much.

5.3 Parameter study

The main results of the base case simulations can be found in Table 1 and Table 2. The parameters varied in this parameter study are the number of boreholes, the size of the ambient air heat exchanger, and the size of the heat pump.

Most values presented below are affected by different interaction effects, which mean the values are dependent on the size of the other components. However, the actual interaction effects are not commented in the text.

5.3.1 Influence of the total electric energy use

The total electricity energy use is mainly affected by the heat pump size. This is quite obviously because the compressor electric energy usage is dominating. The average increase of electricity use is about 56 MWh for warm years and 106 MWh for cold years when the heat pump size is increased from an 18 % smaller to an 18 % larger compared to the actual size. Assuming linear relations the heat pump size influence of the total electric energy usage can be expressed:

- +1.6 MWh per % increase of heat pump size (for warm years)
- +3.0 MWh per % increase of heat pump size (for cold years)

If the number of boreholes increases from 14 to 20 the total electricity use is decreased by 13.5 MWh operational years 1 and 16 MWh operational year 15. These values are not affected by warm or cold years. With a linear relation assumption the number of borehole influence of the total electric energy usage can be expressed:

- -2.3 MWh per borehole (1:st year of operation)
- -2.7 MWh per borehole (15:th year of operation)

Finally, the size of the ambient air heat exchanger has only a minor influence of the total electricity use.

5.3.2 Influence of the additional heating demand

The additional heating demand decreases in all cases if the studied component sizes are increased. An increase of the heat pump size (from –18 % to +18 %) affects the additional heating demand most. The average decrease of the additional heating demand is about 96 MWh for warm years and about 180 MWh for cold years. These values are the same in both a short-term and a long-term perspective. An assumption of linear relations gives:

- -2.7 MWh per % increase of heat pump size (warm years)
- -5.1 MWh per % increase of heat pump size (cold years)

Increasing the number of boreholes from 14 to 20 results in a decrease of the additional heating demand by about 11.5 MWh for a warm first year of operation. The decrease is about two times greater in a long-term perspective (15 years) and about three times greater for cold years. Thus, after 15 cold years the annual decrease is about six times greater. With linear relations this can be expressed:

- -1.9 MWh per borehole (a warm 1:st year of operation)
- -3.8 MWh per borehole (a warm 15:th year of operation)
- -5.7 MWh per borehole (a cold 1:st year of operation)
- -11.4 MWh per borehole (a cold 15:th year of operation)

The annual decrease of additional heating demand when the ambient air heat exchanger size is increased is insignificant in a short-term perspective. After 15 year of operation the annual decrease of additional heating demand is 12 MWh for warm years and 33 MWh for cold years. Expressing this with linear assumptions:

- -0.3 MWh per % increase of the ambient air heat exchanger (warm 15:th year of operation)
- -1.0 MWh per % increase of the ambient air heat exchanger (cold 15:th year of operation)

If to decrease the additional heating demand a large heat pump, a large number of boreholes and a large ambient air heat exchanger are to prefer. This implies a larger plant and a larger plant naturally means a greater heating energy demand coverage factor. The question is of course if this is obtained in a technical optimal way. Therefore, a similar study of the heat pump and system seasonal performance factors are made.

5.3.3 Influence of the seasonal performance factor

The heat pump seasonal performance factor ($SPF_{heat\ pump}$) is most affected by a change of the heat pump size (from –18 % to +18 %). The SPF-value actually decreases if the heat pump size is increased. If increasing the ambient air heat exchanger size or the number of boreholes the SPF-value increases. This can be expressed:

- -0.004 units per % increase of the heat pump size
- +0.002 units per % increase of the ambient air heat exchanger
- +0.015 units per borehole

The system seasonal performance factor (SPF_{system}) is affected in a similar way. This is of course due to the fact that the compressor stands for the major part of the total electric energy use. However, an increase of the number of boreholes is the factor that affects the system seasonal performance factor most.

5.3.4 Economic consequences

The final part of the parameter study is to check whether it is economic justifiable to change the number of boreholes, the size of the ambient air heat exchanger or the size of the heat pump. An assumption of linear relations, an interest rate of 5 %, 20 years of operation, cost for electricity as well as for additional heating of 1.0 SEK/kWh gives the annual savings shown in Table 7.

Table 7. Annual savings

	Annual savings (SEK)
Increasing the number of boreholes by 3	+11 100
Increasing the size of the ambient air heat exchanger by 18 %	+10 100
Increasing the size of the heat pump by 18 %	-14 400

Current exchange rates are: 1 SEK ≈ 0.14 USD ≈ 0.11 EUR.

Table 7 clearly shows that investing in a larger heat pump is not a good option. Whether a larger number of boreholes or a larger ambient air heat exchanger is the best choice is not so easy to say. However, the annual savings of a specific investment can clarify this. The outcome of an investment of 100 000 SEK is given in Table 8.

Table 8. Annual savings due to an investment of 100 000 SEK

	Size change	**Electricity use savings (SEK)**	**Additional heat savings (SEK)**	**Annual savings (SEK)**
Larger number of boreholes	+1.6 boreholes	4 160	10 080	6 200
Larger ambient air heat exchanger	+28 %	0	11 200	3 200

Current exchange rates are: 1 SEK ≈ 0.14 USD ≈ 0.11 EUR.

Table 8 shows that the best choice is to invest in more boreholes. However, it should be noted that these figures are only valid in the vicinity of the base case design due to the assumptions regarding linear relations. A reasonable guess is that the positive response of the number of boreholes or the ambient air heat exchanger size is decreasing for larger numbers. Calculations with 36 boreholes and everything else unvaried gave a negative value of the annual savings, -3 600 SEK (-504 USD) per borehole.

6 CONCLUSIONS

The main conclusion is that a dual-source heat pump seems to be an advantageous configuration in general compared to just using ambient air or the ground as heat source. Therefore, the dual-source concept should always be considered when a ground source heat pump is considered, especially if the available ground area for the ground heat exchanger is limited.

The result of the study shows that the dual source concept is to prefer when comparing with a single source concept. In the case study plant, 17 boreholes is used but the technical performance would improve if more boreholes where added. If a few more boreholes where added, the annual electrical energy demand will decrease about 2.5 MWh per borehole and the annual additional heating energy demand in the range 2.0 – 11.5 MWh per borehole. Adding more boreholes also seems to be economical beneficial, the annual savings are about 3 700 SEK (518 USD) per borehole. The results also show that an increased size of ambient air heat exchanger is economical beneficial. However, if 100 000 SEK (14 000 USD) is invested, the annual savings are about 6 200 SEK (868 USD) if more boreholes are drilled and 3 200 SEK (448 USD) with a larger ambient air heat exchanger. The heat pumps seem to cover about 78 % – 95 % of the annual heating energy demand and in this case, investment in a larger heat pump is not an economical option. The actual figures are highly dependent on the climate (annual mean ambient temperature) and the number of operational years of the plant.

The control strategy used does not influence the performance to a large extent. The most economical strategy seems to be the one without any recharging of the ground. However, the difference between the options studied is rather small.

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