GROUND-SOURCE HEAT PUMPS - RECHARGING OF BOREHOLES BY EXHAUST-AIR COILS

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ABSTRACT

Vertical boreholes in dry bedrock are commonly used as heat source in Swedish heat pump installations. Currently used design depths result in minimum brine temperatures in the range -3 to -5 °C. This temperature can be raised by means of a deeper borehole or recharging from an alternative heat source (e.g. exhaust-air or a solar system). The paper discusses a combination of borehole/exhaust-air heat-recovery including an optional supply-air coil to achieve summer recharging/'free' air-conditioning and high-efficiency heat recovery ventilation.

In situ measurements confirm calculated brine temperatures and improved performance. Furthermore, results underline the importance of minimizing power input to pumps and fans that are used in the distribution systems. With heat-recovery, the brine temperature never goes below zero and hence defrosting of the exhaust-air coil will not be necessary. As a consequence of the raised brine temperature, a smaller amount of anti-freeze suffices and this will both improve heat transfer and decrease electric input to the brine pump.

1 INTRODUCTION

Electric heating is still one of the most frequent types of Swedish residential heating system. To improve the energy efficiency of such systems, installation of a heat pump is a popular alternative. It is, however, even more common to replace oil and wood-boilers with heat pumps. These heat pumps commonly use boreholes in dry bedrock as the heat source and the winner of the 1996 Nordic heat pump competition had this as the first alternative (Fahlén 1997). As a result of the popularity, there were in the year 2000 more than 350 000 heat pumps in Sweden and of these more than 120 000 were GSHP-systems.

1.1 Sizing and installation

For reasons of economy as well as operation, even GSHPs are rarely designed to cover maximum heat demand. A rule of thumb says that a capacity of 50 % of the building design load will cover 80-90 % of the annual demand for space heating (typical outdoor design temperatures vary between -20 to -30 °C).

For bivalent installations with long operating times, sizing of the borehole depth should be based on energy extraction. In case of monovalent systems, maximum refrigeration capacity must also be considered. Currently used depths, with a design annual heat extraction of around 150 kWh/m, result in minimum brine inlet temperatures ranging from 0 to -5 °C. The high value applies to rock with substantial groundwater flow and the low value to rock of poor thermal conductivity and/or a higher than expected energy extraction. In the heat pump competition the minimum value was -3 °C, calculated as well as measured. The performance of a heat pump system typically improves by 2-4 % per K increase in evaporating temperature and hence there is an incentive to raise this temperature.

When retrofitting homes heated by electric heaters noise is an important consideration since these houses are not equipped with boiler rooms. Therefore rotary piston compressors have been popular. However, subsequent to the ban on HCFCs there was a dramatic increase in the number of failures of this type of compressor. In particular, compressors operating with high pressure-ratios suffered from insufficient lubrication. Hence there is an incentive to raise the evaporating temperature and lower the condensing temperature also for reasons of reliability.

1.2 Recharging

One method of raising the evaporating temperature, applicable for existing boreholes as well as new installations, is recharging the borehole to compensate for the extracted energy and induced long-term temperature drop. Use of solar-collectors has been popular and this can indeed restore the borehole temperature to its original condition during summer. It cannot, however, improve the situation in winter over the conditions of the first operational year of the heat pump. Losses from individual boreholes of residential systems are too large to permit seasonal storage of energy. Another alternative is to use the energy of ventilation air. Most Swedish houses have mechanical ventilation and exhaust-air is a high-temperature heat source available all year and is also frequently utilized by means of exhaust-air heat pumps or HRVs (heat recovery ventilators).

2 METHOD AND APPLICATION

To investigate the possibilities of recharging by means of an exhaust-air coil, a house used as reference during the heat pump competition was equipped with a brine-cooled exhaust-air coil.

2.1 Building data

The house was built in 1977 and is a typical, mass-produced, single-family house with direct-acting electric heating, electric storage water heater, mechanical exhaust-air ventilation (0.5 ACH ≈ 165 m^3/h), and a heated space of 140 m^2 plus a heated storage facility of 11 m^2. Annual mean outdoor temperature is 5.8 °C and total use of electric energy was 25 MWh. These data correspond closely to a Swedish standard house for calculation of energy upgrades. The theoretical standard house has a mean outdoor temperature of +6 °C and an energy input of 25 MWh (5 MWh household electricity, 4 MWh hot water and 16 MWh space heating).

2.2 Heat recovery air-coil

Calculations showed that recharging the borehole could improve on the original 9 MWh savings by a further 1 MWh and also improve the life-expectancy of the heat pump in the process. In April 2000 the house was retrofitted with a heat recovery air-coil (figure 1) with the following design data (the brine consists of a 30 % ethanol-water mixture):

$\dot{V}_a = 150\ \mathrm{m^3/h}$, $t_{aci} = +20$ °C, $t_{aco} = +0.1$ °C, $\varphi_{aci} = 0.45$, $\Delta p_{ac} = 0$ Pa, $\Delta \dot{W}_f = 0$ W

$\dot{V}_b =$ 0.54 m^3/h, t_{bci} = 0 °C, t_{bco} = +2.2 °C, Δp_{bc} = 15 kPa, $\Delta\dot{W}_{bp}$ = 2.25 W
$\dot{Q}_{aS}$ = 1.0 kW, $\dot{Q}_{aL}$ = 0.3 kW.

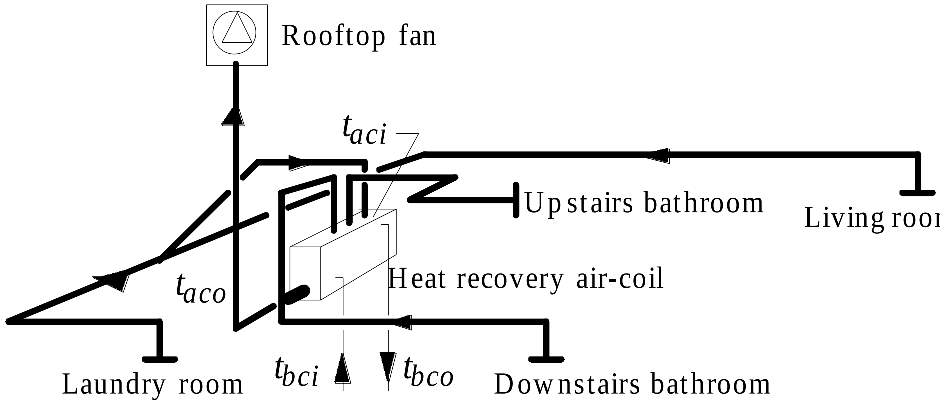
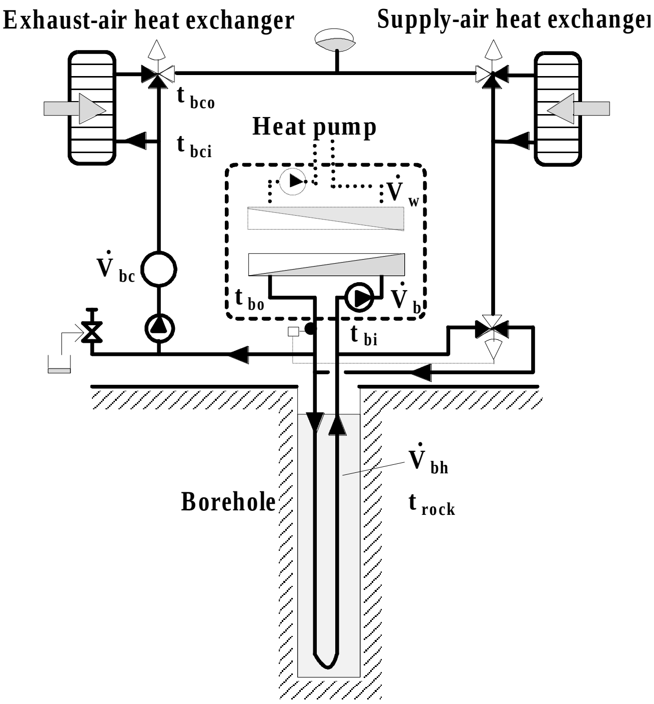


Figure 1. Connection of mechanical ventilation to a heat recovery coil.

Calculated coil data are uncertain since the air inlet velocity is only 0.17 m/s to keep the airside pressure drop low. The manufacturer's coil model has not been validated to such low values and natural convection will also influence results. Therefore the coil was arranged with the air flowing downwards to facilitate condensate drainage and use the natural convection effects. The pressure drop is a penalty paid 8760 h per year since the fan operates continuously and must therefore be kept low. The increases in fan and pump power above have to be divided by the respective efficiencies, typically 5-10 %, to see the effects on drive power.

2.3 Operational strategy



The projected installation has three operational modes, implemented using the two switching valves at the coils:

Recharging with brine being warmed by the exhaust coil, bypassing the supply coil and warming the collector in the borehole.

Air-conditioning with cold brine bypassing the exhaust coil, cooling the supply coil and warming the collector in the borehole.

Heat recovery with brine being warmed by the exhaust coil, then warming the supply coil and finally being pre-heated by the borehole collector.

Figure 2. Brine circuit with borehole, exhaust-air coil and future supply-air coil.

To avoid unnecessary operation, the recharging pump only starts when the heat pump is about to operate. The heat pump controller has a time-relay which delays the compressor start by 15 min (adjustable) after detection of a heat demand. The controller has been modified to provide also an output without delay to start the recharging pump. By the time the heat pump starts the borehole has already been warmed. The pump will continue to run after the heat pump has stopped until a thermostat senses that the brine temperature has been restored to its undisturbed value. Further measurements and analysis will show whether it pays to operate the pump to excess temperatures but this would only be possible during periods with little heat demand. The charging pump has been connected via a switching valve and a stop valve to disconnect the heat pump when charging the borehole during heat pump off-periods. During on-periods, the charging pump works only on the return pipe to the borehole. In this way, the charging circuit will not affect the flowrate through the heat pump and will return the lowest temperature to the coil (see figure 2).

2.4 Measuring program

In March 1996 this house and four neighbouring houses were equipped with heat pumps to check the theoretical and laboratory results used in the evaluation of the heat pump competition. Fahlén (1997) describes this installation, which so far only covers space heating but with a future possibility to include hot water. The installation originally included an electromagnetic flowmeter to measure heating water flowrate ($\dot{V}_w$), PRTs to measure heating water inlet and outlet temperatures (t_{wi} and t_{wo}), brine inlet and outlet temperatures (t_{bi} and t_{bo}) plus room-air temperature (t_{room}) and electric energy meters for the heat pump ($\dot{W}_{ehp}$) and the total input to the house ($\dot{W}_{etot}$). When the heat recovery air-coil was installed, measuring equipment was added to monitor also the brine flowrate to the exhaust-air coil ($\dot{V}_{bc}$), the brine inlet and outlet temperatures at the coil (t_{bci} and t_{bco}), the exhaust-air inlet and outlet temperatures at the coil (t_{aci} and t_{aco}) and operational time meters and start counters for the recharging pump as well as the heat pump.

3 RESULTS

Results consist partly of comparative calculations and partly of comparative measurements.

3.1 Possible savings by means of recharging

The calculated savings on space heating were 9 MWh using data from laboratory tests at SP and the energy calculation program Enorm (Fahlén 1991, Munther 1991). Calculations were verified by monitoring before and after installation of the heat pump (Fahlén 1997). The borehole was modelled according to theories described by Claesson (1985). Using data from this installation, the examples below illustrate the effects of various measures to improve on the temperature conditions in the borehole. The temperature t_{Rm} is the mean peripheral borehole temperature (at radius R). It is given as the long-term steady-state mean value and as minimum temperature after operation at maximum capacity for 30, 60 and 90 days respectively (a typical design capacity will result in operation at close to maximum capacity for a large part of the heating season).

Annual heat extraction: Q_{bh} = 9000 kWh, $\dot{Q}_{bhm}$ = 1.03 kW.
Borehole data: D_{bh} = 0.115 m, H_{bh} = 60 m, λ_{rock} = 3.5 W/m/K.
To illustrate the influence of λ_{rock}, a change from 3.5 to 3.0 will cause t_{Rm} to drop from 1.7 to 0.9 °C in ex. 1 below.

Ex. 1, no recharging:
Long-term mean: t_{Rm} = 1.7 °C
Short-term min. : t_{Rm} = -2.8 °C (30 days), t_{Rm} = -3.2 °C (60 days), t_{Rm} = -3.4 °C (90 days).
Ex. 2, solar recharging of all energy extracted:
Long-term mean: t_{Rm} = 6.5 °C
Short-term min. : t_{Rm} = -1.0 °C (30 days), t_{Rm} = -1.7 °C (60 days), t_{Rm} = -2.0 °C (90 days).
Ex. 3, exhaust-air coil recharging of half the energy extracted:
Long-term mean: t_{Rm} = 4.0 °C
Short-term min. : t_{Rm} = 1.9 °C (30 days), t_{Rm} = 1.7 °C (60 days), t_{Rm} = 1.5 °C (90 days).
Ex. 4, exhaust-air coil recharging of all the energy extracted:
Long-term mean: t_{Rm} = 6.5 °C
Short-term min. : t_{Rm} = 2.8 °C (30 days), t_{Rm} = 2.4 °C (60 days), t_{Rm} = 2.2 °C (90 days).
Ex. 5, increasing borehole depth from 60 to 110 m:
Long-term mean: t_{Rm} = 4.2 °C
Short-term min. : t_{Rm} = 1.6 °C (30 days), t_{Rm} = 1.4 °C (60 days), t_{Rm} = 1.2 °C (90 days).

3.2 Factors which affect the possible savings

Recharging affects total COP_1 positively by increasing the evaporating temperature T_2 and hence reducing compressor electric motor work ($\dot{W}_{em}$) for a given heat extraction. However, increased capacity raises the condensing temperature and additional pump and fan work ($\Delta\dot{W}_{ec}$) to the coil has a negative effect according to:

$$\Delta COP_1 = \frac{\partial COP_1}{\partial \dot{W}_e}\cdot\Delta\dot{W}_{em} + \frac{\partial COP_1}{\partial \dot{W}_e}\cdot\Delta\dot{W}_{ec} = \frac{\partial COP_1}{\partial T_1}\cdot\Delta T_1 + \frac{\partial COP_1}{\partial T_2}\cdot\Delta T_2 + \frac{\partial COP_1}{\partial \dot{W}_e}\cdot\Delta\dot{W}_{ec} \quad \text{(eq. 1)}$$

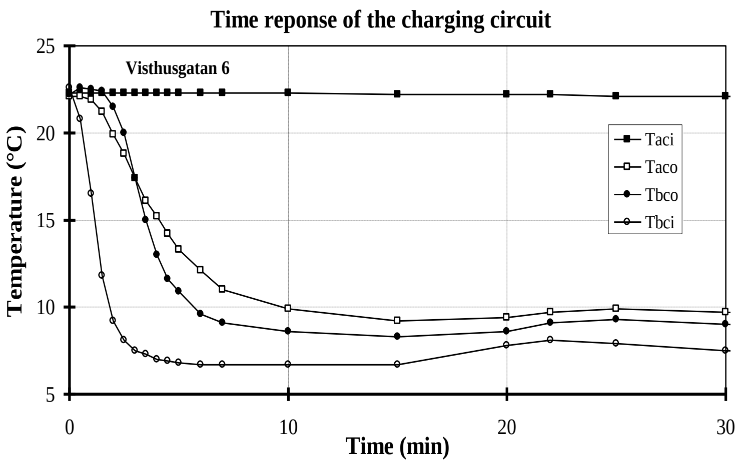
Assuming that the compressor efficiency is constant within modest changes in pressure ratio, the temperature dependence of the actual COP_1 will be close to that given by the theoretical Carnot efficiency and changes in evaporating and condensing temperatures will follow changes in brine and heating water temperatures. Thus the following relation indicates how the recharging work affects the relative change in COP_1 and drive power for a given improvement in brine temperature:

$$\frac{\Delta COP_1}{COP_1} = -\frac{\Delta\dot{W}_{ehp}}{\dot{W}_{ehp}} = \left[-\frac{T_2}{T_1}\cdot\frac{\Delta T_w}{T_1 - T_2} + \frac{\Delta T_b}{T_1 - T_2} - \frac{\dot{W}_{ec}}{\dot{W}_{ehp}}\right] \quad \text{(eq. 2)}$$

Comparing examples 1 and 3 above, then ΔT_b = 4.9 K and for this installation T_1 - $T_2 \approx$ 45 K, $\dot{W}_{ehp} \approx$ 1.1 kW and $\dot{W}_{ec} \approx$ 0.06 kW. $\frac{\Delta COP}{COP} = \left[\frac{4.9}{45} - \frac{0.06}{1.1}\right] = 0.109 - 0.055 = 0.054$, i.e. a net improvement by 5.4 %. This presumes a suitable increase in radiator capacity to keep ΔT_w = 0. The theoretical gross improvement, 10.9 %, is very close to the value measured with a heating water temperature of t_{wo} = 35 °C. However, at t_{wo} = 50 °C the real change for this particular heat pump is much more significant, ≈ 30 %, and this would yield a net improvement of 24 % from the recharging activity.

3.3 Measured temperatures of the heat recovery coil

The exhaust-air flowrate was measured and adjusted before starting the measurements. The brine flowrate is constant and is registered as a weekly mean value. Measured brine temperatures without recharging (Fahlén 1997) agreed well with calculated values (maximum capacity is used approximately one month). Also the measured temperatures with recharging agree with projected values. The efficiency of the heat-recovery coil, however, is not quite as good as calculated by the manufacturer (c.f. figure 3).



The time lag due to the collector volume/flowrate is approximately 15 minutes, as indicated by the graph. The coil also has a thermal inertia of 44 kJ/K.

Measured coil-efficiency is not quite on a par with the calculated result. The pinch T_{aco} - T_{bci} = 1.8 K compared to 0.1 K as given by the manufacturer.

Figure 3. Transient response of the exhaust-air coil after a pump start.

One special effect of the chosen criteria for sizing the heat-recovery coil is that frosting of the coil can be altogether avoided. Hence you avoid the problem of controlling and actuating defrosts, which penalizes conventional exhaust-air heat pumps and heat-recovery ventilators. Figure 4 indicates that the coldest brine temperature, t_{bo}, is higher than $t_{outdoor}$ until the latter falls below 0 °C. At lower temperatures the dew point of the outdoor air will be below the brine temperature and you avoid moisture precipitation in the heat exchanger. You will, however, have condensation at temperatures above 0 °C and then you can benefit from the latent heat of condensation. In cases of large humidity inputs to the exhaust-air, e.g. when you are using a drying cabinet or taking a shower, the heating capacity from the exhaust-air increases to a level when the surface temperature in the coil exceeds 0 °C even though the brine temperature may be several K below zero (t_{bo}, which goes to the heat-recovery coil, is 3-4 K lower than the brine temperature from the borehole to the heat pump, t_{bi}).

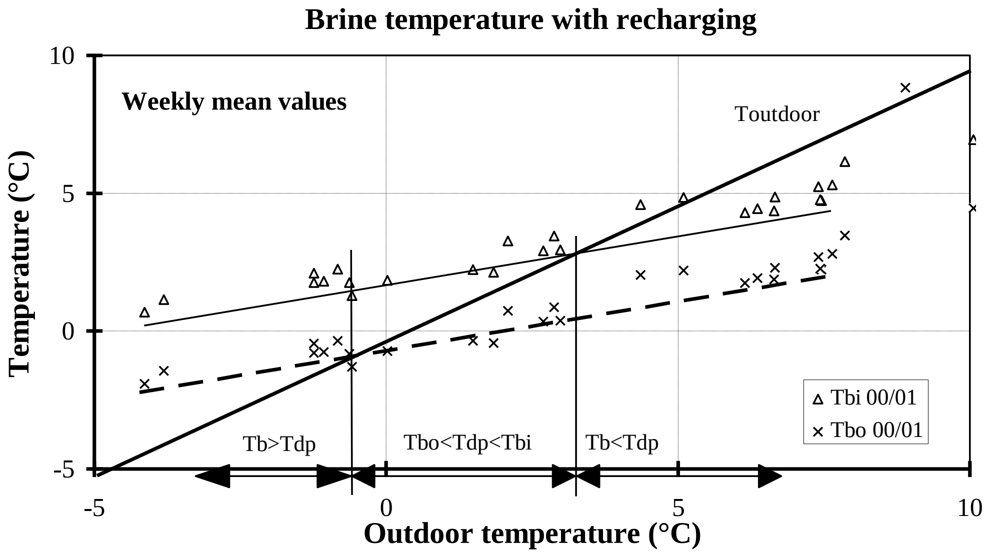


Figure 4. Comparison between the lowest brine temperature and the outdoor air-temperature.

3.4 Measured temperatures of the borehole

Measured brine temperatures without recharging (Fahlén 1997) agree with calculated values (maximum capacity is used for 1-2 months). Figure 5 compares the inlet (t_{bi}) and outlet (t_{bo}) brine temperatures of the heat pump before and after installation of the recharging coil. The diagram indicates that t_{bi} is 3-4 K higher with recharging at a given outdoor air temperature.

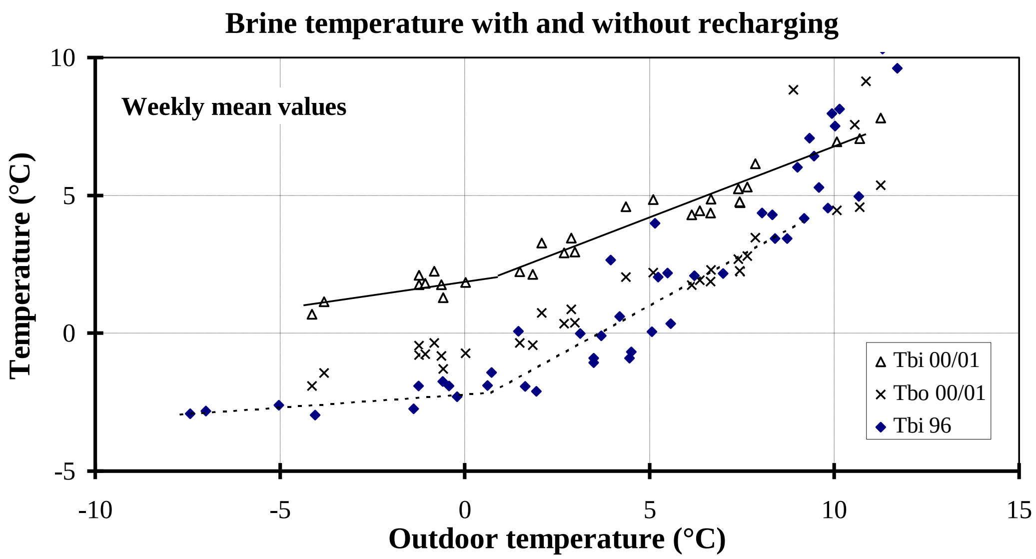


Figure 5. Brine temperature as a function of the outdoor temperature (without recharging in 1996 and with recharging during the winter 2000/2001).

3.5 Measured coefficients of performance

The influence of parasitic drive powers such as pump and fan motors was previously mentioned in connection with equations 1 and 2. Figure 6 once again illustrates the importance of taking even small drive powers into consideration. Measurements show that COP_{hp} increases from 3.2 up to nearly 4 when the outdoor temperature increases from -4 to +12 °C. We expect *COP* to increase because the brine temperature increases and the <u>mean</u> heating water temperature decreases with increasing outdoor temperature. If instead we look at COP_{hps}, which includes the brine and water pumps, then the increase is very weak. This is caused by continuous operation of the heating water pump while the heat pump runs intermittently and hence the fractional power used by the water pump increases. If you take one step further and include the recharging pump (COP_{hps*}) and the fan (COP_{hpsf}) of the fan-coil unit of the heating system, then COP_{hpsf} is almost constant and even decreases at high outdoor temperatures. At +10 °C COP_{hp} = 3.8 while COP_{hpsf} = 2.4!

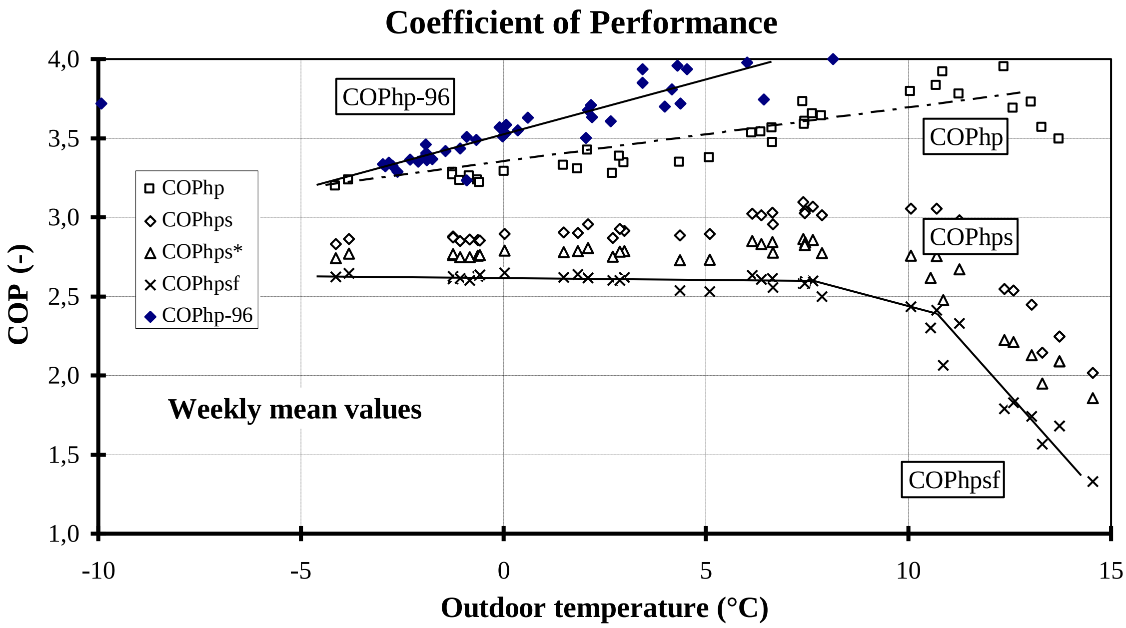


Figure 6. Coefficient of Performance as a function of the outdoor air-temperature with recharging during the winter 2000/2001 (as a comparison the diagram includes COP_{hp} without recharging during 1996).

Figure 6 indicates also that parasitic drive powers are not the only factors, which affect results negatively. A comparison between COP_{hp} with or without (-96) recharging shows that COP_{hp} is actually lower during most of the year with recharging even though the brine temperature is approximately 4 K higher during all the time. This is partly caused by the recharging pump but is mainly due to a large increase in the capacity of the heat pump. The resulting high condensing temperatures and short operating times yield a larger negative influence than the positive effect of higher evaporating temperatures. Hence recharging must be accompanied by larger heat transfer surfaces in the heating system and/or a buffer tank.

3.6 Operational time and start frequency

As a consequence of the raised brine temperature the capacity of the heat pump will increase. Figure 7 shows that the capacity increase due to recharging is sufficient to lower the balance point, i.e. the lowest outdoor temperature without need for supplementary heating, from -3 to -6 °C. This corresponds to a capacity increase of 17 % and is consistent with results from laboratory tests on capacity versus brine temperature on this type of heat pump.

As previously mentioned, the condensing temperature will increase unless the heat transfer capacity of the heating system is improved. The raised capacity also causes operating times to become shorter and the starting frequency to become higher with a maximum number of starts per 24 h of around 70 (3 starts per hour, see figure 8). The high starting frequency is a result of economic considerations in retrofitting electrically heated houses. Direct-acting electric heaters are retained as supplementary heating and the heat pump is connected to a ‘mini’ hydronic heating system consisting of one fan-coil unit (time constant around 5 min) and one radiator (time constant around 10 min). The thermal mass of the system is small and at times with a low heating demand in relation to the capacity of the heat pump, the temperature rises quickly in the heating system. This, of course, can be mitigated by means of a buffer tank but then you loose part of the economic advantage of a ‘mini’ hydronic system. Possibly a more interesting future alternative is the use of speed-controlled compressors, pumps and fans. Karlsson and Fahlén (2001) provide an overview of the technical development in this field and also render diagrams illustrating how this may affect the starting frequency (c.f. figure 8). Figure 9 illustrates the short operating times as a consequence of the high starting frequency (for this particular installation there is a minimum starting delay of 15 minutes). High starting frequencies and short operating times are neither good for energy efficiency, nor for reliability.

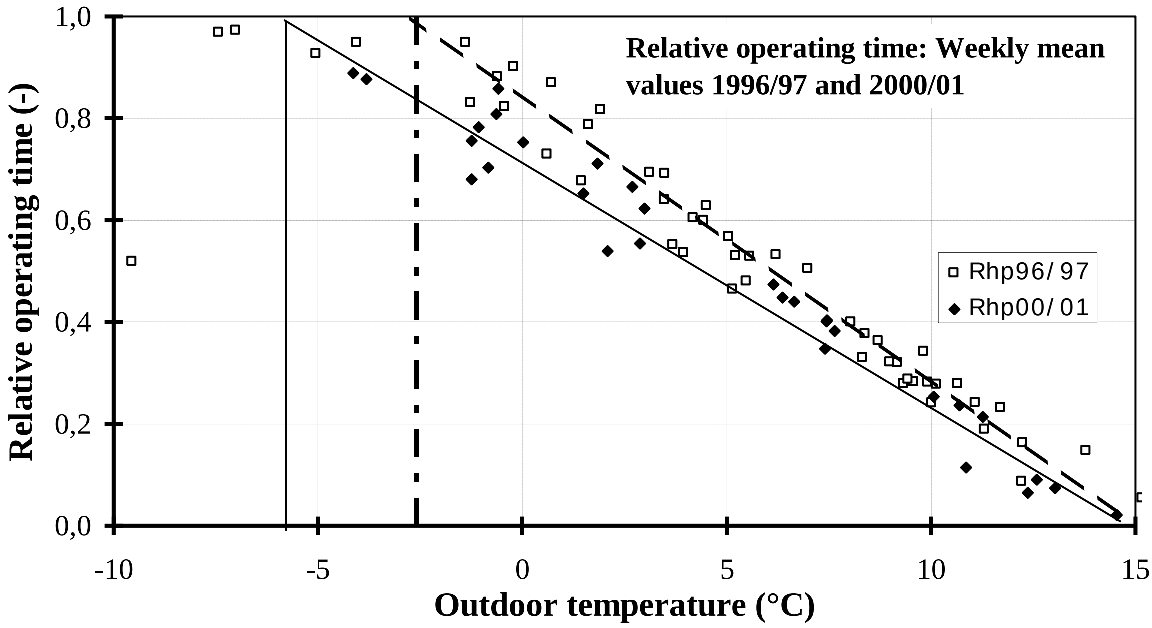


Figure 7. Relative operating time as a function of the outdoor air-temperature (without recharging in 1996 and with recharging during the winter 2000/2001).

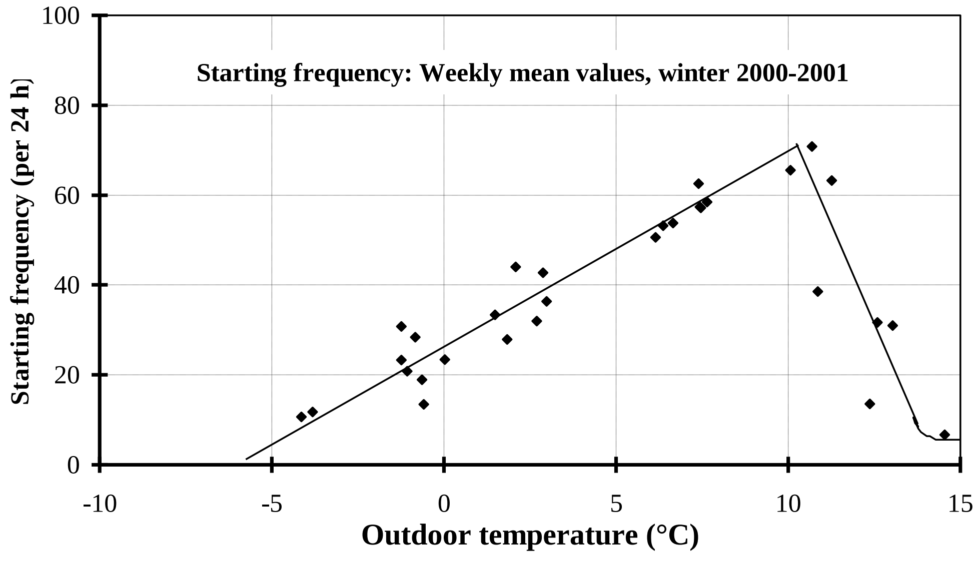


Figure 8. Starting frequency as a function of the outdoor air-temperature (with recharging).

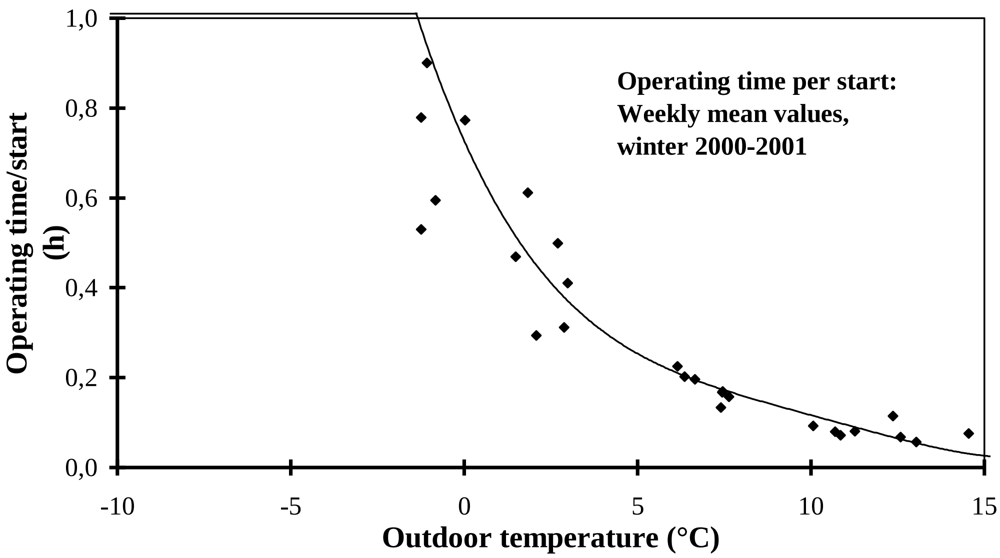


Figure 9. Operating time as a function of the outdoor air-temperature (with recharging).

4 DISCUSSION

The temperature in vertical boreholes used as heat source for GSHPs will slowly drop with time, the more so the more energy is extracted. This can be mitigated either by a deeper borehole (in a new installation) or a system to replenish the energy extracted from the hole (in both new and existing installations). Raising the brine temperature from -5 °C to 0 °C may improve the *COP* by 10-50 % depending on the type of heat pump. Normally, a deeper borehole is the most cost-effective solution but recharging is of great interest for existing, over-taxed holes, for sites with multiple holes drilled within 20 m of each other and in very cold regions.

Recharging with an exhaust-air coil has several advantages in relation to conventional systems. Since heat can be extracted even when there is no heat demand, the borehole can act as a diurnal heat store. The penetration depth of a temperature wave into the rock is only about 0.2 m over a 24 h period and hence heat losses will be miniscule when used as a short-term store. In comparison with a traditional exhaust-air heat pump, this load levelling will eliminate the need for defrosting which results in a very simple recovery system. The brine temperature will be sufficiently low to utilize both sensible and latent heat in the exhaust-air but not low enough for frosting. It should be noted that long operating hours make it important to ascertain low pressure drops on both the air and the brine sides to avoid excessive drive energy inputs. To save on drive energy, it is also desirable to develop more efficient small pumps and fans than those currently available on the market. At present, theoretical improvement may be halved by the pump work.

Increasing the depth of the borehole seems to be the most cost-effective way to improve the efficiency of a GSHP. At a cost of 250-300 SEK/m ($28-$33/m) the added cost of achieving an improvement similar to that of the described recharging system would be around 14000 SEK ($1550) and it is doubtful whether the recharging system could be installed at this cost. Components, including piping, couplings etc., add up to around 10000 SEK but installation work probably adds another 10000 SEK to the bill. In particular, insulation of the cold brine pipes is time consuming and costly but very important for the long-time functionality of the system. In a new house, however, the installation cost could probably be halved. The most interesting application of the exhaust-air coil would be in combination with a supply-air coil to achieve a highly efficient HRV system while at the same time providing the possibility of ”free” air-conditioning and improved operating conditions for the heat pump. Direct heat recovery will provide greater savings than the effect of recharging.

Recharging by means of a solar system has much less effect than either the exhaust-air system or extra drilling. The results used as examples above only look at the situation at the borehole periphery. Including the thermal resistances of the brine pipe-wall and water in the borehole further favours the exhaust-air solution. In this case heat is added directly to the brine when the heat pump is operating whereas with a solar system heat has to be transferred across all heat resistances twice, thus adding a significant temperature drop which will detract from the improvement indicated at the periphery of the borehole.

5 CONCLUSIONS

A vertical borehole used as heat source for a GSHP can be recharged by means of an exhaust-air heat-recovery coil and thus increase the brine temperature by 3-4 K. This raises the capacity of the heat pump and may also raise the *COP*. To raise the *COP*, however, the heating system must be complemented with more heat transfer area and/or a buffer tank. A buffer tank is recommended to avoid high starting frequencies in ‘mini’ hydronic heating systems used to retrofit houses with direct-acting electric heaters. Pressure drops in the recharging coil as well as pump and fan motor efficiencies must also be carefully considered to actually improve on the overall *COP*. Adding a supply-air coil may provide ‘free’ cooling and gives an opportunity for a highly efficient heat-recovery system with no need for defrosting. The exhaust-air coil is a better way of recharging a borehole than is the solar system alternative. Most cost-effective, however, is choosing a deeper borehole from the very beginning.

6 ACKNOWLEDGEMENTS

I would like to thank the Swedish Energy Administration, Eskilstuna, and SP (Swedish National Testing & Research Institute), Boras, Sweden for jointly funding this work.

7 NOMENCLATURE

ACH	Air Changes per Hour	GSHP	Ground-source heat pump
HRV	Heat Recovery Ventilator	PRT	Platinum Resistance Thermometer

Designation of variables

COP	Coefficient of Performance	(-)
N	Air Changes per Hour (ACH)	(h^{-1})
p	pressure	(Pa)
Q	heat (thermal energy)	(J)
$\dot{Q}$	capacity (thermal power)	(W)
t	temperature (Celsius)	(°C)
T	temperature (thermodynamic)	(K)
V	volume	(m^3)
$\dot{V}$	volume flow	(m^3/s)
W	work (mechanical or electric)	(J)
$\dot{W}$	power (mechanical or electric)	(W)
η	efficiency	(-)
φ	relative vapour pressure	(-)
λ	thermal conductivity	(W/m/K)
ρ	density	(kg/m^3)
τ	time	(h.min.s)

Subscripts

1	condenser/heating
2	evaporator/cooling
a	air
b	brine
c	coil
dp	dew point
e	electric
f	fan
hp	heat pump
hps	heat pump system
hpsf	hps plus fan-coil heater
i	inlet
L	latent
m	mean, motor
o	outlet
outdoor	outdoor
R	radius
p	pump
S	sensible
tot	total

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