HEAT PUMP FOR HEATING AND HOT WATER - EXPERIENCE FROM AND IMPROVEMENT OF A RETROFIT, GROUND-COUPLED INSTALLATION

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Abstract: Sweden aims to reduce building-related energy use by 50 % until 2050. The main savings have to be in existing buildings and in this respect heat pumps have a strong position in Sweden. This paper describes a retrofit solution that can be used also in newer, heat efficient single-family houses. The presented heat pump system design was first evaluated by simulations based on laboratory-tested capacity data for the heat pump. Simulations were made for various levels of upgrades of a standard installation. The heat pump was installed in a single-family house with direct-acting electric heating and the various upgrades were made one after the other. Measured values agreed well with the predicted results and, compared to a standard installation, the new system raised SPF from 2.7 for heating-only to 3.7 for heating and hot water. Specific energy use for heating and hot water in this house from 1977 is now on a par with future requirements in Sweden for new housing with electric heating (30 kWh/m^2/year) and less than that of current passive houses.

Key Words: heat pump, retrofit, ground-source, heating, hot water, heat recovery, control, load matching, storage, parasitic powers

1 INTRODUCTION

Sweden has set a target for 2050 to reduce energy use in the built environment by 50 %. The challenge is not to build new low-energy houses but to radically reduce the energy use in existing houses.

The dominating building category for space heating and hot water in Sweden is the single-family house. Figure 1 illustrates a future scenario for reduction of purchased energy as proposed by IVA (Royal Swedish Academy of Engineering Sciences). Note that a large share of Swedish houses use electricity for heating and hot water.

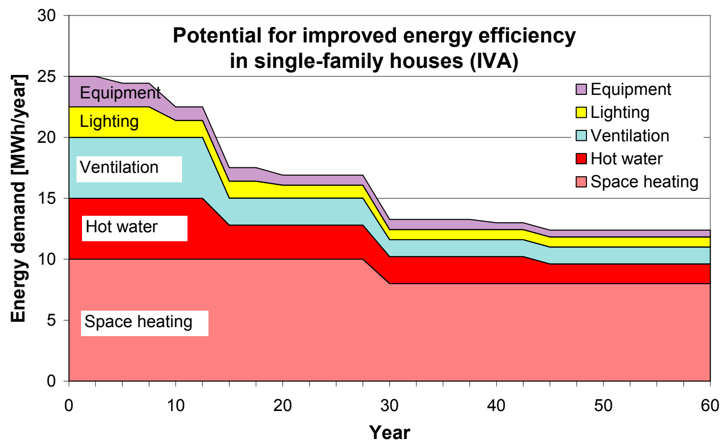


Figure 1: Proposed future target values for purchased energy in Swedish single-family houses.

To find ways of addressing this building category a number of technical procurement competitions were staged in the 1990s. Based on past experience at the time and screening of potential alternatives, retrofitting of heat pumps was identified as one of the most cost-effective alternatives. Hence a heat pump competition[3] was staged in 1995 and this paper presents experience from field tests of the winning heat pump and a number of improvements made to the original concept.

2 TEST HOUSE

Field tests were carried out in a single-family, detached house built in 1977. The house is a standard, mass produced 1½ story building located in Boras in southern Sweden. This location has an annual mean temperature of +6 °C and a design outdoor temperature for heating of DOTH = -19 °C. The test house was also the reference house of the Nordic heat pump competition in 1995. Paragraphs 2.1 and 2.2 provide the basic building specifications and historic energy use respectively.

2.1 Building specifications

Figure 2 below provides some basic information on the test house.

Specifications
Location: Boras, Sweden. Annual mean temperature + 6 °C, DOTH = -19 °C.

Building year: 1977

Building: Timber-framed, single-family. U_{wall} = 0.35 W/m^2/K, 2-pane windows.

Heated area: 140+10 m^2

Ventilation: Mechanical exhaust, CAV = 165 m^3/h (0.5 ACH).

Heating: Direct-acting electric. Heat pump retrofit in March 1996 (see paragraph 3).

Hot water: Electric storage heater, 300 L. Replaced by new TES in 2004 (see 3).

Figure 2: The field-test house in Boras, Sweden.

2.2 Historic energy use

Figure 3 provides the reference level of purchased energy prior to the installation of the heat pump in 1996. Only the total was actually measured, the split between heating and household electricity has been estimated based on the energy use outside the heating season.

The simulation program Enorm [11] provides an estimated value of 25 MWh per year.

The Swedish target level for 2050 is indicated by the dashed green line.

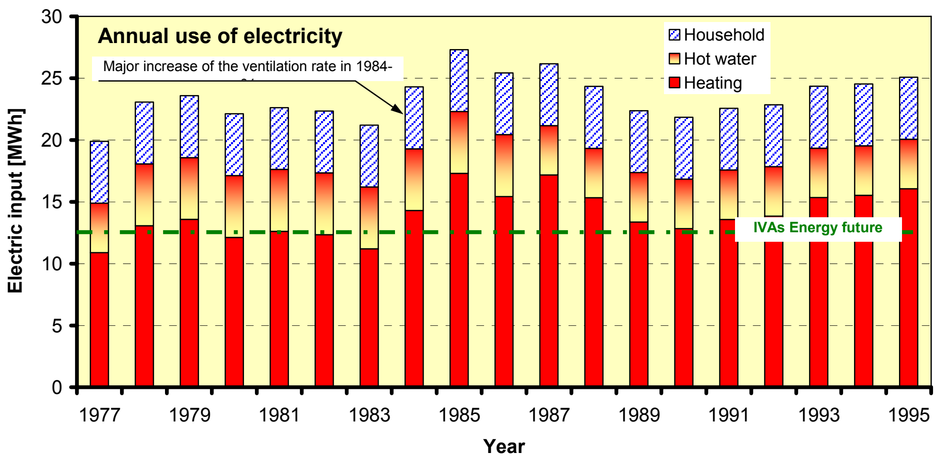


Figure 3: Total purchased energy for heating, hot water and household electricity 1977-95.

3 HEAT PUMP SYSTEM

In 1995 the test house was used as the model house for the Nordic heat pump competition[3]. The competition was targeted towards new construction or retrofitting of electrically heated houses. In both cases heating demand is fairly low and hence the specifications prioritized compact, low-cost units (design heating demand is 6 kW at DOTH = -19 °C). The test house was retrofitted with the winning low-cost ground-source heat pump system in 1996 (around 4600 € for heat pump, borehole, one fan-coil unit, one radiator and installation).

The original system was monitored for a number of years to verify the simulations that were made to evaluate the competition. Annual savings and temperatures of the heat source (brine temperature from the borehole) and the heat sink (hydronic fan coil) agreed well with measured results. Prior to the installation the following upgrades were planned and subsequently introduced and evaluated (c.f. Figure 7):

- Recharging of the borehole from an exhaust-air heat recovery coil.
- Storage tank for heating and hot water, new control system.
- Addition of 4 more radiators.

The order of the modifications differs slightly in practice, c.f. Figure 11, from the planned actions of Figure 7.

3.1 Heat pump

The original heat pump has CSD compressor and CSD pumps. Some characteristic data: *Dimensions*: 0.6x0.6x0.6 m, heat pump system performance at B0/W35 (EN255[2]): *heating capacity* 4.09 kW, total *drive power input* 1.16 kW and COP_{hps} = 3.51 (see also Figure 8). The small dimensions make it an easy retrofit in houses that have electric heating and hence lack an equipment room.

3.2 Heat source

Heat source of the original heat pump system was a vertical borehole of 60 m active depth.

Design value for borehole
Design value: 50 W/m active borehole
Heat pump cooling capacity: 3 kW
→ 3000 W/(50 W/m) = 60 m borehole.

Operating modes for additional system
Three modes of operation: Recharging, air conditioning and heat recovery

Recharging:
- exhaust coil ON
- supply coil OFF

Air conditioning:
- exhaust coil OFF
- supply coil ON

Heat recovery:
- exhaust coil ON
- supply coil ON

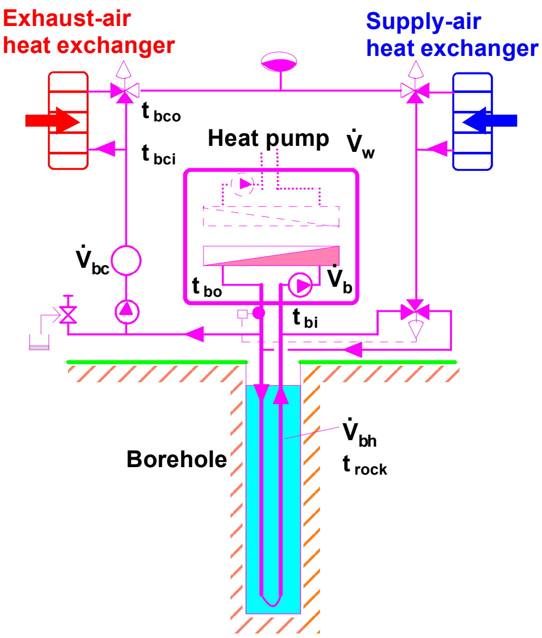


Figure 4: Heat source with vertical borehole plus the additional recharging system.

3.2.1 Borehole
The short borehole was the outcome of standard design values at the time and the low-cost requirement of the competition. As part of a research plan, the piping of the original installation was prepared for the addition of a recharging system. This is illustrated in Figure 4.

3.2.2 Recharging
Although solar recharging of boreholes is regularly proposed in Sweden, it was shown experimentally and theoretically already in the early 1980s that this is not a viable solution for single boreholes (systems of boreholes, however, is another cup of tea). Fahlén[4] has shown that recharging by means of heat recovery from ventilation air is a much better proposition. This solution provides heat when it is actually needed, transforms the system during much of the year to a very efficient exhaust-air heat pump with high recovery potential and no need for defrosting. When it is cold outside, the brine is preheated by the borehole before it enters the exhaust-air coil and in winter borehole temperature will always be higher than the dew-point of outdoor air. Also, heat recovery will be continuous, even when the heat pump is not in operation (in this case the borehole acts as a short-term storage with no losses).

3.2.3 Air conditioning
The concept indicated by Figure 4 can also be integrated with a supply-air heat exchanger to provide summer-time air conditioning and winter-time preheating of the supply air. Air conditioning not only provides "free" cooling (only pump energy is required) but also aids in recharging of the borehole for the next heating season.

3.2.4 Heat recovery
Fahlén[4, 8, 12] and Zhous have analyzed also the possibility of using the system in Figure 4 to provide a run-around-coil heat recovery system. This will provide high temperature-efficiency with no defrost requirement and no risk for carry-over contamination of the supply air.

3.3 Heating system

The original heating system consisted of direct-acting electric heaters in each room. Upon installation, a hydronic fan coil was installed on the bottom floor (part of the fixed price from the competition package; c.f. Figure 5). As part of the research plan, future addition of hydronic room heaters was prepared when the heat pump was installed.

3.4 Hot water system

The test house was equipped with an electric water heater (volume 300 L, power input 1.5/3 kW). This was retained during the first evaluation period but was later replaced by a specially designed[6], integrated heating and hot water TES (c.f. Figure 6). The TES stores heat for heating in the lower part and heat for hot water in the upper part, all in passive water. Hot water is heated on demand by a plate heat exchanger (HEX1). This provides a hot water system with no requirement of corrosion protection of the tank, no problems of limestone build-up on heating surfaces and little risk of legionella (no storage of fresh, sanitary water). It also reduces the pressure level of the tank from around 1 MPa (SHW pressure) to 0.1-0.2 MPa (heating system pressure). Finally, the tank provides a good place for deaeration of the heating water system and mitigates the common acoustic problem of many current Swedish switching designs for heating and hot water (often "anti-click" tanks are provided as retrofits). The design provides extremely good stratification (see 5 Results).

3.5 Control system

The control system of the original heat pump unit was quite simple. This was subsequently upgraded to improve room temperature control, to add a novel hot water heating system and to reduce parasitic drive energy to pumps.

3.5.1 Original control system

Figure 5 illustrates the original control system. The lack of feedback control of room temperature made it necessary to manually modify the settings of the controller from time to time. Also, the lack of heating-water pump control reduced the heat pump system coefficient of performance (COP_{hps}) severely.

Original control of the heat pump operation relied on feed-forward control of the heating water temperature return water (inlet) temperature t_{wi} by means of the measured outdoor temperature t_{out}. The relation is non-linear and has to be adapted to the specific house, hence the moniker “curve control”. There was no feed-back control so the setting had to be adjusted slightly in spring and autumn respectively. The intrinsic heating water circulator, P1, was a CSD unit with continuous operation and no control. It was switched off manually when the heating season stopped.

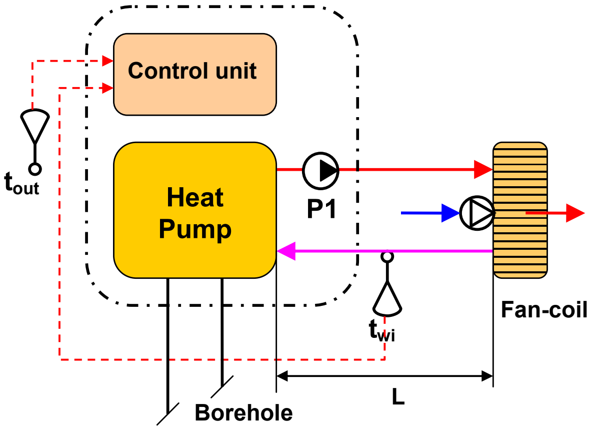


Figure 5: Original control system[5, 6] using a simple feed-forward curve control.

3.5.2 Modified control system

Fahlén[6, 10] has described the new control system in some detail. Figure 6 shows the basic principles.

P1 is the original, intrinsic heating-water circulator of the heat pump. P2 is a new, VSD circulator that has direct feedback COD by the measured mean room temperature t_{room} (e.g. 21 °C). P3, finally, is a new VSD pump that controls the outgoing hot water temperature t_{hw} to the desired value (e.g. 55 °C).

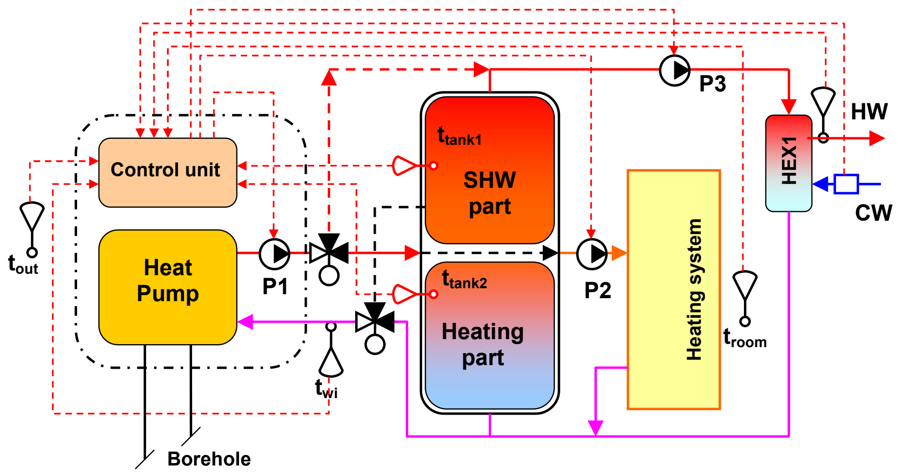


Figure 6: Modified[5, 6] control system that combines feed-forward and feedback control.

The original controller was retained to provide feed-forward control of the heating water temperature by means of the outdoor temperature t_{out} and the condenser water inlet temperature t_{w1}. These control sensors have been complemented by sensors t_{tank1}, t_{tank2}, t_{hw} and t_{room} for the hot water and heating part of the tank, the outlet hot water temperature and the mean room temperature (exhaust air temperature) respectively. The heat pump will start if either of t_{tank1} or t_{tank2} is below their set values (e.g. 55 and 30 °C respectively). If both indicate start, priority is given to t_{tank1}.

Tank control: Heating operation starts by $t_{\tan k2}$ and stops when $t_{w2} > t_{w2}(t_{out})$ as calculated by the curve control. This means that the heating-part tank will have a temperature that is always matched to the current outdoor temperature. In this mode, P1 operates with CSD at the optimal flow for the condenser. Each heating operation terminates by switching to a top-up hot water operation if $t_{\tan k1} < t_{\tan k1.\max}$. Hot water operation starts when $t_{\tan k1.\min} < t_{\tan k1.set}$ - $\Delta t_{\tan k1.set}$ and $t_{\tan k1.\max} > t_{\tan k1.set} + \Delta t_{\tan k1.set}$. P1 now switches to VSD and pump speed is modulated to maintain the supply temperature at $t_{w1} > t_{\tan k1.set}$. Below this value, pump speed has a minimum value which is increased in proportion to the difference $t_{w1} - t_{\tan k1.set}$ until maximum flow is reached.

Heating system control: The room temperature sensor starts the heating system circulator P2 when $t_{room} < t_{room.set}$. A PI-controller regulates the pump speed in relation to the control deviation. Currently there is only one heating system circulator controlled by the mean indoor temperature ($\bar{t}_{room} = t_{ex}$). When new types of very small circulators appear on the market, individual control of each room terminal unit becomes possible as described by Fahlén[7].

Hot tap water control: The flow sensor at the cold water inlet of HEX1 starts the circulator P3. The hot water PD-controller regulates pump speed to maintain the desired outlet temperature of the hot water ($t_{hw} = t_{hw.set}$). This makes it very simple to change the desired hot water temperature (the set value for the tank temperature should be automatically updated when $t_{hw.set}$ is changed to maintain a suitable temperature margin).

4 MODELLING AND SIMULATIONS

For the evaluation of the heat pump competition, results from laboratory tests were used to predict energy savings and temperature levels in the heat source and heat sink systems.

4.1 Building model

The building simulation program Enorm[11] was used to model the heating demands for transmission, ventilation and infiltration. The model considers the detailed construction of all building components, the location and orientation of the building and losses from ventilation ducts, hot water storage and other HVAC components. Experience from many years of electric heating and measured values of air tightness made it possible to verify the model and to assess values of interior heat generation from persons and equipment.

4.2 Heat pump model

The heat pump model, as described by Fahlén[10], was a black-box model based on laboratory measurements:

$$\dot{Q}_{hps}(t_{b1}, t_{w2}) = \dot{Q}_{35}(t_{b1}) + \left(\frac{t_{w2} - 35}{15}\right) \cdot \left(\dot{Q}_{50}(t_{b1}) - \dot{Q}_{35}(t_{b1})\right) \tag{1}$$

with $\dot{Q}_{35}(t_{b1}) = \dot{Q}_{35.0} + \dot{Q}_{35.1} \cdot t_{b1} + \dot{Q}_{35.2} \cdot t_{b1}^{\;2}$ and similarly for $\dot{Q}_{50}(t_{b1})$. The constants of these relations are obtained from regression of test data from standard rating points according to EN255[2] (t_{b1} = -5, 0, +5, +10 ° and t_{w2} = 35 and 50 °C). This type of model works better than general models when a real, tested heat pump is the target (c.f. 5.1 Laboratory measurements).

4.3 Borehole model

Claesson[1] has developed an analytical solution for a single borehole that works very well indeed. For the heat pump competition a borehole model was created in Mathcad based on the following relations by Claesson (t_{R1} is the peripheral borehole temperature, radius R1):

$$t_{R1}(\tau_y) = t_{u.g} - \frac{\dot{q}_{m.bh}}{2 \cdot \pi \cdot \lambda_{rock}} \cdot g(\tau_y) \text{ with } g(\tau_y) = if(\tau_y \leq \tau_{1y}, g_{1e}(\tau_y), g_2) \quad (2)$$

with

$$g_{1e}(\tau_y) = \frac{1}{2} \cdot \left(\ln\left(\frac{16}{9} \cdot \frac{\tau_{nd}(\tau_y)}{D_{nd}{}^2} \right) - \gamma \right) \quad (\gamma = \text{Euler's constant}). \quad (3)$$

The mean specific borehole load, $\dot{q}_{m.bh} = Q_{bh.y} / H_{bh}$, influences the mean reduction of the undisturbed ground temperature at the borehole periphery. Short term reductions are calculated by superimposing the borehole specific load for each short term interval. Using data for the bedrock thermal conductivity, borehole depth etc. the calculated minimum brine temperature for the borehole was -3.0 °C (c.f. measurements in 5.2). The use of an analytical model has many advantages that compensate for the minor loss of computational accuracy that is introduced by simplifications necessary to find the analytical solution. The main advantages are physical understanding and computational speed.

4.4 Calculated heat pump COP for different system modifications

The heat pump model and borehole model, complemented by a heat exchanger model for the recharging coil, were used to calculate the overall heat pump coefficient of performance, COP_{hps}, as a function of outdoor temperature. Figure 7 shows the effect of the various system modifications.

I. Competition solution
II. Recharging
III. Pump operation
IV. Load matching (accumulator)
V. Low temperature system

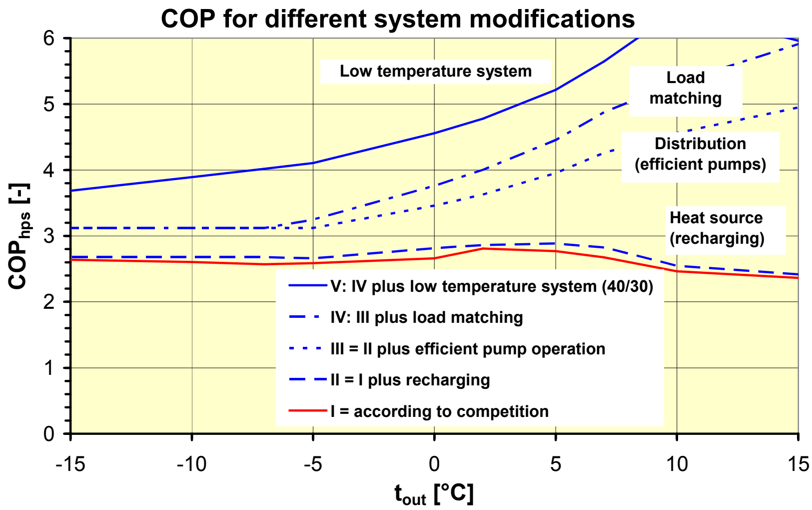


Figure 7: Simulated COP_{hps} as a function of outdoor temperature for the original installation and a number of upgrades.

It is obvious from curves I and II that the effect of recharging appears rather disappointing. One would expect that as outdoor temperature goes up so would also COP. This is clearly not the case and Fahlén[4, 9] has repeatedly pointed out the necessity of addressing the consequences on overall performance by changes to a singular part of the heat pump system. For instance, if one raises the temperature of the heat source by means of recharging, then the capacity will increase and unless something is done to the heat sink, this temperature will also go up and offset the potential advantage of a raised evaporating temperature. Also, during on-off operation, the heating water system that affects COP is the on-temperature which at high outdoor temperature can be higher than the on-temperature of continuous operation at a lower outdoor temperature. Improved load matching (IV) by means of the TES makes it possible to utilize the raised source temperature to its full potential and the diagram shows the substantial improvement of COP_{hps}. The diagram also indicates the effect of improved control of pump operation (III) and an increase of the heating system capacity (V).

5 MEASUREMENTS

The heat pump was installed in the simulated house and monitored from 1996-2000 to verify the calculated savings. A number of improvements were subsequently made to the installation (c.f. Figure 7) and after each improvement the function was verified by measurements.

5.1 Laboratory measurements

During the evaluation of the heat pump competition, SP Technical Research Institute of Sweden tested all heat pumps according to EN255[2]. The tests provided 8 rating points for each heat pump as basis for simple models (c.f. 4.2). Figure 8 shows the excellent agreement between model curves and rating points.

Designations
Heat pump: hp
(excluding circulators)
Heat pump system: hps
(including circulators)

Capacities
Thermal output: $\dot{Q}_{hp}$
Drive power input: $\dot{W}_{e.hp}$

Temperatures
Brine inlet: t_{b1}
Heating water outlet: t_{w2}

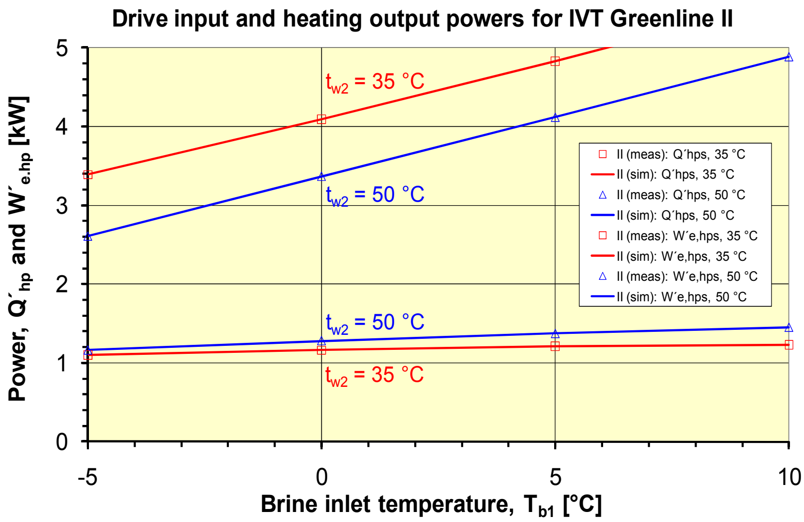


Figure 8: Comparison of simulated (curves) and measured (markers) heating capacity and compressor-motor drive power as a function of inlet brine temperature.

5.2 Field tests - Heat pump system I (original from heat pump competition)

The original system was monitored for a period of 4 years before any changes were made. Weekly mean values were collected of measured heating output and drive power input, temperatures indoors and outdoors, in the heating water and brine system, fan-coil inlet and outlet air, compressor on-time and relative operating time etc. Degree-day corrected savings agreed well with the estimated value of 9 MWh. The calculated minimum brine temperature for the borehole was -3 °C and the measured value was -2.9 °C (c.f. Figure 9).

5.3 Field tests - Heat pump system II (original + recharging of borehole)

In 2000 recharging of the borehole by heat from the exhaust air was introduced. Figure 9 a) shows the effect of recharging. Now the brine temperature rarely goes below 0 °C. However, as explained in 4.4, just raising the brine temperature will not improve COP unless the improvement on the source side is matched with corresponding improvements on the sink side. This is clearly shown by Figure 9 b) which indicates that COP is actually higher without recharging than with recharging most of the year.

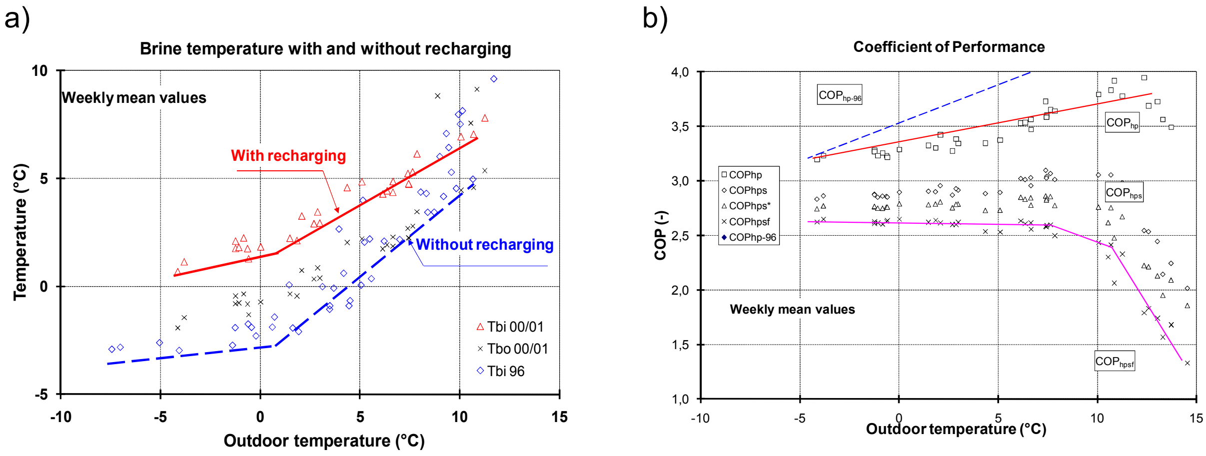


Figure 9: Measured values without and with recharging of a) brine temperatures without and with recharging and b) COP_{hp}, COP_{hps}, and COP_{hs} with and without recharging respectively.

The pressure difference for the heat pump condenser water pump will be quite low and constant when it operates directly on the storage tank. Furthermore, this pump only needs to run when the heat pump is operating and not continuously as in a traditional system. The same pump also effects the charging function for the SHW part but in this mode with a reduced flow and hence a lower power demand. As a consequence, the drive energy will decrease even though the number of pumps has increased. Selecting state-of-the-art motor technology and VSD has a potential to radically reduce the drive energy of pumps.

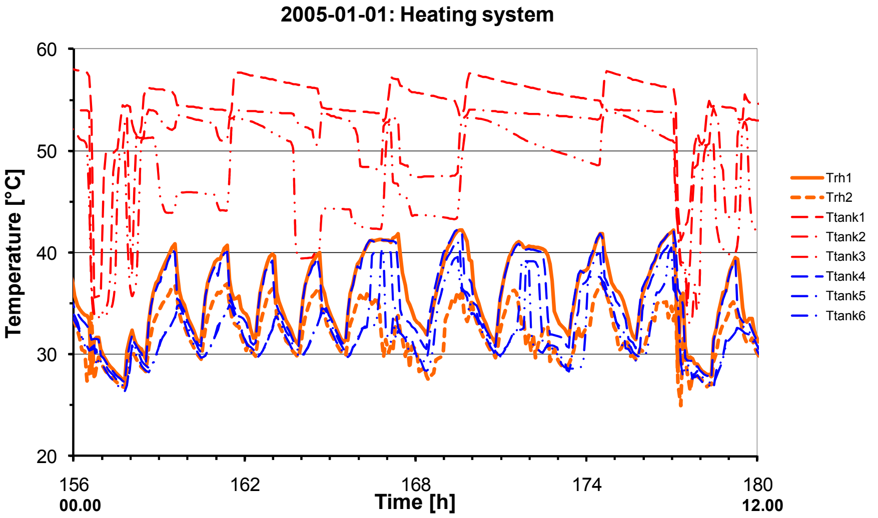
5.4 Field tests - Heat pump system IV (original+recharging+pump operation+TES)

Installation[5, 6] of the TES drastically improved the operating conditions of the heat pump. The maximum diurnal cycling frequency was reduced from 60 to 15, minimum on-times went up from 5 min to 30 min and the outlet heating water temperature from the condenser was reduced by 5-10 °C. The stratification in the tank was extremely good and there was very little interaction between the lower, heating part and the upper hot water part (water to the upper part is always preheated by the lower part). Figure 10 provides an extreme example from a new year's weekend with 9 people in the house, all having showers. Maximum tap flows (see Figure 10b) are almost three times the design value for the hot water heat exchanger HEX1 and still the system copes without destroying the stratification. It should be pointed out that the auxiliary electric heaters, one in the lower and one in the upper part, were not on. Detailed results, as illustrated in Figure 10, were recorded during half a year with occupation ranging between 0 and 9 persons (normally 2).

The installation of the case study originally contained two pumps and a fan (c.f. the system layout in Figure 5). By disconnecting the heat pump from the heating system (c.f. Figure 6), the flow rate in the heating system can be decreased in conditions of reduced heat demand and part-load operation. Fahlén[5] has shown that it is possible to save almost 1000 kWh by adopting state-of-the-art pump technology. The parasitic ratio, i.e. the drive energy to ancillary devices in relation to that of the compressor, as a consequence drops from 38 % to 8 %. In the original installation, unless one were to shut the heating system down manually during the summer period, the parasitic ratio would have been all of 50 %! That is, efficient pumps and intelligent system design and control can bring the parasitic ratio down from 0.5 or 0.38 to 0.08 and COP_{hps} will improve correspondingly.

a) Heating
rh = room heater
1 = outlet
2 = inlet

tank = TES temperatures;
1, 2 and 3 in the hot water part
4, 5 and 6 in the heating water part.



b) Hot water
hw = hot water
cw = cold water

ha.hw1 = heating agent inlet to HEX1
ha.hw21 = heating agent outlet from HEX1

tank = TES temperature;
1, 2 and 3 in the hot water part
4, 5 and 6 in the heating water part

V'_{hw} = hot water flow

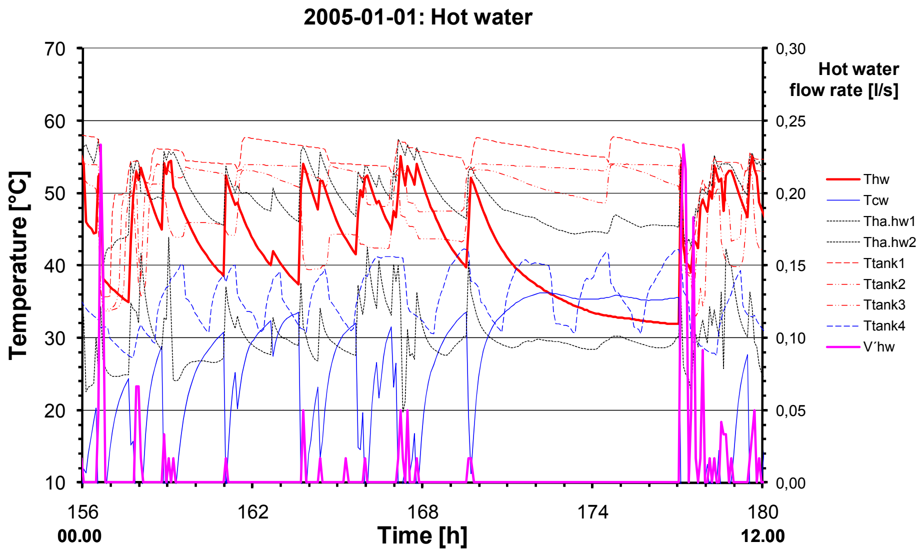


Figure 10: Comparison of temperatures in the TES during 24 h: a) focus on heating system operation and b) focus on hot water operation.

6 DISCUSSION

This paper proves the importance of overall system design. The performance of the original system has been thoroughly improved with no changes to the heat pump per se. Modifications reduced the purchased electric energy through a combination of recharging the ground storage, load matching with a storage tank and better control, increased room heater capacity, and reduction of the parasitic drive power ratio. Figure 7 illustrates the effects of alternative modifications to the original system (I). Neither in theory (Figure 7), nor in practice (Figure 9b) does the addition of a recharging system (II) have any noticeable effect on *COP*. However, together with more efficient pump operation (III), load matching (IV; a storage tank) and improved room-heater capacity (V) there are drastic improvements. SPF_{hps} goes up from 2.7 to around 4.2 for heating only and 3.7 including hot water. Figure 9b shows the *COP* 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). COP_{hps*} includes the recharging pump and COP_{hpsf} includes also the fan in the fan-coil unit.

Results confirm the prediction that performance will not directly improve from raising the borehole temperature by recharging with heat from the exhaust air. Unless something is done regarding the heat transfer capacity of the heating system the actual gains of recharging are miniscule. It is like fitting a larger heat pump to an existing system, the condensing temperature will rise, on-times will be shorter and the parasitic ratio will increase. The quickest savings are simply to go for the best available pump technology and control. By this, the original parasitic ratio of R_p = 0.38 can be reduced to 0.08. No change in the heat pump as such is likely to provide an efficiency improvement of this order (30 %).

The end result, after all modifications, is quite satisfactory. With no change to the heat pump, SPF_{hps} has improved by 30-40 % while at the same time providing both increased energy coverage for heating and now also including hot water (both factors will normally reduce *COP*). The total annual purchase of energy, including household electricity, has been reduced from 25 to < 10 MWh and the specific total purchase is now 67 W/m^2/year (see Figure 11). This is lower than in most modern passive houses being built in Sweden with super insulation, high efficiency heat recovery and solar heat.

Annual purchase, heating + hot water (Q_h+Q_{hw})

No heat pump
Total: 20 MWh
Specific:
133 kWh/m^2

Current hp
Total: 4.6 MWh
Specific:
30 kWh/m^2

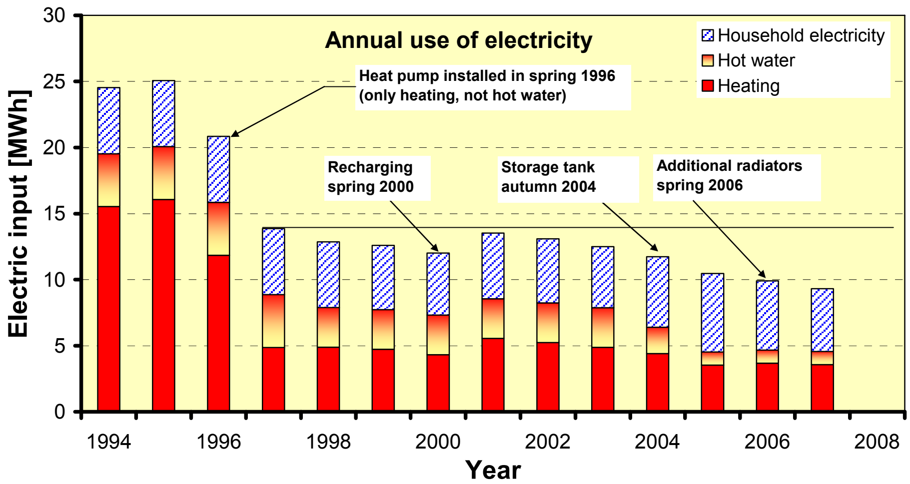


Figure 11: Measured values of electricity for heating, hot water and household purposes.

7 CONCLUSIONS

The initial installation was for space heating only and results indicate a reduction of the total purchased energy (electricity) from 25 to 14 MWh/year with SPF = 2.7. Estimated uses of household electricity and hot water (electric storage water heater) were 5 MWh each. After a number of modifications the total purchased electric energy has been reduced to 9 MWh per year and SPF has increased to 3.7 in spite of much higher energy coverage and the inclusion of domestic hot water. The results show that it is possible by means of a heat pump to achieve a specific energy use in a house built in 1977 that is lower than currently built passive houses in Sweden and that fulfils the future requirements of the Swedish building code. It also matches the vision by IVA for 2050.

The main conclusions are that by careful design and system optimization, it is possible to drastically improve on COP and energy coverage of a standard heat pump installation. In this case, energy coverage was raised by over 50 % and SPF by almost 40 % with no change to the heat pump per se. In particular, control strategies for heating and hot water, including how circulators are positioned and controlled, has a decided effect on overall performance.

8 NOMENCLATURE

Symbols, Latin letters

COP	coefficient of performance [-]	$\dot{V}$	volume flow rate [m^3/s]
Q	heat [J]	*R*	ratio [-]
$\dot{Q}$	power, thermal [W_{th}]	*W*	work, mechanical or electric [J]
SPF	seasonal performance factor [-]	$\dot{W}$	power, mechanical or electric [W]
t	celsius temperature [°C]		**Symbols, Greek letters**
T	thermodynamic temperature [K]	λ	Thermal conductivity [W/m/K]
U	Thermal transmittance [$W/m^2/K$]	τ	time [s], [h], [years]

Subscripts					**Abbreviations**
1	condenser	hp	heat pump	ACH	Air Changes per Hour
2	evaporator	hps	heat pump sys.	CAV	Constant Air Volume flow rate
a	air	hr	heat recovery	CSD	Constant Speed Drive
b	brine	m	motor	DOTH	Design Outdoor Temperature Heating
e	electric	out	outdoor	TES	Thermal Energy Storage
f	fan	p	pump, parasitic	VAV	Variable Air Volume flow rate
		w	water	VSD	Variable Speed Drive

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