

OPTIMIZING AND CONTROLLING MEDIA FLOWS IN HEAT PUMP SYSTEMS

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ABSTRACT

The *COP* of a *heat pump* depends on compressor efficiency and the condensing and evaporating temperatures. The *COP* of a *heat pump system* also depends on the drive powers to pumps and fans. For a given situation, there will be an optimum balance between heat transfer and drive power. The optimum can be found by considering flow related changes in drive power to the compressor, pumps, and fans. This necessitates individual optimization of flows in the heat pump and heating system. Minimizing pressure drops and selecting efficient pumps, fans and motors can improve the optimum. Motors must be suitable for variable speed drive in applications where such control is appropriate.

A case study of a residential heat pump installation underlines the importance of load-matching, optimized flow rates and efficient pumps, fans, motors and drives. Improvements in these aspects may reduce total drive energy by 20-40 % in small systems. Also, variable speed drives for pumps can render superfluous control valves and hence further contribute to reductions in pressure drop and pumping power.

Key words: *control, fan, flow rate, heat pump, motor, optimal, pump, Variable Speed Drive, VSD.*

1 INTRODUCTION

Heat pumps, in particular ground-source heat pumps (GSHPs), dominate the Swedish market for residential heating systems. In 2004 approximately 60 000 heat pumps were sold and all in all there are more than 500 000 installations in a country with only 9 million people. Typical applications include retrofitting of oil, wood or electric boilers in buildings with hydronic heating systems. Hence, the drive energy of heat pumps is beginning to show on the national electricity balance. The most frequent individual heating system in existing single-family houses, however, is direct-acting electric heating. In the future, this sector will be the target of retrofit initiatives due to rising electricity prices.

2 BACKGROUND

Traditionally, retrofitting of boilers based on economical sizing of the heat pump results in a capacity providing 50 % of the maximum heat demand and with energy coverage of around 80 %. Often an electric boiler covers the need for supplementary heat. Due to rising electric energy and power tariffs, there is an increased interest in full coverage heat pumps. This also leads to increased operation at part-load. Residential heat pumps, as indeed most HVAC-systems, will operate predominantly at part-load. A full coverage system will only operate at full load for a few days per year. The design capacity will therefore be much larger than the building need most of the time and to avoid frequent starts and short operating times there are two basic solutions for load matching:

- Adapting the capacity according to load (*capacity control*)
- Adapting the load to accept the capacity (*thermal energy storage* to accept superfluous capacity)

In most cases, pumps and fans for media flows in the heat pump system are designed to match the full-load conditions. At part-load they will be oversized if the compressor has capacity control or their operating times will be excessive unless they are coordinated with the compressor during on-off operation. Even though HVAC-systems employ numerous small and medium-sized pumps and fans, interest in their efficiency and operation has so far been low. This is mainly due to the low rated power of the individual components. The overall efficiency of the smallest units may be well below 10 % and their numbers and long operating hours result in substantial electric energy use (c.f. Table 1 in section 2). For instance, in indirect supermarket cooling systems it is not uncommon that the energy for pump operation exceeds the drive energy for refrigeration compressors. This can also be the case in residential heat pump systems.

3 PROBLEM

Most residential hydronic heat pump systems still operate with on-off capacity control. This causes the heating water temperature during the on-period to be higher than the mean value required by the heating system to satisfy the building heat demand (see Fig. 1).

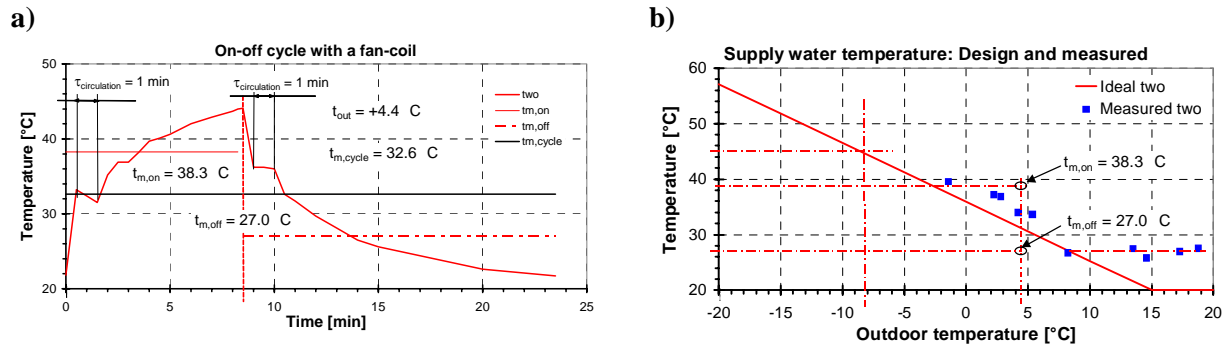


Fig. 1. a) A full on-off cycle with a heat pump directly operating on a fan-coil heating system and b) the actual temperature level compared to the theoretical value with load matching. The mean on-temperature is 38.3 °C while the cycle mean value is 32.6 °C.

At part-load with on-off control operating times will be short and the starting frequency high, which may negatively influence reliability (see Fig. 3 in Results). This also degrades the system efficiency compared to an ideal system with continuous load matching. The problem is accentuated in heating systems with a small water volume and in systems with a high coverage factor, i.e. heat pumps with a large capacity in relation to the heat demand of the building.

4 OBJECTIVE

The research program eff-Sys, sponsored by The Swedish Energy Administration and a number of companies, has looked at viable improvements in residential heat pump systems in two parallel research projects. The research has dealt with possible gains by means of better load matching of heat pump and heating system. One project deals with the possible gains from capacity control (Karlsson and Fahlén 2003) and the other with thermal energy storage (Fahlén 2004). The work has addressed the relation between heat pump, heating system and heating of sanitary water and how different ways of control and sizing affect system efficiency, the fraction of drive power to ancillary devices such as pumps and fans and the required supplementary heat.

One goal was to improve the understanding of optimization and control of hydronic heat pump systems by means of system analysis and component market surveys. Another goal was to gain practical experience from a standard installation upgraded by means of thermal energy storage, better control and more efficient pumps. Developments in the design of pumps, motors, and motor controls provide new possibilities of improving both the efficiency and quality of control.

5 MEDIA FLOWS IN HEAT PUMP SYSTEMS

In a heat pump system, media flows are necessary to transport heat from a source to the evaporator and from the condenser to a heat sink. The optimum values for these flows, however, do not necessarily have to be the same as the optimum values for the heating and hot water systems. The following discussion is based on a brine/water heat pump system but the principle is the same for other sources and sinks.

5.1 Evaporator Flow Rate

When the heat source is practically unlimited, as the case is with GSHP, the flow rate should be high to minimize the temperature difference and provide as high evaporating temperature possible (c.f. Eq. 3). Ground-source heat pumps (GSHPs) universally employ plate heat exchangers as evaporators. In many cases pressure drop on the brine-side has a fundamental influence on the selection of type and size. Therefore the heat exchanger may sometimes be oversized from a heat transfer point of view. A lot of effort has been spent on using the flow efficiently by different designs of distributors at the brine inlet to ascertain that most of the surface will be used for heat transfer. A consequence of the sizing is that for small residential systems the theoretical drive power to handle the pressure drop of the evaporator is only 5-10 W (c.f. the case study, Tables 1 and 2). The smallest available pumps, however, have a power input of 50-100 W.

Standard GSHP installations will only require a brine flow when the compressor is operating. Hence the pump, should be, and always is, switched off during heat pump off-periods. Small pumps, including brine pumps, have very low efficiencies and the losses will not be of use inside the building. Also, they will have fairly long operating hours and contribute substantially to the total drive power (8-9 % in the case study). Therefore, it is important to find more efficient brine pumps for the smallest segment of GSHPs. The number of operating hours will decrease as energy coverage goes up but the relative importance will still be the same. In applications with capacity control of the compressor, the brine pump must also be controlled accordingly. Otherwise its relative share of the total drive power will quickly increase (c.f. Eq. 11).

5.2 Condenser Flow Rate

In a system where the heat pump is isolated from the heating system by means of a buffer tank (c.f. Fig. 2b), the condenser flow rate can be optimized purely from the point of view of heat transfer/pressure drop. With plate type condensers the outlet water temperature will be approximately the same as the condensing temperature in a wide range of flow rate. Depending on the type of refrigerant, the optimal temperature difference and flow rate may vary. With R407C, for instance, a fairly large difference is often used to benefit from the inherent temperature glide of 4-5 K. Hence the optimal flow is normally only half of that in the evaporator. This results in a theoretical drive power for the heating water pump which is even less than that of the brine pump, only around 2-3 W, and it becomes even more difficult to find sufficiently small pumps with acceptable efficiency. Just as the case is for the brine pump, the water pump should only operate during the compressor on-period. If the compressor has capacity control, the heating water pump must also be controlled. Most small GSHPs, however, connect directly to the heating

system of the building. In this situation, the heating system will also influence the magnitude of the optimum flow rate (see 1.3). On the condenser side, pump losses will contribute to the heating. Even so, it is much more efficient to heat the building by means of the heat pump than by pump losses.

5.3 Heating System Flow Rate

Fahlén, Naumov and Voll (2004) have shown that flow control of the heating capacity on the primary side of room-heaters is not very effective until the primary heat capacity flow rate is less than the secondary. Therefore, in a hydronic fan-coil system it is more effective to control the airflow than the water flow. In radiator or under-floor heating systems, this is of course not possible. Still, heating is normally controlled on the waterside by means of valves and thermostats. Controlling the flow by means of VSD pumps, however, can be more efficient and have better control characteristics than throttling. Furthermore, the optimal flow rate in the heating system is generally not the optimal condenser flow rate of the heat pump. Therefore, it is often advantageous to separate pumping and control of these two media flows (c.f. Fig. 2b). Furthermore, there is a need for more or less continuous circulation in the heating system to avoid discomfort through large variations in room-heater temperature and cold draught at windows during off-periods.

5.4 Water Heater Flow Rate

The largest variations in flow rate, both in amplitude and frequency, will be found in water heaters for sanitary water. Depending on whether a direct heating or a storage heating system is used, the requirements on the primary flow will be quite different. Fahlén (2005) has studied a system with direct heating of the sanitary water from storage on the primary, heating-water side. Primary flow is controlled to provide a preset outlet temperature of the SHW. Compared to other pumps, units for SHW will have much shorter operating hours and deliver relatively much higher thermal capacities ($\dot{Q}_{shw,max} \approx 25-30$ kW while $\dot{Q}_{hs,max} \approx 5-10$ kW). Hence, their efficiency will be less important whereas controllability is crucial. On the other hand, hot water circulator pumps, maintaining SHW temperature at the tap, can use unreasonably large amounts of electric energy. If used, efficiency of such units is crucial.

6 OPERATIONAL CLASSIFICATIONS OF PUMPS AND FANS

Pump duty can be classified in terms of the pump power in relation to the heat capacity flow rate (c.f. Table 1). Applications vary from 1) low power and long operating hours, e.g. for heating and cooling systems; 2) high power and short operating hours, e.g. for direct sanitary water heating systems, and 3) average power and average operating hours, e.g. for condenser and evaporator flows of refrigerating equipment and heat pump installations. The principle discussion is valid for both pumps and fans but we will only deal with pumps explicitly. The motor drive energy depends on the number of units and their individual power input and operating hours according to:

$$W_{e,p} = \sum_{i=1}^{i=n} \dot{W}_{e,p}^i \cdot \tau_{on}^i \quad [\text{J}] \text{ or } [\text{kWh}] \quad [\text{Eq. 1}]$$

where $W_{e,p}$ = total energy for pump operation, $\dot{W}_{e,p}^i$ = power input to pump no. i and τ_{on}^i = operating hours of pump no. i . The power input of each pump is decided by functional requirements on flow rate, resulting pressure drops in the system, and the overall efficiency of the pump according to:

$$\dot{W}_{e,p} = \frac{\dot{V} \cdot \Delta p}{\eta_p} \text{ [W]} \quad [\text{Eq. 2}]$$

Equations 1 and 2 indicate that the drive energy can be reduced through the following actions (individually or in combinations): 1) reduce the number of pumps, 2) the operating hours, 3) the flow rates and pressure drops and 4) increase the efficiencies. Alternative system layouts, control strategies, and the selection of coils will influence the first three items. New pump motor controls enhance the possibilities of more efficient control. Improved hydraulic design and more efficient motors will address the final opportunity for savings.

To assist the analysis of pump and fan work, some type of classification is helpful. Table 1 provides an example for the heating side of a small residential heat pump installation (Fahlén 2004; see results). Depending on the function, requirements will be quite different. Long operating hours at fairly constant capacity puts the focus on efficiency at the design point whereas short operating hours in a wide capacity range stresses controllability and dynamic range. Obviously, the higher the efficiency of the pump, the higher the optimal flow rate will be and the less important becomes controllability. Another important aspect is the ratio between delivered heat and drive energy to the pump. For SHW applications, maximum heating capacity is high and on-time is short. Therefore pump efficiency is not as crucial as in the case of the heating system pump or the brine pump. On the other hand, controllability will be important for the SHW pump (regarding the function of this pump, see chapter 4 and Fig. 2b).

**Table 1. Operational time, drive power and transported heat for heating and hot water.
An example from the case study of a small residential ground source heat pump.**

Function of pump/fan	Operating time [h]	Pumping power [W]	Heat [kW]	Comment
Evaporator flow rate	Approx. 3 000	Theory: 5 – 10 Real: 40 - 50	Approx. 4.2	Average on-time, average thermal capacity
Condenser flow rate	Approx. 3 000	Theory: 2 – 3 Real: 40 - 50	Approx. 4.2	Average on-time, average thermal capacity
Heating system flow rate	Approx. 6 600	Theory: 2 – 3 Real: 40 - 50	0 – 4.2	Long on-time, low to middle thermal capacity
Fan-coil	Approx. 6 600	Theory: 3 – 5 Real: 60 - 80	0 – 4.2	Long on-time, low to middle thermal capacity
SHW heat exchanger	Approx. 600	Theory: 1 – 3 Real: 30 - 40	0 – 25	Short on-time, low to high thermal capacity

7 OPTIMIZING AND CONTROLLING MEDIA FLOWS

For a specific heat pump design at given operating conditions, there will be an optimum combination of media flows at the evaporator and condenser respectively. The compressor drive power depends on the compressor and motor efficiency and the operating conditions in terms of condensing and evaporating temperature. Increasing the media flow rates tends to reduce the compressor drive power by lowering the condensing temperature and raising the evaporating temperature via two mechanisms:

- Increase of the mean temperature difference (θ).
- Increase of the heat exchanger thermal transmittance (U).

Drive power to pumps and fans, however, will rise with increasing flow rate. It pays to raise the flow rate as long as the increase in drive power to pumps is smaller than the decrease in power input to the compressor.

7.1 Influence on COP from Temperature and Pumping Power

Assuming that the compressor efficiency is not affected by minor changes in the operating pressures, COP_{hp} will change with temperature the same way as the theoretical COP_{IC} . Differentiating the theoretical expression for COP_{IC} we have (1 refers to the condenser and 2 to the evaporator):

$$\frac{\Delta COP_{hp}}{COP_{hp}} = \frac{\Delta COP_{IC}}{COP_{IC}} = \left[\frac{\Delta T_2}{T_1 - T_2} - \frac{T_2}{T_1} \cdot \frac{\Delta T_1}{T_1 - T_2} \right] \approx - \frac{\Delta \dot{W}_{e, hp}}{\dot{W}_{e, hp}} \quad [-] \quad [\text{Eq. 3}]$$

assuming that the heating capacity is fixed and given by the building need. The overall COP_{hps} of the heat pump system includes also drive powers to brine and heating water pumps:

$$COP_{hps} = \frac{\dot{Q}_1 + \dot{W}_{ep1}}{\dot{W}_{e, hp} + \dot{W}_{ep1} + \dot{W}_{ep2}} = COP_{hp} - \frac{(COP_{hp} - 1) \cdot \dot{W}_{e, p1} + COP_{hp} \cdot \dot{W}_{e, p2}}{\dot{W}_{e, hps}} \quad [\text{Eq. 4}]$$

Assuming $\dot{W}_{e, p1} + \dot{W}_{e, p2} \ll \dot{W}_{e, hp}$, differentiating Eq. 4 yields:

$$\frac{\Delta COP_{hps}}{COP_{hps}} \approx [1 - R_p] \cdot \frac{\Delta COP_{hp}}{COP_{hp}} - \frac{(COP_{hp} - 1)}{COP_{hps}} \cdot \frac{\Delta \dot{W}_{e, p1}}{\dot{W}_{e, hps}} - \frac{COP_{hp}}{COP_{hps}} \cdot \frac{\Delta \dot{W}_{e, p2}}{\dot{W}_{e, hps}} \quad [\text{Eq. 5}]$$

with the parasitic ratio $R_p = (\dot{W}_{e, p1} + \dot{W}_{e, p2}) / \dot{W}_{e, hps}$ (c.f. Tables 3 and 4). Eq. 5 describes how changes in condensing and evaporating temperatures (COP_{hp} via Eq. 3) and drive powers to pumps affect the overall COP. The expression indicates that the brine pump (p2) obviously has a higher influence than the water pump (p1).

7.2 Influence on Temperature from Changes in Media Flows

Changes in media flows will affect condensing and evaporating temperatures and hence COP_{hp} (see Eq. 3). The following discussion concerns the evaporator but the same principle applies also to the condenser. Using the $\varepsilon - N_{tu}$ method for a counter-flow heat exchanger, the thermal effectiveness ε can be expressed as:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{1 - \exp\{-N_{tu} \cdot (1 - R)\}}{1 - R \cdot \exp\{-N_{tu} \cdot (1 - R)\}} \quad \text{with } R = \frac{\dot{C}_{\min}}{\dot{C}_{\max}} \quad \text{and } N_{tu} = \frac{U \cdot A}{\dot{C}_{\min}} \quad [\text{Eq. 6}]$$

In the evaporator, the refrigerant side will have the maximum heat capacity flow and the evaporating part, with approximately zero temperature change, is equivalent to an infinite heat capacity flow rate. Hence $R \approx 0$. This will simplify Eq. 6 and, using the relationship $\dot{Q} = \dot{C} \cdot \Delta t$, Eq. 6 can provide an expression for the difference between the inlet brine temperature and the evaporating temperature:

$$T_{b,i} - T_2 = \frac{\dot{Q}_2}{\dot{C}_b \cdot (1 - \exp\{-N_{tu}\})} = \frac{\dot{Q}_2}{\dot{C}_b \cdot (1 - \exp\{-(UA)_2 / \dot{C}_b\})} \quad [\text{Eq. 7}]$$

The brine inlet temperature is an external factor, which we assume to be constant, i.e. not possible to influence by the heat pump per se or its control system. The thermal transmittance U , on the other hand, will be influenced by changes in the brine flow rate and the thermal capacity. The brine-side heat transfer coefficient can be assumed to vary with the flow rate raised to an exponent n . The value depends on the specific type of heat exchanger ($n \approx 0.3$ in laminar flow and $n \approx 0.8$ in turbulent flow). On the refrigerant side, the thermal capacity will influence the evaporating coefficient of heat transfer. In most designs, however, the heat transfer resistance on the refrigerant side is much smaller than on the brine side and then we need only look at the effect of changes in brine flow. Differentiating Eq. 7 we have:

$$\frac{\Delta \dot{W}_{e,hp}}{\dot{W}_{e,hp}} = -\frac{\Delta T_2}{T_{b,i} - T_2} = \frac{\Delta \dot{Q}_2}{\dot{Q}_2} - \frac{\Delta \dot{C}_b}{\dot{C}_b} \cdot \left[1 + \frac{(1-n) \cdot N_{tu,2}^2}{(\exp(N_{tu,2}) - 1)} \right] \quad [\text{Eq. 8}]$$

Eq. 8 indicates the effect on the evaporating temperature T_2 from changes in capacity and flow rate. An increase in capacity will lower T_2 while an increase in flow rate will raise T_2 . This influence on temperature can be used in Eq. 5, via Eq. 3, to see the effect on COP_{hp} and, more importantly, COP_{hps} . For the latter effect we need also the influence of flow rate changes on drive power to the pumps.

7.3 Influence on Drive Power to Pumps from Changes in Media Flows

Eq. 2 gives the relation between drive power and flow rate. The pressure difference will, of course, also depend on flow rate. This dependence can often be expressed with the flow rate raised to an exponent m ($m = 1$ for laminar flow and $m \approx 2$ for turbulent flow). Using this relation in Eq. 2 and differentiating the resulting equation yields:

$$\frac{\Delta \dot{W}_{e,p2}}{\dot{W}_{e,p2}} = (m+1) \cdot \frac{\Delta \dot{V}_b}{\dot{V}_b} - \frac{\Delta \eta_{p2}}{\eta_{p2}} \approx (m+1) \cdot \frac{\Delta \dot{C}_b}{\dot{C}_b} - \frac{\Delta \eta_{p2}}{\eta_{p2}} \quad [\text{Eq. 9}]$$

7.4 Optimizing and Controlling Media Flow Rates

Knowing the influence of flow rate on COP_{hp} (Eq. 3, 5 and 8), and hence on drive power to the heat pump, together with information on drive power to pumps (Eq. 9) makes it possible to find the optimum flow rates at the evaporator and condenser. The relations above in 3.1 and 3.2 can assist an integrated, optimizing control system in finding this optimum. Different types of optimizing methods, such as the Simplex method (Karlsson 2003), will usually find the optimum when given time. If the operating conditions change quickly, however, direct relations as those described can speed up the process substantially. Having established an optimum at given conditions, when these conditions change the system will know directly in which direction to go and approximately how far. It will pay off to increase the flow rate as long as the decrease in drive power to the compressor is larger than the increase in drive power to the pump. At the optimum they will be equal and there will be no point in going any further. Then we have:

$$\dot{W}_{e,p2} \cdot \left[(m+1) \cdot \frac{\Delta \dot{C}_b}{\dot{C}_b} - \frac{\Delta \eta_{p2}}{\eta_{p2}} \right] = \dot{W}_{e,hp} \cdot \left[-\frac{\Delta \dot{Q}_2}{\dot{Q}_2} + \frac{\Delta \dot{C}_b}{\dot{C}_b} \cdot \left[1 + \frac{(1-n) \cdot N_{tu,2}^2}{(\exp(N_{tu,2}) - 1)} \right] \right] \quad [\text{Eq. 10}]$$

and simplifying by assuming constant pump efficiency we get:

$$\frac{\Delta \dot{C}_b}{\dot{C}_b} = \frac{\Delta \dot{Q}_2}{\dot{Q}_2} \cdot \left[1 + \frac{(1-n) \cdot N_{tu,2}^2}{\left(\exp\{N_{tu,2}\} - 1 \right)} - (m+1) \cdot \frac{\dot{W}_{e,p2}}{\dot{W}_{e,hp}} \right]^{-1} \quad [\text{Eq. 11}]$$

A change in capacity necessitates a change in flow rate. How much will be determined by the characteristics of the heat exchanger (m , n , U , A) and the efficiencies of the pump and compressor respectively (contained implicitly in $\dot{W}_{e,p2}$ and $\dot{W}_{e,hp}$).

8 A CASE STUDY

One of the objectives of this study was to gain practical experience from a standard installation upgraded by means of thermal energy storage, better control and more efficient pumps. For this purpose, an installation that has been monitored since 1996 as part of the evaluation of the Nordic Heat Pump Competition was chosen. The system was converted from the simple, direct operation according to Fig. 2a to the more intricate design of Fig. 2b.

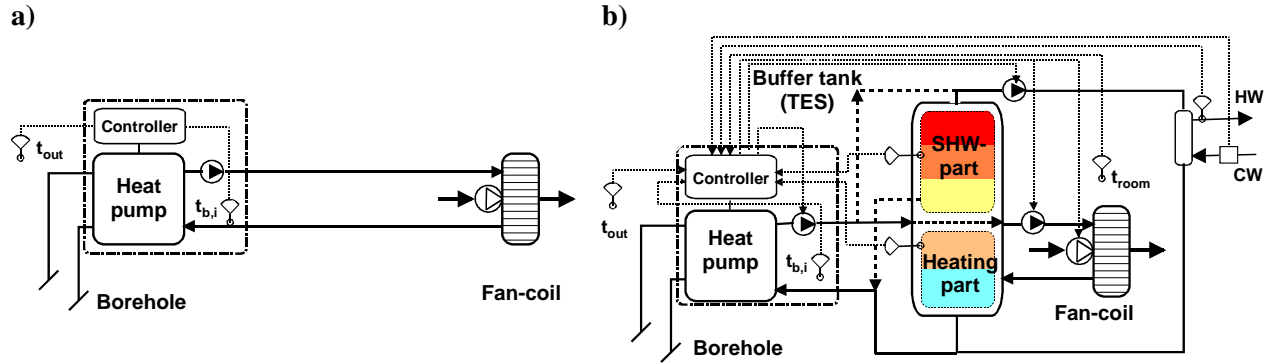


Fig. 2. The original heat pump system (a) and the modified system (b) with thermal energy storage in the heating system and integrated heating of sanitary water. System (b) contains three VSD pumps.

The original heat pump heating system simply controlled the heat supply by operating on-off according to the return water temperature in relation to the outdoor temperature (open-loop control). The modified system controls the heat pump operation by means of a comparison of the state of the storage (temperature, quantity) with the outdoor temperature. The heating water pump of the heat pump operates at an optimized constant flow rate to charge the storage according to demand. Each time when the heating part of the TES has been fully charged, the flow is directed to the top of the SHW part but in this mode the pump speed is controlled to charge with a constant, high temperature. Hence there will be suitable hot water available directly. A SHW sensor in the tank will also trigger the SHW mode. A flow switch starts a second pump as soon as there is a hot water draw-off. A hot water sensor controls its speed to deliver hot tap water at a selectable temperature. Finally, a third VSD pump controls the heat supply to the building via a feedback PID-controller and a room temperature sensor.

Table 2. Linear specific capacity, goodness number and thermal inertia for room heaters in the case study.

Room heater	Linear specific capacity [W/K/m]	Goodness number [\dot{W}_{heat}/\dot{W}_e]	Thermal inertia [min]
Fan-coil	160	40*	1
Radiator (TP11)	15	640**	23
Radiator (TP22)	27	640**	26

* W_e is estimated with recommended flows, fan speed II (85 W) and $\eta_p = 0.1$.

** W_e is estimated with recommended flows, $\Delta p = 2$ kPa and $\eta_p = 0.1$.

Table 2 illustrates the difference in the characteristics of the room heaters used in the case study. A fan-coil was preferred because of a specific linear capacity, which is much higher than that of radiators. One fan-coil unit will deliver the same capacity as five to ten radiators each with the same length as the fan-coil. On the other hand, the fan-coil will require around sixteen times more electric drive power for the heat distribution and much more frequent starts because of a low thermal inertia (c.f. Fig. 3b). This is where the thermal energy storage comes in.

9 RESULTS AND ENERGY SAVING POTENTIAL

Fig. 3 clearly illustrates the effects of the thermal energy storage. With a TES, minimum on-time increased from 6 to 30 minutes and maximum starting frequency dropped from over 60 to 15 starts diurnally. There are two curves without the TES, one without and one with recharging of the borehole. With recharging, the heat source temperature will increase and hence will also the capacity of the heat pump. An increased capacity results in shorter operating times and higher starting frequencies.

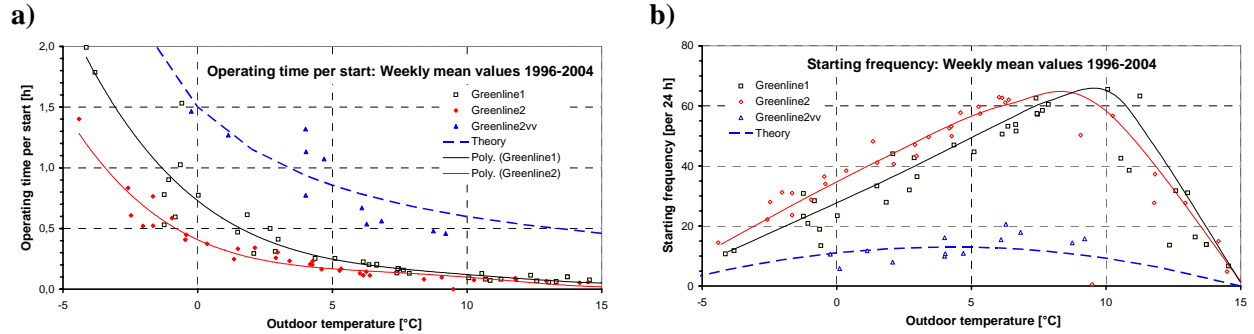


Fig. 3. Weekly mean values of a) the number of starts per 24 h and b) the operating time per start. Without TES and recharging of the borehole (filled squares), without TES but with recharging (unfilled squares), with TES and with recharging (triangles).

The stratification of the TES has worked exceedingly well. Fig. 3 shows the temperature in the tank at three levels in the SHW part and at three levels in the heating system part.

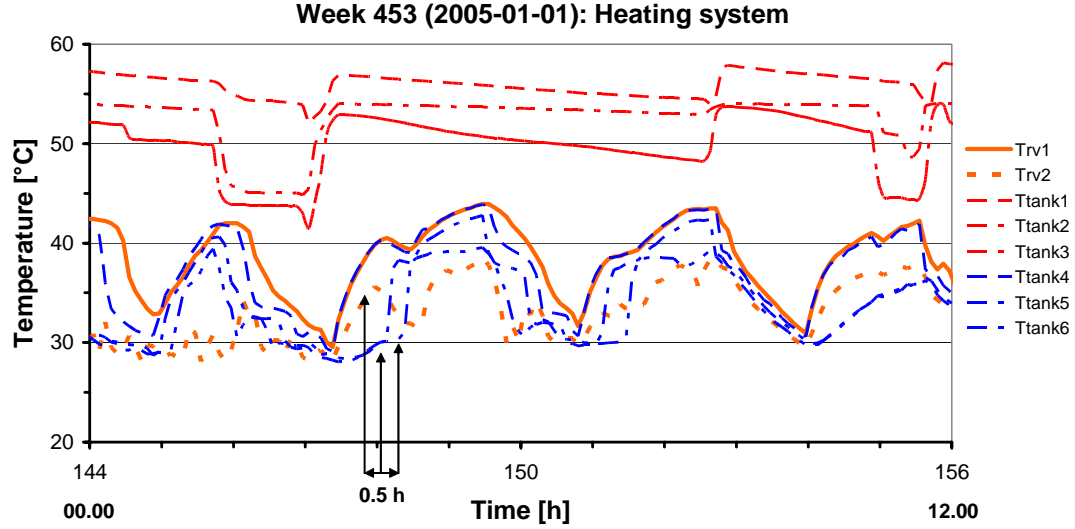


Fig. 4. Temperatures at six different levels in the TES, three in the SHW part and three in the heating part.

The top three temperatures are quite unaffected by large variations in the bottom part. At times there is a difference of more than 20 K between T_{tank3} and T_{tank4} , only 0.1 m apart. Even during very large draw-offs, when the SHW part is almost empty, the top temperature does not fall by much and when recharging starts, T_{tank1} rises directly. The graph also indicates the effect of the control-on-demand heating system pump with a VSD. Some time after a heat pump start, the return temperature drops because the heating system flow is reduced due to a satisfied heating demand. During the half-hour it takes to charge the lower part of the TES, the return-feed comes from the exhausted storage, which is mixed with a gradually reduced temperature from the heating system.

The installation of the case study originally contained two pumps and a fan (c.f. the system layout in Fig. 2a). By disconnecting the heat pump from the heating system (c.f. Fig. 2b), the flow rate in the heating system can be decreased in conditions of reduced heat demand and part-load operation. Also, 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 affects 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. Table 3 presents the function and power requirement of the pumps and fan and compares the drive powers with that of the heat pump compressor. The table indicates that selecting state-of-the-art motor technology and VSD has a potential to radically reduce the drive energy of pumps.

Table 3. Measurements and calculations for the case study during 2002 (total measuring period of 8779 h).
The heating season is January to May and September to December.

Function (motor speed)	Power \dot{W}_e [W]	Operating time τ_{op} hours [h] relative [-]	Drive energy W_e [kWh]	Parasitic ratio R_p [-]
Compressor	1055	2995 0.34	3159	1.00
Brine pump (IV)	96	2995 0.34	288	0.09
Heating water pump (II)	48	6595 0.75	317	0.10
Fan-coil fan (II)	61	6595 0.75	402	0.13
Recharging pump (I)	37	5498 0.63	203	0.06
Parasitic sum	242		1210	0.38

Total sum	1297		4369	
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A comparison of Tables 3 and 4 shows that it is possible to save almost 1000 kWh (1210-242 = 968 kWh) by adopting state-of-the-art technology with around 40 % efficiency (Nipkow and Meyer 1999). 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 % (c.f. equation 4). Unless one shuts the heating system down manually during the summer period, the parasitic ratio would have been all of 50 %! That is, the parasitic ratio goes from 0.5 to 0.38 to 0.08 and COP_{hps} will improve correspondingly.

Table 4. Calculations for the case study of Table 1 but using state-of-the-art pump and fan motor drives.

Function (motor speed)	Power \dot{W}_e [W]	Operating time τ_{op}		Drive energy W_e [kWh]	Parasitic ratio R_p [-]
		hours [h]	relative [-]		
Compressor	1055	2995	0.34	3159	1.00
Brine pump (IV)	19	2995	0.34	58	0.09
Heating water pump (II)	10	6595	0.75	63	0.10
Fan-coil fan (II)	12	6595	0.75	80	0.13
Recharging pump (I)	7	5498	0.63	41	0.06
Parasitic sum	48			242	0.08
Total sum	1103			3401	

To achieve corresponding savings by means of a more efficient heat pump would require an improvement of COP_{hp} from the measured seasonal mean value of 3.13 to 4.09. This, of course, is entirely possible through improved compressor technology and up-sized heat source and sink. On the other hand, the improvements are complementary and hence it is possible to save approximately 2000 kWh by improvements in sizing and choice of components. Using high efficiency pumps, the need for VSD is no longer necessitated by the problem of parasitic drive power but rather from a need for better control of heating and hot water temperature and requirements on thermal comfort.

10 DISCUSSION AND CONCLUSIONS

Optimizing space heating and heating of sanitary water by means of a heat pump first involves finding a suitable balance between building, room and water heaters, conveyors of heat transfer media, heat pump, and heat source. The COP of the *heat pump* depends on compressor efficiency and the condensing and evaporating temperatures. The COP of a *heat pump system* also depends on the drive powers to pumps and fans. A simple expression can illustrate how the base level temperatures, the result of sizing for the design load, affect COP and drive power.

For a given base design, there will be an optimum balance between heat transfer and drive power. The optimum can be found by considering flow related changes in drive power to the compressor, pumps, and fans. Conveying devices with higher efficiencies result in larger optimal flow rates. To find the optimum, one needs relationships describing both the influence on condensing and evaporating temperature of the heat transfer media flows and the influence of media flows on the drive power to the conveying devices. This also necessitates individual optimization of flows in the heat pump, heating and hot water systems.

Obviously, the optimum will differ from the base load situation and it will change continuously in systems with heating capacity control. Hence, there is a need to optimize and control the media flows. In systems with integrated optimizing control (Karlsson 2003), it is possible to find the optimum by means of an iterative process such as the Simplex method. In situations with rapid changes, and to generally

speed up the process, it helps to have an explicit expression that indicates in which direction and how far to go. This can be achieved by using reasonably simple expressions for heat transfer and pressure drop to obtain a direct relationship between COP_{hp} , capacity and media flow rates. Differentiation of these correlations yields a linearized model for the relative effect on COP_{hp} from relative changes in capacity and media flows. This directly indicates how much the condenser or evaporator flow rate should change to maintain optimal operation in response to a change in capacity decided by a change in the heat demand. Minimized pressure drops and use of efficient pumps, fans and motors can improve the optimum. Motors must be suitable for variable speed drive in applications where such control is appropriate.

A case study of a residential heat pump installation underlines the importance of load-matching, optimized flow rates and efficient pumps, fans, motors and drives. The potential improvement of COP_{hps} by use of state-of-the-art motor technology and motor control for pumps and fans is 20 – 40 %. This is noteworthy and must be compared to the effort in achieving corresponding gains from improvements in compressor efficiency and heat exchanger technology. Most likely, the path of addressing the parasitic ratio, i.e. the ratio of drive energy to ancillary devices and that of the compressor, will be much more cost effective at the present state of development. Furthermore, direct flow control with variable speed drives can render superfluous many balancing and control valves of traditional systems. As a result, pressure drop and pump power may be substantially reduced just from a new control principle.

NOMENCLATURE

Roman symbols		Units	Subscripts	
A	area	[m ²]	b	brine
c	specific heat capacity	[J/kg/K]	C	Carnot
C	heat capacity ($C = c \cdot M$)	[J/kg]	e	electric
\dot{C}	heat capacity flow rate	[W/K]	f	fan
COP	Coefficient of Performance	[W _{thermal} /W _{electric}]	h	heating
M	mass	[kg]	hp	heat pump (compressor)
\dot{M}	mass flow rate	[kg/s]	hps	heat pump system
N	Number (of transfer units)	[-]	hs	heating system
\dot{V}	volume flow rate	[m ³ /s], [kg/s]	i	inlet (of heat pump)
Q	heat (thermal energy)	[W]	m	mean; motor; mass
\dot{Q}	heating capacity (thermal power)	[W]	o	outlet (of heat pump)
R	Ratio	[-]	op	operating condition
t	temperature (Celsius)	[°C]	p	pump; pressure; parasitic
T	temperature (absolute)	[K]	rh	room heater
U	thermal transmittance	[W/m ² /K]	$room$	room
W	work (mechanical or electric)	[J]	sh	supplementary heating
\dot{W}	power (mechanical or electric)	[W]	shw	sanitary hot water
Greek symbols			tu	transfer units
ε	heat exchanger effectiveness	[-]	w	water
θ	temperature difference	[K]		
ρ	density	[kg/m ³]	1	condenser (heating)
τ	time	[s]	2	evaporator (cooling)
Abbreviations				
$GSHP$	Ground-source heat pump	TES	Thermal Energy Storage	
SHW	Sanitary Hot Water	VSD	Variable Speed Drive	

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