EFFICIENCY ASPECTS OF HEAT PUMP SYSTEMS - LOAD MATCHING AND PARASITIC LOSSES

P. Fahlén, professor of Building Services Engineering,
Chalmers University of Technology, SE 412 96, Gothenburg, Sweden

Abstract: Heat pumps often provide efficient solutions for refrigeration, cooling, heating and hot water. Although there still remains much to improve on the efficiency of the heat pump per se, in many cases much larger energy savings can be achieved by improving the design and component efficiencies of the systems that connect the heat pump to the load. Also, when building envelopes improve and control systems better match supply and demand, the active operation of the heat pump unit is reduced. Even now, some applications see more electric energy going into distribution and control than to the operation of the heat pump compressor. Examples show how the efficiency of existing heat pump systems may be greatly improved with no changes to the heat pump, how new HVAC system designs may drastically reduce drive energy to pumps and fans and how the use of ambient conditions and ground storage may provide cost-effective and efficient means for space conditioning.

Key Words: ***control, efficiency, fans, heat pump systems, parasitic losses, pumps***

1 INTRODUCTION

Heat pumps often provide efficient solutions for refrigeration, cooling, heating and hot water. Although there still remains much to improve on the efficiency of the heat pump per se, in many cases much larger energy savings can be achieved by improving the design and component efficiencies of the systems that connect the heat pump to the load.

1.1 Heat Pump Applications

Heat pump systems range from applications with few alternative solutions, such as low temperature refrigeration, to those in fierce competition with other options, e.g. heating only heat pumps. Specific conditions of different applications affect the possibilities of utilizing natural ambient sources and sinks. The specific application also affects the utilization factor (relative operating time) of the heat pump and hence the relative importance of parasitic drive energies and heat losses. Applications may be classified in relation to the temperature deviation of the conditioned space from normal ambient conditions (see figure 1):

1. Frozen and deep-frozen food; large negative deviation, difficult to find natural sinks.
2. Chilled food; moderate negative deviation from room temperature but with natural low temperature sinks in outdoor Nordic climates.
3. Space conditioning (cooling only) in warm climates; moderate negative deviation, few natural sinks.
4. Space conditioning (cooling and heating) in moderate and cold climates; small negative to large positive deviations, ample opportunities for low temperature sinks and sources (see example in 4.1).
5. Space conditioning and hot water (heating only) in moderate and cold climates; large positive deviations (see example in 4.2).

This paper will address some general efficiency aspects of heat pump systems providing two examples (categories 4 and 5).

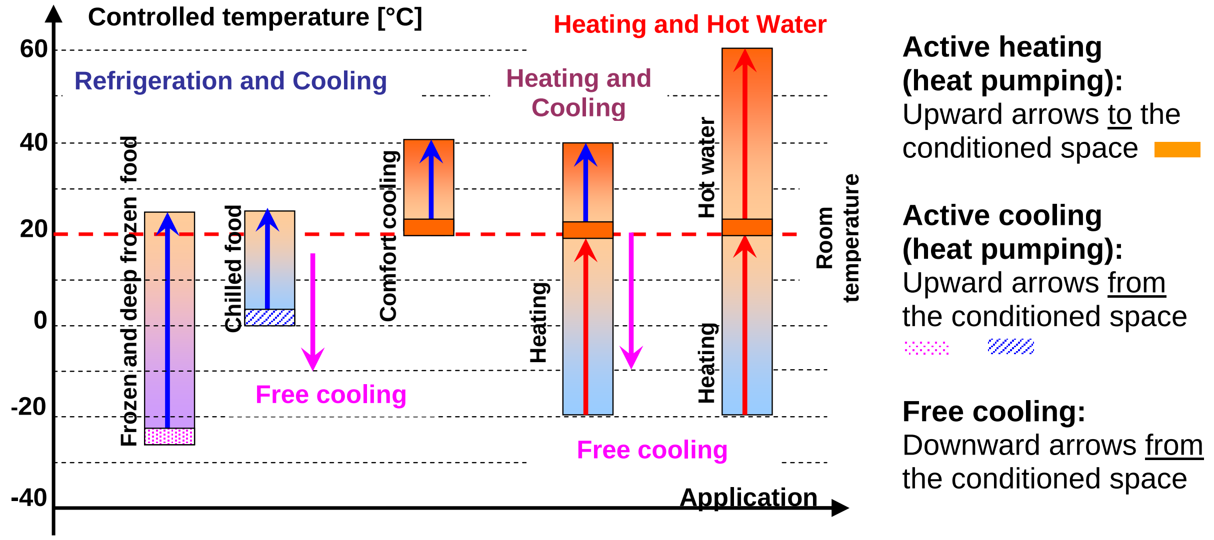


Figure 1: Temperature ranges of typical heat pump applications.

1.2 Heat Pump System Efficiency Aspects

The theoretical energy use of a heat pump system depends primarily on the application and the system design. The application decides the required temperature of the conditioned space and the location provides the ambient climatic condition. In many cases, there are ample opportunities to complement the heat transport achieved by the heat pump (upward arrows in figure1) by heat flows driven by positive temperature differences (free cooling, downward arrows in figure 1). For a specific application there are a number of possibilities to improve the overall System Performance Factor (*SPF*; see 2.3) by optimal uses of free energy and drive power. Some examples (see also section 3) are to:

- Reduce demand by load matching and storage
- Reduce purchase of energy by means of natural sources and sinks
- Reduce drive energy for
  - heat pumping through reduction of temperature lift
  - heat transfer by suitable heat exchangers (new designs!) and optimized flow control
  - heat transport by optimal system design and optimized flow control
  - terminal units by suitable choice (new designs available!) and optimized flow and temperature control

2 BASICS

In the discussion of more efficient use of energy it is essential to distinguish between energy demand and energy use. Energy use typically exceeds actual demand by substantial margins. Actual demand is decided by the quality requirements of the conditioned space, the user pattern, thermal loads, design of the heating, ventilation and air-conditioning (HVAC) and energy supply systems. However, discussion and comparison of energy use and energy efficiency has little meaning unless one has defined the relevant system boundaries.

2.1 Building System Boundaries and Key Numbers

Key numbers for expressing the system energy efficiency will vary depending on purpose and application. This may be confusing, especially for systems using substantial amounts of unpaid-for energy such as direct solar and heat pump systems. Common “efficiency” measures are for instance $kWh_{heat}/m^2/year$ and $kWh_{el}/m^2/year$ (m^2 useable floor area).

The minimum use of heat and electricity is decided by the user demand specification. Real systems will always use more and, depending on whether one includes “free” energy flows or not, there may be widely differing figures for the supply of energy. Typically, building statistics only include purchased energy and hence is rather misleading concerning actual use. Figure 2 provides a simple illustration of energy flows through some alternative system boundaries and the difference between demand, use, net supply, gross supply and purchased supply.

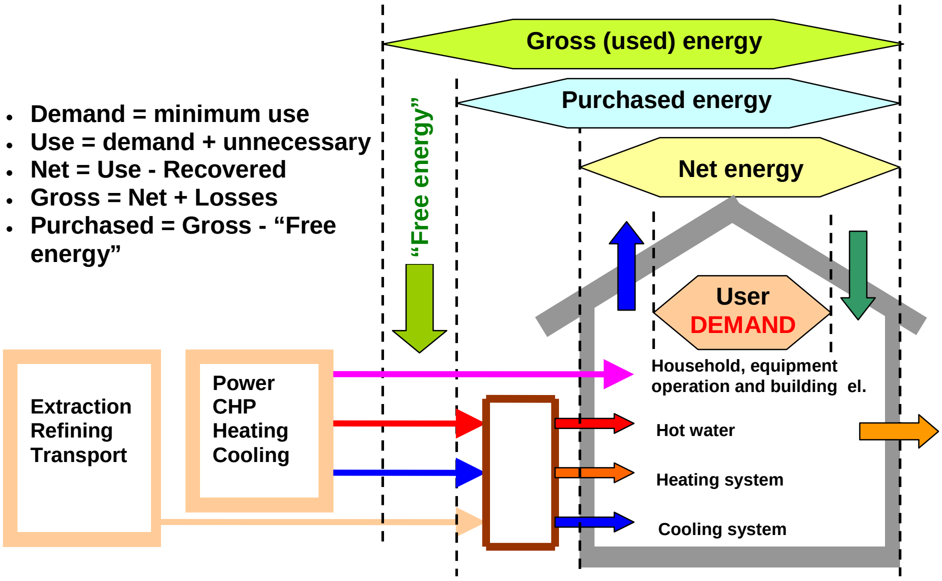


Figure 2: Alternative system boundaries used for efficiency key numbers.

2.2 Demand, Net Use, and Gross Supply in relation to the Load Factor

Looking for improvement of future heat pump heating and cooling systems, one must consider the ongoing developments of improving building envelopes, the use of storage to level out heat deficits and surpluses over time, the use of Building Automation Systems with feed-forward and control-on-demand (COD) to minimize supply in relation to demand etc. Such developments, COD in particular, will reduce the use of energy much more than it will reduce the design capacity. This, in turn, will substantially reduce the thermal load factor L_Q of the HVAC system and hence increase the demand factor D_Q. These factors are defined as:

$$L_Q = \frac{Q_0}{\tau_{year} \cdot \dot{Q}_{nom}} \quad \text{and} \quad D_Q = \frac{Q_{net}}{Q_0} \quad \text{(see figure 3)} \qquad (1)$$

where Q_0 = demand (minimum net use), $\dot{Q}_{nom}$ = nominal capacity and τ_{year} = 8760 h. Adding *h* or *c* to the subscripts will indicate heating or cooling. Demand factors may also be correspondingly defined for the ratio D_W between used drive energy, W_{net}, and the minimum net use, W_0.

It is a common experience that it is more difficult to exactly match supply and demand in systems with a low load factor. When a large part of the energy balance is supplied from uncontrolled heat flows (internal loads, heat stored in a building structure etc.), then it often happens that the conditioned space is overheated or overcooled in relation to demand. Based on simulations and measurements, the German standard VDI2067:20 was developed to describe this. Figure 3 shows an example of how the heat demand factor D_Q increases as the load factor decreases (it has the character of $D_Q = k_1 + k_2 / L_Q$ with k_1 and k_2 being constants). The diagram also indicates that the use of distributed pumps, as indicated in 3.5 and 3.6, will better match demand than one central unit.

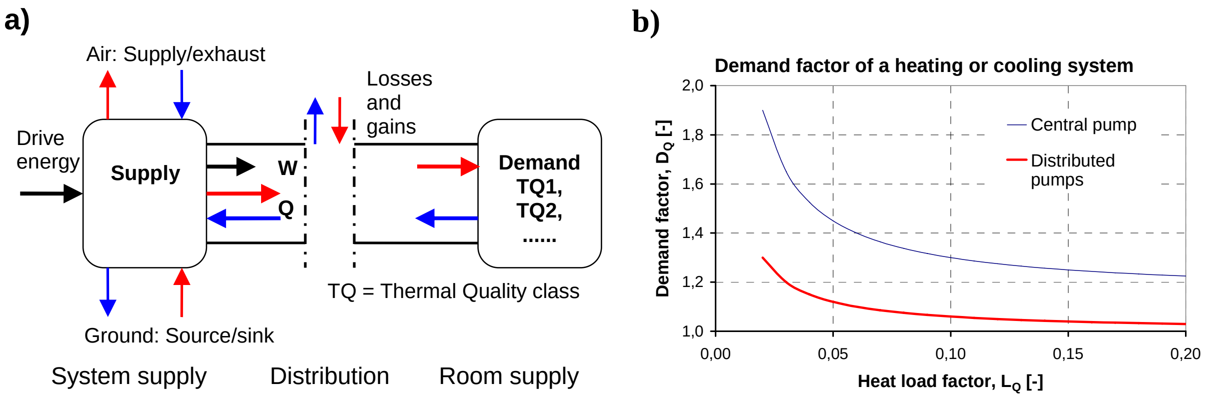


Figure 3: a) System supply, distribution and room supply of a general heat pump system. b) Demand factor of a heating or cooling system as a function of the heat load factor.

Overall use of energy can be reduced by means of reduced demand or improved efficiency. Demand may be tuned by reducing the heating or cooling loads or by changing the indoor environment specifications (e.g. less rigid minimum and maximum temperatures). Figure 3 indicates that there are three major parts of a heat pump heating and cooling system; system supply, distribution and room supply. These are the parts that affect system efficiency and hence will be discussed in this paper.

2.3 Heat Pump System Boundaries and Key Numbers

Key performance indicators of heat pump systems are the Coefficient of Performance (*COP*) and the Seasonal Performance Factor (*SPF*). Both are measures of the delivered heating or cooling energy of the system in relation to the demand for input drive energy. *SPF* is usually the integrated mean value of the instantaneous *COP* over a year, see eq. 2 below. Just as in the case with alternative system boundaries of a building, there will be different *COP*s and *SPF*s depending on the system boundary of the heat pump system. Figure 4 provides an illustration of alternative definitions of *COP*. Using the system boundaries and designations of figure 3 we have in the case of a heating application:

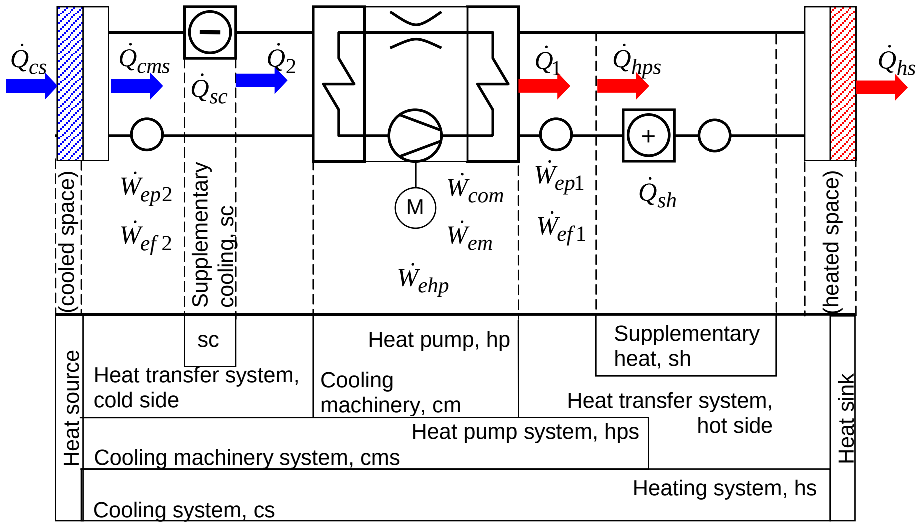
$$COP_{hp} = \frac{\dot{Q}_1}{\dot{W}_{e,hp}} \quad \text{and} \quad SPF_{hp} = \left[\frac{Q_1}{W_{e,hp}}\right]_{annual} \tag{2}$$

$$COP_{hps} = \frac{\dot{Q}_1 + \dot{W}_{e,p1}}{\dot{W}_{e,hp} + \dot{W}_{e,p1} + \dot{W}_{e,p2}} \quad \text{and} \quad SPF_{hps} = \left[\frac{Q_1 + W_{e,p1}}{W_{e,hp} + W_{e,p1} + W_{e,p2}}\right]_{annual} \tag{3}$$

$$COP_{hs} = \frac{\dot{Q}_1 + \dot{W}_{e,p1} + \eta_{sh} \cdot \dot{Q}_{sh}}{\dot{W}_{e,hp} + \dot{W}_{e,p1} + \dot{W}_{e,p2} + \dot{Q}_{sh}} \quad \text{and} \quad SPF_{hs} = \left[\frac{Q_1 + W_{e,p1} + \eta_{sh} \cdot Q_{sh}}{W_{e,hp} + W_{e,p1} + W_{e,p2} + Q_{sh}}\right]_{annual} \tag{4}$$

Corresponding definitions for cooling applications can be obtained by changing subscripts. However, in the heating case, the condenser pump adds to the heating capacity whereas in the cooling case, the evaporator pump detracts from the cooling capacity.

It is obvious from the definitions of SPF_{hp} and SPF_{hps} that the relative influence of the parasitic drive powers increases quickly when improvements regarding the heat pumping process reduces W_{ehp} ($W_{ehp} = W_{em}$ + controls etc.). In the equations 2-4, W_{ep1} is the sum of all parasitic drive powers to pumps and fans on the condenser side and correspondingly for W_{ep2}. Without specifying the system boundaries, comparisons of *COP* may be quite misleading (c.f. figure 9a where at +10 °C COP_{hp} = 3.7 and COP_{hps} = 2.5!).



$$W_{ep1} = \sum_{i=1}^{n} W_{ep1,i} \text{ and } W_{ep2} = \sum_{j=1}^{n} W_{ep2,j}$$

Original Swedish indices have been replaced.

1 = condenser side
2 = evaporator side

hp = heat pump
hps = heat pump system
hs = heating system
sh = supplementary heating system

cs = cooling system
cms = cooling machinery system

e = electric
m = motor
f = fan
p = pump

Figure 4: Heat pump system boundaries based on NT VVS 076, 115 and 116 (Fahlén, P. 1996).

3 SYSTEM DESIGN AND OPERATION

Prior to the discussion of system design and operation, it is obvious that the Thermal Quality (TQ) requirements of the conditioned space must have been decided on. These will set the performance requirements of the technical systems in terms of temperature levels, control deviations and thermal capacities. It then remains for the designer to minimize purchased energy by reducing the demand factors through *load matching* (COD), by maximizing the use of free energy (minimize W_e) through storage and the use of *natural sources and sinks*, and to minimize the *drive power* for heat transfer and distribution.

3.1 Load Matching by Control-on-Demand (COD) and Storage

The heat balance of modern residential and office buildings to a large extent relies on removal of internal heat loads by means of ventilation air. With heat recovery, very little supplementary heat is needed and maximum capacity for cooling will also be reduced (c.f. figure 5a). Central supply units can only handle the building as a total whereas actual demand is created at room level. Individual COD can drastically reduce demand but will also reduce the thermal and drive energy load factors L_Q and L_W. Hence load matching becomes important, otherwise the demand factors D_Q and D_W will become unnecessarily high. For a heat pump system with a given design capacity there are two main possibilities:

- Adapting the supply capacity to match demand (capacity control)
- Adapting the HVAC system to accommodate any excess capacity (storage).

Capacity control: With perfect load matching, the heat pump and heating system operating times will be equal, $\tau_{hp} = \tau_{hs}$ (see eq. 6; correspondingly for cooling). With continuous capacity control of the compressor, the operating time τ_{hp} for the heat pump will increase if a given amount of heat *Q* is to be delivered at a lower mean capacity. From eq. 6 it is obvious that if τ_{hp} goes up, then the drive <u>energy</u> for p11 and p12 will increase unless the respective drive powers are reduced. Karlsson (Karlsson, F. 2007) has shown that controlling the condenser and evaporator pumps linearly in relation to the controlled drive power to the compressor comes very close to an optimal control (see also 3.4).

Storage: With active storage, the same principles apply as described above. In this case, load matching is transferred from a variable capacity compressor to a variable capacity pump that controls thermal output from the storage. In both alternatives there will be parasitic losses of, for instance, frequency inverters and pumps. In the case of storage there will also be thermal losses. Active storage has a general potential for levelling of surplus and deficit (spatial as well as temporal redistribution). Passive storage in the building structure is a more complex issue.

3.2 Use of Natural Sources and Sinks

Obviously the use of free-cooling relies on a natural ambient sink with a temperature that is lower than the required conditioned space temperature (air, lake, sea, ground, stored snow etc.). Mattsson and Malmberg (Mattsson, C.-J. and Malmberg, T. 2008) have demonstrated how the design temperatures of cooling systems affect possible use of free cooling. A case-study in Gothenburg indicates that by designing for a supply cooling water temperature of 14 °C instead of the traditional 7 °C, it would be possible to reduce the annual drive energy for heat pumping from 19.2 to 1.5 kWh_{el}/m^2 using a nearby river as the sink. The reduction is partly due to increased evaporator temperature but mainly from a substantial increase of direct free-cooling. In the example of 4.1, the ground is used as combined storage, heat source and heat sink for a heat pump based heating and cooling system. In this case the fractional supply by free cooling could have been further increased with a different HVAC system.

3.3 Temperature Lift - Drive Energy for Heat Pumping

In the discussion of drive energy to a heat pump heating and cooling system, one should distinguish between drive energy for temperature lift (heat pumping), drive energy for heat transfer between heat pump and the sink or source, drive energy for heat transport and drive energy for heat transfer in the terminal units for air-conditioning. This facilitates the understanding of where savings on drive energy are most easily accomplished.

The temperature levels of the heating system - heat source and cooling system - heat sink will decide the possible *COP* and thus the drive energy for temperature lift. Assuming a constant Carnot efficiency of the heat pumping process, within limited temperature variations ΔT, we have:

$$\frac{\Delta COP_{hp}}{COP_{hp}} = \frac{\Delta COP_{1C}}{COP_{1C}} = \left[\frac{\Delta T_2}{T_1 - T_2} - \frac{T_2}{T_1} \cdot \frac{\Delta T_1}{T_1 - T_2} \right] \approx - \frac{\Delta \dot{W}_{e,hp}}{\dot{W}_{e,hp}} \tag{5}$$

Typically *COP* improves by 2-3 % for each degree of reduced temperature difference $T_1 - T_2$ between condenser and evaporator (relatively more so the smaller the difference becomes). Typical ways of reducing this difference is finding better sources or sinks and upgrading the terminal units by improving their heat transfer capacities. Regarding the latter alternative, it is very often the most cost effective method but it should be noted that there may be a drive power penalty if this requires higher flows and/or pressure drops. However, it pays to increase the drive energy to pumps and fans as long as the corresponding reduction of drive energy to the compressor is larger (see 3.4).

3.4 Heat transfer - Drive Energy to Pumps and Fans

Optimal flow for heat transfer at the heat pump may be quite different from the optimal heat transfer flow in terminal units. Also, demand in terminal units may not coincide with supply, neither for clock time nor for duration. Therefore, it is useful to distinguish between flows and pressure drops for heat transfer of system supply, of room supply and for distribution.

Regarding parasitic drive energy for heat transfer in the condenser or evaporator of heat pumps, Granryd (Granryd, E. 1998) has shown that for optimum *COP* these drive powers should be related to the cooling capacity by fairly simple relations. His conclusion regarding refrigerating applications was that typically pumps and fans were sized for optimum capacity, not optimum efficiency. The former criterion will yield drive powers up to 8 times the optimal for efficiency. Supporting Granryd's findings, Karlsson (Karlsson, F. 2007), in his work on capacity controlled heat pumps, has further underlined the importance of controlling not only the compressor but also the pumps and fans. If not, *COP* is likely to actually decrease instead of the hoped for increase from, for instance, a variable speed drive (VSD).

To achieve the required heat transfer at a lower drive power penalty, new developments of laminar flow design of air-coils show great promise (Haglund Stignor, C., Fahlén, P. and Sundén, B. 2007). However, in the search for the optimal heat transfer flow and drive energy, one must also consider the effect on drive energy for distribution where an increased flow is a penalty with little or no benefit. Fahlén has described the possibilities of reducing substantially the distribution pressure drops by alternative system designs (see 3.5).

Equation 6 illustrates the importance of parasitic drive powers, both regarding maximum input but perhaps more important in terms of the operating times. The parasitic drive powers are separated in components that are heat transfer related, p11 and p21, and distribution related, p12 and p22 (the respective flows and operating times may be quite different; p22 may for instance be a pump for a recharging system = rcs of a ground storage).

$$\overline{COP}_{hs} = \frac{\int_0^{\tau_{hp}} (\dot{Q}_1 + \dot{W}_{e,p11}) \cdot d\tau + \int_0^{\tau_{hs}} \dot{W}_{e,p12} \cdot d\tau}{\int_0^{\tau_{hp}} (\dot{W}_{e,hp} + \dot{W}_{e,p11} + \dot{W}_{e,p21}) \cdot d\tau + \int_0^{\tau_{hs}} \dot{W}_{e,p12} \cdot d\tau + \int_0^{\tau_{rcs}} \dot{W}_{e,p22} \cdot d\tau} \tag{6}$$

Eq. 7 illustrates in a slightly different way how the heat pump system *COP* is reduced by W_{ep1} and W_{ep2}, more so the higher the heat pump *COP* (drive powers are multiplied by COP_{hp}).

$$COP_{hps} = \frac{\dot{Q}_1 + \dot{W}_{ep1}}{\dot{W}_{e,hp} + \dot{W}_{ep1} + \dot{W}_{ep2}} = COP_{hp} - \frac{\left(COP_{hp} - 1\right) \cdot \dot{W}_{e,p1} + COP_{hp} \cdot \dot{W}_{e,p2}}{\dot{W}_{e,hps}} \tag{7}$$

3.5 Heat Transport - Heat Loss and Drive Energy to Pumps and Fans

Efficient heat transport to or from the heat pump should minimize the parasitic losses of drive energy and heat. The effect of such losses is much more important in terms of energy than in terms of power due to the difference in operating times of heating and cooling systems and that of the heat pump (see eq. 6). Heikkilä (Heikkilä, K. 2007) noted in a comparison of alternative air-conditioning systems for office buildings that the drive energy for air distribution totally dominated the environmental impact. In another study, Haglund-Stignor (Haglund Stignor, C., Fahlén, P. and Sundén, B. 2007) discusses the importance of not unduly increasing the distribution flows, and hence the transport work, in the search for minimized supply and terminal unit work (the terminal unit in this case being the cooling coil of a display cabinet). Furthermore, Fahlén (Fahlén, P., Markusson, C. and Maripuu, M.-L. 2007) has indicated the possibilities of new system designs and points out that there are three basic ways of reducing the parasitic drive energy for distribution:

- Improve efficiency. Component development of pumps, fans, motors and motor drives.
- Reduce flow rate. System design, e.g. COD.
- Reduce pressure drop. System design (see for instance figures 5 and 6).

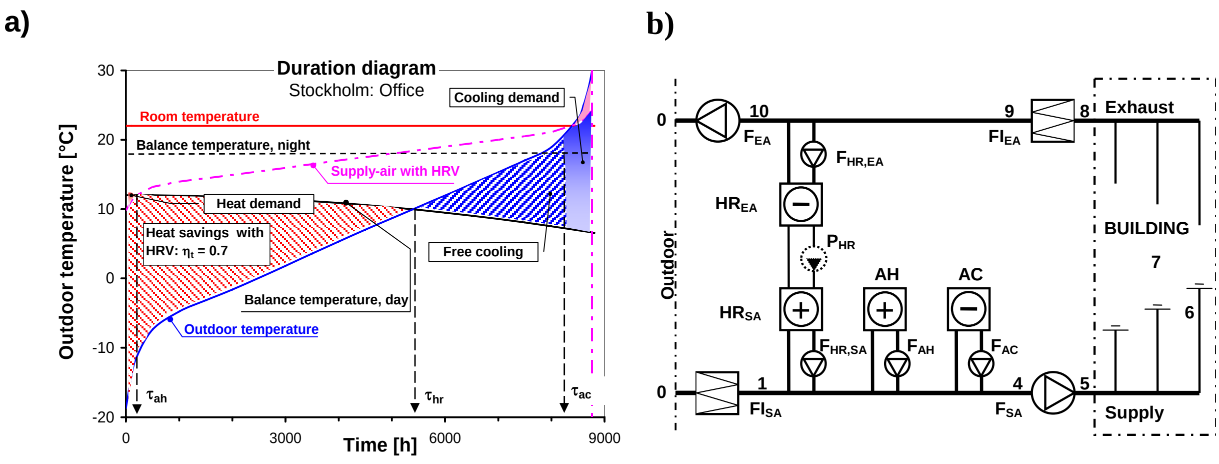


Figure 5: a) Duration diagram of the temperatures of outside air t_{oa}, supply-air t_{sa2} (after heat recovery) and the balance temperature t_{bt} for an office building.
b) Parallel design of an air-handling unit.

Figure 5a shows the duration diagram of outdoor and supply-air to an office building. Variable ventilation flows handle most of the space conditioning and the consequence is that the supply-air heater and cooler operate less than 2 % and 9 % of the time respectively. The rest of the time they constitute unwanted pressure drops. Going from an in-line to a parallel design of the air-handling unit is a possible solution to reduce parasitic drive power on the air-side (Fahlén, P., Markusson, C. and Maripuu, M.-L. 2007). Further gains are possible on the liquid side, see 3.6 below.

3.6 Terminal Units - Drive Energy to Pumps and Fans

Terminal units have a decisive influence on the overall efficiency of a heat pump heating and cooling system. The design and sizing will determine the temperature lift of the heat pump and the possibilities of using direct free-cooling (see 3.2 and 3.3). Cooling beams and actively controlled air-supply devices, which permit wide ranges of air flow rate and supply temperature, are such examples. In all cases, however, it is not sufficient to look only at the thermal side. One has to consider parasitic drive powers as well.

Figure 6 shows how new technology can simplify design and at the same time save on drive energy. By choosing direct flow control of a heater or cooler with a VSD pump instead of the traditional shunt group, the number of components goes down (no control or balancing valves) and the drive energy may be reduced by a factor 10 in COD systems. Examples of possible applications of direct flow control by decentralized pumps are, for instance, indirectly cooled display cabinets, supply-air coolers and heaters with integral pumps and fans (figures 5 and 6) and radiators or fan-coil units with integral VSD pumps.

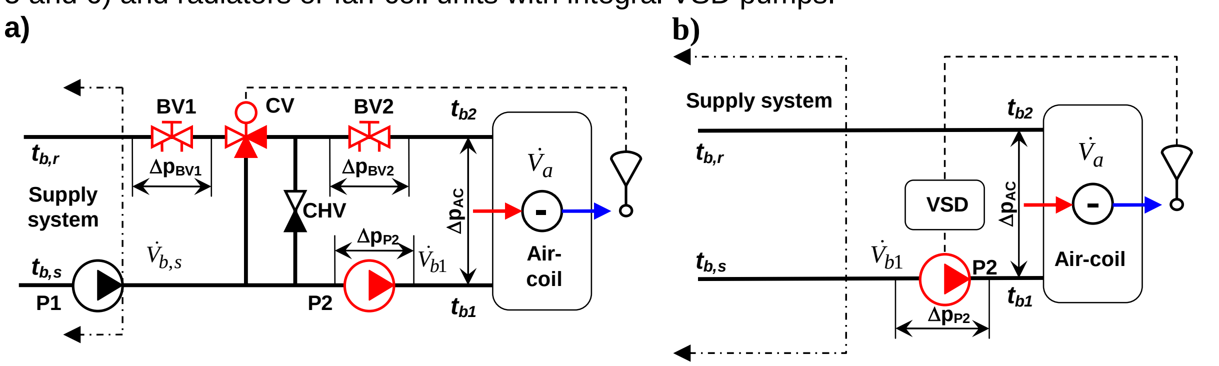


Figure 6: Control by means of a) variable inlet temperature and b) variable coil flow.

4 APPLICATION EXAMPLES

This section provides two examples of how free-cooling and new system designs may reduce the need for drive energy to electric motors of compressors, pumps and fans. The first is a case-study of a ground-source heat pump with combined heating and cooling including seasonal storage and the second a case-study of how the *SPF* of a standard ground-source heat pump can be upgraded.

4.1 An Office Building - Cooling and Heating Application

Space conditioning of office buildings nowadays requires cooling during a large part of the year, in daytime even at outdoor temperatures as low as -10 °C. Using the ground as heat source, heat sink or storage, in combination with a heat pump, is an energy efficient way of satisfying alternating or simultaneous demands for heating and cooling. In the heating mode, the heat pump cools the ground. After the heating season, the cold ground may be used for cooling simply by pumping brine through a heat exchanger in the building. This will heat the ground and at the end of the cooling season the ground temperature may not be sufficiently low. Then the heat pump may have to operate to reduce the brine temperature. Unless there is alternative use for condenser heat it will have to be wasted to the ambience.

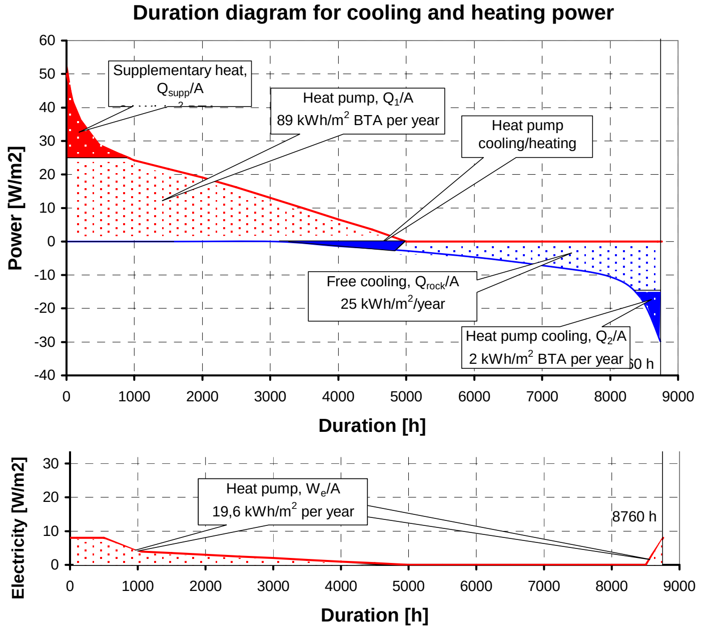
4.1.1 System Specifications

Building: Office building of 5300 m^2 at Lund University, Sweden.
Ventilation: VAV/CAV with heat recovery.
Heating and cooling: Radiators and low temperature supply-air.
Supply system: Heat pump with ground as source, sink and storage; supplementary heat by district heating.

Savings by reducing the purchase factor through heat recovery, variable ventilation flow, free-cooling with ventilation (outside) air and seasonal ground storage. No HVAC system modifications.

4.1.2 Results

Figure 5 provides measured results from February 2002 to February 2003.



Annual energy use	MWh	MWh/ m^2
Heating demand	515	97
Heating supply		
Heat pump	475	89
Supplementary	40	8
Cooling demand	155	29
Cooling supply		
Free cooling	130	25
Heat pump$^{1)}$	15	3
Heat pump$^{2)}$	10	2
Electricity		
Compressor	104	19.6
Pumps	7	1.3

1) Simultaneous heating and cooling
2) Cooling only

Figure 7: Results from a combined heating-cooling-storage system in Lund (Naumov, J. 2005).

4.1.3 Discussion

Measured results indicate that the parasitic ratio, $R_p = W_p/W_{hp}$, is close to optimal (7 %; see 3.4) for maximizing *COP*. As noted in 3.2, however, much greater use of direct free-cooling would have been possible with high supply temperature cooling water and chilled beams. This would also have resulted in higher brine temperatures during the winter heating season and hence less drive power to the heat pump for heating.

4.2 A Residential Building - Heating Only Application

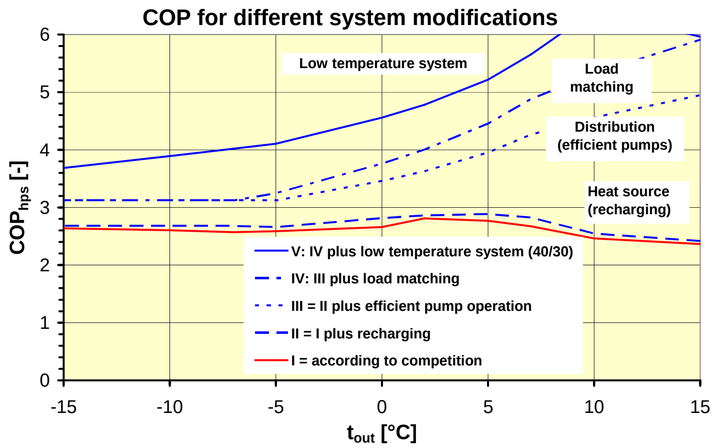
This single-family house was the reference house of the Nordic heat pump competition in 1995 and was retrofitted with the winning low-cost ground-source heat pump system (around 4600 € for heat pump, borehole, one fan-coil unit, one radiator and installation). The system has subsequently been upgraded with a number of modifications (see figure 8a). The order of the modifications differs in practice, c.f. figure 9a, from the planned actions of figure 8a.

4.2.1 System specifications

Building: Timber-framed, single-family house, 150 m^2 in Boras, Sweden.
Ventilation: Mechanical exhaust, CAV.
Heating and cooling: Direct-acting electric plus hydronic fan-coil and radiators (retrofit 1).
Supply system:
1) Heat pump with ground as source, heating only.
2) Recharging of the borehole from an exhaust-air heat recovery coil.
3) Storage tank for heating and hot water, new control system.
4) Addition of 4 more radiators.

Modifications aim to reduce the purchase factor through a combination of recharging the ground storage, load matching with a storage tank and better control, increased room heater capacity, and reducing the parasitic drive power ratio.

a)



b)

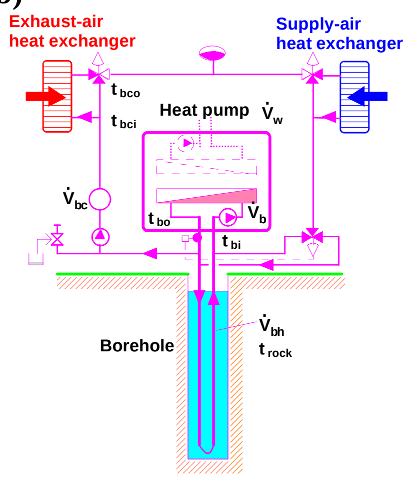


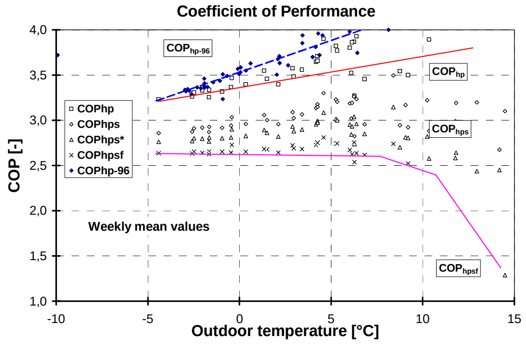
Figure 8: a) Predicted COPhps for alternative modifications of the original installation. b)Schematic of the recharging system with exhaust-air and supply-air heat exchangers.

4.2.2 Results

Figure 8a illustrates the effects of alternative modifications to the original system (I). Neither in theory (fig. 8a), nor in practice (fig. 9a) 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 in theory and 3.7 in practice. The latter figure, however, also includes hot water and also the pumps have not yet been upgraded.

Figure 9a 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.

a)



b)

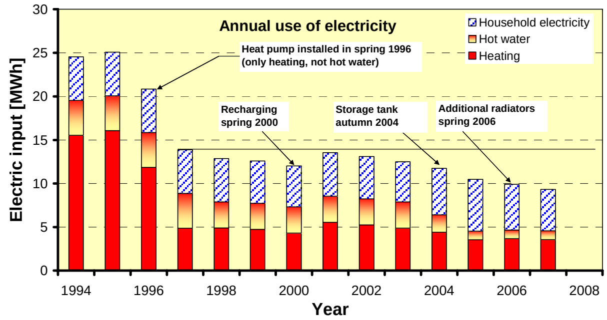


Figure 9: a) Measured values without and with recharging of COP_{hp}, COP_{hps}, and COP_{hs} respectively. b) Measured values of electricity for heating, hot water and household purposes.

4.2.3 Discussion

The results of figure 9a confirm the prediction in figure 8a 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 %). Figure 9a also illustrates the importance of defining the system boundary (c.f. figure 3). With or without pumps and fan makes a tremendous difference.

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 has been reduced from 25 to < 10 MWh and the specific purchase is now 67 W/m^2/year. This is lower than in most modern passive houses being built in Sweden with super insulation, high efficiency heat recovery and solar heat.

5 CONCLUSIONS

The efficiency of *heat pumps* has developed and continues to develop through a number of improvements of components in the refrigerant system such as compressors and compressor motors, condensers and evaporators, expansion devices, but also with new process concepts etc. However, future improvements on the efficiency of *heat pump systems* will to an ever increasing extent depend on component and system developments outside the heat pump per se. Some important conclusions from previous and ongoing research are that:

- Improved design of conditioned spaces such as buildings, display cabinets etc. will reduce the demand for heating and cooling.
- Demand reduction tends to be larger in terms of energy than in terms of power. This will reduce the system load factor.
- Reduced load factors will reduce the HVAC system efficiency and increase the importance of storage and capacity control for load matching.
- Improved system efficiency is possible by reducing the drive powers and heat losses for heat transfer and distribution.
- Distinction between heat transfer and distribution pressure drops facilitates an optimization of the flows and drive powers to pumps and fans (the relative operating times differ).

- New system designs with distributed pumps and fans can drastically reduce the HVAC system drive power (full control authority with no additional control pressure drops).
- Heat pump systems tend to have parasitic energy ratios much greater than optimal, in some applications over 50 %.

Summing up, it is possible by means of heat pump technique to drastically reduce purchased energy in numerous applications for space conditioning and hot water. This, however, requires attention to engineering details in order to fully benefit from the theoretical potential. Many of these details relate to the minimization of parasitic drive energy and heat loss.

6 NOMENCLATURE

	Symbols, Latin letters		
COP	coefficient of performance [-]	*T*	thermodynamic temperature [K]
D	demand factor [-]	$\dot{V}$	volume flow rate [m^3/s]
L	load factor [-]	*R*	ratio [-]
p	pressure [Pa]	*W*	work, mechanical or electric [J]
Q	heat [J]	$\dot{W}$	power, mechanical or electric [W]
$\dot{Q}$	power, thermal [W_{th}]		**Symbols, Greek letters**
SPF	seasonal performance factor [-]	Δ	difference, change (e.g. pressure, temp.)
t	celsius temperature [°C]	η	efficiency, temperature [K/K_{max}]

	Subscripts		
1	condenser	cm	cooling mach.
2	evaporator	cs	cooling sys.
a	air	hcs	heating and cooling system
ac	air-coil		
b	brine	hp	heat pump
e	electric	hps	heat pump sys.
f	fan	hr	heat recovery
m	motor	rcs	recharging sys.
out	outdoor	Q	heat
p	pump, parasitic	W	work

	Abbreviations
CAV	Constant Air Volume flow rate
COD	Control-On-Demand
HVAC	Heating, Ventilation and Air-Conditioning
TQ	Thermal Quality
VAV	Variable Air Volume flow rate
VSD	variable-speed-drive

7 REFERENCES

P. Fahlén, 1996. Field testing of refrigeration and heat pump equipment - General conditions. (SP-AR 1996:22). Boras, Sweden.

Fahlén P., C. Markusson, et al., 2007. Opportunities in the design of control-on-demand HVAC systems. (9th REHVA World congress Clima, 2007-06-10--14.) Helsinki, Finland.

Haglund Stignor C., P. Fahlén, et al., 2007. Design of different types of secondary loop cooling systems in supermarkets – Comparison of energy use and costs. (22nd International Congress of Refrigeration, 2007-08-21--26.) Beijing, China.

Heikkilä K., 2007. Environmental assessment of air-conditioning systems - Design considerations for Swedish conditions. (D2007:01, Chalmers.) Gothenburg, Sweden.

Karlsson F., 2007. Capacity control of residential heat pump heating systems. (D2007:03, Chalmers University of Technology, Building Services Engineering) Gothenburg, Sweden.

Malmberg T., C.-J. Mattsson, 2008. The effect of air-conditioning system design temperatures on the potential for using free-cooling (in Swedish). (E2008:02, Building Services Engineering, Chalmers University of Technology.) Gothenburg, Sweden.

Naumov J., 2005. Optimization of ground-coupled systems for heating and cooling of buildings. (D2005:04, Building Services Engineering, Chalmers.) Gothenburg, Sweden.