

HIGH EFFICIENT HEAT PUMPS FOR SMALL TEMPERATURE LIFT APPLICATIONS

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Abstract: Today, one of the most promising type of heating system for achieving significant primary energy savings in heating of buildings is a heat pump. Their efficiency as measured by the coefficient of performance (COP) is closely related to the inner temperature lift, i. e. the temperature difference between condensation and evaporation. The potential for highly efficient systems is not fully exploited as standard heat pumps are designed for large lifts of 30–60 K. The integration of small temperature lift heat pumps in efficient overall heating systems leads to a considerable reduction in the use of primary energy and operating costs. The development of efficient heat pumps for small temperature lifts requires a new approach in the design and selection of the components. In order to demonstrate the improved efficiency, a system for small internal temperature lifts was developed and tested. By using geothermal heat probes and efficient heat delivery and supply systems a COP above 9 can be achieved for a temperature lift of 20 K. The heat pump can also be employed as a highly efficient chiller for building cooling. For an evaporation temperature of 15°C, the cooling COP is 11.5 at a temperature lift of 12.5 K.

Key Words: heat pumps for small temperature lift, guidelines for the planning and design, experimental investigations, improvement in efficiency

1 INTRODUCTION

1.1 Background and objectives

There are many high demands placed on the design of modern residential and office buildings. Investors and tenants have high standards regarding the quality of use and expect low operating costs for building services, in particular for heating, cooling and ventilation. However, presently little research in this area of primary energy demand as well as the development of highly efficient systems and implementation of practical applications has been undertaken. Any efficiency improvement measures should not result in deterioration of comfort. This goal can only be achieved if an optimal solution is strived for that integrates the branches of architecture, building physics and building services engineering (Wellig et al. 2006).

The goal of this project is the development and experimental study of heat pumps that require significantly lower power consumption. This goal is achieved through the consequent use of small temperature lifts. The project aims at building the foundation in the area of small temperature lift equipment upon which motivation for future developments in industry can be raised to a new level. The findings can also be used for designing highly efficient chillers.

1.2 Temperature lifts for heating and cooling

The efficiency of heat pumps and chillers is strongly influenced by the temperature lift. In compression heat pumps the inner temperature lift corresponds to the temperature difference between the condensation and evaporation temperature.

Heating: Dependent on the heating system as well as the heat source a certain temperature lift is necessary for heating. In geothermal systems, the temperature of the undisturbed soil at a depth of 5 meters is approximately 10°C and at 300 m around 19°C (slope around 0.03 K/m). This permits relatively high evaporation temperatures of up to 10°C when relatively deep geothermal probes (e.g. two-zone probes) are combined with properly sized evaporators. Other potential heat sources for small-lift applications include ground, river or lake water. Also the use of wastewater or the building's exhaust air as "high-temperature" heat sources is steadily increasing.

The heat distribution systems most commonly used are underfloor floor heating or radiators. The temperature lift in most buildings, depending on the chosen heating system, is typically around 20–60 K. For an efficiently heated building a temperature lift in the range of 20 K to 35 K is to be strived for. This refers to e. g. a geothermal probe at 300 m depth and modern low-temperature heating system. Figure 1 (left) shows the COP_{ideal} of an ideal heat pump as a function of the temperature lift for a constant evaporation temperature of 10°C. For a temperature lift of 20 K a COP_{ideal} of 15.2 is possible. If a system with a Carnot efficiency of 50% would be employed, a COP of 7.6 would result (the Carnot efficiency is the effective COP divided by the COP_{ideal} operating between the corresponding cycle temperature levels).

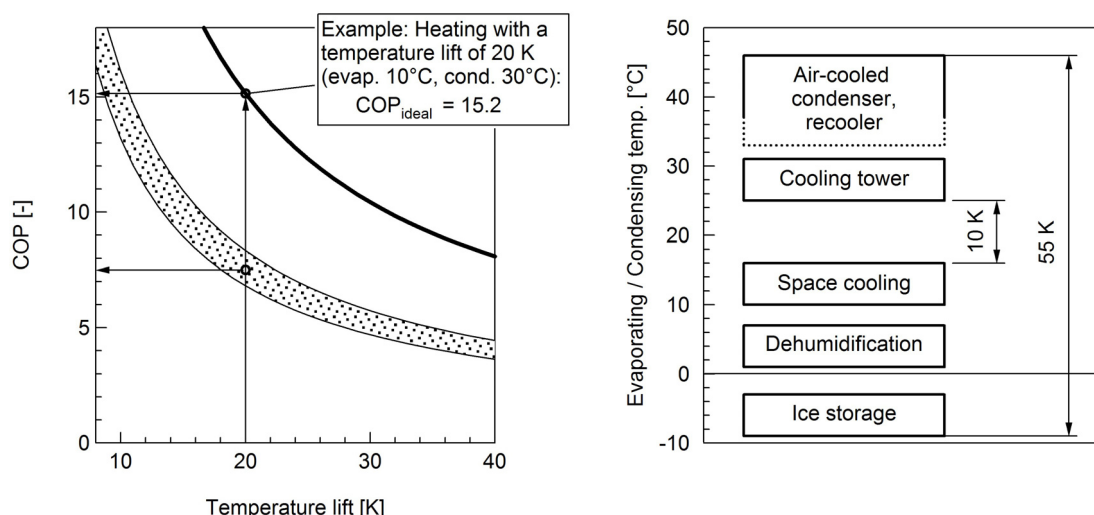


Figure 1: Left: COP of an ideal heat pump and a real heat pump with a Carnot efficiency of 50% as a function of the temperature lift for a constant evaporation temperature of 10°C.

Right: Typical evaporation and condensation temperatures and temperature lifts of different building cooling systems.

Cooling: Figure 1 (right) shows typical temperature lifts for various building cooling systems. The lift varies depending on the system between 10 K and 55 K as extreme examples. Standard chillers are designed for internal temperature lifts of about 20–60 K. For the cooling of buildings, e. g. with a chilled ceilings, concrete core activation or efficient fan coil units combined with efficient hybrid cooling systems, a lift of between 10 K and 20 K is considered possible. However, even though the temperature difference between the heat source and the sink is small, the potential for highly efficient cooling processes is hardly used. The reason is that conventional cooling systems require low pump cold water temperatures that are provided by the cooling system and that dry coolers are used. Typically, the chilled water temperatures (CWT) range from 6°C to 12°C, resulting in a low evaporation temperature. In addition, due to the head pressure control the condensation temperature is maintained at a high level (35–45°C), (Kreider 2001, Stanford 2003). Consequently, conventional building cooling systems often work with an unnecessarily high temperature lift.

Heat pumps and building cooling systems at small temperature lifts only achieve a high efficiency ratio when they are designed specifically for the appropriate operating conditions with small temperature lifts.

2. HEAT PUMP PROTOTYPE FOR SMALL TEMPERATURE LIFTS

2.1 Design

Standard heat pumps are designed for internal temperature lifts of about 20–60 K. This leads to relatively poor efficiencies if they are operated at smaller temperature lifts. In order to demonstrate the potential for improvement in efficiency, a heat pump for small temperature lifts was developed and tested. Therefore, all components of the heat pump were specially designed (Wyssen et al. 2010).

A prerequisite for the development of efficient heat pumps for small internal temperature lifts is to use a new approach in the selection of the expansion device. Using common thermostatic expansion valves, it is difficult to achieve vapour superheating in the evaporator less than 7 K. For efficiency reasons it is advisable to reduce the vapour superheating in the evaporator as much as possible. For this reason the developed heat pump prototype was equipped with an electronic expansion valve instead of a thermostatic one.

The selection of the size of the expansion valve should not be done according to the heating capacity of the heat pump. The main selection criterion is the maximum permissible pressure drop in the expansion valve in a fully open position (figure 2). The pressure loss in the fully opened expansion valve $\Delta p_{Ex\ open}$ for a given operating condition has to be smaller than the difference between the condensing and evaporation pressure reduced by the pressure losses occurring in the evaporator and condenser as well as in piping and fittings Δp_{Ex} , ($\Delta p_{Ex\ open} < \Delta p_{Ex}$). Thereby, the operating condition of the heat pump with the highest evaporation temperature and the smallest temperature lift (lowest pressure ratio and highest working fluid mass flow) is decisive. If the pressure loss in the fully opened expansion valve is greater than the difference between the condensing and evaporation pressure, the desired operating condition cannot be reached. The resulting temperature lift will be higher than the one striven for.

This selection criterion leads to oversized expansion valves with respect to the evaporation capacity denoted by the manufacturers. The evaporation capacity of the developed heat pump prototype is approximately 14 kW (maximum value for small-lift building cooling with evaporation 15°C and condensation 23°C). According to the manufacturer the applied expan-

sion valve is designed for evaporation capacities of about 40 kW. The manufacturer's data are valid for standard systems with temperature lifts of about 30–60 K.

A further prerequisite for high efficiency heat pumps with small temperature lifts is the selection of suitable compressors. It is critical to avoid the use of compressors with a high internal pressure ratio, such as many scroll compressors. For this reason the heat pump prototype is equipped with a common constant speed semi-hermetic, cold gas cooled reciprocating compressor.

The evaporator and condenser are designed as plate heat exchangers. The dimensioning has been carried out in collaboration with the industrial partner with the aim to reduce the required temperature differences for heat transfer as much as possible.

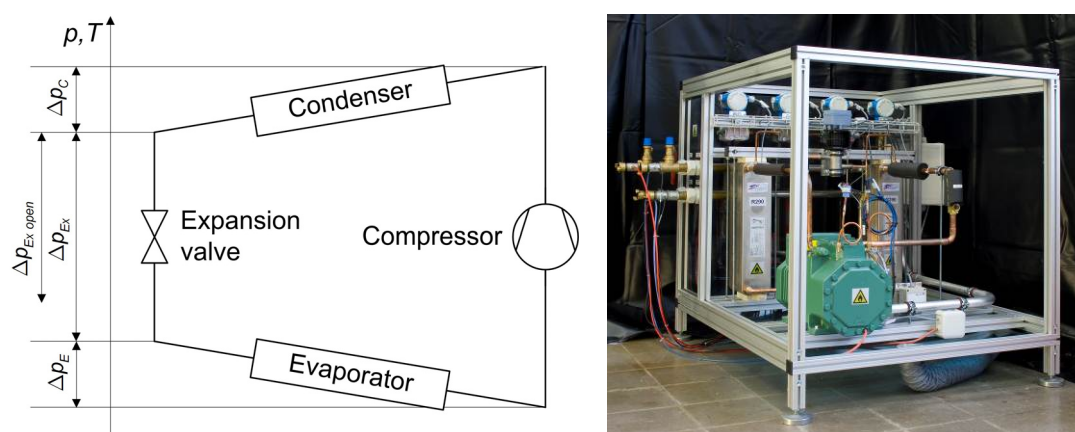


Figure 2: Left: Schematic diagram of the working fluid circuit of the heat pump with pressure losses in the evaporator and condenser. Also shown is the required drop in pressure of the expansion valve as well as the resulting pressure loss in the fully opened expansion valve (see text).

Right: Heat pump prototype for small temperature lifts.

The developed heat pump prototype for small internal temperature lifts uses propane (R290) as working fluid. It does not have any ozone depletion potential (ODP) and a negligible direct global warming effect (GWP). There are no particular material problems using propane as working fluid. The pressure levels and the heating capacity are similar to R22 and R502. A further advantage of propane is its favourable temperature behaviour (Bitzer 2008). This means that in the compressor relatively small vapour superheating occurs with accordingly low discharge gas temperatures. The vapour and liquid densities of propane are small. This leads to high volumetric heating capacities and low working fluid mass flows. As a result of low working fluid mass flows it is possible to minimise pressure losses due to flow in piping and fittings as well as in the evaporator and condenser on the working fluid side. Furthermore, propane ensures relatively high heat transfer coefficients in the evaporator and condenser.

2.2 Experimental Setup

The heat pump was integrated into a testing facility that provides fluid flows with adjustable temperature levels and mass flows and simulates the heat source and the heat sink at the same time. This testing facility is versatile and can be employed in different ways for examining different heat pump systems. This testing facility is described elsewhere in detail (Gasser et al. 2011).

The heat pump is equipped with very precise electric power, temperature, pressure and volume flow measuring devices to retrieve all relevant process data. These process data enable a detailed and reliable energetic and exergetic analysis at different operating conditions. The heat flows in the evaporator and the condenser can be calculated by the energy balances on the side of the heat source and heat sink.

2.3 Procedure

The approach for characterising the heat pump was to adjust the supply flow temperature of the heat source in order to set a constant evaporation temperature. The mass flow of the heat source was controlled by the temperature spread between supply flow and return flow temperature to be 5 K. Then, in a reasonable range between 55°C and 20°C, the condensation temperature was lowered in predefined steps simulating different heating systems. To reach the targeted condensation temperatures, the return temperature of the heating water was controlled. The temperature spread between supply and return water was kept constant at 6 K by varying its mass flow. The electronic expansion valve controlled the superheating to 5 K. The subcooling was not controlled and decreased with decreasing temperature lift from 5.5 K to 1.3 K.

Below, the measured and calculated data of 4 working points with an evaporation temperature of 10°C are shown in order to enlighten the important relations.

3. RESULTS

3.1 Energy analyses

The condensation temperature decreases with lowered temperature of the heat sink supply flow. The condensation pressure decreases with decreasing condensation temperature. Since the evaporation temperature and therefore the evaporation pressure are kept constant, the pressure ratio respectively the temperature lift becomes lower if the condensation temperature decreases.

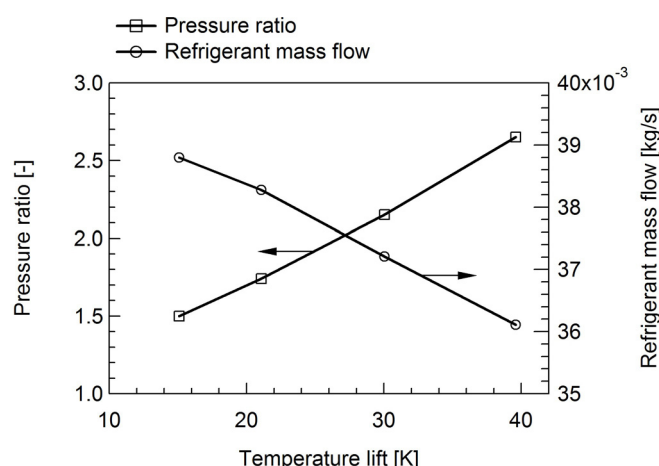


Figure 3: A decreasing pressure ratio entails an increasing mass flow. The diagram is valid for the examined compressor and the working fluid R290 for a constant evaporation temperature of 10°C.

With a decreased pressure ratio the compressor delivers more mass flow. Figure 3 shows this quantitative relation that can be considered as the reciprocating compressor's

characteristic curve. This curve is only valid for the employed compressor, the working fluid R290 and the present temperatures within the heat pump prototype. For higher temperature lifts and therefore smaller mass flows, the heating capacity decreases.

Figure 4 (left) shows the relation of the condenser and the evaporator capacity and also the corresponding consumption of electrical power of the compressor dependent on the above described inner temperature lift. The figure shows that smaller temperature lifts correspond to higher capacities both of the condenser and the evaporator. Smaller temperature lifts result in smaller pressure ratios and therefore in lower power consumptions of the compressor.

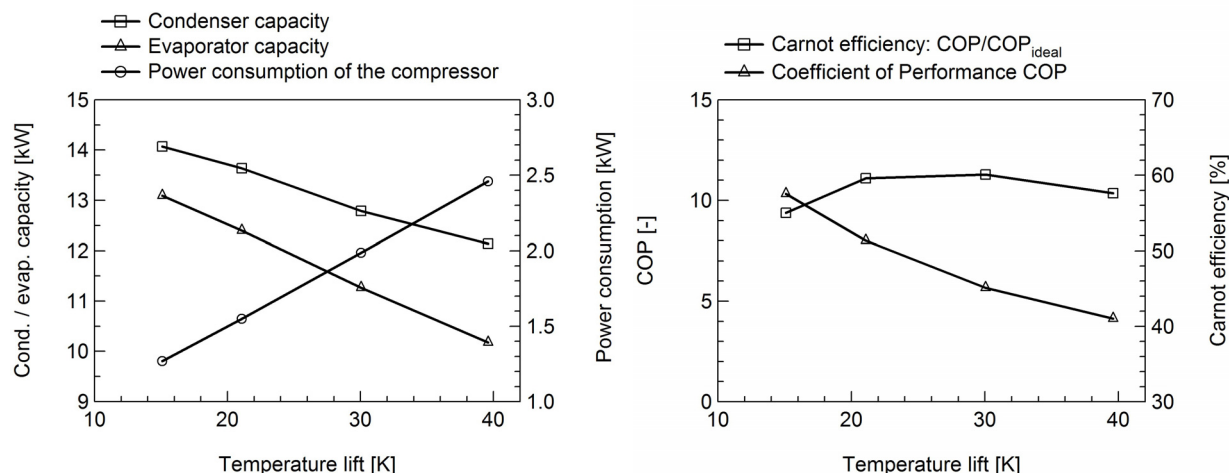


Figure 4: Different characteristics as a function of the inner temperature lift (difference between condensation and evaporation temperature) for a constant evaporation temperature of 10°C. Left: Condenser capacity, evaporator capacity and consumption of electrical power of the compressor. Right: COP and Carnot efficiency.

The COP (ratio between heating capacity and consumption of electrical power of the compressor) and for the heating mode is shown in figure 4 (right) and decreases with increasing temperature lift. For a temperature lift of about 40 K the COP is 4.1 and for about 15 K the COP is 10.3, i. e. that the reduction of the temperature spread by 25 K the COP can be increased by about 150%.

An important characteristic ratio for the energetic evaluation is the Carnot efficiency ($\text{COP}/\text{COP}_{\text{ideal}}$), shown in figure 4 (right) that characterizes the inner heat pump process. The Carnot efficiency stays on a high level of 50% to 60% and is for low temperature lifts well above the Carnot efficiencies of standard heat pumps and compression chillers (Wellig et al. 2006). The curve shows that the chosen reciprocating compressor suits well for applications with a small temperature lift.

The isentropic, compressor and the volumetric efficiency are shown in figure 5. The compressor efficiency includes the electrical and mechanical losses of the compressor (i.e. ratio between internal power and consumption of electrical power of the compressor). The compressor efficiency decreases with increasing temperature lift and stays in the range between 94% and almost 100%. The volumetric efficiency also decreases with increasing temperature lift. For a lift of about 21 K the volumetric efficiency is 90%, for a temperature lift of about 40 K the efficiency is 82%. The isentropic efficiency increases with increasing temperature lift. For a lift of about 21 K the isentropic efficiency is 64%, for a temperature lift of about 40 K the efficiency is 73%.

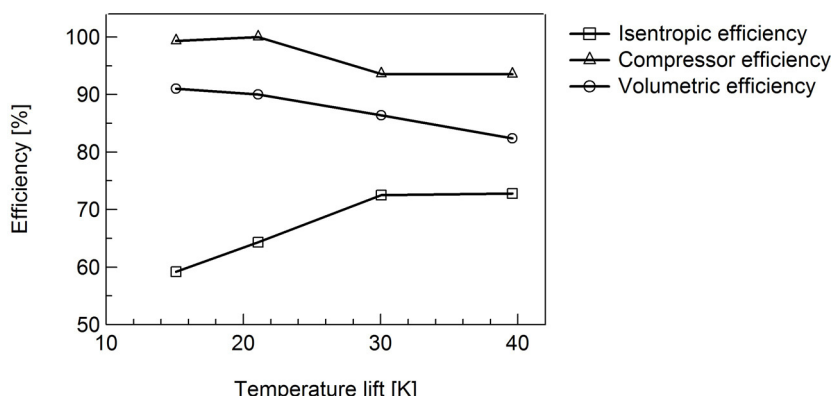


Figure 5: Isentropic, compressor and volumetric efficiency as a function of the inner temperature lift for a constant evaporation temperature of 10°C.

3.2 Exergy analyses

The COP and the Carnot efficiency do not provide any information about the various losses, or rather where they occur and how large they are. Detailed information about the losses can be achieved by using the exergy analysis or by using the more abstract entropy balances (Wellig et al. 2006). Gasser et al. show mathematical equations to quantify the exergy losses of the individual sub-processes of heat pumps and compression chillers (Gasser et al. 2008). The results, see table 1, are based on this mentioned calculation method and are valid for the above mentioned operating point at 10°C evaporation temperature and 21 K temperature lift at 0°C ambient temperature.

The exergy losses \dot{E}_L in the heat pump are: \dot{E}_{LCp} in the compressor, \dot{E}_{LC} in the condenser, \dot{E}_{LEx} in the expansion valve and \dot{E}_{LE} in the evaporator. The exergy losses in the compressor and the expansion valve are caused by dissipative effects. However, the exergy losses during heat transfer in the evaporator and condenser do not originate in the working fluid but are caused by the temperature gradients required for heat transfer (Gasser et al. 2008).

The exergy losses in the evaporator and condenser are not dependent on the temperature lift. Therefore, they cannot be reduced by small temperature lift applications. The losses can only be minimised by reducing the temperature gradients required for heat transfer as a result of an optimal dimensioning of the evaporator and condenser. About 32.5% of the electrical power consumption (exergy) is lost in the compressor. The exergy loss in the compressor is approximately proportional to the specific heat of evaporation of the working fluid and temperature lift and inversely proportional to the isentropic efficiency of the compressor.

The exergy loss in the expansion valve is proportional to the square of the temperature lift. For this reason, the exergy loss in the expansion valve is only about 6% of the consumption of electrical power of the compressor at a temperature lift of 21 K. In total, 888 W, respectively around 57.3% of the electrical power consumption of 1550 W of the heat pump is destroyed and irreversibly converted to anergy, this is the total exergy loss of the heat pump \dot{E}_{Ltot} .

Table 1: Exergy losses of the sub-processes of the heat pump and exergetic efficiency.
Operating point: Evaporator: water flow 2558 kg/h, water inlet/outlet temperatures 15.7/11.5°C, evaporation temperature 9.8°C, superheating 5.1 K, capacity 12.40 kW; **Condenser:** water flow 1798 kg/h, water inlet/outlet temperatures 24.4/30.9°C, condensing temperature 30.9°C, subcooling 2.2 K, capacity 13.64 kW; refrigerant mass flow $\dot{m}_f = 0.038$ kg/s; **Compressor:** isentropic efficiency $\eta_s = 64.3\%$, electric power consumption $P_{el} = 1.55$ kW, inner temperature lift $\Delta T_{Lift} = 21.1$ K ($\Delta h_v =$ specific heat of evaporation, $c_{pl} =$ specific heat capacity of liquid refrigerant).

Components	Exergy loss	Exergy loss ratio
Compressor	$\dot{E}_{LCp} \approx \dot{m}_f \cdot \Delta h_v \cdot \frac{1}{COP_{rev i}} \cdot \left[\frac{1}{\eta_s} - 1 \right] = 503 \text{ W}$	$\frac{\dot{E}_{LCp}}{P_{el}} = 32.5\%$
Expansions valve	$\dot{E}_{LEx} \approx \dot{m}_f \cdot c_{pl} \cdot \frac{\Delta T_{Lift}}{2 \cdot COP_{rev i}} = 93 \text{ W}$	$\frac{\dot{E}_{LEx}}{P_{el}} = 6.0\%$
Evaporator	$\dot{E}_{LE} \approx \dot{Q}_E \cdot \frac{\Delta T_E}{T_A} = 160 \text{ W}$	$\frac{\dot{E}_{LE}}{P_{el}} = 10.3\%$
Condenser	$\dot{E}_{LC} \approx \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H^2} = 132 \text{ W}$	$\frac{\dot{E}_{LC}}{P_{el}} = 8.5\%$
Total	$\dot{E}_{Ltot} = 888 \text{ W}$	$\frac{\dot{E}_{Ltot}}{P_{el}} = 57.3\%$
Exergetic efficiency		$\eta_{ex} = 42.7\%$

If the exergy profit is referred to the input exergy, the exergetic efficiency is obtained. The exergy profit is the exergy of the condenser heating capacity. The input exergy corresponds to the electrical power consumption of the compressor P_{el} .

$$\eta_{ex} = \frac{\dot{E}_{QC}}{P_{el}} = 1 - \frac{\dot{E}_{Ltot}}{P_{el}} \quad (1)$$

At the above mentioned operating point at 10°C evaporation temperature and a temperature lift of 21 K, the exergetic efficiency of the heat pump is 42.7% and therefore clearly above the values of standard heat pumps.

3.3 The heat pump in field application

If small temperature lift heat pumps with geothermal heat probes as heat source are used for modern low-temperature heating systems high COPs and seasonal performance factors (SPFs) can be achieved. This relation is shown by the example given below.

The real building to be heated, see figure 6 (left), is equipped with a low temperature floor heating system that requires a supply temperature of about 28°C at an ambient temperature of 0°C. The corresponding condensation temperature of the small temperature lift heat pump is around 30°C. The two-zone geothermal heat probe has a depth of about 300 m and the resultant evaporation temperature is around 10°C (water 16/12°C). The resulting temperature lift of the heat pump is 20 K.

For these operating conditions a COP of 9.2 and a Carnot efficiency of 60% result. The external exergetic efficiency of the heat pump amounts to around 80% for an ambient temperature of 0°C. The exergy losses, especially in the compressor and the expansion valve, can be reduced due to the small temperature lift. Compared to this, the exergetic

efficiencies of standard heat pumps are often only in the range of 30–50% (Gasser et al. 2008)

If both circulation pumps (geothermal heat probe and heating system) are taken into account, the coefficient of system performance (COSP) is 8.4.

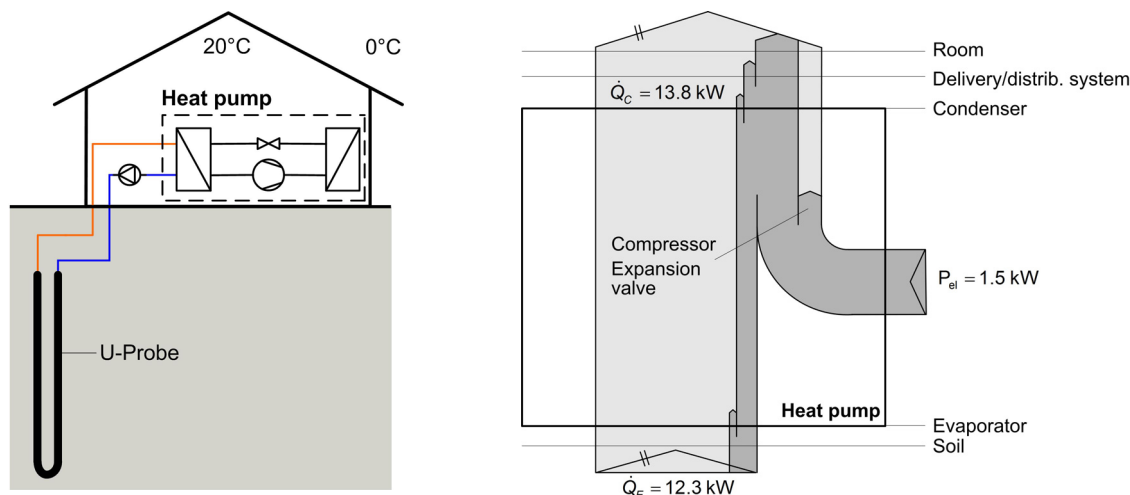


Figure 6: Left: Schematic of a heat pump with geothermal heat probe
Right: Energy and scale exergy flow chart of the small temperature lift heat pump: heating capacity 13.8 kW, power of the compressor 1.5 kW, COP = 9.2, COSP = 8.4 (including both pumps), $\eta_{ex} = 80\%$, ambient temperature 0°C.

A current example is the project B35 (Meggers et al. 2010) that aims at reducing the CO₂ emissions of a building's operation down to zero. To reach this goal, inter alia, an efficient heating system is necessary. Through the application of a two-zone geothermal heat probe with depths of 150 respectively 300 m and an efficient heat delivery system, a temperature lift of about 16 K is to be expected. By the employment of a heat pump, especially designed for small temperature lifts, a COP around 10 and a COSP above 9 can be achieved.

3.4 Use as chiller

The application of the small temperature lift heat pump to building cooling also provides a great potential for efficiency improvements. This can be clearly illustrated through the comparison based on a building cooling system in an office building in Zurich, see figure 7 (Wyssen et al. 2010). The employed building cooling system is very efficient with high chilled water temperatures (CWT), low recooling temperatures and a high free-cooling rate. The cold supply in the rooms is done by using efficient fan coil units with CWTs of 16–20°C. The recooling loop is equipped with a wet cooling tower where the approach to the wet bulb temperature is 1–2 K. In principle, the cooling takes place exclusively by means of free cooling. Once the wet bulb temperature or the required cooling capacity make an efficient free-cooling operation impossible, the chiller is put into operation, see figure 8 (left).

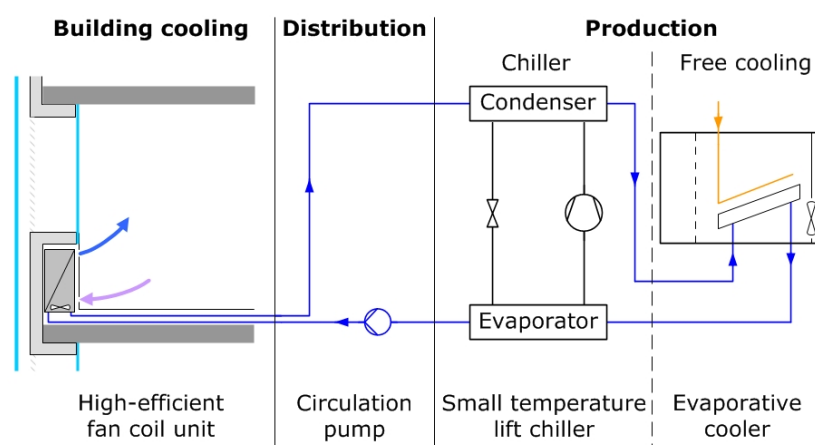


Figure 7: Building cooling system in an office building in Zurich using a small temperature lift chiller.

With a chiller, optimized for small temperature lifts, a relatively high evaporation temperature of 12.6°C and a relatively low condensation temperature of 25.1°C are sufficient to reach the CWT. The resulting inner temperature lift of 12.5 K is far smaller than that for standard cooling systems. For a cooling capacity of 23 kW and a compressor power of 2 kW , this corresponds to a COP of 11.5 . If the auxiliaries (circulation pump, fan of recooling and fan coil units) are taken into account the COSP is 7.2 according to measurements. For standard systems, unnecessarily low evaporation temperatures and unnecessarily high condensation temperatures are required. It is well known that COSP values of standard building cooling systems are mostly below 3 (e. g. Wellig et al. 2006). As a result, the optimized building cooling system, designed for small temperature lifts, exhibits higher COPs.

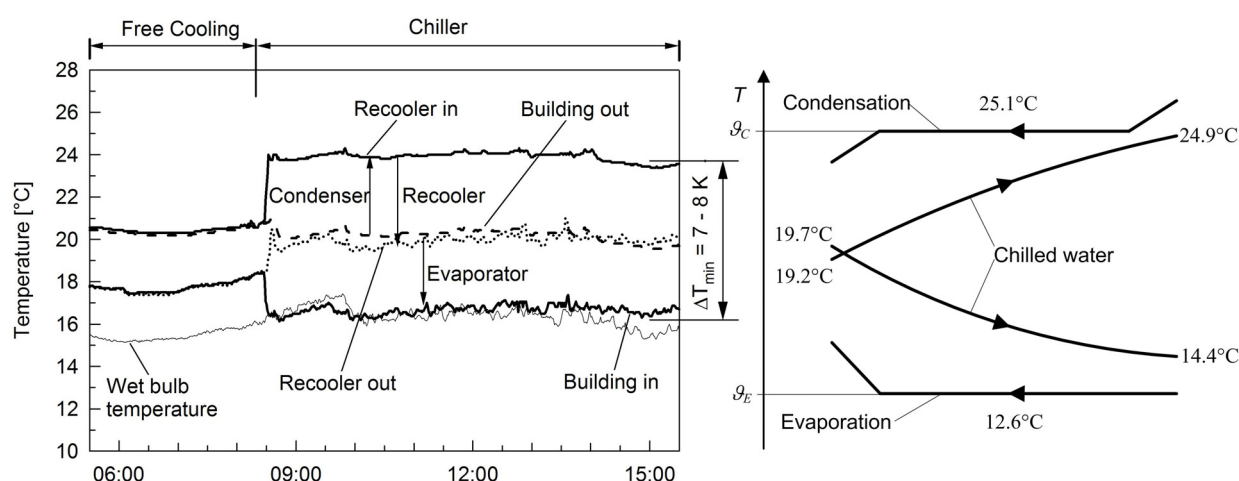


Figure 8: Left: Office building in Zurich: Typical temperature profiles for a summer day. Right: Example for the evaporation and condensation temperatures for a small-lift chiller. Desuperheating, subcooling in the condenser and the superheating in the evaporator are not drawn to scale.

The exergy analyses (Gasser et al. 2008) show the relations for the exergy losses: for the optimized building cooling system, the total exergy loss is around 70% , the external exergetic

efficiency therefore about 30%. For standard systems the exergetic efficiency is often below 20% (Wellig et al. 2006). The better performance of the optimized system is due to the fact that for a reduced temperature lift the exergy losses are reduced. The energy and exergy analyses show the impressive potential of small temperature lifts and specially designed building cooling systems.

4. CONCLUSION AND OUTLOOK

This study shows that in using small temperature lifts when heating and cooling buildings valuable savings in primary energy can be achieved. To reach this goal, not only the heat pump respectively the chiller, but the entire system including the distribution and delivery system, the heat source resp. recooling as well as the operation parameters and the control have to be optimized.

The key to efficiency increases is the often unnecessarily large temperature lift and its associated potential. In standard installations a significant destruction of exergy takes place. The reduction of the temperature lift and the use of highly efficient small lift systems provide the foundation to bring about considerably higher energy efficiencies. With the use of modern low temperature heating systems in combination with relatively deep geothermal probes heating COP values of above 9 can be achieved. In addition, the use of highly efficient chillers may lead to building cooling COP values significantly above 10.

5. ACKNOWLEDGEMENT

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