

## DESIGN OF EFFICIENT CAPACITY CONTROLLED AIR/WATER HEAT PUMPS

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**Abstract;** The popularity of air/water heat pumps (A/W-HPs) has continually increased as a common heating system in residential buildings. However, the potential for improvements in efficiency remains large. Theoretical and experimental analyses indicate that the exergetic efficiency of common A/W-HPs operating in an on/off control mode becomes smaller as the ambient temperature increases. A substantial increase in efficiency requires the adaption of the generated heating capacity to the required one using an appropriate continuous power control of the compressor and the fan. In this contribution, universally valid design and planning criteria for the realisation of efficient and economic A/W-HPs with continuous capacity control are presented. The optimal control strategy and the achievable efficiency strongly depend upon the part load characteristics of the compressor and the fan. Experiments with a prototype confirm the great potential of such capacity control. The seasonal performance factor (SPF) can be increased by approximately 10% to 60% compared to on/off operation. The application of these design guidelines and this control strategy can significantly reduce the primary energy consumption and the associated CO<sub>2</sub> emissions of A/W-HPs as well as the operating costs.

**Key Words:** air/water heat pumps, continuous capacity control, theoretical and experimental investigations, design guidelines, increase in efficiency

### 1 INTRODUCTION

Heating of residential buildings is still a costly and somewhat inefficient process. The use of air/water heat pumps (A/W-HPs) is steadily increasing, since they are easy to install and operate, reliable in their operation and together with a low energy building offer low heating costs. In Switzerland, for instance, the market share of newly built houses with heat pumps reached 83% in 2009. In total 20`600 heat pumps were sold that year. The proportion of the A/W-HPs was in the range of 56% (FWS, Heat Pump Association Switzerland, [www.fws.ch](http://www.fws.ch), 2010). For many home owners, low capital expenditure for the heating system often has priority over low operating costs. As a result, manufacturers are subjected to high price pressures leaving little remaining amounts of funds for important research and development activities. This lack of development results in a still large potential for increasing the efficiency of A/W-HPs.

Therefore, the development of efficient and economic heating systems with A/W-HPs is of great interest. The coefficient of performance (COP) is approximately inversely proportional to the inner temperature lift; the exergetic efficiency of A/W-HPs with on/off control, on the other hand, decreases with increasing ambient temperature, see figure 1 (left). The reason for this is due to the heat pump operation characteristic: the generated heating capacity increases with increasing ambient temperature and the associated decrease in required heating capacity.

It is well known that a substantial increase in efficiency can be achieved by continuous operation with power control of the compressor and the fan in order to adapt the generated to the required heating capacity within specific operating regions. A substantial amount of

research and development efforts have already been undertaken in this area. This paper points out the issues to be considered for the realisation of efficient and economic A/W-HPs with continuous capacity control. The focus lies on universally valid design and planning criteria (“guidelines”) to be used in practice. This focus seeks to give new impulses to manufacturers for the further development of A/W-HPs.

## 2 STATE OF THE ART

### 2.1 Overview

The use of A/W-HPs as heating systems for residential buildings has rapidly grown over the last 20 years. However, one of the most important prerequisites for an even larger distribution lies in the substantial increase of their efficiency. In the last decades, the Swiss Federal Office of Energy (SFOE) has taken on the task of improving the efficiency of heat pumps and of promoting their use in the heating of buildings. Therefore, the SFOE launched a great number of research projects in this area whereby optimal economic efficiency and ecological concerns are considered.

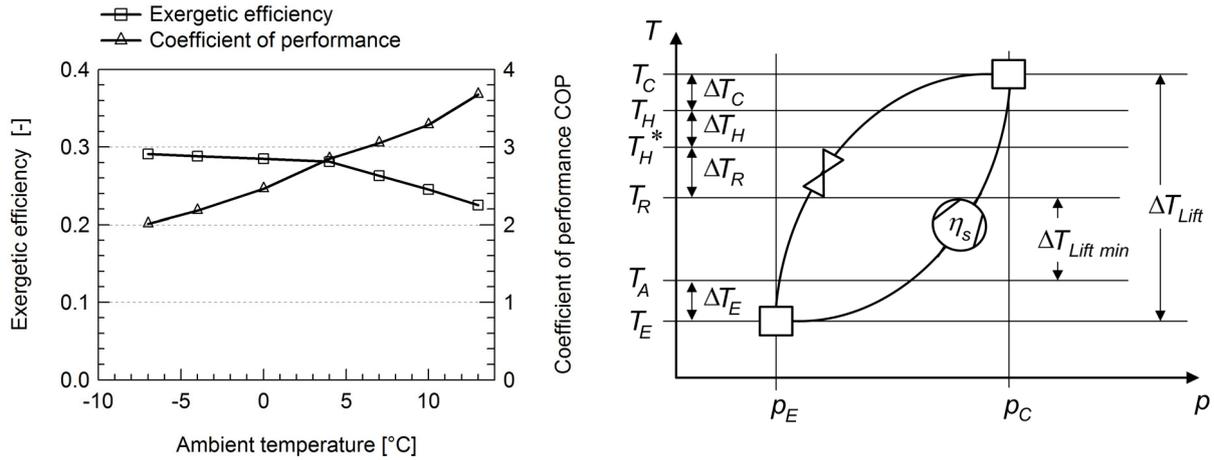
As already stated, heat pump systems exhibit a high potential for increasing their efficiency (Gasser et al. 2008). In around 1980, the seasonal performance factor (SPF) of A/W-HPs was around two in Switzerland, today about three and they have not reached their limits by far. During the same period of time the exergetic efficiency has increased from around 20% to 30%. An ideal heat pump working without any losses has an exergetic efficiency of 100%.

In order to achieve a considerable increase in the efficiency, all sub-processes as well as the heating system must be included in the analysis. Exergy analyses can show clearly where the losses occur, how large they are and how strongly they affect the individual sub-processes. A crucial point is the adaptation of the operating characteristic of the A/W-HP by using continuous capacity control (Gasser et al. 2008, 2011a). Another important issue is the reduction of the negative influence of ice and frost formation in the fin tube heat exchanger as well as the optimization of the necessary defrosting processes (Berlinger et al. 2008).

### 2.2 Exergy analysis of heating systems with A/W-HPs

An energetic evaluation is necessary for making judgements on the heat pump process but it does not completely suffice. The second law of thermodynamics provides information on the quality of the process. In our studies, the application of the second law is not done using abstract entropy balances but with exergy balances (Gasser et al. 2008, 2010). A systematic representation of the thermodynamics of heating has been supplied by Baehr (Baehr 1980). The term “exergy” is easily understandable for heat pump applications: The real driving power of the compressor is higher than the driving power of an ideal (reversible) process by the sum of all exergy losses that occur. If the exergetic efficiency can be improved by further developments, this will also result in an improvement in the COP and SPF, respectively.

The basic factors influencing the exergy losses of the heat pump are the temperature gradients for the heat transfer in the evaporator  $\Delta T_E$  and condenser  $\Delta T_C$  as well as the isentropic compressor efficiency  $\eta_s$ , see figure 1, right (here, only the refrigerant cycle is considered, i.e. the drive losses of the compressor as well as the fan power are ignored). These affect each other mutually and so reduce the exergetic efficiency of the heat pump. Moreover, all exergy losses are dependent on the temperature lift  $\Delta T_{Lift}$  (Gasser et al. 2008).



**Figure 1: Left: Exergetic efficiency and COP of a commercially available A/W-HP with on/off control according to measurements (Gasser et al. 2008)**  
**Right: Schematic representation of temperatures, temperature gradients and pressures of a heating system with an A/W-HP. Indices: A = ambient, E = evaporator, C = condenser, H = generated heating (supply) temp., H\* = required heating temp., R = room.**

For A/W-HPs with on/off control the intermittently generated heating water temperature  $T_H$  at ambient temperatures above the design point is around  $\Delta T_H$  higher than the heating temperature  $T_H^*$  continuously required by the building. The heat with temperature  $\Delta T_R$  supplied to the room is provided by the heat delivery system from a continuously required heating temperature  $T_H^*$  with a temperature gradient  $\Delta T_R$ . The temperature lift  $\Delta T_{Lift}$ , defined as the difference between the condensation temperature  $T_C$  and the evaporation temperature  $T_E$ , is enlarged in comparison to the minimum (ideal) temperature lift  $\Delta T_{Lift min}$  by the temperature gradients  $\Delta T_E$ ,  $\Delta T_C$  and  $\Delta T_R$  as well as by the discrepancy  $\Delta T_H$  between required and generated heating temperature.

**Table 1: Equations for the evaluation of the exergy losses of the sub-processes of an A/W-HP and the heat distribution and delivery system (Gasser et al. 2008). Explanations: Indices according to figure 1;  $\Delta h_v$  = specific heat of evaporation,  $c_{pl}$  = specific heat capacity of liquid refrigerant,  $\eta_s$  = isentropic efficiency.**

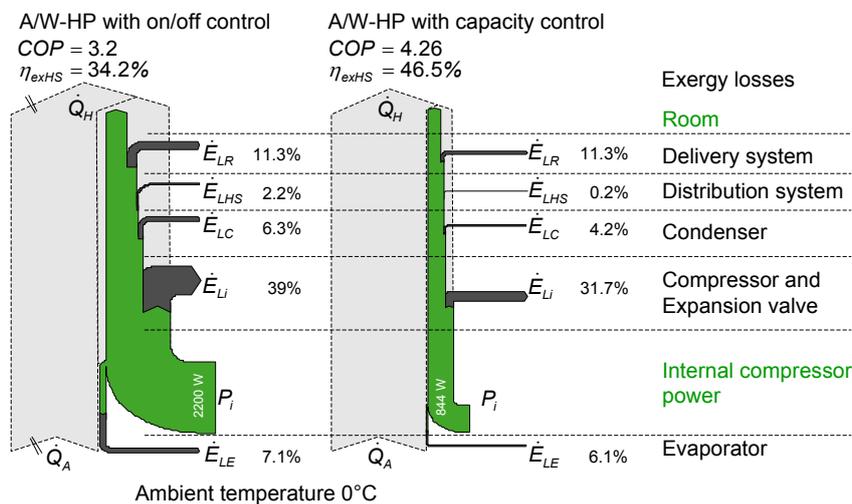
Exergy loss	Equation
Compressor	$\dot{E}_{LCp} = \dot{m}_f \cdot T_A \cdot \Delta h_v \cdot \frac{\Delta T_{Lift}}{T_E \cdot (T_E + \Delta T_{Lift})} \cdot \left[ \frac{1}{\eta_s} - 1 \right] \approx \dot{m}_f \cdot \Delta h_v \cdot \frac{1}{COP_{revi}} \cdot \left[ \frac{1}{\eta_s} - 1 \right]$
Expansion valve	$\dot{E}_{LEx} \approx \dot{m}_f \cdot T_A \cdot c_{pl} \cdot \frac{1}{2} \cdot \left( \frac{\Delta T_{Lift}^2}{T_E^2 + T_E \cdot \Delta T_{Lift}} \right) \approx \dot{m}_f \cdot c_{pl} \cdot \frac{\Delta T_{Lift}}{2 \cdot COP_{revi}}$
Evaporator	$\dot{E}_{LE} = \dot{Q}_E \cdot T_A \cdot \frac{\Delta T_E}{\bar{T}_E \cdot (\bar{T}_E + \Delta T_E)} \approx \dot{Q}_E \cdot T_A \cdot \frac{\Delta T_E}{\bar{T}_E^2} \approx \dot{Q}_E \cdot \frac{\Delta T_E}{T_A}$
Condenser	$\dot{E}_{LC} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H \cdot \bar{T}_C} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H \cdot (T_H + \Delta T_C)} \approx \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H^2}$
Heat distribution system	$\dot{E}_{LHS} = \dot{Q}_H \cdot T_A \cdot \frac{T_H - T_H^*}{T_H \cdot T_H^*} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_H}{T_H \cdot T_H^*}$
Heat delivery system	$\dot{E}_{LR} = \dot{Q}_H \cdot T_A \cdot \frac{T_H^* - T_R}{T_H^* \cdot T_R} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_R}{T_H^* \cdot T_R}$

Gasser et al. (2008) developed physical equations in order to show the quantitative effect of the relevant factors on exergy losses and exergetic efficiency (see table 1). By making appropriate approximations (e.g. linearization or series expansion), relatively clear equations can be established that can easily be discussed without accuracy being affected.

The development engineer is interested both in the magnitude of the individual exergy losses as well as in comparisons made between them. A clear appraisal can successfully be made by using the exergy loss ratio which is defined as the appropriate exergy loss flow with reference to the internal compressor power. The reason for this is that one can directly follow the subtractive effect on exergetic efficiency (figure 2). The exergetic efficiency of the entire heating system with an A/W-HP is calculated as follows:

$$\eta_{exHS} = \frac{\dot{Q}_H \cdot \frac{T_R - T_U}{T_R}}{P_i} = 1 - \frac{\dot{E}_{LCp}}{P_i} - \frac{\dot{E}_{LC}}{P_i} - \frac{\dot{E}_{LEx}}{P_i} - \frac{\dot{E}_{LE}}{P_i} - \frac{\dot{E}_{LHS}}{P_i} - \frac{\dot{E}_{LR}}{P_i} \quad (1)$$

Figure 2 shows the energy/exergy flow diagrams including the exergy loss ratios for the entire heating system with an A/W-HP with on/off control at an ambient temperature of 0°C. For conventional A/W-HPs with on/off control, the generated heating capacity increases with increasing ambient temperature and the associated decrease in required heating capacity. As a result, the temperature gradients for heat transfer increase in both evaporator and condenser so that the exergy losses in the evaporator and condenser increase quasi-progressively with increasing ambient temperature thus reducing exergetic efficiency. Further, the difference between the generated and required heating temperature increases with increasing ambient temperature, thus leading to a further exergy loss. Although this exergy loss originates outside the actual heat pump in the heat distribution system, it must, however, be attributed to the heat pump. The reason for this unfavourable behaviour of the A/W-HP with on/off control is the operating characteristic of the constant-speed compressor.



**Figure 2: Energy and exergy flow diagram of the entire heating system with a commercially available A/W-HP with on/off control (left; evap. temp. -9°C, cond. temp. 42°C, supply temp. 35.5°C) and with a A/W-HP with continuous capacity control (right; evap. temp. -5°C, cond. temp. 36°C, supply temp. 33°C) at 0°C ambient temperature. System information: 5.4 kW nominal heating capacity at -10°C; constant heating water flow (700 l/h) with supply temp. 42°C and return temp. 35° at the balance temp. -10°C; detailed specifications of the heat pump can be found in WEXA (Gasser et al. 2008). The simulations were carried out using the simulation program for A/W-HPs developed in LOREF (Berlinger et al 2008).**

The efficiency can be markedly improved by continuous capacity control of the compressor and the fan, i.e. by the adaptation of the generated heating capacity to that required. In this

way, the temperature gradients for heat transfer in the evaporator and condenser that are encountered when ambient temperatures rise can be reduced effectively. In addition, the generated heating temperature almost always corresponds to that required when this control strategy is used. The temperature lift is distinctly reduced in comparison with on/off control thus leading to a clear increase in exergetic efficiency and COP.

### **2.3 Evaporator**

The optimal design of the evaporator is of vital importance concerning the heat pump's efficiency. At low ambient temperatures, the water vapour contained in the air is deposited on the fins and tubes of the heat exchanger in the form of ice or frost. This formation of ice and frost is influenced by a complex interaction between evaporator, fan characteristic and the characteristic of the heat pump itself.

In the SFOE research projects LOREF, the ice and frost formation in fin tube heat exchangers has been investigated theoretically and experimentally (Sahinagic et al. 2004, Berlinger et al. 2008, Sahinagic et al. 2008, Albert et al. 2008). A layer of ice or frost has basically two negative effects. Firstly, ice or frost is an additional thermal resistance between the boundary layer of the air and the evaporating refrigerant and secondly, the obstruction of the free cross-section of the evaporator. However, this obstruction has a much greater influence on the performance of evaporator and fan than the higher thermal resistance of the ice or frost layer (Sahinagic et al. 2004). As a result of the increased pressure drop across the evaporator, the air flow rate decreases in accordance with the operating characteristic of the fan and the outlet temperature of the air drops also restricting the simultaneous heat and mass transfer. The evaporation temperature drops and, along with it, the efficiency of the heat pump. Therefore, the evaporator can be optimised by providing for a more even distribution of the ice and frost deposited.

A simulation program was developed which allows the calculation of the non-stationary operating behaviour of A/W-HPs with ice and frost formation in the evaporator for any geometries (Sahinagic et al. 2008). The core of the calculation program is a model for simultaneous heat and mass transfer (Sanders 1974) along with an empirically developed correlation for the calculation of the effective thickness of the frost layer formed (Fahlén 1996) and the air-side heat transfer coefficient. The theoretical results have been validated by numerous experiments (Berlinger et al. 2008). The investigation shows that the geometry is of major importance in regard to the requirement of defrosting.

The evaporator and the fan must be considered as a system and aligned as closely as possible to each other so that the fan power is minimized and the fan itself is operated at maximum efficiency. With larger fin spacing the fan power consumption is reduced, the air flow remains high even at high frost mass loadings and longer heating cycles are possible. Moreover, the defrosting can be accomplished at appropriate ambient conditions (i.e. at air conditions above approximately 2°C and well above the fog isotherm at 0°C) by ambient-air defrosting. The application of this technique can significantly reduce defrosting energy consumption and improve the overall efficiency of A/W HPs (Wellig et al. 2008).

### **2.4 Capacity controlled A/W-HPs**

In research as well as in industry, the potential of continuous capacity control is recognized. In various research studies the energy savings potential was studied (e.g. von Böckh et al. 2005, Karlsson et al. 2007) and in others the possibilities regarding the technical implementation were investigated (e.g. Aprea et al. 2006, Kim et al. 2005). Presently several companies already offer fully modulating A/W-HPs which, in some cases, have considerably higher COP values than conventional systems but today, general principles for the design of efficient A/W-HPs with continuous capacity control in practice are still missing.

### 3 OPTIMAL STRATEGY FOR EFFICIENT CAPACITY CONTROL OF A/W-HPs

As described in section 2, the efficiency of an A/W-HP can be increased significantly by adjusting the generated to the required heating capacity. In our studies, the adaptation of the refrigerant flow using power control of the compressor is chosen. In order to further increase the efficiency, a controllable fan is employed that enables an adjustment of the air flow to an optimal value. The power consumption of the fan can have a significant impact on the efficiency of the heat pump, especially in its part load range. In addition, the part load efficiencies of the compressor and fan have a large influence on the COP and strongly influence the control strategy as will be shown in the next sections. The subsequent discussions also clarify how the optimal part loads can be determined for both the compressor and the fan. These findings are further summarized in section 3.3 "Guidelines".

The simulation results presented in the following sections were calculated using a simulation program for A/W-HPs. This validated program allows for a precise calculation of the heat pump behaviour for almost any heat pump configuration, heating curve and ambient condition (Sahinagic et al. 2008), see also section 2.

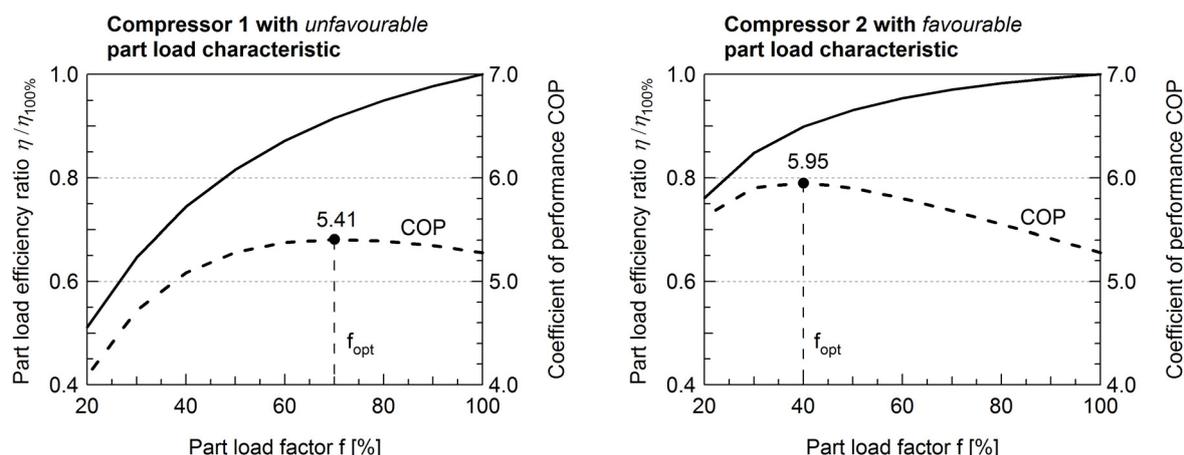
#### 3.1 Capacity control of the compressor

##### *a) Capacity control of the compressor without consideration of the fan*

In order to achieve better understanding, the following only considers the control of the compressor and does not account for the fan. The simulation results presented in section 3 and the measurement results shown in section 4 each have a typical heating curve corresponding to low energy buildings "Minergie" (see also Gasser et al. 2011a, 2011b) using low heating water temperatures (supply/return temperature 30/25°C at an ambient temperature of -10°C) and a low heating limit (10°C).

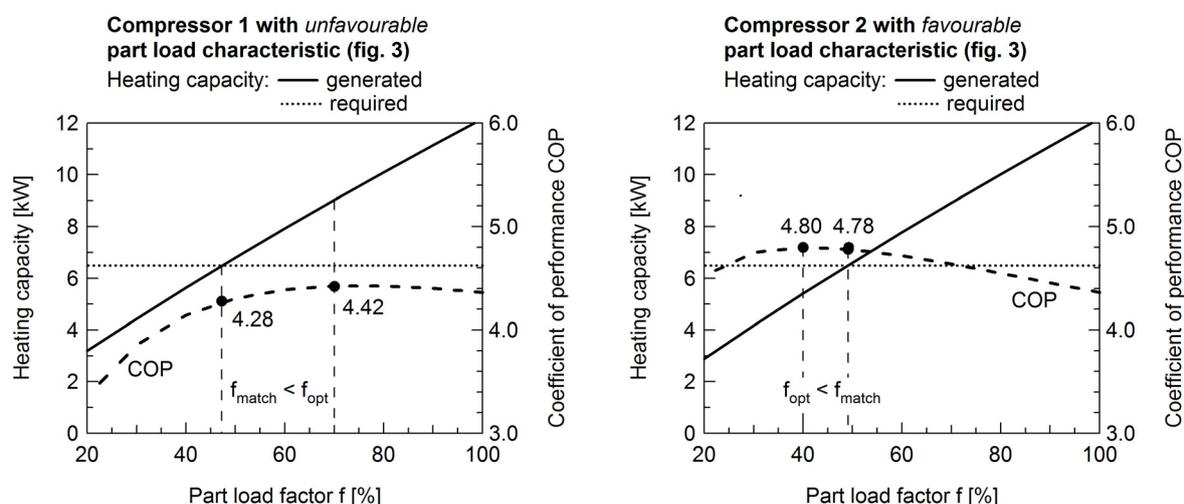
*Selection of the compressor:* Initially, the influence of the compressor's part load efficiency on the heat pump efficiency as well as on the optimal control strategy is discussed. In order to simplify the interpretation, the COP calculation does not include the fan power. The part load efficiency of a compressor is the deciding factor whether a compressor is at all suitable for capacity control. Figure 3 shows two virtual (but realistic) examples of the part load behaviour for a compressor with *favourable* and *unfavourable* performance. The part load performance is represented by the ratio of the respective part load efficiency to the part load efficiency at full load. For the simulations, the part load factor  $f$  is defined as the ratio of working fluid mass flow at part load to the one at full load and is neither related to the speed of the compressor nor its power consumption. The part load factors of the two virtual compressors range from 20% to 100%.

Figure 3 shows the COP as a function of the part load factor for the two virtual compressors using the same heat pump and heating system. Compressor 1 has a maximum COP of 5.41 at a part load of 70% compared to 5.28 at full load, whereas compressor 2 has a maximum COP of 5.95 at a part load factor of 40% compared to a full load COP of 5.28.



**Figure 3: Part load efficiency ratio and COP without fan using a compressor with unfavourable (left) and favourable (right) part load characteristic as a function of the part load factor. The simulations were carried out for 6°C ambient temperature, 85% relative humidity and the heating curve “Minergie” (Gasser et al. 2011a).**

*Control of the compressor:* Adjusting the generated heating capacity to the required heating capacity over the entire part load range of the compressor is not generally advisable. Based on the resulting higher COP, it may be more favourable to operate in on/off control mode at a reduced part load factor for a certain ambient temperature range. This behaviour is illustrated in figure 4, where the curves of COP for compressors 1 and 2 and the required and the generated heating capacities are shown over the part load factor for one example of constant ambient conditions.

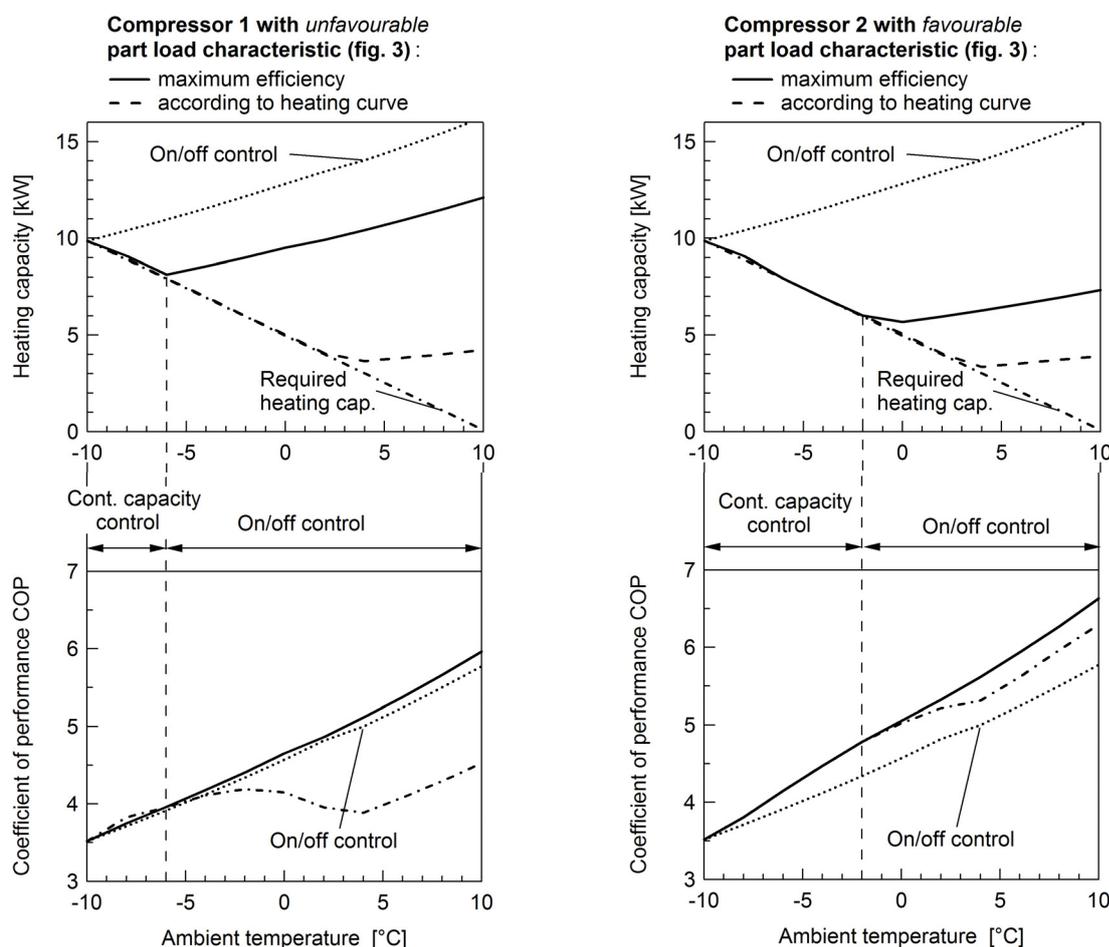


**Figure 4: Required and generated heating capacity and COP without fan as a function of the part load factor according to the part load characteristic in figure 3. The simulations were carried out for -2°C ambient temperature, 85% relative humidity and the heating curve “Minergie” (Gasser et al. 2011a).**

The part load factor  $f_{match}$  is where the required heating capacity, shown as a horizontal dashed line, and the generated heating capacity meet. Above  $f_{match}$  the heat pump is operated in either on/off or capacity control mode. Within the part load range below this point the heat pump cannot be operated since the generated heating capacity is insufficient. In the case where  $f_{opt}$  is greater than  $f_{match}$  as in the example of figure 4 (left), the heat pump is operated in on/off control at reduced power when in the range between  $f_{match}$  and  $f_{opt}$ . If  $f_{opt}$  is smaller than  $f_{match}$ , as in the example of figure 4 (right), the heat pump can be continuously capacity controlled down to the part load factor  $f_{match}$ .

The determination of  $f_{opt}$  is shown in figure 4 for one specific ambient state ( $-2^{\circ}\text{C}$ , 85 r.h.). The shape of the COP curve remains qualitatively the same for other ambient conditions and heating curves so that this specific  $f_{opt}$  of the compressor can be taken as a basis for all operating conditions (Gasser et al. 2011a). If these guidelines are implemented, the heating capacities as shown in figure 5 (top) result.

The point where the generated heating capacity, when controlled at maximum efficiency, separates from the required heating capacity, corresponds to the above discussed part load factor  $f_{opt}$ . From figure 5 it can thus be concluded that for compressor 1 the generated heating capacity should only be adapted to the required one in the temperature range from  $-10^{\circ}\text{C}$  to  $-6^{\circ}\text{C}$ . Above  $-6^{\circ}\text{C}$  the heat pump should be operated in on/off control at the part load factor  $f_{opt} = 70\%$ . The heat pump with compressor 2, on the other hand, can be operated in a temperature range from  $-10^{\circ}\text{C}$  to  $-2^{\circ}\text{C}$  with maximum efficiency using continuous capacity control. Above  $-2^{\circ}\text{C}$  the heat pump should be operated in on/off control at the part load factor  $f_{opt} = 40\%$ . Above an ambient temperature of  $4^{\circ}\text{C}$  for compressor 1 and  $2^{\circ}\text{C}$  for compressor 2, the heat pump must be operated in on/off control anyway because the minimum possible part load factor of 20% is reached.



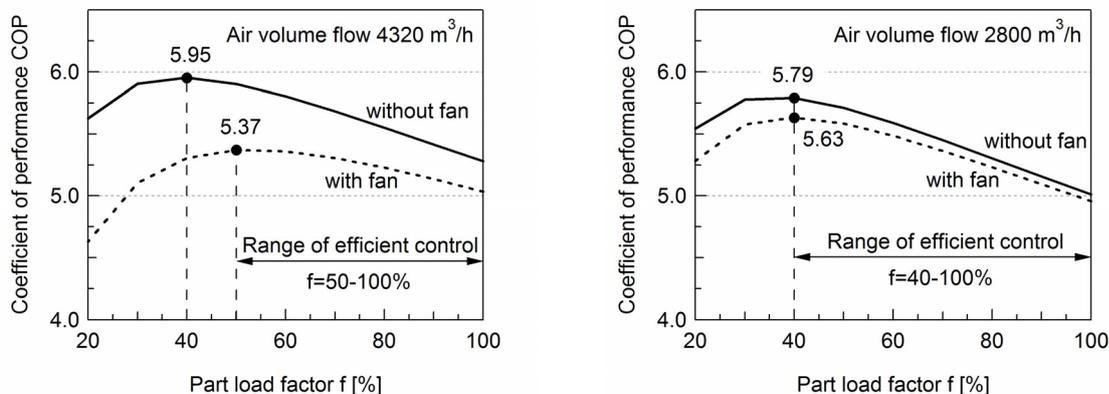
**Figure 5: Heating capacity (top) and COP without fan (bottom) of the heat pump as a function of the ambient temperature according to the part load characteristic in figure 3. The simulations were carried for the heating curve “Minergie” (Gasser et al. 2011a).**

The COP values resulting from this control strategy are shown in figure 5 (bottom). It becomes clear that in the case the controllable part load range of compressor 1 is fully utilized, a significant loss of efficiency results when compared to an on/off control with full load. Even when always controlled at maximum efficiency and the part load factor does not

drop below  $f_{opt}$ , no considerable increase in the COP is possible using continuous capacity control. In contrast is the behaviour of compressor 2: over the entire controllable part load range of the compressor the COP never falls below the value when on/off controlled. The efficiency improvement compared to on/off control operating at full load would be about 9% for an ambient temperature at 10°C. The improvement in efficiency using continuous capacity control at maximum efficiency is about 15% for this ambient temperature.

*b) Capacity control of the compressor with consideration of fan power*

Since the fan, like the compressor, has a larger operating time for continuous capacity control compared to on/off control, its efficiency as well as the matching in design of the evaporator and fan are of great importance. In addition, for the case where the fan is not under continuous capacity control, the fan power must be considered in the evaluation of the optimal control strategy. This conclusion is made clear in figure 6: For two different air flow rates the curves of the COP are shown (compressor 2). With a volume flow of 4320 m<sup>3</sup>/h (left) using continuous capacity control down to a part load factor of 50% is beneficial. At a lower air flow the control strategy should utilise the part load range down to part load factor of 40% (right). The above established optimal part load factor  $f_{opt}$  decreases with a decreasing air flow when considering the fan power and approaches the optimal part load factor when the fan is not considered. For a heat pump operating at constant fan speed the speed can only be lowered to a minimum, because the heat pump requires a minimum air flow for achieving the required heating capacity at full load. This shows that  $f_{opt}$  can only be determined in the case of single control of the compressor when the fan power is taken into account and therefore, in selecting the optimal speed a trade-off must be achieved. But if the compressor and the fan are both controlled, the determination of  $f_{opt}$  can ignore the fan power since at part load range this power has minimal influence.



**Figure 6: COP with and without fan as a function of the ambient temperature according to the part load characteristic in figure 3 right (compressor 2). The simulations were carried out for 4320 m<sup>3</sup>/h (left) and 2800 m<sup>3</sup>/h (right) air volume flow at 6°C ambient temperature, 85% relative humidity and the heating curve “Minergie” (Gasser et al. 2011a).**

### 3.2 Simultaneous control of compressor and fan

To achieve high efficiencies, both the compressor and the fan have to be power controlled. They should have part load ranges as large as possible with high part load efficiencies. The determination of the optimal fan rotational speed and thus the optimum air flow rate is done similarly to determining the optimal compressor speed. Given the previously ascertained compressor rotational speed and almost any combination of heat source and sink conditions, the COP curve over a range of fan rotational speeds can be calculated. Figure 7 (right) shows an example of a measured curve profile for a real A/W-HP. Apparently, an optimal part load ratio  $f_{opt,F}$  exists for the maximum COP that must not be exceeded. Similarly, an optimum fan rotational speed  $f_{opt,F,full}$  can be established when at full load. The fan rotational

speed is then linearly reduced between  $f_{opt,F,full}$  and  $f_{opt,F}$  while the heat pump is running in the part load range between full load and  $f_{opt,C}$  of the compressor. In on/off operation, the compressor and the fan each operate at the optimum speed. Therefore, at each operating point there is a separate optimal rotational speed for both the compressor and the fan.

### 3.3 Guidelines

The following set of guidelines summarize the findings presented in the previous sections. To achieve the best possible efficiency of a capacity controlled A/W-HP, the maximum effective range of the part load of the compressor and the fan are utilised. In order to determine the optimal control strategy for simultaneous control of these components the following procedure is recommended (Gasser et al. 2011a):

*1. Determination of the optimal part load factor of the compressor:* First, the optimal part load of the compressor  $f_{opt,C}$  and therefore the optimal compressor speed without considering the fan are to be determined. These values when under part load operation cannot be exceeded due to efficiency reasons. This determination can be done at almost any source and sink conditions.

*2. Determination of the optimum air flow rate at part load operation:* In a further step, the optimal part load of the fan  $f_{opt,F}$  and thus the optimal air flow rate can be determined when the heat pump is running at the step 1 optimum compressor speed. The fan speed should never drop below this value. This finding again can apply at almost any source and sink operational state.

*3. Determination of the optimum air flow rate at full load:* The maximum of the profile given by the COP versus air flow curve shows the optimal part load factor  $f_{opt,F,full}$  of the fan when the heat pump is at full load.

In the capacity range of the heat pump between  $f_{opt,C}$  and the nominal load of the compressor the speed of the fan is to be varied linearly between  $f_{opt,F}$  and  $f_{opt,F,full}$ .

## 4 EXPERIMENTAL PROOF

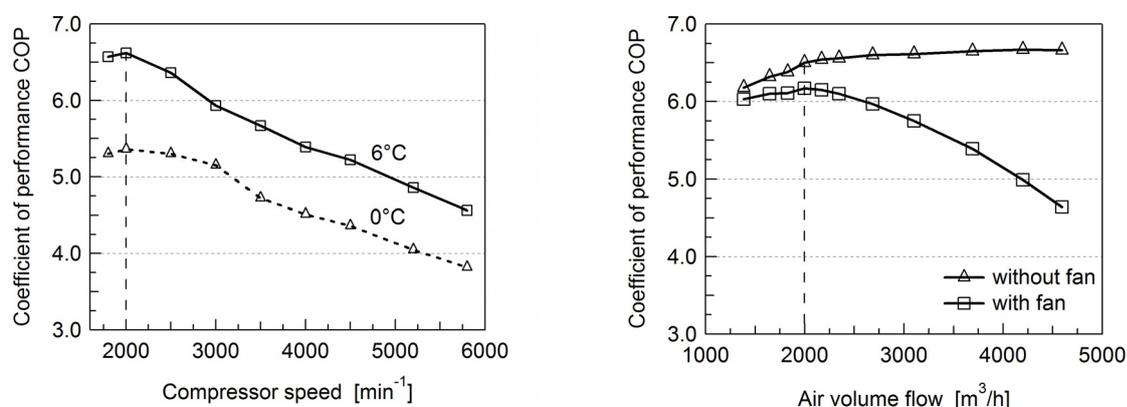
### 4.1 Prototype of a capacity controlled A/W-HP

In order to confirm the potential of the capacity control and to verify the control strategy described above, an A/W-HP prototype has been realised. The design criterion was  $-10^{\circ}\text{C}$  ambient temperature for monovalent operation. The A/W-HP prototype was equipped with a variable speed scroll compressor identified as 1<sup>st</sup> Generation (Gasser et al. 2011a, 2011b) that enables continuous capacity control over a wide range of the part load factor ( $f = 26\text{--}100\%$ ). In the design and layout stage of the prototype it was decided that it was to be built as simple as possible. From this basis all additional components not totally necessary were excluded from the prototype. The design of the evaporator (fin tube heat exchanger) had to be made in conjunction with the selection of the fan (Berlinger et al. 2008); a capacity controlled fan with permanent magnet motor was selected for the prototype. Also, to minimize the fan power consumption, special care had to be taken to ensure that the air side pressure drop was kept as low as possible across the evaporator and that the fan was maintained at maximum efficiency. A typical plate heat exchanger is used as a condenser. As throttle device an electronic expansion valve from the compressor manufacturer is employed. Detailed information about the components of the heat pump and especially about the heat transfer characteristics can be found elsewhere (Gasser et al. 2011a). The A/W-HP prototype and the testing facility were equipped with extensive measurement devices, so that all relevant process variables could be determined. This extensive measurement system allowed detailed analyses of the heat pump process (Gasser et al. 2011a, 2011b).

## 4.2 Optimal part load factors of compressor and fan

For the experiments, the part load factor represents the ratio of rotational speed of the compressor respectively the fan to rotational speed at full load.

In a first step, the rotational speed of the compressor was varied across the whole controllable range for constant ambient conditions and the resulting COPs were calculated and analysed without considering the fan power. The controllable range extends from  $1800 \text{ min}^{-1}$  to  $7000 \text{ min}^{-1}$  which corresponds to the above mentioned part load range. The following figure 7 (left) shows the profile of the COP for the ambient temperatures of  $0^\circ\text{C}$  and  $6^\circ\text{C}$ . Each COP profile has its maximum at  $2000 \text{ min}^{-1}$ , which results in a wide range for efficient capacity control. After this, the rotational speed of the fan was varied for the constant compressor speed of  $2000 \text{ min}^{-1}$  and the profile of the resulting COP was analysed again. The maximum COP points at the optimal air volume flow of  $2000 \text{ m}^3/\text{h}$ , see figure 7 (right).



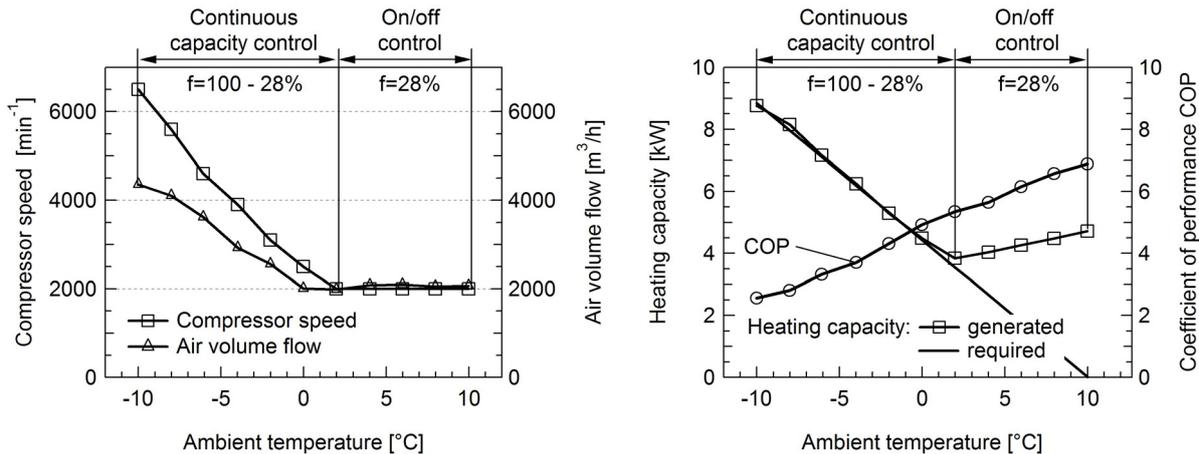
**Figure 7: Left: COP without as a function of the compressor speed at  $0^\circ\text{C}$  and  $6^\circ\text{C}$  ambient temperature, 85% relative humidity,  $2150 \text{ m}^3/\text{h}$  air volume flow and the heating curve “Minergie”. Right: COP with and without fan as a function of the air volume flow at  $2000 \text{ min}^{-1}$  compressor speed,  $6^\circ\text{C}$  ambient temperature, 85% relative humidity and the heating curve “Minergie” (Gasser et al. 2011a).**

After determining the optimal load of the compressor and the optimal air volume flow for full load of the heat pump the range of the admissible part load factors of the compressor and the air volume flow of the fan were defined completely.

## 4.3 Operating characteristics and efficiency

The A/W-HP prototype is controlled in a way to ensure the efficiency is always at a maximum. To achieve this, the compressor is operated at part load within the temperature range of the ambient air from  $-10^\circ\text{C}$  to  $2^\circ\text{C}$ , where  $-10^\circ\text{C}$  corresponds to the part load factor of 100% and  $2^\circ\text{C}$  to 28%, see figure 8 (left). Within this range the part load factor of the compressor and fan are reduced linearly from full load to the respective optimal part load factor. The compressor speed and the air flow are shown in figure 8 (left). Above an ambient temperature of  $2^\circ\text{C}$  the heat pump is operated in on/off mode at the optimal part load factors of 28% for the compressor and  $2000 \text{ m}^3/\text{h}$  air volume flow for the fan, see figure 8 (right).

Through continuous capacity control the inner temperature lift could be reduced significantly compared to common heat pumps operating intermittently at full load. The reduction of the inner temperature lift is especially attributed to smaller temperature gradients for the heat transfer in the evaporator and the condenser (i.e. smaller exergy losses).



**Figure 8: Compressor speed and air volume flow (Left) and heating capacity and COP (Right) as a function of the ambient temperature based on the heating curve „Minergie“.**  
(f is the part load factor of the compressor)

As a consequence of a distinct increase of the condensation temperatures and the temperature lifts below an ambient temperature of  $-6^{\circ}\text{C}$  the COPs are low within this temperature range of the ambient air. The reason is the high filling level of the refrigerant which is, however, necessary for proper operation at part load. Because of this refrigerant quantity the liquid column in the condenser increased, resulting in a reduced surface for condensation, less heat transfer and consequently higher condensation temperatures.

The continuous capacity control results in higher evaporation temperatures. Consequently, ice and frost formation can be reduced considerably starting at a lower ambient temperature of about  $3.5^{\circ}\text{C}$ , in contrast to about  $5.5^{\circ}\text{C}$  for on/off control, each depending on the relative humidity of the air. By lessening the ice and frost formation, the number of defrosting processes can be reduced, which has a positive effect on the achievable SPF.

On top of this, an efficient defrosting control was integrated to exploit additional potential. This control is capable of determining the optimum time for initiating the defrosting process, of choosing the optimal defrosting method (ambient air or reverse cycle defrosting) and of determining the process at the optimum time (Gasser et al. 2011a).

Based on the calculation method by von Böckh (von Böckh et al. 2005, bin method without defrosting), for the city of Zurich and the above described “Minergie” heating curve the A/W-HP prototype achieved a SPF of 5.1, which is far better than average SPFs for newly installed intermittently operated common A/W-HPs in this region. Given the same heating curve, this performance of the A/W-HP prototype corresponds to a SPF of 4.8 for Stockholm and a SPF of 5.8 for Tokyo.

## 5 CONCLUSIONS AND OUTLOOK

Exergy analysis shows that the potential for highly efficient heating systems with A/W-HPs is not fully exploited yet. The reason for the comparatively moderate efficiency of common A/W-HPs with on/off control is due to the unfavourable operating characteristic, which results from the use of a constant speed compressor. By adapting the generated heating capacity to the required one by means of continuous capacity control, considerably higher exergetic efficiencies and COP values can be achieved. An indispensable prerequisite for achieving efficient A/W-HPs with continuous capacity control is the use of compressors and fans with a wide control range and high part load efficiencies.

In order to attain high efficiencies in building heating, integral and optimised solutions are needed. A prerequisite for this are, in addition to highly efficient heat pumps and heating systems, well thought out architectural designs and the consideration of the various physical qualities of buildings. Further, the building, its heating system, the heat pump and control (strategy) employed must be matched to each other in the best possible way.

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