CONTINUOUS CAPACITY CONTROL – THE KEY TO HIGH EFFICIENT AIR/WATER HEAT PUMPS

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Abstract: Air/water heat pumps require only low investments, are easy to install and reliable in their operation. However, one of the most important prerequisites for the increasing proliferation of air/water heat pumps lies in the substantial increase of their efficiency. Theoretical and experimental analyses indicate that the exergetic efficiency of common air/water heat pumps become smaller when the ambient temperature increases. As a consequence, a substantial increase in efficiency requests continuous operation with capacity control of the compressor and the fan. The aim of this project is to develop universally valid design and planning criteria for the realisation of efficient, reliable and economic air/water heat pumps with continuous capacity control. An optimised process control strategy was developed using simulations as well as experiments with a prototype. The experimental investigations approve the great potential of the capacity control. As a result the seasonal performance factor can be increased by approximately 10 to 60% compared to air/water heat pumps with on/off control.

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

1 INTRODUCTION

The use of air/water heat pumps as heating systems for residential buildings has rapidly grown over the last decades. Air/water heat pumps (A/W-HP) require only low investments, are easy to install and reliable in their operation. However, one of the most important prerequisites for the increased proliferation of heat pumps lies in the substantial increase of their efficiency. Common heat pump systems still have a large potential for improvement in their energy efficiency.

The Swiss Federal Office of Energy (SFOE) has initiated and funded two research projects titled LOREF “Air cooler optimization with reduction of ice and frost formation” (Berlinger et al. 2008) and WEXA “Exergy analysis for increasing the efficiency of A/W-HPs” (Gasser et al. 2008). Both projects have provided valuable knowledge related to the optimization of A/W-HPs which are now being implemented in practice.

The aim of the actual SFOE research project is to develop universally valid design and planning criteria for the realisation of efficient, reliable and economic A/W-HPs with continuous capacity control.

Energy balances are necessary for the assessment of the heat pump, but not sufficient. The second law of the thermodynamics describes the quality of thermodynamic processes. Instead of evaluating the quality of the thermodynamic processes with abstract entropy balances, exergy analyses are suitable for our purposes. A systematical representation of the thermodynamics of heating has been supplied by Baehr (Baehr 1980) in two contributions. He shows that exergy is a very practical and descriptive way of evaluating technical processes involving thermodynamic cycles and heat transfer. Its results are easy to follow and immediately demonstrate possibilities for improvement in thermodynamic efficiency because, using exergy analysis, exergy losses can be determined for both the complete system and their subsystems.

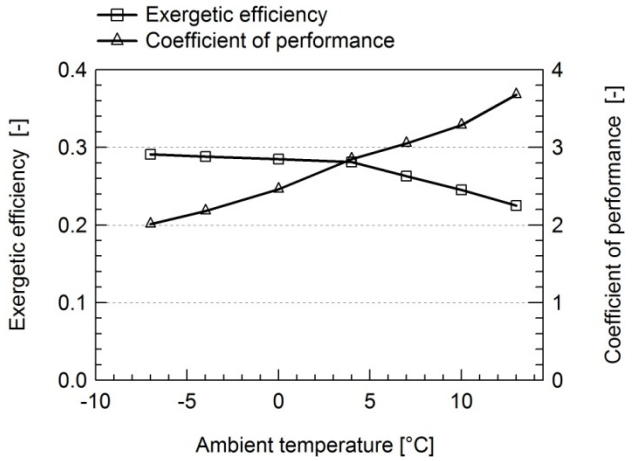


Figure 1: Exergetic efficiency and COP of commercially available A/W-HP with on/off control according to measurements (Gasser et al. 2008)

The coefficient of performance (COP) is approximately inversely proportional to the inner temperature lift. However, the exergetic efficiency of common A/W-HPs with on/off control decreases with increasing ambient temperature and the lower temperature lift associated with it, see figure 1. The WEXA project (Gasser et al. 2008) showed that the reason for this is the heat pump operating characteristic and that through continued adaption of the generated heating capacity to the required heating capacity the energetic and exergetic efficiency can be increased significantly.

2 FUNDAMENTALS

2.1 Air/water heat pumps with on/off control

The operating characteristics, the exergy loss ratios and the exergetic efficiencies presented in figures 2 and 3 (left) are valid for a commercially available A/W-HP with on/off control and 5.4 kW nominal heating capacity at -10°C ambient temperature (detailed specifications can be found elsewhere (Gasser et al. 2008)). The heating system of the considered building has a constant heating water flow (700 l/h) with a supply temperature of 42°C and a return temperature of 35° at the balance temperature of -10°C. For this specific case the heat pump is slightly oversized. The simulations were carried out using a simulation program for A/W-HPs which was developed and documented in LOREF (Berlinger et al 2008).

The reason for the poor efficiency of common A/W-HPs with on/off control is their unfavourable operating characteristic, which results from the compressor running at constant rotational speed. With increasing ambient temperature, the COP of such heat pumps does indeed increase. In contrast to this, the exergetic efficiency decreases, see figure 1. The thermodyna-

mics of heating would, on the other hand, make an increase of the exergetic efficiency possible. With increasing ambient temperature, the required heating capacity for a building decreases. The behaviour of a heat pump with on/off control is the opposite, as described above, and the discrepancy between required and generated heating capacity and temperature increases with increasing ambient temperature, see figure 2.

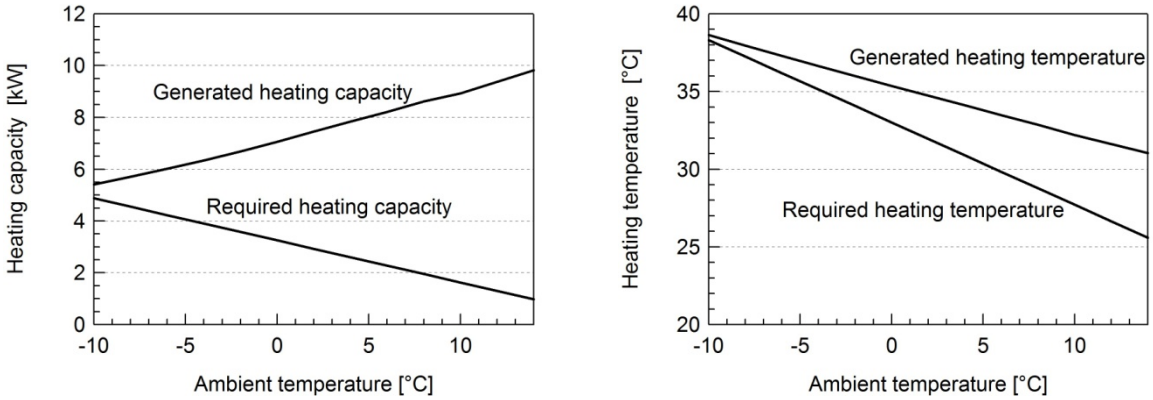


Figure 2: *Left*: Generated and required heating capacities
***Right*: Generated and required heating temperatures (i. e. supply temperatures)**

This behaviour has the effect of causing temperature gradients for heat transfer in the evaporator and condenser to increase with increasing ambient temperature. Moreover, this leads to a clear discrepancy between required and generated heating temperature. The actual temperature lift generated decreases less significantly in comparison with the ideal temperature lift (Gasser et al. 2010). As a result the exergetic efficiency itself is considerably reduced.

Figure 3 (left) 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. The exergy loss ratios, i. e. the exergy loss flow with reference to the internal compressor power, show directly the subtractive effect of the exergy losses on the exergetic efficiency.

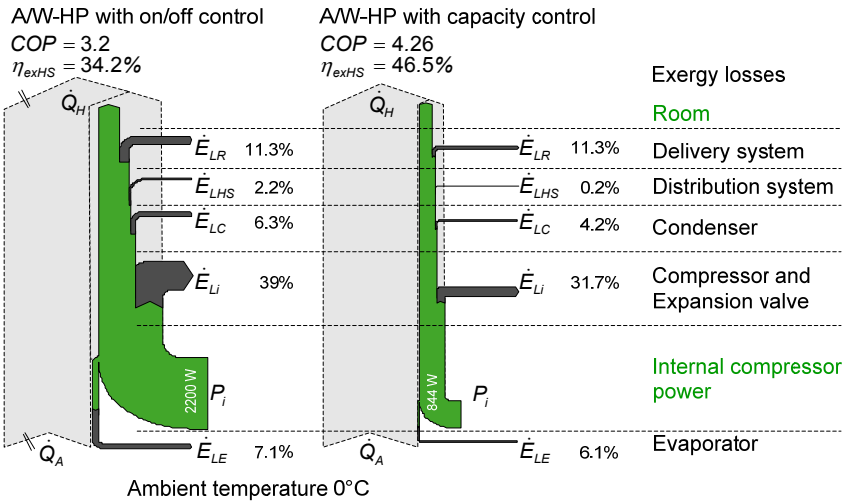


Figure 3: Energy and exergy flow diagram for the entire heating system with a commercially available A/W-HP with on/off control (*left*) (evap. temp. -9°C, cond. temp. 42°C, supply temp. 35.5°C) and with a A/W-HP with continuous capacity control (*right*) (evap. temp. -5°C, cond. temp. 36°C, supply temp. 33°C) (Gasser et al. 2008)

2.2 Air/water heat pumps with continuous capacity control

As previously stated, the discrepancy between required and generated heating capacity and temperature allows only a moderately high efficiency to be achieved for heat pumps with on/off control. In order to allow temperature gradients for heat transfer in the evaporator and condenser to be reduced, i.e. for increasing ambient temperature, any discrepancy between required and generated heating capacity and heating temperature must be avoided. Consequently, the A/W-HP should not be operated in on/off mode, but should be continuously operated – except during any defrosting processes necessary. Different strategies are available for the adaption of the generated heating capacity to that required. For example the compressor can be equipped with speed control. The best efficiencies, see figure 3 (right), can be achieved by using continuous capacity control of the compressor and the fan. Theoretical investigations show (without consideration of the electrical and mechanical efficiencies of the compressor and the fan) that when using this control strategy, the (SPF) can approximately be doubled in comparison to on/off control (see Gasser et al. 2008).

Detailed investigations show that, in the case of continuous capacity controlled A/W-HPs, the part load efficiencies of the fan and especially of the compressor play an important role and have a significant effect on the SPF and the annual average exergetic efficiency. For an easier interpretation of the influence of the part load behaviour of the compressor on the COP of the heat pump, simulations were carried out without consideration of the fan power. The fan is always running at constant rotational speed.

Figure 5 (top) shows the curves of the COP with on/off control and with capacity control of the compressor as a function of the ambient temperature. The part load efficiency of the compressor has a considerable influence on the curve of the COP as well as on the optimal control strategy when continuous capacity control is used for the compressor.

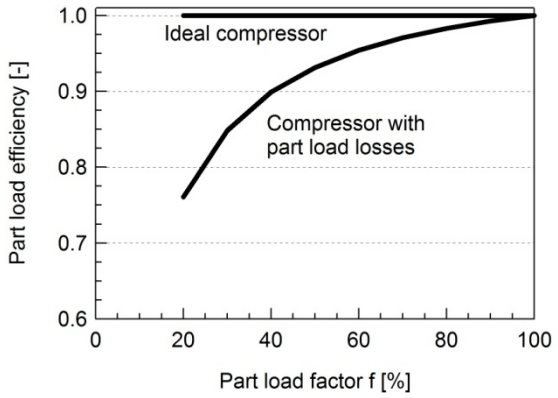


Figure 4: Part load efficiency of the compressor (here: ratio of respective part load efficiency to the efficiency at full load) as a function of the part load factor

Figure 5 (bottom) shows the heating capacity by using different control strategies as a function of the ambient temperature. If the generated heating capacity is consistently adapted to the required heating capacity using a compressor with part load efficiency according to figure 4, the efficiency of the heat pump becomes lower in comparison to the heat pump that is adapted to reach a maximum efficiency. If the heat pump control is always adapted to reach a maximum efficiency, a slight increase in the generated heating capacity results, if the A/W-HP is operated in on/off control with a capacity below the nominal capacity, starting at an ambient temperature of -2°C. A prerequisite for highly efficient A/W-HPs with continuous capacity control is the use of compressors and fans with high part load efficiencies.

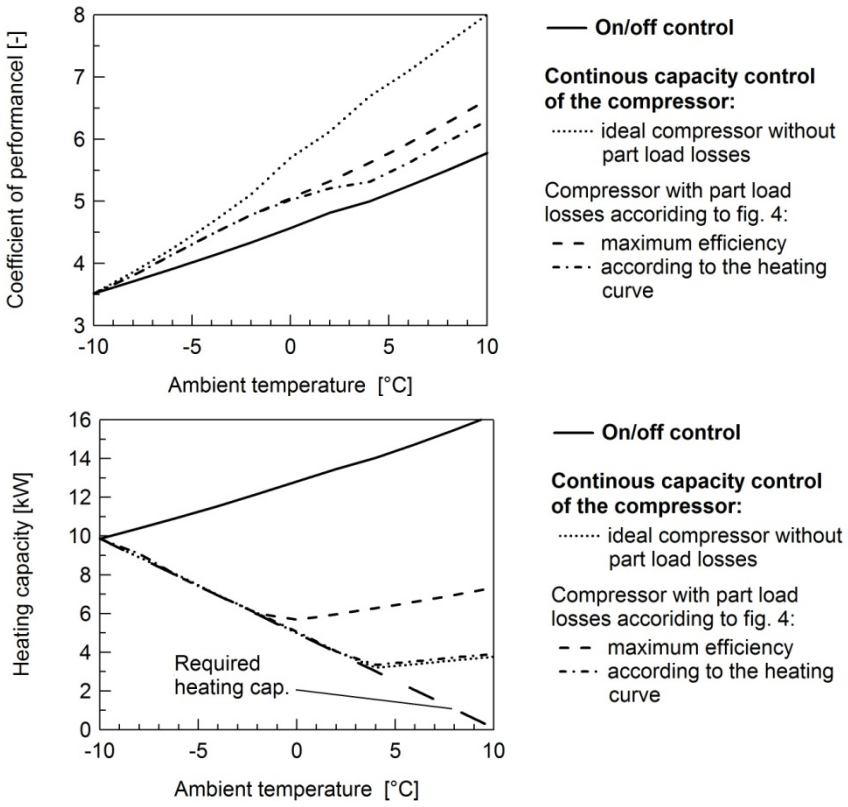


Figure 5: COP (top) and heating capacity (bottom) with and without continuous capacity control of the compressor without consideration of the fan as a function of the ambient temperature

3 EXPERIMENTS

To verify the theoretical findings as well as to investigate different control strategies a heat pump prototype with continuous capacity control as well as a heat pump test rig have been realized.

3.1 Test facility rig

Figure 6 (left) shows the schematic diagram of the heat pump test rig. The test rig consists of two air circuits, a primary and a secondary. This design allows a high dynamic of the air conditioning (necessary for defrosting processes) and stable test conditions. The resulting deviations from the target values are significantly lower than those specified in the EN 14511 code. The built-up heat pump test rig is shown in figure 6 (right).

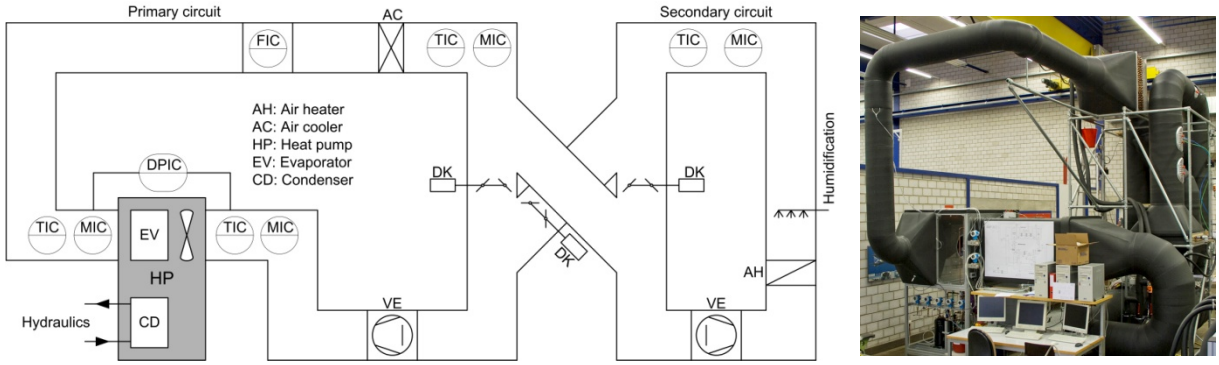


Figure 6: *Left*: Schematic diagram of the heat pump test rig
***Right*: Photograph of the heat pump test rig and the heat pump prototype**

3.2 Heat pump prototype

In order to confirm the potential of the capacity control and to verify the control strategies developed, three heat pump prototypes of continuous capacity controlled A/W-HPs were realized. The design criterion was -10°C ambient temperature for monovalent operation. The prototypes differ primarily in the type of the compressor used during testing. In this paper, only two of the compressors are considered.

The A/W-HP prototype was equipped with compressors that enable continuous capacity control over a wide range of the part load ratio (f = 25–100%). The first was a variable speed compressor prototype identified as 1st Generation. After the completion of the experimental investigations using the compressor of the 1st Generation, the HP was rebuilt and a variable speed compressor of the 2nd Generation was installed. The compressor of the 2nd Generation also enables continuous capacity control over a wide range of the part load ratio (f = 25–100%). This compressor is optimized for HP applications and is equipped with a vapour injection system.

In the design and layout stage of all prototypes, it was decided that each prototype 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 (capacity controlled fan with permanent magnet motor) (Berlinger et al. 2008). 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 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. 2011).

The A/W-HP prototypes and test rig 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.

4 RESULTS

The main objective of the overall experimental study was the confirmation of the distinct efficiency improvement potential of a capacity controlled A/W-HP. Several different control strategies were investigated, e.g. capacity control only of the compressor or the simultaneous capacity control of both the compressor and fan. However, this paper discusses only simultaneous control of the compressor and fan due to the much higher efficiency improvement that was attained. The study was done only for monovalent operation. To ensure the best testing conditions the evaporator was defrosted before each test run and measurement campaign.

4.1 Heating curves

The experiments were carried out for two different heating curves. The first one is a typical heating curve of a building based on the “Minergie” standard (Swiss standard for low energy buildings, www.minergie.ch) using a low heating water temperature and a low heating limit, see figure 7 (left). This heating curve (supply/return temperature 30/25°C at an ambient temperature of -10°C) is very typical for “Minergie” buildings using self-adjusting underfloor heating systems (i. e. constant heating water flow, no room controllers). The second one is a heating curve of a reconstructed old building that required a higher heating water temperature and heating limit, see figure 7 (right).

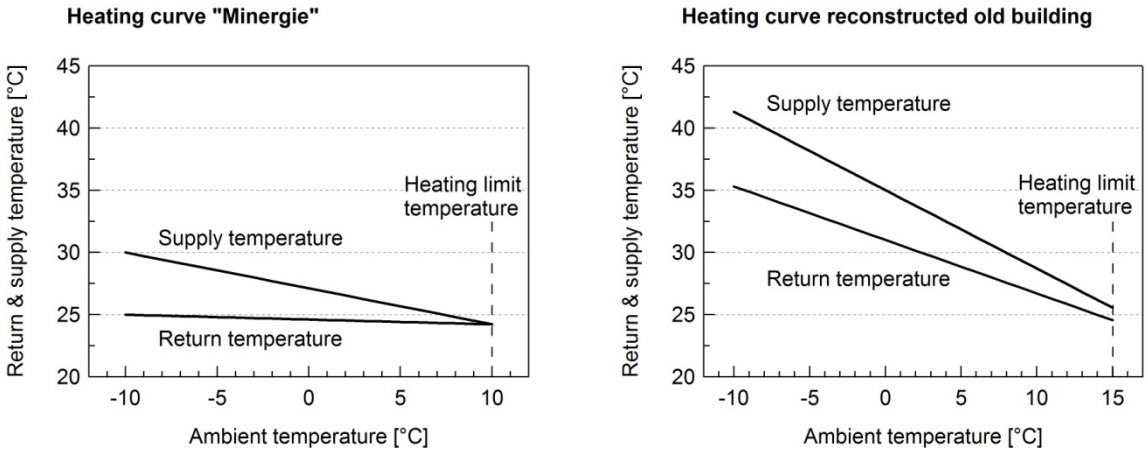


Figure 7: *Left*: Heating curve of a “Minergie” building.
***Right*: Heating curve of a reconstructed old building**

4.2 Capacity control using 1st Generation compressor

Heating curve of a “Minergie” building: Figure 8 (left) shows the required and generated heating capacity as a function of ambient temperature for the "Minergie" heating curve using the 1st Generation compressor. The A/W-HP prototype was capacity controlled so that the

efficiency was always at maximum value. The generated heating capacity at ambient temperatures in the range between -10°C and 2°C was adjusted continuously to the required heating capacity. In this range the part load ratio of the compressor varied from 100% at -10°C to about 28% at 2°C ambient temperature. Over the same temperature range the fan speed was controlled in a way that the air flow was reduced in an approximately linear manner from 4300 m^3/h to 2000 m^3/h. For ambient temperatures above 2°C, both the part load ratio of the compressor and the fan were no longer reduced since the efficiency of heat production was found to decrease. Therefore, the A/W-HP at ambient temperatures above 2°C needed to be operated in an on/off control mode (constant compressor part load ratio of 28%, constant air flow of 2000 m^3/h). As a result, the generated heating capacity above 2°C ambient temperature rises slightly.

Heating curve of a reconstructed old building: Figure 8 (right) shows the results for the reconstructed old building using the A/W-HP prototype with a compressor of the 1st Generation. Based on the heating curve it was found that the continuous adaption of the generated to the required heating capacity was possible over a range of ambient temperatures between -10°C and 6°C. In this range the part load ratio of the compressor varied from 100% at -10°C to about 30% at 6°C ambient temperature. Over the same temperature range the fan speed was reduced in a linear manner from 4300 m^3/h to 2000 m^3/h. In order to achieve the best COP valves, the A/W-HP at ambient temperatures above 6°C was operated in on/off control mode (constant compressor part load ratio of 30%, constant air flow of 2000 m^3/h).

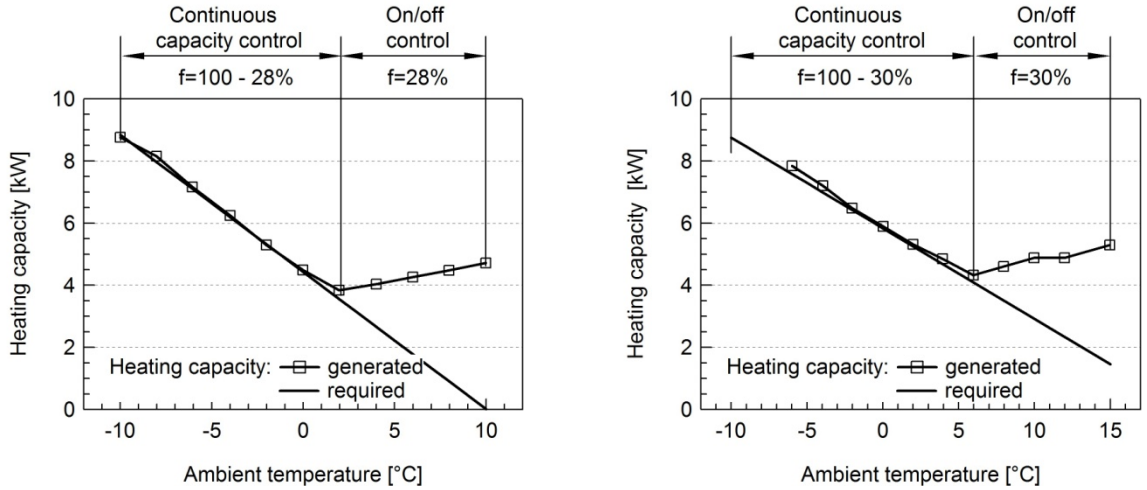


Figure 8: Heating capacity as a function of the ambient temperature (f is the part load ratio of the compressor)
***Left*: Heating capacity with the capacity controlled compressor of the 1st Generation based on a heating curve of a "Minergie" building**
***Right*: Heating capacity with the capacity controlled compressor 1st Generation based on a heating curve of a reconstructed old building**

4.3 Capacity control using 2nd Generation compressor

Heating curve of a "Minergie" building: Figure 9 (left) shows the results for the "Minergie" test case using the A/W-HP prototype with a compressor of the 2nd Generation. In order to achieve the best possible COP, continuous adjustment is only possible in a relatively small range of ambient temperatures. The continuous capacity control of the compressor and fan was determined to be only applicable in the ambient temperature range between -10°C and -2°C. In this range the part load ratio of the compressor varied from 100% at -10°C ambient temperature to about 54% at -2°C ambient temperature. Over the same temperature

range the fan speed was reduced in a linear manner from 3250 m^3/h to 2500 m^3/h. In addition, to improve the performance for ambient temperatures between -10°C to -4°C, vapour injection was used. However, at ambient temperatures above -4°C the vapour injection in the heating mode was disabled.

Heating curve of a reconstructed old building: Also for the heating curve of the reconstructed old building, the range of ambient temperatures was smaller, in which the continuous capacity control of the compressor of the 2nd Generation was beneficial, see figure 9 (right). In this case, the best COP through continuous control of the heating capacity was only possible for the ambient temperature range of -10°C to 2°C. In this range the part load ratio of the compressor varied from 100% at -10°C ambient temperature to about 51% at 2°C ambient temperature. Over the same temperature range the fan speed was controlled to reduce the air flow in a linear manner from 3400 m^3/h to 1900 m^3/h. However, compared to the A/W-HP prototype with the compressor of 1st Generation, this temperature range was found to be less, between -10°C and 2°C). This result is due to the clearly smaller region of efficient operating of the compressor of the 2nd Generation. Again, to improve the performance for the A/W-HP the compressor was operated with vapour injection for ambient temperatures between -10° C and -2°C.

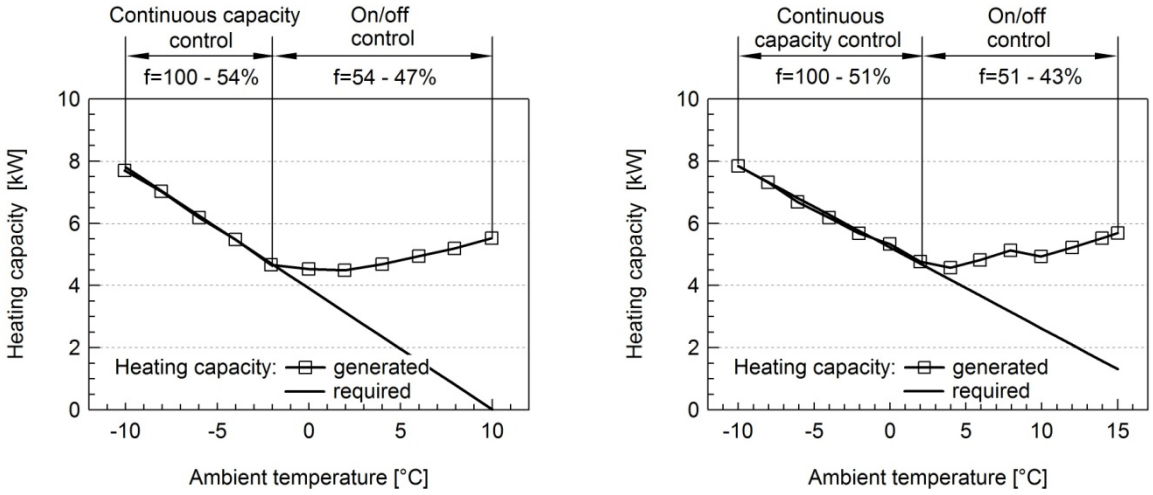


Figure 9: Heating capacity as a function of the ambient temperature (f is the part load ratio of the compressor)
***Left*: Heating capacity with the capacity controlled compressor of the 2nd Generation based on a heating curve of a "Minergie" building,**
***Right*: Heating capacity with the capacity controlled compressor of the 2nd Generation based on a heating curve of a reconstructed old building.**

4.4 COP and SPF

The achieved COP values, including the fan, for the A/W-HP prototypes based on a "Minergie" heating curve as a function of ambient temperature are shown in figure 10 (left). The resulting COPs for the variable speed compressor of the 2nd Generation above an ambient temperature of -2°C were found to be lower than those for the compressor of the 1st Generation. At ambient temperatures below -2°C the reverse situation was found and the COP of the compressor of the 2nd Generation was higher. This result shows that the compressor of the 2nd Generation, which is optimized specifically for the HP application, has the best performance in the case of higher temperature lifts. In the case of operation at higher ambient temperatures and correspondingly smaller temperature lifts, the COP does not increase to the same extent as when using the compressor of the 1st Generation.

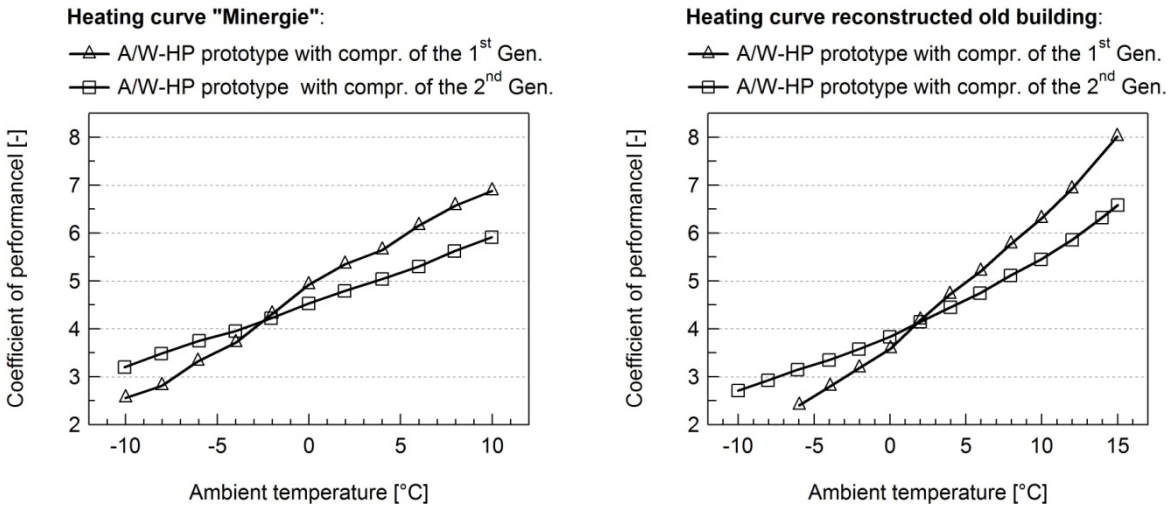


Figure 10: *Left*: COP based on a heating curve of a "Minergie" building
***Right*: COP based on a heating curve of a reconstructed old building**

The resulting COPs, including the fan, achievable with the A/W-HP prototypes for the heating curve of a reconstructed old building are shown in figure 10 (right). It was found that below ambient temperatures of 2°C higher COPs were obtained when using the compressor of the 2nd Generation compared to the 1st Generation compressor. However, the resulting COPs for ambient temperatures above 2°C were found to be below the values produced by the capacity controlled A/W-HP prototype using the compressor of the 1st Generation.

Table 1 shows the SPF, including the fan, of the capacity controlled A/W-HP prototype with the compressor for both the 1st and 2nd Generation compressor for the "Minergie" heating curve and that for the reconstructed old building for climatic conditions of Zurich. The method does not take into account defrosting (von Böckh et al. 2005). However, the SPF based on the heating curve for the reconstructed old building and compressor of the 1st Generation the COP at -10°C ambient temperature was extrapolated (Gasser et al. 2010).

Table 1: SPF, including the fan, for the capacity controlled A/W-HP prototypes with the compressors of the 1st and 2nd Generation for the "Minergie" heating curve and that for the reconstructed old building

Heating curve	Minergie		Reconstructed old building	
Heat pump prototype	1st Generation	2nd Generation	1st Generation	2nd Generation
SPF	**5.0**	**4.6**	**4.9**	**4.5**

The compressor of the 2nd Generation is especially optimized for HP applications and thus allows efficient operation at low ambient temperatures, i.e. high temperature lifts. Table 1 shows that the SPFs for the compressor of the 2nd Generation are slightly lower, since the distribution of the cumulative frequency at higher ambient temperatures in Zurich were more strongly weighted. The COP of the compressor of the 2nd Generation rises less significantly at higher ambient temperatures compared to the compressor of the 1st Generation, see figure 10. The capacity controlled A/W-HP prototypes with the compressors of both the 1st and 2nd Generation achieved SPFs well above the values of standard A/W-HPs running with on/off control. Today, typical SPF values for A/W-HPs are in the range of 3 to 3.5.

For comparison: the heat pump system was also investigated employing an on/off-controlled scroll compressor with constant rotational speed for the "Minergie" heating curve. In this case the SPF turned out to be 4.2 which is significantly below the values shown in table 1.

The COP and SPF values are comparatively close together for the two investigated heating curves. The reasons for this similarity are the wider range of capacity control for the heating curve of the reconstructed old building (upper limit +2°C instead of -2°C) and also the higher part load efficiencies at higher temperature lifts.

5 CONCLUSIONS

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. However, through the continuous adjustment of the generated to the required heating capacity the efficiency of A/W-HPs can be increased considerably. A necessary 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 (Gasser et al. 2010).

Additional potential for an increased efficiency of A/W-HPs can be achieved by defrosting with continued ventilation. The key to defrost with continued ventilation is the state of the ambient air. In addition to the air`s temperature the air humidity plays an important role and must be considered mandatory. To exploit this potential an appropriate defrosting process is developed that is capable of determining the optimum time for initiating the defrosting process, of choosing the optimal defrosting process and of determining the process at the optimum time (Gasser et al. 2010).

6 ACKNOWLEDGEMENTS

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