EXERGY ANALYSIS: STALKING THE POTENTIAL FOR MORE EFFICIENT AIR/WATER HEAT PUMPS

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Abstract: In this study, the behavior of air/water heat pumps over a wide range of operating conditions is investigated using the exergy method. An analytical model based on the relevant thermophysical properties has been developed to simulate the system at steady state and to evaluate the exergetic efficiency. The exergy losses of the main process components and the heating delivery and distribution system are discussed with mathematical equations demonstrating the influence of important variables. Theoretical and experimental analysis indicate that the exergetic efficiency of common heat pumps become smaller when the ambient air temperature increases. The reason for this is the heat pump characteristic: the less heat is required the more it delivers. In common heat pumps this is compensated by on/off operation. As a consequence, a substantial increase in efficiency requests continuous operation with power control of compressor and where necessary of the fan: The generated heat flow of the heat pump shall be continuously adapted to the required one.

Key Words: ***heat pump, exergy, exergy losses, heat pump characteristic, exergetic efficiency, control strategies***

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 require only low investments, are easy to install and are reliable in their operation. They are not subject to official approval and in the future will run with less noise as a result of improved flow geometries and optimized fans. However one of the most important prerequisite for the increased proliferation of heat pumps lies in the substantial increase of their efficiency.
The coefficient of performance (COP) is approximately inversely proportional to the temperature lift; the exergetic efficiency of the air/water heat pump with on/off control, on the other hand, decreases with increasing ambient temperature and the lower temperature lift associated with it. The reason for this is the heat pump operation characteristic: In the case of conventional air/water heat pumps with on/off control, the generated heating capacity, paradoxically, increases with increasing ambient temperature and the associated decrease in required heating capacity.
The WEXA study (Gasser *et al.* 2008) was started as a result of the extensive LOREF research project (Berlinger *et al.* 2008), in which it was increasingly realised that, in order to achieve an important increase in the efficiency of air/water heat pumps, all sub-processes and the heating system must be included in the analysis.
The 2nd law of thermodynamics describes the quality of thermodynamic processes. It states that all technical processes are irreversible. In thermodynamics, one must not speak of energy losses (in accordance with the 1st law) but may speak of irreversible increases in

entropy. Instead of evaluating the quality of thermodynamic processes with abstract entropy balances, exergy analysis is suitable for our purposes. Its results are easy to follow and immediately demonstrate possibilities for improvement in thermodynamic efficiency because, using exergy analyses, exergy losses can be determined for both the complete system and their subsystems. For heating and cooling systems, exergy losses always mean that additional mechanical work is necessary in comparison with ideal process in which no exergy losses occur.
In this study, exergy balances will be calculated for the whole heat pump process and all sub-processes. In order to do this, we aim to clearly show the exergy losses mathematically in relation to the relevant process variables and their effect on other sub-processes. For this purpose, suitable approximations will be made, e.g. where permissible by linearisation or by series expansions.

2 THE RELEVANT FACTORS OF THE EXERGY LOSSES IN HEAT PUMPS

The basic factors influencing the exergy losses of the heat pumps are the temperature gradients for heat transfer in the evaporator ΔT_E and condenser ΔT_C as well as the isentropic compressor efficiency η_s. 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. The temperature lift ΔT_{Lift} is given by:

$$\Delta T_{Lift} = T_C - T_E = (T_R - T_A) + \Delta T_H + \Delta T_R + \Delta T_E + \Delta T_C = \Delta T_{Lift\,min} + \Delta T_H + \Delta T_R + \Delta T_E + \Delta T_C \quad (1)$$

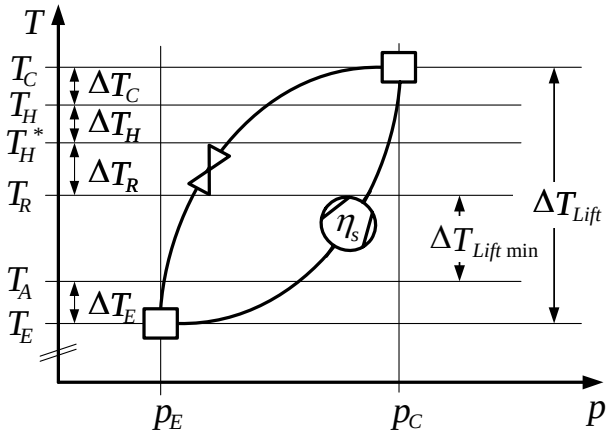


Figure 1: Temperatures, temperature gradients and pressures in the heat pump process

For air/water heat pumps with on/off control the intermittently generated heating (water) temperature T_H, at ambient temperatures above the design point is around ΔT_H higher than the heating temperature T_H^* continuously required by the building. The heat with temperature 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 ΔT_R. The temperature lift ΔT_{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 shift $\Delta T_{Lift\,min}$ by the temperature gradients ΔT_E, ΔT_C and ΔT_R as well as by the discrepancy ΔT_H between required and generated heating temperature (cf. Equation (1)).

3 EXERGY LOSSES OF THE HEATING SYSTEM WITH A HEAT PUMP

3.1 Exergy Losses of the Sub-processes of the Heat Pump Working Fluid

The exergy losses in a simple, single-stage compression 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 pressure drops. On the other hand, the exergy losses during heat transfer in evaporator and condenser do not originate in the working fluid but on account of the temperature gradients ΔT_E and ΔT_C required for heat transfer. Pressure losses due to flow in piping and fittings as well as in the evaporator and condenser on the working fluid side are negligible small and are not considered. Also, the drive losses of the compressor as well as the fan power are not considered here.
Physical equations will now be developed in order to show the quantitative effect of the relevant factors on exergy losses and exergetic efficiency. By making approximations, relatively clear equations can be established that can easily be discussed without accuracy being affected.
The derivation of the equations for the calculation of the exergy losses is presented here in an exemplary way for the compressor. The remaining equations are not derived in detail here; they will, however, be discussed in each case (the detailed derivations can be found in WEXA (Gasser *et al.* 2008).
In Figure 2 the heat pump process is represented by a detailed representation of the individual state points of the heat pump's working fluid in the T,s diagram. Evaporation and condensation are taken as being isobar-isotherm. In the evaporator, vapour superheating ΔT_{cGsh} occurs. In the compressor, vapour superheating ΔT_{hGsh} with reference to T_C occurs along with condensate subcooling ΔT_{Csc}.

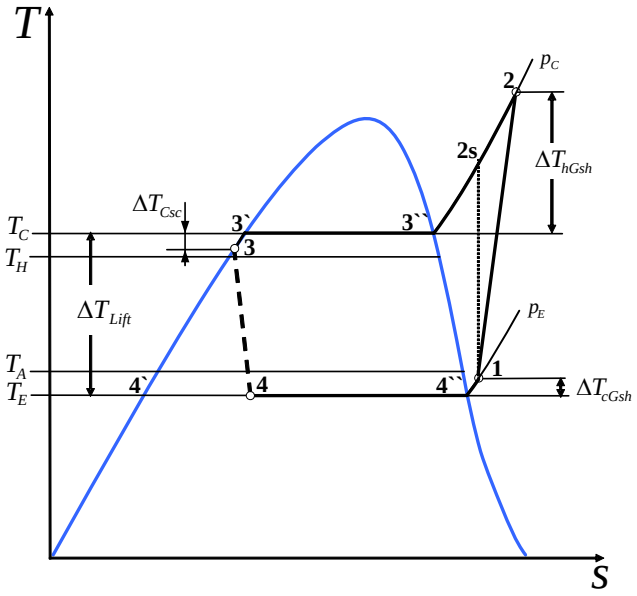


Figure 2: T,s diagram with state points noted (cGsh: cold gas superheating; hGsh: hot gas superheating; Csc: condensate subcooling)

3.1.1 Exergy losses in the compressor

We refer to an adiabatic, single-stage compressor. The isentropic compressor efficiency η_s is particularly important for the exergy loss in the compressor. The following is valid for the exergy loss flow in the compressor:

$$\dot{E}_{LCp} = \dot{m}_f \cdot T_A \cdot (s_2 - s_1) \tag{2}$$

We now treat the vapour in the compressor as a perfect gas with the corresponding averaged gas properties c_p, R and κ. Thus, the following is valid for the irreversible increase in specific entropy in the compressor:

$$s_2 - s_1 = \left[c_{pg} \cdot \ln\left(\frac{T_2}{T_1}\right) - R \cdot \ln\left(\frac{p_2}{p_1}\right) \right] \tag{3}$$

And the consequent exergy loss flow in the compressor is:

$$\dot{E}_{LCp} = \dot{m}_f \cdot T_A \cdot \left[c_{pg} \cdot \ln\left(\frac{T_2}{T_1}\right) - R \cdot \ln\left(\frac{p_2}{p_1}\right) \right] \tag{4}$$

The compressor exit temperature T_2 is dependent both on the pressure ratio $\varphi = p_2 / p_1$ as well as on the isentropic efficiency η_s of the compressor:

$$T_2 = T_1 \cdot \left[\frac{1}{\eta_s} \cdot \left(\varphi^K - 1\right) + 1 \right] \tag{5}$$

whereby K is defined as:

$$K = \frac{\kappa - 1}{\kappa} \tag{6}$$

Here, with p_1 the evaporation pressure p_E and with p_2 the condensation pressure p_C is meant. This means that the pressure ratio is:

$$\varphi = \frac{p_2}{p_1} = \frac{p_C}{p_E} \tag{7}$$

As, primarily, the temperatures T_E and T_C and not the pressures p_E and p_C are the decisive factors in the dimensioning of the heat pump process, is it sensible that in our analysis the vapour pressures should be expressed by their boiling point temperatures. In order to be able to discuss the results analytically, we will mathematise the vapour pressure curve using the Clausius-Clapeyron relation. The following is valid (r = specific enthalpy of evaporation of the working fluid):

$$p = p_0 \cdot e^{\frac{r}{R}\left(\frac{1}{T_0} - \frac{1}{T}\right)} \tag{8}$$

By insertion in Equation (7), the pressure ratio as a function of the relevant process temperatures T_E and T_C is obtained:

$$\varphi = \frac{p_C}{p_E} = e^{\frac{r}{R}\left(\frac{1}{T_E} - \frac{1}{T_C}\right)} = e^{\frac{r}{R}\left(\frac{T_C - T_E}{T_E \cdot T_C}\right)} \approx e^{\frac{r}{R} \cdot \frac{\Delta T_{Lift}}{T_A^2}} \tag{9}$$

Using Equations (5) and (9) in (4), the following is valid for the exergy loss flow in the compressor:

$$\dot{E}_{LCp} = \dot{m}_f \cdot T_A \cdot \left(c_{pg} \cdot \ln\left[\frac{1}{\eta_s} \cdot e^{\frac{K \cdot r}{R} \cdot \frac{T_C - T_E}{T_C \cdot T_E}} - \frac{1}{\eta_s} + 1 \right] - r \cdot \frac{T_C - T_E}{T_C \cdot T_E} \right) \quad (10)$$

Making a simplification concerning the working fluid in the compressor by treating it as a perfect gas and with the idealisation of the vapour pressure curve according to Clausius-Clapeyron, reliable results can be obtained. For a clear interpretation we approximate the logarithmic and the exponential functions with series expansions. In order to achieve an important simplification, the series expansions must be broken off after the first term. Moreover, the following identity is employed.

$$\frac{c_p}{R} = \frac{\kappa}{\kappa - 1} = \frac{1}{K} \quad \Rightarrow \quad \frac{c_p \cdot K}{R} = 1 \quad (11)$$

Using this, the essential interrelationship with respect to exergy loss flow in an adiabatic, single-stage compressor can be clearly shown.

$$\dot{E}_{LCp} = \dot{m}_f \cdot T_A \cdot r \cdot \frac{\Delta T_{Lift}}{T_E \cdot (T_E + \Delta T_{Lift})} \cdot \left[\frac{1}{\eta_s} - 1 \right] \approx \dot{m}_f \cdot r \cdot \frac{1}{COP_{revi}} \cdot \left[\frac{1}{\eta_s} - 1 \right] \quad (12)$$

The exergy loss in the adiabatic compressor is approximately proportional to the specific enthalpy of evaporation r and to the temperature lift ΔT_{Lift}. In which ΔT_E, ΔT_C, ΔT_R and ΔT_H are included (cf. Equation (1)). Also, the effect of η_s is immediately clear - see the square brackets in Equation (12).

3.1.2 Exergy losses in the expansion valve

Because of the pressure reduction in the expansion valve from the condensation pressure p_C to the evaporation pressure p_E, the temperature of the working fluid falls from T_3 to T_4 and the specific entropy increases from s_3 to s_4. The following is valid for the exergy loss flow in the expansion valve:

$$\dot{E}_{LEx} = \dot{m}_f \cdot T_A \cdot (s_4 - s_3) \quad (13)$$

The specific entropy difference $s_4 - s_3$ and, therefore, the exergy loss flow in the expansion valve, can, whilst taking the specific heat capacity c_{pl} of the boiling liquid working fluid into consideration, be determined as a function of the relevant process temperatures (Gasser *et al.* 2008):

$$\dot{E}_{LEx} = \dot{m}_f \cdot T_A \cdot c_{pl} \cdot \left[\ln\left(\frac{T_E}{T_E + \Delta T_{Lift} - \Delta T_{Csc}} \right) + \frac{\Delta T_{Lift} - \Delta T_{Csc}}{T_E} \right] \quad (14)$$

Here, too, the logarithmic function will be further approximated by a series expansion in order to obtain a clear interpretation:

$$\dot{E}_{LEx} \approx \dot{m}_f \cdot T_A \cdot c_{pl} \cdot \frac{1}{2} \cdot \left(\frac{\Delta T_{Lift}^{\,2}}{T_E^{\,2} + T_E \cdot \Delta T_{Lift}} \right) \approx \dot{m}_f \cdot c_{pl} \cdot \frac{\Delta T_{Lift}}{2 \cdot COP_{revi}} \quad (15)$$

Consequently, the exergy loss in the expansion valve is proportional to c_{pl} and to the square of the temperature lift. In practice, ΔT_{Lift} can be influenced by ΔT_E, ΔT_C, ΔT_R and ΔT_H (cf.

Equation (1)). The more exact equation (14) shows that by increasing condensate subcooling ΔT_{Csc} the exergy losses in the expansion valve can be reduced.

3.1.3 Exergy losses in the evaporator

The exergy loss flow in the evaporator can be roughly calculated using the following equation:

$$\dot{E}_{LE} = \dot{Q}_E \cdot T_A \cdot \frac{\Delta T_E}{\overline{T}_E \cdot \left(\overline{T}_E + \Delta T_E\right)} \approx \dot{Q}_E \cdot T_A \cdot \frac{\Delta T_E}{\overline{T}_E^{\,2}} \approx \dot{Q}_E \cdot \frac{\Delta T_E}{T_A} \tag{16}$$

The exergy loss flow in the evaporator is proportional to the heat flow $\dot{Q}_E$ transferred, to the temperature gradient ΔT_E as well as being inversely proportional to the square of the average evaporation temperature $\overline{T}_E$.
In order to determine the exergy loss in the evaporator through the transfer of heat more precisely, e.g. by taking the temperature glide of the working fluid and the superheating into consideration, one must base on the differential exergy loss. Consequently, it follows that:

$$\dot{E}_{LE} = \int d\dot{E}_{LE} = T_A \cdot \int \frac{T_{Air} - T_E}{T_{Air} \cdot T_E} \cdot d\dot{Q} \approx T_A \cdot \sum_{i=1}^{n} \frac{T_{Air,i} - T_{E,i}}{T_{Air,i} \cdot T_{E,i}} \cdot \Delta \dot{Q}_i \tag{17}$$

3.1.4 Exergy losses in the condenser

Similarly to Equation (16), the following is valid for the calculation of the exergy loss flow in the condenser:

$$\dot{E}_{LC} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H \cdot \overline{T}_C} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H \cdot \left(T_H + \Delta T_C\right)} \approx \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_C}{T_H^{\,2}} \tag{18}$$

The exergy loss in the condenser is proportional to the generated heating capacity $\dot{Q}_H$, the temperature gradient ΔT_C for heat transfer as well as being inversely proportional to the square of the generated heating temperature T_H. For more precise investigation, it would be advantageous if the condenser could be divided into several sections (elimination of vapour superheating, condensation, condensate subcooling).

3.2 Exergy Losses in the Heat distribution and delivery System

If the heating water circuit (heat distribution and delivery system) is also considered, two further exergy losses occur in addition to the exergy losses of the four elementary sub-processes of the compression heat pump: In the heat distribution system $\dot{E}_{LHS}$ and in the heat delivery system $\dot{E}_{LR}$. The equations for the calculation of these exergy losses are listed below. The detailed derivation of the individual equations can be found in WEXA (Gasser *et al.* 2008).
Exergy loss in the heat distribution system:

$$\dot{E}_{LHS} = \dot{Q}_H \cdot T_A \cdot \frac{T_H - T_H^{\,*}}{T_H \cdot T_H^{\,*}} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_H}{T_H \cdot T_H^{\,*}} \tag{19}$$

Exergy loss in the heat delivery system:

$$\dot{E}_{LR} = \dot{Q}_H \cdot T_A \cdot \frac{T_H^* - T_R}{T_H^* \cdot T_R} = \dot{Q}_H \cdot T_A \cdot \frac{\Delta T_R}{T_H^* \cdot T_R} \quad (20)$$

The exergy loss in the heat distribution system results from the discrepancy caused by the generated heat flow being higher as that required and, corresponding, because the generated heating temperature T_H is higher than that required T_H^*. This exergy loss is, for a generated heat flow of $\dot{Q}_H$, directly proportional to the discrepancy between the generated and required heating temperatures ΔT_H. The cause of this exergy loss lies in the unfavourable operating characteristic of the air/water heat pump with on/off control. For the purpose of thermodynamic evaluation, this exergy loss is, therefore, to be attributed to the air/water heat pump. On the other hand, the heat pump may not be charged with causing the exergy loss $\dot{E}_{LR}$ in the heat delivery system; the heating system itself must take responsibility for this loss. $\dot{E}_{LR}$ is proportional to the temperature gradient ΔT_R for heat transfer to the room. The temperature gradient ΔT_R and, therefore, the exergy loss $\dot{E}_{LR}$ can be influenced during the planning and design phase of the heat delivery system. The aim here is to minimise the temperature gradient by using low required heating temperatures and thus, additionally, reducing the temperature lift.

4 OPERATING CHARACTERISTIC OF AIR/WATER HEAT PUMPS WITH ON/OFF CONTROL

The operating characteristics, the exergy loss ratios and the exergetic efficiencies presented below are valid for an air/water heat pump with on/off control with 5.4 kW nominal heating capacity at $\vartheta_{A\,min} = -10$ °C. Detailed specifications of this heat pump can be found in WEXA (Gasser *et al.* 2008). These simulations were carried out, however, in great detail using a simulation program for air/water heat pumps which was developed and documented in LOREF (Sahinagic *et al.* 2008).

With increasing ambient temperature, the heating capacity required by the building decreases. The behaviour of common heat pumps, whose compressor is operated at constant rotational speed, is, on the other hand, exactly the opposite: The lower the heating capacity and heating temperature required by the building, the higher the heating capacity and heating temperature actually generated. Figure 3 shows the resultant discrepancy between the continuously required heating capacity and that intermittently generated.

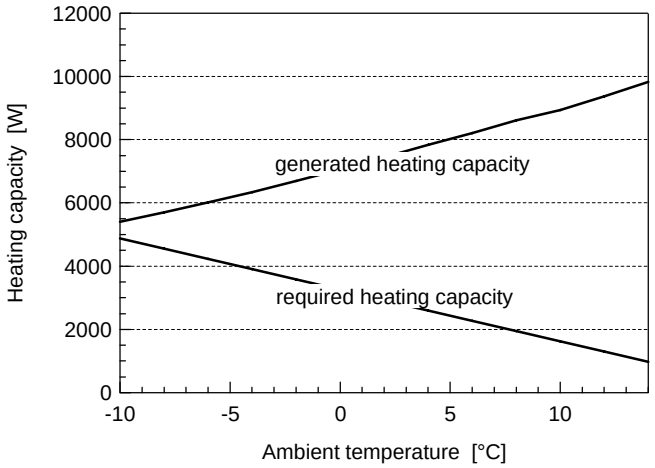


Figure 3: Generated and required heating capacities as a function of ambient temperature

This behaviour results in common heat pumps working in intermittent mode (on/off control). Moreover, the temperature gradients for heat transfer in the evaporator and condenser increase; also, the discrepancy between required and generated heating temperature in the heat distribution system increases with increasing ambient temperature, thus leading to bad part-load efficiencies, lower seasonal performance factors and lower annual average exergetic efficiencies. Figure 4 shows the discrepancy between required and generated heating temperatures.

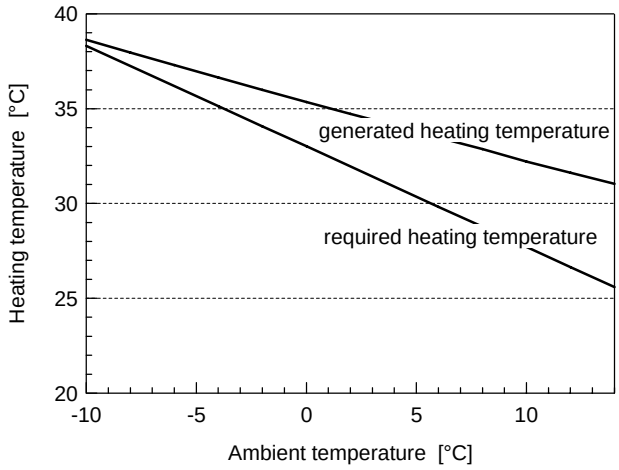


Figure 4: Generated and required heating temperature for an air/water heat pump with on/off control

5 DEFINTION OF EXERGETIC EFFICIENCIES

In order to evaluate the quality of a (irreversible) heat pump thermodynamically correctly, the exergetic η_{ex} is introduced: the exergy flow of heat for heating purposes, which can be calculated as a function of the temperature of a given heat flow, is related to the internal compressor power P_i (the mechanical and electrical efficiencies of the compressor as well as the fan power are not considered here).
The internal exergetic efficiency is used to judge the thermal efficiency of the heat pump's working fluid circuit with the fluid temperatures T_E in the evaporator and T_C in the condenser:

$$\eta_{exi} = \frac{\dot{Q}_H \cdot \frac{T_C - T_E}{T_C}}{P_i} = COP \cdot \eta_{Ci} \tag{21}$$

The internal exergetic efficiency considers the total temperature lift $\Delta T_{Lift} = T_C - T_E$ and therefore includes the temperature gradients in the evaporator and condenser for heat transfer as well as the discrepancy between the required and generated heating temperatures but not, however, their exergy losses. Consequently, it is influenced by the exergy losses in the expansion valve and compressor only in accordance with the pressure ratio corresponding to the total temperature lift.
For the external exergetic efficiency of the generated heating temperature T_H, the exergy of the generated heating capacity $\dot{Q}_H$ is referred to the generated temperature of the heating water T_H in the condenser. It considers in addition to the internal exergetic efficiency the exergy losses caused by the temperature gradients for heat transfer in the evaporator and condenser.

$$\eta_{exe} = \frac{\dot{Q}_H \cdot \frac{T_H - T_A}{T_H}}{P_i} = COP \cdot \eta_{Ce} \tag{22}$$

The external exergetic efficiency of the required heating temperature serves as third means of judgement: in this case, the exergy of the generated heating capacity $\dot{Q}_H$ is put into relation to the heating temperature T_H^* continuously required by the building. It considers in addition to the external exergetic efficiency of the generated heating temperature the exergy losses caused by the discrepancy between the required and generated heating temperature (exergy losses in the heat distribution system).

$$\eta_{exe}^* = \frac{\dot{Q}_H \cdot \frac{T_H^* - T_A}{T_H^*}}{P_i} = COP \cdot \eta_{Ce}^* \tag{23}$$

As in Equation (23), a fourth evaluation of the external exergetic efficiency of a heating system with a heat pump (exergetic efficiency of the heated building) is referred to the ambient temperature T_A along with the room temperature T_R required:

$$\eta_{ex\,HS} = \frac{\dot{Q}_H \cdot \frac{T_R - T_A}{T_R}}{P_i} = COP \cdot \eta_{C\,HS} \tag{24}$$

In addition to the external exergetic efficiency of the required heating temperature, this also takes the exergy losses occurring in the heat delivery system (e.g. underfloor heating) into consideration. These are not to be attributed to the heat pump.

6 RESULTS OF THE EXERGY ANALYSIS FOR AIR/WATER HEAT PUMPS WITH ON/OFF CONTROL – WITHOUT DRIVE LOSSES

6.1 Comparison of the Exergy Loss Ratios

The development engineer is interested both in the magnitude of the individual exergy loss flows as well as in comparisons made between them. A clear appraisal can successfully be made by using the exergy loss ratios. This is defined here as the appropriate exergy loss flow with reference to the internal compressor power P_i. The reason for this is that one can so directly follow the subtractive effect on exergetic efficiency. This can be calculated from the several exergy loss ratios:

$$\eta_{exe} = 1 - \frac{\dot{E}_{LCp}}{P_i} - \frac{\dot{E}_{LC}}{P_i} - \frac{\dot{E}_{LEx}}{P_i} - \frac{\dot{E}_{LE}}{P_i} \tag{25}$$

In Figure 5, the curves of the exergy loss ratios are compared. They are referred to the internal power P_i of a heat pump with a scroll compressor (in accordance with the specification in WEXA (Gasser *et al.* 2008)).

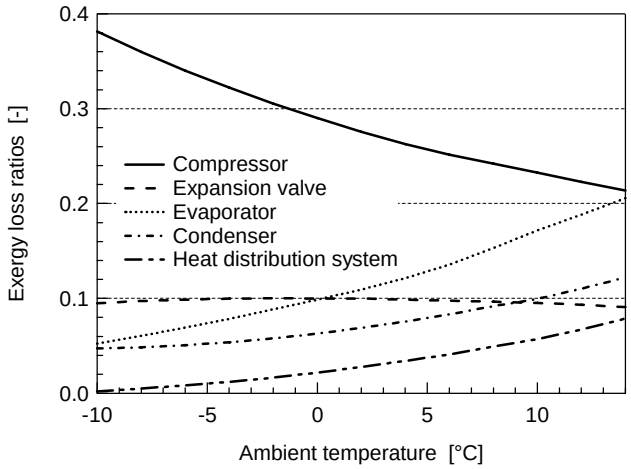


Figure 5: Overview of the exergy loss ratios of an air/water heat pump with on/off control

The exergy loss ratio of the compressor is quite predominant. At outdoor temperatures below 0°C it is even four to eight times as high as the other three sub-processes. The exergy loss ratio is 38% at -10°C and then decreases to 22% at 15°C, but is still much higher than the other losses. Here, an important potential for making improvements in the co-operation between manufacturers of heat pumps and compressors still exists.
The exergy loss ratios for the heat transfer in the evaporator and condenser at -10°C is only 5%. These could no doubt be halved again by using a larger heat exchanger surface and, in addition, with the use of electronically controlled expansion valves. Decisively dramatic, however, is the increase in the exergy loss ratios: at 14°C it is 13% in the condenser and even 20% in the evaporator – still without frost formation. The consequences of this are shown in Figure 6.
Finally, the exergy loss ratio of the expansion valve: it is always around 10% and is thus comparatively high. For heat pumps that are designed for low temperature lift, the exergy loss flow and exergy loss ratio must become more favourable as $E_{LEx} \sim \Delta T_{Lift}^{2}$.
Also, the exergy loss ratio in the heat distribution system increases quasi progressively with increasing outdoor temperature: from 0% at -10°C to 8% at 14°C.

6.2 Comparison of the Exergetic Efficiencies

In Figure 6, the curves for exergetic efficiencies defined in Section 5 are compared. The electrical and mechanical efficiencies of the compressor as well as the fan power are still not considered here.
The curves show that the internal exergetic efficiency increases somewhat with increasing ambient temperature. The reason for this is the decrease of temperature lift.
Figure 5 shows that the exergy losses in the evaporator and condenser increase quasi progressively with increasing ambient temperature because of the increasing temperature gradients required for heat transfer. As can be seen in Figure 6 this results in the curves for the external exergetic efficiency of the generated heating temperature.
The external exergetic efficiency of the required heating temperature falls with increasing outdoor temperature even more strongly than the external exergetic efficiency of the generated heating temperature. Reason for this is the exergy loss in the heat distribution system, which increases with increasing outdoor temperature and is added on.
The curve for the external exergetic efficiency of a heating system with a heat pump takes the exergy losses of the heat transfer from the heating water circuit to the room in addition to the external exergetic efficiency of the required heating temperature into consideration and is once more much lower.

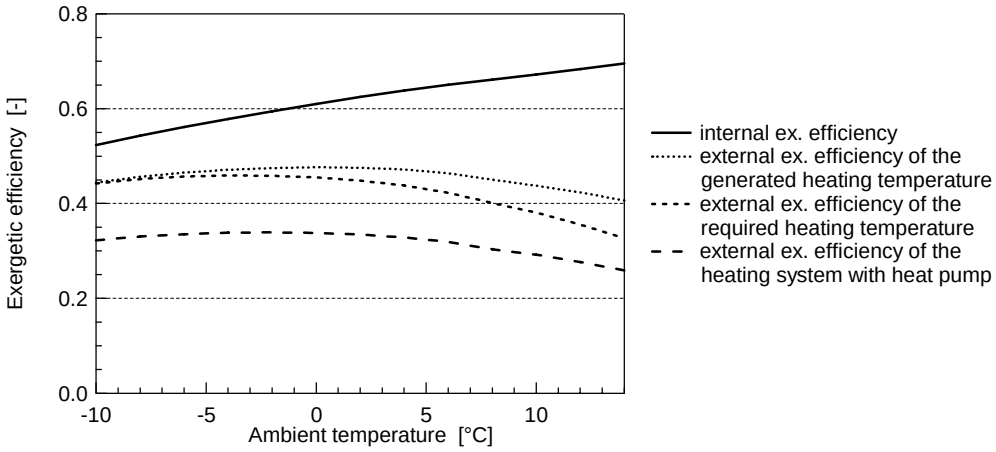


Figure 6: Overview of the exergetic efficiencies of an air/water heat pump with on/off control

6.3 Findings

The most important findings are briefly summarised below:

- The exergy losses of all four sub-processes of the air/water heat pump with on/off control are adversely influenced by the temperature gradients ΔT_E and ΔT_C required for heat transfer as well as by the discrepancy between required and generated heating temperatures ΔT_H.
- The temperature gradients ΔT_E and ΔT_C have a direct influence on the exergy losses in evaporator and condenser. They also have an indirect effect on the expansion valve and the compressor.
- The temperature gradients ΔT_E and ΔT_C in the evaporator and condenser that increase with increasing ambient temperature and also ΔT_H in the heat distribution system are caused by the discrepancy between required and generated heating capacities.
- The unfavourable operating characteristics of on/off controlled air/water heat pumps (which are driven with a constant rotation speed compressor) are the cause for generated heating capacities that increase with increasing ambient temperature as well as for generated heating temperatures that are too high.
- The air/water heat pump process can be clearly improved by the reduction of the temperature gradients in the evaporator and condenser as well as by adaptation of the generated heating temperature to the required heating temperature.
- By reducing the required supply temperature, the temperature gradient ΔT_R, the required heating temperature, the temperature lift and, therefore, the exergy losses in the compressor, expansion valve and the heat delivery system can be reduced. As a result, the internal exergetic efficiency as well as the external exergetic efficiency of heating systems using heat pumps can be improved.
- The compressor and where necessary the fan should be equipped with continuous power control to avoid the discrepancy between required and generated heating capacity and heating temperature and, as a result, effectively reduce the temperature gradients for heat transfer in evaporator and condenser. The air/water heat pump should no longer operate in intermittent on/off mode but continuously - cf. WEXA (Gasser *et al.* 2008).

- If only the compressor in air/water heat pumps is equipped with continuous power control, the efficiency of the fan has a decisive influence on the seasonal performance factor and the annual average exergetic efficiency to be achieved (Gasser *et al.* 2008).
- If, in addition to the compressor, the fan is also continuous power controlled, the influence of the fan efficiency becomes much lower in comparison to the sole control of the compressor (Gasser *et al.* 2008).
- If, for an air/water heat pump with continuous power control of the compressor, a fan with a sufficiently high efficiency is used, considerably better seasonal performance factors and annual average exergetic efficiencies can be obtained using this control strategy than with on/off control of the heat pump. The choice of the fan plays, therefore, an important role here (Gasser *et al.* 2008).
- Best seasonal performance factors and annual average exergetic efficiencies can be achieved through the simultaneous power control of the compressor and the fan (Gasser *et al.* 2008).

7 CONCLUSION AND OUTLOOK

It was mentioned that 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 instead of increasing in the case of partial load, i.e. for increasing ambient temperature, any discrepancy between required and generated heating capacity and heating temperature must be avoided. Consequently, the air/water heat pump should not be operated in on/off mode, but be continuously in operation - except during any defrosting processes necessary. Different strategies are available for the adaptation of the generated heating capacity to that required: the compressor can, for example, be equipped with speed control.
Considerably higher exergetic efficiencies can be achieved by using continuous power control of the compressor and where necessary of the fan. At 14°C, the external exergetic efficiency of the required heating temperature for a continuously power-controlled heat pump is about twice as high as in the case of the air/water heat pump with on/off control - cf. WEXA (Gasser *et al.* 2008).

8 REFERENCES

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