

DEFROSTING THE FIN TUBE EVAPORATORS OF AIR/WATER HEAT PUMPS USING AMBIENT AIR

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Abstract: Air/water heat pumps for heating systems are becoming increasingly popular. A major challenge for these systems is the formation of ice and frost in the fin-tube evaporator. Defrosting is usually accomplished by the reverse-cycle or hot-gas defrosting technique. In both defrosting modes, the compressor is in operation. An alternative method is ambient-air defrosting, where the heat pump's compressor is simply turned off and defrosting is achieved by exposure to ambient air, which is conveyed through the fin tube evaporator by the fan. A model has been developed which permits the calculation of the simultaneous heat and mass transfer during the melting of ice and frost deposits. The theoretical results have been validated by numerous defrosting experiments. The theoretical and experimental investigations revealed that both temperature *and* humidity of the ambient air strongly influence the total heat flow and the time needed for defrosting. Already at an ambient air temperature of 2°C and a dew point temperature of 0°C, the heat flow of ambient-air defrosting is comparable to that of hot-gas defrosting; energy consumption is decreased by a factor of 8–10, however. On the other hand, ambient-air defrosting is less effective at air conditions below the extended fog-isotherm at 0°C. The application of this technique can significantly reduce defrosting energy consumption and improve the overall efficiency of air/water heat pumps.

Key Words: *air/water heat pump, defrosting using ambient air, simulation*

1 INTRODUCTION

Due to cooling in the evaporator of air/water heat pumps (A/W-HP) constructed as a fin tube heat exchanger, the water vapour in moist ambient air is precipitated. At fin temperatures below the freezing-point of water, this water is deposited in the form of ice or frost on the fins and tubes of the evaporator. The ice or frost layer thus built up reduces the free cross-section for the air passing through. Accordingly, the air flow rate decreases, which leads to a decline of the simultaneous heat and mass transfer and, as a result, the efficiency of the whole heat pump is reduced; periodic defrosting processes are therefore required. The ice and frost formation in the evaporator of an on/off controlled A/W-HP is highest in the neutral temperature range between approximately 2-8°C.

Today, defrosting is mainly achieved using reverse cycle or hot-gas defrosting. In both cases, the compressor of the heat pump remains running, resulting in the defrosting process being correspondingly energy-intensive. An alternative way of removing ice and frost is defrosting with ambient air. Heating operation is interrupted by stopping the compressor while the fan further provides for the continuing flow of ambient air through the evaporator. The heat required for melting is provided by heat and mass transfer from ambient air to the ice or frost layer.

In the LOREF research project (Berlinger *et al.* 2008), a mathematical-physical simulation program was developed, with which the simultaneous heat and mass transfer in the evaporator was simulated and thus permitted the systematic examination of the ambient-air defrosting method as a function of the air's state. This model was validated by means of experimental investigations made using a commercial A/W-HP.

2 DEFROSTING PROCEDURES AND METHODS (STATE OF THE ART)

The defrosting procedures for the evaporators of A/W-HP most widespread in practice today are hot-gas and reverse cycle defrosting. Today, some manufacturers already offer A/W-HP that defrost at sufficiently high outdoor temperatures using ambient air which is conveyed through the evaporator by the fan. Detailed research on various defrosting procedures were made in the course of the "Verbesserung des Abtauens bei luftbeaufschlagten Verdampfern" project that was carried out in three phases (Hubacher *et al.* 2000, Bertsch *et al.* 2002, Hubacher *et al.* 2004). In particular, phase 3 is to be mentioned, in which defrosting with ambient air was experimentally examined (Hubacher *et al.* 2004). Using theoretical and experimental methods, Unternaehrer (2006) demonstrated the limits of defrosting with ambient air. Here, it was made clear that the influence of humidity on the defrosting operation cannot be neglected.

3 MATHEMATICAL-PHYSICAL MODELING

3.1 Balance equations

The surface-segment of a fin tube with an ice or frost layer used for the derivation of the equations used subsequently is shown in figure 1. It is assumed that a water film has been formed by the melting of the ice or frost layer. The water film consists only of melted ice or frost or of melted ice and frost together with water deposited from water-vapour in the moist air (condensate), this depending on the state of the ambient air. The following simplifications are made:

- The heat and mass fluxes are constant over the length l of the element.
- The temperature gradients within the melted water film and the ice or frost layer are neglected.
- The energy needed to heat up the ice to 0°C is neglected. The ice or frost layer as well as the water film have a constant temperature of $\vartheta_F = \vartheta_W = \vartheta_B = 0^\circ\text{C}$.

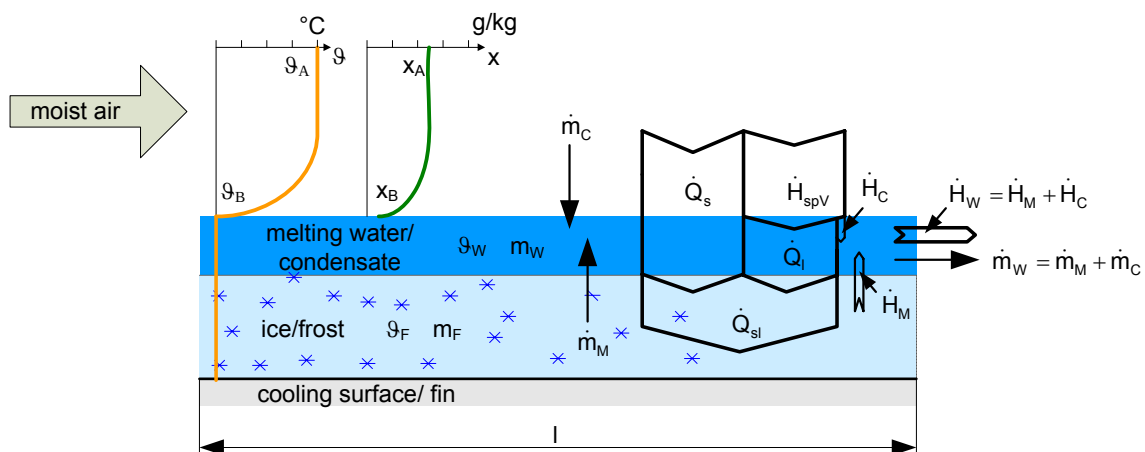


Figure 1: Surface segment of a fin with ice or frost and water film in the case of defrosting with ambient air.

For the situation shown in figure 1, with positive humidity gradient from the air to the boundary layer ($x_A > x_B$), the following mass and energy balance equations apply.

Balance boundary around the air flow:

The condensate mass flow $\dot{m}_C$ can be determined from the mass flow of dry air $\dot{m}_{Ad}$ and the difference in absolute humidity x between the input and output of the surface segment I:

$$\dot{m}_C = \dot{m}_{Ad} \cdot (x_{Ai} - x_{Ao}) \quad (1)$$

The enthalpy flow difference of the moist air between inlet and outlet ($\dot{H}_{Ai} - \dot{H}_{Ao}$) is equal to the sum of sensible heat flow $\dot{Q}_s$ and the enthalpy flow of the precipitated water-vapour $\dot{H}_{spV}$ (spV: separated vapour):

$$\dot{Q}_s + \dot{H}_{spV} = \dot{H}_{Ai} - \dot{H}_{Ao} = \dot{m}_{Ad} \cdot (h_{Ai} - h_{Ao}) \quad (2)$$

$$\dot{Q}_s + \dot{H}_{spV} = \dot{m}_{Ad} \cdot [c_{pAd} \cdot (\vartheta_{Ai} - \vartheta_{Ao}) + r_v \cdot (x_{Ai} - x_{Ao}) + c_{pV} \cdot (x_{Ai} \cdot \vartheta_{Ai} - x_{Ao} \cdot \vartheta_{Ao})] \quad (3)$$

Here, c_{pAd} stands for the specific heat capacity of dry air, c_{pV} for the specific heat capacity of water vapour and r_v for the specific enthalpy of evaporation of water. To a good degree of approximation, (3) may be expressed as follows (cf. Berlinger *et al.* 2008):

$$\dot{Q}_s + \dot{H}_{spV} = \dot{m}_{Ad} \cdot [c_{pAd} \cdot (\vartheta_{Ai} - \vartheta_{Ao}) + r_v \cdot (x_{Ai} - x_{Ao})] \quad (4)$$

Balance boundary around the water film:

Assuming that the sum of condensate and melted ice and frost mass ($\dot{m}_C, \dot{m}_M$) continuously flows off as $\dot{m}_W$, the following applies for the mass balance:

$$\frac{dm}{dt} = \dot{m}_C + \dot{m}_M - \dot{m}_W = 0 \quad (5)$$

$$\dot{m}_W = \dot{m}_C + \dot{m}_M \quad (6)$$

The energy balance for this control area is:

$$\dot{Q}_s + \dot{H}_{spV} + \dot{H}_M = \dot{Q}_{sl} + \dot{H}_M + \dot{H}_C \quad (7)$$

From the energy balance equation it follows that the total heat flow for melting $\dot{Q}_{sl}$ is identical to the sum of sensible heat flow $\dot{Q}_s$ and latent heat flow $\dot{Q}_l$. In this case, the latent heat flow corresponds to the difference between the enthalpy flow of the precipitated water-vapour $\dot{H}_{spV}$ and the enthalpy flow of the condensate $\dot{H}_C$ running off (with the specific heat capacity c_{pC} and the temperature ϑ_C of the condensate).

$$\dot{Q}_{sl} = \dot{Q}_s + \dot{Q}_l = \dot{Q}_s + \dot{H}_{spV} - \dot{H}_C \quad (8)$$

$$\begin{aligned} \dot{Q}_{sl} &= \dot{Q}_s + \dot{Q}_l = \dot{Q}_s + \dot{H}_{spV} - \dot{H}_C = \dot{m}_{Ad} \cdot (h_{Ai} - h_{Ao}) - \dot{m}_C \cdot h_C \\ &= \dot{m}_{Ad} \cdot [c_{pAd} \cdot (\vartheta_{Ai} - \vartheta_{Ao}) + r_v \cdot (x_{Ai} - x_{Ao}) + c_{pV} \cdot (x_{Ai} \cdot \vartheta_{Ai} - x_{Ao} \cdot \vartheta_{Ao}) - c_{pC} \cdot \vartheta_C (x_{Ai} - x_{Ao})] \end{aligned} \quad (9)$$

The following is approximately valid for $\dot{Q}_{sl}$:

$$\dot{Q}_{sl} = \dot{m}_{Ad} \cdot [c_{pAd} \cdot (\vartheta_{Ai} - \vartheta_{Ao}) + r_v \cdot (x_{Ai} - x_{Ao})] \quad (10)$$

The heat flow with which the frost layer is *melted* is designated as *defrosting capacity* $\dot{Q}_D$. It corresponds to the sum of sensible $\dot{Q}_s$ and latent heat flow $\dot{Q}_l$. If this sum is negative, the frost layer can not be reduced by *melting*, but only by *sublimation*.

$$\dot{Q}_D = \dot{Q}_{sl} = \dot{Q}_s + \dot{Q}_l \quad (11)$$

Balance boundary around the frost layer

It follows from the mass balance that the change in the frost mass corresponds to the mass flow of the melting ice $\dot{m}_M$.

$$\frac{dm_F}{dt} = -\dot{m}_M \quad (12)$$

The energy balance equation is:

$$\frac{dE}{dt} = \dot{Q}_{sl} - \dot{H}_M \quad (13)$$

Whereby:

$$\frac{dE}{dt} = \frac{d}{dt} [m_F(t) \cdot u_F(t)] = \frac{dm_F}{dt} \cdot u_F + \underbrace{\frac{du_F}{dt} \cdot m_F}_{=0 \text{ since } \vartheta_F \approx \text{konst.}} = -\dot{m}_M \cdot [-r_{sd} + c_{pF} \cdot \vartheta_F] \quad (14)$$

Here, u_F designates the specific internal energy of the ice or frost. Further, r_{sd} stands for the specific enthalpy of solidification of water and c_F for the specific heat capacity of ice or frost. Thus, the following applies for the energy balance equation:

$$-\dot{m}_M [-r_{sd} + c_F \cdot \vartheta_F] = \dot{Q}_{sl} - \dot{H}_M \quad (15)$$

$$\dot{m}_M \cdot r_{sd} = \dot{Q}_{sl} - \dot{H}_M + c_F \cdot \vartheta_F \quad (16)$$

Approximately:

$$\dot{m}_M \cdot r_{sd} = \dot{Q}_{sl} \quad (17)$$

The melted ice mass flow can be calculated using:

$$\dot{m}_M = \frac{\dot{Q}_{sl}}{r_{sd}} \quad (18)$$

For the defrosting time for *melting* of the entire frost mass m_F the following is valid:

$$t_D = \frac{r_{sd} \cdot m_F}{\dot{Q}_{sl}} \quad (19)$$

3.2 Simultaneous heat and mass transfer and its representation in the Mollier diagram

In order to devise the simultaneous heat and mass transfer processes on a frost layer covered with a water layer, a local fin element as shown in figure 1 is considered. Due to the temperature gradient ($\vartheta_A - \vartheta_B$) between the moist air and the boundary layer (water surface), the transfer of a sensitive heat flow $\dot{Q}_s$ occurs. The magnitude of the heat flow can be expressed independent of evaporator surface area by the sensible *heat flux* $\dot{q}_s$. The sensible heat flow causes the air to be cooled down. A latent heat flow $\dot{Q}_l$ is transferred simultaneously along with the sensible heat flow $\dot{Q}_s$ (partial condensation, evaporation of water or sublimation of ice). As for the temperature gradient in heat transfer, the humidity gradient ($x_A - x_B$) is the driving force for mass transfer. In most technical applications, heat and mass transfers can be examined together, whereby the Lewis number Le describes the relationship between heat and mass transfer. For the application under consideration we can calculate with $Le = 1$ (Sahinagic *et al.* 2004). The usable heat flux for the defrosting is:

$$\dot{q}_{sl} = \alpha_A \cdot (\vartheta_A - \vartheta_B) + \dot{m}_C \cdot r_v = \underbrace{\alpha_A \cdot (\vartheta_A - \vartheta_B)}_{\dot{q}_s} + \underbrace{\frac{Le \cdot \alpha_A}{c_{pAd}} \cdot (x_A - x_B) \cdot r_v}_{\dot{q}_l} \quad (20)$$

The humidity gradient ($x_A - x_B$) and, therefore, the latent heat flux $\dot{q}_l$ can, depending on humidity, be positive or negative. If this gradient is positive, water vapour in moist air condenses at the boundary layer (water film). The condensation heat transferred to the water film is, in addition, added to the sensible heat flow for the *melting* of the ice or frost layer and thus speeds up the defrosting process. If for low humidity the humidity gradient is negative, the melted ice evaporates at the boundary layer. The evaporation heat required for this is taken from the melted ice and frost, which reduces the total heat flow $\dot{q}_{sl}$ for melting the frost layer.

The sensible and latent heat flow can be graphically represented and interpreted in the Mollier diagram. Through the transformation of (20), one obtains the following:

$$\underbrace{\dot{q}_{sl} \cdot \frac{c_{pAd}}{\alpha_A}}_{\dot{q}_{sl}} = \underbrace{c_{pL} \cdot (\vartheta_A - \vartheta_B)}_{\dot{q}_s} + \underbrace{Le \cdot (x_A - x_B)}_{\dot{q}_l} \quad (21)$$

$$\dot{q}_{sl} = \dot{q}_s + \dot{q}_l \quad (22)$$

By transformation, the heat fluxes $\dot{q}$ in W/m^2 are converted into specific heats q with the unit J/kg . The specific heats refer to the mass of the dry air. Assuming that $Le = 1$ these specific heats can be represented in the Mollier diagram in a simple way (cf. Bošniaković *et al.* 1997). The sensible, latent and the resulting total specific heat for five different air states are shown in figure 2. All five air states exhibit the same temperature but different humidities. Therefore, the sensible specific heat remains unchanged for all air states while the latent specific heat assumes different values depending on the air state.

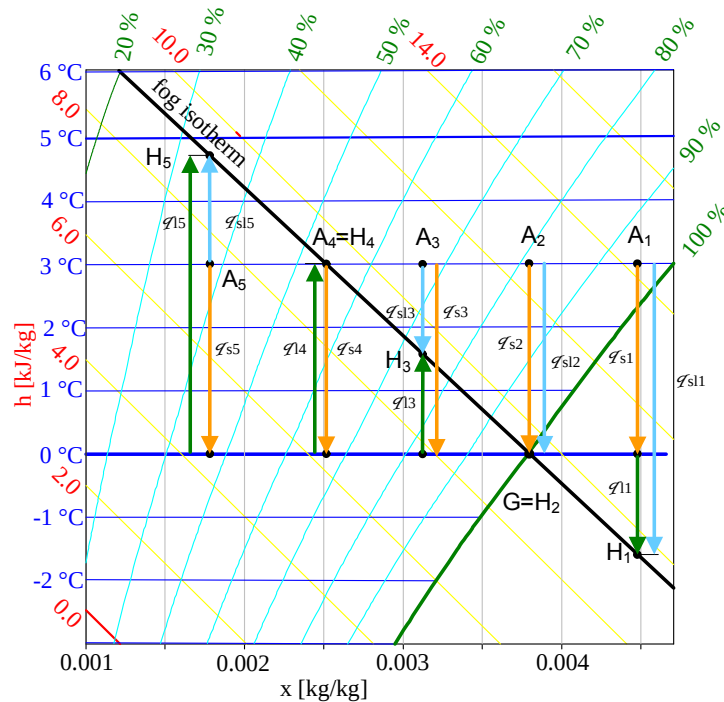


Figure 2: Representation of the sensible (q_s) and latent (q_l) specific heats for various air states for air flowing over a frost layer covered with a film of water at 0°C . The boundary layer G designates the surface of the film of water with a temperature of 0°C . The air states are noted as A_1 to A_5 . The points H_1 to H_5 are auxiliary points for the graphical ascertainment of the specific heats.

Air state A_1 (3°C , 95% rH, $x_A > x_B$)

The air at the state point A_1 is almost saturated with water vapour. One can see from the Mollier diagram that the difference for the absolute humidity ($x_{A1} - x_B$) is positive. This means that water vapour in the moist air condenses on the water film and that a latent heat flow is transferred to the layer of melted ice and frost. The melting of the frost layer is supported by the sensible and latent specific heat and is optimal in this area ($|q_{s1}| + |q_{l1}| = |q_{sl1}|$) (cf. figure 3). The total heat flow for defrosting by *melting* is highest for air state A_1 .

Air state A_2 (3°C , 81% rH, $x_A = x_B$)

For air state A_2 , the absolute humidities in the air and for the melted ice and condensate film are of the same size ($x_{A2} = x_B$). No latent heat flow occurs. The total specific heat for defrosting by *melting* is identical to the sensible specific heat ($|q_{s2}| = |q_{sl2}|$).

Air state A_3 (3°C , 67% rH, $x_A < x_B$)

The difference between the absolute humidities of the air with state A_3 and the boundary layer is negative, which results in the evaporation of water in the water film. The totally usable specific heat for defrosting by *melting* is lower than the sensible specific heat ($|q_{s3}| - |q_{l3}| = |q_{sl3}|$) by the amount of latent specific heat. The totally usable specific heat is, in this case, less than half of that for air state A_1 .

Air state A_4 (3°C , 47% rH, $x_A < x_B$)

The air state A_4 is located on the extension of the fog isotherm. The latent specific heat given off by the water film to the air is of the same magnitude as the sensible specific heat transferred from the air to the water film ($|q_{s4}| = |q_{l4}|$). Consequently, the resultant specific

heat for defrosting is ($|q_{sl4}| = 0$). Therefore, no heat flow to or from the water and frost layer occurs. This air state forms the theoretical marginal case: Defrosting using fan operation to induce *melting* of the frost layer is no longer possible. Nevertheless, the frost layer is reduced for this air state. The frost layer is slowly removed by sublimation of water from the frost due to mass transfer. The sublimation heat necessary for this is around eight times higher as the heat for melting, which makes the process correspondingly slow and not relevant for the defrosting of the evaporator in air/water heat pumps.

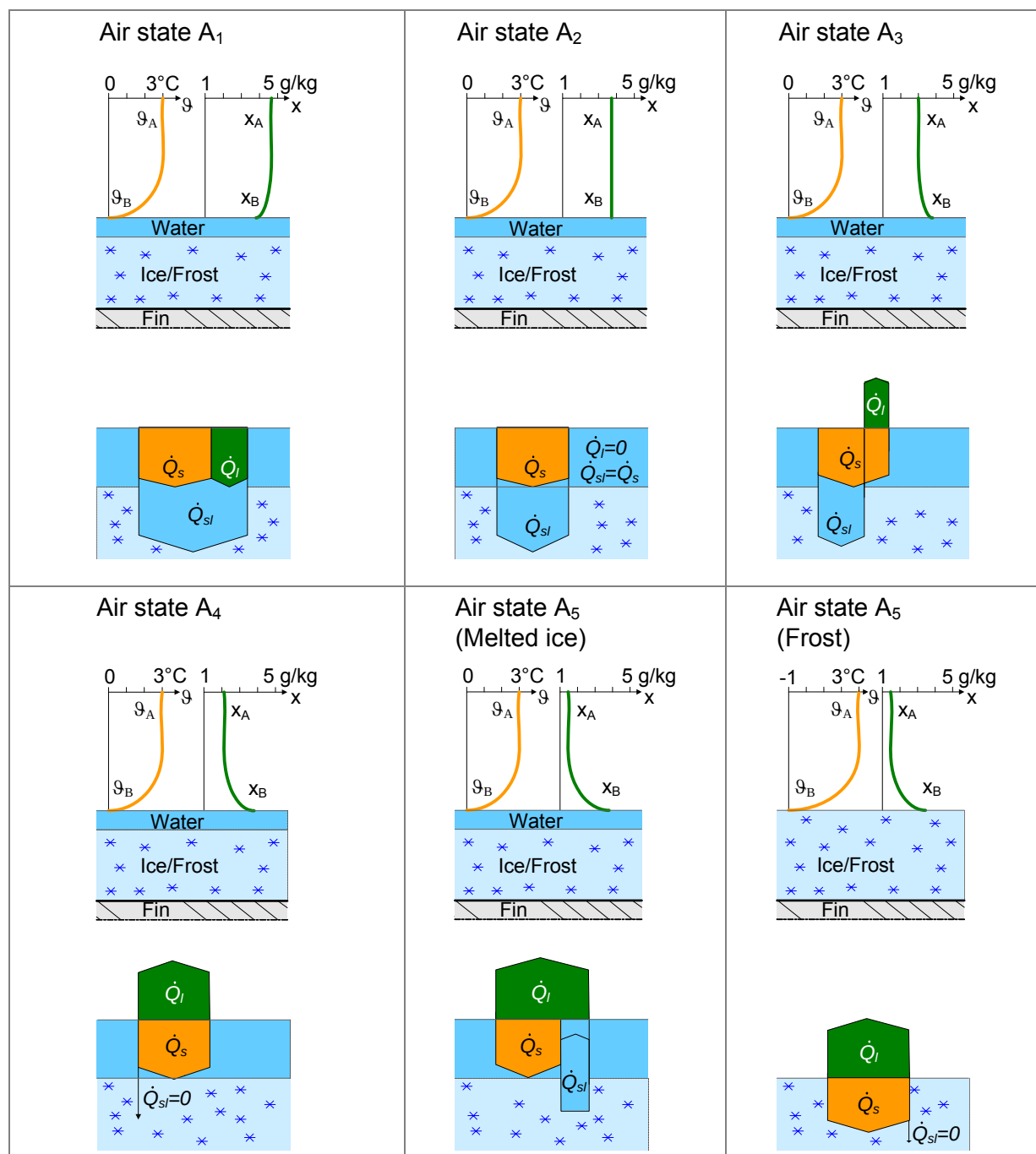


Figure 3: Temperature curves and curves for the absolute humidity for the air flow up to the boundary layer for the air states A₁ to A₅ showing the heat flows occurring.

Air state A_5 (3°C , $32\% \text{ rH}$, $x_A < x_B$)

a) Without water film at the start of defrosting:

For this air state, no defrosting is possible through *melting* by using ambient air as the evaporator can not be warmed up to 0°C ; no melted ice film results. The latent heat flow does not result from evaporation but from the sublimation of water from the frost to the air. The frost automatically falls to a temperature below 0°C so that the latent and sensible specific heats are of the same magnitude (figure 4). As for the air state A_4 , here, too, the frost layer is reduced by sublimation; this is, on account of the slow ice removal rate, not interesting for the defrosting of the evaporator in air/water heat pumps.

b) With water film at the start of defrosting:

The air state A_5 is to the left of the extended fog isotherm (Mollier diagram). The latent specific heat transferred from the water film to the air is greater than the sensible specific heat transferred from the air to the water film. Consequently, the total amount of heat for defrosting by *melting* is negative. As a result, the water film freezes to ice. If the water film has frozen completely, the processes continue as in the case of the state without water film at the beginning of the defrosting.

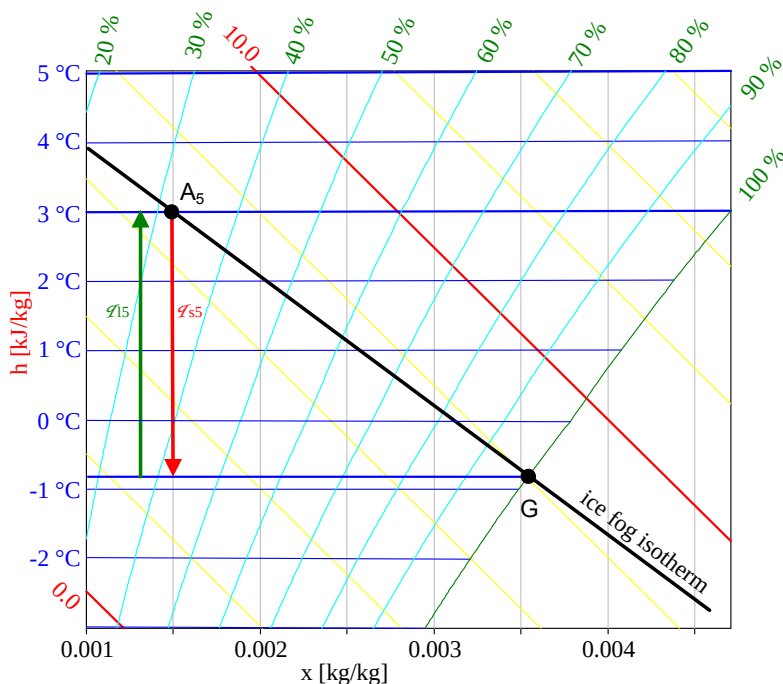


Figure 4: Sensible and latent specific heat when air with state A_5 flows across a frost layer. The air state A_5 and the state of the boundary layer are on the same ice fog isotherm. (For sublimation, the ice fog isotherm must be employed: the difference in gradient compared with the fog isotherm is, however, marginal and is of low importance for this representation)

4 RESULTS

4.1 Theoretical and experimental parameter study

Both theoretical and experimental parameter studies on defrosting using ambient air were carried out in the LOREF research project (Berlinger *et al.* 2008). Measurements were made using an experimental air heat exchanger with a similar geometry as can be found in the evaporators of A/W-HP. The average defrosting capacity ($\dot{Q}_D = \dot{Q}_S + \dot{Q}_I$) is shown in figure 5 as a function of the air inlet temperature. The air velocity between the fins is $w = 2 \text{ m/s}$ and the dew point temperature is $\vartheta_{DP} = 0^\circ\text{C}$, independent of the air inlet temperature. Since the dew point temperature corresponds here exactly to the boundary layer temperature (frost, condensate and melting-ice water film), no latent heat flow occurs (cf. air state A_2 in figure 3).

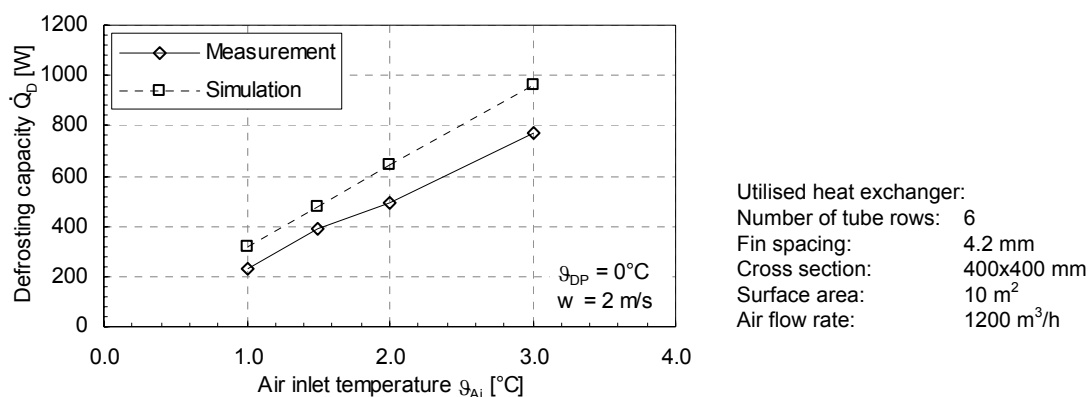


Figure 5: Measured and simulated defrosting capacity as a function of air inlet temperature for an air velocity of 2 m/s between the fins and a dew point temperature of 0°C.

Figure 5 shows that both the measured and the simulated defrosting capacity increase linearly with increasing air inlet temperature. With increasing air inlet temperature, the temperature gradient between the air and the boundary layer increases and, therefore, the sensible heat flow does so too.

Figure 6 shows the defrosting capacity as a function of the dew point temperature of the air on inlet into the fin tube heat exchanger. The air inlet temperature is in each case $\vartheta_{Ai} = 2^\circ\text{C}$ and the air velocity between the fins $w = 2 \text{ m/s}$.

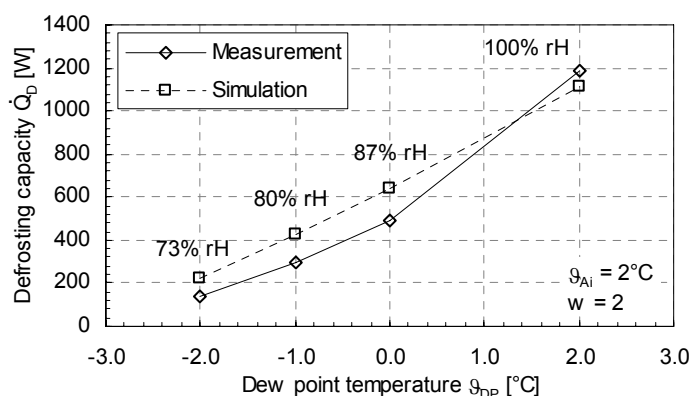


Figure 6: Measured and simulated defrosting capacity as a function of dew point temperature for an air velocity of 2 m/s and an air temperature of 2°C. For each measurement, the relative air humidity is indicated in % [rH].

It is easy to see, for efficient defrosting using ambient air, that the air temperature alone as a single criterion does not satisfy our needs. At the measurement point with $\vartheta_{DP} = -2^\circ\text{C}$, the defrosting capacity is below 200 W and the defrosting time is approximately 40 mins (the resultant latent heat flow is negative here on account of the evaporation of melted ice that occurs). At the measurement point with saturated air (100% rH), a defrosting capacity of clearly over 1000 W is achieved and the defrosting takes only approximately 5 mins, although, with 2°C , the air temperature is only slightly above the melting point of ice. The reason for this fast defrosting is the high latent heat flow.

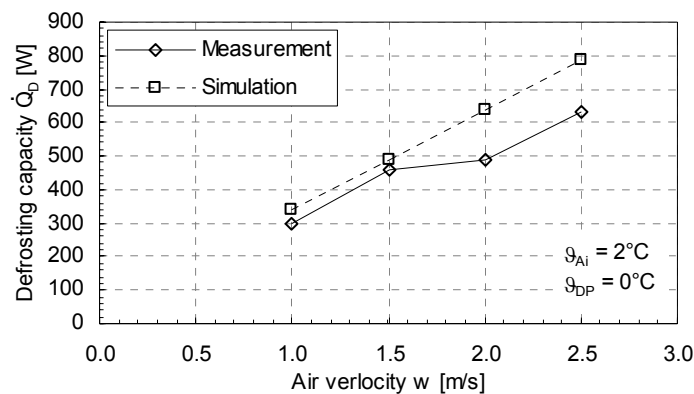


Figure 7: Measured and simulated defrosting capacity as a function of air velocity for a dew point temperature of 0°C and an air temperature of 2°C

The dependence of defrosting capacity on air velocity is illustrated in figure 7. The simulation shows a linear rise of defrosting capacity with increasing air velocity. The cause of this is that, in simulations, the air-side heat transfer coefficient α_A is calculated with a linear correlation as a function of air velocity. The measurements up to 1.5 m/s are almost on the simulated curve whereas from 2 m/s air-speed onwards defrosting capacities were measured that are lower than those determined by simulation. Simulated defrosting capacity is tendentially a little higher than the values measured.

4.2 Comparison with hot-gas defrosting

As a comparison, the defrosting capacity $\dot{Q}_D$ of the used commercial A/W-HP with the known compressor power of P_K is estimated (Berlinger *et al.* 2008):

$$\dot{Q}_D = P_K \cdot f_A \cdot f_w = 2300 \cdot 0.43 \cdot 0.5 = 500 \text{ W} \quad (23)$$

In the case of hot-gas defrosting (without additional electrical resistance heating) the entire defrosting capacity is provided by the compressor. As the surface area of the evaporator is larger than the surface area of the experimental heat exchanger examined (ch. 4.1), the compressor power in equation (23) is corrected using a surface factor f_A . In the case of hot-gas defrosting, the pressure ratio across the compressor is much lower than in heating operation. In addition, losses to the environment must be taken into account. This is why the decrease in the defrosting capacity compared to the theoretical compressor power is taken into consideration using the factor f_w . Recalculated for the experimental air cooler, a defrosting capacity for hot-gas defrosting of approximately 500 W results. Here, it is assumed that this capacity is independent of ambient temperature.

In figure 8 the estimated defrosting power for hot-gas defrosting is compared with defrosting using ambient air for various air inlet temperatures, each with a dew point temperature of 0°C . Already for an air inlet temperature of 2°C , the defrosting capacity is of the same magnitude as in the case of hot-gas defrosting. Assuming that the fan, when well

dimensioned and with a high efficiency, only consumes around 10% of the electrical power of the compressor, the energy consumption for defrosting can be reduced by a factor of approximately 10.

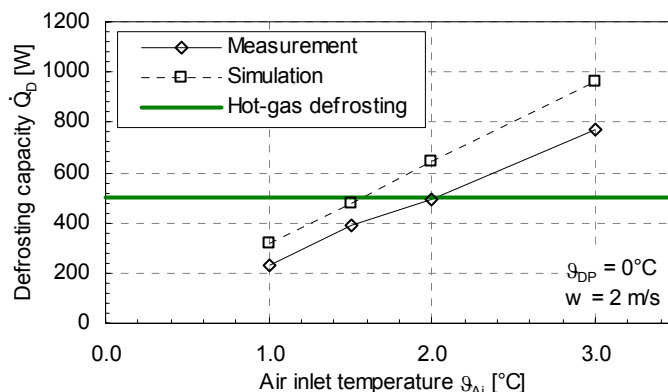


Figure 8: Measured and simulated defrosting capacity as a function of air inlet temperature. For an air inlet temperature of 2°C, the measured defrosting capacity by using ambient air is already of the same magnitude as in the case of hot-gas defrosting.

5 CONCLUSIONS AND OUTLOOK

The defrosting capacity when defrosting with ambient air is strongly dependent on the temperature and the humidity of the air. Above all, at air temperatures up to around 3°C, defrosting by *melting* ice and frost under low humidity conditions can be slowed down considerably or even rendered impossible. If one wants to defrost at temperatures just a little above 0°C using ambient air, it is imperative that the air's humidity must be considered. The fact that the required defrosting times get longer at temperatures slightly above 0°C is not critical for heat pumps working at these ambient temperatures on account of the high periods of inactivity that are encountered in on/off operation. A further positive factor is that defrosting by using ambient air is especially effective for air states with much frost formation (neuralgic temperature range 2–8°C).

6 REFERENCES

- Berlinger L., M. Imholz, M. Albert, B. Wellig, K. Hilfiker 2008. LOREF: Optimierung des Lamellenluftkühlers/Verdampfers von Luft/Wasser-Wärmepumpen, Teil 1: Theoretische und experimentelle Untersuchungen, Swiss Federal Office of Energy, Berne.
- Bertsch S., P. Hubacher, M. Ehrbar 2002. Verbesserung des Abtauens bei luftbeaufschlagten Verdampfern, Phase 2: Bewertung der Abtauprozesse, Swiss Federal Office of Energy, Berne
- Bošnjaković F., K.F. Knoche 1997. Technische Thermodynamik, 2. Teil, 6. Auflage, Dr. Dietrich Steinkopff Verlag, Darmstadt.
- Hubacher B., M. Ehrbar 2000. Verbesserung des Abtauens bei luftbeaufschlagten Verdampfern, Analyse gängiger Abtauverfahren, Swiss Federal Office of Energy, Berne.
- Hubacher P., C. Bernal, M. Ehrbar, S. Schwendener 2004: Verbesserung des Abtauens bei luftbeaufschlagten Verdampfern, Phase 3: Technische Umsetzung, Labor- und Feldversuche, Swiss Federal Office of Energy, Berne

Sahinagic R., M. Imholz, L. Berlinger, H. Huber, K. Hilfiker 2004. LOREF: Untersuchung der Frostbildung für Lamellenluftkühler von Wärmepumpen, Swiss Federal Office of Energy, Berne.

Unternaehrer A. 2006, Luftabtauung des Verdampfers einer Luft/Wasser-Wärmepumpe mit Ventilatornachlauf, Diplomarbeit, Lucerne University of Applied Sciences and Arts – Engineering & Architecture, Horw.