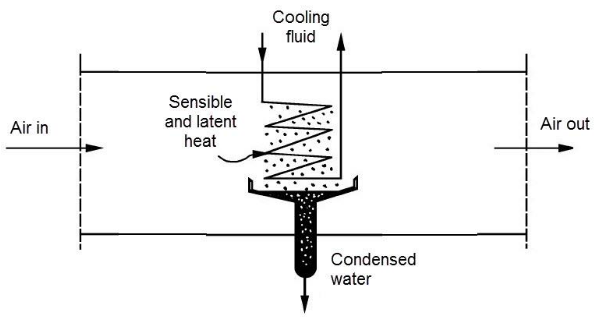

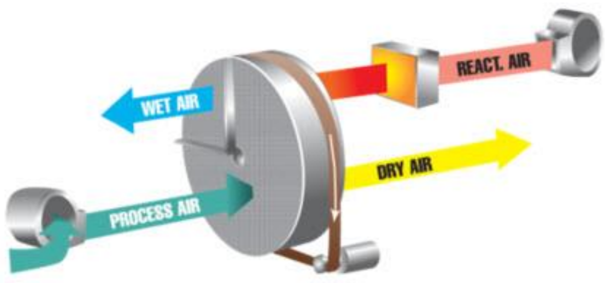

NERIS – Part 5

Comparison of refrigeration and sorption-based dehumidification in ice rinks

(Jämförelse av kyl- och sorptionsavfuktning i ishallar)

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Summary

The project name NERIS is an acronym for Nordicbuilt: Evaluation and Renovation of Ice halls and Swimming halls. NERIS is led by the department of Civil Engineering at the Royal Institute of Technology (KTH) in Stockholm, Sweden.

This report is part five in a series of five addressing humidity issue in ice rinks. This knowledge is very important when designing and operating an ice rink, where proper air moisture management is crucial. In this part refrigeration and sorption technologies are compared as to different aspects of performance and ultimately the annual energy use.

The analysis suggests that refrigeration based technology with frost-free conditions can provide with only 4 kg/h dehumidification capacity at design conditions, while higher dehumidification capacity can be obtained in sub-zero conditions, but defrosting must be considered. The modelling results show that for a nominal design dehumidification capacity (20 kg/h) the required installed cooling capacity is around 85 kW. There are field examples of design with underestimated cooling capacity requirements for the desired dehumidification capacity, which is due to neglection of latent heat as well as defrosting implication. For a large arena, although potentially frost free operating mode may be allowed due to acceptance of higher moisture content in air, in case of 60 kg/h dehumidification capacity, as much as 360 kW of cooling capacity is needed.

When it comes to the sorption technology, many field examples are analysed. The specified equipment size suggests that it is needed around 30-50 kW of heating capacity. As to the airflows, these differ depending on reactivation air temperature. Generation two configuration needs around 50°C air temperature, 2 and 1 m3/s airflow rates for process and reactivation air respectively. This can be put into comparison to refrigeration based technology which for the same capacity needs around 3.5 m3/s (coil temperature -8°C), which is however strongly dependent at what temperature level the cooling is provided.

When integrating dehumidification system in an ice rink, it is important to understand how well it fits with other systems. Refrigeration based dehumidification drives the required installation capacity of the main refrigeration plant of the ice rink, thus the costs. Sorption on the other hand does not have a cooling demand as such, and it can even contribute to heat recovery utilisation, as the demand for heat is highest when there is plenty available from the refrigeration system.

As regards annual energy requirements, both technologies are compared, with different heat source options considered for sorption technology. What matters most is how much energy is eventually purchased. And the best scenario is found to be sorption “generation two technology” with full heat recovery, requiring around 14 MWh of electricity on annual basis. Refrigeration based dehumidification energy use over a same period of time, is calculated to be around 35 MWh.

Table of Contents

Table of Figures

1 Introduction

1.1 Background and scope of the NERIS-project

The overall project name NERIS is an abbreviation based on Nordicbuilt: Evaluation and Renovation of Ice halls and Swimming halls, which is managed by the Department of Civil and Architectural engineering at the Royal Institute of Technology (KTH) in Stockholm, Sweden.

The financing stems from the organizations: Formas (The Swedish Research Council for Environment, Agricultural Sciences and Spatial Planning) and Energimyndigheten (the Swedish Energy Agency).

The overall target for the project is the following: "This project aims at the proposal of methods for inspection and evaluation of the functionality of buildings of this kind and by demonstration of different methods for renovation for improving their performance". This implies building a knowledge bank related to moisture handling in ice rinks and swimming halls. The NERIS-project was initiated in 2014 and will be finished in 2018.

This report is part 5 in a series of 5 reports addressing the humidity issue in ice rinks. In these 5 reports the mechanisms of humidity in ice rinks will be explained and analysed. It all starts with the specifics of ice rinks as applications and how the humidity issue comes into the picture. Further, the idea is to build a logical order of reports that describe the moisture challenges in ice rink applications ranging from the moisture sources through the building physics challenges to dehumidification methods and the associated energy usage. In conclusion, the different parts should be linked together containing practical advice and instructions for design and sizing of ice rink dehumidification systems.

1.2 Scope of NERIS – Part 5: Comparison of refrigeration and sorption-based dehumidification in ice rinks

This report was added to the original planned 4 reports since the question "what about refrigeration based versus sorption-based dehumidification" often came up in the discussion and presentations though the work. There are many "opinions" out there as which process is the best. What is the best or the most suitable depends on a number of constraints such as what is the desired humidity level in an ice rink? Which source of energy do you have available for your dehumidification process? We could probably extend the list quite far, however, in this report the findings from Neris part 1 to 4 have formed the basis for the assumptions and these are of course coupled with our never-ending desire to save energy in general!

The report covers the theory of condensation and frost formation which are the basis for the refrigeration based dehumidification. Further, the practical implication of designing such systems and especially the cooling demand and the defrosting is described and modelled. Since this method is relatively unusual in Sweden there was little field data available. In practical design data have been used for the modelling and later comparison with both theoretical and practical sorption dehumidification data.

The aim of the study was to illustrate the design and investment implications of both studied methods to allow the reader to judge what is the most suitable method in his/her application.

2 Refrigeration based dehumidification

2.1 Theory

A literature survey on the topic is made and the insights are used to the best of knowledge from what is studied in other investigations. There are not many installation examples where refrigeration-based dehumidification technology is applied in such relatively low temperature and humidity levels. However, there are a lot of investigations done either for heat pump or air cooler frosting issues. For the most part, these studies have been found to be applicable as close condition cases for refrigeration type dehumidification that would be applied in an ice rink.

2.1.1 Working principles

When any surface that is surrounded by air has a temperature lower than the dew point of air, the water vapor content of the air will condense on the surface, making the air drier. This is the principle upon which the refrigeration type dehumidifier is based. A coil with a circulating fluid in it has a cold surface that "attracts" the water vapor in the air. This means that due to the necessity of having a cold surface, a cooling energy source as well as a medium of some kind are needed. Depending upon the configuration of these components, refrigeration type dehumidification can be classified into direct or indirect. Not many ice rinks use this technology, as it is found to be have some challenges with applicability for the specific needs of ice rinks, which will be discussed later.

Figure 1. Moisture condensation from air to a surface with lower temperature than the dewpoint of air.

2.1.2 Direct and indirect configurations

There are two main types of refrigeration-based dehumidification. The distinction can be made between a direct and an indirect configuration.

2.1.2.1 Direct

Figure 2 shows the working principle of the refrigeration type dehumidifier in the direct expansion configuration. First, humid ambient air passes the cold surface of the evaporator, where water vapor is removed through the condensation process, then air should be heated in order to supply it warm to the rink space. To eliminate the need for an external heat source for reheating, condenser heat is normally utilized.

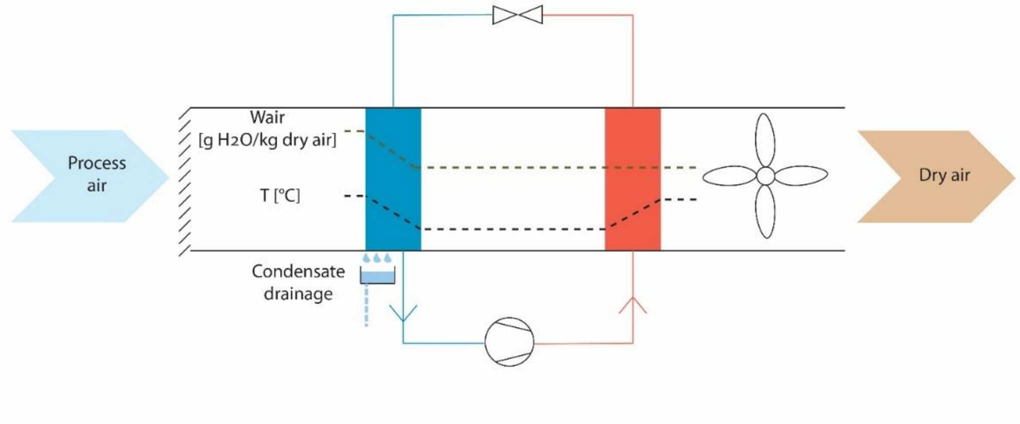

Figure 2. Schematic of the direct refrigeration type dehumidifier.

2.1.2.2 Indirect

The indirect refrigeration type dehumidification has the same working principle as the direct type in terms of the physical processes involved on the air side. The difference is in the cooling and heating energy supply configuration. In Figure 3 it can be seen that the secondary fluid of the ice rink's refrigeration system provides the dehumidification unit with the necessary cooling capacity and the rejected heat in the coolant loop is used for reheating. The major concern about the applicability of this configuration in ice rinks is whether it is a cost-effective solution, since the required additional cooling capacity to the ice rink's refrigeration system might increase its investment costs significantly. This is because the peak load for dehumidification and ice cooling occur at the same part of the season, i.e. when ambient air has the highest temperature and is the most humid, meaning that the requirements of both systems rather stack on each other than complement each other.

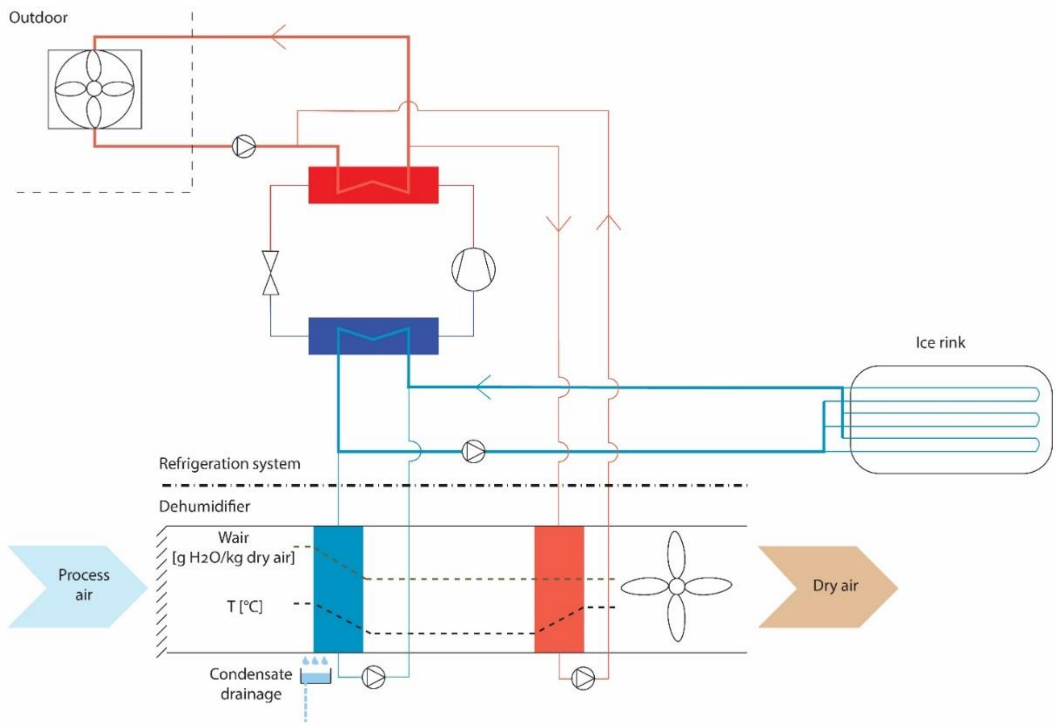

Figure 3. Schematic of the indirect refrigeration type dehumidifier.

Figure 4. A ventilation unit with an indirect refrigeration dehumidification function.

In the air handling unit above the air direction is from the right to the left where the cooling coil comes first followed by an internal heat recovery circuit. The last coil connected to the what insulated pipes is the heating coil. This system is later illustrated in the chapter "Examples from the field".

2.1.3 Air side energy balance

Air is a mix of gases that carries energy and according to The First Law of Thermodynamics energy is a finite unit. In a refrigeration type dehumidifier, the released energy amount from the air is equal to the total energy amount that is absorbed by the cooling fluid and the energy that is in the condensate water. The latter however is relatively small and can even be neglected. The rest of the energy remains with the outlet air. Figure 5 gives an indication of process flows in a refrigeration type dehumidifier and the energy balance may be deduced better by using this illustration.

It must be kept in mind that when air is dehumidified using this technology, the required cooling capacity consists of sensible and latent heat. It is not uncommon that only sensible heat is taken into consideration during the design, which accounts only for capacity needed to cool the air and not to remove the water from it as well.

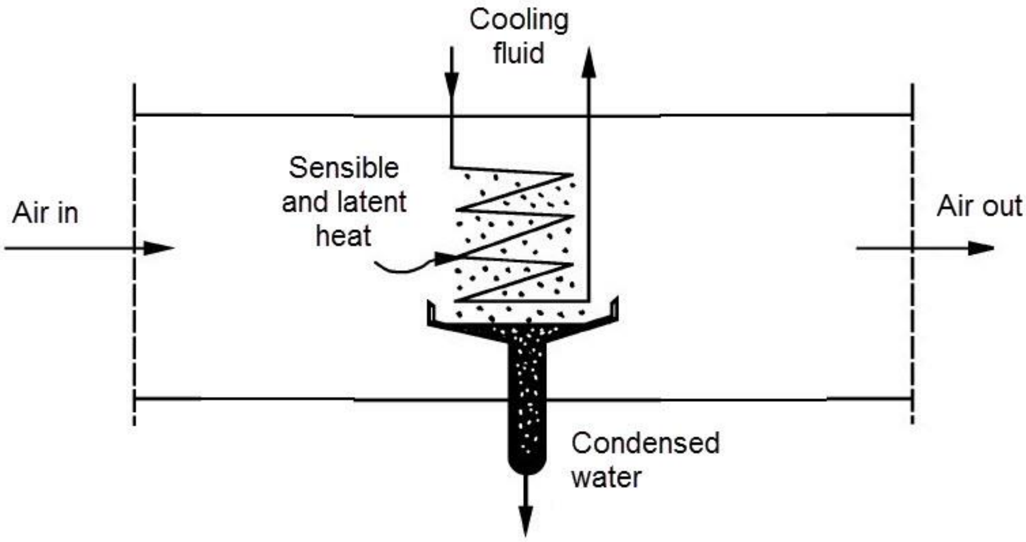

Figure 5. Schematic of energy carrier flows in a refrigeration based dehumidifier. (ASHRAE, 2017)

This solution requires an additional consideration as to the actual air mass flow needed, because in reality when air passes the cooling coil the heat exchange process is not perfect, and therefore the outlet air temperature will be higher than the surface temperature of the coil. To predict the effectiveness of the heat exchange process between the cooling coil and the air that is being processed from one state to another, the concept of bypass factor can be used. The idea is to simplify complex processes that happen in every specific heat exchanger by using a predictable factor, that can be expected in different applications. Bypass factor tells what proportion of air has not been changed along a heat exchanger. Zero bypass factor means that all of the air has been in contact with the walls of the coil, meaning a perfect heat exchange, which is not possible in reality.

The amount of air that bypasses depends upon different aspects like velocity of air, construction of the coil and surface temperature of the coil. By knowing the bypass factor value and inlet conditions, it is possible to estimate the outlet conditions. Bypass factor is somewhat exclusive for every unit and operating conditions, but a typical expected range for different applications can be seen in Figure 6 below, suggested by manufacturer. In system like is analysed in this work bypass factor values can range between 0.002 to 0.012 and the chosen value is 0.1, which is on the safe side of the range, not overestimating the performance. (Carrier United Technologies, 2017)

Equipment Type	Available Cooling Coil Rows	Bypass Factor (BF) Range
Residential Cased Coil	1-2	0.20-0.30
Small Packaged Unit (<10 tons)	2-4	0.05-0.30
Packaged RTU (> 10 tons)	3-4	0.03-0.20
Central Station AHU	3-10	0.002-0.12

Typical Bypass Factors (BF) for Various Equipment Types and Coil Rows

Figure 6. Expected coil bypass factor ranges for different applications. (Carrier United Technologies, 2017)

2.1.4 Frost accumulation

Achieving dew point temperature of the air to be close to 0°C, as is the case in an ice rink, requires in practice a coil with a sub-zero surface temperature. Inevitably conditions for frost formation are favorable. As the frost layer accumulates on the coil surface during the operation, heat transfer will continuously deteriorate, reducing the cooling output and thus limiting capability of the dehumidifier to remove water.

2.1.4.1 Physics behind frost formation

The visually very well recognized process of freezing is not as simple to be fully perceivable and predictable as far as the physics are concerned.

Similar as with snow, the structure of frost can vary. The properties of frost are strongly influenced by the conditions during which the frost is formed. As an example, the density of frost is lower at lower temperatures, while increasing at higher temperatures. Moreover, the properties are also dependent on temperature and humidity in the air. In terms of depth, the inner layer of frost closer to the cold surface is more dense than the outer boundary layer facing the air. This process has been extensively investigated both in theoretical studies as well as laboratory tests, however this work is not dedicated to analyse frosting process itself in depth as it is not within the scope.

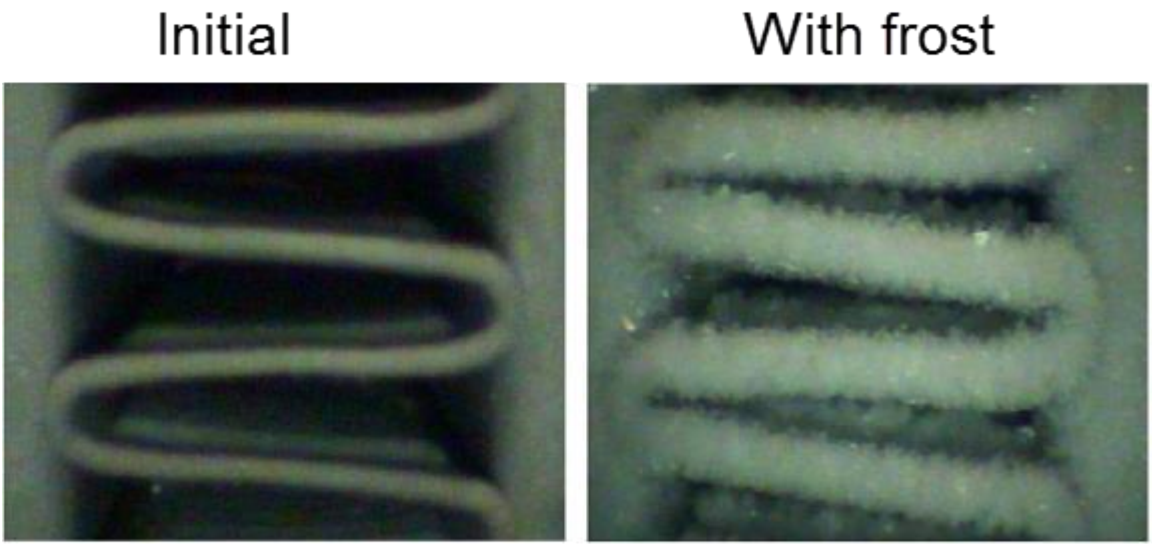

Figure 7. Frost formation on the surface after some time of operation.

2.1.4.2 Effects on the performance of a dehumidifier

To reach a low enough dewpoint in the rink space with reasonable cooling capacity and volume flowrate, cooling coil wall temperatures below 0°C are normally required, which is going to generate frost. As it accumulates on the heat exchanger surface, it will deteriorate the heat transfer rate due to:

- Reduction of the air flow due to increased flow resistance
- The insulating effect of the frost layer

The first effect has generally a more significant impact and therefore results in the need for defrosting, particularly in heat exchangers with a small fin spacing. Depending on the design of the cooling coil, sensitivity to the frost formation is different. In a heat exchanger with fins packed closely together, only a relatively small amount of frost needs to be accumulated before performance will already become poor. A simple example can be given: in a heat exchanger with 4 mm fin spacing distances, a 1mm frost layer on the two opposite fin surfaces will eventually cut the free flow area to half. If the same air flow volume needs to be carried through, it would correspond to a doubled flow velocity, meaning that the pressure drop would rise four times roughly.

It is therefore important to understand to what extent both the cooling capacity and airflow are reduced over the time of operation. As the system continues to operate, the negative effects will develop further, reaching a level where means of frost removal are ultimately needed, i.e. defrosting. It can be done by heating the coil from the fluid side, but configurations with air side heating also exist. A better understanding as to how the cooling capacity deteriorates over time may be obtained by examining Figure 8, where also defrosting initiation and energy use are graphically shown.

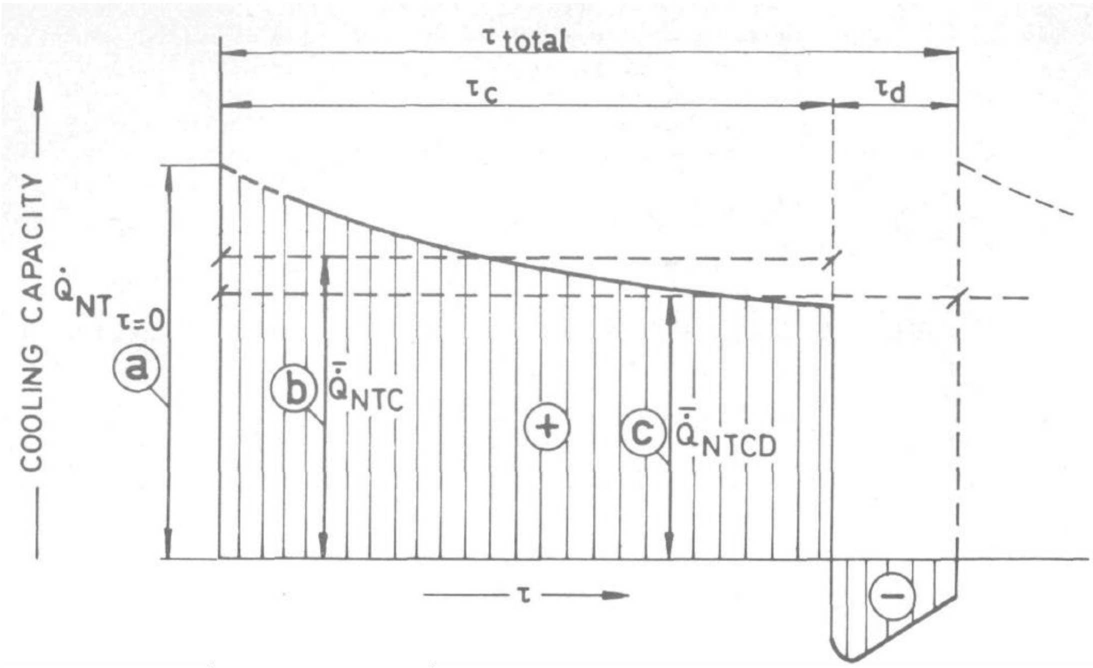

Figure 8. Principle of the deterioration of cooling capacity and required defrost capacity during operation.

When the exact melting procedure of the frost starts depends on the control of the unit. An empirical correlation of cooling capacity deterioration from a relevant study is used, thus transient conditions can be modelled. (Lenic, Trp, & Frankovic, 2009)

While this investigation aims to understand how frost layer resistance impacts the heat flux, there is still another very important aspect to remember: the free flow area for the air becomes smaller resulting in a higher pressure drop and, more critically, a lower air flowrate. Nevertheless, it can be possible to have a reasonably constant airflow, by controlling the fan speed. Any fan has unique characteristics which are dependent on the air volume rate and static pressure head, so called fan performance curves. Thus, to predict the impact on fan power, unfrosted and frosted coil pressure drop has to be known. A rule of a thumb suggests that a completely dry coil, removing only sensible heat, offers approximately one-third less resistance to airflow than a dehumidifying coil which removes both sensible and latent heat. (ASHRAE, 2016)

2.1.5 Defrosting

Successful removal of frost is mostly dependent on interruption time of the dehumidifier operation, which obviously should be as short as possible. Likewise, energy usage must be as low as possible, prioritizing reclaimed heat usage. These are acknowledged defrost design questions that have to be addressed.

2.1.5.1 Defrosting methods

Various ways of defrosting have been used in a finned heat exchanger. No method is perfect and there are cons and pros for each of them. Some more known methods are:

- Self defrosting
- Electrical defrosting
- Hot gas defrosting
- Subcooling heat defrosting
- Sprayed warm water defrosting

From these the least defrosting energy is required for self-defrosting, however due to limited possibility to control the process and the long time it takes, normally self-defrosting is not used in a dehumidifier. Electric defrosting is a rather expensive method as manufacturing resistance cords into the cooling coil is complicated, as well as the energy use is higher, moreover it is difficult to melt the frost completely. Hot gas defrosting is a reversed refrigeration cycle, where evaporator is used as a condenser during the defrosting process. It is a quite popular method, as it allows for a fast and energy efficient frost removal, however mainly applicable in direct expansion systems.

An energy-wise solution is to use heat that is generated by the refrigeration system. Since the temperature requirement for defrosting heat can be relatively low, it can be generated by subcooling the refrigerant after the condenser. In order to have this heat available anytime, a thermal storage is most likely needed. This method would suit the best such system that is analysed, therefore it is chosen for this study.

2.1.5.2 Defrosting control strategy

A proper control strategy is crucial for the optimum performance of the refrigeration type dehumidifier. Two important aspects of defrost control are:

- Initiation of defrosting after reasonable intervals
- Termination of defrosting at the right time, just when all frost is removed

If the latter aspect is simpler, the first is more complicated to design as the whole process is transient and predicting the actual moment at which to start defrosting is the most optimum is a challenge. In general if one aims to achieve higher energy efficiency, the priority is longer intervals (provided capacity is enough), while to achieve maximum net cooling capacity, relatively shorter intervals are needed.

<u>Initiating</u>

Several defrost initiating strategies used in practice and most recognized are:

- Time control
- Demand control
    - Based on UA-value deterioration over time
    - Based on changes in the pressure drop on the air side due to frost accumulation
    - Based on changes in the fan electric power demand (due to flow resistance on the air side)
    - Based on optical sensors measuring the frost thickness at suitable positions

None of these methods are ideal and the most common is time control. For applications where air temperature and humidity are variating during operation it is favourable to use ”frost map” to adapt the intervals between defrost cycles. This allows to initiate defrost when capacity has dropped by around 30%. (Zhu et al., 2015) The optimum intervals for defrosting have been treated by Granryd and as the energy saving approach is concerned, intervals should be longer and it takes about twice the time compared to what is optimal for the maximum capacity. Intervals can vary a lot depending on operating conditions and moisture load.

2.2 Modelling

A theoretical background alone does not give a complete picture of the technology potential or possible limitations. Based on the theoretical background several case studies are done to analyse the performance of refrigeration based dehumidification in a manner of striving to imitate conditions as they would be in reality.

2.2.1 General assumptions for ice rink application

The study is conducted using a theoretical model of a refrigeration type dehumidification. In ice rinks where this technology is applied, most likely dehumidification is done in the air handling unit (AHU), before being heated to the required supply temperature, and a schematic is similar as shown in Figure 3. AHU in an ice rink serves for space heating function primarily, in addition providing the ventilation when required, which in ice rinks is rarely needed. In this study fresh outdoor air intake is assumed to be zero. Another aspect is that cooling is achieved using the same refrigeration plant as for ice cooling with a connection to the secondary loop. Indirect configuration would be most likely implemented when the refrigeration technology is chosen for dehumidification, as a separate refrigeration unit for dehumidification adds complexity and components. Therefore, this study aims to analyse only such type of solutions.

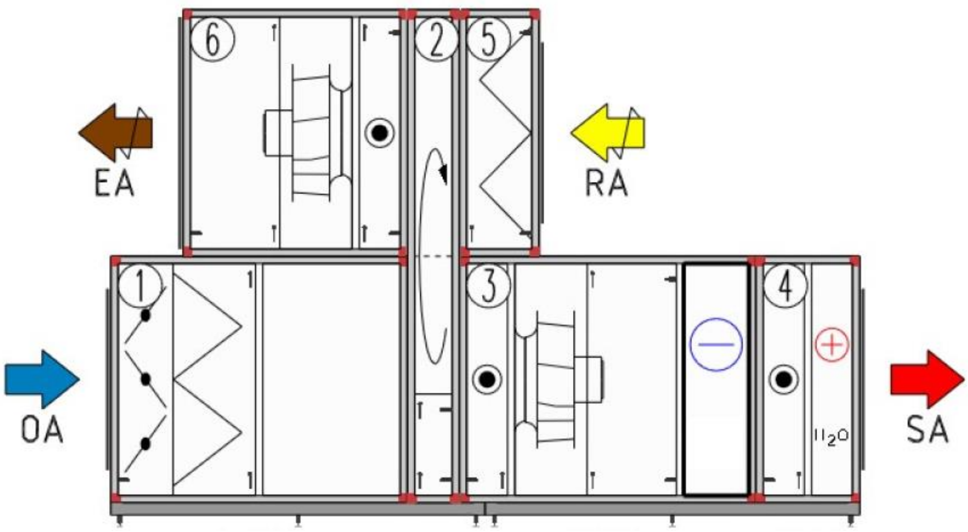

Figure 9 An example of an air handling unit configuration that represents this study case.

Some of the key assumptions are indicated in the Table 1, and these are according to what would best represent such system in reality.

Table 1 Several key assumptions for the theoretical model.

Aspect	Assumption
Type	Indirect
Cooling coil placement	Embedded within the air handling unit
Cooling unit	Ice rink’s refrigeration plant
Outdoor air intake	None

Spectator seats (ice rink size)	500 to 1000
Volume of the rink space	20000 to 25000 m3

2.2.2 Nominal conditions

First case is set for design conditions in a typical size ice rink in Sweden. The dehumidification function in this setup would provide moisture safe indoor climate over the season.

Table 2 Constant variables for nominal case.

Variable	Value	Unit
Water removal capacity $\dot{m}_{dh}$	20	kg H2O per hour
Inlet temperature t_{in}	+8	°C
Inlet relative humidity rh_{in}	60	%
Inlet air dewpoint $t_{d,in}$	+0.7	°C
Air flowrate $\dot{V}_{supply}$	5	m^3/s
Supply temperature t_{supply}	+8	°C
Defrosting initiation time	80	min
Defrosting time	12	min
Defrosting heating capacity	20	kW

2.2.3 Limitations for frost free operation

The first aspect to consider is one of the main drawbacks in a refrigeration type dehumidifier - the frost formation. Obviously, it is not a problem when the surface wall temperature is above 0°C, however a system is modelled to show better why in an ice rink it must be below 0°C. In order to have model working some of the conditions must be assumed constant and these are also based on the design conditions, compiled in Table 2.

The indoor air conditions in this study are set to 8°C and 60% RH, which can be converted to 0.7°C dewpoint, that is why there is no way physically how it could be achieved with a wall temperature higher than that, and again the minimum is 0°C for no frost conditions. In this case the temperature range for the wall surface is between 0 to +0.6°C.

What is a practical potential for frost free configuration or in other words what dehumidification capacity would be achieved in reality? As it turns out with nominal airflow it would be up to around 4 kg/h at 0°C wall temperature. Clearly it is far from what this function is expected to provide. Dehumidification capacity and wall temperature relation is illustrated in Figure 10.

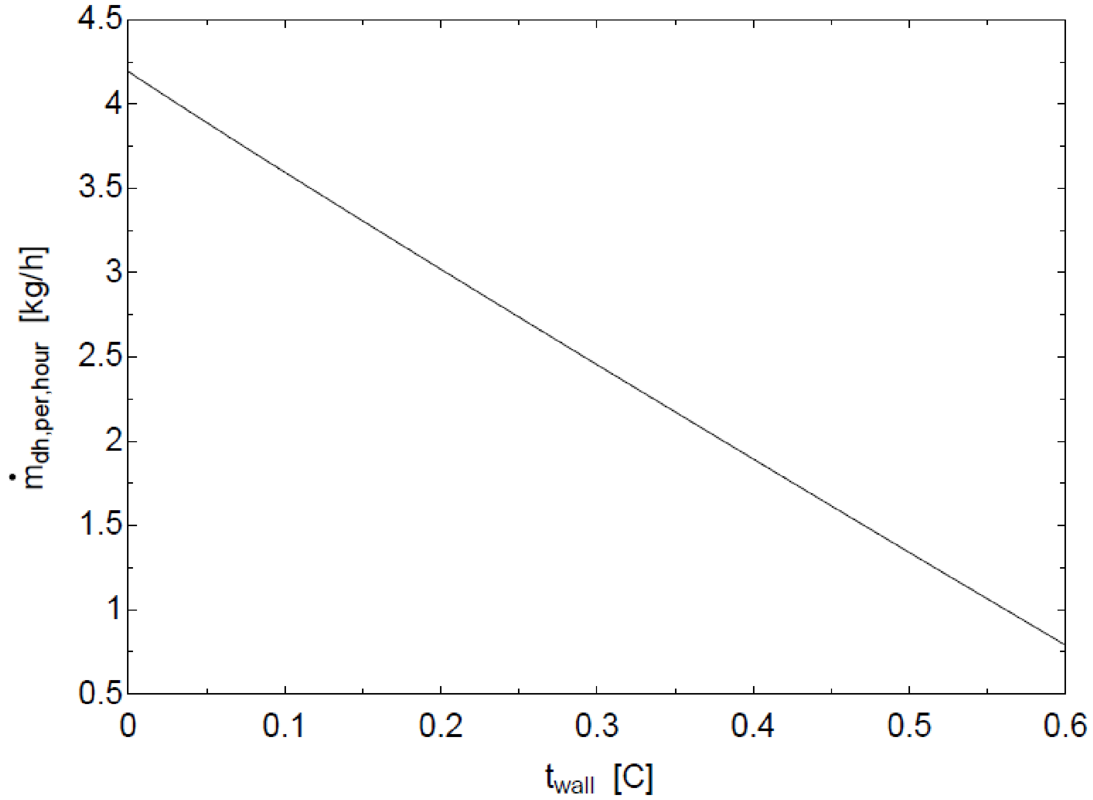

Figure 10 Water removal capacity for coil temperature above 0°C at constant airflow.

Again this is an evident argument for the fact that refrigeration type dehumidification in reality may deliver the required capacity only with a wall temperature under 0°C thus frost formation must be dealt with.

2.2.4 Cooling capacity requirements

One of the main questions are about the machinery size in refrigeration type configuration, as it would help to acknowledge the capital cost investment and to conclude whether such is a reasonable solution compared to the sorption type.

The assumptions are again made as indicated in Table 2. The wall temperature is assumed to be -8°C, which is reasonable if considering a commonly designed indirect ice rink refrigeration system with a secondary fluid temperature in that order of magnitude. The deterioration of the cooling capacity as it is calculated in the model is illustrated in Figure 11, and it must be noted that the required installation capacity can depend on several aspects. For example, if aiming for a particular dehumidification capacity at certain inlet air conditions, then defrosting interval will impact the requirement for installed capacity. It can be better explained using the graph, where cooling capacity change over time during frost formation are shown, with a dehumidification capacity of 20 kg per hour in both cases, but with a different defrost interval.

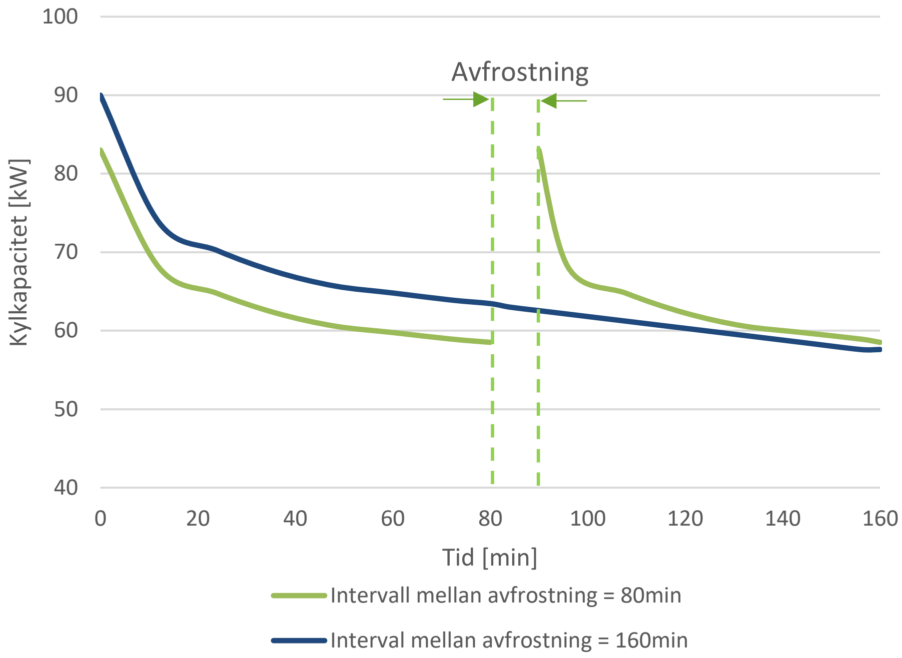

Figure 11 Cooling capacity deterioration over time with two different intervals between defrosting cycles.

First is an interval of 80 minutes, which as mentioned before is considered when optimizing for the highest capacity. The other being 160 minutes, when optimizing for energy efficiency, due to the fact that less frequent defrosting cycles are needed. So in order to have the same dehumidification capacity, average cooling capacities over time must be equal in both intervals, which implies that the initial cooling capacity of the cycle with defrost interval of 160 minutes must be higher than for the shorter one.

In Figure 12 installation cooling capacity can be found for a range of water removal rates, and it suggests that for a nominal dehumidification capacity of 20 kg of water per hour, 84 kW of installed cooling capacity would be required.

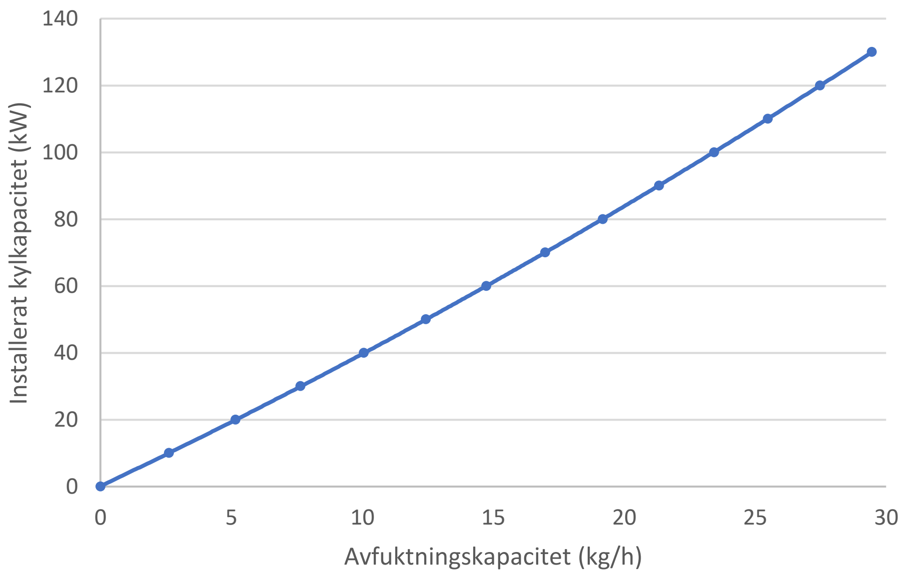

Figure 12 Cooling capacity requirement for the respective dehumidification capacity with respect to the assumptions in Table 2.

2.2.5 Air flowrate requirements

Air flowrate is assumed to be constant (5 m^3/s) because the most likely arrangement for the equipment would be a dehumidification function embedded within the main ventilation and space heating AHU, as it is a cheaper solution with respect to the capital cost. However it is interesting to understand what airflow rate is actually needed, if maybe a separate dehumidifier would be installed which means the airflow would be controlled according to the dehumidification capacity.

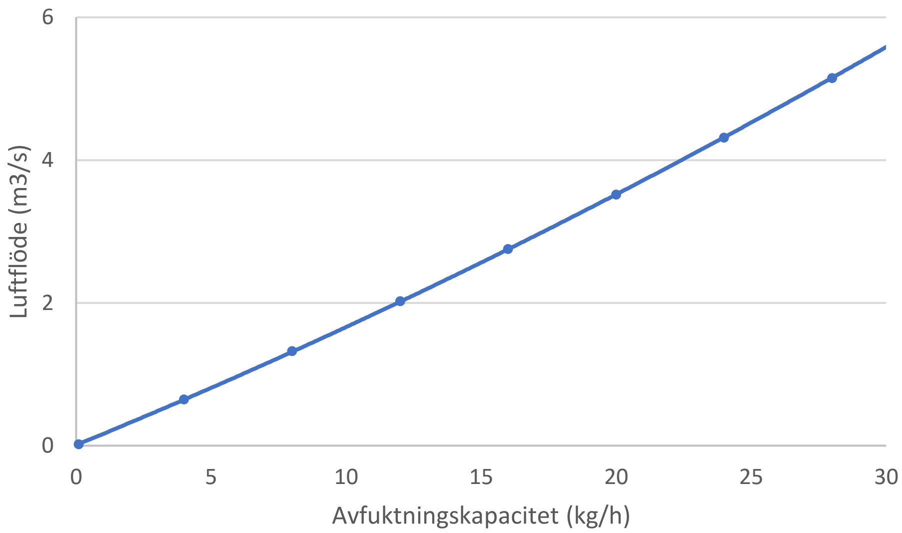

Figure 13 Airflow rate required for the respective dehumidification capacity.

Figure 13 suggest that an airflow rate in the range of 3.5 m^3/s would be required for the nominal dehumidification capacity (20 kg of water per hour). Compared to sorption 2nd generation technology with similar dehumidification capacity, which requires around 2 m^3/s.

2.2.6 Energy use

The performance over a season is a rather important aspect because the energy bill should never be compromised. When energy efficiency is concerned, interval between defrost initiations should be as long as possible, still with respect to the required capacity. As mentioned previously, a rule of a thumb suggest that this interval should be twice as the one for optimum UA value (80 minutes) and for energy calculation it is set to 160 minutes.

Table 3 Some key assumptions and average parameters for energy performance evaluation.

Month	Days	Defrost time [min]	Interval between defrost [min]	Volume flowrate [m3/s]	Average water removal rate [kg/h]	Average cooling capacity [kW]	COP2
Jul	6	10	160	5	15.5	48.4	3
Aug	31				15	46.9	3
Sept	30				12	37.5	3
Oct	31				4	12.5	3
Nov	30				4	12.5	3
Dec	31				2	6.2	3
Jan	31				0.2	0.6	3

Feb	28				0.4	1.2	3
Mar	15				0.5	1.6	3

Results in Table 4 show that main portion of energy is required to provide cooling in coil, but at the same time fan energy cannot be neglected and accounts for a quarter of total electricity use of the process. Heating energy is also relatively high, but since the source can be heat recovery from the process itself, it should not be considered as an additional expense.

Table 4 Energy use results monthly and total.

Month	Cooling demand [kWh]	Electricity use for refrigeration [kWh]	Electricity for the process fan [kWh]	Total electricity use [kWh]	Heating energy for defrosting [kWh]	Post heating energy [kWh]
Jul	6561	2248	137	2386	219	5011
Aug	32797	11239	710	11949	1087	25054
Sept	25397	8703	687	9390	841	19397
Oct	8747	2997	710	3707	290	6680
Nov	8465	2901	687	3587	281	6465
Dec	4375	1499	710	2209	145	3340
Jan	437	150	710	860	14	334
Feb	790	271	641	912	26	603
Mar	529	181	343	525	18	404
Total	**88098**	**30190**	**5333**	**35524**	**2920**	**67288**

2.2.7 Limitations

A constant fluid temperature is assumed inside the cooling coil. A more precise analysis would be when using a stepwise temperature gradient so that having a more realistic model. It would affect the dehumidification capacity and so other parameters; however it should not be to a great extent. Such analysis would also require much more information for the cooling coil heat exchange properties and would be limited to specific size coil.

The deterioration in UA value is assumed using an empirical relation, which is reasonably justified. However more arguments that would prove this relation would help.

As the frost is building up, the coil surface temperature is not anymore the one that is in direct contact with the air, but it is the ice surface itself. It should be examined, whether it is a significant effect. And it is not clear whether this effect is already accounted within the cooling capacity deterioration.

Available airflow passage in the cooling coil becomes less as the frost is accumulating. In the model it is treated as the pressure drop increase and assumed that fans can provide with the same airflow by increasing speed according to the rise in pressure drop. In such scenario, velocity will increase, and it may impact the bypass factor. It is not accounted in the model because it requires very detailed analysis with specific cooling coil and fan designs, and this is not within scope of this study.

2.3 Examples from field

There are not many ice rinks with a refrigeration based dehumidification in the field, but some references are available as to how the systems are designed.

The first example is shown in Figure 14, where dehumidification is intended to be achieved with a secondary fluid temperature around -8°C at 42 kW cooling capacity and air flowrate 16000 m3/h. The design inlet air conditions are quite dry (dewpoint -1.3°C) and it may not be fair to evaluate the system with such low moisture content, therefore the typical summer indoor air conditions are used (8°C and 65% RH or 1.8°C dewpoint). Defrosting is needed for this configuration since sub-zero temperatures in the coil. The modelled attainable dehumidification capacity is 3.5 kg/h, which is a very low figure, and this is because the cooling capacity requirement is underestimated.

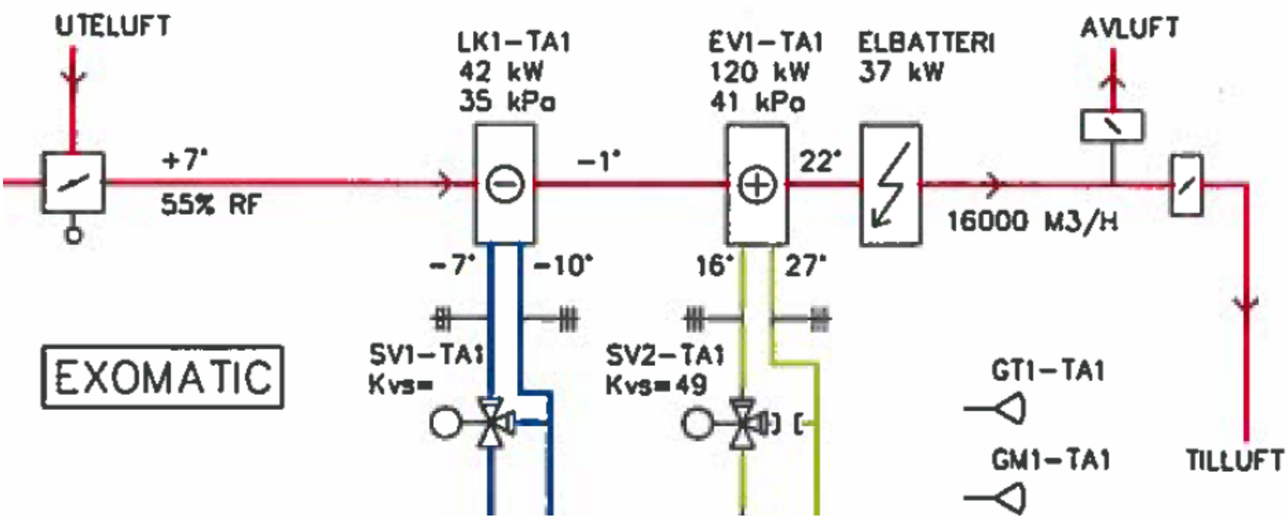

Figure 14 An example design of air handling unit with undersized dehumidification capacity due to low cooling capacity dimensioning.

Another design example is illustrated in Figure 15, where for a large arena refrigeration dehumidification is selected. The cooling fluid temperatures are close to 0°C to avoid frost formation. Inlet air conditions are not fully specified, with only temperature given as 12°C. In such a large arena air can be assumed to be more humid, as the critical dewpoint can be allowed to be higher than in a regular size rink. In this case the assumed dewpoint temperature is around 3.5°C. For a larger event, which may take place since there are seats for 7000 spectators, the indoor air temperature is assumed even higher (15°C) same as the dewpoint temperature (4°C).

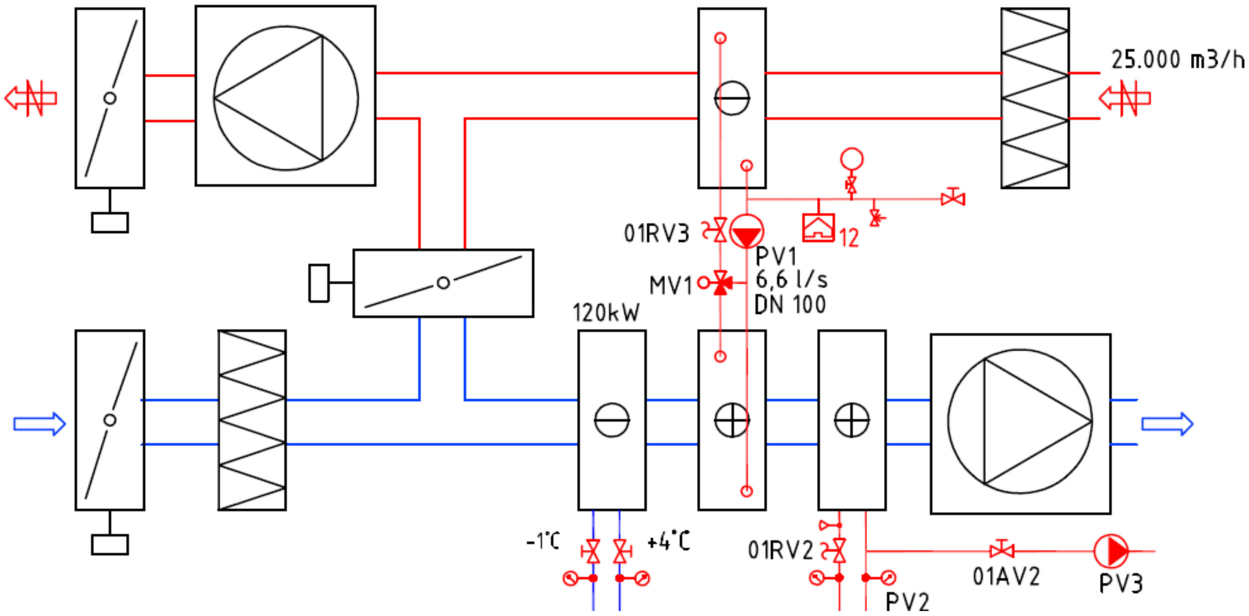

Figure 15 An example design of air handling unit with undersized dehumidification capacity due to low secondary fluid temperatures.

The results of two cases are shown in Table 5. The design cooling capacity is likely to be sized for high inlet air temperature during event, and obtainable dehumidification capacity is around 23 kg/h. For a regular operation less cooling demand is needed however a lower dehumidification capacity is achievable (18 kg/h). From this example it is clear that even with such a high cooling capacity and airflow the unit can provide what sorption would be able to deliver at drier inlet conditions and lower airflow. In addition as this is for a large arena, the dehumidification demand is also significantly higher than in a normal size arena. A rule of a thumb is that as much as 3 times higher dehumidification capacity is needed, which would mean around 360 kW of extra cooling capacity to be installed being a very costly solution.

Table 5 Performance of a refrigeration based dehumidifier in large arena with non-frost conditions.

Case	Indoor air temperature [°C]	Dewpoint temperature [°C]	Cooling capacity [kW]	Dehumidification capacity [kg/h]
Event	15	4.0	120	23
Regular	12	3.5	95	18

3 Sorption type dehumidification

The alternative to refrigeration is the sorption technology, which works with a different principle. The adsorption technology makes use of certain chemical compound properties in order to absorb water molecules without condensation and freezing being involved. This is the most used technology in ice rinks today due to its effective performance even in the sub-zero dewpoint range.

3.1 Theory

The in-depth processes in sorption technology are more associated with chemistry rather than physics. But it is not required to study it completely because the energy flows are easy to follow as this technology does not involve sublimation and eventual frost formation.

3.1.1 Working principles

The desiccant rotor behaves like a water vapor filter, where air passing through the rotor gives off a big part of its moisture content to the desiccant that constitute the major content of the rotor matrix. Alternate flat and wave-shaped thin walls create narrow channels where the airstream can pass at a low pressure-drop and in close contact with the desiccant walls.

All desiccants have limited moisture capacity and need to be reactivated to get the capability to adsorb water vapor back. In a rotor-dehumidifier this is achieved by letting a small hot airstream pass through a small sector of the rotor, while the rotor slowly rotates between the sector where moisture is picked up and the sector with hot air where moisture is released from the desiccant. The two airstreams are kept apart by internal ducting and seals against the flat rotor surface. While the dry air is blown into the area to be kept dry, the exhaust of the reactivation-airstream is led to the outside. By this arrangement the rotor-dehumidifier can offer a continuous supply of dry air for a large number of applications.

There are several types of desiccants, all with the common ability to attract and trap the water molecules from the surrounding air, but otherwise different making them more or less suitable for a certain application.

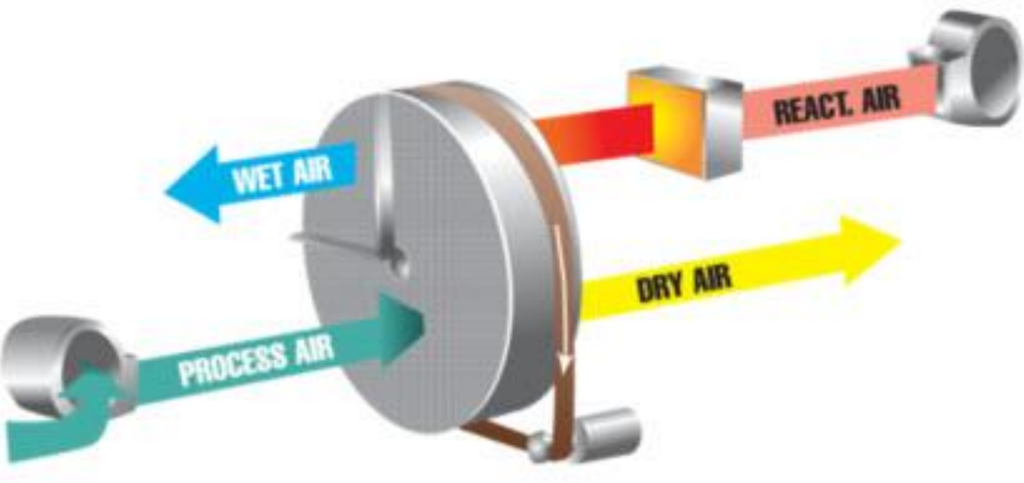

Figure 16 Working principle of the sorption type dehumidifier.

Due to the physical process that is involved in this cycle, dew point temperatures lower than 0°C can be reached without problems related to frost formation (for refrigeration type dehumidifiers this is not possible), lowering the requirements of air volumes in the process as well. The highest share of energy usage in the cycle goes to the reactivation air heating, therefore it is where the highest potential for energy savings can be found. As the refrigeration system has a plenty of surplus heat, it could be utilized for reactivation air heating instead of being rejected to the outdoor air. However, a relatively high temperature level is necessary for this process, which makes it challenging to reactivate the wheel completely with heat recovery only.

3.1.2 Reactivation heat choice

Sorption dehumidification technology utilizes heat to regenerate desiccant wheel, which is the major energy input into the unit. Source of heat can be any origin, as long as capacity is at temperature level high enough for the reactivation process to be complete. From available and economically justifiable heat sources in Sweden, a priority list is as follows:

1. Recovered heat
2. District heat
3. Electricity

Such preference order should always be followed, as it proves in all normal cases to be the least expensive choice sequence from operational energy perspective. The only question is whether it is doable in the particular ice rink.

3.2 Examples from field installations

A number of ice rinks in Sweden with sorption type dehumidifiers have been analysed using an online monitoring system that saves the measurement data from periods of operation. The data consists of air parameters, electrical and heating energy measurements. Using these various aspects of dehumidifier performance can be assessed which supports the theoretical knowledge.

3.2.1 Heating capacity requirements

One of the key components in the sorption type dehumidifier is the regeneration heater and this is where most of the energy is utilized. Air that is taken from the outside must be heated to the design regeneration air temperature. Depending on heat source, different temperatures and eventually airflows are required. With electric heaters the regeneration temperature is 120°C while with heat recovery 50°C is needed. But in the end heating capacity for a nominal case is around 40 kW regardless of the supply temperature, that is why generation 2 requires higher air flowrate for reactivation.

3.2.2 Airflow requirements

To obtain certain dehumidification capacity with certain heat source temperature, reasonable air flowrates are sized by the dehumidifier manufacturers. Examples of units from several field installations are given in Table 6. All of the units in these examples are sorption technology, but in different configurations, four of them are generation one, while in ice rink 5 there is the generation two dehumidifier. It is apparent that for generation two, since the reactivation temperature is lower, significantly more air is needed for reactivation – 1.1 m3/s compared to 0.3 m3/s for similar dehumidification capacity. However, considering the fact that heat input for generation two can be from recovered heat, it is more advantageous, because fans account for around 16% in generation two.

Table 6 Air flow rates and fan capacities of sorption dehumidifiers from several ice rinks.

Ice rink	Type of unit	Dehumidification capacity [kg/h] (@ 8°C 60% RH)	Process fan airflow rate [m3/s]	Process fan rated power [kW]	Reactivation fan airflow rate [m3/s]	Reactivation fan rated power [kW]
Ice rink 1	Gen 1	20	1.53	3	0.3	1.1
Ice rink 2	Gen 1	12	0.77	3	0.19	1.5
Ice rink 3	Gen 1	23.5	1.67	n/a	0.47	n/a
Ice rink 4	Gen 1	19	1.11	3	0.35	1.5
Ice rink 5	Gen 2	20.6 kg/h (@ 10°C and 61% RH)	2.08	n/a	1.06	n/a

3.2.3 Heat recovery feasibility

One of the solutions is to use the discharge heat of the refrigeration system, which is otherwise rejected to the ambient if not utilised for other functions. However, it is important that the heat must be available at a high temperature level in order to be usable in dehumidification process, which makes it a challenge to heat the regeneration air only with recycled heat from the cooling system. Most of the traditional dehumidifiers need up to 120°C hot regeneration air, but in the recent years, improvements allowed to reduce the requirement. Today there are sorption dehumidifiers which can operate with around 55°C, which opens up the possibility to cover the entire heat demand in the dehumidification process with recycled heat from the cooling system. The drawback of these models is that the fan power often increases as the flowrate rises, which is mostly acceptable because it is a matter of using a heat which is “for free”, making the solution a profitable alternative.

The refrigeration system needs to deliver 30-50 kW heat at around 60°C to be able to cover the demand of the sorption dehumidifier. Recently cooling systems that are based on CO_2 as the refrigerant have become more popular, and these can generate heat which fulfils this requirement. Another solution is to connect a heat pump to the coolant side in the cooling system which can boost the discharge heat of the cooling system to the necessary temperature level.

4 Comparison between refrigeration and sorption dehumidification technology performance

4.1 Other studies

A comparative study has shown a significant difference between the technologies applied in an ice rink. One of the major drawbacks for refrigeration dehumidification technology is concluded to be the airflow requirement. For the same dehumidification capacity – 24.3 kg of water per hour at 2°C dewpoint process air, 17000 m3/h air flowrate is necessary for the refrigeration technology, while sorption can provide the same with around 5600 m3/h air flowrate. (Controlled Dehumidification, 2012)

4.2 Equipment size

When the investment cost is concerned, size of the equipment has a decisive role. Table 7 gives an indication of the required capacities for cooling, heating and fans. It serves however only as an approximate comparison, because the actual size for fans for instance depend on fan and cooling coil efficiencies and can vary by manufacturer. It is clear however that cooling capacity requirement will size up the refrigeration plant significantly, as larger compressors are needed, while heating capacity can be a matter of extra heat exchanger and connection piping if heat recovery solution is used.

Table 7 Comparison of the size of the equipment for both technologies.

Technology	Cooling capacity [kW]	Heating capacity [kW]	Process air fan power [kW]	Reactivation air fan power [kW]
Sorption gen 1	-	30 - 50	~ 3	1 to 1.5
Sorption gen 2	-	30 - 50	~ 4.5	~ 2
Refrigeration	80 - 90	50	~ 7	-

4.3 Energy use

Monthly average dehumidification demand data in an ice rink over the whole season is available which allows to evaluate how much energy the process requires. As for the sorption technology - direct energy use measurements are compiled, while refrigeration technology is evaluated using the theoretical modelling method as described in previous chapters.

Several key assumptions are made when energy use for the dehumidification function is calculated. To better understand the approach, re-examination of Figure 3 can help. As the dehumidification effect is achieved in a cooling coil built in the air handling unit, assumption about the fan energy for dehumidification is that pressure drop across the cooling coil will define it. So, because fans in an air handling unit are running constantly to keep the rink space warm, even when there is no dehumidification demand, the cooling coil imposes an additional pressure drop for fans to overcome. Eventually pressure drop across the coil in relation to the total fan pressure is considered as the extra fan energy input for the refrigeration based dehumidification.

Sorption technology can have various heat sources, which will definitely make installation and operation costs different. In Table 8, with the conventional configuration is considered the fully electricity reactivated desiccant wheel with heat temperature up to 120°C. Even though less energy is required for the fans, heaters consume a lot of electricity, thus the total purchased electricity is the highest compared to other alternatives.

The benefit of generation one sorption dehumidifier is that it would utilise either excess heat from the refrigeration system or district heating as a preheating source while still completing reactivation with electric heaters. In both cases the purchased electricity is going to be reduced obviously, however district heating energy is also not for free and has to be considered in case when its chosen as a source - though cheaper than electricity a better option is to use the heat recovery.

The generation two is an even more advanced solution to replace electric heaters completely with either heat recovery or district heating. It is doable as the heating temperatures are up to 60°C, on the cost of an increased airflow requirements. When heat recovery is feasible at this temperature level it is evident that this solution is the least energy intense from the operational point of view, even if it results in higher fan energy use.

The refrigeration based dehumidification would consume significantly more than the sorption 2nd generation configuration, but according to the results less energy is needed than for the conventional than 2nd generation sorption dehumidifier.

Table 8 Energy use results for various possible dehumidifier configurations.

Typ	Värmekälla	Elenergi kylsystem [kWh]	Elenergi fläktar [kWh]	Värmeenergi behov [kWh]	VåV [kWh]	Tillägsvärme [kWh]	**Köpt elenergi [kWh]**	**Köpt värmeenergi [kWh]**
Sorption vanliga	Elvärme	0	9043	55700	0	55700	64743	0
Sorption "gen 1"	1:a steg - VåV 2:a steg - Elvärme	0	9043	55700	23900	31800	40843	0
Sorption "gen 1"	1:a steg - Fjärrvärme 2:a steg - Elvärme	0	9043	55700	0	55700	40843	23900
Sorption "gen 2"	VåV	0	14116	55700	55700	0	14116	0
Sorption "gen 2"	Fjärrvärme	0	14116	55700	0	55700	14116	55700
Kyl	-	30190	5333	70208	70208	0	35523	0

From the energy use perspective, the refrigeration based dehumidification does not seem to be the worst choice, but conclusions can be made by evaluating systems holistically, by knowing total life cycle costs and considering practical performance of the system.

It has to be acknowledged though that for the refrigeration dehumidification evaluation is done with a theoretical model while for sorption technology the numbers are from field measurements. It means there may be deviations from the obtained value. Another important aspect is how to account for the energy use only for dehumidification as in this case it is a part of the air handling unit, e.g., fan energy is assumed to be the portion of energy that is due to additional pressure drop over the cooling coil.

4.4 Efficient load matching

In an ice rink cooling and heating loads are varying significantly over the season. These variations can either be taken advantage of or mismanaged causing even increased energy use. As a matter of fact, dehumidification system is one of the best representations of how loads can be wisely managed to have a synergy.

First thing is to prove how the refrigeration plant in an ice rink is going to be magnified if a refrigeration type dehumidification is used. Using field data from an existing ice rink where energy systems are monitored over the whole season, cooling capacities at corresponding ambient temperatures are included in Figure 17. And obtained cooling capacity requirements for the refrigeration type dehumidification based on water removal rate are added to the base cooling load of the ice rink. This shows firstly how size of the refrigeration unit is increased when this dehumidification technology is used, due to coinciding peak loads for both functions. Such concept increases the capital and possibly operational cost because the performance of the cooling plant is usually lower when it is warmer outside.

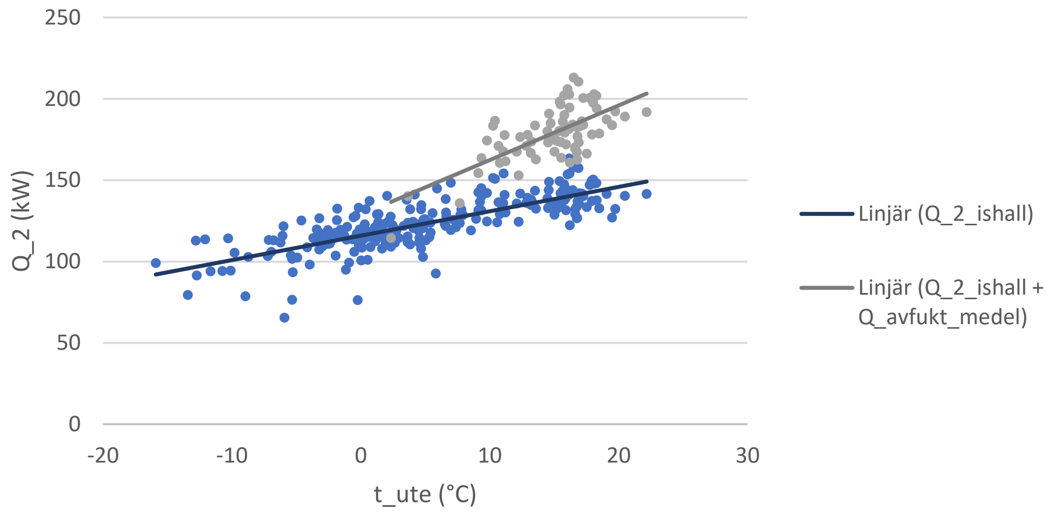

Figure 17 Cooling capacity requirement for the system where dehumidification cooling capacity adds load significantly on top of the base ice rink load.

To put this into perspective with sorption technology, Figure 18 is shown, with focus on monthly heating energy that is generated by the refrigeration unit and either can be recovered or otherwise must be rejected to the ambient. In addition, heating energy demand for the sorption type dehumidification is shown, and as can be concluded the pattern of demand follows the pattern of what is available. It can be called as efficient match, as the required heating energy for the dehumidification process is covered by energy that would otherwise be simply rejected to the outdoors, i.e., heat recovery utilization is increased with sorption technology.

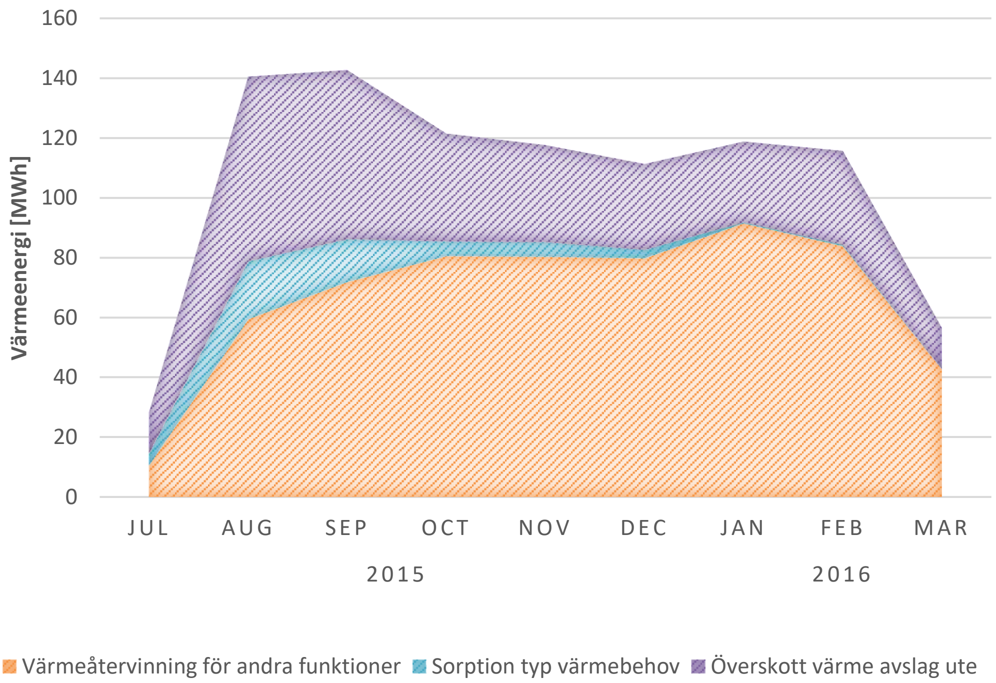

Figure 18 Monthly rejected heat sinks from a refrigeration system.

Intuitively there is a reasoning that refrigeration based dehumidification may be beneficial due to increased heat recovery potential, as it generates additional heat. As concluded from Figure 18, there is not a real need for an additional heat during the warm part of the season. In fact, by examining Figure 19 for monthly profile of the heat generated by the refrigeration plant, it turns out that the refrigeration type dehumidification generates even more excess heat that has to be discharged to the ambient. It may be said that heat recovery is increased slightly, but it is less efficient, because the process uses this heat for itself and in the end, more has to be discharged to the ambient, consuming more energy for the fans.

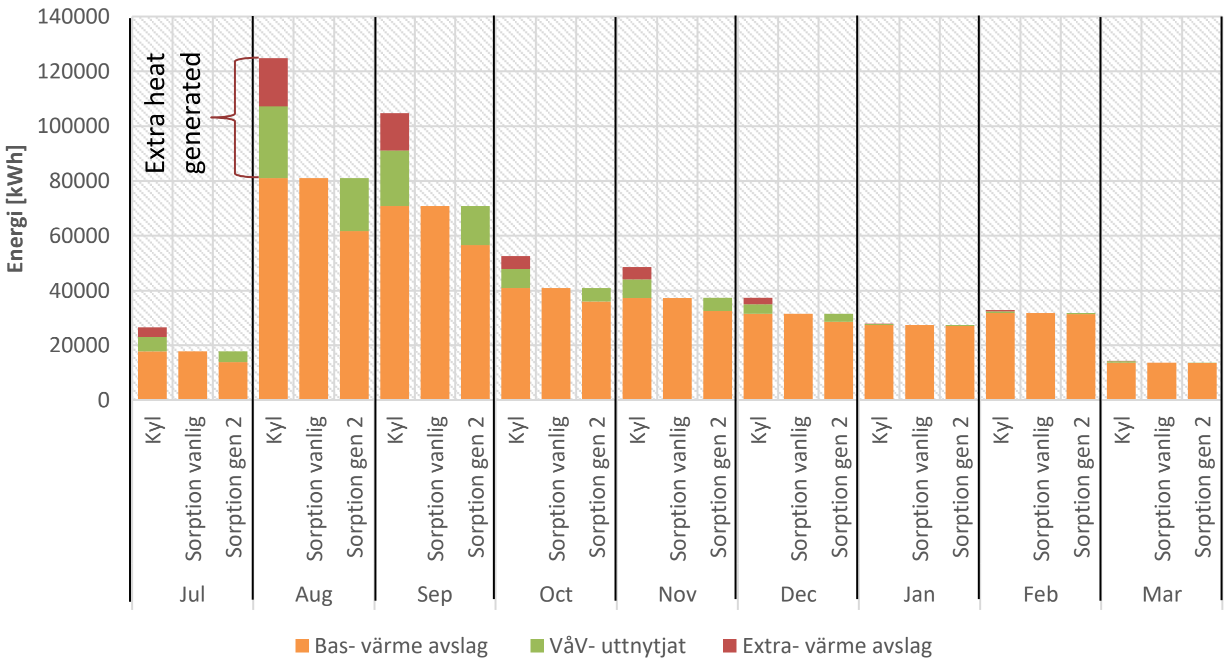

Figure 19 Comparison of the heat rejection requirements for standard sorption, "generation 2" sorption and refrigeration dehumidification technologies.

The total distribution of the heat from the high pressure side of the refrigeration system with comparison between dehumidification technologies used is shown in Table 9. As was concluded before, that even heat recovery with refrigeration type dehumidification is higher than sorption by 2% it is because the unit requires heat for itself. What is a concern is that more heat must be rejected, and it turns out to be by around 30% more. The assumed benefits of a higher heat recovery have not proven to be valuable, and only in case if there would be a potential for heat export in the warm part of the season then it may be beneficial, but less likely that heat demand in other applications is higher during warm periods. As long as sufficiency for ice rink itself is concerned, there is excess of heat energy, especially during the warm periods.

Table 9 Heat recovery and rejection from a refrigeration system with two dehumidification technologies.

	Heat generated [MWh]	Heat recovered [MWh]	Heat rejected [MWh]
With sorption technology	994	669	325
With refrigeration technology	1112	683	428
Refrigeration vs sorption	+12%	+2%	+32%

5 Conclusions

Part 5 report of NERIS – Comparison of refrigeration and sorption dehumidification technologies in ice rinks – discusses about what are the differences in performance of two most common dehumidification technologies in ice rinks as to the function as well as annual energy footprint.

One method that is treated in depth is refrigeration based technology, where moisture is condensed and eventually removed from the air. Two working modes in a refrigeration based dehumidifier can be distinguished – with or without frost formation. The advantage of a frost-free operation is that defrosting cycles are avoided. The simulation results however show that dehumidification capacity in a frost-free is limited to only around 4 kg/h in design conditions, which is far from enough what ice rinks normally need, as concluded in previous parts a typical ice rink in Northern Europe would demand around 20 kg/h dehumidification. Therefore this leads to conclusion that sub-zero temperatures are needed to achieve high enough drying of air and eventually frost formation cannot be avoided.

Computer simulation results show that higher dehumidification capacity can be obtained in sub-zero conditions. This is however not as simple as reducing the temperature of the fluid – frost formation on coil surface reduces the performance and installed cooling capacity is higher than what would be needed if no frost formation is taken into account. The modelling results show that for a nominal design dehumidification capacity (20 kg/h) the required installed cooling capacity is around 85 kW.

Examples of designed refrigeration based dehumidification plants show that underestimation of cooling capacity requirement is common. First of all, latent heat must be accounted for, as it can be a significant part of the total demand, and secondly frosting and defrosting impact needs to be included in the design. For a large arena, although potentially frost free operating mode may be allowed due to acceptance of higher moisture content in air, in case of 60 kg/h dehumidification capacity, as much as 360 kW of cooling capacity is needed.

There is a lot of agreement in the literature that sorption technology is a better fit for such dry conditions as in ice rink. This is the reason why much more theoretical and field examples in ice rink applications are available. The equipment size can be determined in terms of design heating capacity and fan power. There are two sorption dehumidification modifications possible – known as generation one and two. These are distinguished by temperature required for regeneration process and main difference in terms of equipment is fan power. Generation one needs air temperature up to around 100°C with process airflow around 1.5 m3/s, reactivation airflow 0.3 m3/s, while generation two requires approximately 50°C air temperature, process airflow of 2 m3/s, reactivation airflow of 1 m3/s. The main reason in favor to generation two is the potential to utilize low grade heat from heat recovery of the refrigeration system.

When refrigeration and sorption technologies are compared, one key point that stands out is how load profiles of both processes fit with the loads of an ice rink. Refrigeration based dehumidification has a cooling demand that is highest when it is warmest outside, which is exactly how it is with cooling demand to maintain the ice. This implies that refrigeration system of an ice rink has to be sized according to the sum of these demands, while if sorption technology is chosen, there is no cooling demand and refrigeration units are covering only the load of the ice sheet. What is also in favour to sorption technology is that heating demand for the process is highest when there is most refrigeration system excess heat available and it can be can be recovered.

As regards annual energy requirements, both technologies are compared with different heat source options considered for sorption technology. What matters most is how much energy is eventually purchased and the best scenario is found to be sorption generation two technology with full heat recovery, requiring around 14.1 MWh of electricity on annual basis. Refrigeration based dehumidification energy use over a same period of time is calculated to be around 35.5 MWh.

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