

ICE RINK DEHUMIDIFICATION SYSTEMS ENERGY USAGE AND SAVING MEASURES

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Abstract: Ice skating rinks are one of the largest energy consumers in the building sector. The dehumidification function is one of the dominating energy users but is vital for the climate control in the ice rink. High humidity level causes extra diffusion load on the ice which affects the ice surface structure and risk to form condensation in the building structure. This study focuses on mass transfer phenomena in ice rinks. Three elements of the mass transfer mechanism are investigated: the mass transfer on the ice surface; the infiltration rate; and the dehumidifier energy consumption. The energy usage related to dehumidification of an ice rink is concluded to be as high as 15% of the total energy usage, which annually may correspond up to 150 MWh. A moisture balance model including infiltrations, moisture sources, condensation and dehumidification is proposed. A possible saving measure is to replace the electricity used for regeneration with recovered heat from the refrigeration system. Results from a case study show an electrical energy saving potential of 40% using other sources than electricity to heat the regeneration air.

Key Words: dehumidification, ice rink, infiltration, condensation loads

1. INTRODUCTION

1.1. Dehumidification in ice rinks

Ice rinks are public buildings using a large amount of energy due to simultaneous needs of heating, cooling, lighting and ventilation. In Sweden, the average energy consumption for an ice rink is 1000 MWh per year and Swedish ice rinks altogether use around 300 GWh per year (Rogstam 2010). Since the various needs entail various energy processes, reducing energy consumptions in ice rinks can be done in different ways but it requires a thorough knowledge of each energy system. Energy saving measures related to dehumidification systems in ice rinks is a subject that needs to be further investigated. This study focuses on understanding the underlying principles of the moisture balance and the potential energy savings related to the dehumidification function.

Moisture in ice rinks have various sources such as spectators, ice resurfacing water and outdoor air entering either through the ventilation system or through the building envelope. Humidity may cause discomfort issues, corrosion or rotting of the structure and deterioration of the ice quality (Karampour 2011). Moreover, humidity may lead to higher heat loads, directly by increasing the condensation loads and indirectly by increasing the number of ice resurfacing necessary to keep a good ice quality. Therefore, dehumidification is an important function, however, it is also important to consider from an energy usage perspective.

1.2. Dehumidification technologies

Two types of dehumidification technologies may be used in ice rinks: adsorption type and refrigeration type.

1.2.1. Adsorption type

The adsorption technology is based in the principle of dehumidifying using chemical sorbents, with an adsorption rotor coated with a special substance, such as silica gel that adsorbs the water molecules of the moisture in the transitory air. When the wheel is

saturated, it continues to a regeneration zone, where it is dried with warm air. The warm and humid air is led away out of the building. The rotor is again prepared to adsorb the water molecules of the air inside the building. The most recognized equipment which uses the adsorption type technique is referred to as “desiccant wheel”.

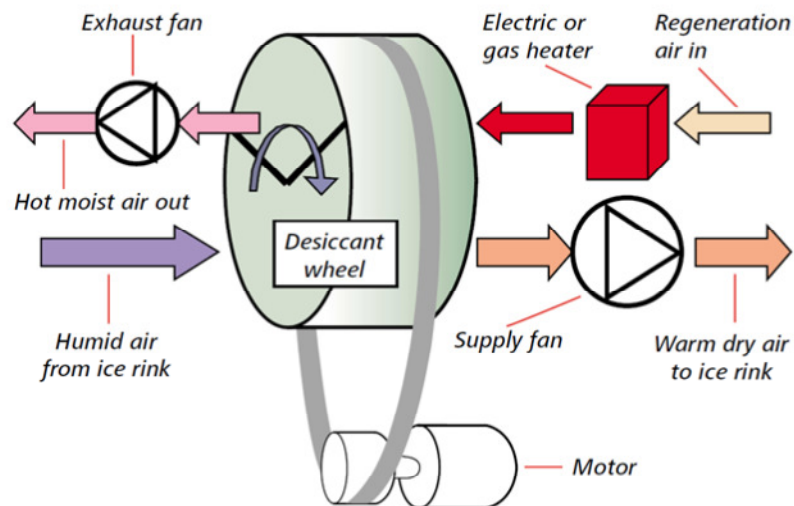


Figure 1: Dehumidification by desiccant wheel (IIFH 2010)

1.2.2. Refrigeration type

The refrigeration type is another technology to dehumidify the air, acting by the decrease of the air temperature below the dew temperature point, in order to condense the moisture of the air. The working principle of a refrigeration dehumidifier unit is that the air crosses the evaporator coil, cooling down until the dew temperature point, where condensation occurs. The cooled air is heated up in the condenser coil and, afterwards, the dry warm air is discharged to the room. For cooling the air, part of the cold brine from the refrigeration system unit can be used. Dehumidification by condensation can be integrated in the ventilation system together with the refrigeration system (Bermejo Pereira Rodrigues da Silva 2013).

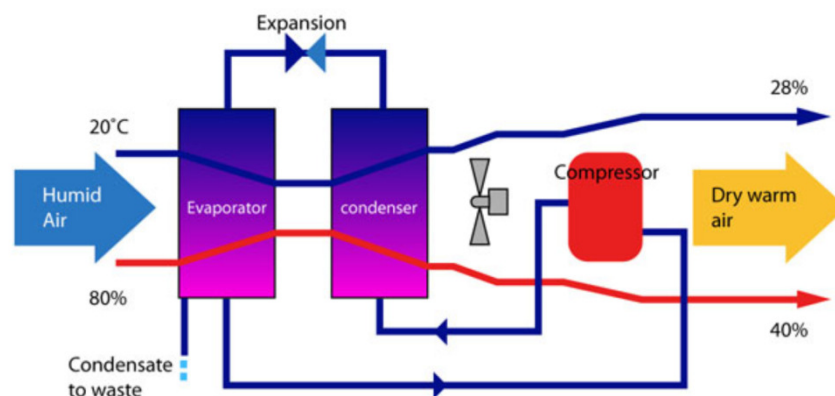


Figure 2: Principle of refrigeration type dehumidifier

1.3. Mass balance for moisture in ice rinks

The moisture balance in ice rinks can be expressed in a similar way like Jin-Taung and Yew Khoy (2010), but including internal moisture sources, \dot{S} (e.g. players, spectators, etc.)

$$\dot{m}_{inf} + \dot{S} + \Delta \dot{m}_{ven} = \dot{m}_{cond} + \dot{m}_{dh} + \dot{m}_{exf} \quad (1)$$

In smaller ice rinks where only a limited number of people are present fresh (outside) air should not be brought in. Indeed, the volume of air in the ice hall is significant and airborne

pollution is well diluted. Therefore, it assumed in this study that the mass flow term corresponding to ventilation equals 0. Thus, the air mass flows of infiltration and exfiltration are equal Considering the previous assumptions, eq.(1) can be rewritten using absolute humidity ratios, (not to confuse with , see eq.(8))

(2)

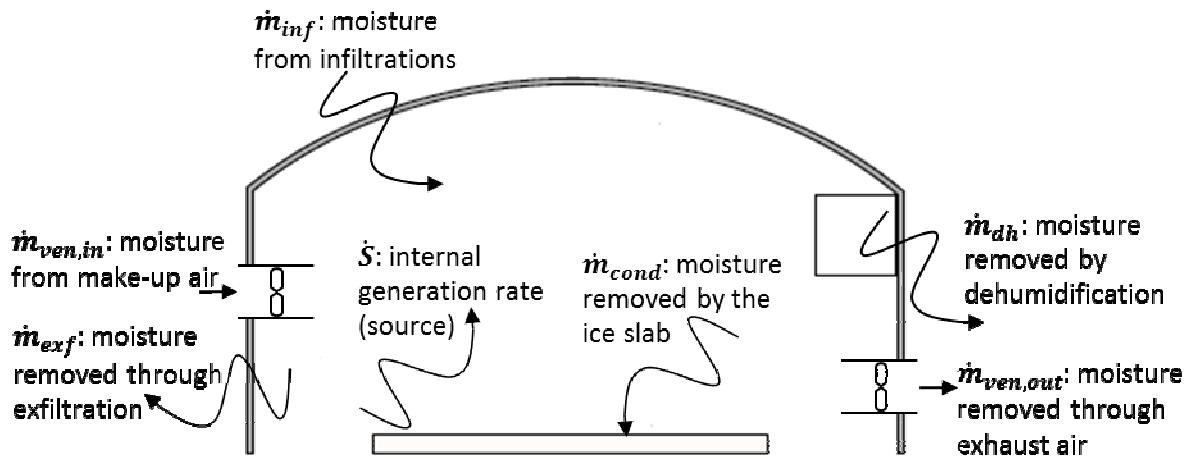


Figure 3: Moisture balance in an ice rink (Bermejo Pereira Rodrigues da Silva 2013)

The left-hand side of eq.(2) is the water vapor excess in the building which corresponds to the amount of water that will be removed either by the ice slab or the dehumidifier. The second term on the right hand side of eq.(2) may be deduced from the dehumidifier power or energy consumption and using the manufacturer data. Hence it should be possible to get an estimation of the infiltration rate of an ice rink building evaluating the mass transfer to the ice slab and the dehumidifier energy consumption.

2. DESCRIPTION AND MODELLING OF THE CONDENSATION LOADS AND DEHUMIDIFIERS

2.1. Condensation Loads

Due to the various physical mechanisms involved in the heat and mass transfer on the ice surface, finding accurate analytical solutions which fit experimental figures is a challenge. Moreover, these solutions may not be convenient to use when designing ice rink energy systems. Therefore, correlations may be considered to estimate the heat loads including condensation loads. Some correlations found in the literature are listed in the next part.

2.1.1. Correlations for condensation loads and transfer coefficients

Since the convection mechanism is referred to as the energy transfer due to random molecular (diffusion) and bulk (advection) motions (Incropera et al. 2011), it is expected to find similarities between heat and mass transfer both involving energy transfer.

The different condensation load correlations found in the literature often relate, indeed, heat and mass transfer coefficients. In the ice rink case, both mechanisms are influenced by the air velocity and the ice temperature.

Granryd et al. (2011) expressed the condensation heat flux, q_{cond} , with an equation analogous to the Newton's law of cooling, hence using a coefficient of heat transfer, h_{cond} , dedicated to the condensation load

where $h_{cond} = \dots$ (3)

With numerical values of 101 325 Pa, 18,015 g.mol⁻¹, 28,97 g.mol⁻¹, 1,003 kJ.kg⁻¹.K⁻¹ and 2850 kJ.kg⁻¹ for p , M_a , M_w and $c_{p,a}$ respectively, we obtain

$$\dot{q}''_{cond} = 1,74 \cdot 10^{-2} \cdot h_{cv} \cdot (P_v(T_a, \varphi) - P_{sat}(T_{ice})) \quad (4)$$

Kwan-Soo et al. (1997), Zhang and Niu (2002) and as well as Incropera et al. (2011) give a correlation for the coefficient of moist air mass transfer rather than for the coefficient of condensation heat transfer. They state that the coefficient of mass transfer could be calculated knowing the convection heat transfer coefficient, the specific heat capacity of the fluid in motion (moist air) and the Lewis number, Le , which is the ratio between the thermal diffusivity of moist air and the mass diffusivity of water vapor in air. The two first groups of authors assume a value of 0 for n (Reynolds analogy), while Incropera, et al. (2011) state that is reasonable to assume a value of $\frac{1}{3}$ for most applications. When n equals $\frac{1}{3}$, this correlation is referred to as the Chilton-Colburn analogy (1934)

$$h_m = \frac{h_{cv}}{C_{p,a} \cdot (Le)^{1-n}} \quad (5)$$

with $Le = \frac{\alpha}{D_{va}}$

The correlation given by Granryd et al. (2011) is equivalent to take the Lewis number equal to 1 in eq.(5).

The section dedicated to ice rinks in the ASHRAE Handbook (2009) expresses the condensation load with the water vapor molar transfer coefficient, h_n , the enthalpy difference and the difference in mole fractions of water vapor in the air bulk and at the ice surface, which can also be written as partial versus total pressure ratios considering water vapor as an ideal gas.

$$\dot{q}''_{cond} = h_n \frac{(P_v(T_a, \varphi) - P_{sat}(T_{ice}))}{P} \Delta h_{lat+sens} \cdot M_v \quad (6)$$

The paper suggests using the Chilton-Colburn analogy to calculate the coefficient of moist air molar transfer. A numerical value of $0,23 \text{ g.s}^{-1} \cdot \text{m}^2$ is provided, but the unit of the coefficient should be $\text{mol.s}^{-1} \cdot \text{m}^2$ to be dimensionally consistent with the equation provided. In fact, the value of 0,23 can be found dividing the mass transfer coefficient value found with the Chilton-Colburn correlation by the molar mass of air. Similarly, in a paper about ice rink heat loads simulation (Daoud et al. 2008), the condensation load on the ice sheet is calculated as

$$\dot{q}''_{cond} = h_m (\omega_a - \omega_{sat,ice}) \Delta h_{lat+sens} \quad (7)$$

In this study, the coefficient of moist air mass transfer h_m is assumed to be a constant value equal to $10 \text{ kg.h}^{-1} \cdot \text{m}^2$. As written here, the equation (6) cannot be directly deduced from equation (7) because the ratios between vapor mass and dry air mass, ω , are used instead of using ratios with moist air mass, w . A modified version of eq.(7) would then be

$$\dot{q}''_{cond} = h_m (w_a - w_{sat,ice}) \Delta h_{lat+sens} \quad \text{with } w = \frac{\omega}{1+\omega} \quad (8)$$

However, since ω is very close to 0, $w \sim \omega$. Eq.(6), (7) and (8) give identical results as shown in Table 1, which compares the different ways of calculating the heat or mass transfer and the corresponding condensation heat flux. The assumptions made for the calculations are: ice temperature of $-5 \text{ }^\circ\text{C}$; air bulk temperature of $3 \text{ }^\circ\text{C}$ and relative humidity 80%; air is saturated with water at the ice surface; all the thermo-physical properties of moist air and water depend on the temperature; particularly, the following set of equations is used (Egolf et al. 2000)

$$P_{sat} = P_T \cdot \exp\left(\frac{\Delta h}{R_v} \left(\frac{1}{T_T} - \frac{1}{T}\right)\right) ; \quad \varphi = \frac{P_v}{P_{sat}} ; \quad \omega = \frac{M_v}{M_a} \frac{\varphi \cdot P_{sat}}{P - \varphi \cdot P_{sat}} \quad (9)$$

The mass diffusivity of water vapour in air is calculated with the Sherwood and Pigford (1952) correlation as suggested in the ASHRAE Handbook (2009)

$$D = \frac{0,926}{P} \left(\frac{T^{2,5}}{T + 245} \right) \quad (10)$$

The total pressure is assumed to be $101,325 \text{ kPa}$; the convection heat transfer coefficient is calculated with a velocity-dependent relation (ASHRAE 2009)

$$h_{cv} = 3,41 + 3,55 \cdot u \tag{11}$$

Assuming a velocity of $0,5 \text{ m.s}^{-1}$ the convection heat transfer coefficient equals $5,185 \text{ W.m}^{-2}.\text{K}^{-1}$.

Table 1: Comparison between different heat and mass transfer coefficients and the corresponding condensation heat flux

			Condensation heat flux (W.m^{-2})				
			h_m ($\text{kg.s}^{-1}.\text{m}^{-2}$)	Eq.(6)	Eq.(7)	Eq.(8)	Eq.(3)
Mass transfer	Reynolds analogy	$n=0$	$5,44.10^{-3}$	19,4	19,6	19,4	-
	Chilton-Colburn	$n=1/3$	$5,58.10^{-3}$	19,9	20,1	19,9	-
	Daoud et al.	-	$2,78.10^{-3}$	9,89	9,99	9,93	-
Molar transfer	DOE / ASHRAE	h_n ($\text{mol.s}^{-1}.\text{m}^{-2}$)	$2,30.10^{-4}$	23,7	24,0	23,8	-
Heat transfer	Granryd et al.	h_{cond} ($\text{W.m}^{-2}.\text{K}^{-1}$)	2,00	-	-	-	16,03

2.1.2. Sensitivity analysis of the condensation loads correlations

The sensitivity of the Chilton-Colburn and the heat transfer analogies (Granryd et al. 2011) to ice temperature and indoor air velocity is assessed in this part. Figure 4 shows the condensation for both analogies and for different ice temperatures and air velocities. The assumptions are the same as in the previous part.

As it may be noticed, the condensation loads given by the two analogies are not similar. Indeed, since the Granryd et al. (2011) correlation assumes a Lewis number of 1, the resulting condensation heat flows are lower than what is found using the Chilton-Colburn analogy. The air velocity does not seem to have a significant effect but further investigation should be conducted on the convection heat transfer coefficient correlation, in order to check to what extent the correlation is acceptable to use. At $-5 \text{ }^\circ\text{C}$ ice temperature, the relative difference between the two correlations is 17% for an air velocity of $0,5 \text{ m.s}^{-1}$. This represents a difference in the condensation heat load of $34,9 \text{ kW}$ vs. $28,8 \text{ kW}$.

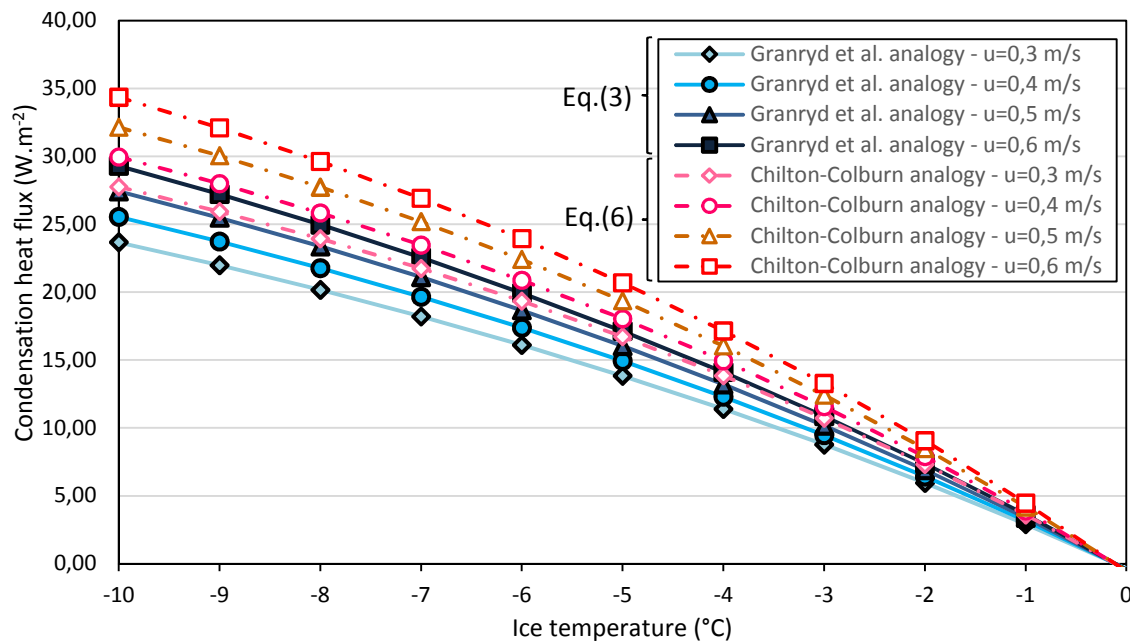


Figure 4: Condensation heat flux sensitivity to ice temperature and air velocity using mass and heat transfer analogies

2.1. Characterization of the dehumidifiers

In both ice rinks, the dehumidifiers are of adsorption type. The nominal characteristics of each dehumidifier given the respective data sheets are given in Table 2 for air intake at 20 °C and 60% of relative humidity.

Table 2: Dehumidifiers characteristics under nominal conditions (20 °C / 60% RH)

Characteristics	Ice rink A	Ice rink B
Air flow (m ³ .h ⁻¹)	6000	4000
Power (kW)	54,2	46,8
Drying capacity (kg _v .h ⁻¹)	39	32

The amount of water removed by the dehumidifier is only given for nominal conditions but it is necessary to know what this amount is for other conditions. In order to do that, the dehumidifiers' respective capacity diagrams are used. It is further assumed that the amount of water removed in a day is proportional to the energy consumption for given indoor air conditions. Knowing the indoor air conditions and the energy consumption from the measurements; it is possible to deduce the amount of water removed from Figure 5.

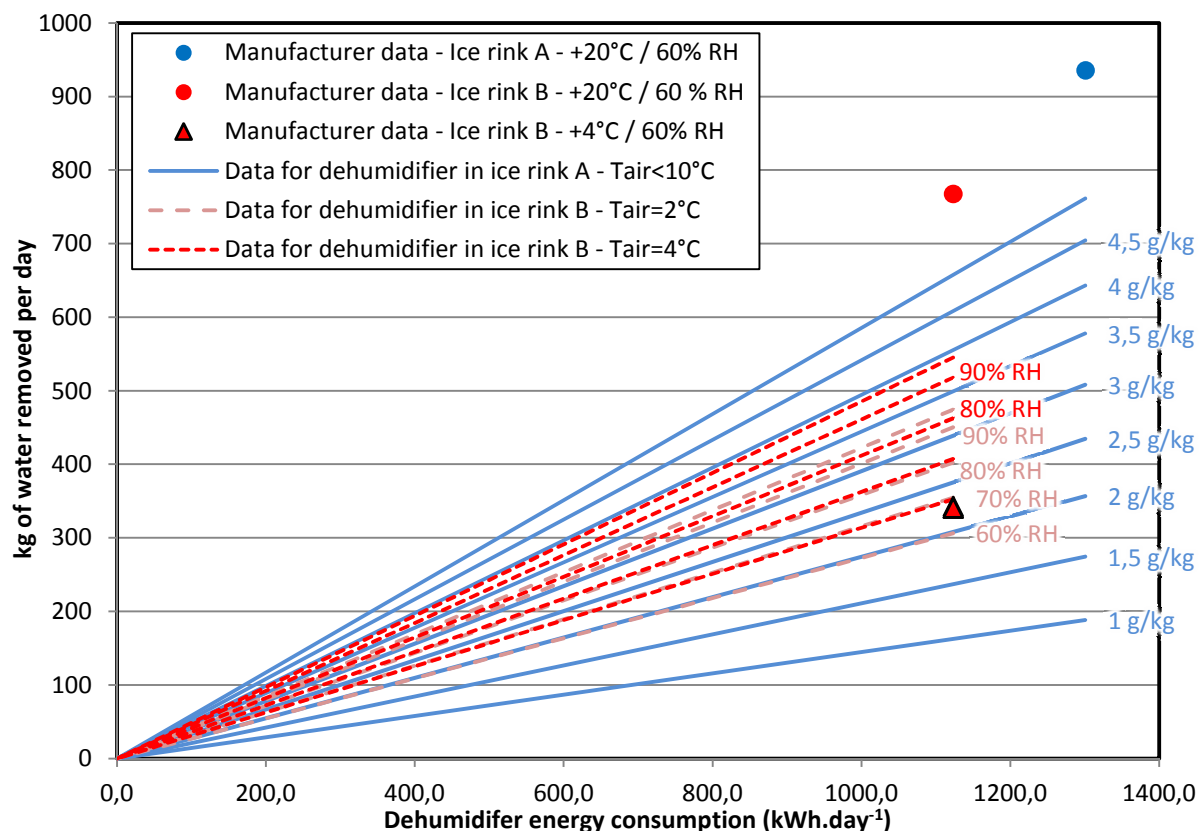


Figure 5: Characteristic curves of the dehumidifiers in ice rinks A and B

The dehumidifiers in these ice rinks are of different brands but similar type, consequently the performance is comparable. The indoor air conditions in the studied ice rinks are quite representative for the bulk of the Swedish ice rinks. Often the temperature will float within a range which in this case is 1-5 °C and the relative humidity is normally controlled to 50 to 70%. Relative humidity is, however, not a good reference for humidity control, especially not when the temperature is floating. The consequence is that the dew point (absolute water content) will vary as well. It is for that reason preferred to control to a set dew point around 0 °C to avoid over drying (at low air temperatures) or too high humidity levels (at higher air temperatures). The typical "cost" of dehumidification with the systems above lies in the range of 2.5 to 3 kWh/kg of removed water.

3. FIELD MEASUREMENTS AND ANALYSIS

3.1. Assumptions and methodology

The mass balance as expressed in eq.(2) is evaluated for a time period of 24 hours. The average values are used. The mass transfer is estimated with the Chilton-Colburn analogy. In a previous study about dehumidification in ice rinks (Bermejo Pereira Rodrigues da Silva 2013), it was shown that the absolute humidity content of indoor air was homogeneously distributed within the ice hall bulk. On the contrary, relative humidity ratios were shown to be varying because of temperature stratification. Therefore, spatial variation of the indoor humidity ratio is not included in the mass balance in this study, meaning that the humidity contents of the air bulk above the ice and at the dehumidifier inlet are the same. The ice area is assumed to be $1800 m^2$.

Field measurements were performed in two indoor ice rinks (*A* and *B*) in the Stockholm area. The measured properties used in this study are: outdoor and indoor temperatures; outdoor and indoor relative humidity; dehumidifier electricity consumption and ice temperature. The value of each property is recorded every minute. Note that data from the Swedish Meteorological and Hydrological Institute (SMHI) were used for the outdoor relative humidity of ice rink *B*. Other needed properties are then deduced from eq.(9).

More details about the instrumentation and data acquisition method can be found in Berglöf (2010), Karampour (2011) and Khalid and Rogstam (2013).

3.2. Dehumidification and condensation shares in water vapour removal

Figure 6 shows the dehumidifier and ice slab share in the removal of the water vapour excess. The bases for the analysis are the measured values for the ice, in- and outdoor temperature as well as the in- and outdoor relative humidity levels. For both ice rinks the indoor temperatures are slightly different with ranges of 3-4,5°C and 1-2°C in ice rink *A* and *B*, respectively. The RH levels are nevertheless similar with values in the range of 67-75%. The ice temperatures vary between -5 and -4°C. The outdoor temperatures are typical for the Stockholm area in autumn and winter with +10°C on the high end side and -10°C on the low end side. Even though the dehumidifiers are different, their average share in the removal of water vapour is equivalent: 26% for ice rink *A* and 23% for ice rink *B*. These percentages correspond also to the potential of condensation load decrease due to the use of a dehumidification system.

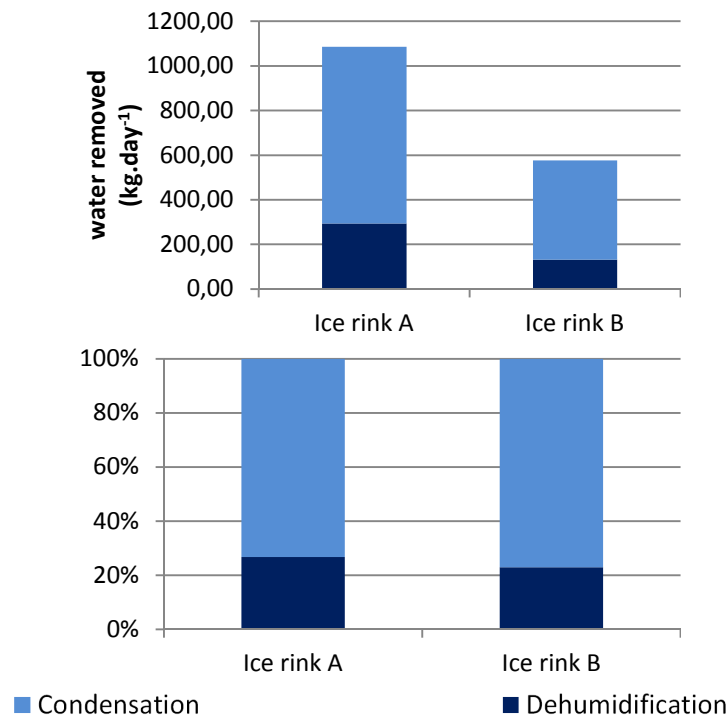


Figure 6: Average quantity of water removed per day by condensation and dehumidification and the corresponding shares

3.3. Dehumidification energy consumption in different ice rinks

The electricity consumption of four dehumidifiers in four different ice rinks close to the Stockholm area was measured over a full season, from July 2012 to April 2013 as shown in Figure 7.

The ice rink *D* has the highest electricity consumption for the dehumidification system, while the ice rink *A* (2012/2013) has the lowest due to its connection to district heating. There may be several reasons why the ice rink *D* shows such high electricity consumptions: 1) the indoor ice rink is coupled with an outdoor one and a large door allows the ice resurfacing machine to go between them; 2) the ice resurfacing machine does not have a dedicated garage and is cleaned inside the ice rink (dirt accumulates during the drive between the two rinks); 3) the transducer controlling the dehumidifier is located close to this large door and the cleaning area. As it could be expected, the electricity consumption is higher during summer, when the outdoor air is more humid. This also confirms that the infiltration has a significant impact on the dehumidification energy consumption and the overall energy usage of the building.

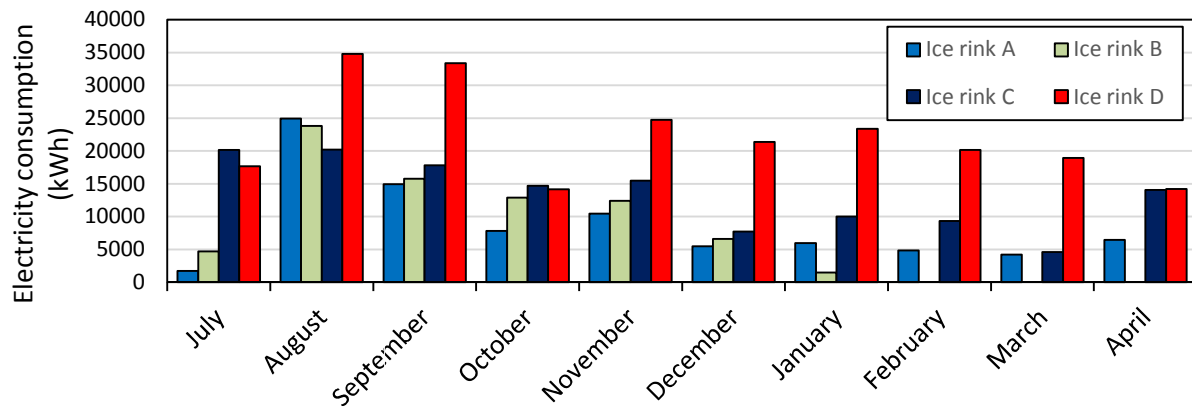


Figure 7: Electricity consumptions of dehumidification in four different ice rinks (2012/2013)

One way to illustrate the dehumidification energy usage and its relation to the outdoor air is to look at so called energy signature. In this case the energy usage is plotted vs. the difference in absolute moisture content in the indoor and outdoor air. This will isolate the dependence on the infiltration load. The infiltration will, however, depend on several factors such as outdoor temperature, wind speed and the usage pattern of the ice rink.

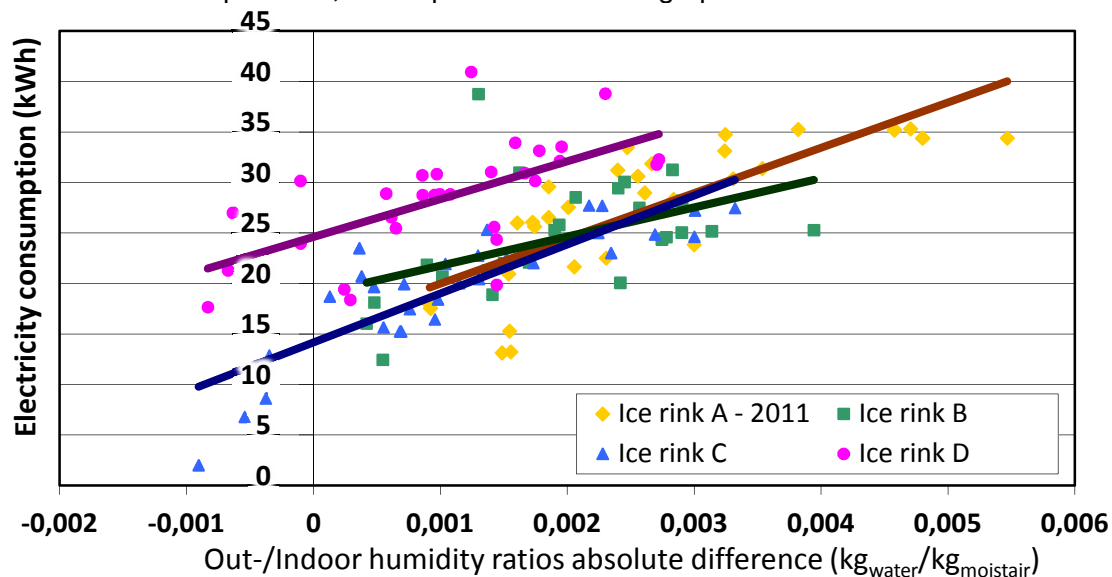


Figure 8: Dehumidification energy consumption VS outdoor/indoor humidity ratios difference

It can be concluded from figure above that the energy signatures are similar or show similar behavior in the studied ice rinks. A general view and analysis of the results reveals that the level of the lines indicates the overall moisture load resulting from internal activities and outside air infiltration. The inclination most likely indicates the buildings sensitivity to infiltration caused by the in- and outdoor temperature difference – often referred to as the stack effect. The intersection with the y-axis should theoretically give the base dehumidification need based on the internal moisture loads, since at this point the air infiltration stop affecting the total moisture load.

The exact physical meaning, interpretation and consequences of the patterns above are not deeply analyzed in this report. It is, however, subject to further analysis in coming research work.

3.4. Dehumidification compared with total energy usage

Several references point out that dehumidification in ice rinks consumes only a share of 4-6% of the total electricity consumption (Rogstam 2010; IIFH 2010; Natural Resources Canada 2003). Since the total and the dehumidifier electricity consumptions were measured, it is possible to evaluate the shares of dehumidification systems in the global energy

consumption of ice rinks. The study for ice rink A and C during the season 2011/2012 and 2012/2013, respectively.

The results shown in Table 3 indicate that the dehumidification share in the global energy consumption is higher than the previously anticipated 6%.

Considering the percentage found in ice rink A and C, it is likely that the dehumidification energy consumption is generally underestimated in ice rinks.

Table 3: Dehumidification shares in the yearly global energy consumption of ice rink A and C

Electricity consumption (MWh)	Ice rink A (09/2011-04/2012)	Ice rink C (04/2012-03/2013)
Dehumidifier	135,3	108,4
Global	885,0*	977,4
Dehumidification share	15,3 %	11,1 %

*the global energy consumption includes district heating

3.5. Energy saving potential – case study ice rink A

Ice rink A installed a pre-heating system for the regeneration air between the seasons 11/12 and 12/13 as shown in Figure 9. The preheater uses district heating to preheat the regeneration air before the regeneration process. Thus, it is possible to evaluate the energy saving potential associated with the use of a pre-heating system.



Figure 9: The preheater installed on the existing dehumidifier in ice rink A.

In the Figure 10 below the difference in electricity usage between the seasons 2011/2012 and 2012/2013 can be seen. In this case the electricity is replaced by district heating so no real energy saving is achieved but the owner sees a cost saving due to lower cost of energy. The heat could be supplied from by a heat recovery system as well, but in this case district heating water of 65°C was used.

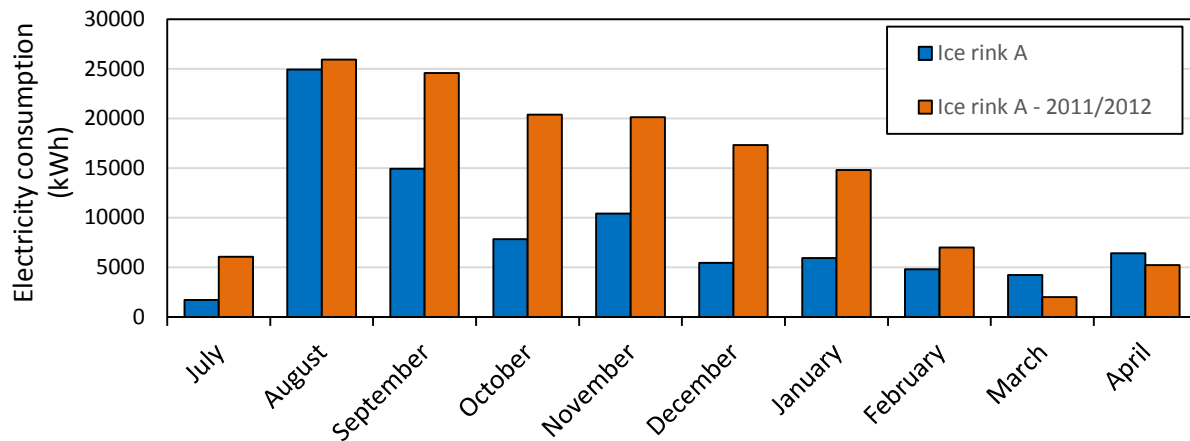


Figure 10: The reduction in electricity consumption 11/12 vs. 12/13 when using a preheater.

Assuming comparable weather conditions during the seasons 2011/2012 and 2012/2013, the energy saving induced by the pre-heating system in ice rink A could be calculated as the difference between the energy consumptions during the two seasons.

The resulting energy saving is of *56,6 MWh* per year in terms of the electricity consumption. If the energy saving potential is defined as the ratio between energy saving and energy consumption, the corresponding energy saving potential is *39,5%*.

4. CONCLUSION

Several correlations were compared for the mass transfer and condensation loads on the ice surface. All the correlations considered in this study are based on a similar heat and mass transfer analogy (Chilton and Colburn 1934) but a relative difference as high as *17%* was found under operational conditions, highlighting the importance of the correlation choice and accuracy. Three elements of the mass transfer mechanism are investigated: the mass transfer on the ice surface; the infiltration rate; and the dehumidifier energy consumption. A moisture balance model including infiltrations, moisture sources, condensation and dehumidification is proposed.

It was found that about *75%* of the moisture load in the ice rink deposits on the ice and the remaining *25%* is removed by the dehumidifier. According to the proposed mass balance model the water removed by the dehumidifier and, thus, its energy consumption increase with increasing outdoor/indoor humidity ratios difference. The energy signature results tend to confirm this statement. However, in order to have a better interpretation, validate or invalidate the proposed mass transfer model, the analysis needs to be further developed by including altogether dehumidifier energy consumptions, condensation loads, infiltration rates and moisture source rates. Particularly, wind speed measurements should be added to list of measurements currently performed.

The energy usage related to dehumidification of an ice rink is concluded to often be in the range *10* to *15%* of the total energy usage, although lower values are found as well. The annual energy consumption may reach, or sometimes even exceed *150 MWh*.

Possible saving measures are the use of recovered heat from the refrigeration system and/or heat pumps. Results from a case study show an electrical energy saving potential of *40%* using other sources than electricity to heat the regeneration air. In the studied cases the dehumidification can reduce the condensation loads of about *23-26%* and it limit the number of necessary ice resurfacings.

Nomenclature				
A_{ice}	ice area	m^2	<i>Subscripts</i>	
C_p	specific heat capacity	$J.kg^{-1}.K^{-1}$	a	air /moist air
D	mass diffusivity	$m^2.s^{-1}$	$cond$	condensation
ELA	effective leakage area	m^2	cv	convection
g	acceleration of gravity (9,81)	$m.s^{-2}$	dh	dehumidifier
h	heat or mass transfer coefficient	$W.m^{-2}.K^{-1}$ or $kg.s^{-1}.m^{-2}$	exf	exfiltration
H	height of the building	m	i	indoor
Le	Lewis number	-	ice	ice surface
M	molar mass	$kg.mol^{-1}$	inf	infiltration
\dot{m}	mass flow	$kg.s^{-1}$	$lat + sens$	latent and sensible heat
n	exponent for heat and mass transfer analogy	-	m	mass / mass transfer
P	pressure	Pa	n	molar / molar transfer
Q	volumetric infiltration rate	$m^3.s^{-1}$	o	out / outdoor
\dot{q}	heat flux	$W.m^{-2}$	reg	regression
\dot{S}	moisture source / generation rate	$kg.s^{-1}$	s	stack effect
T	temperature	K	sat	saturated state
u	velocity	$m.s^{-1}$	T	triple point
w	absolute humidity ratios (per kg of <u>moist</u> air)	$kg.kg_{moistair}^{-1}$	v	water / vapor
			ven	ventilation
			w	wind
<i>Greek letters</i>			<i>Superscripts</i>	
α	thermal diffusivity	$m^2.s^{-1}$	"	flux
Δh	enthalpy difference	$J.kg^{-1}$	-	average
φ	relative humidity	-		
ω	humidity ratios (per kg of <u>dry</u> air)	$kg.kg_{dryair}^{-1}$		

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