SECONDARY FLUID IMPACT ON ICE RINK REFRIGERATION SYSTEM PERFORMANCE

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ABSTRACT

Sweden has 352 ice rinks in operation which annually use approximately 1000MWh. A refrigeration system usually accounts for about 43% of the total energy consumption and can present a significant energy saving potential. More than 97% of the Swedish ice rinks use indirect refrigeration system and thermo-physical properties of secondary fluid have a direct impact on the heat transfer and pressure drop. A theoretical model and two case studies focusing on the importance of the secondary fluid choice were investigated. The results showed that potassium formate had the best heat transfer properties while ammonia lead to the lowest pressure drops and pumping power. Propylene glycol showed the worst performance in both cases. Ammonia and potassium formate showed respectively 5% and 3% higher COP than calcium chloride for typical heat loads of 150kW. When controlling the pump over a temperature difference (ΔT), the existence of the optimum pump control or optimum flow was highlighted. For typical cooling capacity of 150kW optimum pump control temperature difference ΔT was around 2,5K for calcium chloride and around 2K for ammonia. Järfälla case study showed a potential energy saving of 12% for the refrigeration system when increasing the freezing point of the secondary fluid. An energy saving of around 10,8 MWh/yr per 1K increase of the secondary fluid freezing point was found.

Key Words: ice rink, secondary fluid, refrigeration system, heat transfer, pressure drop

1 INTRODUCTION

In 2010, around 6700 indoor ice rinks were recorded all over the world. Countries having most ice rinks are Canada and United States of America with 2703 and 2500 facilities, respectively (IIHF 2010). The energy usage in Swedish ice rinks varies a lot and depends on many factors like length of season, number of activity hours and building characteristics (Khalid and Rogstam 2013). Statistical studies show an average yearly consumption of 1000 MWh/yr for the ice rink in Sweden (Rogstam 2010). The inefficient ice rinks may use even up to 2000 MWh/yr whereas the most efficient ones may use as low as 700 MWh/yr (Karampour

2011). Meanwhile, the average energy consumption of ice rinks in Quebec province in Canada reaches 1500 MWh/yr with highest energy consumptions of 2400 MWh/yr (Nichols 2009). Studies show that refrigeration accounts for the largest energy share of 43% of total energy consumption (Rogstam 2010). There are 352 ice rinks currently in operation in Sweden and their number is still growing. More than 97% of Swedish ice rinks refrigeration systems are designed as indirect or partly-indirect systems on the evaporator side (Makhnatch 2011). The cooling capacity usually reaches 300 to 350kW and is directly depended on the heat loads and gains. Thus, reducing the cooling demand is an important step while trying to decrease the overall energy consumption of ice rink. In some cases, one refrigeration system is used to chill two ice rinks simultaneously, e.g.: one indoor and one outdoor. In this case, the total cooling capacity may be even higher. While improving the ice rink refrigeration system design, it became important to find a common feature applicable to all ice rinks which would help in decreasing the energy consumption. Therefore, thermo-physical properties of secondary fluids have become significant.

In Sweden, the majority of ice rinks use calcium chloride based secondary fluid (Makhnatch 2011). (ASHRAE 2010) states ethylene glycol, propylene glycol, methyl alcohol and ethyl alcohol besides calcium chloride as the potential secondary fluids used in the ice rinks. (Caliskan and Hepbasli 2010) and (Stegmann 2005) have considered ammonia as a potential secondary fluid for the ice rink application. Ethyl alcohol was mentioned as a long-time used secondary fluid by (Wang et al. 2010) and may be used in ice rinks. Potassium formate and potassium acetate are not widely used in ice rinks although they could be considered (Hillerns 2001). However, methyl alcohol as secondary fluid is currently banned in Sweden due to its toxicity and was not included in this study. The most common primary refrigerant used in Swedish ice rinks is ammonia (R717), which accounts for about 85% of all facilities. The remaining 15% use R404A, R134a or other HFC refrigerants (Makhnatch 2011). Several compressors in parallel may be used to stagger the cooling demand and improve the refrigeration system performance. Most of the ice rink refrigeration systems in Sweden have two compressors.

2 OBJECTIVES

The aim of this study was to evaluate the energy saving potential in ice rinks with respect to the type of secondary fluid used. Additionally, a comparison between the theoretical and real thermo-physical properties in terms of the heat transfer and pressure drop related to different secondary fluids was assessed by analyzing samples from two ice rinks and comparing results with values for the pure mixtures of calcium chloride at same concentration. Finally, an attempt to define the best control strategy of the pumps was made by defining the secondary fluid temperature difference (ΔT) for different secondary refrigerants.

3 METHODOLOGY

In order to fulfill previously mentioned objectives, a theoretical model has been developed and two case studies have been conducted. The theoretical model allows comparing the different secondary fluids that may be used in the ice rinks. The model has been performed using Microsoft Excel and its programming interface Visual Basic as well as COMSOL Multiphysics simulation software. The two case studies have been conducted using the ClimaCheck performance analyzer which is an “internal method” used to analyze the performance of the ice rink refrigeration system as described by (Khalid and Jörgen 2013). However, the ClimaCheck analyzer did not allow assessing the influence of the secondary fluid used in the refrigeration system. Therefore, samples from the real facilities have been collected and tested in the laboratory in order to link the refrigeration system performance with secondary fluid thermo-physical properties.

3.1 Theoretical Study of Heat Transfer and Pressure Drop

The ice rink design was fixed throughout the study and s teady-state conditions were assumed. The study was conducted in the way that one main input variable (secondary fluid average temperature, ice temperature, temperature difference / pump control) in a given range was chosen. Fixing the ice temperature or the secondary fluid average temperature as the main input parameter may be considered as the control strategy.

3.1.1 Ice rink design

In case of the ice rink refrigeration system two components were of biggest interest: evaporator and the ice rink floor. In this study, the heat gains from the ground were neglected due to the presence of insulation below the ice rink. Numerical simulations had shown that the heat gains were negligible for the assumed insulation thickness in this study in comparison to the heat loads. The sum of ice floor heat loads, heat gains from the pumps, the headers and the distribution pipes represented the total cooling capacity (Mazzotti 2014). According to (Karampour 2011), the heat loads and heat gains shares were taken as 90% and 10%, respectively. The distribution pipes were assumed to have a length of 20m and 150mm inner diameter and were made out of steel. The ice rink was assumed to be used for the ice hockey purpose and it is considered 60m long and 30m wide. From the supplying header to the return one, the cooling pipes had a U-shape. The headers were located on one of the shorter sides of the ice rink so that the U-pipes were placed along the length. The spacing between two successive parallel straight pipes was 10cm. The U-shaped pipes were embedded into a concrete layer of 15,5cm. Additionally, an ice layer of 3cm and an insulation layer of 15cm was assumed. The pipe outer diameter was taken as 25 mm and its wall thickness as 2,3mm. The back-and-forth length of a U-pipe is approximately 120m and the total number of U-pipes was 150 (Mazzotti 2014).

3.1.2 Refrigeration system and primary refrigerant

Since this study focuses on the secondary fluids less detailed analysis was performed for the cooling machine. The compressor isentropic efficiency was assumed as 0,65. Since evaporator was considered to be the flooded type, rather small superheating of 1K was chosen. 5K subcooling and a constant condensation temperature of +20°C were assumed. Since the evaporation temperature depended on the type of secondary fluid used, its average temperature and the heat loads needed to be known in order to perform calculations. Thus, for each required data a respective function of input data value versus the temperature was created based on EES Property Calculator Software and (Granryd 2011) which used data from NIST software REFPROP 6.01.

3.1.3 Assumptions on the evaporator side

In order to be able to compare all secondary fluids on the same basis, the plate heat exchanger (PHE) design was assumed to be the same for all cases. The evaporator was considered to be a flooded type evaporator and its parameters were chosen based on the literature study and simulations results from Alfa Laval calculation software. The evaporator was assumed to be a single-pass counter-flow heat exchanger which is the most common type for heat pump applications (Claesson 2004). In this study the two boundary plates were not included in the total number of plates. The plates were considered to be made from titanium due to corrosive character of calcium chloride used as the secondary fluid (Ignatowicz 2008). The heat transfer area for one plate was calculated as (Huang 2010). In this study no fouling was considered on either sides of the PHE and the evaporator was assumed to be insulated. Correlation by (Martin 1996) was used for Nusselt number in PHE calculations since it showed a good agreement between the theoretical and experimental results for commercial plates. Studies stated that this correlation could be used for ($400 < Re < 10000$) since it was originally varied in this range by (Claesson 2004) and (Huang 2010).

For (Re < 400), the correlation by (Muley and Manglik 1999) was employed. The correlation developed by (Ayub 2003) is suitable for any chevron angle for commercial plates using ammonia for determining the boiling heat transfer and was used in this study. (Huang 2010) claimed that it was one of the most promising correlations although it was not dimensionally consistent. To calculate the UA value, one-dimensional heat transfer process was considered within the PHE. For pressure drop calculations Moody friction factor equation provided by (Martin 1996) was used since it was already accounting for the ports head losses (Palm and Claesson 2006).

3.1.4 Characterization of the heat transfer

The total heat transfer process occurring in the ice rink floor was based on convection and conduction mechanisms. The convection heat transfer coefficient strongly depends on the flow regime; hence two different equations were used. For (Re > 2300), the widely used (Gnielinski 1976) correlation was employed to calculate the Nusselt number. (Seghouani and Galanis 2009) recommended using Gnielinski correlation to calculate Nusselt number for the single-phase secondary fluids and turbulent flow case. However, in case of laminar flow inside the ice floor pipes, the effect of the entry region needed to be considered due to the high Prandtl numbers implied by high viscosity values. In order to do so, Hausen's equation was used. To assess the equivalent heat transfer resistance from the pipe wall to the ice surface, a two-dimensional numerical model has been developed with COMSOL Multiphysics software.

3.1.5 Characterization of the pressure drop

The pressure drop in the distribution pipes was calculated assuming 20m as the pipe total length and 150mm as its diameter. Only major losses were considered since the system design differs from one case to another. In case of the U-pipes total pressure drop, the minor losses created by U-shaped bends were accounted. Additionally, the pressure drop in both supply and return headers had to be taken into account. The inner pipe diameter of headers was assumed to be 0,2m (Mazzotti 2014). The choice of correlations used in this study was based on the literature study and (Haglund Stignor et al. 2007), which focus on the heat transfer and pressure drop in cooling coils using secondary fluids.

3.1.6 Secondary fluids assumptions

After analyzing secondary fluid samples from the real installations, it was discovered that average secondary fluid's freezing point was around -30°C, thus two different freezing points of -30°C and -20°C were investigated in this study. Lower concentrations could lead to better performance still preventing the secondary fluid from freezing if lower operating evaporation temperatures were maintained. The decrease in concentration was one of the energy saving options to consider. Both theoretical pure secondary fluids and commercial secondary fluids were included in this study. In case of the pure secondary fluids, the data were retrieved from (Melinder 2010). Thermo-physical properties (specific heat capacity, dynamic viscosity, density and thermal conductivity) of samples from Järfälla and Nacka ice rinks and reference pure solutions were measured in the laboratory. The freezing points of those secondary fluids were also experimentally measured to define the concentration.

4. RESULTS

4.1 Theoretical Comparison

Figure1 and Figure 2, show the convection heat transfer coefficients in one U-pipe in ice floor and in one plate of PHE for: calcium chloride ($CaCl_2$); propylene glycol (PG); ethylene glycol (EG); ethyl alcohol (EA); ammonia (NH_3); potassium acetate (K-acetate) and potassium formate (K-formate) for a given concentration corresponding to the freezing point of -30°C.

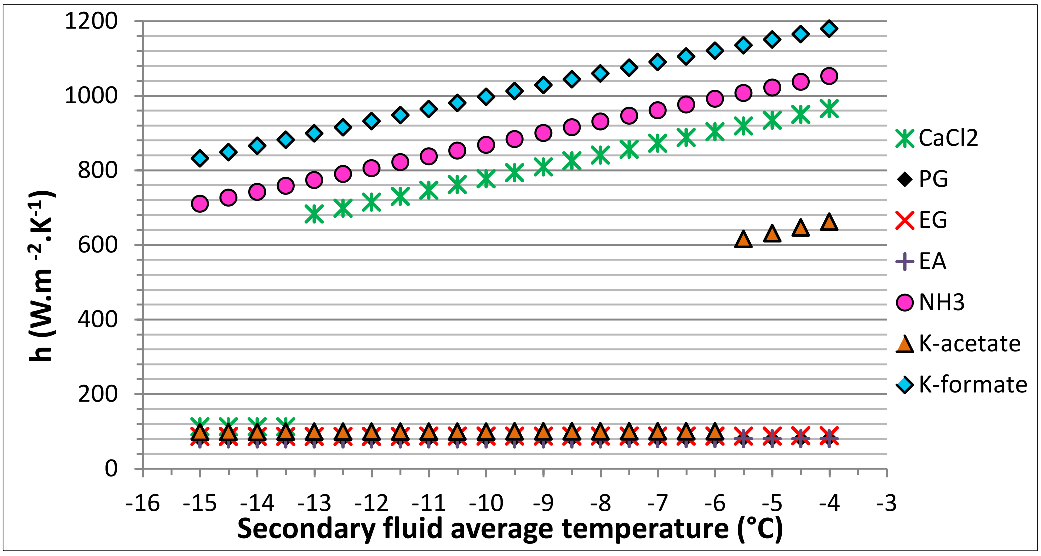

Figure 1: Convection heat transfer coefficient in one U-pipe (Q = 200 kW; ΔT = 2 K)

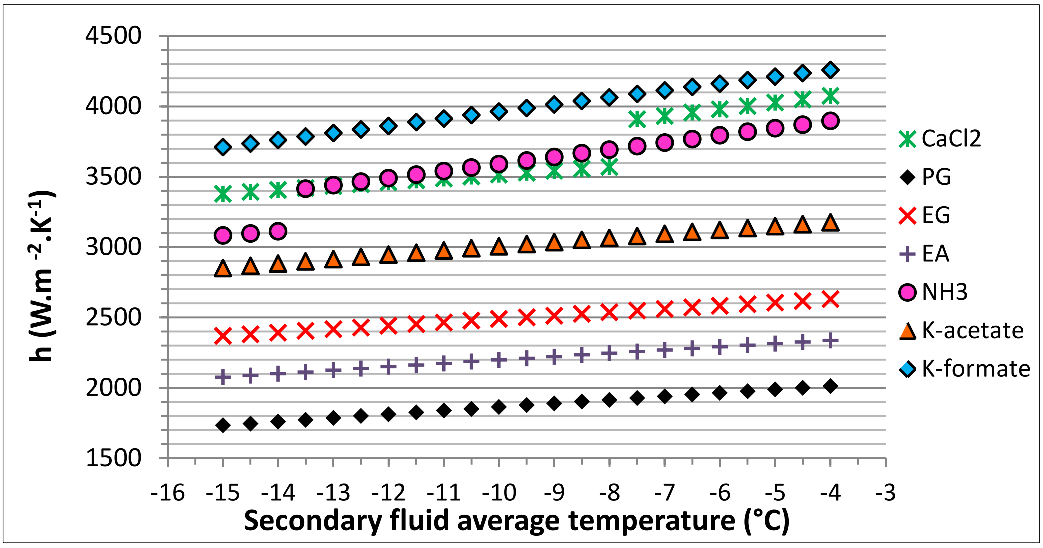

Figure 2: Convection heat transfer coefficient in one PHE plate (Q = 200kW; ΔT = 2 K)

It is possible to see that K-formate gave the best heat transfer features while PG had the poorest heat transfer properties. $CaCl_2$ and NH_3 had good heat transfer properties. The convection heat transfer coefficient decreased with decreasing temperature. Moreover, lower secondary fluid temperatures led to lower evaporation temperature which was reducing the performance. The chosen cooling capacity was close to the typical operational cooling capacity for the ice rink. In Figure 1, a discontinuity may be observed in the $CaCl_2$ and K-acetate cases due to the change in the flow regime to laminar. Similarly in Figure 2, a discontinuity may be observed for $CaCl_2$ and NH_3 due to the change in correlation used from the laminar to turbulent flow. For the same given conditions, PG, EG and EA were in the laminar flow regime whatever the operating temperature was. $CaCl_2$ case showed a flow regime shift at -13°C, a temperature that will only be reached for the high heat loads. K-acetate had a change in flow regime at -5,5°C which was close to normal operating temperatures. NH_3 and K-formate remained also in the turbulent regime. For higher heat loads and the same pump control strategy, the flow would be higher leading to the turbulent flow. Decreasing the temperature difference over which the pump is controlled allows having higher flow that may lead to turbulent flow and higher evaporation temperature. On the other hand, decreasing the temperature difference leads to higher pumping power.

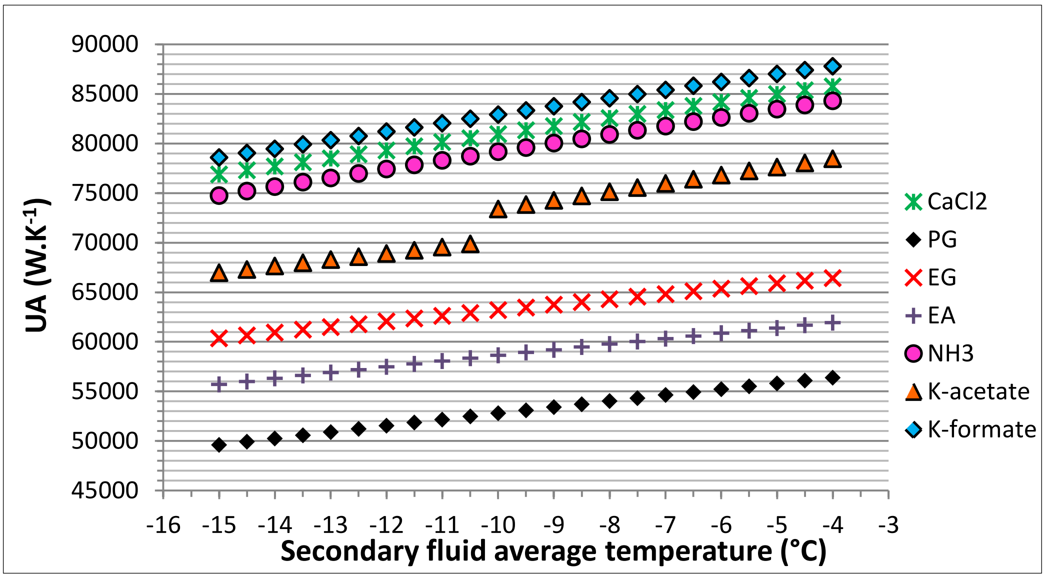

Figure 3: Evaporator UA values versus T_{av} (Q = 300 kW; ΔT= 2 K)

Figure 3, shows the heat transfer coefficient UA, for the evaporator versus the secondary fluid average temperature. Lower UA values led to higher temperature difference for a given cooling capacity, and lower refrigeration system performance. In reality, PHE has been designed for a given UA value or a given temperature difference. Therefore, PHE with same design but with a larger heat transfer area for PG for example would be used.

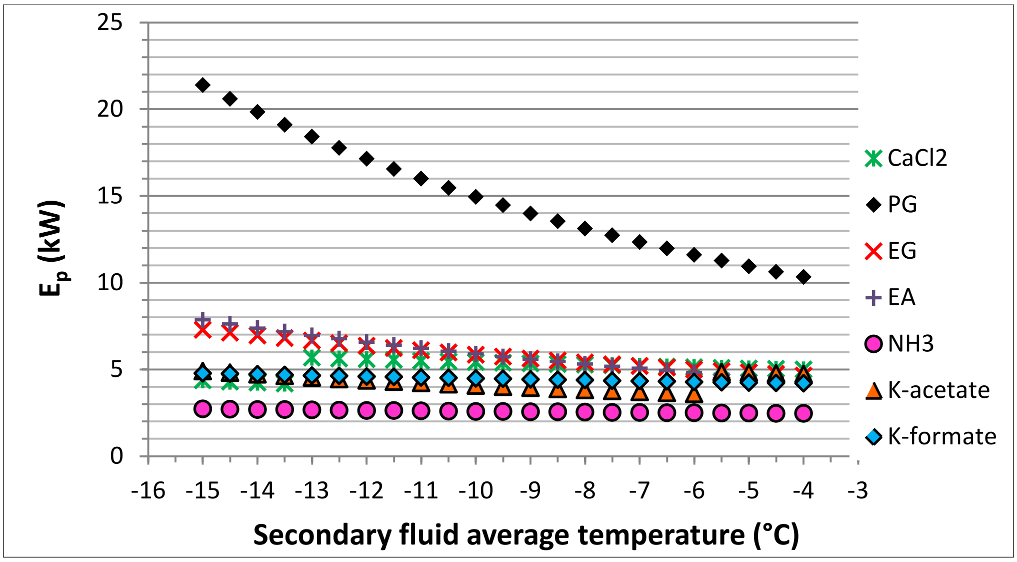

Figure 4: Pumping power versus T_{av} (Q = 200 kW; ΔT = 2 K)

Figure 4, shows the pumping power for all secondary fluids versus their average operational temperature. PG was having the largest pumping power due to high viscosity at low temperatures. Additionally, NH_3 led to low pressure drop and the lowest pumping power. Except for PG, the pumping power was varying between 2 and 7 kW which was rather low due to the variable speed pump assumption. Once again, the discontinuities that may be observed due to the change in flow regime. The higher the temperature difference over which the pump was controlled, the lower the pumping power but lower evaporation temperature could be obtained.

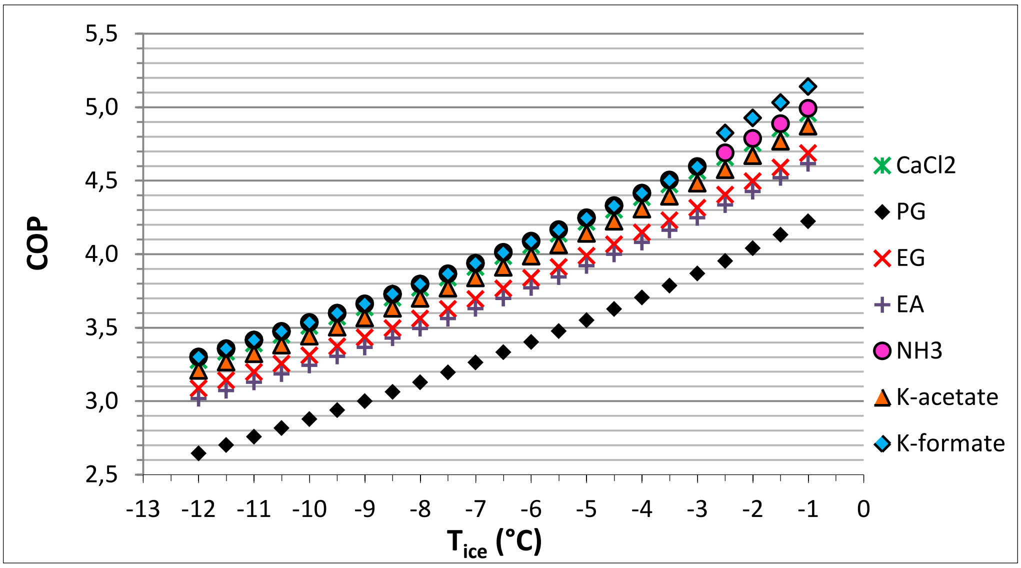

Figure 5: COP versus T_{ice} (Q = 150 kW; ΔT = 2,5 K)

Figure 5, shows the COP variation with the ice temperature for each secondary fluid and operational cooling capacity of 150kW. $CaCl_2$, K-formate and NH_3 were the best in terms of COP. K-acetate also gave rather high COP values while PG, EG and EA showed lower COP values. PG was having the lowest COP. The performance of other secondary fluids was rather close in this case since the pump control temperature difference (ΔT) was high and the cooling capacity low, leading to low volumetric flow and the laminar flow regime.

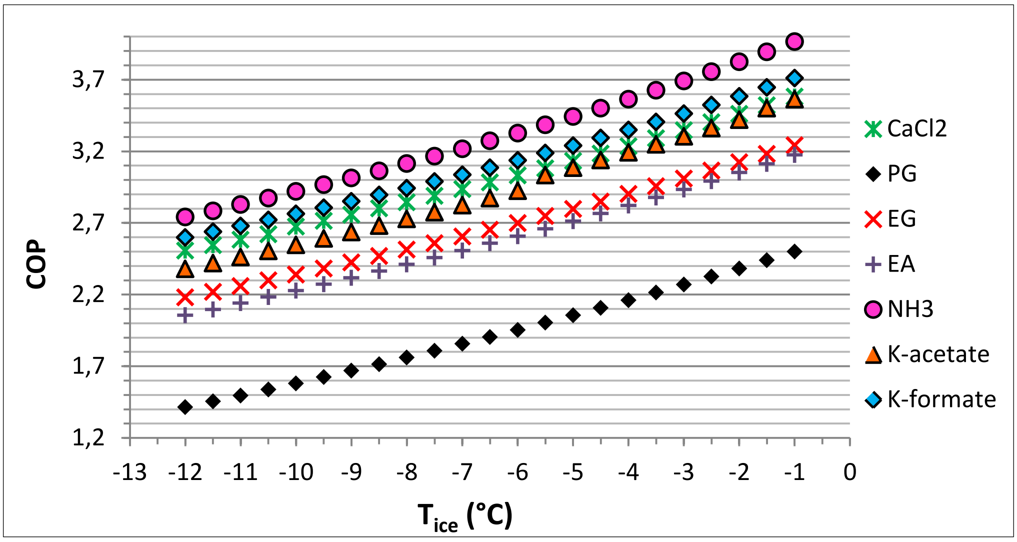

Figure 6: COP versus T_{ice} (Q = 300 kW; ΔT = 2 K)

Differences can be observed between the various secondary fluids in Figure 6 for a lower pump control temperature difference and higher heat loads. Again NH_3; K-formate and $CaCl_2$ are having the highest COP since the dynamic viscosity is an important factor. Additionally, high cooling capacities lead to lower evaporation temperature and hence to lower COP. For each secondary fluid, an optimum pump control or secondary fluid temperature difference (ΔT), exists for a given heat load. Indeed, low ΔT lead to high pumping power but also lower compressor power. On the contrary, high ΔT implies low pumping power but higher compressor power. A lower ΔT is in general more desired since it gives more uniform temperatures over the ice surface. Figure 7 and 8, show the COP variations due to the pump control change for the heating loads of 150 and 400kW, respectively. Once again the discontinuities were due to the flow regime change. NH_3 and K-formate showed the highest COP for the lowest optimum pump control temperature difference, ΔT.

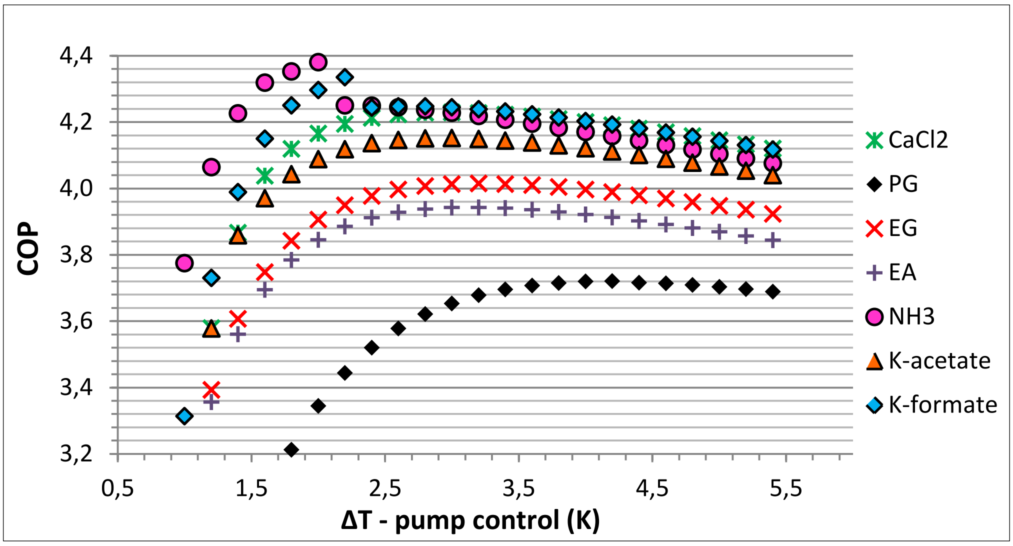

Figure 7: COP versus ΔT (Q = 150 kW; T_{ice} = -5 °C)

The largest optimum pump control temperature difference ΔT was obtained for PG. Moreover, it can be noticed that the optimum pump control ΔT increases with increasing cooling capacity, regardless of the secondary fluid type. A better way of controlling the pump would be to define the optimum temperature difference for the different cooling capacities with a temperature difference glide for instance. It is not always possible, thus a constant temperature difference is often chosen as the control parameter. For instance, ice rinks in North America have often control temperature difference closer to 1K although it does not lead to the best efficiency according to this study. In Sweden, the pump control temperature difference ΔT is often close to 2K. One of the advantages of the constant temperature method is better uniformity of the surface temperature of ice.

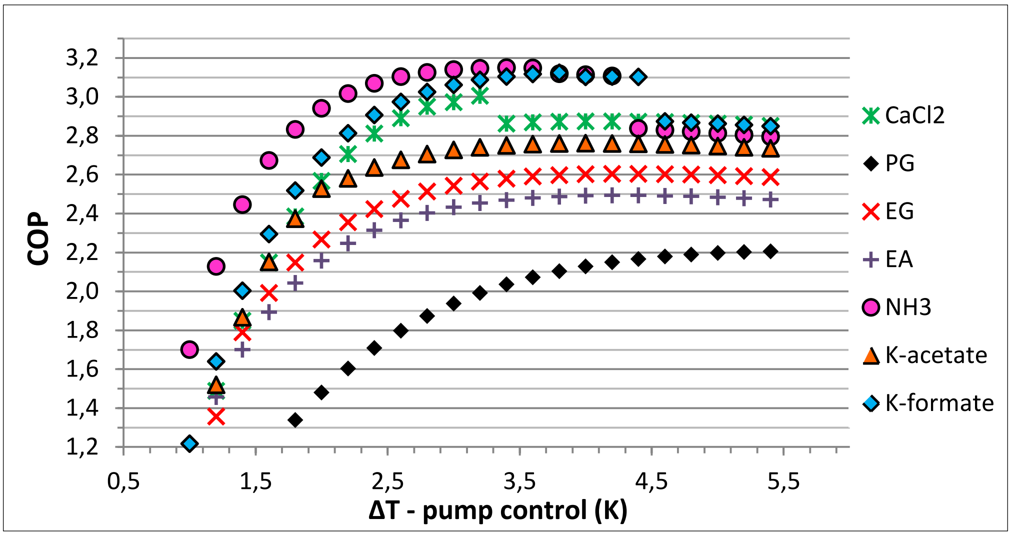

Figure 8: COP versus ΔT (Q = 400 kW; F_p = -30 °C; T_{ice} = -5 °C)

Since $CaCl_2$ is used in more than 97% of the Swedish ice rinks, it was interesting to compare the performance of other secondary fluids. As a result NH_3 and K-formate showed better COP than $CaCl_2$. All other secondary fluids had lower COP than $CaCl_2$. For the ice temperature of -5°C NH_3 led to 5% higher COP while K-formate led to 3% higher COP.

Based on measurements performed in the real facilities using calcium chloride the energy consumption associated with the refrigeration system is around 300 MWh/yr. Considering that 150kW is the average cooling capacity over a year and that the ice temperature of -5°C is maintained using NH_3 or K-formate instead of calcium chloride would lead to energy savings of 15 MWh/yr or 10,5 MWh/yr, respectively.

The freezing point, or concentration, directly influences the thermo-physical properties and thus, the heat transfer and pressure drop. The recommended concentration in ice rinks should be 24wt-% for $CaCl_2$ (F_p= -26,5°C). However, measurements for several Swedish ice rinks showed freezing points closer to -30°C rather than recommended -26,5°C. Additionally, the freezing point could be even higher since the normal operating temperatures rarely exceed -10°C in ice rinks. (Melinder 2007) recommends having a c oncentration giving a freezing point 10K lower than the normal operating temperature. In the two ice rinks used as case studies minimum measured temperature of the secondary fluid and ev aporation temperature was -9,1°C and -17°C, respectively. Hence, a freezing point of -20°C instead of -30°C would be sufficient. If using the same secondary fluid with the freezing point of -20°C the highest gains could be seen for the secondary fluids which thermo-physical properties are poor: PG, EG and EA. While $CaCl_2$, K-acetate and K-formate show gains equal or slightly higher than 3% for high cooling capacities and COP gains for NH_3 never exceed 1%. For all secondary fluids COP gain increases with increasing cooling capacity. Considering same yearly consumption for refrigeration system (300 MWh) and average cooling capacity of 150 kW, $CaCl_2$ with the freezing point of -20°C instead of -30°C would lead to 3,9 MWh energy savings. For cooling capacity of 150 kW the gains are: 1,3 % for $CaCl_2$; 10,5 % for PG; 3,6 % for EG; 2,8 % for EA; 0,4 % for NH_3; 1,7 % for K-acetate and 1,4 % for K-formate. For all secondary fluids, the COP gain increases with increasing cooling capacity.

4.2 Case Study - Järfälla and Nacka

Järfälla is a rather old ice rink having a partially indirect refrigeration system with ammonia as the primary refrigerant and having air-cooled condenser. Initially, the refrigeration plant was cooling down an indoor and outdoor ice rink. However, the outdoor ice rink is not in operation anymore. The refrigeration system has three reciprocating compressors and a shell-and-tube heat exchanger working as the evaporator. The pumps work only in full-speed mode and consume around 18kW. 25,54wt-% calcium chloride (F_p= -31°C) is used as the secondary fluid. When using thermo-physical properties of calcium chloride with recommended concentration of 24wt-% (F_p= -26,5°C), the calculations gave an energy consumption of 176 MWh for compressors and a potential energy saving of roughly 46,8 MWh/yr. Considering the freezing point difference, the potential energy saving is 10,8 MWh when increasing the freezing point by 1K (or 26,9 MWh per each 1wt-% concentration). This potential energy saving represents 12% of the total energy associated with the refrigeration (compressors and pumps). These values show that the energy saving potential related to the concentration of secondary fluid may be significant for some of the ice rinks. The energy consumption for pumps is 178 MWh/yr which is rather high due to the fact that they are running full-time at full-speed. The pumping energy consumption accounts for 44% of the total refrigeration consumption.

Nacka refrigeration system is fully-indirect system and heat recovery from the condenser and desuperheater is being done. The refrigeration system is used to cool down two rinks: one indoor and one outdoor. The refrigeration system uses the reciprocating type compressors and ammonia as the primary refrigerant. 22wt-% calcium chloride (F_p= -24°C) is used as the secondary fluid. Since the freezing point is below the recommended -26,5°C energy savings obtained by lowering secondary fluid's concentration were not assessed. Unfortunately, the pumping power was not measured and no i nformation was found on the pump control, except the fact that variable speed pumps were used. Additionally, the sensor measuring the ice temperature was disabled and no i nformation could be r etrieved regarding the ice temperature profile. Nacka refrigeration system is having ON-OFF control strategy for the compressors. Yearly energy consumption for the compressors is 250 MWh. The total pumping power was estimated as 41 MWh/yr which account for 16% of the total refrigeration energy consumption, contrasting with 44% in Järfälla case due to variable speed pumps. It is important to underline that some energy consumptions such as lighting in the locker rooms and domestic electrical devices were not measured. If assuming that the non-measured shares were 10% a total energy consumption of 1060 MWh/yr would be obtained which corresponds to 38% of the total energy consumption.

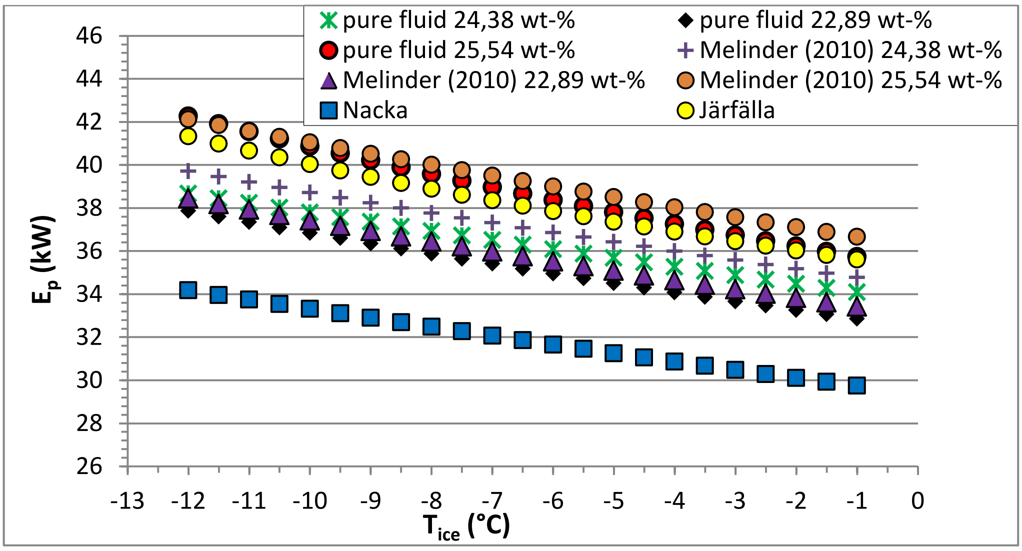

Figure 9: Total pumping power versus T_{ice} (Q_2=400 kW; ΔT = 2K)

Commercial secondary fluids may have different thermo-physical properties than those given for pure secondary fluid mixtures due to various additives added e.g.: corrosion inhibitors that may modify properties. In order to perform the comparison pure mixtures of secondary fluids having the same freezing points as samples from Nacka and J ärfälla were prepared in laboratory and results are presented in Figure 9 and 10.

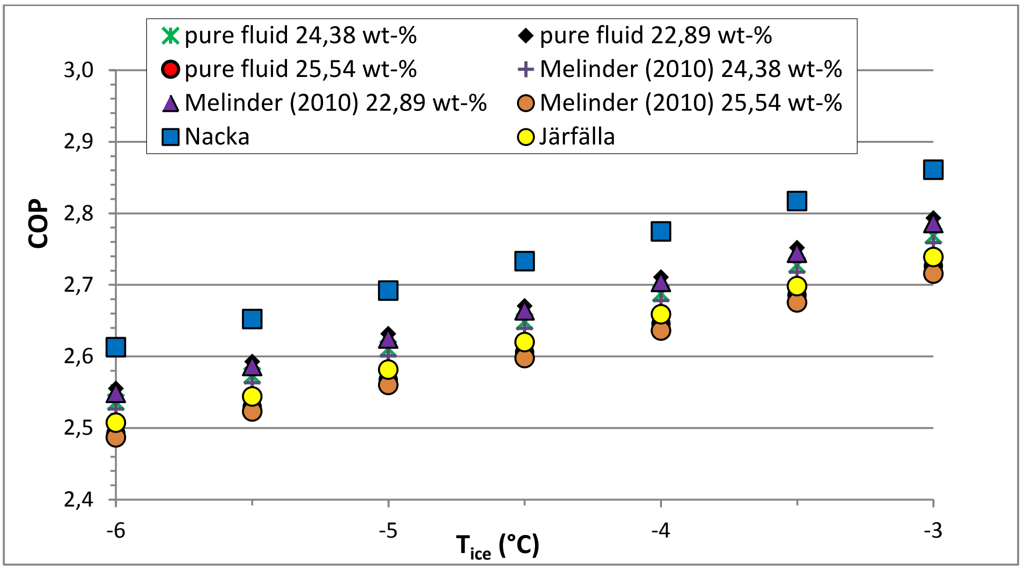

Figure 10: COP for different secondary fluids (Q = 400 kW; ΔT= 2 K)

Figure 9, shows the total pumping power for different samples and reference data from (Melinder 2010). Nacka sample had lower viscosity and density; and higher specific heat capacity (around 4% higher) comparing to pure fluid with same concentration. These results were unexpected since commercial fluids usually have worse thermo-physical properties than pure ones due to the additives (e.g. corrosion inhibitors). As a result Nacka had the lowest pumping power and the highest COP among all samples as seen on Figure 9 and 10. The exact chemical composition of Nacka and Järfälla samples was not investigated.

5. CONCLUSION

It was shown that potassium formate has the best heat transfer properties while ammonia leads to the lowest pressure drops and pumping power. Propylene glycol shows the worst performance in both cases. A relation was made between the secondary fluids efficiency and viscosity. Indeed, the secondary fluids showing the lowest viscosity at low temperatures are the ones leading to the best refrigeration efficiency. Particularly, NH_3 and K-formate show respectively 5% and 3% higher COP than calcium chloride for typical heat loads of 150 kW.
When controlling the pump over a temperature difference ΔT, the existence of the optimum pump control or optimum flow was highlighted. This optimum pump control temperature difference, ΔT depends on the secondary fluid type as well as on the heat loads. For typical heat loads of 150 kW this optimum pump control ΔT is around: 2,8K for $CaCl_2$; 4,2K for PG; 3,2K for EG; 3K for EA; 2K for NH_3; 3K for K-acetate and 2,2K for K-formate. The optimum pump control temperature difference is increasing with increasing cooling capacity independently of the secondary fluid type. The theoretical model was also used to assess how the reduction of the freezing point depressant concentration would affect the system performance. It was found that increasing the freezing point by 10K gives COP improvement of: 1,3% for $CaCl_2$; 10,5% for PG; 3,6% for EG; 2,8% for EA; 0,4% for NH_3; 1,7% for K-acetate and 1,4% for K-formate for cooling capacity of 150 kW.

A comparison between two secondary fluids with different freezing point was performed for the Järfälla case and the results showed a potential energy saving of 12%, corresponding to 46 MWh/yr. An increase of 1K in the freezing point may lead to 10,8 MWh/yr savings. Due to the lack of the ice temperature measurement, it was not possible to conduct the same type of comparison for Nacka. However, yearly energy consumption shares could be measured showing that refrigeration accounts for 38% of the total energy consumption, which is similar to 43% obtained from statistical studies.

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