EVALUATION OF HEAT RECOVERY PERFORMANCE IN A CO2 ICE RINK
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ABSTRACT

Ice rinks have a high coinciding cooling and heating demand which turns them into ideal heat recovery applications. Using carbon dioxide (CO_2) as a refrigerant in a trans-critical refrigeration system allows the ice rink to be self-sufficient in terms of heat. High temperatures can be supplied which in combination with a suitable heating system design can result in very high heat recovery performance. This study is based on the system solution used in the Gimo ice rink, first of its kind in Europe, using a 100% CO_2 trans-critical system, with direct expansion. This system is designed with a temperature cascade on the heat recovery side to fit the CO_2 properties. This implies a low return temperature from the heating system to increase the sub-cooling and thus the system efficiency. To further improve efficiency and the return from the heat recovery system a geothermal storage equips the installation as well. A numerical model has been created using real data from the field which allows to compare actual operating parameters with the desired optimal values. The aim is to find the optimum control of the head pressure and the sub-cooling to reach the maximum overall efficiency at all conditions. The results suggests that the heat recovery coefficient of performance is in the range 3.5-4, which is in line with or exceeds most conventional heat pumps considering the relatively high supply temperature of 60°C. An advantage is that this system solution does not need supplementary heat or heating equipment – one system covers the complete refrigeration and heating function.

Keywords: ice rink, CO_2, heat recovery, geothermal.

1. INTRODUCTION

Ice rink operation requires significant amounts of energy and a typical single sheet ice rink uses about 1000 MWh per year. A well designed modern ice rink should not use more than about 500 MWh per year, consequently the potential energy saving by using best possible technology and design is therefore considerable. Carbon dioxide refrigeration systems suits the ice rink application very well due to the favourable properties for heat recovery. The first 100% CO_2 ice rink in Sweden is in operation since 2014 which makes it very interesting from a demonstration and evaluation point of view. This ice rink is unique, not only because of the CO_2 system, but also due to the integration of all the energy systems. To further utilize the advantages of the CO_2 process a geothermal storage is connected to the refrigeration system which enables energy storage and heat pump operation. To make the heat recovery process as efficient as possible the heating system is designed to fit the characteristics of CO_2. It has been proven that an ice rink can be self-sufficient with heat when properly designed.

2. CONCEPT AND SYSTEMS DESIGN

2.1. Technical introduction to ice rinks

In order to operate an ice rink there are essentially five basic energy systems required, as can be seen in Figure 1; refrigeration, heating, dehumidification, lighting and ventilation. These systems are sometimes referred to as the “big five” because they normally account for more than 90% of the energy used in the ice rink. When operated the systems will interact which can be an advantage when utilised correctly. As illustrated in Figure 1 it is evident that the heat, air and light provided inside the arena room will have an impact on the ice surface heat transfer. In order to keep the ice surface at the desired temperature the refrigeration system extracts the heat absorbed by the ice. All heat absorbed together with the compressor work is eventually rejected on the warm side of the refrigeration system. From an energy conservation perspective it is evident to reclaim the heat from the refrigeration system as Figure 1 suggests. Space heating and hot water are obvious heat demands, however, today modern dehumidifiers may be operated with low grade heat for reactivation which account for a very interesting saving potential. It has been shown that when an ice rink has a well design heat reclaim system and uses heat to all available demands, it may be self-sufficient with heat, Rogstam and Bolteau (2015).

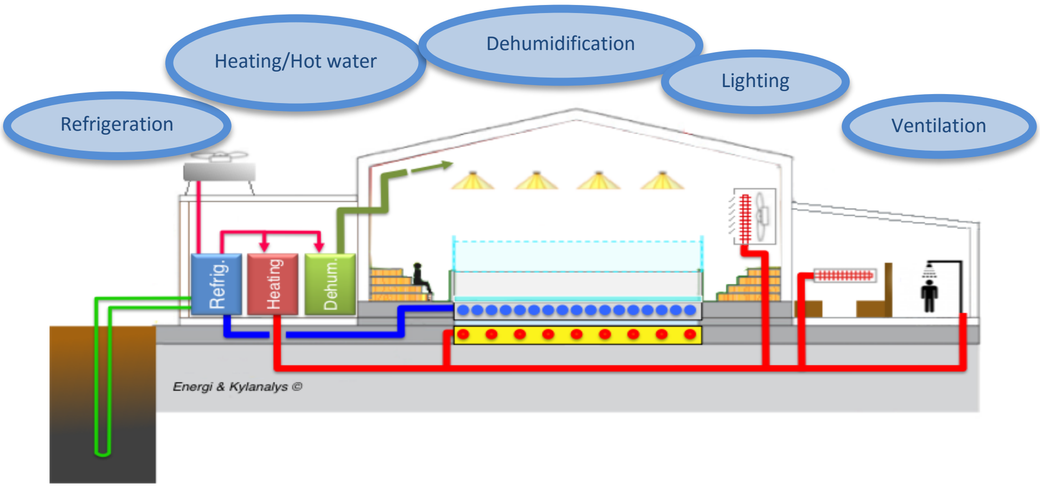

Figure 1 The general ice rink energy system layout of the Gimo ice rink.

2.2. Gimo ice rink – the energy hub

To realise the general requirements outlined above a system concept was developed according to Figure 2 which illustrates the refrigeration/heat pump system. In the centre of the "energy hub" there is a trans-critical CO2 system connected to a receiver tank which circulates liquid CO2 to the rink floor. By evaporating the liquid in the rink floor heat is absorbed and transported to the refrigeration system. On the top right the heat recovery function is illustrated with its "water fall" principle where the heat demands are lined up in order of "temperature need". The latter enables a priority order of the heat supplied and provides a desired high temperature difference in the heat recovery heat exchanger. If there, after the heat recovery, still is a heat surplus it may be rejected to the ambient or to the geothermal storage. In the geothermal storage the heat may be stored short or long term. Lastly this concept may retrieve the heat from the storage via a second evaporator which allows the system to operate as a heat pump, which can be done complementary to the ice rink operation or solely outside ice rink season.

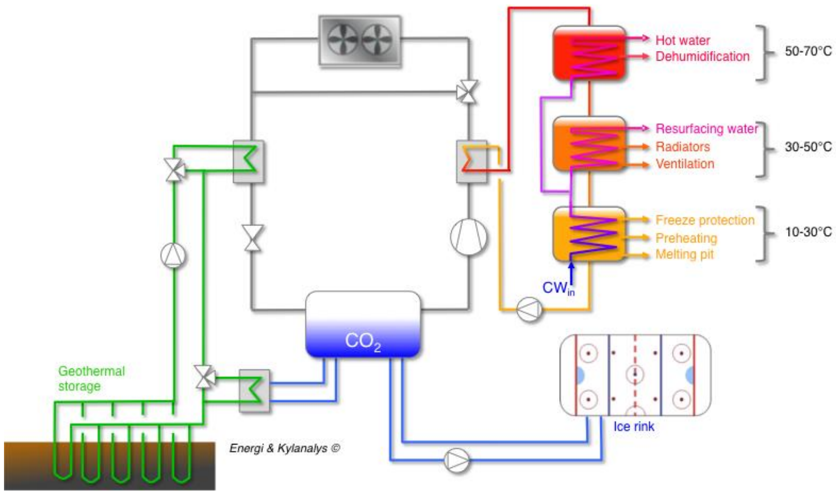

Figure 2 Overview of the refrigeration system in GIMO ice rink.

2.3. Refrigeration system

The specifics of the CO_2 refrigeration system loop may be described starting with the compressors marked with A in Figure 3 below. These evacuate the evaporated refrigerant from the receiver tank and circulate the refrigerant in the loop. The working pressure on the high pressure side is normally above the critical pressure of CO_2 making the system trans-critical. After the compressor the refrigerant passes through a heat recovery heat exchanger marked as B. If required the refrigerant is then cooled further using a gas cooler marked as C and/or a geothermal heat storage marked as D. After expanding the refrigerant back into the tank it may be passed through the ice rink tube system which is the primary evaporator, marked as E. There is a secondary evaporator connected to the geothermal storage where additional heat can be provided to the system when the ice rink cooling demand is low. The refrigerant is stored and separated in the accumulator tank marked as F.

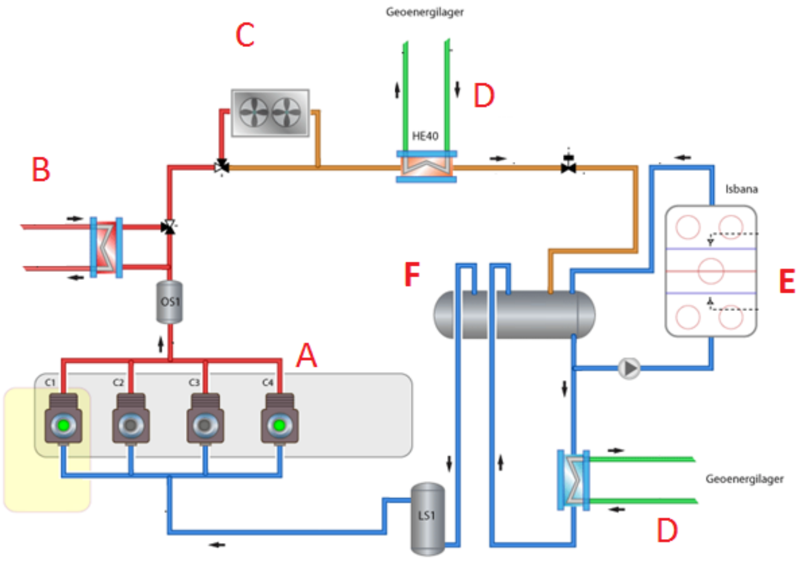

Figure 3 Overview of the refrigeration system in the GIMO ice rink.

2.4. Heating and heat recovery system

The heating system is connected to the refrigeration system through a heat exchanger marked as B in Figure 4Figure . The primary heating circuit with a minimum temperature of 60°C is passed through high temperature accumulator tanks A where domestic hot water, DHW, is heated to at least 55°C. After the DHW the heat required for the dehumidifier is supplied in a secondary circuit via a heat-exchanger marked as C. Typically this supply temperature is between 55 and 58°C. Next function is the radiator system D which has a maximum design temperature level of 55°C at lowest outdoor temperature. Typically at low ambient temperatures there is little or no dehumidification need so there is no problem supplying the required temperature level.

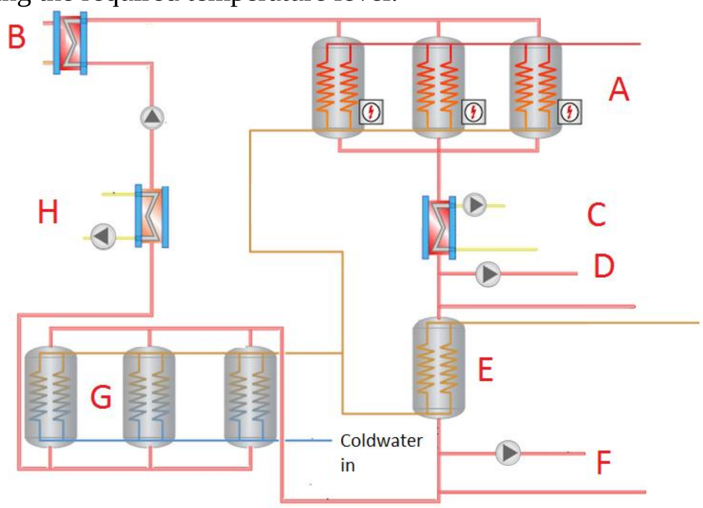

Figure 4 Schematical view of the heat recovery system.

In the tank marked as E the resurfacing water is heated with a target temperature of about 45°C. It is however less critical so therefore it has a lower priority. The dominating heating demand, but on a lower temperature level, are the ventilation units F which should manage with about 35°C. Further, the preheating of cold water takes place in the accumulators G followed by the last function where the subfloor heating H is provided at the lowest temperature level which normally range between 20 and 30°C. As mentioned above the idea is to create a large temperature difference since CO2 can easily provide high supply temperatures. To get a good overall efficiency a CO2 heat recovery system is however dependent on a low return temperature, which is indeed the idea with the "water fall concept".

3. RESULTS

3.1. Evaluation method

The system has a process monitoring as well as data acquisition tool called IWMAC. This measures and stores the system data such as energies, temperatures, pressures, etc. with a 1 minute resolution. Two ways to analyse the refrigeration and heat recovery system have been applied. Firstly to evaluate the refrigeration system the internal method which uses the compressors as flow meters have been applied. This allows the performance such as cooling capacity and coefficient of performance, COP to be calculated. Secondly the heating system was equipped with an energy meter measuring flow and temperature difference across the heat exchanger B allowing the total recovered heat to be calculated. Further, by measuring the temperature before and after each heat demand the different heating systems could be analysed respectively.

3.2. Heat available

The system was analysed during the 2014/2015 season so the amount of heat per month has been calculated. In the table below the total available heat from the refrigeration system has been compiled together with the recovered and rejected heat. The latter is divided into what goes to the geothermal storage and what is released to the ambient air.

Table 1 Total heat available, recovered and rejected from the refrigeration system vs month.

	Oct '14	Nov '14	Dec '14	Jan '15	Feb '15	Mar '15	TOTAL [MWh]	TOTAL [%]
Total heat available [MWh]	120.3	109.0	119.4	121.3	108.0	117.2	**695.3**	**100%**
Heat recovered [MWh]	75.8	70.2	93.2	93.6	80.5	86.3	499.6	72%
Heat rejected via gas cooler [MWh]	37.7	34.1	21.1	22.2	23.7	25.8	164.7	24%
Heat rejected in geothermal storage [MWh]	6.8	4.8	5.0	5.5	3.9	5.0	31.0	4%

As Table 1 above shows the available heat is relatively constant around 100 MWh per month throughout the season. The available heat is mainly dependent on the load situation in the ice rink in the sense that the more cooling capacity the more heat is available on the warm side of the system. In general a large portion of the available heat is used. During the first season, although there are still some tuning and improvement to do, the monthly average recovered heat is around 70 MWh corresponding to above 70% of the available heat.

Since the Gimo ice rink has a geothermal storage the surplus heat may be stored or rejected to the ambient air. Table 1 shows the amount of surplus heat which is distributed as 24% to the ambient air and 4% to the geothermal storage. This may seem surprising but can be explained by the fact that the geothermal storage and the subcooling function should primarily be used during warm conditions and the ice rink was taken into operation in October so the conditions were not ideal for evaluation of the geothermal storage.

3.3. Performance of the heat recovery

One way to illustrate the total performance of the refrigeration and heating system is to form a so called global COP which includes both the useful cooling capacity and the recovered heat according to Eq. 1.

$$COP_{Global} = \frac{Q_{cooling} + Q_{HR}}{E_{comp}} \tag{1}$$

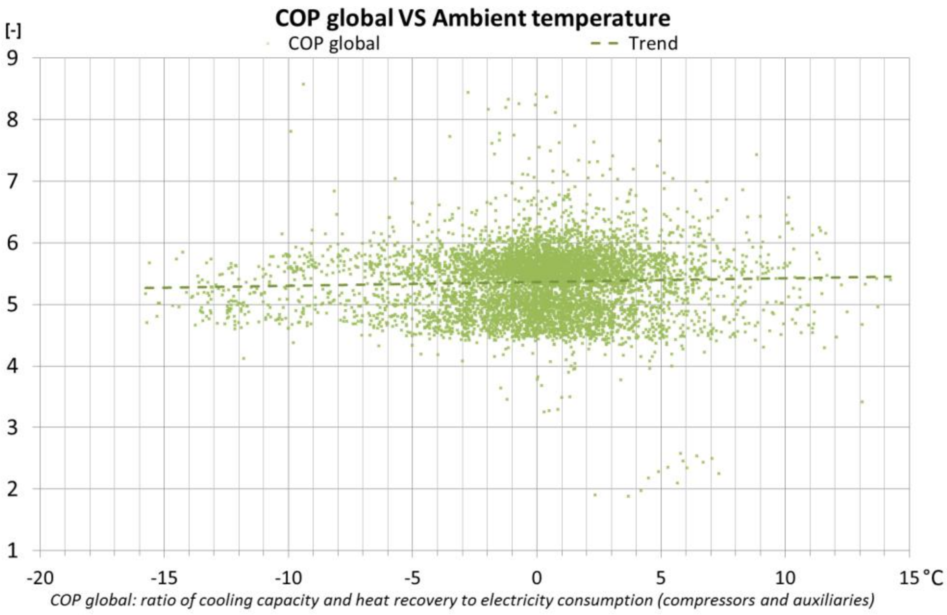

Figure 5 Global COP including cooling and recovered heat vs the ambient temperature.

Figure 5 shows that the global COP has an average of around 5 during a season. In order to take the performance analysis of the heat recovery function one step further another COP is defined which is referred to as heat recovery COP, COP_{HR}, and defined according to the Eq. 2 below where Q_{HR} is the recovered heat, E_{comp} is the actual compressor power input and E_{FC} is the compressor power in floating condensing (no heat recovery). As basis for the analysis the measured data is used but the compressor power E_{comp} is recalculated to a floating condensing condition E_{FC} rather than the measured power with elevated head pressure.

$$COP_{HR} = \frac{Q_{HR}}{E_{comp} - E_{FC}} \quad (2)$$

In case of floating condensing; the calculated condensing temperature is determined by the measured ambient temperature. If the ambient temperature is lower than 5 °C the condensing temperature is fixed at 10°C, otherwise, the condensing temperature is equal to the ambient temperature plus 5 K.

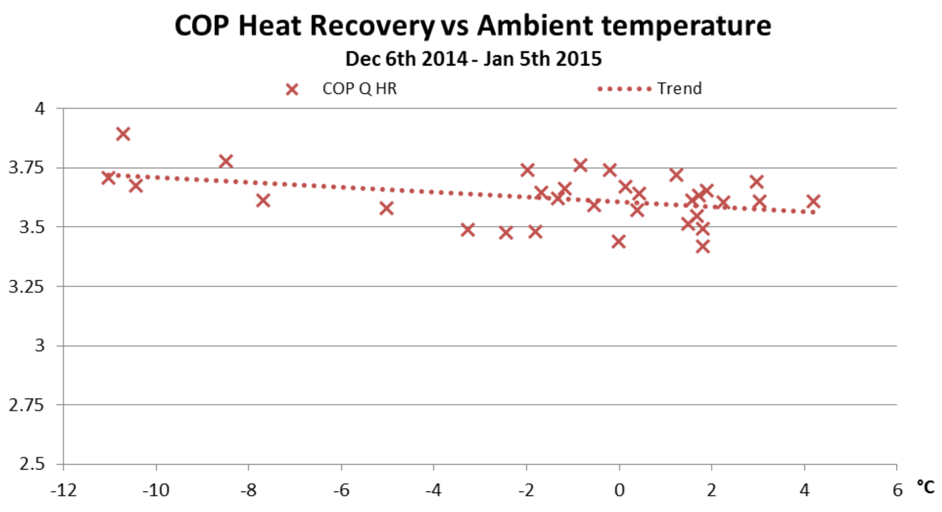

Figure 6 The heat recovery COP, COP_{HR}, vs the ambient temperature.

The COP_{HR} versus ambient temperature is shown in Figure 6 and it can be concluded that the level of the COP_{HR} is between 3.5 to 4 and increases as the ambient temperature decreases. This COP can be compared with another heating system such as a heat pump. Very few traditional heat pumps, depending on the heat source, can supply heat at an average COP of this level. Another advantage is that this system is refrigeration and heat pump function integrated in the same unit, so the same power feed, control system, refrigerant, etc.

The integrated heat pump option which uses heat from the geothermal storage to support the heat recovery function was never in operation during the first season. With the current heat recovery system control strategy it was never necessary since the elevated head pressure control managed to cover the heat demand. As a result of this evaluation it was concluded that there is room for improvement in the control strategy which would may include operating the heat pump function more frequently.

4. AREAS OF IMPROVEMENT

Operating an integrated refrigeration and heat pump system like this is more challenging than may be first expected. The art of controlling the head pressure to maximise the refrigeration performance in a CO2-system has been discussed since the late 80-ies. When optimising from a combined refrigeration and heat recovery perspective the control strategy becomes more complex. With the current system layout a third dimension comes into the picture namely the associated subcooling control which adds additional challenges. In this section these challenges are illustrated.

4.1. Head pressure control

This specific system have the options of using the geothermal storage and/or the gas cooler for cooling or subcooling the loop. The theory behind the "optimum control" of CO2-systems has been investigated by S. Sawalha (2008). As shown in Figure 7, an important fact of a transcritical mode was highlighted: for a same gas cooler exit temperature, different pressures are possible. A certain head pressure has to be found to achieve an optimum COP_2 at varying GC outlet temperatures which is illustrated in Figure 8.

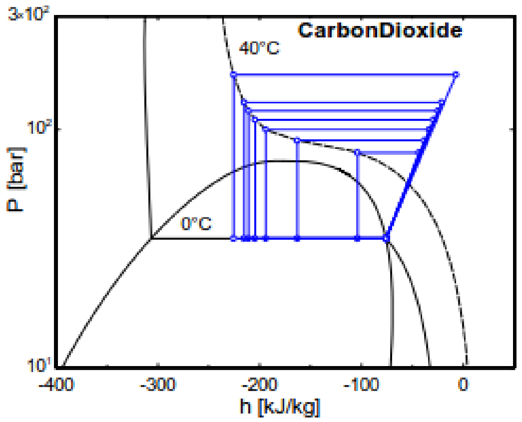

Figure 7 CO_2 transcritical cycles with a 40°C gas cooler exit and different discharge pressures

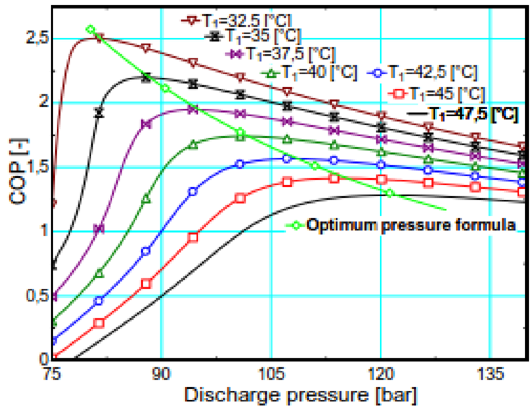

Figure 8 COP_2 of transcritical CO2 cycle vs discharge pressure for different gas cooler exit temperatures

Depending on the gas cooler exit temperatures and the compressor isentropic efficiency (Brown et al. 2002), the COP_2 versus discharge pressure as shown in Figure 8 will lead to an equation for the optimum head pressure:

$$P_{opt} = 2.7 \times (\mathrm{T_{amb}} + \Delta\mathrm{T_{GC}}) - 6.1 \qquad (2)$$

Where the temperature in brackets is the temperature before expansion device. The corresponding evaporating temperature does not significantly influence the optimum pressure; Kauf (1998), Chen and Gu (2005).

4.2. Heat recovery and influence of subcooling

The control of the gascooler and/or the geothermal storage will affect the subcooling and as Sawalha illustrated, Figure9, that there is a trade-off between subcooling and head pressure to achieve the best heat recovery performance. For a given cooling capacity a larger the subcooling will reduce the mass flow in and consequently reduce the potential heat recovery. In other words, with subcooling, a higher discharge pressure is needed to reach an equivalent heat recovery capacity as without subcooling.

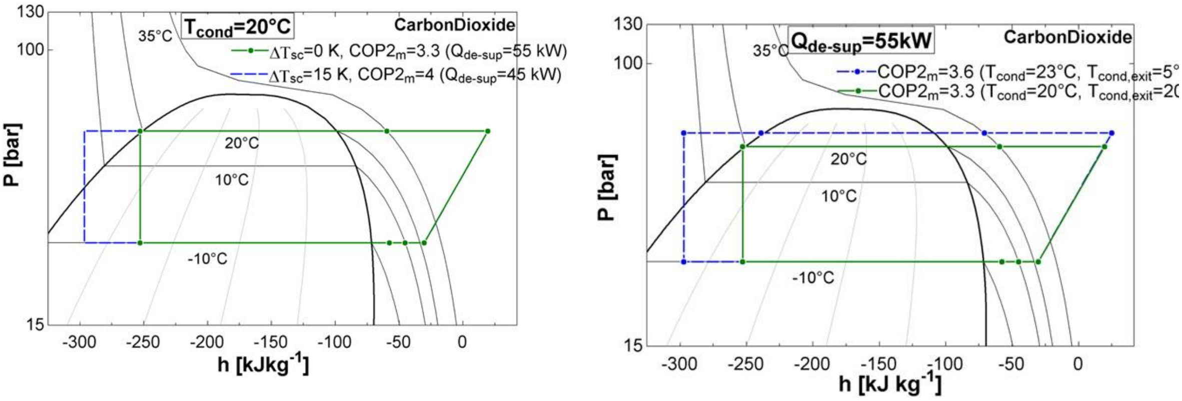

Figure 9 The impact of subcooling on a CO_2 system with heat recovery (Sawalha).

In order to extend the heat recovery control analysis Sawalha (2013) introduced a new parameter, the Heat Recovery Ratio, HRR which is the relation between the recovered heat and the cooling capacity. In order to maximise both the HRR is used as control parameter. In Figure 10 the combined control of the head pressure as well as the gas cooler outlet temperature is illustrated as a function of the HRR.

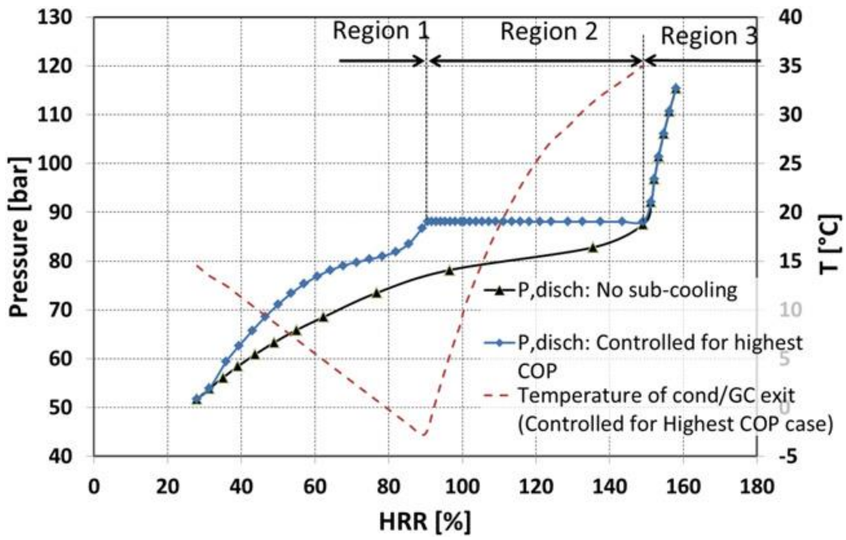

Figure 10 Control concept to optimise the heat recovery control (S. Sawalha).

In essence this control concept presents a pressure control to find a trade-off between the refrigeration and heat recovery functions based on discharge side pressure and CO_2 temperature before the expansion device. Whether this control concept is practically applicable on this system is not evaluated within the scope of this investigation, but will be studied in future work. A first attempt was however carried out by Tazi (2015) with an EES (Engineering Equation Solver) model based on measured data from Gimo. The analysis indicates that the actual discharge pressure was relatively close to the optimum pressure along the season but a sensitivity analysis will follow in future work.

Further, Tazi (2015) showed that for low HRR (< 65%), reducing the gas-/subcooler outlet temperature was beneficial to increase the COP_2. With high HRR (> 100%) however it would reduce the COP_2, which is in line with the results from Sawalha. Based on Tazi's model, the recommendation is to keep the subcooler outlet at 10°C for HRR lower than 82%, which can be done when operating the geothermal storage. At HRR higher than 82% the gas-/subcooler outlet temperature may increase linearly to the current heat recovery heat exchanger outlet temperature.

5. CONCLUSIONS

The use of CO2 is rather new in ice rinks but is since long time a proven system solution in supermarkets. The overall energy performance is very good with a daily energy usage of 1630 kWh including all required functions in a single sheet ice rink. The corresponding figure for the average Swedish ice rink is about 4500 kWh/24 hours. For the 6 months season this summed up to 296 000 kWh of electricity with a forecasted energy usage in a standard 8 month season of about 400 MWh. This can be compared with the average ice rink annual consumption of 1000 MWh.

What makes a difference in an ice rink is the heat reclaim. In ice rinks with a temperature controlled arena room the heat demand is similar to the refrigeration energy demand which makes it evident to recover the heat from the refrigeration system. The Gimo ice rink only uses recovered heat – no heat is purchased or produced outside what comes off the refrigeration system. The recovered heat is used for space heating, hot water production, subfloor heating, dehumidification etc. which implies it is used everywhere it can be used.

Although the 2014/2015 season was relatively warm the total recovered heat was 466 000 kWh in 6 months. Even during the coldest days with -15°C ambient temperature the facility manged to fulfil the heating requirements, which proves that 100% heat recovery works.

To further reduce the overall energy usage, an improved control strategy on the heat recovery system side may be developed. It has been observed that the geothermal storage has not been utilized as expected. The control sequence for the subcooling function is not fully developed and the discharge pressure control may be further optimized. When this strategy is set, the integration of the heat pump function could be improved since it has not been used so far. One reason being that the heat demand was covered with the heat available coming off the ice sheet. Another reason being that the evaluated winter season, 14/15, was very mild which reduced the need for geothermal heat.

NOMENCLATURE

COP_{global}	Global Coefficient of performance (-)
COP_2	Cooling coefficient of performance (-)
E_{comp}	Electricity consumption of compressors (kW)
COP_{HR}	Heat recovery coefficient of performance (-)
E_{FC}	Compressors power with floating condensing (kW)
$Q_{de\ sup}$	Desuperheating capacity (kW)
ΔT_{GC}	Temperature difference at the gas cooler
$Q_{cooling}$	Cooling capacity (kW)
P_{disch}	Discharge pressure (bar)
HRR	Heat recovery ratio (%)
T_{cond}	Condensing temperature (°C)
Q_{HR}	Heat recovered (kW)
T_{amb}	Ambient temperature

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