KTH Industrial Engineering and Management

CARBON DIOXIDE IN ICE RINK REFRIGERATION

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Abstract

The average energy consumption of one ice rink is around 1000MWh/year, which approximately 69% is occupied by the refrigeration unit and heating demand. With the aim of decreasing the energy consumption, a new concept of refrigeration system with CO2 as a refrigerant has been developed and it is promising to become a high potential next generation for refrigeration system in ice rink.

This thesis is to evaluate a new refrigerant application in ice rink refrigeration system under three different aspects; energy performance, heat recovery potential and economic efficiency. In order to make this evaluation, three main tasks are executed. Firstly, literature review and market statistic are processed to give a general picture of the CO2 development as a refrigerant. Secondly, a software Pack Calculation II is used for the simulations of CO2 refrigeration system and traditional ice rink refrigeration system. Älta ice rink located in Sweden, is chosen as a reference case for simulation's input data. The simulation results is to compare these system in terms of energy performance and heat recovery potential. Finally, life cycle cost of these systems is calculated to investigate the economic benefits from this new application.

Results from this study show good benefits of the new CO2 application in ice rink. From the market statistics, CO2 has become a successful refrigerant in supermarket food and beverage industry with 1331 CO2 refrigeration system installed until 2011 in Europe (Shecco2012). In ice rink industry, 24 ice rinks have been applied CO2 in the second cycle of refrigeration system; one ice rink in Canada applied a refrigeration system with only CO2 in the first cycle and the distribution system.

From the simulation's result, CO2 full system has been proven as the most efficiency system with the lowest energy consumption (30% lower than NH3/Brine system and 46% lower than CO2/Brine system) and the highest COP (6.4 in comparison with 4.9 of NH3/Brine system and 4.37 of CO2/Brine system). Regarding heat recovery potential, CO2 full system has highest energy saving in comparison with the other two systems.

Due to lower energy cost and service cost, the life cycle cost of CO2 full system is lower around 13% than the traditional NH3/Brine system, furthermore, the component cost of CO2 system is promising to decrease in the next years thanks to the rapid development of this market in supermarket industry.

To conclude, CO2 full system has high potential to become a next generation of refrigeration system in ice rink, however, because of its transcritical working, this application can be restricted in the regions of warm climate.

Acknowledgment

First and foremost, I would like to express my special gratitude and respect to my supervisor at KTH, Samer Sawalha, who have been very patient and spend much of time to give me the very first knowledge of CO2 refrigeration technology, also method of processing this thesis.

This thesis could not be successful with valuable support, critical data and experience sharing from Jörgen Rogstam, who leads me at the very first step to the completion of my Master thesis.

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1 Introduction

1.1 Background

Two important aspects of refrigeration systems are environmental perspective and energy cost. Some traditional refrigerants known as harmful to environment has been excluded from the market, the second generation represented by natural refrigerants, such as NH3 which has been developed over 50 years with many advantages. However, it still presents some disadvantages in term of safety and efficiency of the system. The risk of NH3 leakage is the main reason for indirect designs which NH3 is restricted as much as possible in the system, so it requires another liquid in the distribution cycle. Consequently, energy consumption of the indirect NH3/brine system is higher than a direct system because of heat exchange between different working fluids in the system. CO2 has been discovered as another candidate of natural refrigerant which is promising to solve the problem. The first concept for CO2 application in ice rink was in 1999 in Austria (GEA2011), CO2 is used in the second cycle with 80% decrease in secondary pump power (Rogstam2007). Then, with the successful story of CO2 application in supermarket, the second concept of CO2 is used for the whole cycle was applied in one ice rink in Quebec, Canada in 2011 what makes the results very positive. There has not been many research work of CO2 application in ice rink, and this study is to give a general picture of CO2 application in ice rink refrigeration system and a base study of evaluation of CO2 system in ice rink.

1.2 Aim of study

The aim of this study is to evaluate the saving potential in ice rinks by the use of transcritical CO2 systems. The first part of the study is based on the results from a number of previous work in the field of ice rinks and supermarkets; in addition, the ideas of CO2 application in ice rinks or in other industries are investigated. The second part of the study is to make a simulation and comparison between CO2 refrigeration system and the tradition refrigeration system in ice rink in terms of energy performance, heat recovery potential and economic efficiency.

1.3 Scope & limitations

This study focuses mainly on indoor ice rinks and only one typical ice rink in Sweden with 300kW cooling capacity is chosen for the simulation. The load profile or other input data, such as climate for the simulation is based on Sweden. The result can be applicable apparently for similar climate or similar ice rink.

The limitation of this study is the software used for simulation, some assumptions and calculations attached in the software can effect to the simulation results. The heat recovery potential of the systems is difficult to have a proper evaluation because the system control is fixed in the software and there are not many options to choose for different control strategy for the system.

2 CO2 as a refrigerant

2.1 Properties of CO2

CO2 has been known as a popular natural refrigerant from 1850 due to its safe properties in comparison with ammonia or sulphur dioxide. Until 1930, CO2 was stop using because of the entry of artificial synthetic chemical refrigerants. With high cooling capacity at high ambient temperature and the ability to work at lower pressure than CO2 that is an advantage for artificial synthetic chemical refrigerants to develop low cost and light heat exchanger components. However, till 1990, when CFC and HCFC present the ozone depleting ability, CO2 was introduced again as one of alternatives (Padalkar A.S.2010).

Safety

CO2 is non-toxic, non-flammable and non-explosive gas while ammonia is mild flammability, pungent smell and low threshold limit value. It is environmental friendly refrigerant since its ODP is zero and GWP is one.

Thermal-physical properties

One of the most different property of CO2 is the high operating pressure, as shown in figure 1, it is as 6 to 7 times as higher as NH3 at 0^0C. At the saturation temperature -10^0C, the operating pressure of CO2 is around 25bars, so operating pressure of CO2 is often in range between 25bars when it is evaporating around -10^0C and it can reach to 120bars at high pressure side in the cycle.

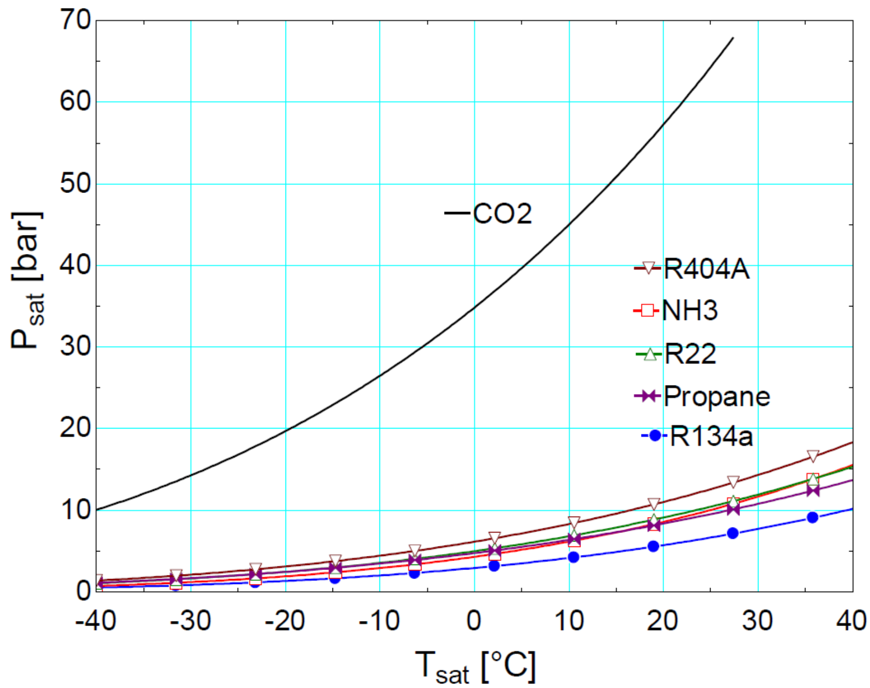

Figure 1: Saturation vapour pressure versus temperature for selected refrigerants (Sawalha2008)

Other important properties of CO2 resulting from high operating pressure is a higher volumetric refrigeration effect and lower vapor pressure drop than other refrigerants. Figure 2 shows the volumetric refrigeration effect of CO2 in comparison with other refrigerants. It is around 5 times higher than R22 and NH3 at 0^0C. Then, it can be seen in figure 3 that the vapour pressure drop of CO2 from -30^0C to -10^0C is much lower than other refrigerants and lower than NH3 also, it is stable to the change of temperature and it is only higher than NH3 when temperature is higher than 0^0C.

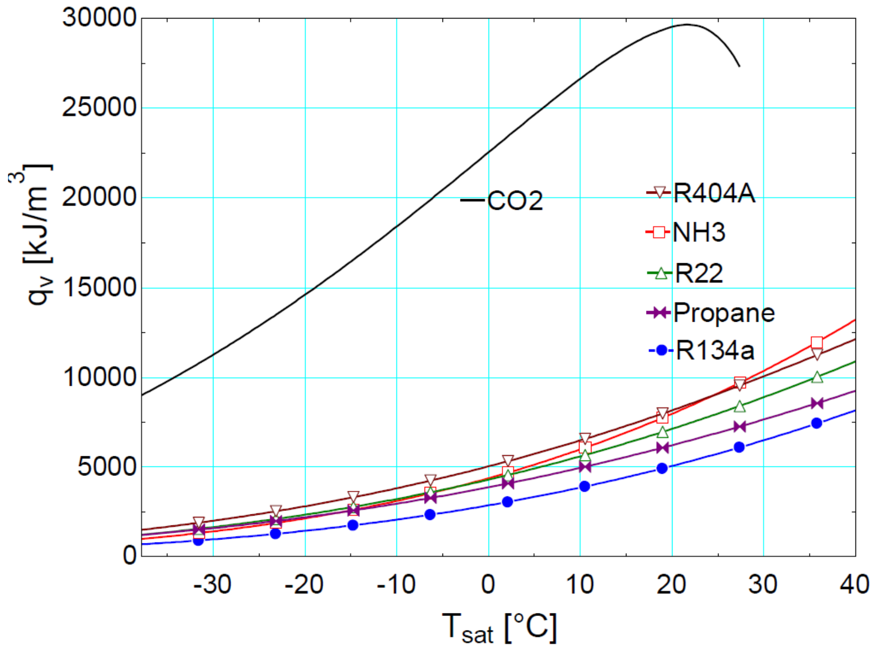

Figure 2: Volumetric refrigeration capacity of selected refrigerant (Sawalha2008)

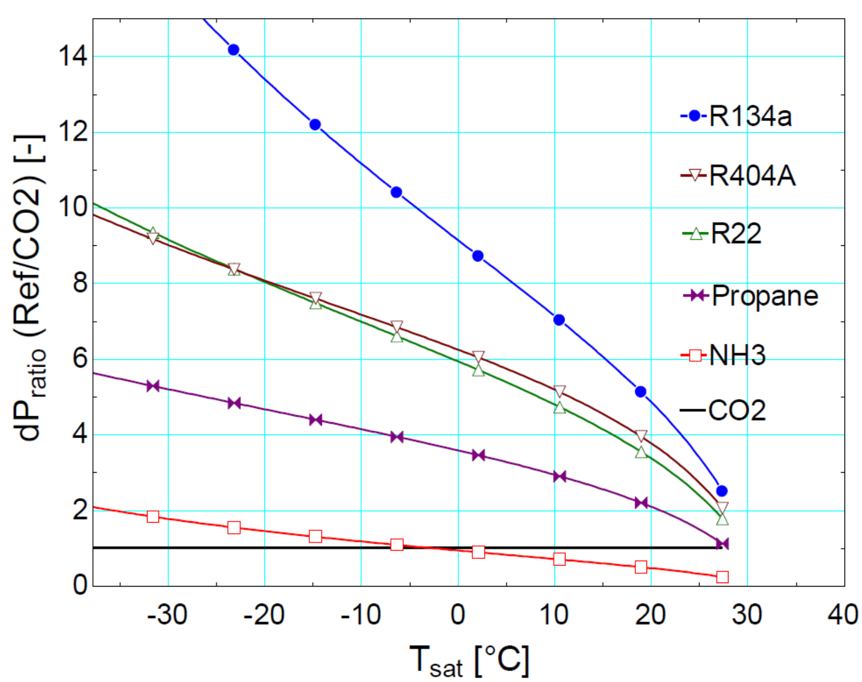

Figure 3: Vapour pressure drop ratios for selected refrigerant s a n d CO2 (Refrigerant/ CO2) at different saturation temperatures (Sawalha2008)

In the paper of Samer Sawalha (2008), these properties result that CO2 also operates in lower temperature drop, consequently, size of compressors, heat exchangers and tubing for CO2 system can reduce dramatically. Figure 4 shows the tube size of the suction vapour before compressor and the liquid after the condenser, it is much smaller than normal tube size of R134a and R22.

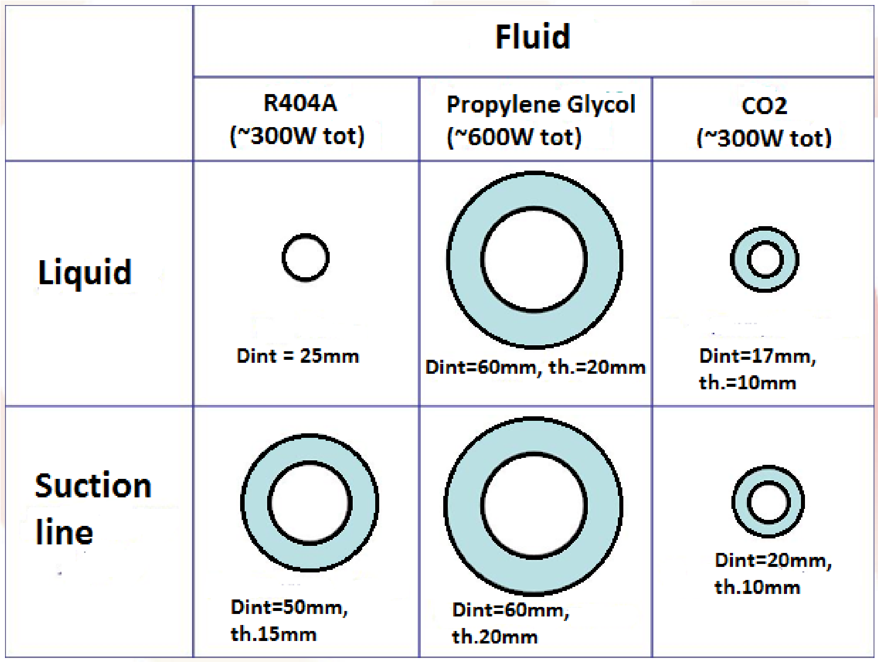

Figure 4: Tube size of three refrigerants (Torben2011)

Triple point of CO2 is at 5.2 bars and -56.6°C as shown in figure 5, which means that operation in the refrigeration cycle is always higher than 5.2 bars limit, at below this limit, solid CO2 will be formed. Together with low triple point, it can be seen in figure 5 that CO2 has a quite low critical temperature also, 31.1 ^{o} C.

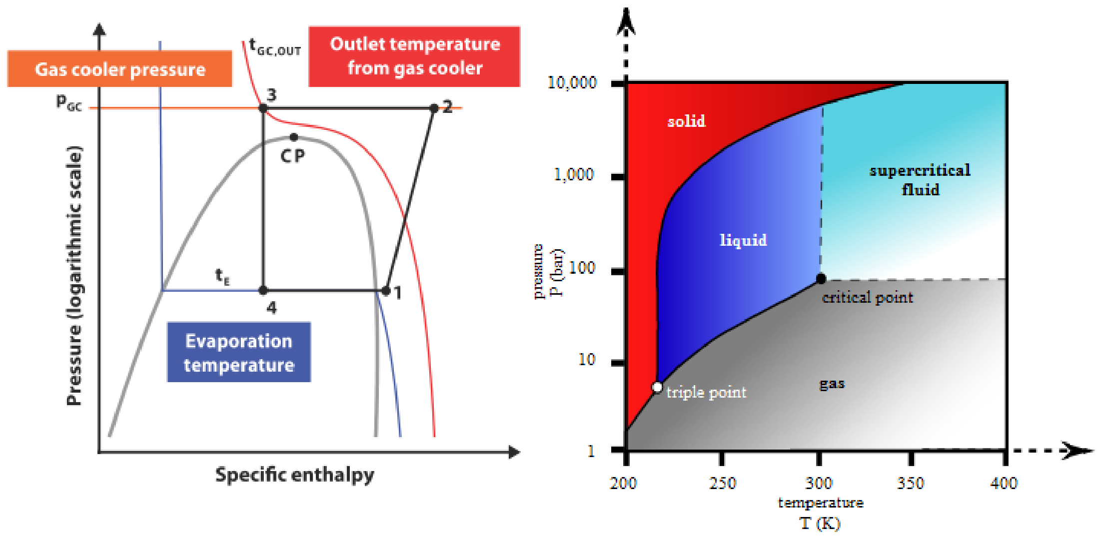

Figure 5: Transcritical cycle of CO2(Danfoss2010)

In a system using CO2 as a refrigerant, if ambient temperature is higher than 30^{o} C, the heat rejection process will operate at supercritical condition as shown in figure 5. Operation in this cycle creates a loss in cooling capacity and higher power consumption in compressor so a loss of COP will happen. As the results, the COP of system is changing to the ambient temperature, a higher COP can be achieved with a lower outside temperature. Also, when condensing temperature closes to the critical point, the cycle is jumped to operate in transcritical also and reject heat above the critical point, the systems suffered loss in cooling capacity and efficiency. Figure 6 shows COP2d of a basic cycle of CO2 in comparison with NH3 at -10^{o} C evaporating temperature, COP2d of CO2 cycle is lower then NH3 in all of condensing temperature. However, the CO2 discharge gas is warmer and stores higher energy density due to higher pressure and higher volumetric refrigeration effect, so the heat of CO2 vapor can be recovered better than other refrigerants. This can improve the performance of system if the heat recovery is utilized properly.

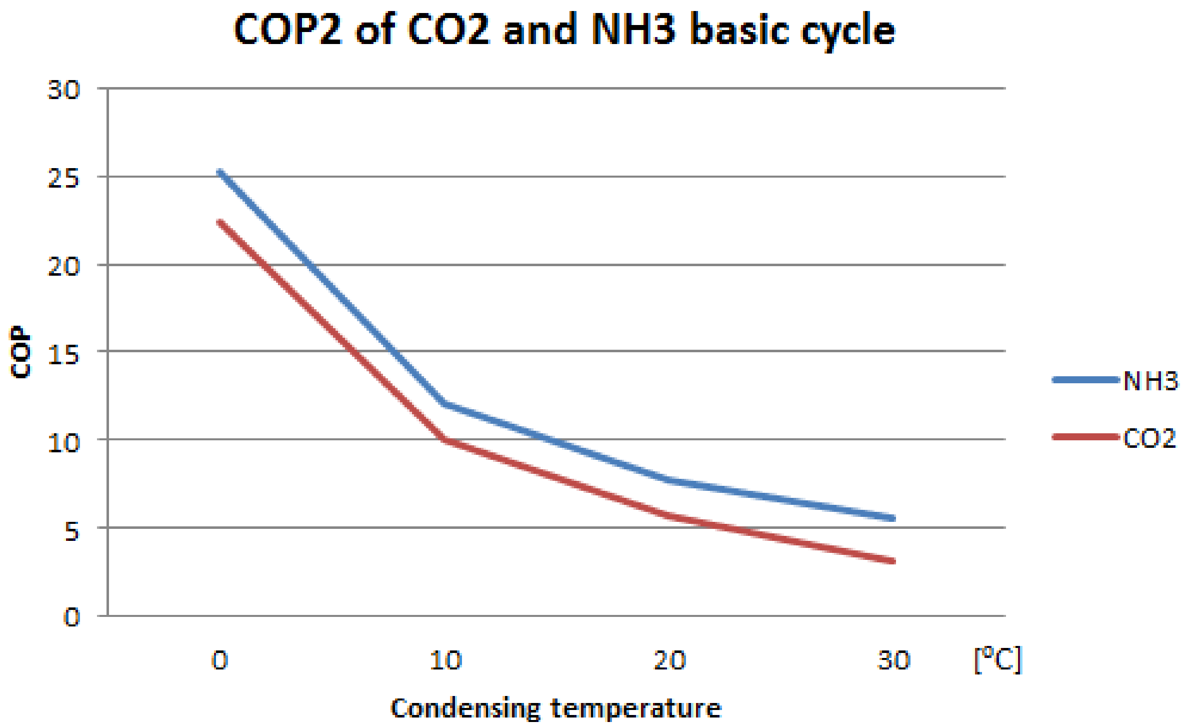

Figure 6: COP2 of a basic cycle at -10^o C evaporating temperature (Björn2009)

When the heat rejection works in supercritical, the condenser is called the gas cooler, the process is becoming a temperature change process, not a phase change process, so the pressure and temperature is not dependent. Then the discharge pressure is defined by refrigerant charge and in paper of Danfoss (2012), it proves that discharge pressure makes an influence on the compressor power or COP of system. Figure 7 illustrates the influence of discharge pressure to refrigerating capacity and COP, it can be seen that at the point of the highest COP is the optimum discharge pressure, so the discharge pressure can be controlled that the system can operate at highest COP. In Sawalha paper (Sawalha2008), the optimum discharge pressure is proved changing to ambient temperature or discharge gas cooler temperature. At every different ambient temperature and exit gas cooler temperature, the CO2 system has one specific discharge pressure that the system can generate highest COP, and it calls optimum discharge pressure, is illustrated in equation 1.

$$P_{opt} = 2.7 * (T_{amb} + T_{gascooler,app}) - 6.1 \quad (1)$$

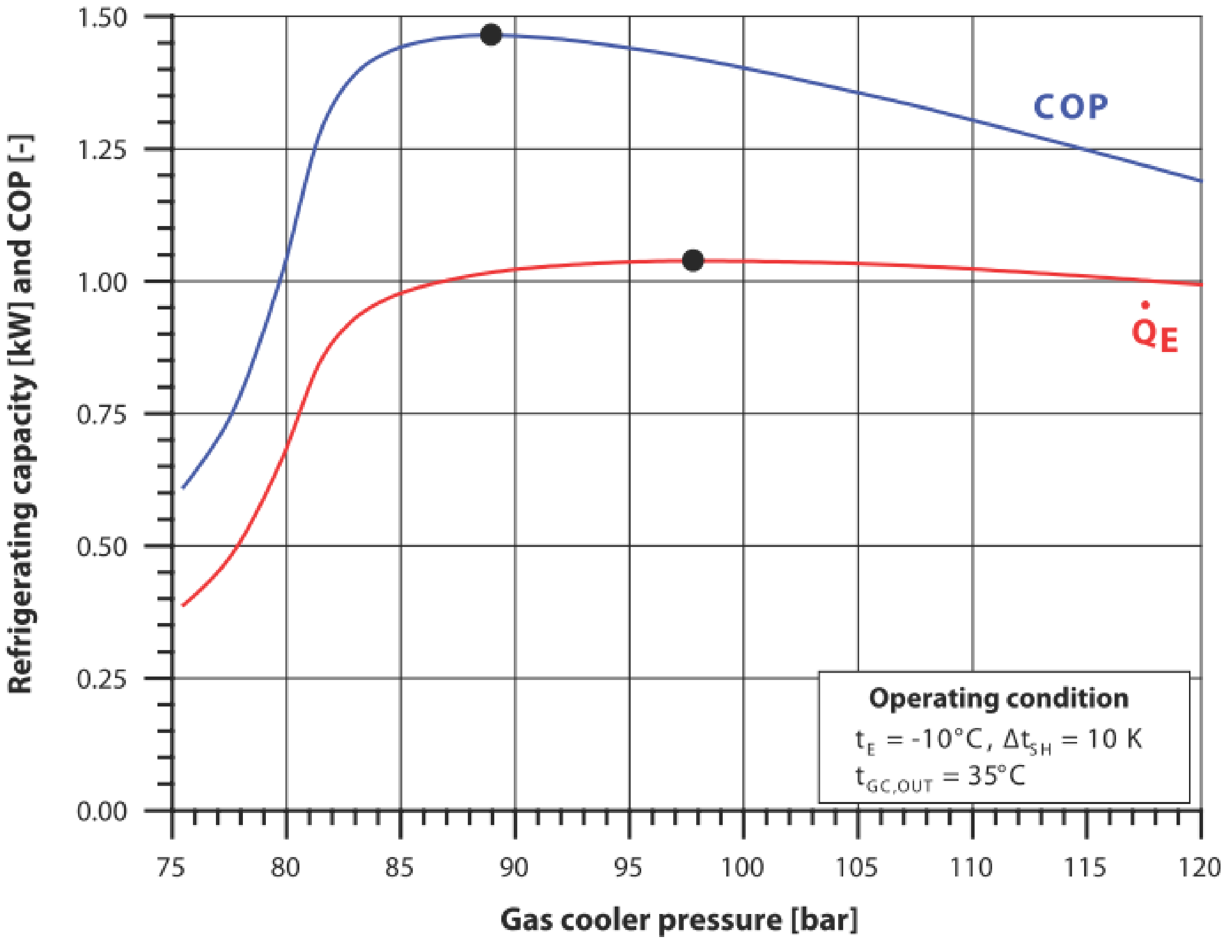

Figure 7: Effect of gas cooler pressure on refrigerating capacity and COP(Danfoss2012)

Safety issues

The vapour pressure of CO_2 is higher about 4-5 times in comparison with other refrigerants (Sawalha2008). It can be noticed as a dangerous problem for the components and demands for a thickness of material. However, the risk of explosion depends on the stored energy in vessel which decides by the refrigerant charge quantity in the components, vapour fraction, inside pressure and temperature (Padalkar2010). With really high volumetric refrigeration effect, the refrigerant charge of CO_2 is much lower (Padalkar2010).

Another issues of safety is the effect of CO_2 on human if they expose to high concentration. At more than 3% on volume basis, it causes hyperventilation, 5% causes narcosis and 10% causes coma (Padalkar2010). Ammonia has even more dangerous if it leakages into the enclosed space and the refrigerating technology also is really advanced now in handling with refrigerant even more toxic and heavier than CO2 ((Padalkar2010).

Conclusion

As conclusion, CO_2 is becoming a promising substitute refrigerant due to its very good thermo-physical, transport properties and safety to both environment and human in comparison with ammonia. The disadvantage of low performance for operating in transcritical cycle maybe overcome with better heat recovery system utilization.

2.2 Applications of CO2 as a refrigerant

Until 2012, CO2 has been applied as refrigerant in many different industries, from heat pumps, food industry, supermarket, ice rink or ice making to transport air conditioning. CO2 heat pumps have been applied widely in Japan, Ireland and UK for public building or domestic hot water for private residential housing as an efficient, renewable alternative to the oil fired boiler system. In Denmark, CO2 heat pump is on development for the district heating system also (Shecco2012). One of the most widely spread application of CO2 is in supermarket refrigeration system due to its non-toxic property and competitive performance in comparison with traditional R404A system. The design of CO2 supermarket refrigeration system is quite similar to design of CO2 refrigeration system for ice rink. Therefore, a detail of CO2 application in supermarket is discussed in this part which can be used to see a view for development of CO2 refrigeration system for ice rink.

2.2.1 CO2 supermarket refrigeration market

After HFC has been confirmed as a source for Global Warming, CO_2 has been increased as a substitute refrigerant in supermarket refrigeration system. Until now, about 1331 CO2 supermarkets in Europe 20 and 25 supermarkets in Canada are installed with CO2 refrigeration systems (Shecco2012). Denmark is the first country in developing the systems with HFC/CO_2 as a cascade system, then pure CO2 booster system at the end (Shecco2012). Today, there are 424 of CO2 refrigeration systems in Denmark; UK, Germany and Switzerland are the following countries that have a wide application of CO2 (Shecco2012).

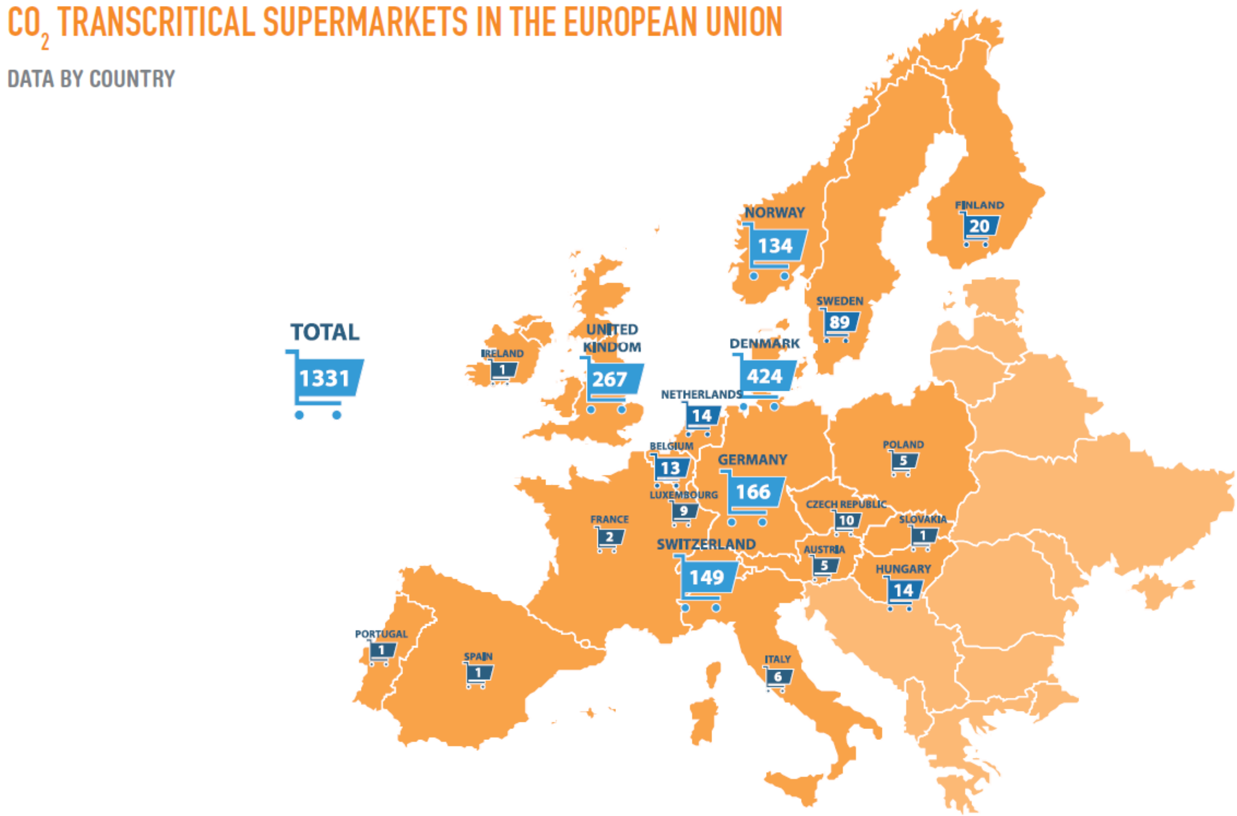

Figure 8: CO2 transcritical supermarkets in the European Union (Shecco2012)

In addition to this market growth, both CO2 sales and volumes are rising sharply in recent years as shown in figure 9, 88% in three year period from 2008-2010, and 117% in volume (Shecco2012). It is explained by the growing demand of Western Europe, while ammonia has a small increase because the NH3 market is becoming mature in recent years.

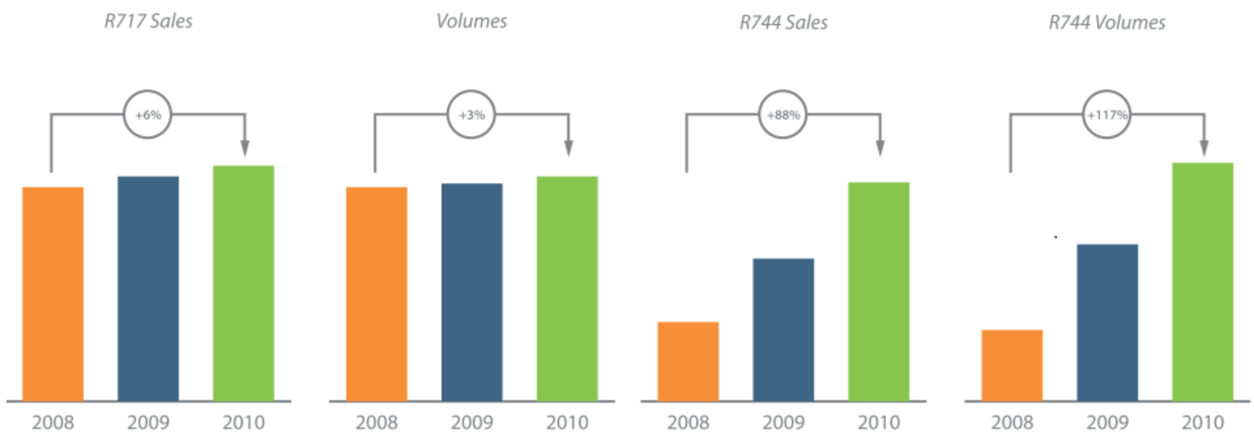

Figure 9: CO2/NH3 sales and volumes (Shecco2012)

The reasons behind this rapid development in EU are the tax, regulations and energy saving programs that put a pressure on a replacement for an environmental friendly refrigerant and higher energy efficiency. CO2 is a natural refrigerant that can meet the requirements of environment friendly, safety, reliable as well as competitive energy efficiency in comparison with other natural refrigerants like ammonia or synthetic refrigerants. In the market report of Shecco 2012, a survey is made to forecast the potential development of three natural refrigerants CO2, NH3 and hydrocarbons. It presents that CO2 is forecasted to become a new refrigerant, planed to use with 42% in comparison with 18% of NH3.

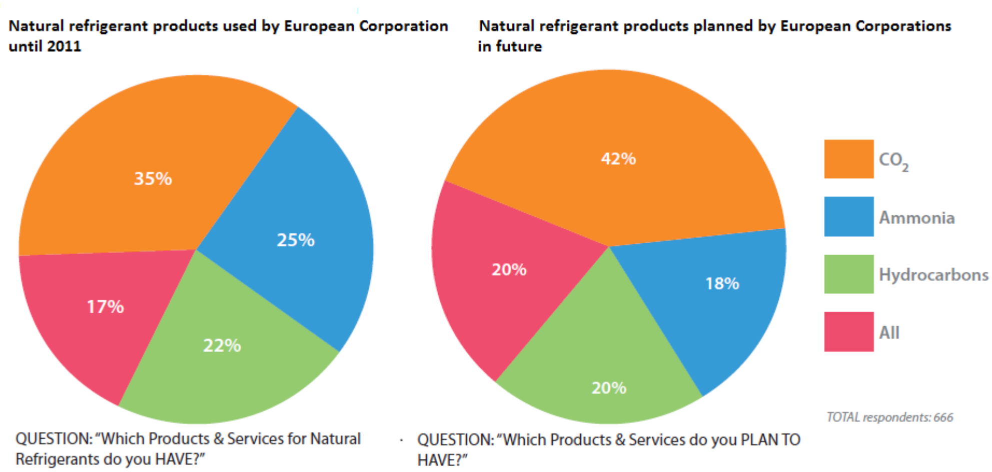

Figure 10: Prospect of CO2 market (Shecco2012)

2.2.2 CO_2 supermarket refrigeration system

The first design of CO2 refrigeration supermarket system is indirect system with ammonia, hydrocarbon or synthetic refrigerants in the high stage cycle and carbon dioxide in the low stage cycle as shown in figure 11.

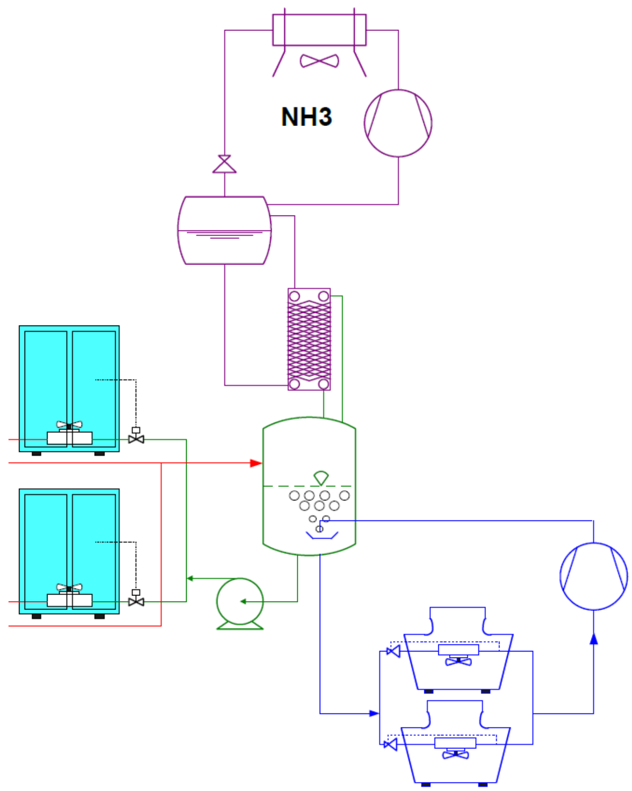

Figure 11: An example of a cascade NH3/CO2 system (Sawalha2008)

The later design coming after the cascade is the fully CO2 system or CO2 booster system as shown in figure 12.

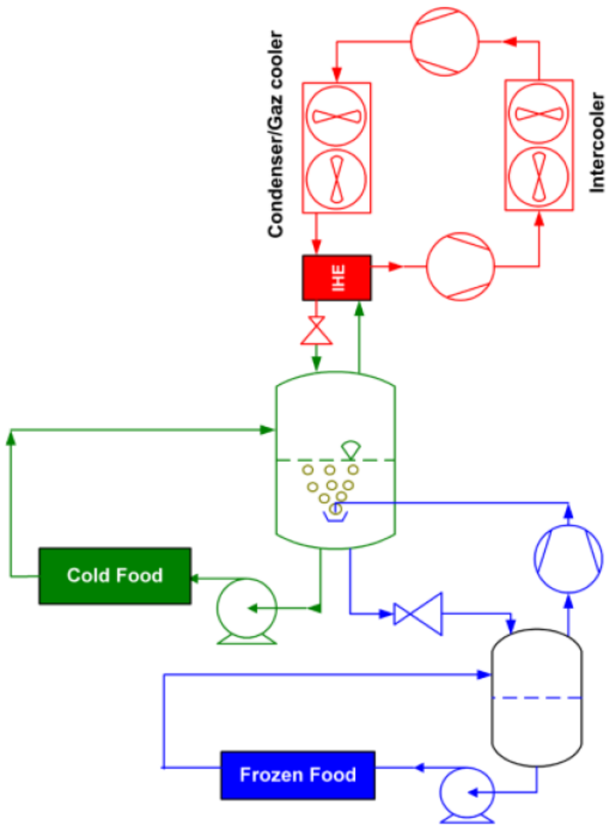

Figure 12: An example of a CO2 only centralized system (Sawalha2008)

Theoretical analysis of the R134a/CO2 cascade system results in a 16% reduction in energy consumption compared to a traditional DX R404A system. The R404A/brine indirect system consumes 41% more energy than the cascade system, mostly due to the brine pump (Elefsen and Micheme 2003). In Sawalha's paper (2008), the cascade system shows higher COP than the traditional R404A system at the calculated range of ambient temperatures. In case of Stockholm weather conditions, the full CO2 modified centralized system solution has the lowest energy consumption over the year, 12% less than R404A and 4% less than NH3/CO2 cascade, followed by the reference centralized system solution and NH3/CO2 cascade. All the investigated CO2 systems except the CO2 parallel solution consume less energy over the year compared to the traditional R404A system. In the case of Frankfurt, the modified centralized system also has the lowest energy consumption and consumes 2% less energy compared to the NH3/CO2 cascade. The result also shows that CO2 systems except parallel system are better for cold climates, such as the case of Stockholm. Whereas NH3/CO2 cascade system is better for hot climates, such as the case of Phoenix-USA; in this climate case the CO2 modified centralized system has only 1% higher annual energy consumption than the conventional R404A. As a result, from energy consumption point of view the CO2 centralized and the NH3/CO2 cascade solutions are good for low ambient temperatures whereas the NH3/CO2 cascade is good for high ambient temperatures. Both system solutions proved to be good alternatives to DX R404A system for supermarket refrigeration (Sawalha 2008).

In the paper of Torben Funder (2011), the cascade systems consume less energy than the traditional solution with two single stage R404A systems. It can be seen in the figure 13, NH3/CO2 cascade has 9%-11% energy saving, the cold climate has lower energy saving than the mild climate. Propane/CO2 cascade gives a similar pattern, however, this system has lower energy performance than NH3/CO2 cascade. CO2 booster system gives a highest energy saving up to 16% in cold climate like Stockholm, and 11% in Berlin, however this system does not give high percentage of energy saving in mild climate like Madrid and Beijing.

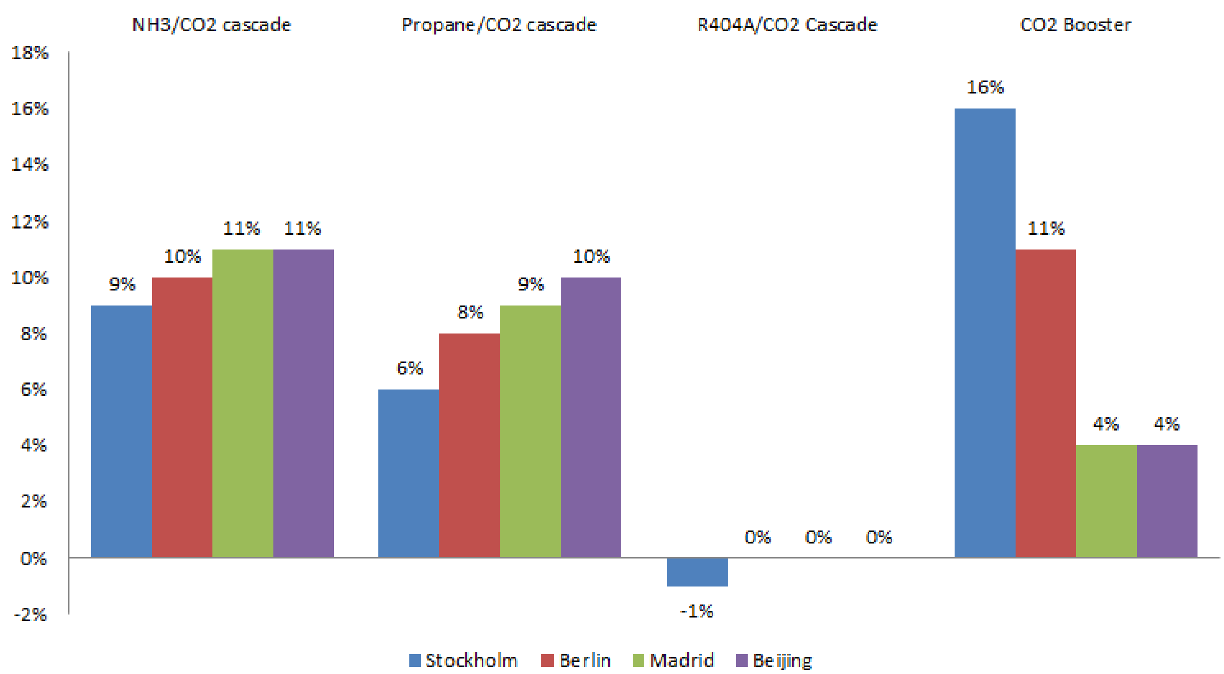

Figure 13: Energy savings of different CO2 cascade systems with traditional two singel stage R404A system (Torben2011)

2.2.3 Capital Investment, Installation and Maintenance cost

The capital cost of the R134a/CO2 cascade system is lower than the indirect option and slightly more expensive than the conventional DX system (Elefsen and Micheme 2003). With the characteristic of toxic and flammable, refrigeration system with NH3 as a refrigerant requires higher safety devices and special safety equipment (Rolfsman 1996). According to Girotto et al. (2003), the installation of the transcritical CO_2 system is more expensive than the R404A system by at least 10%.

Other industry sources estimate that, at the current pace of development, transcritical CO_2 refrigeration system could become established as a standard for the food retail market in about 2- 3 years. While present investment costs would be about 50% higher than for conventional systems, in 3-5 years the difference might have dropped to 20-25% [GTZ, 2008]. The interaction between policy drivers and costs as seen above could probably become the decisive factor to trigger a mass production of necessary technology at affordable prices (Nareco2009).

3 Ice rink energy demand

Ice rink is one of the industry that consumes a huge amount of energy every year. According to a statistic of Sweden in 2012, average energy usage of an ice rink is around 1000MWh per year (Rogstam 2012). The highest part comes from refrigeration system with a typical cooling capacity around 300-350 kW (IIHF, 2010). As shown in figure 14, the refrigeration system consumes about 43% of total energy consumption in ice rink, it can be explained by a high demand for freezing and maintaining a quite large ice surface area up to 2000m^2 at around -4^{0}C. Other energy demands are heating with 26%, lighting 10%, ventilation, dehumification and miscellaneous.

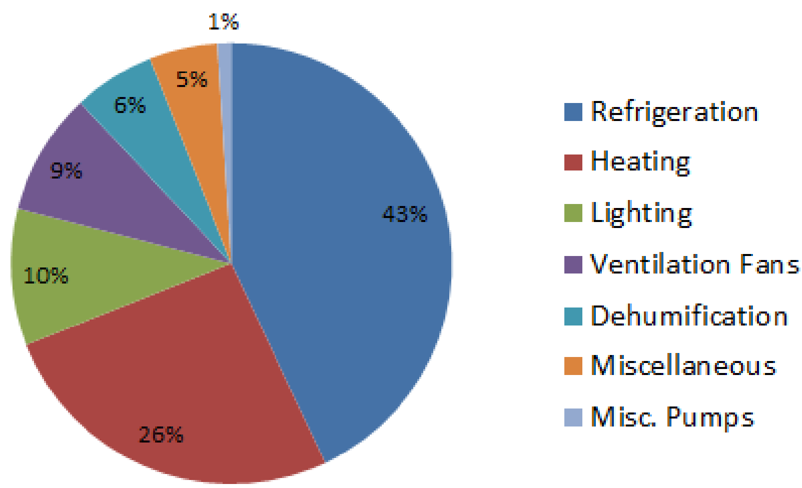

Figure 14: Total energy consumption in a typical ice rink (Rogstam2010)

Heating

There are variety of heating requirements in an ice rink:

- Space heating: in order to have a comfortable zone for both audiences and players, the space heating is not only the audience areas or public areas but also the space over the rink as shown in figure 15. The temperature should be around 10^{o}C in the stand and 20^{o}C in the public areas like dressing room, offices and restaurant.
- Domestic hot water (DMW): hot water is used in toilet, bathroom, ice resurfacing and ground heating.

- Snow melting, and subsoil or ground heating: snow after resurfacing process or frosting on the equipment.

The feeding energy for these systems is supplied by heat recovery from refrigeration system and district heating for hot water and sometimes electricity is used for hot water.

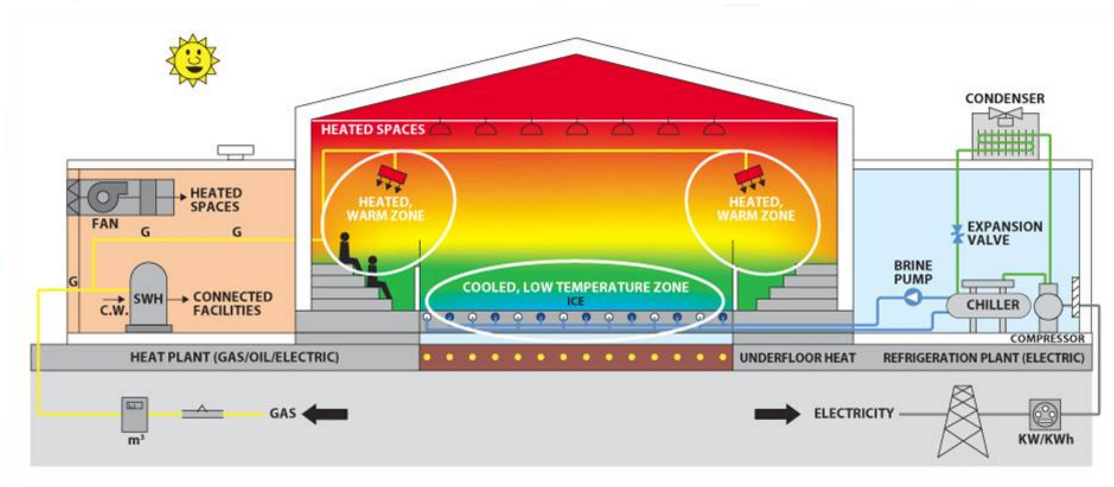

Figure 15: Energy System in Ice Rink (Retscreen 2005)

Lighting

The lighting system consumes around 10% of the total energy. Light intesity is different in different areas in ice rink and the light intensity can be controlled to decrease the energy usage (B.K. Dietrich1980). However, the lighting also contributes on the energy loss through radiation to the ice surface, so the energy of lighting system is not only electricity, but heat loss of lighting radition.

Ventilation

Ventilation system is designed with two purposes, to supply fresh air and to secure space heating also. Normally, it is also attached together with dehumidification system. Two zones of ventilation are applied in ice rink: the ice rink and public areas. Energy for ventilation is mainly consumed by fans, so the energy-saving factor in ventilation can be found in demand-controlled fresh-air intake and optimising the airflow rates according to the needs for minimizing fan power. However, it often works together with the dehumidification system, and the dehumification system has to run all the time, so the fans are running all the time with dehumidifaction also without consideration of fresh air need.

Dehumidification

Operation of dehumidification depends on local climate where the ice rink locates. The inside air of the ice rink often has high humidity because of many sources of water vapour. The main sources are coming from the fresh air of ventilation and uncontrolled air infiltration leakage through door opening, other sources can be moisture load of occupants, evaporating water from ice resurfacing process (IIHF2010). The energy usage for dehumidification will be coil cooling and fan power.

4 Refrigeration System in Ice Rink

The refrigeration system is the most vital part of the ice rink, not only because it is the main energy consumer but the capital investment for this system occupies more than 60% of total capital investment of an ice rink. The refrigeration system comprises of two main parts, a mechanical refrigeration unit to produce the cooling energy and a distribution system or secondary cooling system that transfers the cooling to the ice surface.

There are two types of refrigeration design, indirect system and direct system as shown in figure 16, about 97% of ice rink in Sweden are indirect systems or partly indirects. Ammonia is used as the refrigerant in about 85% of ice rinks in Sweden while the remaining use R404A, R134a or other HFC refrigerants (Makhnatch P. 2010).

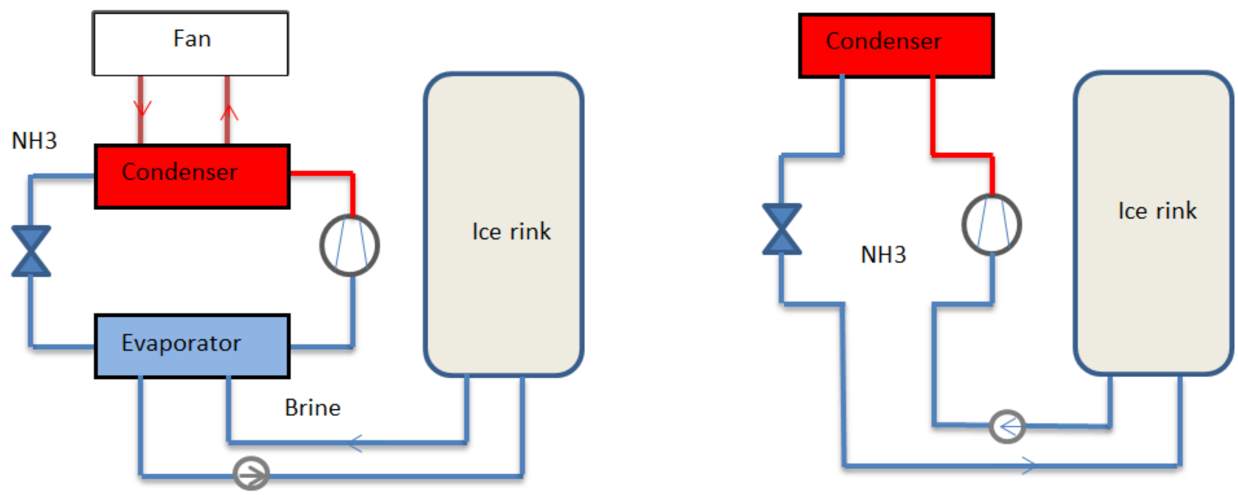

Figure 16: Indirect and direct system in ice rink

4.1 Direct system

In direct system, distribution system works as the evaporator, only one type of refrigerant runs for the whole system. So, it demands for a huge amount of refrigerant charge. R-22 and ammonia are the most used refrigerants for the direct systems but R-22 is banned now in many countries due to its global warming potential and ammonia has a charge limit according to its hazards and cannot be used in large systems including ice rink direct systems. In addition, capital investment of a direct system is higher than that of an indirect system and a direct system also requires more professional skills in design and installation (IIHF2010). Consequently, this method is not applied much in the icerink although the energy efficiency of the direct system is in general better than the efficiency of the indirect system.

4.2 Indirect system

An indirect system comprises of a mechanical refrigeration unit and a secondary cooling cycle, ammonia is used in the first cycle and a brine, such as $CaCl_2$ used in the distribution system. A heat exchange occurs in the evaporator between the refrigerant and the distribution system. First, the refrigeration unit generates a temperature around -10 ^oC to -12^oC at the evaporator. Then the secondary fluid in the distribution system is cooled to -5^oC to -2^oC which depends on the function of the ice rink, for example, ice surface used for icehockey or only for ice skating. This fluid is circulated between the ice pad and the evaporator in a closed circulation loop.

With this method, the amount of required refrigerant is minimized, the refrigeration unit size at same cooling capacity is reduced, and the most vital factor is the risk of refrigerant leakage can be avoided, especially to indoor ice rinks. However, the efficiency of this system will be lower in comparison with the direct system because of an additional temperature difference in the heat exchanger between the primary refrigerant and the brine resulting a necessary lower evaporation temperature, then a higher required pump power for secondary brine. Furthermore, some refrigerant such as ammonia can not be used as a secondary coolant because of safety reason.

I guess you mean that this solution implied that there is an additional temperature difference in the heat exchanger between the primary refrigerant and the brine resulting in lower evaporation temperature in the refrigeration system. The indirect solution also has additional pumping power for brine circulation.

Figure 17 shows the percentage of energy usage in the indirect system. The energy input is electricity where 80% is used by compressors, then 10% for the pump power of brine distribution system.

4.3 Distribution system

Figure 18 shows a typical piping arrangement for an ice rink. It comprises of many tubes as a U shape from the supply headers then return back. The ice pad is built over this pipe network and the brine runs through the network by a brine pump with frequency control.

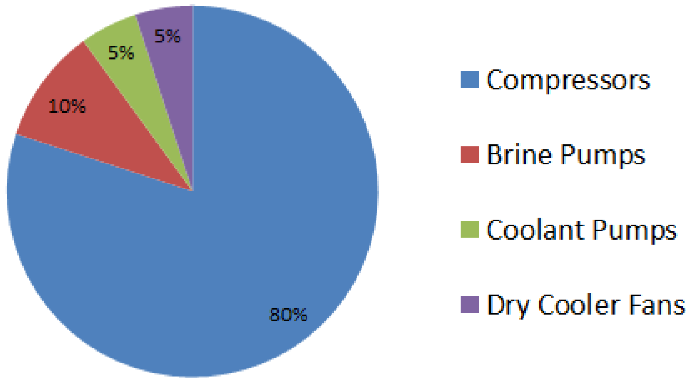

Figure 17: Monthly energy usage (kWh/month) for 19 indirect ice rink refrigeration systems (Rogstam 2010a)

Figure 18: Piping arrangement in distribution system (Katrineholm ice rink project 2006)

There are typically three types of pipe used in ice rinks, plastic pipe, steel pipe and copper tube. The most popular pipe installed in ice rinks is plastic pipe because of its light weight, easy for installation and low cost. Although steel pipe and copper tubes have higher heat transfer conductivity than plastic pipe but higher cost. In Sweden, only a few quite old ice rinks were installed with steel pipes, and only two ice rinks that CO2 is used in the distribution system, installed the copper tubes which can stand for high working pressure of CO2.

4.4 Heat recovery system

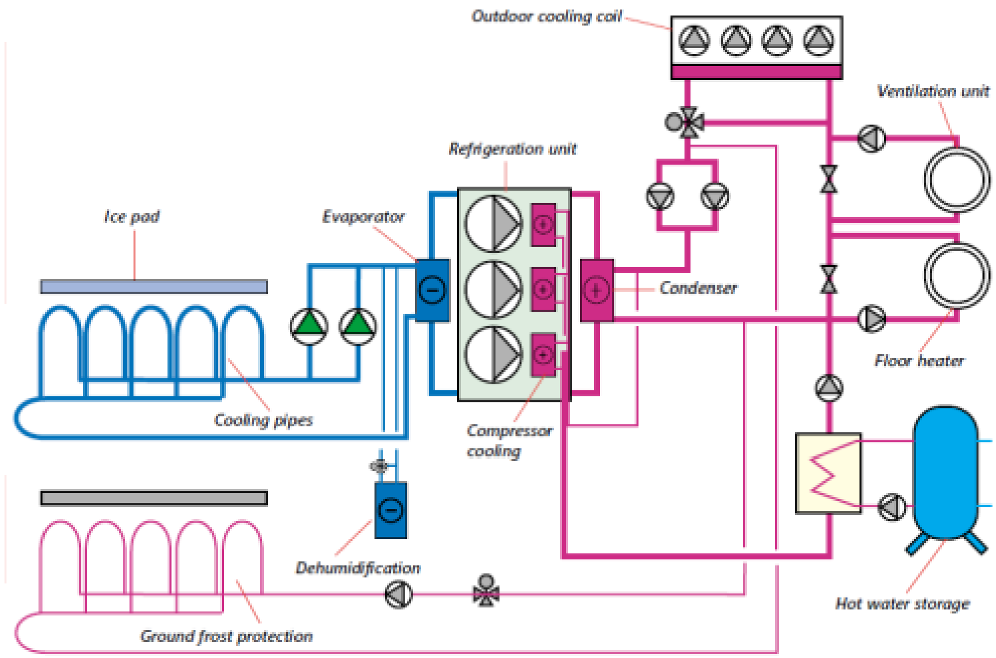

Figure 19: Refrigeration system with Heat recovery (IIHF2010)

A high quantity of saving energy can be acquired by a heat recovery system in ice rink. As shown in figure 19, the heat recovery system is used for ground frost protection, hot water, space heating and floor heating. The heat is collected by the dry cooler system and distributed to the heat exchangers under different temperature levels. The highest temperature is used for water heating, warm air is used for ventilation, space heating, dehumidification system or ground floor heater.

According to Olivier Piché, Nicolas Galanis, a direct use of this warm air for ventilation can reduce around 24.2% of the yearly heating consumption of ventilation system. In addition, it depends on the size of the heat exchanger, it can gain to 60.8% saving of heating energy consumption (Olivier Piché2010).

5 CO2 refrigeration system for ice rink

5.1 Refrigeration system of CO2 in ice rink

CO2 used as brine in secondary cycle - C1

This system is similar to the traditional indirect ice rink refrigeration system as described in figure 4 with NH3 in the first cycle but CO2 is used as brine in the secondary cycle. CO2 liquid from the tank goes through the pipe system and comes back to the tank as a mixture of vapour and liquid. The evaporator acts as a cooler to condense the CO2 gas, then CO2 liquid returns to the vessel and continues to the distribution system. The pump work is evaluated to save up to 90% of energy consumption compared to the conventional brine system (Rogstam2010a). However, as explained in part 2.1, operating pressure of CO2 is really high (4-5 times higher than NH3), so copper tube is used in distribution system instead of plastic pipe and the vessel should be able to work at a high pressure condition also. Consequently, it requires a quite high capital investment in this case for the vessel and the copper tube distribution system.

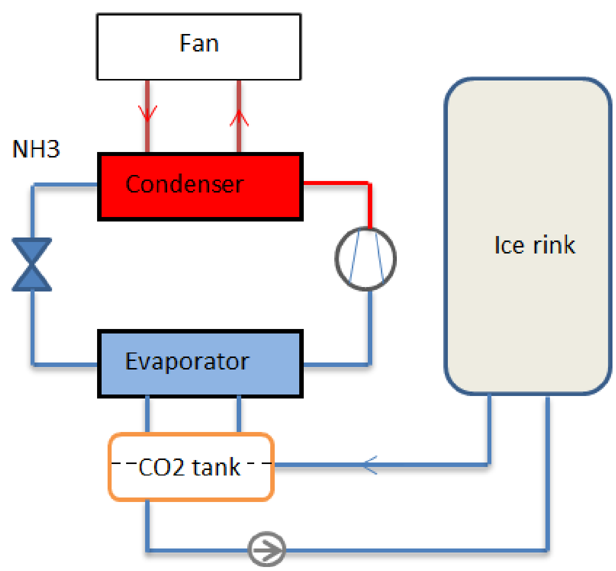

Figure 20: C1 - NH3/CO2 indirect system

CO2 as a refrigerant in the first cycle-C2

This system is similar to the conventional system (NH3/Brine), however, CO2 is replacing NH3 in the first cycle. If the sink temperature is higher than 30^{o}C, the heat rejection process will run in transcritical, and the cycle performance of CO_2 may be lower than cycle performance of NH_3. However, the advantages are a direct condenser and higher heat recovery potential of CO_2.

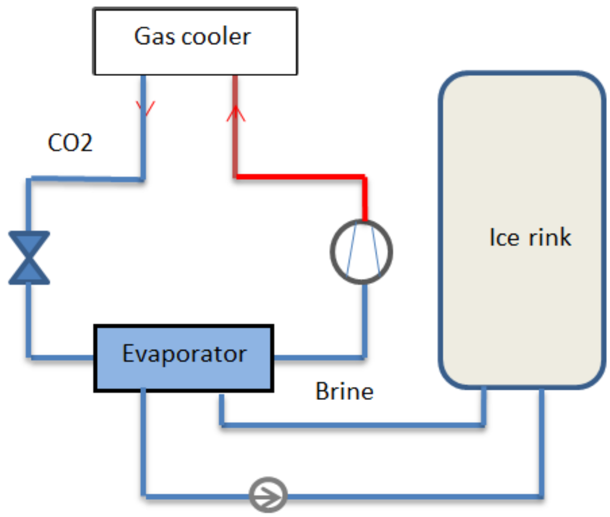

Figure 21: C2 - CO2/brine partly indirect system

Fully CO2 system-C3

In this system, CO_2 is used fully both in primary cycle and distribution system as shown in figure 22 and it operates as a direct expansion system. There is no heat exchange loss between evaporator-distribution system and refrigerant-coolant in the condenser side. First, due to working without evaporator, the system can work with a higher evaporation temperature profile also in comparison with traditional indirect system. Secondly, the pump work in the distribution system can be decreased by up to 90% in comparison with a traditional of brine system. Thirdly, heat recovery potential is higher due to higher temperature and volumetric heating capacity of CO_2 gas out of compressor. Finally, there is no capital investment for the evaporator and cheaper CO_2 compressor can be another advantage of this system.

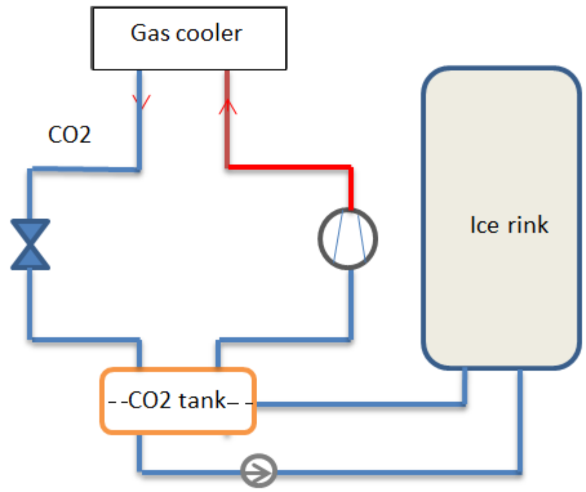

Figure 22: C3 - Fully CO2 direct system

CO2 indirect condenser-C4

CO2 is used as a refrigerant in the primary cycle and brine is used in the distribution system, but the refrigerant is condensed under dry cooler mode. This design can be possible in ice rink application, however it shows many disadvantages in comparison with other designs. The heat loss happens in the condenser because of indirect condensing and CO2 is a non-toxic and non-flammable refrigerant, so it does not demand to restrict the refrigerant charge. Direct condensing or air condensing can be applied to avoid the heat loss. In term of system performance and energy saving, it seems not to show any advantages in comparison with C2 system.

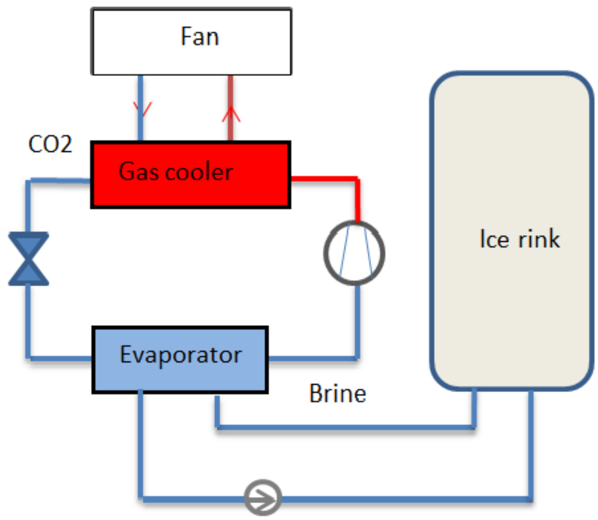

Figure 23: C4 - CO2/Brine indirect system

5.2 Case studies

Until 2011, almost all of the ice rink refrigeration systems in Sweden are running with NH3 as a refrigerant in the first cycle and brine in the secondary cycle. The first application of CO2 in ice rink was in Austria in 1999 with CO2 used in the secondary cycle instead of brine and NH3 in the primary cycle (GEA2011). In Sweden, similar technology was applied in Backvallen ice rink in 2006 and the other two ice rinks (Tingvalla Ice Stadium, Göransson Arena) were built on this technology (GEA2011). About 25 ice rinks in the world as shown in table 1, mainly in EU have applied the similar technology with CO2 in the secondary cycle (GEA2011). When CO2 is used in the secondary cycle, the pump power in the secondary cycle can save up to 80% in comparison with using brine as the conventional system.

In 2010, the first ice rink in the world, Arena Marcel Dutil in Quebec, Canada was built on a new technology of using CO2 for the whole system (Simard2012). It resulted a quite good performance in comparison with the traditional NH3/brine refrigeration system. In this part, Backvallen Ice rink in Katrineholm, Sweden and Arena Marcel Dutil, Quebec, Canada are discussed in term of system design and energy performance for CO2 application in ice rink.

No.	Ice rink	Country	Year
1	Dornbirn	Austria	1999
2	Wil	Switzerland	1999
3	Ascona	Switzerland	1999
4	Lyss	Switzerland	1999
5	SportFinanz Arena	Switzerland	1999
6	Bern Weyermannshaus	Switzerland	2000
7	Zug	Switzerland	2000
8	Wengen	Switzerland	2002
9	Lausanne	Switzerland	2002
10	Ravensburg	Germany	2003
11	Bern Ka We De	Switzerland	2004
12	Mannheim	Germany	2003
13	Haarlem	Netherland	2004
14	Torino 1	Italy	2006
15	Katrineholm	Sweden	2006
16	Karlstad	Sweden	2007
17	Sandviken	Sweden	2009
18	Enschede	Netherland	2008
19	Breda	Netherland	2010
20	Zug	Switzerland	2010
21	Arena Marcel Dubil	Canada	2010
22	Karuizawa	Japan	2011
23	Kristiansand	Norway	2011
24	Vahterus Ring	Finland	2011
25	Paranomovo	Russia	2011

Table 1: Ice rinks with CO2 application (GEA2011)

5.2.1 Backvallen Ice rink, Sweden

Backvallen locates in Katrineholm, Sweden, it is an indoor ice rink with 300kW cooling capacity supplied by two refrigeration units KA1 147kW and KA2 153kW. Figure 24 shows the simple layout of the refrigeration in Backvallen ice rink.

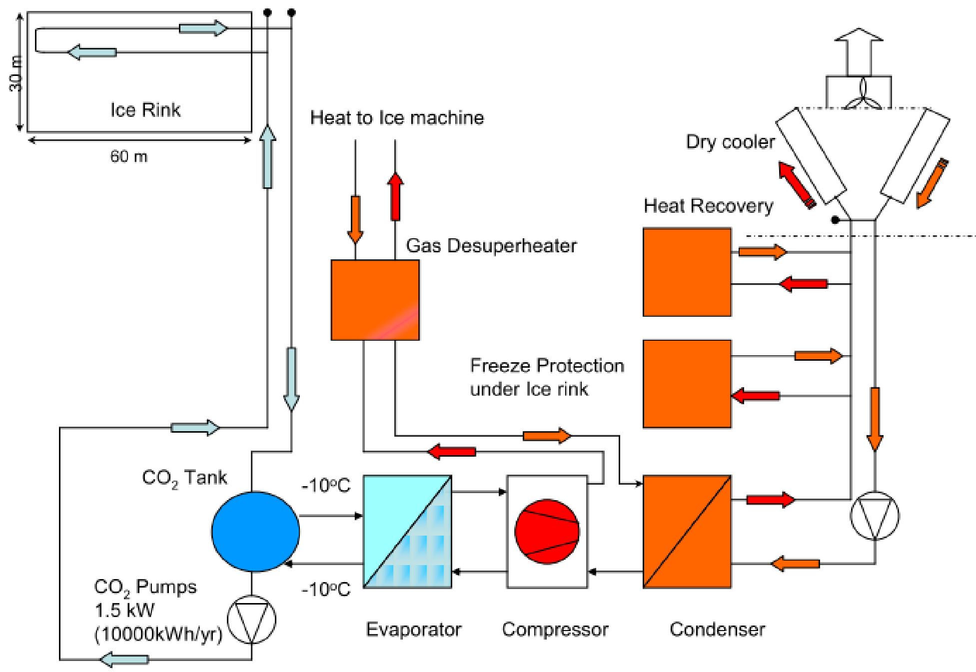

Figure 24: Simple layout of the refrigeration system (Rogstam2007)

The system in figure 24 is running with NH3 in the primary cycle and CO2 in the secondary cycle. NH3 is evaporated at -11^0 C and condensed at 38^0 C. Scroll compressors and dry cooler with ethylene glycol as a coolant are designed for the system. A desuperheater and another heat recovery system after condenser are attached to the system which can provide heat to ice refurbishment machine, freeze protection heating, space heating and heat for locker rooms.

The secondary cycle is running with CO2 and a CO2 tank installed in the system. CO2 liquid is pumped through copper tube circuits under ice surface and the heat absorbed from the ice surface makes the CO2 liquid evaporate then CO2 vapour and liquid mixture comes back to the tank. CO2 vapour is condensed into liquid at -10^0 C by the NH3 in the evaporator and return back to the CO2 tank.

Two important differences between this system and the traditional NH3/Brine system is the CO2 tank and pump power in the secondary cycle. The tank is necessary in this system because CO2 operates in two phases, liquid and vapour while in the traditional system NH3/brine, the brine is only operating in one phase of liquid. Secondly, pump power in the secondary cycle of CO2 system is much smaller than pump power of brine system due to low pressure drop and high operating pressure of CO2. Table 2 compares normal pump power of a traditional brine system and a system with CO2 in the secondary cycle for similar cooling capacity. It is obvious that the pump power of the traditional brine system is 5 times lower than the one of CO2 system

(Rogstam2007,Thomas2010). The energy analysis of Backvallen Ice rink also shows that pumping power has been reduced to 75%-80% resulting in a saving of 75-80MWh electricity for 8 months season in comparison with CaCl2 system (Rogstam2007).

	Cooling capacity	Brine pump power	CO2 pump power
Backvallen Ice rink	300kW	15kW	3kW
Göransson Arena	1316kW	60kW	12kW

Table 2: Brine pump power/CO2 pump power (Rogstam2007, Thomas2010)

However, the high operating pressure of CO2 also creates another difference. Copper tubes evaluated to be able to work at high pressure condition is used for CO2 system instead of plastic pipe. Installation and material cost of copper tube are higher than plastic pipe, but the copper tubes can pay back in long term with higher heat transfer that can reduce the heat loss between distribution system and ice pad. To a report of Thomas (2010) about energy performance of Göransson Arena, another advantage of the usage CO2 in the secondary system is a decrease of maintenance cost due to non-corrosive property of CO2 in comparison with the brine (CaCl2) that often has problem with corrosion.

5.2.2 Arean Marcel Dutil, Canada

Refrigeration unit

This ice rink was built in 2010 and was the first ice rink using 100% CO2 in the system. Figure 25 shows the design of refrigeration unit with heat recovery and CO2 is the only fluid for the whole system (Simard2012). The capacity is 317kW operating by four compressors, one expansion tank, gas cooler, one internal heat exchanger and 3kW pump power in distribution system.

There is no second fluid in the system, so evaporating temperature acts as distribution temperature also and the distribution system operates as an evaporator. Due to the absence of the heat exchanger between the primary cycle and the secondary cycle, evaporating temperature of a full CO2 system is higher than the one of the traditional system. With this advantage, energy consumption of CO2 full system can be decreased. The requirement temperature of the ice sheet in this case is around -5^0 C so the evaporating temperature is set at -7^0 C in this full CO2 system instead of -11^0 C in traditional NH3/brine system. CO2 liquid is pumped over the distribution circuit and it evaporates by heat absorption from ice surface. Then CO2 vapour and liquid mixtures return to the low pressure tank where the low pressure CO2 vapour goes through the compressors. Finally, it is condensed/cooled by gas cooler and comes back to the tank as liquid. The distribution system is plastic-coated soft copper tube.

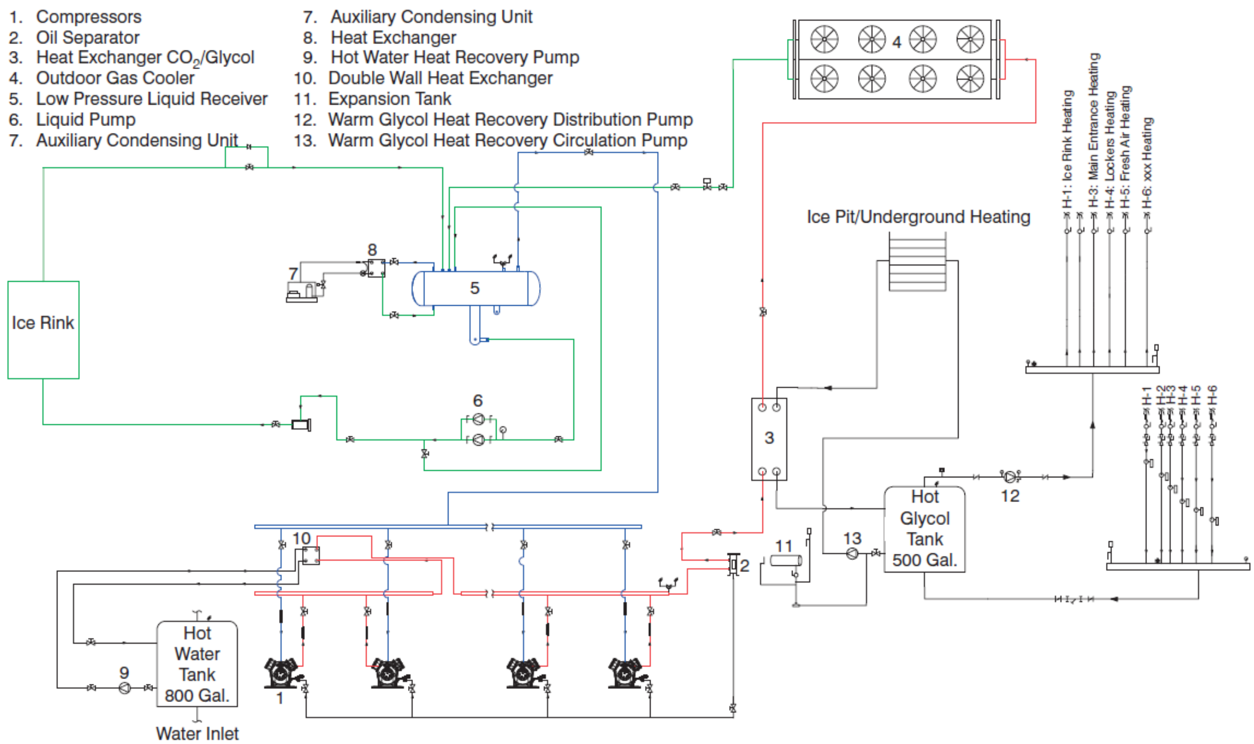

Figure 25: Simple layout of CO2 full ice rink refrigeration system (Simard2012)

Heat recovery and energy performance

There are two heat recovery units attached in the system, one is desuperheater for hot water storage tank on one side and another is a warm glycol loop through another side after compressors. The hot water tank delivers 75^0 C water at constant temperature to the building, and it is reported to be able to meet full heating demand for the building, district heating and back-up electric heating are never used during last season 2011-2012 (Simard 2012). The glycol loop operates with input temperature of 45^0 C and output of 55^0 C and this heat is delivered to ventilation system, locker rooms and central air unit (Simard 2012).

According to a published paper of Simard (2012), annual coefficient of performance (COP) of this refrigeration unit is 3.55 which is achieved due to higher evaporation temperature and lower pump power in the distribution cycle. Energy consumption of Arean Marcel is compared with a tradition ice rink (NH3/Brine) at similar capacity in the same area and the result shows a reduction of 25% in total energy costs in Arean Marcel (Simard 2012).

6 Simulation and Results

The software Pack Calculation II is used for the simulations. One ice rink in Stockholm, Sweden is chosen as a reference case, cooling capacity and evaporation temperature profile of this reference ice rink are used for all of the simulations. The first simulation is made for this ice rink and results from this simulation are compared with the measurement data of this ice rink. Other simulations are based on the reference case.

6.1 Reference case

6.1.1 Description

The reference ice rink is Älta locating in Nacka district, Stockholm. It is about 1800$m^2$with capacity of 700-800 people. It is opened in the period from April to August annualy. The refrigeration system is indirect with amonia as refrigerant, calcium chloride 24% - water as secondary refrigerant and the coolant is propylene glycol 40%-water.

Two GRAM reciprocating compressors with total nominal cooling capacity of 400 kW and 90 kW nominal motor capacity are the driving forces of the refrigeration plant in Älta ice rink. The evaporator is a flooded type plate heat exchanger. The condenser is an Alfa Laval plate heat exchanger and the desuperheater is a shell and tube heat exchanger. A heat recovery system is installed after the compressor, however it is not designed in order to meet the heating demand of ice rink, so the compressor power, pump and fan power are not effected by the demand in the heating system. The measured data of the icerink is taken by ClimaCheck instrument.

Figure 26 shows the principle of refrigeration system in Älta with following technical specification:

- Design evaporation temperature -15^{o}C, condensing temperature 30^{o}C
- Constant condensing temperature 25^{o}C
- NH3 is forced to circulate in the primary cycle, so no pump in the primary cycle
- Two nominal 15 kW brine pumps
- Two 11 kW coolant pumps and none of the machinery is controlled by frequency converters
- Fan power 12kW
- No internal heat exchanger

- Dry cooling condensing and constant condensing temperature with speed controlled fans and pumps

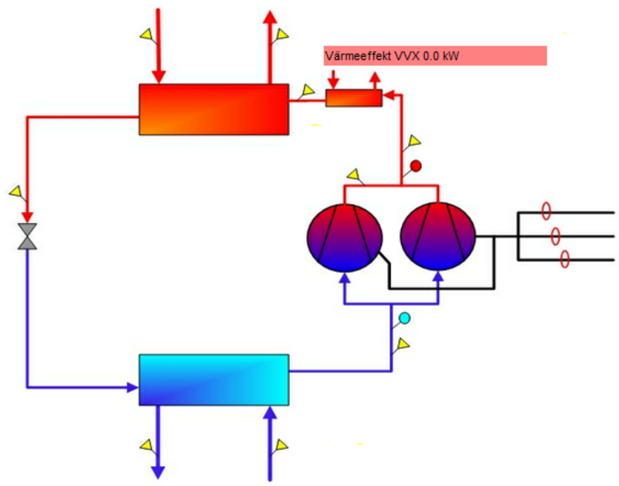

Figure 26: Principle of refrigeration system in Älta ice rink

6.1.2 Load profile

Period choosing

The cooling capacity profile and evaporation temperature profile (load profile) are calculated from the measured data of Climacheck during the period from April 2011 to April 2012. Normally, the ice rink does not open from May to August, but in 2011, Älta opened all around the year, except July. In order to have a year load profile (April 2011-April 2012), a load profile of one typical week is calculated then it is applied for the whole year.

The calculation to choose this typical week is to following steps:

- Step 1: Calculate the average weekly energy consumption
- Step 2: Calculate the energy consumption of each week by summing up the hourly energy consumption
- Step 3: Calculate the differentiation between average weekly energy consumption and the energy consumption of each week
- Step 4: Choose the week having smallest differentiation

The result shows that energy consumption of the week 24th October-31st October generates the smallest differentiation in comparison with the average weekly energy consumption. Then, the raw data of this week is chosen to make load profile calculation.

Evaporation profile and cooling capacity calculation

The evaporation temperature data doesn't have in the raw data, but it has the data of low pressure side. So evaporating temperature profile is defined as the saturation temperature of the low pressure side.

The cooling capacity is calculated by energy balance through the evaporator and if ambient temperature is over 20^{o}C, the cooling profile is changing to ambient temperature at 1%/K which references to cooling capacity changing to ambient temperature of Älta ice rink report (Rogstam2012).

$$Q_{cooling} = m * (h_{evap.out} - h_{evap.in})$$

where

m: Refrigerant mass flow rate

$Q_{cooling}$: Cooling capacity at the evaporator

$h_{evap.out}$: Enthalpy after the evaporator

$h_{evap.in}$: Enthalpy before compressor

The reference mass flow is calculated to the method of Mazyar (Mazyar2011)

$$m = \frac{\eta_{el}.P_{el} - Q_{compressorloss}}{h_{comp.out} - h_{comp.in}}$$

where

η_{el}: Electric motor efficiency, is assumed 90%

P_{el}: Electric power to the compressor motors

$Q_{compressorloss}$: Heat loss from compressor body and/or compressor cooling by oil/water

$h_{comp.out}$: Enthalpy after compressor

$h_{comp.in}$: Enthalpy before compressor

The calculation and the assumptions of cooling capacity are based on the thesis of Mazyar Karampour (2011). The heat loss from compressor body is assumed 7% of the electric power to the compressor motors and the other 3% is assumed for the cooling power, so the $Q_{compressorloss}$ is 10% of the electric power after the compressor motors.

Figure 27 presents the cooling profile and evaporating temperature for one week of Älta ice rink, this profile is applied for load profile of one operation period, September 2011-April 2012.

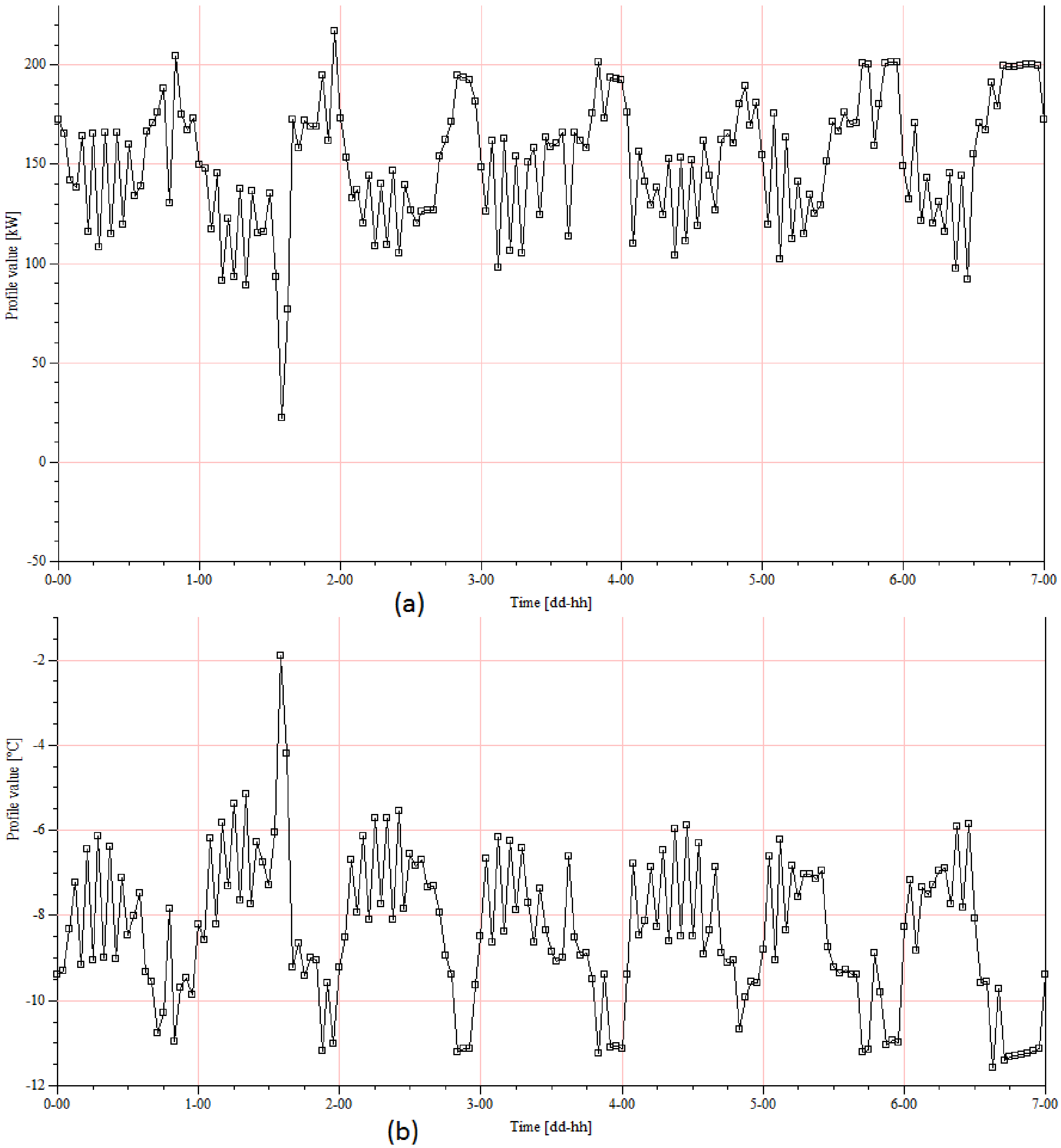

Figure 27: Cooling capacity profile (a) and evaporation temperature profile (b) of one week in Älta

6.2 Simulation

6.2.1 System Description

System N1 - reference system

N1 has the same specifications to Älta refrigeration system with the load profile to figure 27, so the simulation has following specifications and assumptions:

- NH3 in the primary cycle, brine in the second cycle
- Evaporation temperature/Condensing Temperature: Fluctuate to load profile/Constant at 25^{o}C
- Flooded evaporator
- Two reciprocating compressors 50Hz with external drive control, electric motor efficiency 92%
- Subcooling 5K
- Ammonia is forced circulation in the first cycle
- No internal heat exchanger
- Condenser capacity 400kW
- Indirect dry cool condenser and constant condensing temperature at 12^{o}C
- Fan power 12kW and coolant power 11kW
- Speed controlled fans and pumps
- Pump power in the secondary circuit 15kW

System N2 - Indirect NH3 system with floating condensing temperature

This system is same as the reference system, however, because this system is used as a typical NH3/brine system to compare with CO2 systems, some assumptions are modified to be able to equivalent to CO2 systems.

- Condensing temperature is fluctuated to the ambient temperature with Tc = Tamb + 10K
- Subcooling 5K
- Minimum condensing temperature 10^{0}C

System C2 - Indirect CO2 system

This system is also similar to N2, but CO_2 is used as the primary refrigerant and brine in the secondary cycle. An air cooled condenser is designed for this system and the difference between condensing temperature and ambient temperature is 7^{o}C that is lower about 3^{o}C than N2 system due to direct condensing. Design of the system is described in 5.1 chapter.

- Evaporation temperature -15^{o}C, transcritical operation at 95bar
- 6 reciprocating CD 3500M Dorin compressors, external drive control
- One small pump 0.1kW is used in the first cycle to circulate CO2
- Internal heat exchanger with effectiveness assumed 50%
- Air cooled condensing mode and fluctuated condensing temperature, Tc = Tamb + 7K.
- Minimum condensing temperature 10^{0}C
- Fan power 15kW
- Subcooling 5K
- Speed controlled fans

System C3 - Full CO_2 direct system

The system design is described in chapter 5.1, because CO2 is used for the whole system, so some assumptions are modified in this simulation:

- The pump power in the secondary cycle is assumed to be 3kW which is similar to size of the pump used in Backvallen ice rink, Katrineholm (Rogstam2007).
- This is a direct system, no heat exchanger between the evaporator and the secondary cycle, so CO2 evaporation temperature profile is same as brine temperature profile in secondary cycle. An average of input and output brine temperature is calculated from measured data of Älta, then this temperature profile is applied as evaporation temperature profile for this simulation.

6.2.2 Simulation

All of the system simulation uses same load profile and evaporating profile of Älta calculated in part 6.1. System N1 is simulated for 11 months of the year except July and without heat recovery system. This simulation result is used to compare with the real power consumption data in Älta during period from April 2011 to April 2012. Three systems N2, C2, C3 are simulated without heat recovery system for 8 months period, except May, June, July and August. It is known as typical period for of ice rinks in Sweden.

6.2.3 Results

Reference case result

Figure 28 shows energy consumption of the simulation and measured data from Älta ice rink. Compressor power result from the simulation is lower than the measured data of 18%, pumps and fans power in the simulation is also lower than measured data of 2%. It can be explained that auxiliary power measurement in Älta takes into account the pumps of heat recovery system. Regarding the compressor, efficiency of current compressors in Älta should be lower than the assumed efficiency for compressors in the simulation, so compressor power in Älta is higher than the simulation's result.

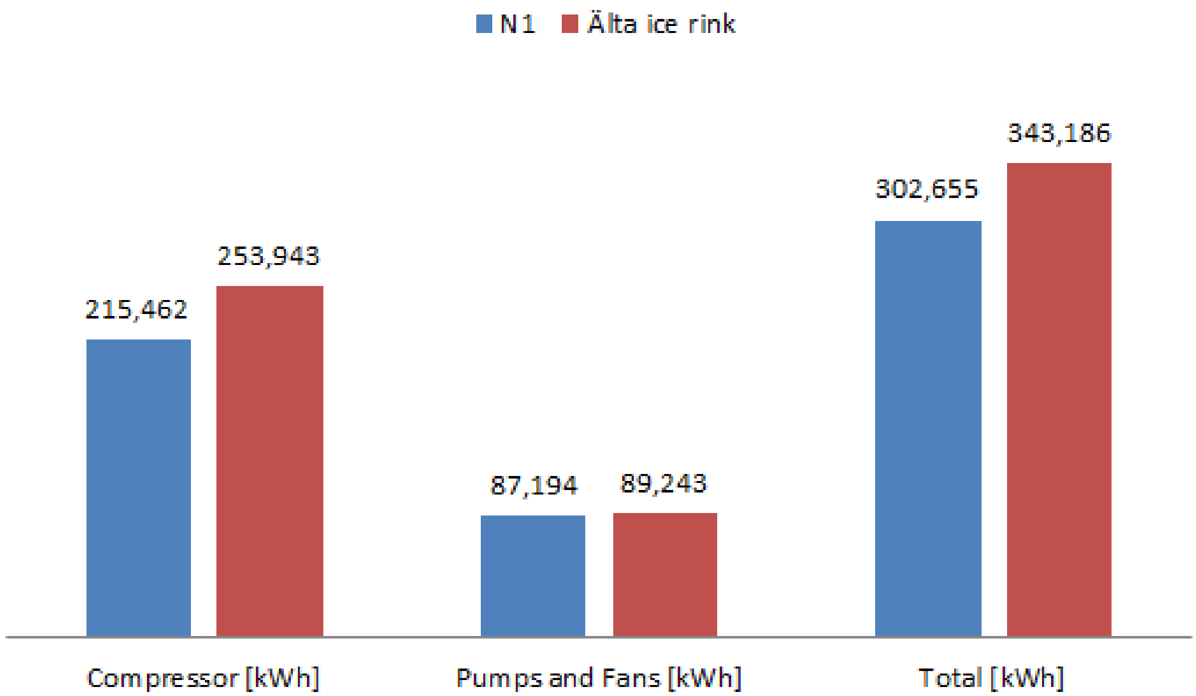

Figure 28: Energy consumption comparison of N1 and Älta ice rink

Three systems N2, C2, C3 without heat recovery system

Figure 29 shows the energy consumption of three systems N2, C2, C3 without heat recovery function. The pumps and fans power of N2 is the highest in three systems because of heat loss in indirect condensing and coolant pump and C3 has lowest pump and fan power. Pump power in the secondary cycle is similar between N2 and C2, however, pump power in the secondary cycle of C3 is 80% lower than C2 and N2 (3kW to C3 simulation and 15kW to N2 and C2 simulation respectively), it explains for the low pump power consumption of C3. Compressor power of N2 is lowest in three systems, it is 26% lower than C2 and 12% lower than C3. C3 compressor power is lower than the one of C2 due to lower evaporation temperature profile for C3.

Finally, C3 presents as the best performance in terms of COP2 and energy consumption. Average COP of C3 is 6.4 that is 24% higher than N2 and 30% higher than C2. Total power consumption of C3 is lower around 30% than N2, and 46% than C2.

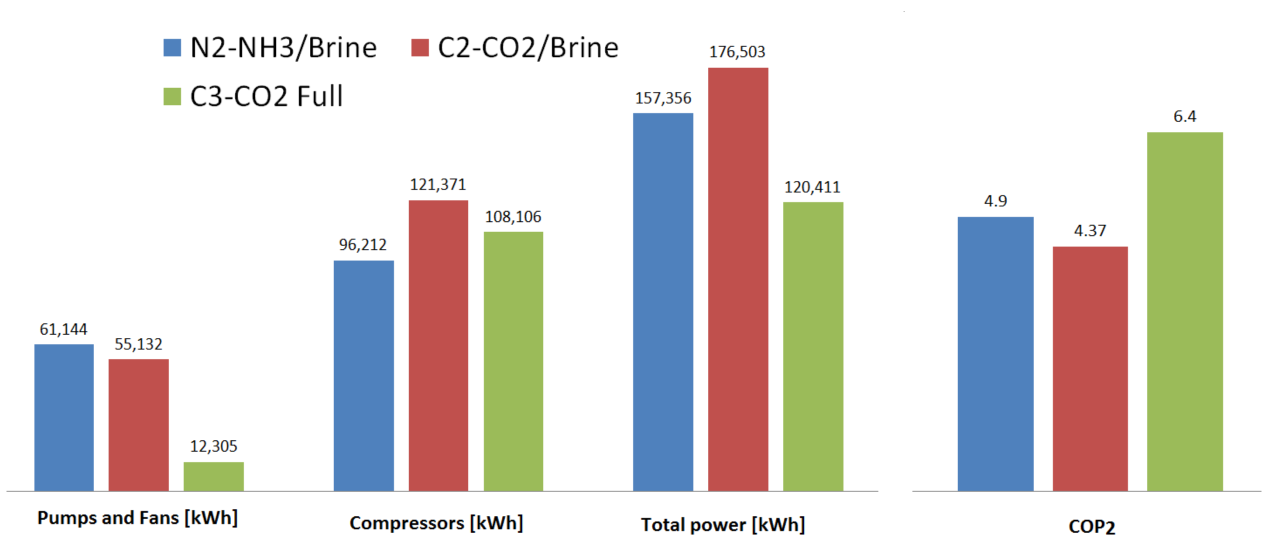

Figure 29: Three systems energy consumption without heat recovery

Three calculations are made by data exported from Pack Calculation II with the aim to investigate the software's results. Three calculations are named as energy balance checking, secondary effect checking and optimum discharge pressure checking for CO2 systems when they are running with heat recovery function.

Energy balance checking

Losses and efficiency of the compressors are investigated by energy balance method. Energy balance for three systems is calculated to following equations:

$Q_e + E_{compressor} = Q_c + Q_{loss}$

where

Q_e: cooling capacity

$E_{compressor}$: Total compressor power consumption

Q_c: condenser capacity

Q_{loss}: Total loss of compressors

Then, fraction of total loss and total compressor power is percentage of compressor loss, as follow:

$Percentage of compressorloss = \frac{Q_{loss}}{E_{compressor}}$

Figure 30 shows the calculation result of the percentage of heat loss and isentropic efficiency of the compressors.

Isentropic efficiency of CO2 systems is quite low, 65% in comparison with normal isentropic efficiency of CO2 compressors. In the thesis of Cottineau (2011), the total efficiency of CO2 compressors in supermarket refrigeration system are measured. It shows that the total efficiency of CO2 compressors for medium temperature stage is around 60%-70% which includes heat losses, motor efficiency and isentropic efficiency. Therefore, the isentropic efficiency of CO2 compressors should be higher than 65% if the total efficiency is around 60%. With this checking, it can be concluded that the compressor power of CO2 simulation C2, C3 can be lower in reality.

Regarding to the total loss of compressor, it can be seen that the percentage of compressor loss of N2 is only 1% much lower than C2 and C3 compressor losses 5%. In the software, motor electrical loss of N2 is assumed to be 92% of the total electric consumption ($E_{compressor}$), so the total compressor loss should be

and this 8% loss is assumed to be absorbed by the refrigerant because it . This amount of 8% loss in the refrigerant will go to condenser, so condenser load, Qc of three systems will be higher than the "real" condenser load. Consequently, a higher amount of heat can be recovered after compressor as heat recovery in case the data from software are used for heat recovery calculation.

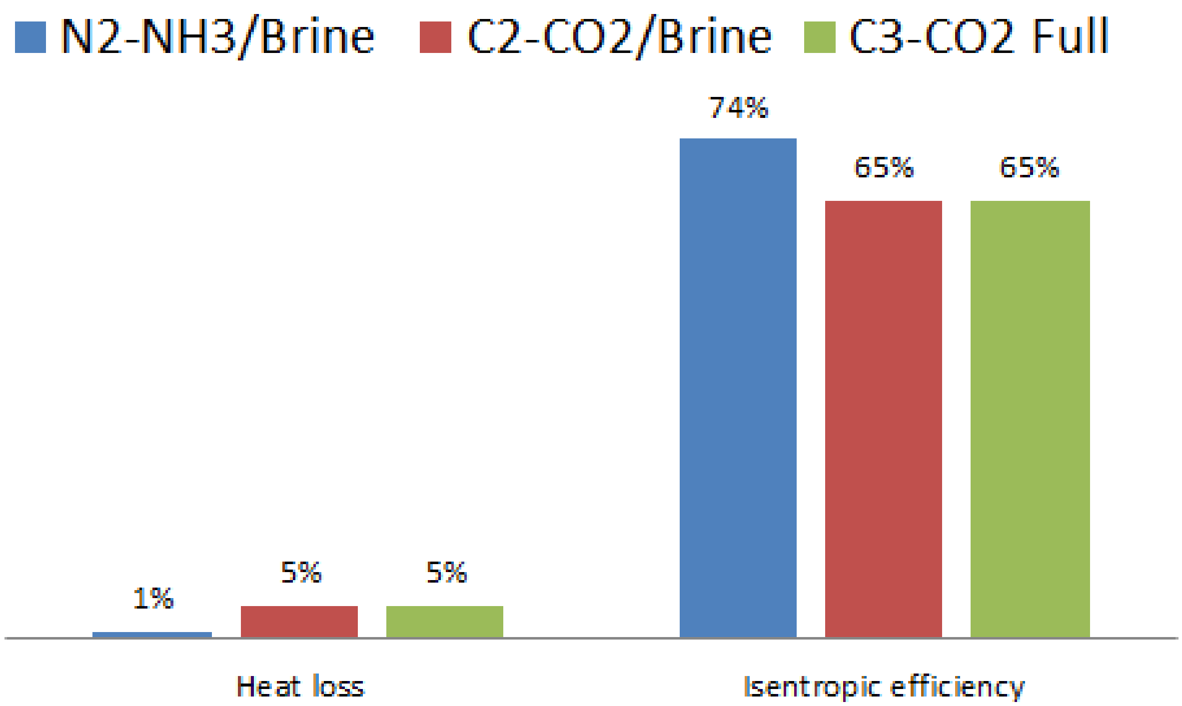

Figure 30: Heat losses and isentropic efficiency of compressors

Optimum pressure control checking

CO_2 systems running with heat recovery function in Pack Calculation II are examined to investigate if the systems are running at optimum COP when meeting the heating demand. Three systems are running with heat recovery function in the software with assumptions:

- Refrigerant temperature out of recovery heat exchanger is constant at 35^0C
- Start heat recovery when ambient temperature at 12^0C
- At start, keep high pressure at 22^0C
- Maximum heat recovery when ambient temperature at -5^0C
- High pressure at maximum heat recovery 40^0C

The results of three systems running with heat recovery function in the software shows that compressor power and condenser load are changing when the heat recovery function is attached in the simulation. It can be explained that the system is running to meet both cooling capacity and heating demand.

Regarding CO_2 systems, Sawalha (2008) presents a method to define optimum discharge pressures that the systems can operate at optimum COP to meet the heating demand. The optimum discharge pressure for system running with heat recovery function is calculated by equation to Sawalha (2008):

$$P_{optimum} = 2.7 \times T_{refrigerantoutofdesuperheater} - 6.1 \tag{2}$$

where

$P_{optimum}$: optimum discharge pressure

$T_{refrigerantoutofdesuperheater}$: refrigerant temperature out of desuperheater, is assumed 35^o C

The yearly average discharge pressure of C2 to simulation data is also calculated to evaluate if C2 system operates at the optimum discharge pressure when the system is running to meet both cooling capacity and heating demand.

The calculation results show that yearly average discharge pressure of C2 is 51.3bars while yearly average optimum discharge pressure calculated to equation 2 is 88.4bars. So, the discharge pressure in the software is lower than the optimum discharge pressure. It can be concluded that the C2 system with heat recovery function is not controlled in order to run at optimum discharge pressure in the software.

Secondary effect checking

In ice rink refrigeration system, the pump in the distribution system or it calls secondary pump that also makes an effect to the compressor power consumption. Heating from the secondary pump that spreads to the around environment creates a heat loss in the fluid, then it requires more cooling load that makes compressor consume more power. The heat loss gets higher if the secondary pump power is higher and it calls secondary effect in this part.

A trial is made in the software, for the same simulation with different secondary pump power to investigate whether the compressor power consumes more energy in case of higher secondary pump power. The results shows that there is no change in compressor power consumption if the secondary pump power is changing.

In conclusion, compressor power in three systems should be higher than simulation's results when secondary effect is considered and C3 system with lower secondary pump power can get lower compressor power consumption in comparison with N2 and C2.

7 Heat recovery potential

7.1 Heating demand

In section 3, it specifies all of the heating demand for an ice rink. In the reference case of Älta ice rink, heating for ventilation, dehumidification system and ice resurfacing is supplied by heat recovery system; other heating demand, like hot water and lock room heating are supplied by district heating. Table 3 shows only the heating supplied by district heating in Älta, data of total heating demand in Älta is not measured. In order to get the data of total heating demand, another ice rink, Järfälla that does not have heat recovery system and has similar capacity with Älta is chosen. All of the heating demand in Järfälla is supplied by district heating shown in table 3.

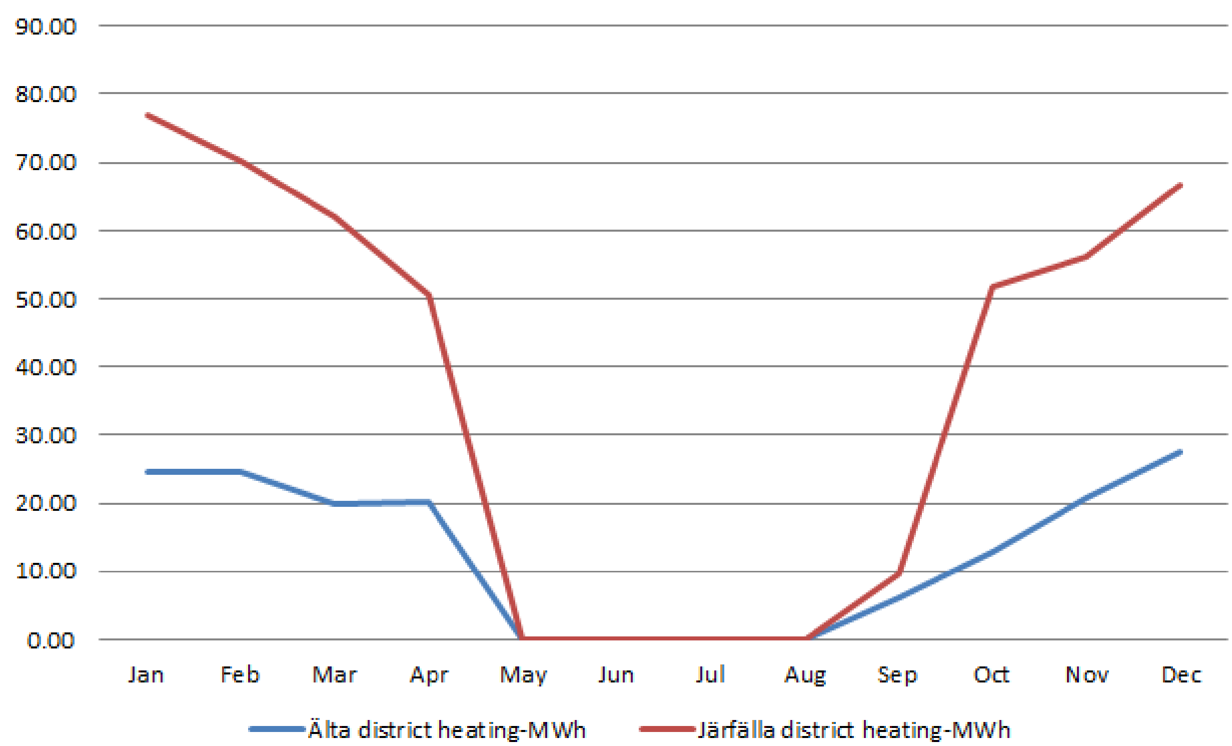

Figure 31: Heating demand in Järfälla and heat recovery in Älta

	Älta district heating [MWh]	Järfälla district heating [MWh]
January	24.53	76.90
February	24.52	70.17
March	19.96	61.87
April	20.26	50.70
September	6.23	9.53
October	12.82	51.76
November	20.65	56.16
December	27.42	66.70
TOTAL	156.38	443.79

Table 3: Heating demand in Älta and Järfälla in the period of September 2011-April 2012

7.2 Heat recovery potential

Heat recovery potential of three systems N2, C2, C3 are discussed in this part. Three different strategies for heat recovery are presented, only heat recovery from desuperheater is calculated and exit temperature from desuperheater is assumed 35^0C. Energy saving in each case is calculated by the quantity of heat recovered and the quantity of energy consumption increase in comparison with the systems without heat recovery.

The first scenario is that heat recovery is calculated in case the systems are operating only to meet cooling capacity, the second scenario is that the systems are operating to generate more heating to ambient temperature, finally, the C2, C3 systems are operating to meet heating demand and achieve highest COP by controlling discharge pressures.

7.2.1 Heat recovery without changing the systems

In this solution, data from three simulations N2, C2, C3 without heat recovery function are exported from Pack Calculation II and heat recovery quantity-$Q_{\text{heatrecovery}}$ is calculated to following equation:

$Q_{heatrecovery} = m \cdot (h_{thotgas,pdischarge} - h_{35^0C,pdischarge})$

where

m: Refrigerant mass flow rate

$Q_{heatrecovery}$: heat recovery

$h_{thotgas,pdischarge}$: Enthalpy of discharge gas after compressor

$h_{35^0C,pdischarge}$: Enthalpy of discharge gas at 3

Figure 32 shows that C2 gives the highest amount of heat recovery, then N2 is the second and C3 gives the lowest heat recovery potential. With this strategy, the compressor power of the systems do not change to generate more heat, so energy consumption does not change. Under this scenario, energy saving is the heat that can be recovered, and this heat recovery is taken in account in the COP to following equation:

$$COP = \frac{(Qe+HR)}{Totalenergyconsumption}$$

Where:

Qe: cooling capacity

HR: heat recovery

It can be seen that C3 still has highest COP, the second is N2 and C2's COP is the lowest one which is similar to the case of system without heat recovery presented in part 6.2.3.

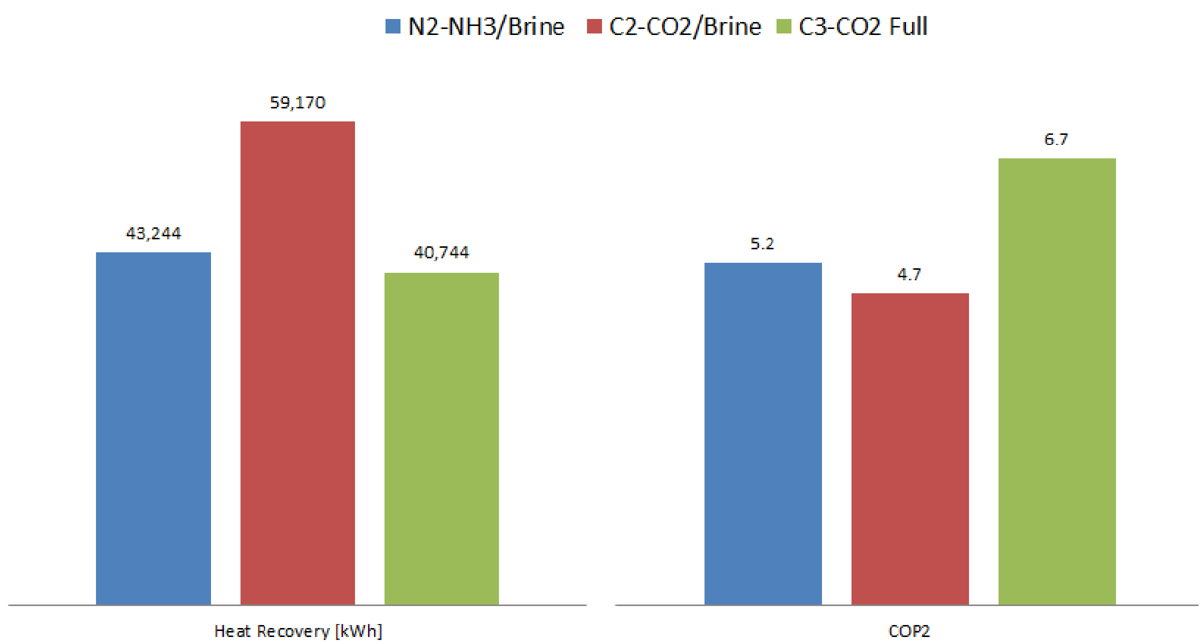

Figure 32: Heat recovery without changing the systems

7.2.2 Systems with heat recovery to ambient temperature

Heat recovery function in the software is attached in three simulations N2, C2, C3 with the similar assumptions in chapter 6.2.3.

Figure 33 shows results of three systems running with heat recovery function to ambient temperature by the software Pack Calculation II. In the results, C2 presents the highest heat recovery potential in three systems, the second is C3, and N2 shows the lowest result. C2 can recover nearly 14% more heat than N2 and C3 has 11% more heat than N2. It also shows that compressor power is nearly as twice as the systems without heat recovery function, however, pumps and fans power does not decrease as much, so N2 total power consumption increases to 52%, C2 has 56% higher and C3 has 81% higher in comparison with systems without heat recovery. COP of three systems is included heat recovery also, and C3 still has highest COP, then N2 and C2 is the lowest one.

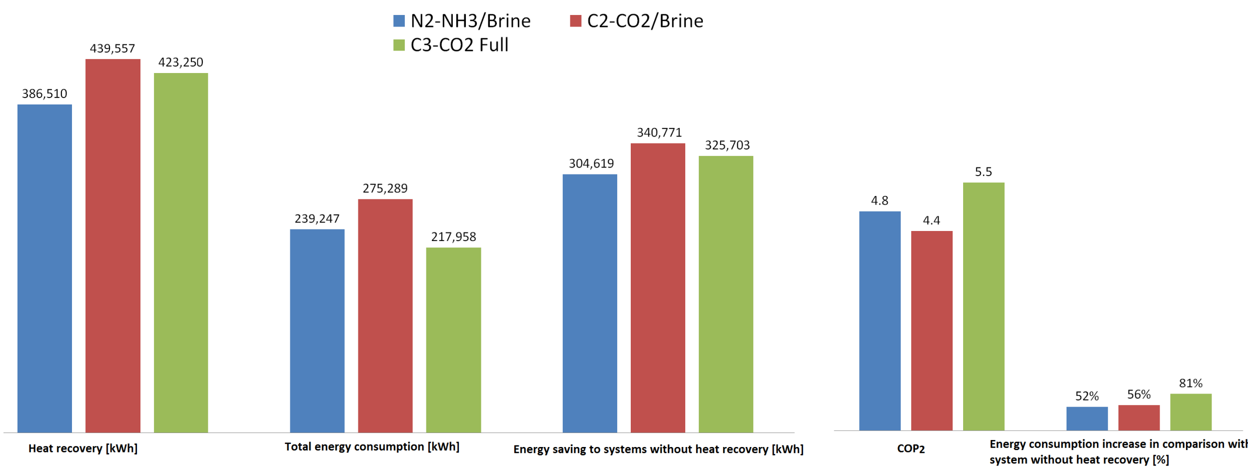

Figure 33: Three systems energy consumption with heat recovery

7.2.3 Heat recovery with optimum discharge pressure control for CO2 systems

In this case, the refrigeration system is operating to meet the heating demand specified in part 3. The systems are simulated to Sawalha's strategy (2012), in which the systems are controlled to meet the heating demand and operated at optimum discharge pressure. If the discharge pressure is getting higher than the maximum discharge pressure to generate more heat, then the discharge pressure will be controlled at this maximum discharge pressure. The maximum discharge pressure is calculated to the equation 2.

The heating demand, cooling capacity, evaporating temperature, ambient temperature profile are monthly average data. Some assumptions are made in this part:

- Desuperheater exit temperature/return temperature: 35^{o} C/30^{o} C
- No subcooling is assumed
- Exit gas cooler temperature is 5K different to ambient temperature
- Heating demand data is in table 3

With this method, C2 and C3 can produce the same heating but C3 has higher COP than C2, however, C3 consumes more energy to produce this heat than C2. It can be explained that C3 compressor power is lower than C2 because of lower evaporation temperature, in order to meet heating demand, the systems consume more energy in the compressors.

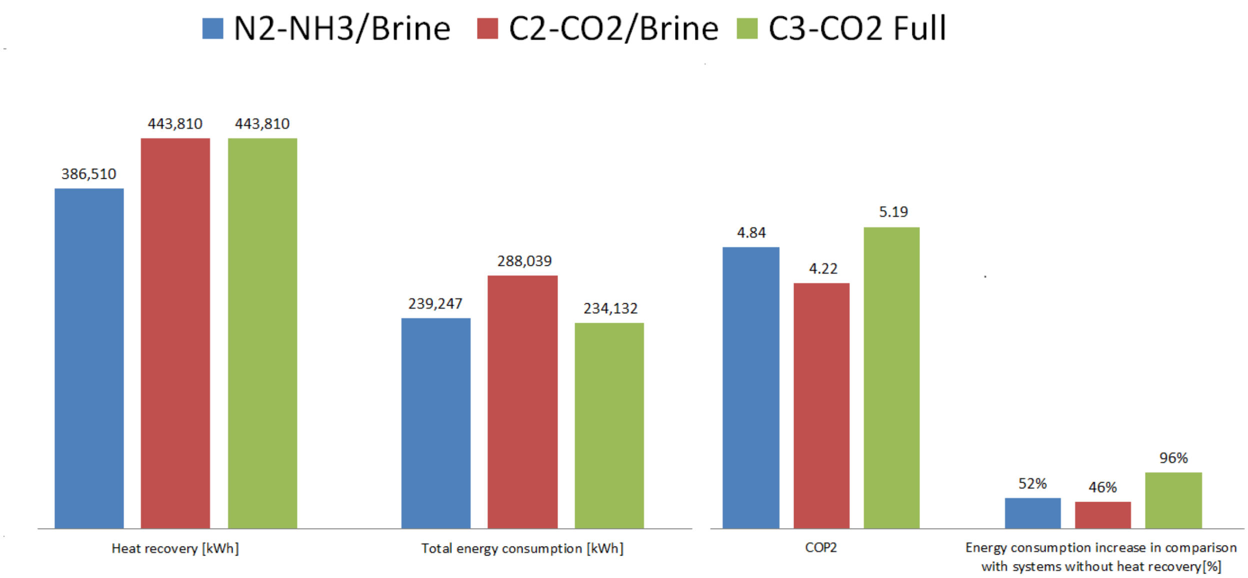

Figure 34: Three systems with heat recovery to optimum pressure control strategy

8 Life cycle cost

The simulation results of three systems without heat recovery function are used for the life cycle cost analysis. However, in the capital investment assumption, the cost of heat recovery system 50kW is included in the three systems. These three systems are assumed to have heat recovery equipment but the systems are not controlled to meet heating demand or generate higher heat quantity. Therefore, the energy consumption used in LCC cost calculation are the energy consumption result from the part "three systems without heat recovery system".

Some assumptions are made for the life cycle cost calculation:

- Life cycle 15 years
- Electricity price 1sek/kWh
- Electricity price increase rate 3%/year
- Energy consumption is assumed equal through the period
- Inflation rate 2%
- Discount rate 5%
- Installation cost is included in the capital cost
- Installation cost is included in the capital cost
- Capital cost is assumed 3,000,000kr for NH3/Brine system and CO2 Full system. CO2/Brine system is assumed 3,900,000kr
- Annual service cost is assumed
  - NH3/Brine: 70,000kr
  - CO2/Brine: 55,000kr
  - CO2 full: 50,000kr
- No repair cost or equipment replacement
- No remain cost after life cycle period

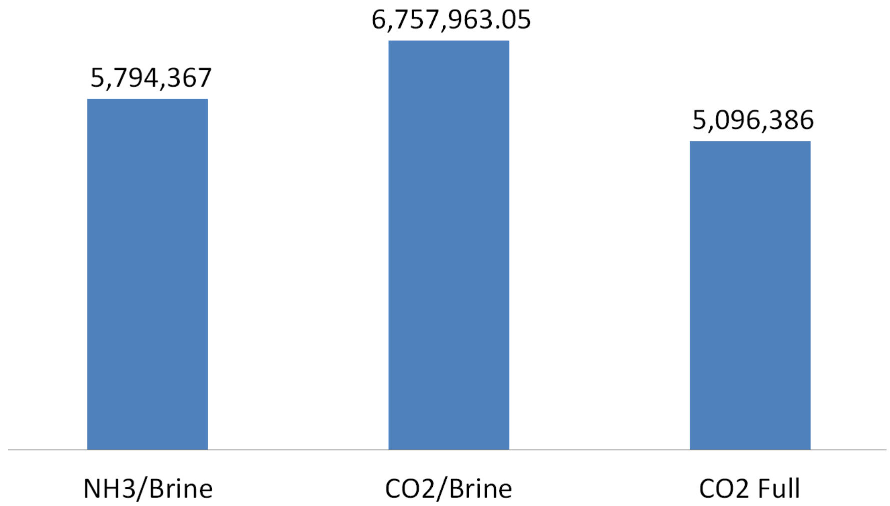

Figure 35: Life cycle cost of three systems

Figure 35 displays the life cycle cost of three systems. The CO2 full system has the lowest LCC in three systems due to the lower energy cost which benefits from lower power consumption, in addition, lower maintenance cost is also another advantage of CO2 full system. LCC of the traditional NH3/Brine system is higher than CO2 full system around 13% which results from higher energy cost and maintenance cost. Finally, CO2/Brine system generates a higher LCC because of the highest energy consumption in three systems.

9 Conclusions and suggestions for future work

9.1 Conclusions

CO_2 full system is proving as the most efficiency system in three systems. It has low energy consumption (30% lower than NH_3/Brine system) which benefits from direct condensing, no heat exchange between refrigerant-brine and higher evaporation profile. Heat recovery potential to CO_2 full system is also higher than NH_3/Brine system.
COP of CO_2 refrigeration system will be higher in the cold climate and lower in the hot climate because of transcritical operation of CO_2 cycle. These simulations are made in the cold climate of Sweden, furthermore the systems is only running in the cold season (September-April). As the result, these factors are becoming advantages for CO_2 refrigeration system in ice rink, and the energy consumption of this CO_2 refrigeration system will be much higher if it is simulated in warmer climate condition.
Regarding to the software, there are some assumptions in the software that can generate the better results of NH_3/Brine system in comparison with CO_2 systems. Lower heat loss of NH_3 compressors 1% in comparison with 5% of CO_2 compressors is a favor of the software to NH_3 compressors, secondly CO_2 compressors isentropic efficiency is lower than a normal CO_2 compressor isentropic efficiency measured in CO_2 supermarket refrigeration system.
The life cycle cost of three systems is affected dramatically by energy cost. If the energy cost is getting higher and higher in the next few years, the CO_2 full system would become a potential candidate for the next energy efficiency system generation. Another factor relating to economic performance of three systems is the initial capital cost. With the strong market development in the period 2005-2012 of CO_2 refrigeration system in supermarket, the mass production of CO_2 components promises lower component cost in the next years while NH_3 component market has been matured after more than 50 years of development, it is difficult for a downing in component cost of NH_3 system.
In conclusion, CO_2 ice rink refrigeration system is still in the first stage of development. Energy performance, investment cost and service cost needs to be measured for the first system installation in order to have a validate evaluation of CO_2 refrigeration system in ice rink. Lacking of experiences, skilled workers for CO_2 system can be an obstruct for the first application of this system in ice rink.

9.2 Suggestions for future works

Some suggestions for future studies can be applied in the next studies to have better understanding of CO2 application in ice rink refrigeration system:

- In these simulation, only one week cooling profile applied for the whole period, the results can be changed if the whole period load profile applied.
- Some assumptions in the software Pack Calculation II for compressors need to be modified to generate a higher accuracy result.
- Simulation for different climate conditions can be practiced to have different evaluation to CO2 systems.
- Maintenance and operation aspect, or total service cost of these systems can be investigated to have deeper view to the cost of each system.

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Appendix

A N1 System Description

	NH3
	One stage
Configuration	Reference system
Refrigerant [-]	R717
Design condition [-]	Custom, MBP (Te/Tc = -15.0 / 30.0 °C)
Capacity [kW]	322.7 / 416.3
Comp 1 [-]	Recip 4, R717, 50Hz
Comp 2 [-]	Recip 2, R717, 50Hz
Suction side	
Cooling capacity	
Profile [-]	cooling capacity-Oct
Dimensioning capacity [kW]	216.70
Tamb at dim [°C]	32.0
Profile change [-]	1.0
Profile const below Tamb [°C]	20.0
Flooded evaporators	
Forced circulation [-]	Yes
Nominal pump power [kW]	0.000
Pump power [-]	Proportional to capacity
Evaporation temperature	
Evaporation temp profile [-]	Teva-Oct
Temp for const profile [°C]	-1.9
Additional	
Internal hx eff. [-]	0.0
Discharge side	
Condenser type [-]	Dry cooler
Cond. cap. ctrl.	
Tc profile [-]	-
Temp for const profile [°C]	-
Const Tc [°C]	25.0
Tc = A*Tamb+DT [°C]	-
Fan with compressor [-]	-
Minimum Tc [°C]	20.0
Subcooling [K]	0.0
Speed ctrl. fans [-]	True
Speed ctrl. pump [-]	True

B N2, C2, C3 System description

	N2	C2	C3
Configuration			
Refrigerant	R717	R744	R744
Capacity	352.2/454.1	383.6/646.7	329.7/531.7
Compressor 1	Recip 4, R717, 50Hz	CD 3500 M, R744, 50Hz	CD 3500 M, R744, 50Hz
Compressor 2	Recip 3, R717, 50Hz	CD 3500 M, R744, 50Hz	CD 3500 M, R744, 50Hz
Compressor 3	-	CD 3500 M, R744, 50Hz	CD 3500 M, R744, 50Hz
Compressor 4	-	CD 3500 M, R744, 50Hz	CD 3500 M, R744, 50Hz
Compressor 5	-	CD 3500 M, R744, 50Hz	CD 3500 M, R744, 50Hz
Suction side			
Cooling capacity	Älta Ice rink profile	Älta Ice rink profile	Älta Ice rink profile
Profile change [%/K]	1	1	1
Tamb at change [^{o} C]	20	20	20

C EES Model for heat recovery system with optimum pressure control

```
"Input parameters and assumptions"
"run t2 as monthly average, run in table"
t_hr_r=30
"return temperature from heat recovery"
dt_app_desup=5
"approach temperature difference in de-superheater"
dt_app_gc=5
"approach temperature difference in gas cooler"
"t_desup_exit is the CO2 temp exiting the de-superheater"
"run t_amb as monthly average, run in table"
{P_disch=P_disch_max}
"overall compressor efficiency"
eta_tot=0.65
```

```
"Heat loss from the compressor as percentage of compressors power"
Q_hl_per=0.07
"Heat demand also the heat recovered from the de-superheater"
"run Q_hr as monthly average heating demand, run in table"
"run Q_2 as monthly average cooling capacity in table"
t_gc_exit=t_amb+dt_app_gc t_desup_exit=t_hr_r+dt_app_desup
"This is the maximum pressure that can be used for heat recovery"
P_disch_max=2.7*t_desup_exit-6
p2=P_sat(CarbonDioxide,T=T2)
h_ev_exit=Enthalpy(CarbonDioxide,T=T2,x=1)
s_ev_exit=Entropy(CarbonDioxide,T=T2,x=1)
h1k_is=Enthalpy(CarbonDioxide,s=s_ev_exit,P=P_disch)
"this is the isentropic efficiency"
eta_is=(h1k_is-h_ev_exit)/(h_comp_exit-h_ev_exit)
h_desup_exit=Enthalpy(CarbonDioxide,t=t_desup_exit,P=P_disch)
h_gc_exit=Enthalpy(CarbonDioxide,t=t_gc_exit,P=P_disch)
h_ev_in=h_gc_exit
Q_2=m_dot_r*(h_ev_exit-h_ev_in)
"Heat recovered in de-superheater"
Q_HR=m_dot_r*(h_comp_exit-h_desup_exit)
Q_gc=m_dot_r*(h_desup_exit-h_gc_exit)
E_el=m_dot_r*(h1k_is-h_ev_exit)/eta_tot
E_shaft=E_el*(1-Q_hl_per)
E_shaft=m_dot_r*(h_comp_exit-h_ev_exit)
COP2=Q_2/E_el
"This is for heat balance check: Q_balance should be close to zero"
{Q_gc=Q_2+E_shaft-Q_HR}
{Array Enthalpy}
Enthalpy[1..6]=[h_ev_exit,h_comp_exit,h_desup_exit,h_gc_exit,h_ev_in,h_ev_exit]
{Array Enthalpy end} {Array Pressure}
Pressure[1..6]=[P2,p_disch,P_disch,P_disch,P2,P2]
{Array Pressure end}
```