Techno-Economic Feasibility Study of an Urban Rooftop Ice Rink

Integrating with Rosenlundsverket District Energy in Central Gothenburg

Master's thesis in
Architecture and Civil Engineering

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Abstract

This master's thesis presents a comprehensive techno-economic feasibility study for implementing an innovative urban rooftop ice rink at Rosenlundsverket in Gothenburg, Sweden. The research addresses the growing need for sustainable recreational infrastructure by evaluating advanced energy system alternatives that integrate with existing district energy networks while minimizing environmental impact.

The study focuses on a 1,100 m^2 recreational ice rink positioned 20 meters above ground level, designed for year-round operation with a design effective cooling load capacity of 762.5 kW, excluding HVAC requirements due to the open roof proposal. The research emphasizes CO_2 transcritical refrigeration technology (R744) as the primary cooling system due to its superior environmental credentials and high efficiency in Nordic climates. Through detailed performance modeling using Danfoss Coolselector®2 professional software and ASPEN Plus® validation, combined with meteorological data from SMHI Säve station, six integration scenarios excluding the reference scenario were systematically evaluated over a 25-year lifecycle.

The methodology incorporates comprehensive cooling load calculations adapted for open-roof conditions, considering factors such as solar radiation, convective heat transfer, condensation effects, and wind exposure characteristic of rooftop installations. Advanced design features encompass integrated photovoltaic shading systems utilizing Midsummer BOLD flexible films that cover 65% of the roof area (958 m^2), wind barriers that achieve a 70% reduction in wind speed, and rainwater harvesting systems that can decrease municipal water consumption by 60-80%. Findings indicate significant economic and technical viability across all assessed scenarios, with notable outcomes from hybrid system configurations. The COMBO Scenario with Heat Recovery demonstrates significant leadership in transformation, achieving a net present value enhancement of +40.4 million SEK, annual savings of 41.5 million SEK, and an exceptional return on investment of 3,576%. The COMBO Scenario (nominal) demonstrates significant second-place performance, yielding an improvement of +30.3 million SEK in NPV and an ROI of 5,411%. The River Cooling integration yields an NPV improvement of +22.8 million SEK without necessitating further capital investment, whereas the Cascade District Cooling scenario presents an NPV improvement of +22.7 million SEK with an ROI of 4,042%. The Combo+ DHW scenario demonstrates significant technical performance, achieving an overall normalized efficiency gain of +88%.

Additionally, the Combined DH exhibits a +53.4% improvement in overall normalized efficiency through advanced waste heat recovery, capturing up to 268 kW of thermal energy for district heating applications. Environmental benefits encompass annual energy consumption reductions of 12-18% relative to traditional ice rink designs, with the optimized COMBO scenarios attaining 56-64% of the Swedish national average consumption (1,137 MWh annually). Reductions in greenhouse gas emissions of up to 5.8 tonnes CO2 per year compared to the reference are achieved through the mitigation of off-site emissions from electrical generation and the adoption of natural refrigerants. Additionally, enhanced urban sustainability is realized through the utilization of waste heat in district energy networks, resulting in carbon footprint reductions of 26-37% relative to reference systems. The integrated photovoltaic system, with an 11% capacity, facilitates renewable energy generation, while the rainwater harvesting system enhances sustainable water management practices. The study demonstrates that sustainable urban recreational infrastructure can significantly enhance economic performance while benefiting district energy systems. The COMBO scenarios exhibit significant potential, with the heat recovery configuration achieving a 72% reduction in total costs relative to conventional methods, thereby establishing a clear technological advantage in ice rink energy systems. The findings offer a solid technical basis and economic rationale for stakeholders to adopt this innovative rooftop ice rink concept, creating a replicable model for analogous urban development initiatives in Nordic climates. This study provides important insights into sustainable building design, district energy integration, and urban planning practices, illustrating how advanced engineering can yield synergistic benefits for facility operators and urban energy networks while achieving carbon-negative operations and significant economic returns.

Keywords: Ice rink, CO_2 refrigeration, district energy integration, hybrid systems, techno-economic analysis, Open-roof Ice Rink

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Abbreviation List

Abbreviation	Definition
AI	Artificial Intelligence
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BTUS	British Thermal Units per Second
CO_2	Carbon Dioxide (R744)
COP	Coefficient of Performance
COSP	Coefficient of System Performance
CapEx	Capital Expenditure
DC	District Cooling
DH	District Heating
ETFE	Ethylene Tetrafluoroethylene
GC	Gas Cooler
GWP	Global Warming Potential
HX	Heat Exchanger
IIHF	International Ice Hockey Federation
IRR	Internal Rate of Return
KTH	Royal Institute of Technology (Kungliga Tekniska högskolan)
LCC	Life Cycle Cost
LMTD	Log Mean Temperature Difference
LP	Low Pressure
LT	Low Temperature
LiBr	Lithium Bromide
MP	Medium Pressure
MSEK	Million Swedish Kronor
MT	Medium Temperature
MW	Megawatt
MWh	Megawatt-hour
NH_3	Ammonia (R717)
NPV	Net Present Value

NTU	Nephelometric Turbidity Unit
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
PV	Photovoltaic
R134a	Refrigerant 134a
R717	Refrigerant Ammonia
R744	Refrigerant CO_2
ROI	Return on Investment
SEER	Seasonal Energy Efficiency Ratio
SMHI	Swedish Meteorological and Hydrological Institute
SPF	Seasonal Performance Factor
UV	Ultraviolet
WHR	Waste Heat Recovery
kW	Kilowatt
kWh	Kilowatt-hour
°C	Degrees Celsius

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1 Introduction

The integration of energy-efficient technologies into urban infrastructure is essential for achieving sustainability goals. Ice rinks are among the most energy-intensive public facilities, with refrigeration systems accounting for approximately 43% of total energy consumption, in Nordic climates like Gothenburg's, where year-round ice rink operation is common. Optimizing refrigeration systems is crucial for reducing environmental impact and operational costs (ARENAS, 2023).

This study focuses on evaluating system alternatives for a proposed ice rink on the rooftop of Rosenlundsverket, a central facility in Gothenburg, Sweden. The goal is to develop an innovative and sustainable energy system design that aligns with the city's environmental, technical, and economic objectives.

1.1 Scope of the study

This study is strictly focused on energy system alternatives for the proposed ice rink and does not include structural or construction-related aspects of the building. It is assumed that the rooftop placement of the ice rink is feasible. The specific scope includes:

1. **Ice Rink System Alternatives:** The study evaluates refrigeration system options for the ice rink, emphasizing their technical feasibility, environmental impact, and economic performance in integration with Rosenlundsverket's district energy systems.

2. **Exclusion of Building Construction:** The structural assessment and construction-related aspects of the ice rink are outside the study's scope, except for proposed solar shading over the Ice pad, as well as wind barriers, which have a significant effect on overall energy consumption from a systematic approach. It is assumed that the necessary requirements for the implementation of a rooftop Ice rink have already been met from a structural perspective according to the Swedish building codes.

3. **Exclusion of HVAC and Side Amenities while Focus on Refrigeration Technology:** The study prioritizes the CO_2-based refrigeration system due to its high efficiency and low global warming potential (GWP). Additionally, two alternative technologies will be evaluated for comparative analysis:

   - Absorption chillers, which utilize low-cost waste heat but have efficiency limitations in colder periods.
   - Compressor chillers use ammonia, which has a long history in ice rink applications but comes with safety and environmental considerations.

4. **Energy Integration with Rosenlundsverket:** The study examines how the proposed reference refrigeration system can integrate with existing district heating, cooling, and river water systems at Rosenlundsverket. While this research acknowledges the technical design specifications at a basic level, the main focus is on estimating energy requirements, power demand, and operational duration for the different scenarios in order to obtain the data needed for the economic assessment, followed by LCC analysis.

1.2 Research Questions

1. Which refrigeration technologies among (CO_2-based, ammonia-based, or absorption systems) are most suitable for rooftop ice rink applications in Nordic climates, in terms of energy efficiency, environmental impact, and operational feasibility?
2. How can the selected refrigeration systems be effectively integrated with Rosenlundsverket's existing district heating, cooling, and river water systems to maximize energy recovery and offset the operation cost of energy?
3. What technical and environmental challenges may arise from implementing the proposed integration scenarios, considering the rooftop design of the ice rink? How would it be possible to overcome these challenges through well-designed and established integration strategies?
4. To what extent can innovative features, such as heat recovery, solar shading, rainwater harvesting, or an Auxiliary ORC unit, have the potential to enhance the overall performance and sustainability of the ice rink system?
5. What are the projected energy savings, environmental and economic benefits (e.g., payback period, NPV, ROI) associated with each proposed scenario under seasonal operating conditions in Gothenburg?

1.3 Objectives

The overarching objective of this study is to develop a technically and economically viable energy system design for the ice rink, while leveraging Rosenlundsverket's existing infrastructure. The specific objectives are:

1. Identify and compare three refrigeration system alternatives CO_2-based systems, absorption chillers, and ammonia-based compressor chillers, based on technical performance, environmental impact, and cost-effectiveness.

2. Select the most viable reference refrigeration system for the ICE Rink application and outline the most feasible integration Scenario among several alternatives based on the compatibility level to the selected reference System.

3. Assess integration strategies considering the potential infrastructure located at Rosenlundsverket district energy substation.

4. Conduct the performance benchmarking in terms of either the potential energy saving or recovery over the nominated scenarios from technical perspectives. Estimate energy consumption, power demand, operational feasibility, and potential environmental credits under different seasonal patterns.

5. Conduct LCC analysis to assess the economic assessment based on the given data obtained by energy saving projection to find the most attractive scenario and build up the NPV sensitivity Tornado 0 as well as the corresponds scenario decision matrix 5.4.5.

1.4 Context and Relevance

Rosenlundsverket presents a unique opportunity to develop a sustainable energy system for an urban ice rink due to its oversized district energy infrastructure. According to initial assessments from Göteborg Energi and discussions with Anders Strand, Rosenlundsverket has sufficient capacity for:

- Electricity supply
- District heating (potential for condenser cooling with temperature lift of 50–75°C)
- District cooling (evaporator heating with a temperature drop from 15°C to 6–2°C)
- River water utilization (6,000 m^3/h available for condenser cooling)

The available energy sources facilitate an energy-efficient integration of refrigeration technologies, thereby decreasing dependence on standalone cooling systems. This study prioritizes CO_2-based refrigeration and utilizes waste heat recovery through integration with the district heating network at the enclosure. It aims to demonstrate how an urban ice rink can support Gothenburg's sustainability goals by optimizing energy consumption while ensuring economic viability.

1.5 Methodological Approach

To achieve the study's objectives, the research follows a structured approach that includes:

- Literature Review: Analysis of modern ice rink refrigeration technologies and district energy systems.
- Site Analysis: Examination of Rosenlundsverket rooftop characteristics and available energy resources.
- Conceptual Design: Development of alternative refrigeration system configurations.
- Energy Modeling & Simulation: Estimation of energy use, power demand, operational duration, and CO_2 emissions reductions under various seasonal conditions.
- Economic Analysis: Comparison of capital costs, operational expenses, and potential revenues for each refrigeration system alternative.

2 Literature Review

2.1 Overview of modern ice rink systems

Conventional ice rinks are ranked as highly energy-intensive end-users, requiring advanced subsystems to preserve high-quality ice surfaces while optimizing energy efficiency and minimizing their environmental footprint. The refrigeration system stand-alone, which is responsible for maintaining the ice surface at the desired temperature, accounts for the largest portion of energy consumption in an ice rink, approximately 43% of total energy use (Shirvani and Youssef, 2018). Addressing the substantial energy demands needed to maintain consistent ice quality involves prioritizing cutting-edge refrigeration technologies and exploiting synergies with district energy networks.

In Nordic climates, specifically Sweden, the average age of active ice arenas exceeds 30 years, with the majority built between 1980 and 1989. This indicates that the current infrastructure, which typically operates year-round, is considerably outdated and requires modernization. Sweden has recognized the need to upgrade energy-intensive facilities as a strategic approach to improve energy efficiency and reduce environmental impacts. The incorporation of advanced ice rinks with dynamic response subsystems is essential in the development of sustainable facilities (Makhnatch, 2011). Currently, ice rinks function mainly as facilities for multiple sports, such as ice hockey, curling, figure skating, and recreational skating. The dimensions of ice surfaces vary considerably, from compact 60 m^2 areas intended for figure skating practice to expansive public arenas that accommodate ice sports events. Furthermore, bobsleigh tracks meeting Olympic standards necessitate refrigerated ice surfaces that may cover an area of up to 10,000 m^2 (Heating et al., 1994).

2.2 Energy consumption in ice rinks, with a focus on Nordic climates

Currently, Sweden operates approximately 365 ice rinks, with an average annual energy consumption of around 1,000 MWh (including both electricity and heat). The average total purchased energy in Swedish ice rinks is 1091 MWh/year (Pomerancevs et al., 2020).

According to the International Ice Hockey Federation (ARENAS, 2023), the five primary technical subsystems—lighting, heating, refrigeration, and ventilation are also the largest energy consumers, as depicted in the diagram of a typical ice rink in Figure 1. These subsystems collectively account for 95% of total energy consumption.

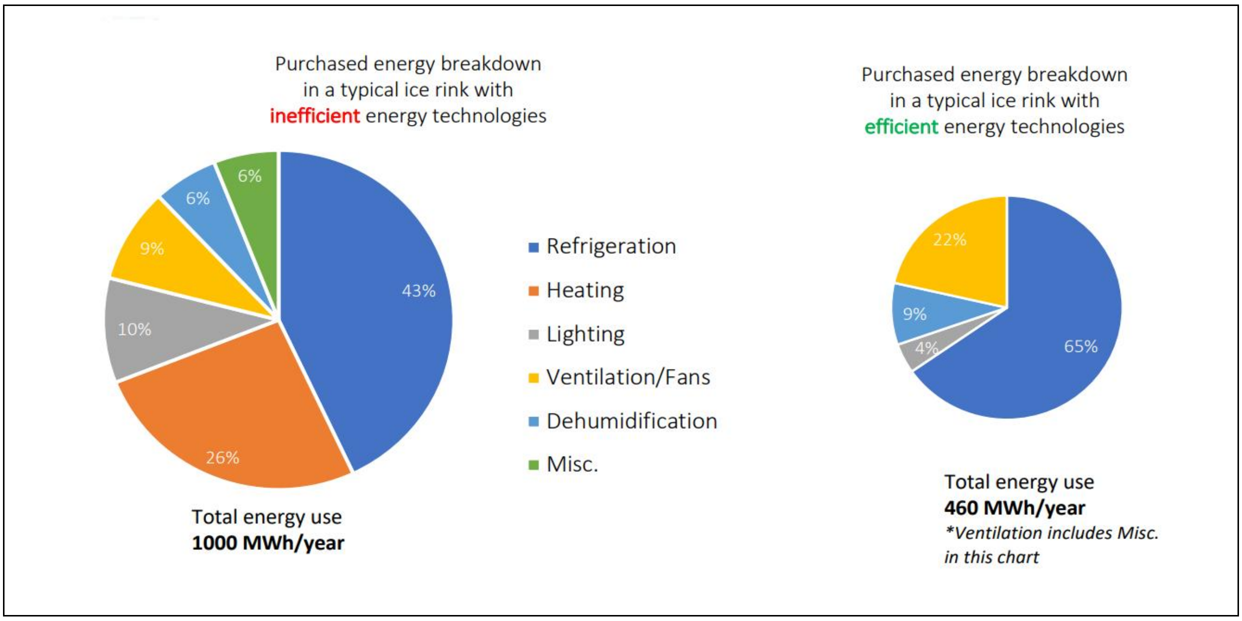


Figure 1: Energy distribution in a Typical vs a Modern Ice rink (ARENAS, 2023)

Ice arenas are categorized as energy-intensive commercial facilities, where the energy use of corresponding refrigeration subsystems, standing alone, is dominantly the largest share of energy consumption, up to 43% (430 MWh) of this total (Hemati, 2023). This significant energy demand stems from the need to maintain ice surfaces at low temperatures year-round, even during colder months typical of Nordic climates.

2.3 Share of the absorption heat gain in the Ice Rink

As mentioned earlier, the cooling system is the largest energy consumer in ice rinks, accounting for approximately 43% of total energy use in a typical facility. (Hemati, 2023). This significant energy demand is primarily due to the need to maintain the ice surface at low temperatures year-round, even during warmer months. (Karampour, 2011).

Various heat sources within the ice rink also influence the cooling load. The major heat loads include (Karampour, 2011):

1. Convection and condensation (30-36%)
2. Radiation (15-20%)
3. Ice resurfacing (12-15%)
4. Lighting (5-7%)
5. Miscellaneous Latent & sensible heat gain (15-20%)

Within this research, all construction-based segments have been excluded in order to keep the primary focus on the Ice-Rink energy System stand-alone, excluding side amenities like space heating for changing rooms, etc. This may develop in conjunction with building demolition and construction later.

2.4 Refrigeration Systems in Ice Rinks

A representative ice rink for northern climates requires about 340 MWh per year to cover its thermal demands, with the majority going to the cooling system (Shirvani and Youssef, 2018). To calculate the refrigeration load for an open-roof ice rink, the methodology integrates principles from both enclosed ice rink modeling and adaptations for open-air exposure. The cooling system typically consists of the following thermal main hardware:

1. Refrigeration plant (compressors, condensers, evaporators)
2. Secondary fluid circulation system: Brine pump
3. Ice rink floor track piping network

To optimize the cooling energy use, managers can consider several strategies. (Karampour, 2011):

- Upgrading to more efficient refrigeration equipment
- Improving building envelope insulation
- Installing heat recovery systems
- Using CO2 as a refrigerant in transcortical systems
- Implementing advanced control strategies (Dynamic Response System)

A typical ice rink refrigeration system comprises compressors, condensers, expansion valves, and evaporators (Calisir and Karlsson, 2024). These systems have undergone substantial evolution over the years to enhance energy efficiency and minimize environmental impact.

2.4.1 Direct System

The refrigeration system designed for ice rinks could be either a direct or indirect system. Direct systems are commonly feasible to employ when there are no precautionary considerations regarding the type of refrigerant used in the system, for the choice between these systems impacts energy efficiency and environmental considerations.

For example, while it is theoretically feasible to use carbon dioxide in a direct system, the current unavailability of appropriately sized compressors makes this approach impractical. Consequently, the focus of this study will be on the indirect components of such systems, which are currently of greater relevance (Nilsson, 2009).

The refrigeration cycle in ice rinks typically involves several key components:

- Evaporator: Where the refrigerant absorbs heat from the ice surface or secondary coolant.
- Compressor: Increases the pressure and temperature of the refrigerant gas.
- Condenser/Gas Cooler: Rejects heat from the refrigerant to the environment or for heat recovery.
- Expansion Device: Reduces the pressure of the refrigerant before it re-enters the evaporator (Gao et al., 2024).
- Piping and other subsidiary system components, including:
  Track pipes, Fasteners, Collection pipes, Tanks, Heat exchangers, Pumps, Valves and filters, Pressure maintenance systems, Control and security systems. (Nilsson, 2009)

2.4.2 Indirect System

Indirect systems use a secondary refrigerant like calcium chloride, propylene glycol, ethylene glycol, or carbon dioxide, while direct systems use the primary refrigerant directly in the ice rink piping (Shirvani and Youssef, 2018). Whether using carbon dioxide or a water-based coolant, all current systems employ an indirect cooling method. In this setup, distinctions between the systems only become apparent at the heat exchanger, where the refrigerant is responsible for cooling the intermediate coolant. Indirect refrigeration systems are widely used in ice rinks due to their flexibility and safety advantages. In an indirect system, a primary refrigerant circulates in a closed loop within the refrigeration cycle, and a secondary coolant circulates through pipes embedded in the concrete slab beneath the ice surface. The role of the secondary coolant is to collect heat from the ice and transfer it to the primary side via a heat exchanger to be revamped or discharged. (Nguyen, 2013) (Nilsson, 2009).

This arrangement allows for better control and reduces refrigerant charge, but comes with some efficiency losses compared to direct systems (Hemati, 2023).

Advantages of indirect systems:

- Reduced refrigerant charge: The primary refrigerant is confined to the machine room, reducing the risk of leaks in public areas (Nguyen, 2013).
- Flexibility in refrigerant choice: More options are available for the primary refrigerant, including those that may be toxic or flammable, as they are isolated from the ice rink area (Nguyen, 2013).
- Easier leak detection and repair: Leaks are more easily identified and fixed in a smaller, confined system (Nguyen, 2013).

Disadvantages:

- Lower efficiency: The additional heat exchange step reduces overall system efficiency compared to direct systems (Nguyen, 2013).
- Higher initial costs: More components and materials are required, increasing investment costs (Nguyen, 2013).

2.5 Key Factors for Open-Roof Ice Rink Load Calculation

2.5.1 Environmental Exposure

Unlike enclosed rinks, open-roof designs are exposed to:

- Solar radiation (direct and diffuse)
- Wind-driven convection
- Precipitation (rain/snow)
- Sky radiative exchange (longwave radiation to the atmosphere)

2.5.2 Refrigeration Load Components

The total refrigeration load (Q_{total}) includes:

- Conductive heat transfer through the ice pad and foundation
- Convective heat loss to ambient air
- Radiative heat exchange (solar gain and sky radiation)
- Latent heat from resurfacing and ambient humidity
- Sensible heat from occupants, lighting, and equipment

2.5.3 Impact of Protective Roof Design

Retractable roof systems, convertible shading, and photovoltaic arrays can effectively decrease refrigeration loads in open-roof ice rinks by reducing solar and convective heat transfer. Research (Wang et al., 2024) indicates that shading can obstruct 30–50% of direct solar radiation, whereas high-albedo materials can reflect up to 80% of incident energy, in contrast to the 20% reflection from conventional surfaces. Protective Roofs also lower convective loads by reducing wind exposure; for Gothenburg's average wind speed of 4 m/s, wind shielding cuts the convective heat transfer coefficient by about 31%, lowering refrigeration demand by 18–25% under windy conditions. At the system level, this dual mitigation decreases annual energy use by 12–18%, reduces peak summer loads by up to 22%, and minimizes frost formation by stabilizing surface temperatures.In Gothenburg's climate, retractable designs featuring angled photovoltaic panels are notably effective, optimizing winter solar gain management while facilitating passive radiative cooling in sub-zero temperatures.

2.6 Ice rink refrigeration system alternatives:

The refrigeration system is crucial for sustaining ice surfaces within optimal temperature ranges of -3°C to -5°C. This presents a challenge in Nordic climates, where seasonal fluctuations necessitate efficient operation to maintain stable ice quality. The increasing interest in sustainable refrigerants like carbon dioxide (CO_2) and ammonia is attributed to their advantageous thermodynamic properties, notably higher coefficient of performance (COP) values ranging from 3.0 to 4.5, in contrast to 0.55 to 0.70 for absorption systems, as well as their low global warming potential (Nilsson, 2009, Shirvani and Youssef, 2018).CO_2-based systems are notable for their environmental advantages, high energy efficiency, effective operation in cold climates such as Sweden, and significant potential for heat recovery, rendering them particularly appropriate for contemporary ice rinks connected to district energy networks (Calisir and Karlsson, 2024).

2.6.1 Transcritical CO_2 - R744 System as the Preferred Refrigerant Cycle

There has been a move toward more eco-friendly refrigerants in the last few years, especially CO2. Using CO2 as a refrigerant in ice rinks has shown good results in terms of saving energy and having less of an impact on the environment (Girip et al., 2023a).In trans-critical mode, CO2 systems can work at greater temperatures for rejecting heat and improved chances for recovering heat. The initial costs of CO_2-based systems can be significant because they need specialized high-pressure parts, but these costs are frequently offset over time by decreased energy use and less need for maintenance. Also, economic incentives like rebates for using low-GWP technologies can make CO_2 systems more cost-effective overall (Sjoquist, 2021). A similar system with parallel compression can save up to 42.6% more energy than traditional ammonia systems, and its Seasonal Performance Factors (SPFs) can reach 7.5 (Shirvani and Youssef, 2018).

R744 Properties

Carbon dioxide (CO_2), often known as R744 in refrigeration, is one of the best alternatives to traditional refrigerants since it has very little effect on the environment. It has a Global Warming Potential (GWP) of 1 and no Ozone Depletion Potential (ODP) (FRIGO, 2024, CAREL, 2025).CO_2

is better than hydrofluorocarbons because it has minimal temperature glide (≤3°C), great heat transmission at high pressures (≥80 bar), and it works well in tough situations. For ice rink applications that need ice temperatures close to -3°C, CO_2 must be cooled to roughly -9°C, which is 2.72 MPa of pressure. The system's working pressures are between 2.6 and 3 MPa, and the design pressure is 4 MPa for safety (Granryd, 1998) ; (Nilsson, 2009).With a critical temperature of 31°C and critical pressure of 7.38 MPa, CO_2 transitions into the transcritical region above these limits, lacking a distinct liquid–vapor phase (Shahzad, 2006).

- Nominal Volumetric mass flow

The hourly volume of liquid carbon dioxide required to be circulated through the Chiller skids at cooling capacities of 300, 400, and 500 kW, respectively, is presented in the Table below (Nilsson, 2009) :

Table 1-Volume Flow Rates Required for Various Cooling Effects

Cooling effect	Volume flow
300 KW	8.61 m3 /h
400 KW	11.49 m3 /h
500 KW	14.36 m3 /h

- Steep Pressure-Temperature Curve:

The vapor pressure of CO2 increases steeply with temperature compared to other refrigerants (see Figure 2 and Figure 3).

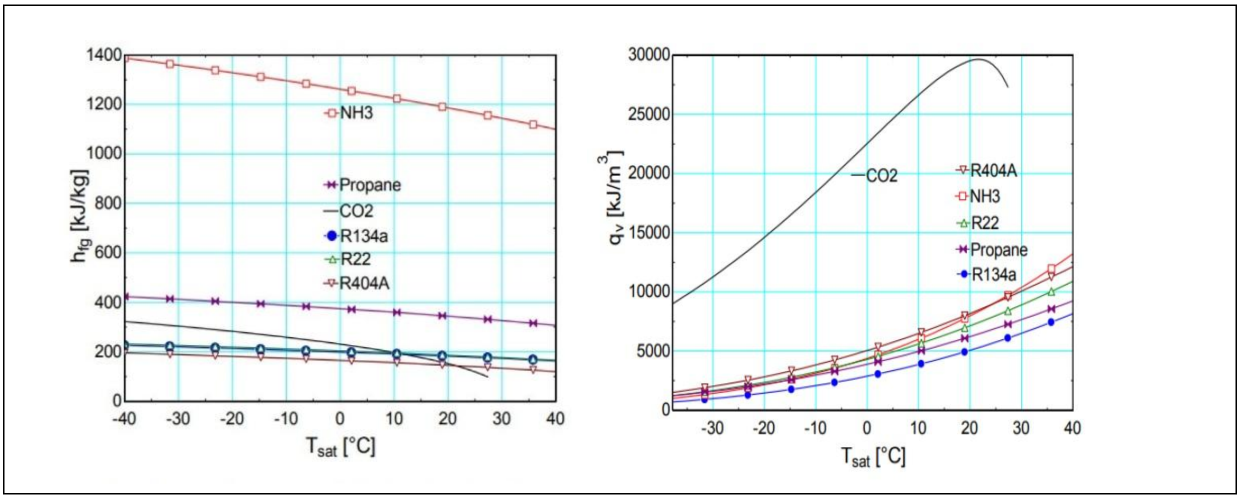


Figure 2- Key Characteristics of R744 in a P-T Diagram

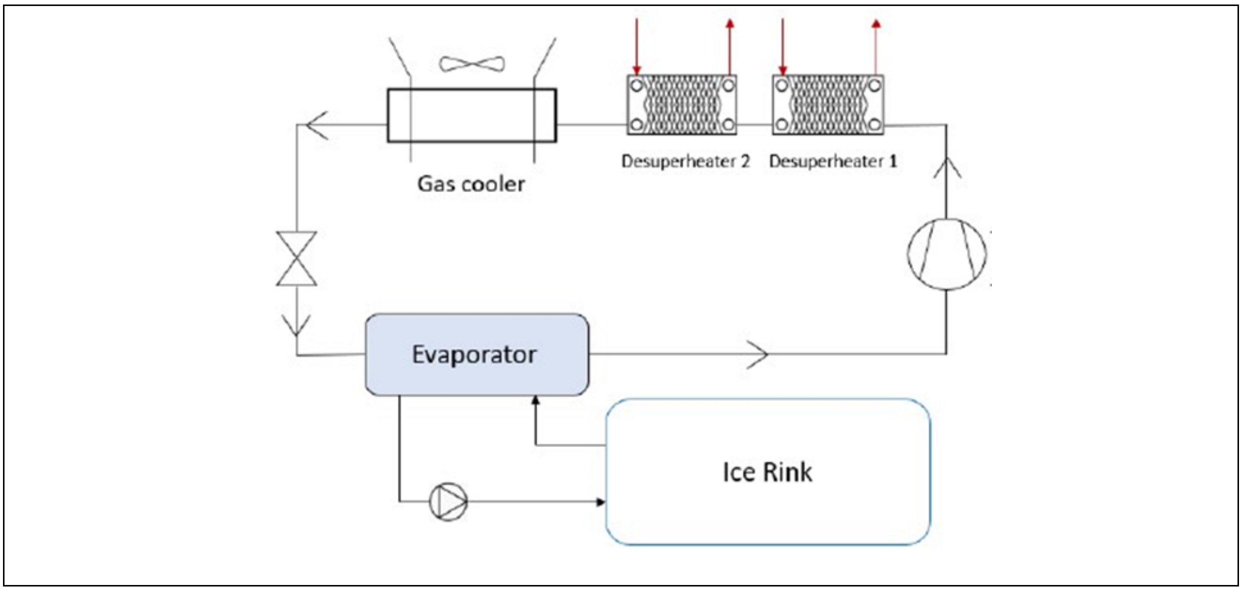


Figure 3_ Typical CO2 transcritical chiller (Hemati, 2023)

Potential benefits and Advantages of R744 as a refrigerant

R744 (CO_2) is a great refrigerant for ice rinks and other uses since it has several benefits. It is a safe, natural, non-toxic, and non-flammable fluid that can be used in a lot of different ways and works well in both subcritical and transcritical settings (FRIGO, 2024).CO_2 is safe for the environment because global rules will not phase it out. This is in line with plans to get rid of refrigerants with a high global warming potential (Northstarhvacr, 2024). Its good thermodynamic features make it particularly energy efficient, especially in colder locations. It has a high COP, is small, and can recover much heat (Matthew Martinez, 2024). Secondly, multi-compressor redundancy increases liability even more by making sure that the system keeps working even when parts break (hillphoenix, 2024), Using it as a secondary refrigerant in copper piping has also shown promise for saving up to 20% more energy than ammonia-based systems (Calisir and Karlsson, 2024).

R744 Key Challenges and Drawbacks

R744 systems face several challenges, primarily due to their high saturated working pressure (407 psig at +20°F), This means they need special high-pressure components like several-stage compressors, heat exchangers, pipelines, and valves that are often made of stainless steel like SS304L for safety in high-pressure operation (hillphoenix, 2024). Compared to ammonia, CO_2 systems feature compressors that practically less efficient and usually need a bigger initial expenditure to buy, install, and run. However, depending on the size of the system, they can still be cost-competitive (Sjoquist, 2021). The market also does not have many compressors that are the right size for direct refrigeration systems with changing needs, such as ice rinks. This makes it harder to use (Nilsson, 2009). Also, CO_2 has no color or smell (odorless); thus, accurate leak detection and monitoring systems are necessary for safe operation.

In the context of Rosenlundsverket, where district heating and cooling systems are operational, the heat recovery capability of CO_2 systems could enhance energy efficiency by reintegrating recovered heat into the network. This synergy improves overall system efficiency and diminishes dependence on an external energy source.

2.6.2 NH3 - R717 (Ammonia)

Ammonia is widely employed in industrial refrigeration systems across various sectors, such as food processing, cold storage, pharmaceuticals, and marine operations, and it constitutes the primary refrigerant in ice rink refrigeration systems in Sweden, representing about 85% of installations (Makhnatch, 2011). Having been utilized for almost a century in refrigeration, it is a proven and dependable option. Ammonia provides superior thermodynamic efficiency and is regarded as environmentally sustainable. Its appeal arises from various advantages that render it a favored option for extensive refrigeration requirements. Nonetheless, ammonia has various obstacles, including its toxicity, potential flammability, and compatibility concerns with specific materials. To mitigate the possibility of leaks and pipe ruptures, these systems are frequently engineered as indirect refrigeration systems, particularly in applications such as ice rinks.

NH3 Properties

Ammonia (NH_3), referred to as R-717, is a natural refrigerant that possesses neither ozone depletion nor global warming potential. It possesses a low boiling point of -28°F at atmospheric pressure and a critical temperature of 221°F. Despite being colorless, it is readily recognizable due to its powerful, acrid scent. Ammonia is industrially synthesized by reacting nitrogen and hydrogen gases in the presence of an iron catalyst, a process that may necessitate up to 60 GJ of energy (Danfoss, 2015).

NH3 Principal Challenges and Drawbacks

Ammonia systems face several challenges, including the potential for product contamination, restricted compatibility with yellow metals like brass, bronze, and copper, and dependence on evaporative condensers that necessitate substantial water consumption, drainage, pumping, and chemical treatment. They experience comparatively low efficiency and subpar heat recovery quality, necessitate intricate and expensive regulatory compliance, and require an additional secondary coolant subsystem (Danfoss, 2015).

Potential benefits and Advantages of NH3 as a refrigerant

R-717 (ammonia) is an economical and readily accessible refrigerant with a demonstrated history of dependable performance across several applications. It provides somewhat superior theoretical efficiency relative to R134a or propane, necessitates reduced circulation energy than many synthetic alternatives, and functions at pressures akin to numerous conventional refrigerants (see Figure 4). Moreover, its unique scent functions as a natural leak warning, hence improving safety (Bantillo et al., 2024).

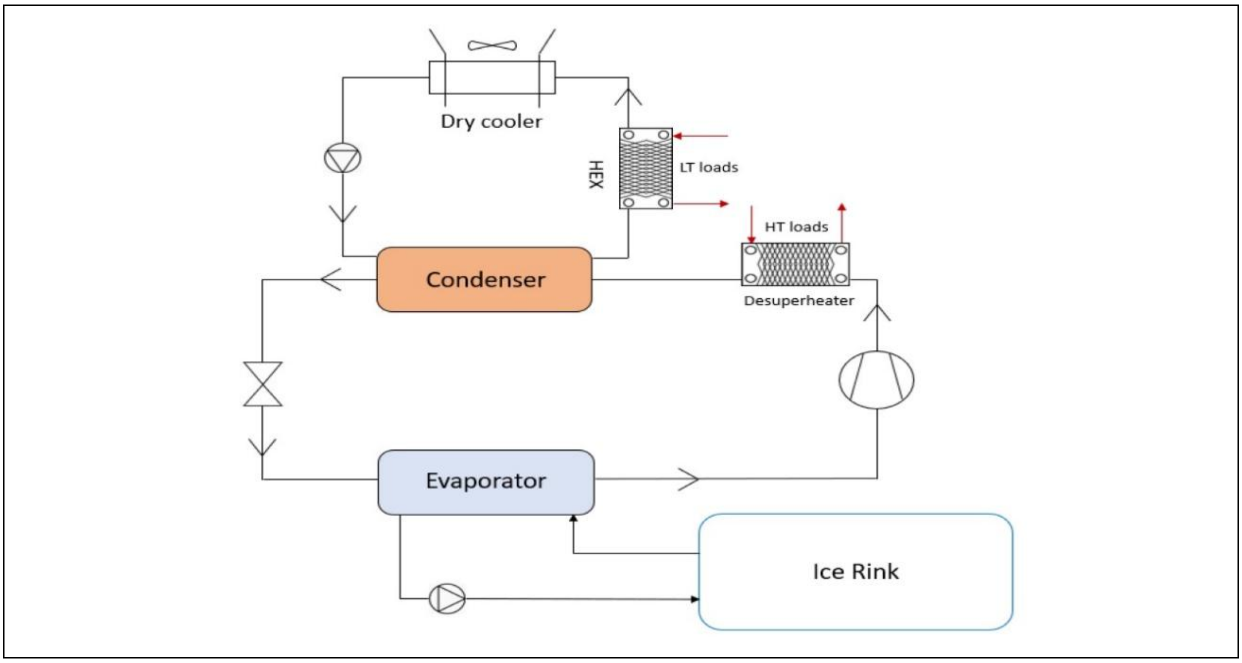


Figure 4 __Typical NH3 refrigeration System (Hemati, 2023)

2.6.3 Heat-Driven (Absorption Refrigeration System)

Absorption systems use heat (instead of mechanical energy) to drive the refrigeration cycle; the key components are like a conventional vapor-compression system: an evaporator, condenser, and expansion valve. However, it replaces the mechanical compressor with a generator (Desorber) and absorber. The system operates using two fluids, including an absorbent (typically lithium bromide or ammonia) and a refrigerant (commonly water). The absorbent cycles between the generator and absorber, whereas the refrigerant traverses the condenser, evaporator, and expansion valve, engaging with both the generator and absorber. Absorption chillers are characterized by their dependence on heat instead of electricity as the main energy source for producing a cooling effect (Zabian and Chan, 2022). Standalone absorption chillers are unsuitable for dynamic ice rink systems that necessitate swift load adjustment when deploying absorption chillers as the primary refrigeration system (rather than auxiliary) in scenarios with sufficient available heat, 80–120°C (e.g., waste steam or hot water). Several critical factors determine feasibility and efficiency (Villa et al., 2019). Applying this system reduces reliance on compressors, cutting energy costs. As a proven case study, the Pirkkala rink achieved a 30% reduction in consumption with enhanced heat recovery (iifiir, 2025). Absorption chillers have restricted compatibility with dynamic response systems in ice rink applications due to their slower thermal inertia and temperature limitations; however niche implementations are feasible under some conditions. The systematic study was performed utilizing thermodynamic concepts and practical applications as shown in Figure 5.

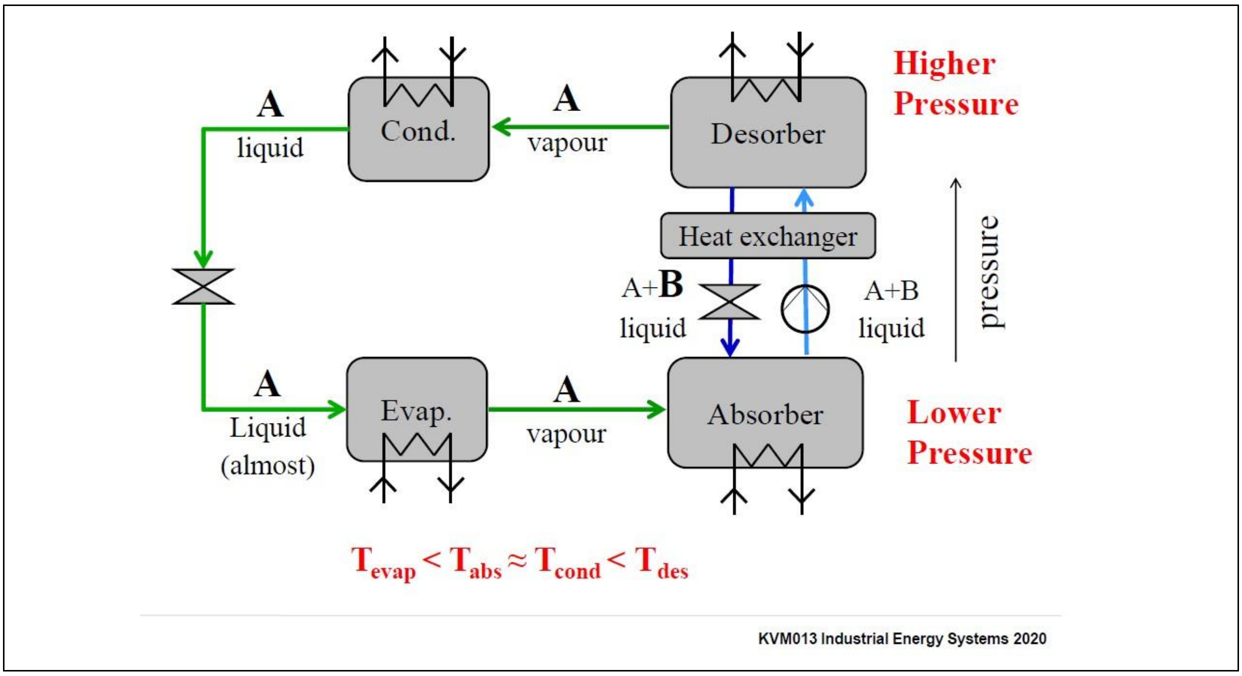


Figure 5- Absorption Refrigeration Cycle

Key Challenges for Absorption Chillers

Absorption chillers present several challenges that restrict their use in ice rink refrigeration. Their sluggish dynamic response, generally necessitating about 100 minutes to stabilize following load variations like resurfacing or event-induced thermal spikes, stands in stark contrast to the swift adjustment capabilities of electric compressor-driven systems, which can modify cooling capacity within minutes. Lithium bromide–water chillers are typically engineered to supply cooled water at approximately 7°C, which complicates achieving the low evaporation temperatures of −10°C to −15°C necessary for preserving ice quality in rinks (Villa et al., 2019). Further disadvantages encompass heightened maintenance intricacy, as lithium bromide solutions necessitate corrosion inhibitors and regular purging of crystallized absorbent, alongside elevated initial investment costs, projected to be 30–50% higher than electric systems; however, payback periods of 5–7 years may be attainable in energy-intensive applications (Grossman et al., 1995; Rivera et al., 2018).

Potential benefits and advantages

Absorption chillers provide numerous advantages when included in energy systems. They can substantially lower energy expenses by diminishing dependence on grid electricity, attaining savings of up to 60% in facilities equipped with on-site heat-generating systems, such as CHP plants (EnergyLink, 2024). Furthermore, in contrast to mechanical chillers, absorption chillers expel a greater volume of high-grade excess heat, necessitating larger cooling towers and facilitating energy recovery options (U.S. Department of Energy, 2017). In hybrid setups, they enhance compressor-driven units by alleviating startup delays during load fluctuations and decreasing variable energy costs through an enhanced overall coefficient of performance (COP). Their connection with CO_2 or ammonia systems facilitates cascade refrigeration, improving efficiency in elevated outside temperatures where transcritical CO_2 cycles typically underperform as presented in Figure 6 (Thanasoulas, 2018).

Table 2-Comparison of Chiller Technologies for Ice Rink Applications

Feature	Absorption Chillers	Electric/CO_2 Chillers	NH_3-R717 Chillers
Response Time	100+ minutes	5–15 minutes	~5 minutes
Cooling Capacity	7°C chilled water	−15°C evaporation	−30°C to −40°C evaporation
Energy Source	Waste heat (80–120°C)	Electricity	Electricity or waste heat
COP (Cooling)	0.55–0.70	3.0–4.5	5.0–6.0
Ice Rink Suitability	Limited to auxiliary cooling	The standard for dynamic loads	Ideal for ice rink applications

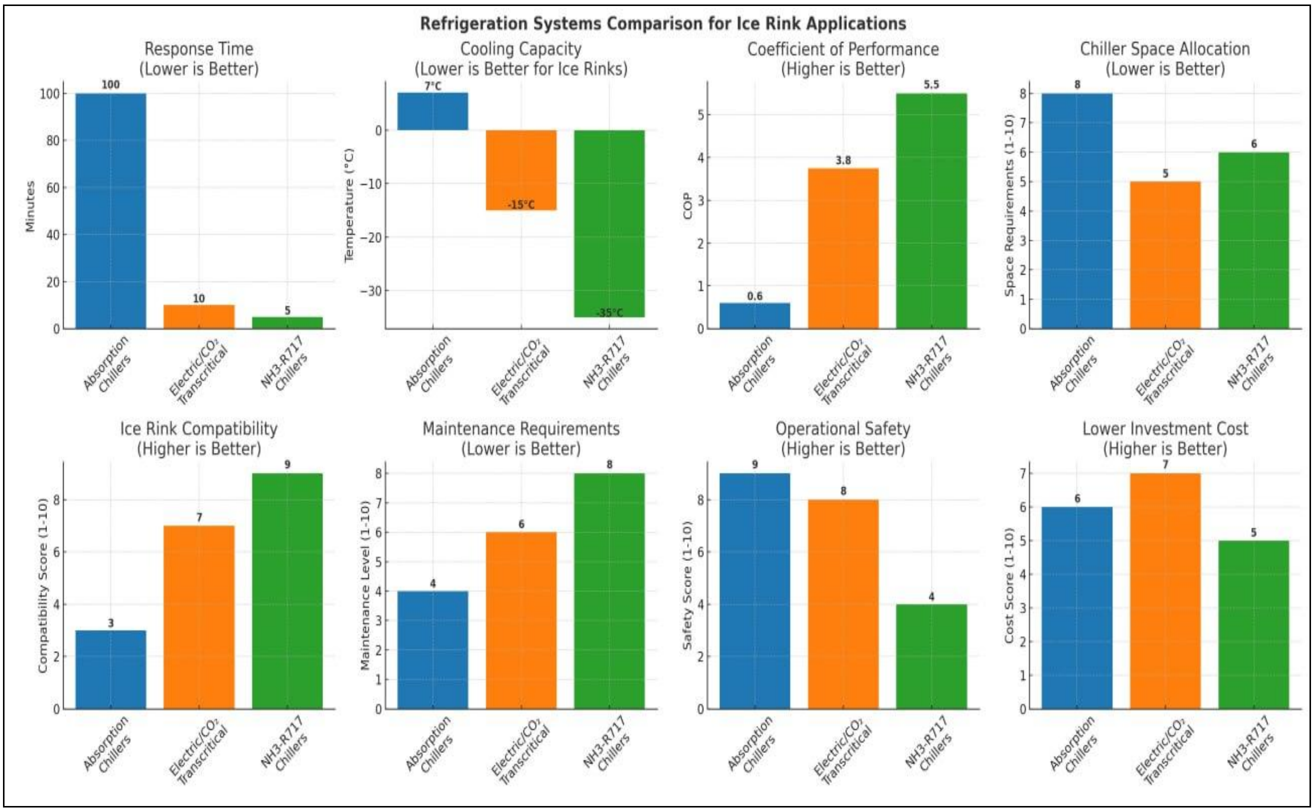


Figure 6_ Refrigeration Technologies Overview

2.6.4 Hybrid system concept (Combo Scenario)

System Characteristics and Specifications

Within the Hybrid System concept, an absorption chiller (preferably ammonia–water) with an electrical COP of about 20 and capacity to cool down to -15 degrees is considered as a base-load portion, with a backup CO_2 Transcritical compression chiller sized as Peaker to deal with the dynamic load swings and fast response demand for the rooftop Ice rink, specifically during the hot season. No capacity "buffer" is required unless specified by ASHRAE safety margins for critical applications. This arrangement could be an efficient option for indoor Ice rink where the indoor climate supposed to be under control by HVAC system, but when it comes to open roof rink still there would be the uncertainty whether could be a robust solution or not, as the excess heat source would be surplus for specific time of years (summer time) to serve the absorption chiller it is coincident exactly to when the ice rink machines needs to deal with critical moments in terms of dynamic load fluctuations for open roof ice rink, this system setup needs to be tuned in such a way to be able shift into the 100% compressor chillers when the excess heat at substation needs to be utilized elsewhere such as district heating (Tao et al., 2022).

Moreover, practically it's not possible to design the Ice rink system for optimal Ice surface Temperature -3 as it proposed for entirely run by compressor chiller and it should be overestimate to below down the Ice set point around -5 degrees considering the absence of indoor climate control by HVAC due to open roof design as well as the potential lagging response time to sync two chillers together in a minimal response time, subsequently will affect the $Q_{effective}$ design in higher value than completely run by CO_2 compressor chiller. Lowering the ice surface design temperature from -3°C to -5°C significantly increases the required $Q_{effective}$ (cooling capacity= 762.5 KW), leading to higher system energy consumption and reduced overall efficiency according to findings by (Karampour, 2011), shows roughly 10-15% growth in total refrigeration system heat load when the designed ice temperature is decreased by 2°C, assuming other conditions are constant. This is directly proportional to the effective heat load Q_{eff} calculated in analytical models used for Swedish ice rinks.

ASHRAE typically recommends sizing the base-load chiller to cover around 60–80% of the total cooling requirement, with the Peaker chiller handling the remainder during load swings or peak periods. This aligns with best practices for hybrid systems to maximize efficiency and manage dynamic loads.

A practical split would be:

- Base-load chiller (ammonia–water absorption): 70% of Q $effective_{design}$ ≈ 553.7 kW
- Peaker chiller (CO2 transcritical): 30% of Q $effective_{design}$ ≈ 228.7 kW

This concept needs to be equipped with auxiliary advanced and complex controllers and drivers, as well as extension of the system through an intermediate loop, which imposes the extra complexity and capital, both for initial investment as well as maintenance cost over the years, as fast-response capability is a crucial factor; technically absorption chiller would not be possible to run the system standalone. Absorption chillers are quite different from compression chillers in how their COP (electricity to cooling) is defined, because they mainly use thermal energy (steam, hot water, direct-fired gas) as the driving source, with only a small fraction of electricity for pumps, fans, and controls. COP (electric): ~20–40, but this justification could not be applicable for all setups and misleading could be inevitable since the main energy driver is heat, rather than electricity, specifically for subzero operation, which, when converted to electrical SPF equivalents (if using waste heat as input and accounting for parasitic electricals), is much lower than reality (Thanasoulas, 2018).

Practically, designers use the following formula (Equation 1) to scale up the potential QCond in the absorption cycle as well.

Equation 1

$$Q_{cond} \approx Q_{evap} + Q_{gen} / COP_{abs,\ thermal} \cdot (1 - COP_{abs,\ thermal})$$

Since both condenser and absorber reject heat to a common cooling-water loop, engineers usually speak of the total rejection duty (see Equation 2):

Equation 2

$$Q_{rej} \approx Q_{cond} + Q_{abs} \approx Q_{eva} \cdot (1 + 1/COP_{abs,\ thermal})$$

For double-effect, with COP ≈ 1.1, based on rule of thumb (see Equation 3).

Equation 3

$$Q_{rej} \approx 1.9\ Q_{eva}$$

So, if your chiller is producing 553.7 kW cooling, the rejected heat is ~1050 kW, but the temperature levels within span 45-55, which is suitable for local heating.

2.7 Energy Efficiency Measures and Practice in the Context of Nordic Ice Rink

2.7.1 High-efficiency refrigeration systems

Nordic ice rinks can significantly reduce energy use and maintain high-quality ice by adopting efficient refrigeration technologies that also enable waste heat recovery. Growing interest is directed toward sustainable refrigerants such as CO_2 and ammonia, which not only have low global warming potential but also offer superior thermodynamic performance, providing lower operating temperatures and higher COP values (3.0–4.5 compared to 0.55–0.70 for absorption systems) (Nilsson, 2009, Shirvani and Youssef, 2018) while minimizing thermal losses .

2.7.2 Choosing the Secondary Coolant for Minimal Pumping Work

Aqua ammonia (ammonia–water solution) has exhibited remarkable longevity in more than 35 Swedish ice rinks since 2007, necessitating minimal maintenance and providing distinct energy and environmental benefits compared to conventional secondary coolants such as calcium chloride ($CaCl_2$) and ethylene glycol. It diminishes pumping power requirements by 45–47%, reducing circulation energy consumption to merely 1–3% in contrast to 10–20% for traditional fluids, while facilitating COP enhancements of 4.7% relative to $CaCl_2$ and 11.6% compared to ethylene glycol. Its compatibility with natural refrigerants like transcritical CO_2 enables economical retrofits, and despite its heat transfer coefficients being 9–27% lower than those of traditional fluids, the 40–55% reductions in viscosity-related pumping power compensate for the thermal performance deficit (Kilberg, 2020).

2.7.3 Optimal Design Flow &Temperature of Ice Resurfacing Feedwater

In Nordic ice rinks, utilizing resurfacing water at approximately 40°C enhances CO_2 chiller efficiency by reconciling diminished refrigeration demands with efficient high-temperature heat recovery; conversely, lower temperatures, while decreasing resurfacing heat gains, fail to exploit this recovery capability fully (Pomerancevs et al., 2020, Karampour, 2011, Thanasoulas, 2018).

The resurfacing process provides latent heat (from water freezing) and sensible heat (from cooling hot water to 0°C), with frequency having a direct impact on overall energy demand (Lind, 2018).

2.7.4 Well-Placement of Chiller room with respect to Ice Pad Location

(Yu et al., 2024) investigated the impact of spatial layout design on ice rink energy consumption. The study revealed that optimizing the arrangement of functional areas, such as locating heat-generating spaces (chiller room) away from the ice sheet, can reduce overall energy demand by 5-10%. Additionally, proper insulation and strategic placement of windows and doors can minimize heat transfer between different zones of the ice rink. Modular and removable designs could be a solution. Some rooftop rinks are being designed with modular components that can be easily installed and removed seasonally. This approach allows for flexible use of rooftop spaces and can reduce the structural load during non-winter months.

2.7.5 Self-Sufficient Rink (Passive Arena)

a) Integrated cooling systems:

Advanced cooling technologies are being designed to merge the refrigeration requirements of the ice rink with the building's comprehensive HVAC system. This method can markedly enhance energy efficiency by harnessing waste heat from the refrigeration process for space heating or the generation of household hot water (Zabian and Chan, 2022).

b) Integrated Heating System

According to (ASHRAE, 2010), the amount of excess heat rejected through the refrigeration system could be sufficient to cover a great share of the heating demand, up to 100% needed (Karampour, 2011).Ice rinks are especially conducive to waste heat recovery because of their simultaneous heating and cooling demands. Instead of releasing surplus heat at high pressure, this thermal energy can be harnessed and utilized to satisfy the facility's heating requirements. In certain cases, the recovered heat is adequate to meet all heating requirements, potentially allowing the ice rink to function autonomously from external heating sources (Rogstam et al., 2016). The typical heating requirements associated with ice rink operations, essential to fulfill by either internal integration or external sources, are enumerated below (see Figure 7):

- Ventilation (local heating) and Side Amenities
- Ice resurfacing water
- subfloor heating
- Snow melting (melting pit)

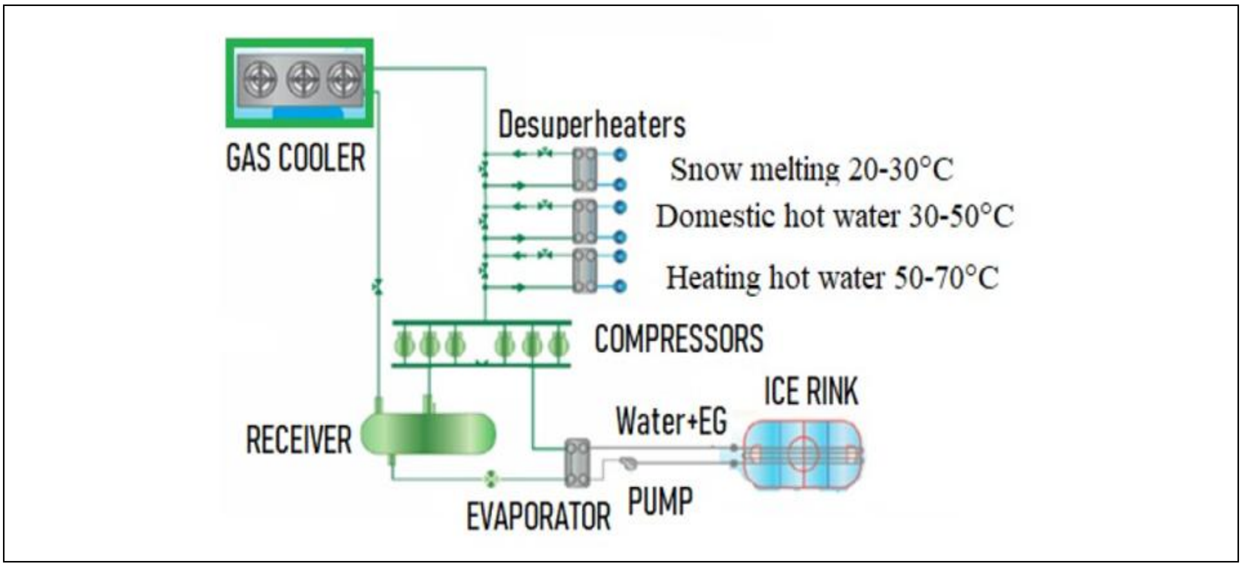


Figure 7_Typical trans-critical CO2 system for ice rink with heat recovery (Girip et al., 2023b)

2.7.6 Process calibration, well-configuration and advance control System

Optimizing refrigeration system configuration is vital for energy efficiency, with advance features such as economizer cycles, parallel compression, ejectors, and subcooling providing superior COP values despite increased capital expenditures (Hemati, 2023). Efficient calibration of ice rink sub-systems via sensor optimization, control modifications, and supplementary equipment, along with sophisticated monitoring that adjusts to real-time conditions, guarantees optimal performance and sustained energy conservation. Advanced sensor networks and AI-driven control systems are implemented to perpetually monitor ice conditions and enhance cooling system performance. These systems can modulate cooling output in response to variables such as ambient temperature, humidity, and usage patterns (Tiusanen et al., 2012).

2.7.7 Employed WHR Practice

a) Integration of Ice Rink Refrigeration System's heat Sink into Enclosed District Energy

Ice rinks represent an attractive opportunity for integration with district energy systems because of their high cooling demand and the significant potential for waste heat recovery.

The refrigeration process necessary for sustaining ice surfaces at -3°C to -7°C generates substantial waste heat at 30°C to 35°C, which can be effectively redirected to district heating networks rather than being expelled into the environment (Lind and Rundgren, 2017, Pieper, 2019). This integration improves overall energy efficiency while providing community-wide energy savings and environmental advantages. The Resenlundsverket location, designated in Gothenburg's urban development plan as one of the first hubs of a district energy system in the south southwest of Sweden, and its adjacent to an ice rink on its rooftop, offers a persuasive argument for process integration. Here, waste heat from refrigeration systems could be recovered to either operate absorption chillers for district cooling or preheat the district heating return line, while the DH network simultaneously covers the rink's other heating demands. Successful precedents already exist, such as Gothenburg's Angered Arena, where recovered heat up to 68°C is supplied to the local district heating network through heat pumps, demonstrating practical potential highly relevant to this research (Jangsten et al., 2020, Pieper, 2019)

b) Excess Heat recovery into power through an Auxiliary ORC solution

The Organic Rankine Cycle (ORC) functions as an auxiliary augmentation to augment energy efficiency and operational flexibility by converting waste heat into power. ORC systems are validated for localized power generation; however, their efficacy is contingent upon adequate incoming temperatures from the chiller's gas cooler or condenser. Research indicates that a minimum heat source temperature of around 58°C is necessary for effective functioning (Tipton et al.).with 80°C commonly used in theoretical models (Wernersson, 2023). 90–95°C marking the threshold for practical efficiency, and 100–150°C representing the optimal range for industrial waste heat recovery (Iyengar et al., 2022).

2.7.8 Optimal Spatial Design of Shading over the Ice Pad

The spatial design of an ice rink can have a substantial impact on its energy efficiency. (Wang et al., 2024) Assert that the configuration of shade in an airtight system, such as a tent over an ice rink, significantly influences its thermal efficiency. Their research indicated that a curved spatial design (Context) can diminish the cooling load by as much as 15% relative to a flat surface, owing to enhanced air circulation and less solar heat gain. Rooftop ice rinks pose distinct problems and opportunities for creative design strategies. Such facilities necessitate meticulous evaluation of structural integrity, energy efficiency, and environmental regulation. A variety of inventive design techniques, as below, have been employed to tackle the aforementioned challenges.

a) Photovoltaic Shading Cover

Various shading and renewable energy solutions were assessed to enhance energy efficiency and operational sustainability in ice rink design. Upon evaluating alternatives, including traditional rigid solar panels, reflective coatings, and passive shading systems, Midsummer BOLD flexible PV films proved to be the most feasible choice for incorporation into curved spatial designs. This feasibility evaluation examines their distinct technical specifications, energy-generating capabilities, and alignment with the thermal and structural needs of the ice pad.

b) Rainwater harvesting as an ice resurfacing reservoir:

Advanced techniques are being created to gather and purify rainwater for application in ice resurfacing activities. This method can diminish water usage and offer a sustainable solution for ice maintenance (Birkeland, 2018). The Rosenlundsverket rooftop ice rink has potential to integrate rainwater harvesting embedded to its spatial shading structures to Captures ~23% of rainfall from impervious surfaces during extreme events, to support sustainable operations as reservoir Delivers low-turbidity water (<5 NTU) for ice resurfacing and Cuts municipal water use by 60–80% (3,000–5,000 L/day savings despite of potential challenges corresponds to winter freezing risk and Water quality conditioning to control pH and citric acid dosing to prevent scaling in brine loops. The table in the Appendix is the proposed design aligned with the case study technical framework and Gothenburg's hydrological context.

c) Solar reflective coatings:

Highly Reflective roof coatings are utilized to decrease solar heat gain and lessen the cooling demand on the ice rink. These coatings can reflect as much as 85% of solar radiation, aiding in the preservation of stable ice conditions and diminishing energy usage (Lind and Rundgren, 2017)

d) Wind barriers and aerodynamic design:

Rooftop rinks frequently experience elevated wind velocities compared to ground-level facilities. Innovative wind barrier designs and aerodynamic roof configurations are being utilized to mitigate wind-induced heat transfer and enhance the rink's energy efficiency (Jangsten, 2020a).

2.8 Rosenlundsverket site and District Energy System:

Resenlundsverket History:

Rosenlundsverket, a historically significant energy plant in Gothenburg, has been essential to the city’s industrial and urban development since 1846, when Scandinavia’s inaugural gasworks were erected on the site to provide gas lighting. The facility became an important part of Gothenburg's energy infrastructure over time. It changed from an oil-fired combined heat and power (CHP) plant in the 1950s to one that used natural gas and flue gas condensation technology by the 1980s to increase efficiency. The plant has a maximum production of 662 MW of thermal energy and 36 MW of electrical energy, and it commenced district cooling services in 2007, utilizing absorption and compressor systems with a total capacity of 77 MW. It is recognized as a main intersection of the district energy network. Rosenlundsverket is architecturally distinguished by its sky-colored chimneys, crafted by Nils Andréasson, and is acknowledged for its technical and cultural significance. The scheduled decommissioning demonstrates Gothenburg's overarching dedication to sustainable energy transitions and environmentally aware urban planning (Risell and Selberg, 2015, Zhou et al., 2021).

Göteborg Energy’s role in the city's energy infrastructure

Goteborg Energy plays a crucial role in Gothenburg's energy infrastructure, serves as the principal operator of the city's district heating and cooling systems(see Figure 8). The company is responsible for:

1. Operating and maintaining the district heating and cooling networks.
2. Producing and distributing heat and cooling to connected buildings.
3. Implementing energy efficiency measures and exploring innovative solutions to improve system performance.
4. Collaborating with building owners and operators to optimize energy use and reduce greenhouse gas emissions (Jangsten, 2020b).

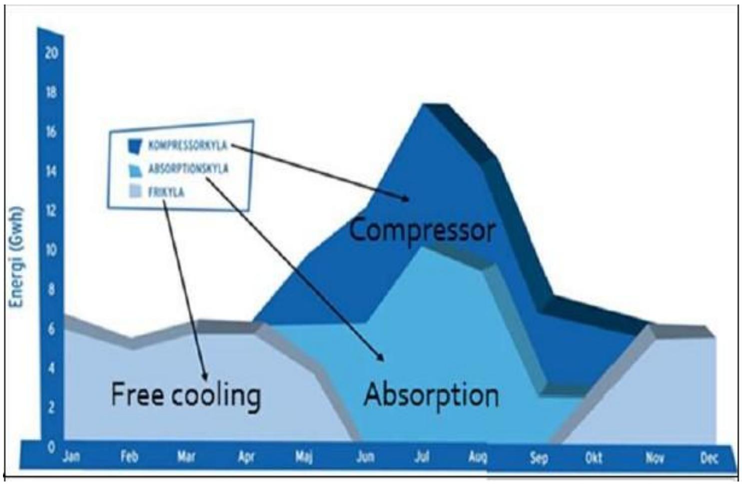


Figure 8-Seasonal Operational Order _source GE Thesis brochure

Operational Seasonal Pattern at Rosenlundsverket:

- *Direct (Free) Cooling 25% Period:*(Nov-March)

Leveraging river water for free cooling during colder months to enhance efficiency.

- Absorption Chiller 50% period (April- Oct)

Using waste heat from industry as a supply to serve an absorption heat source

- Compressor-driven Chiller 25% period (backup summer)

A centrifugal compressor chiller is a common type of chiller used in district cooling systems with a coefficient of performance of above 7. It makes use of the vapor compression cycle and commonly uses R-123 or R-134a as the refrigerant as a backup to support.

Figure 9 respresents Rosenlundsverket trnasformations during the time.

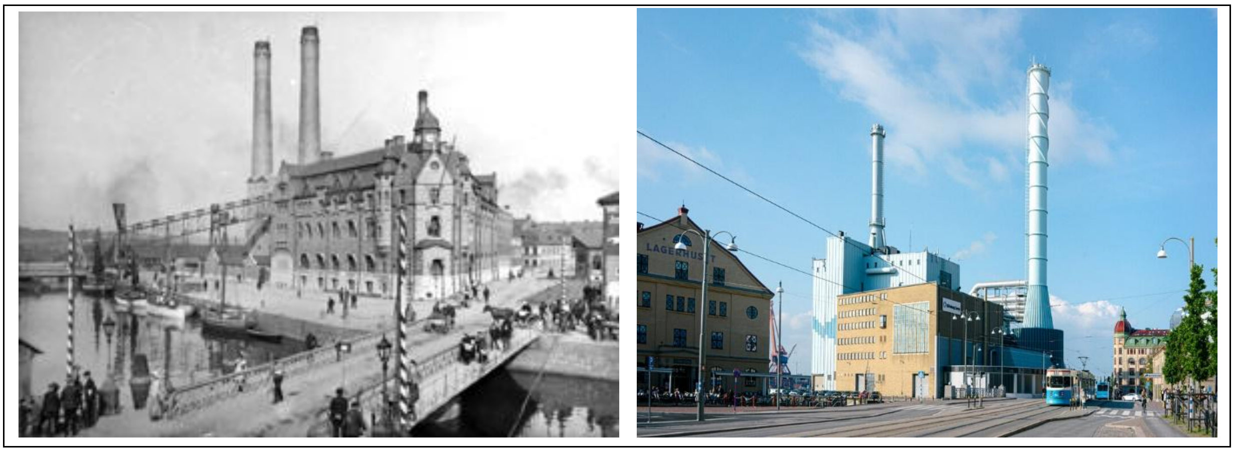


Figure 9 Rosenlundsverket Goteborg Energy

2.8.1 Gothenburg district heating and cooling systems

District Heating System

Gothenburg has a well-established district heating system that has been developed over several decades. The district heating network in Gothenburg is extensive, providing heat to a large portion of the city's buildings. The system utilizes waste heat from industrial processes and waste incineration, as well as other renewable and non-renewable sources, to generate hot water that is distributed through a network of underground pipes (Lind and Rundgren, 2017).

District Cooling System

Gothenburg's district cooling system is the second largest in Sweden, with a total installed capacity of 70 MW and 30 km of piping. The system primarily relies on absorption chillers running on waste heat, with free cooling from the river used during winter months. In 2018, the Gothenburg district cooling system delivered around 100 GWh of cooling (see Figure 10) (Lind and Rundgren, 2017).

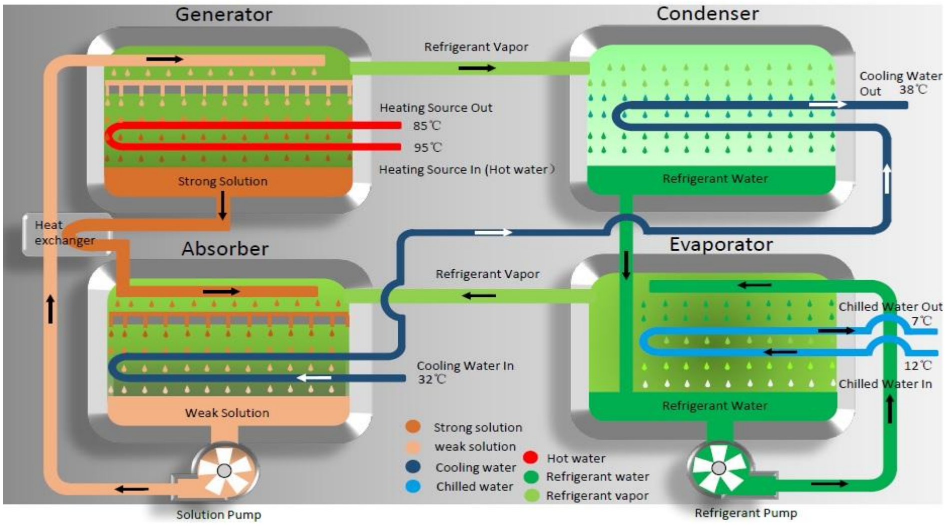


Figure 10 _District cooling Sys Diagram source

The technical principles of the district cooling system in Gothenburg involve centralized chilled water production and distribution. Chilled water is generated at large production plants and distributed to connected buildings through underground pipes. The system uses plate frame heat exchangers in energy transfer stations to separate the distribution system from the connected buildings' chilled water systems (Jangsten, 2020b).

The conventional infrastructure for the embedded district cooling system includes:

1. Production plants: Where chilled water is generated using absorption chillers and free cooling sources.
2. Distribution network: Underground pipes that transport chilled water to connected buildings.
3. Energy transfer stations: Where plate frame heat exchangers transfer cooling energy from the district cooling network to the building's internal cooling system.
4. Building chilled water systems: The internal cooling systems of connected buildings that utilize the chilled water provided by the district cooling network (Jangsten, 2020b).

2.8.2 Drawbacks Vs strengths behind the integration to the Substation

Potential Advantages:

Integrating the ice rink with Rosenlundsverket's district energy systems provides several Economic and Environmental Benefits and advantages.

I. **Energy Efficiency**: By either utilizing the waste heat from industry to serve the base chiller of ice rinks, or recovering the rejected heat from chillers, the overall energy efficiency of both the ice rink and the district energy system can be improved (Pieper, 2019) Waste heat recovery reduces primary energy consumption by repurposing otherwise discarded thermal energy according to IIHF (Hayes, 2023).

II. **Load balancing:** Ice rinks typically have high cooling demands during summer months when heating demands are low, providing an opportunity to balance seasonal loads in the district energy system (Lind and Rundgren, 2017).

III. **Cost Savings**: The sale of recovered heat to the district heating network generates an extra revenue stream for the ice rink operator and reduces operational expenses (Industries, 2025). The selling of surplus heat to the district heating network can generate an extra revenue stream for ice rink owners, while district heating corporations can gain from an economical heat supply (Lind and Rundgren, 2017).

IV. **Efficient Resource Utilization**: Leveraging existing infrastructure minimizes capital expenditure compared to standalone systems.

V. **Environmental impact**: Utilizing waste heat reduces the need for additional heat production, potentially lowering greenhouse gas emissions associated with district heating (Pieper, 2019). Reduced Carbon Emissions through the substitution of other fuels for recovered heat aligns with Gothenburg's sustainability goals (Gummesson, 2014, Hayes, 2023)**.**

Potential Challenges

A few Challenges were identified in implementing such integrations:

I. **Temperature discrepancy:** The comparatively low temperature of waste heat generated by ice rinks frequently necessitates further enhancement to satisfy district heating standards (Lind and Rundgren, 2017).

II. **Seasonal Variations:** The cooling requirements of ice rinks may not consistently correspond with the heating needs of the district energy system, demanding meticulous planning and possible thermal storage options (Pieper, 2019).

III. **Economic viability:** The initial capital outlay for heat recovery and enhancement apparatus must be weighed against the prospective energy savings and income from heat sales (Lind and Rundgren, 2017).

Notwithstanding these limitations, the amalgamation of ice rinks with district energy systems signifies a viable strategy for enhancing energy efficiency and mitigating environmental effects in metropolitan locales. As district energy systems advance and pursue varied heat sources, ice rinks are expected to assume a progressively significant function in the energy framework of cities with frigid climates.

2.9 Case Studies

Successful integrations of ice rink refrigeration systems with district energy networks demonstrate significant efficiency gains and emissions reductions. (Hayes, 2023).

2.9.1 Angered Arena, Gothenburg:

Angered Arena exemplifies the successful integration of ice rink refrigeration with district energy systems. This project highlights the viability of mid-scale waste heat recovery in recreational facilities, providing a template for retrofitting. While not directly connected to Gothenburg's main district network, the arena's heat pump-based system demonstrates scalable principles for urban energy integration. (krahnspecialtyfluids, 2024)

Ice rink cooling Uses 11,000 liters of Temper -20 heat transfer fluid (non-toxic, biodegradable) with 30% lower viscosity than glycol, enabling smaller pumps and pipes. Heat recovery loop captures waste heat from refrigeration dry coolers (typically wasted) to warm the adjacent 25-meter swimming pool.

➢ ***Operational Outcomes***

- Energy savings**:** Direct heat transfer reduces swimming pool heating costs by 40–50% annually.
- Environmental impact**:** Eliminates 180 tons CO_2/year compared to conventional electric pool heating
- Reliability**:** Year-round operation maintained despite Gothenburg's temperature extremes (-10°C to 30°C).

2.9.2 Stockholm District Cooling Network

A Stockholm installation pairs a 2 MW transcritical CO_2 system with an NH_3-H_2O chiller, achieving (Yuwardi, 2013).

- Desorber input: 85°C from CO_2 gas cooler.
- Chiller output: 6°C chilled water for district cooling at COP 0.55.
- Net energy savings: 28% vs. separate systems.

2.9.3 Tallinn District Heating

A pilot project achieved 28% primary energy savings by integrating CO_2 waste heat (80°C) into a LiBr absorption chiller, preheating DH return flow from 55°C to 68°C (Nouri, 2020).

➢ ***Key outcomes:***

- Heat Recovery: 4.6 MW from a 6 MW CO_2 compressor.
- Emission Reduction: 2,100t CO_2/year avoided.

3 Methodology

This study adopts a case study research design, focusing exclusively on alternative refrigeration systems for the proposed ice rink on the rooftop of Rosenlundsverket in Gothenburg. The case study approach provides an in-depth examination of the technical and economic feasibility of integrating advanced refrigeration technologies with the existing district heating and cooling infrastructure at Rosenlundsverket. Importantly, this study does not include detailed construction or building design considerations except proposing PV films as the cover of shading over the roof footprint, assuming that placing an ice rink on the roof is structurally feasible.

The research design incorporates the following key elements:

1. **Literature Review**: A comprehensive analysis of existing research on refrigeration technologies for ice rinks, with a focus on CO_2-based systems and other environmentally friendly alternatives such as ammonia and absorption chillers. The review also examines best practices for integrating refrigeration systems with district energy networks.

2. **Site Analysis**: An evaluation of Rosenlundsverket's existing district cooling and heating infrastructure to assess its compatibility with the proposed refrigeration systems. This includes analyzing energy flows, available capacities, and potential integration points for heat recovery and cooling support.

3. **Conceptual Design Development**: Creation of preliminary designs for baseline refrigeration systems, based on A CO_2-based trans-critical compressor chiller as the primary focus due to its high energy efficiency and low Global Warming Potential (GWP), and then develop in accordance with Preliminary Design Concepts according to 3.3.2

4. **Energy Modeling and Process Integration**: Simulation tools will be utilized to model the energy performance of each proposed refrigeration system under various seasonal conditions. The analysis will explore optimal integration strategies with Rosenlundsverket's district heating and cooling networks, including waste heat utilization and load balancing.

5. **Economic Analysis**: Evaluation of capital costs, operational expenses, and potential revenue streams (e.g., selling recovered heat to the district heating network). The economic viability of each alternative will be assessed using metrics such as Net Present Value (NPV), Internal Rate of Return (IRR), and payback periods.

6. **Techo-economic Sensitivity analysis:** Quantification of potential energy savings, and economical benefits associated with each system alternative. The assessment will also consider compliance with potential stakeholder initiatives and goals.

The flowchart below (Figure 11Figure 11)outlines the structured steps taken from initial data gathering and site analysis to system modeling, economic assessment, and final design recommendations.

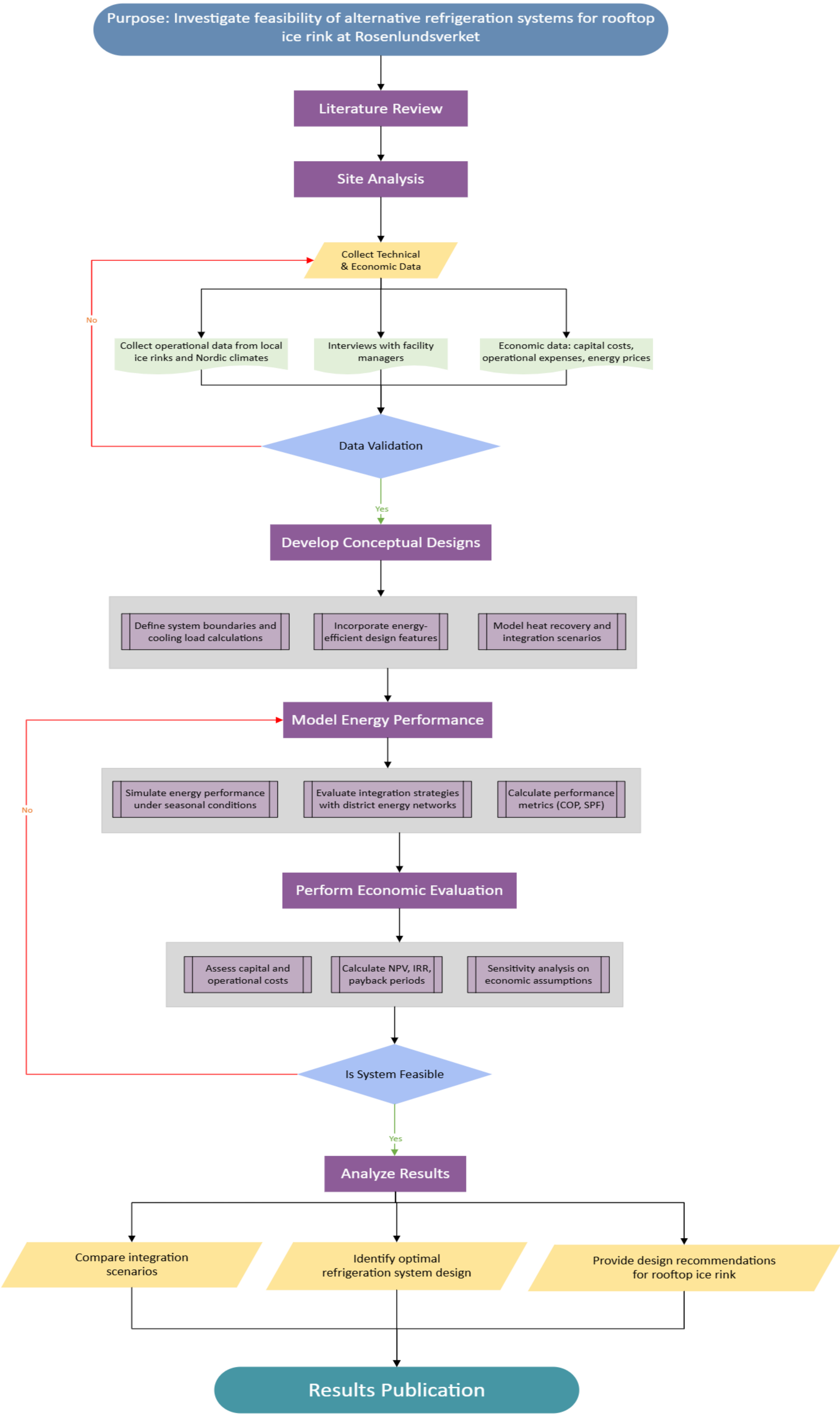


Figure 11-Methodology Flowchart

3.1 Data Collection Methods

The data collection process for this study will employ a multi-faceted approach to gather comprehensive information on ice rink technologies, district energy systems, and their integration potential. The following methods will be used:

3.1.1 Literature Review

An extensive review of academic literature, industry reports, and technical documents will be conducted to establish a strong theoretical foundation for the study. Key databases such as Scopus, Web of Science, and Google Scholar will be utilized to access peer-reviewed articles, conference proceedings, and technical reports. Search terms will include combinations of keywords such as "ice rink energy efficiency," "district heating and cooling," "waste heat recovery," and "CO2 refrigeration systems."

3.1.2 Site Analysis and Technical Data Gathering

In conjunction with several consultation meetings with Göteborg energy representatives, a detailed analysis of the Rosenlundsverket site in central Gothenburg was held to assess the feasibility of integrating an ice rink with the existing district energy infrastructure. This will involve:

- On-site inspections to evaluate the available roof space and structural capacity
- Review of existing district cooling machine configurations and capacities
- Analysis of current energy production and distribution systems
- Assessment of potential integration points for the proposed ice rink

3.1.3 Site Analysis and Constraints Assessment

The site analysis will evaluate Rosenlundsverket's roof space and its compatibility with the proposed refrigeration systems. This includes:

- Assessing the existing district heating and cooling infrastructure to identify integration points for heat recovery and cooling support.
- Reviewing local climate data to understand seasonal variations that influence system performance.
- Analyzing the energy flow capacities of Rosenlundsverket's oversized district heating and cooling networks, including river water availability for condenser cooling.

3.1.4 Case Study Data Collection

Data from existing ice-skating facilities in Gothenburg and similar Nordic climates will be collected to provide benchmarks and best practices. This will include:

- Energy consumption patterns and operational data from local ice rinks
- Interviews with facility managers to gather insights on energy-saving measures and challenges
- Collection of technical specifications and performance data of existing ice rink refrigeration systems
- Cooling Load Calculations: Estimation of ice rink cooling demands based on factors such as ambient conditions, usage patterns, and resurfacing schedules.

- Seasonal Performance Factor (SPF).

3.1.5 Economic Data Collection

To assess the economic viability of the project, the following data will be gathered:

- Capital cost estimates for equipment and infrastructure from manufacturers and industry databases
- Operational and maintenance cost data from similar facilities and utility providers, and the ASHRAE guideline.
- Revenue projections based on local market analysis and comparable facilities
- Energy price forecasts and incentive programs relevant to the Gothenburg area
- Interest rate, energy escalation rate, and inflation will be taken into account.

3.1.6 Limitations and Delimitations

Methodological Limitations

- Uncertainty about the future and a precise plan for the Rosenlundsverket infrastructure made it hard to predict the best system design. Thus, it was assumed this proposal was meant to be a considered part of the developing agenda of this facility.
- Constraints related to the ramp-up and ramp-down rates of cooling units are not considered (Zabian and Chan, 2022).
- The district cooling and district heating systems are treated as separate entities, and any interactions between them are disregarded in this analysis. (M. Jangsten.2022)
- Economic projections are based on current market conditions, which may change over time. Future energy prices for heat and electricity remain uncertain, potentially influencing the model's outcomes. In this study, the electricity price data is based on a rough estimation.

Data Limitations

- Limited availability of operational data from similar integrated systems may affect benchmarking accuracy.
- Historical climate data may not fully reflect future climate variability, impacting long-term performance predictions.

Technical Limitations

- Integration challenges related to compatibility with Rosenlundsverket's future district heating and cooling infrastructure, as the facility owner and energy distributor plan to refurbish the infrastructure at this station.

- Gothenburg's District Cooling system has been affected by Low Delta-T Syndrome, as identified by (Jangsten et al., 2020) with the temperature difference between supply and return water falling below the 10 °C design target, typically ranging between 6–8 °C. This issue is mainly caused by insufficient temperature separation in heat exchangers, excessive chilled water flow operating in the saturation zone, and seasonal mismatches. As a result, Rosenlundsverket experiences 18–22% higher annual pumping energy costs.

3.2 Rosenlundsverket Proposed Roof Top Ice Rink

Rosenlund has 1,400 m^2 of roof space for the city to build an ice rink on. Smaller skating rinks: For smaller public skating rinks, a common size is approximately 12.5 meters long and 8.5 meters wide, providing an area of about 106 m^2. (1,800 m^2 for a hockey rink)

Table 3-Roof Top Ice Rink Proposal

Ice Pad Size	**1,100 m^2**
Type	Recreational – Open Roof public skating
Anticipated Operational Seasonal length	12 months
Optimal Operational Length	Sep-April (Thanasoulas, 2018)
Cumulative absorption duty of Ice formation in addition to heat gain by the Ice surface	1428.92 KW
Energy efficiency practices	Hybrid system, Waste Heat Recovery ORC system

3.3 Conceptual Design Development

The conceptual design for integrating an ice rink into Rosenlundsverket's district energy infrastructure focuses exclusively on investigating and exploring several practical integration scenarios based on compatibility and flexibility for process integration through waste heat recovery from the Ice rink's refrigeration system, ensuring optimal energy efficiency and sustainability. The primary refrigeration focus is on CO_2-based systems and compressor chillers as reference refrigeration systems, as highlighted comprehensively within the literature review chapter 2.6.1.

3.3.1 Design Hypothesis

I. *Methodological assumptions:*

- The design process prioritizes system integration while excluding structural analysis or detailed building design.
- The study relies on simulation tools that simplify real-world complexities; results may vary under actual operating conditions.
- To simplify the analysis, this research will disregard network effects. It is assumed that all pipelines within the network have sufficient capacity to meet the required cooling demand.
- The operation of cooling units is independent of the previous hour's conditions.

- To calculate the ice pad's heat load externally through empirical formulas and relevant theories, all the variables and key design parameters inspired by the ASHRAE guideline, with Special Considerations for Gothenburg's climate.
- For simplification of Ice formation/depletion, considered as a steady-state heat sink.

II. *Key Temperature Considerations*

- ***Primary Reference:*** Mean Temperature Used for baseline monthly calculations (McQuiston et al., 2023, Taebnia et al., 2020). Represents average thermal conditions affecting vapor pressure differentials. Aligns with ASHRAE's load calculation methodologies for HVAC systems (Thevenard and Humphries, 2005).
- ***Peak Load criteria:*** Maximum monthly average high Temperature +2°C
  - Critical for sizing refrigeration capacity
  - Governs worst-case condensation scenarios (summer afternoon conditions)
  - Combined with 85-95% RH values from historical data (Pomerancevs et al., 2020)

Figure 12 represents future plans designs for Rosenlundsverket .

Figure 12-Render of Future Design

3.3.2 Preliminary Design Concepts

Preliminary designs will focus on refrigeration system integration rather than structural elements. Key considerations include:

- **Outline the System Boundaries**, focusing on finding the proper refrigeration cycle size and capacity. (Calculate the needed cooling effect for an open-roof Ice rink)
- **Refrigeration System Alternatives**: Developing concepts for CO_2-based trans-critical systems (focus).
- **Heat Recovery**: Define the baseline system potential for maximum waste heat utilization.
- **Energy Efficiency**: Incorporating practical and feasible design practices in real-world operation by including energy-efficient features to enhance the baseline refrigeration system into enhanced models. (adding at least 2 practical energy efficiency promoter solutions for a standalone refrigeration system, like parallel comparison, Flash Gas Bypass with internal HX, ejector, Ice storage, etc.)
- Define and configure the developed model as the reference system prior to integration.

- Conduct the performance analysis over the stand-alone reference to investigate for maximum waste heat recovery, before employing the integration according to the Selected Scenarios. (4 Scenarios)

3.3.3 Cooling System Design (Case study definition)

The cooling system design will be initiated by establishing a baseline stand-alone default CO2 transcritical refrigeration system; then, a preliminary design will be developed to reach the appropriate reference system. Through a comparative study of the most viable integration alternatives, the best-integrated system Design was identified and considered as a reference system.

I. **Baseline System Definition and Boundaries:**

- incorporates empirical correlations from Salib (2021) and ASHRAE standards, with open-roof adjustments per Karampour (2011) to define the system boundary in terms of needed cooling effect and capacity.
- Define the COP, COSP, Win, and Q_H of the baseline system.

II. **Reference System Definition based solely on the compressor chiller**

- A developed system has been modeled via Danfoss coolselectoe2® incorporates advanced design practice enhanced by feasible energy-efficient features based on the Transcritical CO_2 refrigeration system performance (Danfoss Climate Solutions, 2023).
- Establishing performance benchmarks such as Coefficient Performance (COP) and COSP to identify the most appropriate selection. SPF is more representative of real-world efficiency than COP, which is measured under ideal, fixed conditions (Energiamegújítás, 2025).

III. **Integrated Conceptual System Design**

- Developed the default system based on Integration Scenarios into an advanced model,
- Evaluating dynamic operation scenarios to optimize load balancing between the ice rink and district energy networks.
- Conducted performance benchmarks again to track the escalation in electrical (COP/COSP) Coefficient of Performance, (SEER) seasonal energy efficiency ratio as well as electrical Seasonal Performance Factor (SPF) to identify the most efficient scenario of Integration.

IV. **Selected Integration Scenarios based on the compressor chiller**

Among a bunch of different investigated alternatives, considering the enclosed potential infrastructure, excluding the reference scenario 6 of the most practical and feasible Scenarios were nominated to be developed as below. A series of different temperature-level (High-Mid-low) desuperheaters proportional to operational pressure and available heat applied before the gas cooler, which achieves temperature reduction by injecting a cooling medium (often water) into the flow of superheated gas or steam. The water absorbs heat and evaporates, cooling the gas down to near its saturation point, while the evaporated water can be used in the preheating application.

Reference Scenario:

Transcritical R744 Transcritical chiller + chilled water from cooling Tower and 4 primary scenarios proposed based on Solely compressor chiller:

1st Scenario: Compressor Chiller + WHR (Cascade to DC)

Utilize the excess heat rejected from the gas cooler (condenser) to serve the absorbent Cycle (desorber) of the District Cooling Cycle. (Within warm season)

Rosenlundsverket's district cooling system can serve as the primary heat sink for the CO_2 gas cooler, enabling efficient waste heat rejection while simultaneously driving absorption chillers. In this setup, low-grade waste heat from the gas cooler powers the generator of the absorption cycle, where the refrigerant is evaporated and the absorbent concentrated. The resulting vapor condenses, releasing heat, while the regenerated solution is throttled to the absorber to take up vapor from the evaporator before being recirculated to the generator. This closed-loop process sustains absorption chiller operation, with the generator and condenser operating at high pressure and the evaporator and absorber at low pressure. In warmer seasons, district cooling helps manage peak loads while the recovered waste heat supports absorption chiller demand (Omar Zabian, 2022; Fernqvist et al., 2023).

2nd Scenario: Compressor Chiller + WHR (Combined to DH)

Utilize the excess heat rejected from the gas cooler through a single or a series of desuperheaters to precondition the return line of the DH network before feeding it into the chiller. (preferably Within Mid-season and colder operation time).

3rd Scenario: Compressor Chiller + Direct River cooling

Hybrid ORC system. (Could be regulated appropriately with the time of year and surplus thermal energy)

4th Scenario: Compressor Chiller +ORC

Direct cooling the gas cooler through the Göta Canal.

2 advances Scenario introduced based on the Hybrid chiller arrangement

5th Scenario :(COMBO) Ammonia-water Absorption chiller + CO_2 compressor chiller

Conceptual COMBO Scenario for Hybrid system 2.6.4 which is not solely based on a compressor chiller, considering recovering the surplus heat from industry on the supply side to drive the absorption chiller as a base load, and leveraging a reference CO2 Chiller system as the speaker.

6th Scenario: COMBO + DHW (local Heating)

Conceptual Scenario according to previous arrangement (COMBO), while utilizing the waste heat rejected from the system through a series of desuperheaters to either serve the Domestic hot water (DHW) or Heat recovery to cover the potential Rink's heating demand for procurement and start-up heating (subfloor heating, Snow melting).

3.3.4 Meteorological Data on a Monthly Basis

Error! Reference source not found. In Appendix 1, summarizing monthly values for solar radiation, wind speed, temperature, and precipitation in Gothenburg, Sweden (SMHI, 2025, Bekiaris et al., 2005).

- Temperature profile, relative humidity, Wind speed and precipitation data are gathered based on monthly averages from the SMHI database recorded at the closest weather station (Säve station).
- Solar radiation data is derived from SMHI STRÅNG model datasets (2015–2024) and consistent with NASA-compatible methodologies for Rosenlundsverket, Gothenburg coordination (57.702792°N, 11.954889°E).

Based on available data from SMHI and long-term water temperature records for the Gothenburg area (Göta river, mouth at Gothenburg), the monthly mean water temperatures are as follows (See appendix 1, **Error! Reference source not found.**):

- The warmest month is August, with average maximum temperatures around 18.5–20.7°C.
- The coldest month is March, with average minimum temperatures around 0.3–2.6°C

Nearest SMHI Weather Station

The Swedish Meteorological and Hydrological Institute (SMHI) runs multiple stations near Gothenburg. Although none are located directly at Rosenlundsverket, the Säve station (located ~10 km northwest of the city center), which is derived from 30 years of observations (1992–2021), is often used as the reference source. It reliably represents central Gothenburg and the Södra Älvstranden area in scientific and engineering studies (Nilsson and Kryh, 2012).

Table 4- SMHI station specification

Weather Station	Approximate Distance to Rosenlundsverket	Typical Collected Local Data	Reference
Säve	~10 km NW	Wind, temperature, and climate	Used in studies for central Gothenburg

3.3.5 Climatic Data Adaptation

The Climatic meteorological data has been adapted Proportional to the Rooftop Ice Rink's Elevation from Ground Reference and Shading:

I. Wind Speed adjustment:

Considering the characteristics of designated site and the rink location on the rooftop of the 6th floor, which can interpret it around 20 m height from the ground reference base on Swedish building code, Since the conventional and standard height for installing an anemometer to measure average wind speed in the city climate station is commonly 10 meters above ground level, Ice rink surface supposed to be in 20 m elevation, an adjustment and correction on reported value needs to applied to get precise value for our load calculation.

This adjustment could be conducted in two different ways, whether through a rough estimation of average Wind Speed at 20 m elevation using the power law to 10-meter data, or using a reasonable wind shear exponent for the local terrain (see Equation 4).

Equation 4

$$v20 = v10\left(\frac{20}{10}\right)^{\alpha}$$

Where:

- v_{10} is the average wind speed at 10 meters.
- α is the wind shear exponent known as the Hellman exponent, typically ranging from 0.14 (open terrain) to 0.22 (urban/forested areas).

For Gothenburg, a value of α around 0.20 is reasonable due to its urban and coastal character (Simiu and Scanlan, 1996)

Alternatively, using the marine forecasters and data sources for wind speed, 20-meter height, closer to typical ship anemometers. This adjustment could be performed to compare marine data modelers vs published data by the weather station, specifically for coastal areas like Gothenburg. (Aponte-Roa et al., 2018).

Example Calculation: If the average wind speed at 10 meters is 4 m/s (a typical value for Gothenburg coastal areas): v_{20}= 4× $(2)^{0.20}$ ≈ 4 ×1.14 = 4.59 m/s ≈ 4.6 m/s

II. Temperature depletion by altitude:

To adjust the extracted temperature values of Gothenburg to an 18- 20-meter elevation difference from ground level, the environmental lapse rate method was applied, which describes how temperature decreases with altitude. The standard lapse rate in the troposphere is 6.5°C per 1,000 meters (or 0.65°C per 100 meters) (López-Moreno et al., 2019) , In the context of Gothenburg, the lapse rate may be closer to 6°C/km (moist adiabatic rate)

For small elevation changes (e.g., 20 meters), the temperature adjustment can be calculated as (see Equation 5):

Equation 5

$$\Delta T = Lapse\ rate \times \frac{\Delta elevation}{1000}$$

The standard elevation for recording the temperature profile at SMHI (Swedish Meteorological and Hydrological Institute) weather stations is 2 meters above ground level (see Equation 6) (Petersson, 2014).

Equation 6

$\Delta_{elevation}$ = 18 m So: ΔT = (6.0). 18/1000 = 0.108

As a rule of thumb, for most practical purposes (e.g., urban planning, climate studies), this minor correction (~0.1°C) is negligible.

III. Impact of Shading on Radiation:

Within this proposal, protective solar shading is considered, whether as fixed shading or convertible (maximizing the sky cooling effect at night), as part of the preliminary plan to minimize the heat gain radiation on the Ice surface.

Shading effect reduces solar gains by 30–45 kW (per 1100 m^2 ice sheet). This corresponds to a 49–50% reduction in solar heat load during peak summer (Maleki, 2011). For a Coastal Climate like Gothenburg, High humidity (85–97% RH in summer) amplifies condensation heat gains, making shading critical despite radiative trade-offs through sky radiative cooling at night.

Roof Shading Potential Benefits proportional to η_{hading} (Shading Efficiency)

Protective roofs (e.g., retractable or photovoltaic systems) reduce solar loads by:

- Direct shading: Blocking 30–50% of incident radiation
- Albedo enhancement**:** Reflecting 50–80% of the remaining sunlight

Table 5-Example of shading impact

Condition	Min Solar Gain (Wh/m²/day)	Max Solar Gain (Wh/m²/day)
Unshaded ice	350	5400
$\eta_{shading} = 0.5$	175	2700

Seasonal Variation

- Winter irradiance: 0.35 kWh/m^2/day (minimal solar gain)
- Spring/autumn**:** 2.6–5.4 kWh/m^2/day (moderate loads)

Protective shading provides the greatest benefit during summer, reducing refrigeration demands by 18–25% when combined with convective mitigation strategies. (Dubois, 1997).

Photovoltaic Films as a shading surface have been one of the innovative features proposed as part of the Rooftop design of the Ice rink. Midsummer BOLD panels are ultra-light (2.9 kg/m^2), thin (2 mm), and flexible (0.25 m bend radius). Made from CIGS cells, they yield ~116–123 W/m^2 (~12% efficiency), are fully black/opaque, and are suitable for curved or flat roofs. Certified for outdoor use with a 30-year warranty. It was assumed that almost one-third of the ceiling would be covered by PV panels, and the rest by convertible transparent glass. The technical assumption of PV installation is summarized in Annex VI.

IV. Impact of using protective windbreaker pillar (barriers)

As the maximum design Air velocity over the ice surface needs to be regulated up to 0.5 m/s to eliminate the wind exposure according to ASHRAE higher value exceed this limit would leads to higher energy loss, and more refrigeration compression is required to compensate subsequently, so the optimal nominal V_{wind} needs not to exceed its higher allowed cap of 0.5 m/s, that means for a city like Gothenburg the wind barriers needs to be design by architectural features in such a way to eliminate the wind penetration up to 90% on Ice surface to make the system energy efficient for operation otherwise the wind on rooftop could be one of the greatest challenges in practical.

In contrast, the maximum recorded effectiveness for windbreaks in terms of wind speed reduction can reach up to 73.7% under optimal conditions. This was achieved using a solid windbreak configuration (minimal porosity), specifically with a 2-meter-high windbreak placed at an optimal distance from the protected area (Hashemi Monfared et al., 2019). This also led to a 74.1% reduction in evaporation, demonstrating the strong link between wind speed reduction and its practical benefits. Therefore, the design wind speed is considered based on a 70% reduction in reference values to be closer to reality.

3.4 Step-by-Step Method for the calculation of the effective cooling load

This model incorporates empirical correlations from Salib (2021) and ASHRAE standards, with open-roof adjustments per Karampour (2011). The CO_2 system performance factors had been taken into account based on Scandinavian case studies.

3.4.1 Initial One-Time Freezing Load (Ice Formation)

Use the time-to-freeze method to estimate the refrigeration system capacity (see Equation 7) (Mazzotti, 2013, Salib, 2021):

Equation 7

$$Q_{\text{freezing}} = C_{\text{loss}} \cdot \rho \cdot V_W [Cp_{,w} T_W + q_{\text{fusion}} + Cp_{,\text{ice}} (0 - T_{\text{set}})] \;/\; H_{\text{freeze}} \cdot SecInHour$$

- V_W: Volume of feed water to freeze, Feed Water Volume=Area×Thickness (*1100 m^2 rink with 25 mm ice thickness requires **27,500 liters** of feed water.*)
- T_{set}: Ice surface temperature setpoint (e.g −3°C for recreational skating rink)(P.hemati)
- $T_{feed,W}$: The feed water temperature of Ice formation: Nordic Setting 40 °C (M. Karampour, 2011)
- $Q_{freezing}$, is the capacity of the refrigeration system. (J/s; Btu/s)
- $Cp_{,w}$ is the specific heat of water. (kJ/(kg.K);Btu/lb.°F)
- Q_{fusion} is the latent heat of the freezing water (334 kJ/kg or 144 BTU/lb)
- Cp_{ice}, is the specific heat of ice. (kJ/(kg.K);Btu/lb.°F)
- H_{freeze} is the desired freezing time. (hrs to freeze the water) **48-72 hrs for 25mm** ice
- C_{loss}, is a correction coefficient to account for any thermal losses. (C_{loss}: 1.5 (air exposure adjustment according to Willem Mazzotti ,2013)
- $SecInHour$, number of seconds in an hour 3600 S)

ASHRAE emphasizes:

Distribute the ice formation volume across multiple thin flooding cycles (typically 1–3 mm per layer) to ensure optimal ice quality.

- Applying thin layers of water (400–700 liters per flood for a standard rink)
- Using the lowest practical water temperature to reduce refrigeration load
- Ensuring each layer freezes within approximately 15–20 minutes for optimal ice formation and energy use (Karampour, 2011).

According to the ASHRAE guideline for the one-millimeter thickness of Ice, for an Ice area of1100 square meters, and 500 liters of feed water volume per flood, we need 2.2 flooding cycle for each 1 mm ice layer, multiplied by 25 mm we find that need at least 55 flooding cycles for entire ice formation. If each cycle takes time for Flooding duration: 15–30 minutes per cycle and Freezing time for each thin layer should freeze within 15–20 minutes at suitable temperatures (around –

4°C or colder), it will take roughly about +50 to 55 min for each cycle, we have 55 flooding cycles. Hence, we need approximately 3025 min, equal to 50 hours, as H_{freeze} as a rough estimation.

3.4.2 Solar Radiation Gain

The solar heat load on ice surfaces is calculated by incorporating solar radiation as an added heat load according to (profileSOLAR.com, 2025):

As mentioned in 3.3.4, protective solar shading is included in the initial plan to reduce heat radiation affecting the ice surface, so the shading effect on mean irradiation values needs to be applied with a 50% mitigation (see Equation 8).

Equation 8

$$Q_{\text{solar}} = (1 - \eta_{shading})\alpha \cdot G_{\text{solar}} \cdot A_{\text{ice}}$$

α: Absorptivity of ice (typically 0.1−0.3)

G_{solar}: Incident solar irradiance (W/m^2)

A_{ice}: Ice surface area

η_{shading}: Shading Efficiency ≈ 0.6–0.9 for typical systems (Maleki, 2011, Dubois, 1997).

For Rosenlundsverket, Gothenburg coordination (57.702792°N, 11.954889°E) conditions:

Minimum solar gain: 0.35 KWh/m^2/day

Maximum solar gain: 5.4 KWh/m^2/day

3.4.3 Lighting Radiation Sensible Heat Gain

Lighting in ice rinks is essential for creating a comfortable and functional environment, particularly during the winter months when daylight is limited. The required lighting intensity varies depending on the specific area within the facility and the nature of the activity taking place on the ice. As stated by the IIHF Ice Rink Guide, the recommended minimum illuminance level for recreational skating rinks is 300 lux. ASHRAE guidelines suggest assuming 15–25% of lighting heat is latent (depending on humidity). However, in ice rinks, latent effects are minimal due to low ambient moisture, and when it comes to an open roof scenario, they could be negligible for preliminary design.

3.4.4 Wind-Driven Convective Heat Gain

To estimate the convection and condensation heat load on ice, it is necessary to know the wind speed of the air very close to the ice surface (Khalid, 2012). Based on the adjusted value of Wind in accordance to Equation 10 a wind-speed-dependent convection coefficient (h_c) is used for the calculation of Wind-Driven Convective Heat Gain, according toEquation 9:

- h_c=5.7+3.8$\cdot v_{\text{wind}}$ (ASHRAE correlation for horizontal surfaces, v_{wind} in m/s).

Equation 9

$$Q_{\text{conv}} = h_C \cdot A_{\text{ice}} \cdot (T_{\text{air}} - T_{\text{ice}})$$

Wind Speed Adjustment close to the Ice source

ASHRAE-compliant calculations require adjusting wind speed measurements enclosure of the ice surface. For example, if the local meteorological station reports 5.0 m/s at 10 m elevation, the speed at 0.1 m above ice (typical for load calculations) can be estimated using the logarithmic wind profile (Khalid et al., 2019):

Equation 10

$$V_{ice} = V_{\mathrm{ref}} \times \frac{ln\,(\frac{Z_{Ice}}{Z_0})}{ln(Z_{ref}/Z_0)}$$

- Z_0 (roughness length) ≈ 0.01–0.05 m for smooth ice surfaces.
- Example: V_{ref}=5.0 m/s at 10 m → V_{ice}≈2.2 m/s at 0.1 m.

Design Recommendations

Air velocity over ice should ideally be kept below 0.5 m/s to eliminate the convective loads, as higher speeds drastically increase refrigeration demands (Dietrich & MacAvoy, 1980). For open-roof rinks, windbreaks or architectural features (e.g., partial walls, strategic landscaping) are recommended to reduce ambient wind exposure (Khalid, 2012).

Within the calculation, it was assumed there would be a windbreaker with an effectiveness factor of 70% in order to protect the ice surface from harsh winds, then the design V_{ice} considered was 0.5 m/s.

3.4.5 Sky Radiative Cooling

Account for longwave radiation exchange with the sky (see Equation 11):

Equation 11

$$Q_{\mathrm{sky}}=\psi.\epsilon\cdot\sigma\cdot A_{\mathrm{ice}}\left(T_{\mathrm{ice}}{}^4 - T_{\mathrm{sky}}{}^4\right)$$

According to (Swinbank model for clear skies) $T_{\mathrm{sky}} \approx 0.0552\cdot T_{\mathrm{air}}{}^{1.5}$

- ϵ: Emissivity of ice

For open-roof ice rinks in coastal areas like Gothenburg, the optimal ice emissivity value for salt contamination should be adjusted to ϵ = 0.93 during high-season salt deposition (summer) and ϵ =0.95 for winter. Salt crystals reduce effective emissivity by 2–4% through:

- Surface roughness scattering (1.2–1.8% reduction)
- Brine inclusion absorption (0.8–1.5% reduction)
- σ=5.67×10^{-8} W/m^2·K^4 (Stefan-Boltzmann constant)
- Effective sky view factor (ψ = 0.3–0.7 for common shading systems.

Night radiative cooling offsets ~5-15% of daytime heat loads due to clear skies without shading (Holzlöhner et al., 2021) The impact of protective shading on sky radiative cooling. Result: 20–50% reduction in nocturnal cooling capacity; this effect constantly helps balance the cooling system design in the context of CO_2 trans-critical chillers. Its value is universally consistent across physics and HVAC calculations.

3.4.6 Condensation effect (convective mass transfer)

The heat transfer resulting from mass transfer, specifically due to the latent heat released during the condensation of water vapor (a form of convective mass transfer), is accompanied by heat exchange with the surface where the condensation takes place (Khalid, 2012). The primary driving mechanism for the condensation of water vapor is the difference between the partial pressure of water vapor in the air adjacent to the ice surface and the saturation pressure corresponding to the temperature at the top surface of the ice.

The fundamental equation governing heat transfer during condensation resembles that of convective heat transfer, with the key distinction being the use of a condensation heat transfer coefficient in place of the convective counterpart. Condensation, occasionally referred to as diffusion, can be quantified using the following Equation. In summer, condensation becomes another significant heat load, accounting for about 10% of the total (see Equation 12) (Karampour, 2011).

Equation 12

$$Q_{diffusion} = \alpha_d A_{ice} \, (t_{air} - t_{ice})$$

According to(Melinder and Granryd, 2005):

α_d is the diffusion/condensation heat transfer coefficient *[W/m2. K]*, which can be calculated by

- $\alpha_d = 1740\ \alpha_C\ (\Delta P/\Delta T)$
- α_c = 3.41 + 3.55V
- ΔP: differential pressure of the water vapor in the air and the air in the boundary layer enclosure to the ice surface.
- ΔT: temperature difference between the ambient air temperature and T_{air} on the boundary layer to ice surface

α_c is the convective mass transfer coefficient, representing the ability of a surface to transfer heat during the condensation of vapor. It can be calculated by an experimental equation (ASHRAE, 2010) which uses the air velocity over ice to take into consideration both natural and forced convection, expressed in units of $W/(m^2 \cdot K)$.

For an open-air rink adjustment, we use a logarithmic wind profile, similar to Equation 10, to estimate the air velocity over the ice surface as a basis for our calculation:

- α_c = 3.41 + 3.55(0.4 to 0.5) = 5.0 to 5.2 $W/(m^2 \cdot K)$

This aligns with the value of 5 $W/(m^2 \cdot K)$ cited in research for similar air velocity conditions (Karampour, 2011).

- $\Delta P = \varphi\ P_1 - P_2\ (8)$ in *K/bar, bar*
- φ air relative humidity of Gothenburg on a monthly basis
- P_1 and P_2 *water vapor saturation pressures in air and on the ice top surface. [bar]*

$$\boldsymbol{P_2 = e^{\left(17.391 - \frac{6142.83}{273.15 + t_{ice}}\right)}}, \qquad \boldsymbol{P_1 = e^{\left(12.03 - \frac{4025}{235 + t_{Air}}\right)}}$$

When the ice surface temperature is below 0 ˚ C, the deposits form as ice. Therefore, the constant value is set to 1740, which accounts for both the latent heat released during the condensation of water vapor into frost and the latent heat involved in the freezing of water into ice (Melinder and Granryd, 2005).

3.4.7 Resurfacing Load

Ice resurfacing contributes 11–17% of total refrigeration loads in Swedish ice rinks (Kaya, 2017). (For a typical rink with a 1,000 MWh/year total energy use, resurfacing accounts for ~110–170 MWh/year (≈ 12.6–19.4 kW average annual load). Include latent heat from hot water applied during resurfacing. Studies have shown that periods with ice resurfacing can result in 30% higher cooling demand compared to periods without resurfacing. (Karampour, 2011). This substantial increase in cooling demand highlights the importance of optimizing the resurfacing process to improve energy efficiency in ice rinks (see Equation 13).

Equation 13

$$Q_{\text{resurfacing}} = \dot{m}_{\text{water}}[Cp_{,\text{w}}(T_{\text{water}} - 0) + q_{\text{fusion}} + Cp_{\text{ice}}(0 - T_{\text{ice}})]$$

or by (ASHRAE,2010): $Q_{resurfacing} = 1000\ V_f\,[4.2\,(T_{\text{water}} - 0) + 334 + 2(0 - T_{\text{ice}})] * \sum f_{\text{Correction}}$

- **Resurfacing frequency**: The number of resurfacing events directly scales the total daily/monthly refrigeration load (4-6 times/day). [A.Salib,2016]. For example:
- **Daily load** = $Q_{\text{resurfacing}}$ × Number of resurfacing events/days

ASHRAE Recommendations regarding the Resurfacing Frequency would be as below:

- **Winter (Low Use)**: 4–6 resurfacings/day (≈ every 4 hours).
- **Summer (High Use)**: 8–12 resurfacings/day (≈ every 2 hours).

- $Q_{resurfacing}$ Heat load due to resurfacing (kW or BTU/hr)
- $\dot{m}_{water}$: Mass flow rate of resurfacing water (typically 0.5−1.0 L/m²)
- $Cp_{,w}$: Specific heat capacity of water (4.18 kJ/kg\cdotpK or 1 BTU/lb\cdotp°F)
- T_{water}: Temperature of the hot water applied during resurfacing (°C or °F)
- T_{ice}: Temperature of the Ice surface
- $Cp_{,\,ice}$: Specific heat capacity of Ice (2.0 kJ/kg
- Q_{fusion}: Latent heat of fusion for water (334 kJ/kg or 144 BTU/lb)
- V_f = flood water volume (m^3)
- ρ = Density of water (kg/m^3)
- f = Correction factors (typically 1.1-1.3 to account for additional heat transfer mechanisms)

Water Volume =($\dot{m}_{\text{water}}$)L/m² × 1100 m² = L

The ice resurfacing process typically involves applying a thin layer of warm water (30-60°C) to the ice surface, which then freezes to create a smooth skating surface. This process introduces a considerable amount of heat into the ice rink system. According to (Karampour, 2011)Ice resurfacing accounts for approximately 14% of the total heat load in a typical ice rink during winter conditions.

3.4.8 Model Adjustments for Open-Roof Design

For open ice rinks, resurfacing feed water flow rates should be adjusted based on outdoor temperature to ensure proper ice formation. Temperature-driven feed Water Flow Adjustments need to be implemented in accordance with the following criteria for a standard-size ice rink (Kaya, 2017):

- **Colder temperatures (<–5°C):** Higher flow rates (~0.3–0.4 L/m^2) are feasible due to faster freezing, allowing thicker layers (1–2 mm) per pass.
- **Warmer temperatures (0°C to –5°C):** Reduce flow rates to ~0.1–0.2 L/m^2 to prevent pooling or partial melting. Thinner layers (0.5–1 mm) freeze more reliably.

The heat load from ice resurfacing can be estimated using rules of thumb and correction factors as well. The HVAC Rule of Thumb Handbook (VEDAUARZ, 2007) states that this calculation offers a foundational estimate that may be modified according to particular rink conditions and resurfacing methods. In open-roof ice rinks, supplementary factors must be taken into account when assessing the heat load resulting from ice resurfacing. Although research on open-roof ice rinks is few, certain overarching themes may be relevant:

I. **Solar radiation:** Direct sunlight can significantly increase the heat load on the ice surface, especially during resurfacing when the water is exposed (Bellache et al., 2005).

II. **Wind effects:** Wind can increase evaporation rates and affect the freezing process of resurfacing water (Wang et al., 2024)

III. **Ambient temperature fluctuations:** Open-roof rinks are more susceptible to changes in outdoor temperature, which can impact the resurfacing process and subsequent heat load (Daoud et al., 2008).

To account for these factors in the Rosenlundsverket location in Göteborg, Sweden, additional correction factors should be applied to the indoor calculation method. Through the generalization of the theories mentioned, we reach this equation (see Equation 14):

Equation 14

$$Q_{\text{resurfacing}} = \dot{m}_{water}\left[Cp_{,w}(T_{\text{water}} - 0) + q_{\text{fusion}} + Cp_{\text{ice}}(0 - T_{\text{ice}})\right] * (f_{\text{solar}} * f_{\text{wind}} * f_{\text{temp}})$$

Where:

- f_{solar} = Solar radiation correction factor
- f_{wind} = Wind effect correction factor
- f_{temp} = Temperature fluctuation correction factor, known as $f_{\text{diffusion}}$(Condensation due to humidity change subsequent to Temperature fluctuations)

These correction factors would need to be calculated through adjustment according to Gothenburg Climatic statistical data for Seasonal variations in solar radiation and wind patterns from external data sources.

Open-Roof Corrections reference

Add adjustment factors for Gothenburg's coastal climate:

Table 6-Correction factors Definition for Gbg costal climate

Factor	***Formula***	***Example Value***
f_{solar}	=1 + (5.2 / Mean Solar $_{Irradiation}$) *0.3	1.57(July)
f_{Wind}	=1 + (4.31 / Mean Wind$_{speed}$) *0.2	1.23 (August)
f_{temp}	=1 + (18.8 / Mean T_{air}) *0.15	1.3 (July)
Gothenburg's coastal climate		
Mean Wind speed (V_{wind})	4.39	m/s
Mean dry bulb temp (T_{air})	9.28	°C
Mean Solar irradiance (G_{solar})	2.80	kWh/m2/day

3.4.9 Q $_{(Misc.\ loads)}$ inclusion according to the rule of thumb (ASHRAE)

For ice rink cooling load calculations per ASHRAE guidelines, miscellaneous heat loads (Q_{misc}) are typically allocated 15-20% of the total calculated refrigeration load as a rule of thumb. It could also be forecasted through the equation below (see Equation 15) (Girip et al., 2023b, Joudi and Hussien, 2015):

Equation 15

$$\boldsymbol{Q_{misc,}} = \left[\left(\boldsymbol{Q_{Sum}} \times \mathbf{0.15}\right).\left(\mathbf{1} + \mathbf{0.05} \times \left(\boldsymbol{T_{avg}}/\mathbf{10}\right)\right]$$

This allocation accounts for unmeasured or complex-to-model factors like:

Table 7_ Q, misc. allocation in context of Ice Rink

Factor	**Contribution**	**Load Impact**
Pump work	15–25 kW	contribution from brine circulation pumps
Ground conduction	2–5%	of total load through insulation layers
Header pipe losses	3–8%	from uninsulated brine headers
Snow melt systems	5–12 kW	for ice surface maintenance
Electrical equipment	3–5%	on scoreboards/Zambonis

3.4.10 Total absorption heat gain from by ice pad

Summary of all components, see Equation 16, Equation 17:

Equation 16

$$Q_{total,} = [Q_{freezing} + Q_{Con,} + Q_{\mathrm{diffusion},} + (Q_{solar,} - Q_{Sky,}) + Q_{\mathrm{resurfacing},} + Q_{misc,}]$$

3.4.11 CO_2 System Adjustment

Account for trans-critical CO_2 system efficiency:

Equation 17

$$\text{Final Capacity } = \left[Q_{total,} \,/\, COP\right]$$

Where: COP = 2.8-3.2 (typical for CO_2 systems in Scandinavian climate).

Seasonal COP Patterns in Gothenburg

Gothenburg's climate features cold winters (avg. −0.4°C in February) and mild summers (avg. 16.6°C in July). Seasonal COP Data trends breakdown, followed by table 1x (Fricke et al., 2016):

Table 8_COP data breakdown proportional to Göteborg

Season	Avg. Temp (°C)	Estimated COP	Key Factors
Winter	−3 to 2	4.0–4.5	Low compressor work, optimal ΔT
Spring/Autumn	1 to 16	3.5–4.0	Moderate pressure optimization
Summer	12 to 20	2.5–3.5	Reduced heat rejection efficiency

Efficiency Drivers:

- Low ambient temperatures allow efficient heat rejection, reducing compressor workload.
- Moderate temps enable stable operation near optimal gas cooler pressures (~90–95 bar)
- Higher temps force the system into less efficient transcritical modes, with COP declines of ~15–20% compared to winter.

By prioritizing cooling system efficiency and minimizing overall heat loads, ice rink managers can substantially lower their energy expenses while preserving optimal ice quality.

3.4.12 Calculation of the Nominal Duty of the Evaporator

The evaporator duty must account for:

- Ice pad heat absorption (already calculated through empirical theory)
- Distribution system losses (e.g., heat gains in pipes, pumps, and heat exchangers)

The total cooling capacity (Q_{total}) for indirect arrangement is given by (see Equation 18):

Equation 18

$$Q_{\mathrm{total}} = Q_{\mathrm{ice}} + Q_{\mathrm{dist,Secondary}}$$

For indirect cooling systems (e.g., aqua ammonia secondary loops), empirical data (M. Karampour,2011) incorporate cross-reference with ASHRAE Standard 90.1 for insulation, suggesting distribution losses of ~16% of the total cooling load (Q_{ice}) during system dimensioning (see Equation 19).

Equation 19

$$Q_{dist,Secondary} = Q_{ice} .(0.16)$$

3.4.13 Secondary coolant (Aqua Ammonia) Parameters

Experimental data for ammonia-based secondary systems show 9–27% lower heat transfer coefficients in updated models, but this has minimal impact on COP, The evaporator duty depends on the secondary fluid's heat transfer as Equation 20 (Karampour, 2011, Kilberg, 2020):

Equation 20

$$Q_{evap} = \rho \; . \; \dot{m}_{secondary} \; . Cp \; . \Delta T$$

Where;

- $\dot{m}_{secondary}$: Mass flow rate of aqua ammonia
- CpSpecific heat capacity of aqueous ammonia is approximately 3.8 (NH_3 40%, T=-10)
- ρ_{NH3} : 955 Kg/m^3 witin temperature range -5 to -10
- ΔT: Temperature difference across the secondary circuit

3.4.14 CO_2 Evaporator Sizing

For the primary CO_2 system, the evaporator duty is influenced by:

- Log mean temperature difference (LMTD) between CO_2 and the secondary fluid
- Heat transfer coefficients (impacted by fouling, flow arrangement, and phase change)

Figure 13 _Brazed plate heat exchangers by Alfa Laval are commonly used in CO_2 chillers

CO_2 transcritical refrigeration systems are sensitive to ambient temperature and discharge pressure fluctuations. A safety factor of 10–20% is typically recommended to add to the Maximum calculated Q_l equal to $Q_{l,\ Adjusted}$ 663 (kW), account for load variations and transient conditions for Heat exchanger dimensioning and product development (see Equation 21):

Equation 21

$$Q_{adjusted\_\ evap} = U \cdot A \cdot LMTD$$

ASHRAE Standard 90.1-2022 Addendum M states that plate-type *liquid-to-liquid heat exchangers* (see Figure 13) *must be rated in accordance with AHRI 400* (for I-P units) or AHRI 401 (for SI units) (see Equation 22) (Tomlinson, 2017).

Equation 22

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

- $\Delta T_1 = T_{hot,\,in} - T_{Cold,out}$ and $\Delta T_2 = T_{hot,\,out} - T_{Cold,\,in}$
- U is the overall heat transfer coefficient. ≈ 1500 W/m^2°C (Gas-Liquid)
- A is the evaporator surface area.

In heat exchanger sizing, increasing mass flow rate reduces Δ*T*, which lowers LMTD and minimizes the heat transfer area. However, this can be mitigated by improved turbulence, adding turbulators (higher *U*) and optimized theta values. Designers must balance these factors against pressure drop and operational costs (Laval, 2004).

3.4.15 Energy balance over the Compressors

Compressor power in a CO_2 transcritical system using positive displacement compressors is calculated by the mass flow rate times the enthalpy rise across the compressor, divided by the isentropic efficiency (assumed to be 70%). Accurate property data can be extracted whether Use CO_2 property tables or software (like REFPROP) to find h_{in} and h_{out} (see Equation 23).

Equation 23

$$W_{compressor} = \dot{m} \times (h_{out} - h_{in}) / \eta_{comp}$$

3.4.16 Nominal mass flow rate of refrigerant R744

Following the approach applied by (M. Karampour 2011), the refrigerant mass flow rate is calculated through an energy balance over the compressor, the hourly rate of flow passing through the compressor stages was considered as the basis for predicting the nominal Design mass flow, for the given peak value (663 kW) assuming the Transcritical operating setup (P_{Design}=120 bar, $T_{GC,out}$= 102.8 °C); it has been proportionally scaled up the required mas flow rate of CO_2 Based on available property date for CO_2 (enthalpy values) and corresponding density within mentioned operational region extracted from Danfoss Coolselector data set library for R744 refrigerant (Danfoss Climate Solutions, 2023). The nominal value for design mass flow is forecasted to be 15160 kg/hr for running the system in its full capacity.

3.4.17 Condenser (Gas cooler) Dimensioning

For systems with known evaporator duty Q_{evap} and compressor power $W_{compressor}$ (see Equation 24).

Equation 24

$$Q_{condenser,} = Q_{evaporator} + W_{in}$$

And the size of the Condenser (Gas cooler) can be calculated subsequently through the empirical formula, similar to the evaporator (Equation 25):

Equation 25

$$Q_{Cond} = U \cdot A \cdot LMTD$$

Impact of Ambient outdoor Temperature on Gas cooler Design outlet Temperature (T-Sgc)

In Gothenburg's climate, $T_{sgc,\ out}$ varies from 5–10°C in winter (near ambient) to 25–35°C in summer (with heatwaves exceeding 40°C). According to the rule of thumbs, at 10° C ambient rise (e.g., 10° C → 20° C), T_{sgc} increases by ~15° C and P_{gc} by ~15 bar. Its equal to boost 1.5 °C in T_{sgc} per 1 degree increase in ambient temperature, This criteria has been taken into consideration to model a seasonal pattern of Design T_{sgc} for monthly simulation as one of the key factors to outline the system dynamic while The lowest practical gas cooler outlet temperature T_{sgc} for transcritical CO_2 chillers in cold climates, according to Danfoss guidelines, is 15 °C during winter design conditions and maximum 35 °C for summer design This value balances efficiency and operational stability while maintaining transcritical operation (Danfoss Climate Solutions, 2023).

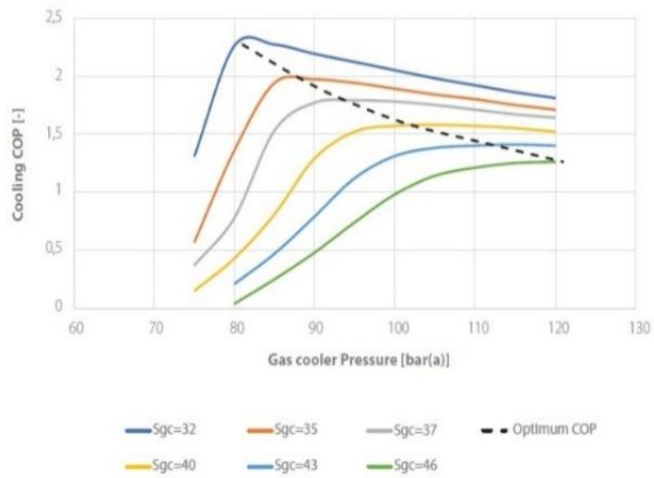


Figure 14-(Nominal COP operational line)

3.4.18 Needed pump work for Direct cooling into the river (see Equation 26).

Equation 26

$$Q_{Cond} = \rho\, \dot{v} . Cp\ . \Delta T\ . SF$$

- ρ = Fluid density (kg/m^3)
- $\dot{v}$ = Volumetric flow rate (m^3/s)
- Cp= Specific heat capacity (kJ/kg°C)
- ΔT = Temperature difference between gas cooler outlet and river (°C)
- SF = Safety factor (typically 1.1–1.3)
- ΔP_{total} in Pascals (Pa)
- η_{pump} = Pump efficiency (typically 0.6–0.75)
- $\Delta P_{total\ differential}$ pressure between suction and discharge

Use Equation 27 for pump power:

Equation 27

$$W_{pump} = \Delta P_{total} \cdot \dot{v}\ /\ \eta_{pump}$$

$$\Delta P_{total} = \Delta P_{pipe} + \Delta P_{equipment} + \Delta P_{fittings} + \Delta P_{elevation}$$

Note: The work needed of the Brine pump in secondary cooling can be calculated based on the same empirical formula.

3.4.19 Effective operational hours prediction

To determine cumulative monthly effective operational hours for energy calculations in open-roof ice rinks, use 24-hour continuous operation for refrigeration systems, as Equation 28:

Equation 28

$$\text{Effective}_{hrs} = [24 \times (1 + 0.03) \times (T_{avg} - 5)]$$

Where:

- T_{avg}: Monthly average temperature (°C)
- +3% adjustment per °C above freezing for coastal humidity effects

For the 1,100 m^2 Gothenburg rink, expect 18-24 effective compressor hours/day, depending on season, with total system energy calculated as Equation 29:

Equation 29

$$Daily\ Energy\ (kWh) = \frac{Q_{total,}(kW) \times \text{Effective}_{hrs}}{COP}$$

This aligns with Scandinavian case studies showing 1,700-2,300 annual operating hours for similar systems (Gummesson, 2014, Karampour, 2011).

3.4.20 (COSP) Systematic approach behind Performance Benchmarking

To calculate the Coefficient of Performance (COSP) for a transcritical CO_2 chiller integrated System cascade with a waste-heat-driven absorption cycle, the system's total cooling output and energy inputs must be evaluated as Equation 30.

Consider the COP of a Single-effect absorption chiller for this Scenario equal to 0.7,

Equation 30

$$COSP_{Casccade} = \frac{Q_{Primary} + Q_{waste\ heat\ utilization}}{W_{in}} = \frac{Q_{CO2} + Q_{\ absorption}}{W_{Compressor}}$$

Only a fraction (η) of the $Q_{gas\ cooler}$ is usable due to temperature or logistical constraints (see Equation 31):

Equation 31

$$Q_{gen} = Q_{recovered} = \eta \cdot Q_{gas\ cooler,CO2}$$

The $Q_{recovered}$ by generator (desorber) from transcritical CO_2 systems, which is known as Q_{gen}, is typically 30–40% of the total duty of the Gas cooler, so $Q_{absorption}$ could be calculated by (Equation 32);

Equation 32

$$Q_{absorption} = COP_{absorption} \cdot Q_{gen}$$

3.4.21 Seasonal Performance Factor (SPF)

SPF stands for Seasonal Performance Factor. It is a key indicator used to measure the annual energy efficiency of heat pumps, chillers, and similar energy systems. Unlike COP (Coefficient of Performance), which measures efficiency at a single operating point, SPF reflects the system's actual performance over an entire season-taking into account real-world conditions, temperature fluctuations, and the energy use of all ancillary components (like pumps and fans) (Energiamegújítás, 2025).

The standard formula for SPF is Equation 33:

Equation 33

$$SPF = \frac{\sum(Q_{Cooling} + Q_{recovered\ heat}) \quad (kWh)}{\sum E_{input} \quad (kWh)}$$

SPF < 2.5: Low efficiency	**2.5 ≤ SPF < 3.5:** Medium efficiency	**SPF ≥ 3.5**: High efficiency (modern, well-designed systems)

3.4.22 SEER value in Energy Modeling

SEER stands for Seasonal Energy Efficiency Ratio, it is a key metric used in energy modeling to evaluate the energy efficiency of air conditioners and heat pumps over an entire cooling season and refers only to cooling performance, not including heat recovery. SEER is calculated as the ratio of the total cooling output (measured in British Thermal Units, BTUs) produced during a typical cooling season to the total electrical energy input (measured in watt-hours) consumed during the same period (carrier, 2025) and the Lowest possible electricity consumption identified by SEER >10 (see Equation 34).

Equation 34

$$SEER = \frac{Total\ Cooling\ output\ (BTUs)}{Total\ electrical\ energy\ Input\ (Wh)}$$

Where 1 kWh = 3,412.142 BTUs

3.4.23 KB-Temperature criteria of Secondary subsystem

KB-Temp typically refers to the refrigeration unit operating temperature, which is a critical parameter in Scandinavian HVAC engineering. This term specifically denotes the secondary coolant temperature maintained by the refrigeration system to preserve ice quality (hardness ≈ 45–55 Shore D) and COP (2.8–3.2 for modern systems) while optimizing energy efficiency. A 1°C reduction in KB-Temp increases energy consumption by ~7% in CO_2 systems (Shahzad, 2006).

For the specified 1,100 m^2 Gothenburg rink, to achieve an ice surface temperature of **-3°C** using aqua ammonia as the secondary coolant needs to be maintained KB-Temp at -8°C evaporator/-7°C return temperatures, as the optimal kB design ±1°C during summer peak loads (June–August) and ±0.5°C in winter. However, for precise Load-Dependent Adjustments based on monthly value, it can be estimated based on (see Equation 35):

Equation 35

$$T_{KB} = T_{set,ice} - (5 + 0.2 \times G_{solar,monthly})$$

- A 5°C ΔT between the secondary coolant and ice surface is ideal
- Aqua ammonia at -8°C absorbs heat from the ice (-3°C) with minimal thermal resistance.
- Larger ΔT (e.g., 7°C) increases pumping power but reduces heat exchanger size
- A 1°C reduction in KB-Temp increases energy consumption by ~7% in CO_2 systems (Shahzad, 2006).

T_{KB} = *Kältemaschine Betriebs-Temperatur*

4 Energy Performance Analysis and Results

4.1 Preliminary Performance Modeling

Performance modeling started by cohas been the calculation of heat absorption by the Ice pad through the empirical theory and adjustment the load based on nominal COP recorded in ideal lab simulation in accordance to theory (3.3.8), then the potential distribution losses of indirect cooling system has been taken into account and the duty of evaporator obtained in accordance to (3.4.12), system dimensioning and constrain was been scaled up subsequently through energy flows over major component of System such as evaporators, compressors, condensers, and expansion valves to outline the potential peak design for refrigeration system envelope and boundaries.

In order to tackle the limitations of Aspen Plus® against the potential challenges stemming from CO_2's unique supercritical behavior and thermodynamic model constraints, for simplification and to avoid dealing with complex binary interactions and solubility coefficients of CO2 gas properties within the Transcritical setup, an open-source Danfoss Coolselector®2 professional tool was utilized as an reliable alternative to model the Stand-alone Ice rink's CO_2 Transcritical chiller. The data libraries in which this software handled the calculations for R744 (CO_2) and other refrigerants are the ASEREP database (v3.5.0), which is standard for thermodynamic properties and used across Danfoss products.

4.1.1 Performance Simulation

Given by data obtained through the empirical dimensioning of the refrigeration system from previous steps, and knowledge of the Design cooling capacity of the system and vital operational parameters like mass flow rate of refrigerant, a standalone baseline Refrigeration system has been modeled thermodynamically through dedicated Modeling of a standalone refrigeration System via Coolselector 2® Built-in App Chiller:

The chiller model has been created and developed under various System setups to evaluate the best potential alternatives and overall performance.

- **Seasonal Variations**: Assessed how the system's performance is affected during Gothenburg's distinct seasonal conditions (winter vs Summer).
- **Operational Strategies**: Exploring different configurations for managing ice rink operations, including dynamic load adjustments.

Coolselector®2 (Danfoss) key features:

Modeling tools for a CO2 transcritical chiller via Danfoss Coolselector 2® and progressed through the following steps:

1. Component selection (expansion valves, heat exchangers) and system optimization
2. Real-time calculations for pressure, temperature, and efficiency
3. Built-in CO_2 refrigerant properties database
4. User-friendly interface for commercial/industrial applications

Developing the baseline model based on the most novel, practical and commercial proven setup, by adding the following features as below (see Figure 15- (Danfoss Climate Solutions, 2023):

1. parallel compression
2. Gas bypass valve and internal HX
3. High/Low-pressure ejector

In order to enhance the Overall COP Value and make it closer to the real-world practice (see Figure 15).

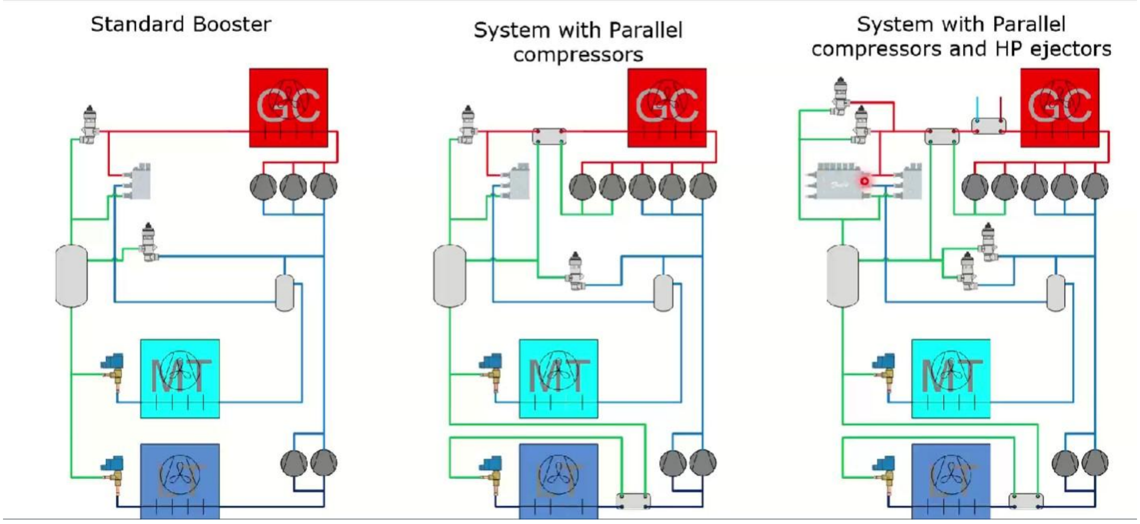


Figure 15- (Danfoss Climate Solutions, 2023)

- Define the month of July as a hotspot and adjust it by considering the 20% extra safety margin for the dimensioning of the Maximum design capacity of the system.
- Within this simulation via Coolselector 2 ®, the secondary coolant loop was not extended explicitly, as we already obtained the design duty of evaporators based on the design KB Temperature of secondary system and adjustment of absorption heat from the Ice pad within the secondary loop, considering potential losses (between the heat source (Evaporator) toward the Ice pad and backward.
- Run the Modeling for different seasonal cooling demands proportional to the maximum outlet gas cooler temperature specified in the Coolselector® 2 as the T_{Sgc} parameter.
- Summary of the values after running the simulated system and extracting the report, including all result values for each component, PI preliminary layout as well as log(p)h-diagram motivated by the monthly input parameters obtained within the previous steps (See Appendix 2).

Performance Metrics:

The resulting values outline the system size and evaluate the *COP*, Q_C, Q_H, energy consumption, and heat recovery potential of the reference refrigeration system under seasonal conditions (Figure 16Figure 17), motivated on a monthly basis. Some standards (e.g., ISO 23953 for refrigeration) may require including pump energy in specific contexts, but this is not part of the baseline COP definition of refrigeration system, while it has been taken into account when it comes to COSP value, which refers to the COP of the whole ICE rink System (See Appendix 3).

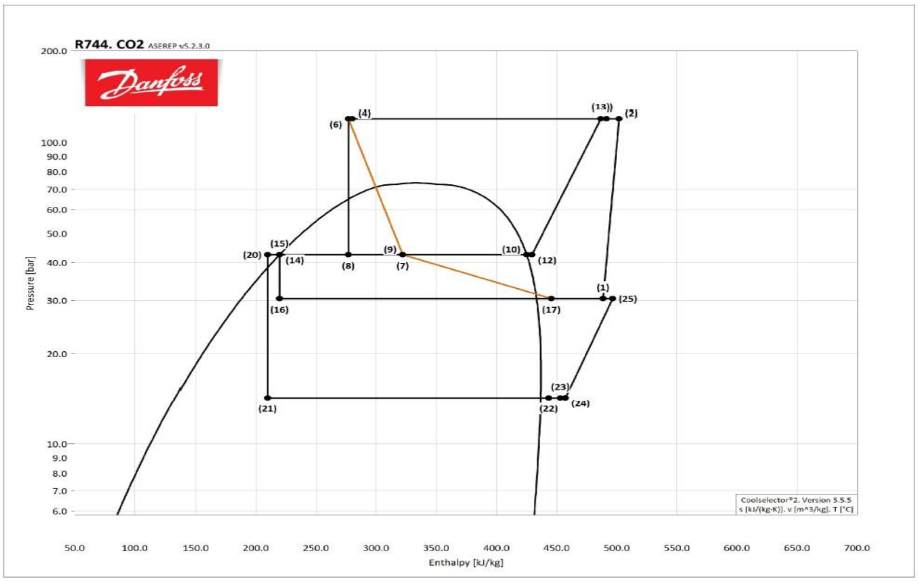


Figure 16- Summer Setup

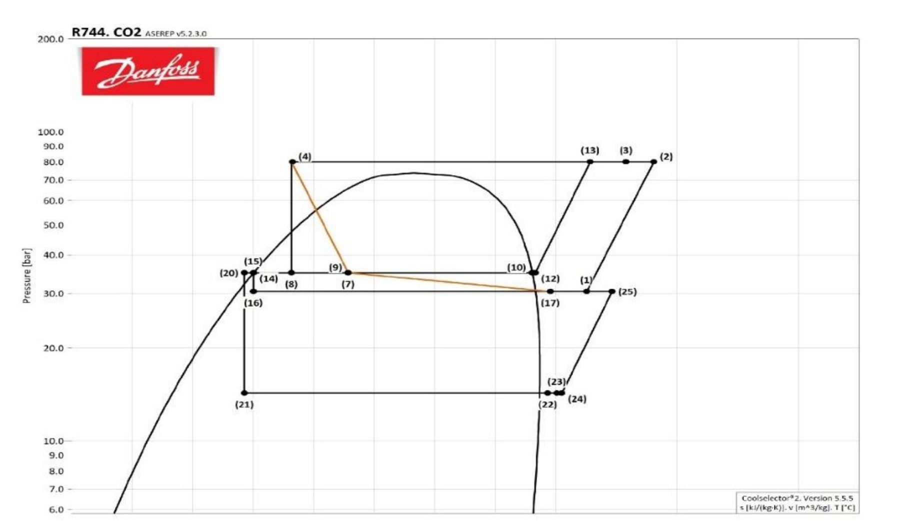


Figure 17 Winter Setup

Model Validation against Empirical Theory

Comparison analysis was performed to validate the energy model's reliability by contrasting operational theoretical values calculated through empirical formula against the results obtained from simulation tools. This validation has been progressed through ASPEN PLUS ®

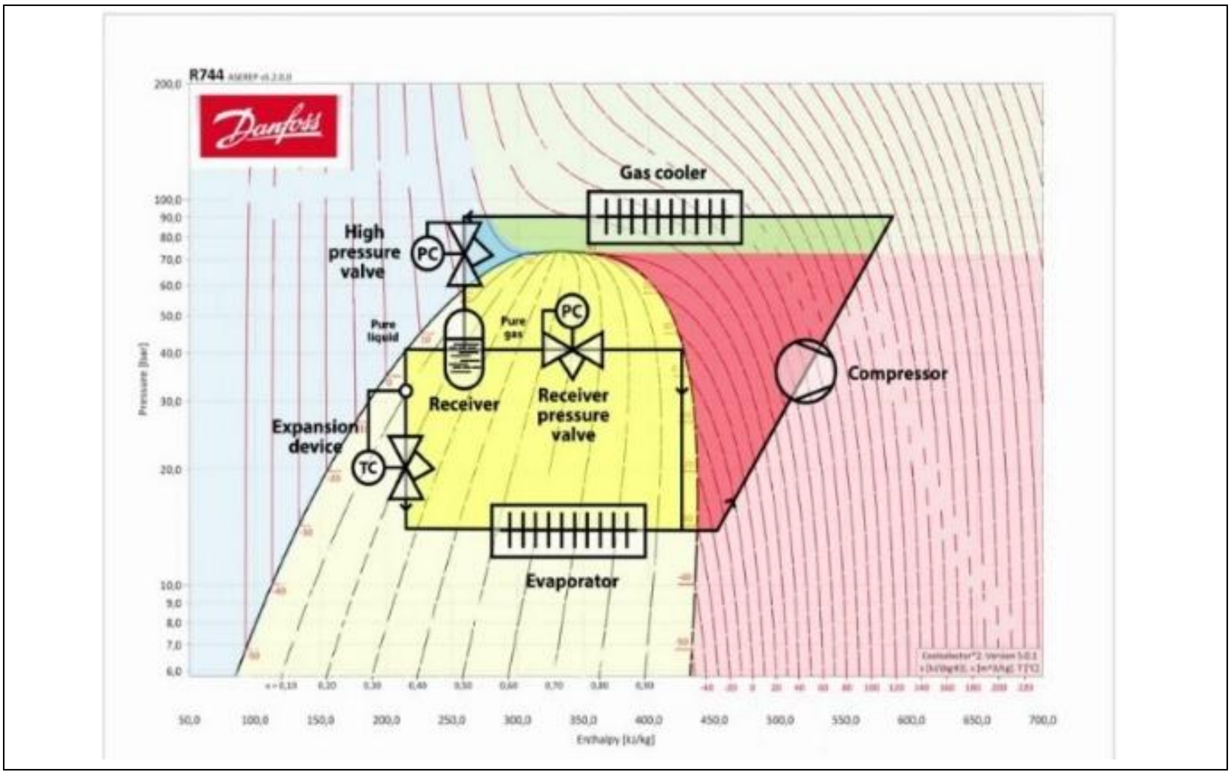


Figure 18-(R744 P-h Diagram)

4.1.2 Data Validation and Integration via ASPEN PLUS®

The energy modeling and process integration phase of this study focuses on evaluating and optimizing alternative setups for the conceptual ice rink refrigeration System at Rosenlundsverket. The compatibility analysis utilizes a relevant simulation tool (ASPEN⁺) to simulate the mass and energy-related performance over the Gas cooler of CO_2 chillers based on resulted monthly value obtained by Coolselector® 2 simulation of the stand-alone CO_2 chiller, to explore how the designed system incorporated against predefined scenario and discover the potential to partially offset the input energy for running the compressors and brine pumps through recovering waste heat of Condenser (Gas cooler) by adding indirect desuperheater without mixture, while ensuring seamless integration of different Scenarios in accordance with (2.8). That means the secondary (Absorption) chiller has not been simulated in detail. The simulation investigates how the desuperheater can utilize the excess heat to run the secondary System and dimension the intermediate loop between the systems in terms of temperature guidelines and mass flow. The aim of modeling in ASPEN⁺ would be explicitly to validation and in-depth analysis the integrated model against the outcomes from Empirical Theory and simulation results through Coolselector tools in which utilize ASEREP data set library which is generally reliable for day-to-day design and selection. However, considering the extraordinary behavior of Pure CO_2 refrigerant within the Transcritical region (Figure 18), professional databases for cutting-edge energy optimization, system upgrades, or research high-accuracy advanced property libraries like REFPROP should be take into consideration specifically when it comes near or over the critical point, ASPEN PLUS have REFPROP internal addons and could be a reliable option to conduct the data Validation.

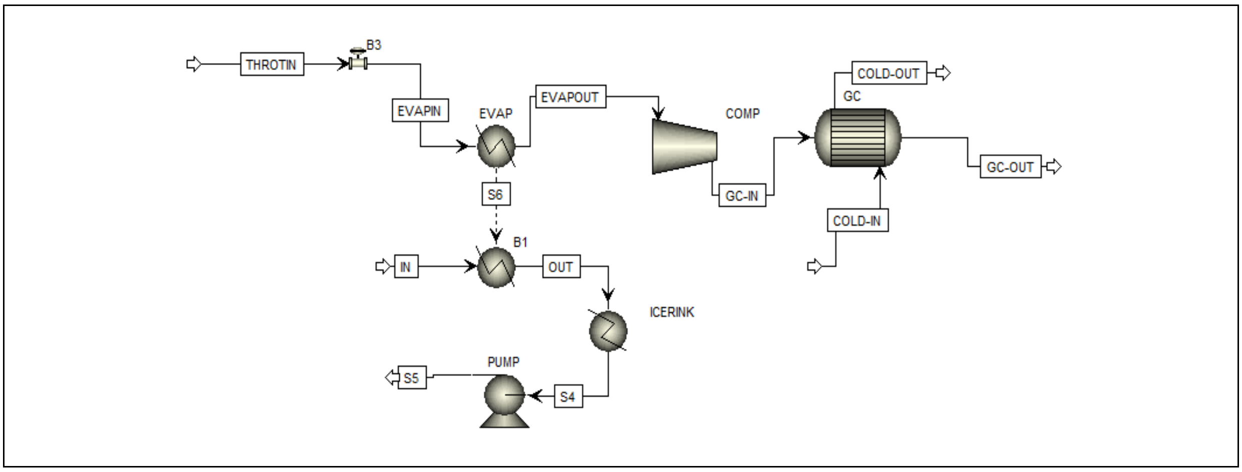


Figure 19-Flowsheet of Reference System

Set up Simulation Environment:

ASPEN PLUS simulations are "structured into four main phases," including simulation setup, where "chemical species and mathematical method for resolution need to be defined (Viola, 2018). This systematic approach (Figure 19) is well-suited for modeling the thermodynamic interactions between CO2 transcritical systems interpolated with different integration Scenarios initiated by establishing a thermodynamic framework by defining fluid Packages as below:

- Select CO_2 (R744) as the refrigerant and REFPROP method as the property package.
- Enable Henry's Law for dissolved gases (if applicable).
- Set Binary Interaction Parameters to NIST REFPROP defaults for accuracy.

REFPROP uses the Span-Wagner equation of state, which is validated for CO_2 properties in transcritical regions (pressures >73.8 bar, temperatures >31.1°C) (Huber et al., 2022).

Then, the modeling progressed following the Scenarios addressed in 3.3.3 (IV)

4.2 1st Scenario: (to serve the Desorber(generator) of district cooling absorption chiller)

The conventional Coefficient of Performance (COP) for single-effect LiBr-H_2O absorption chillers used in district cooling, as it assumed within this Scenario, typically ranges between 0.6 and 0.8 under standard operating conditions (Yuwardi, 2013). This efficiency range applies to systems driven by heat sources at 80–100°C, with water-cooled condensers and absorbers. Integrating waste heat recovery from a transcritical CO_2 refrigeration system to drive the desorber (generator) of a district cooling absorption chiller is technically viable. Research indicates a ~23% enhancement in the coefficient of performance for CO_2 systems when integrated with absorption chillers (Bellos and Tzivanidis, 2019). Success, however, hinges on meticulous thermal and operational alignment.

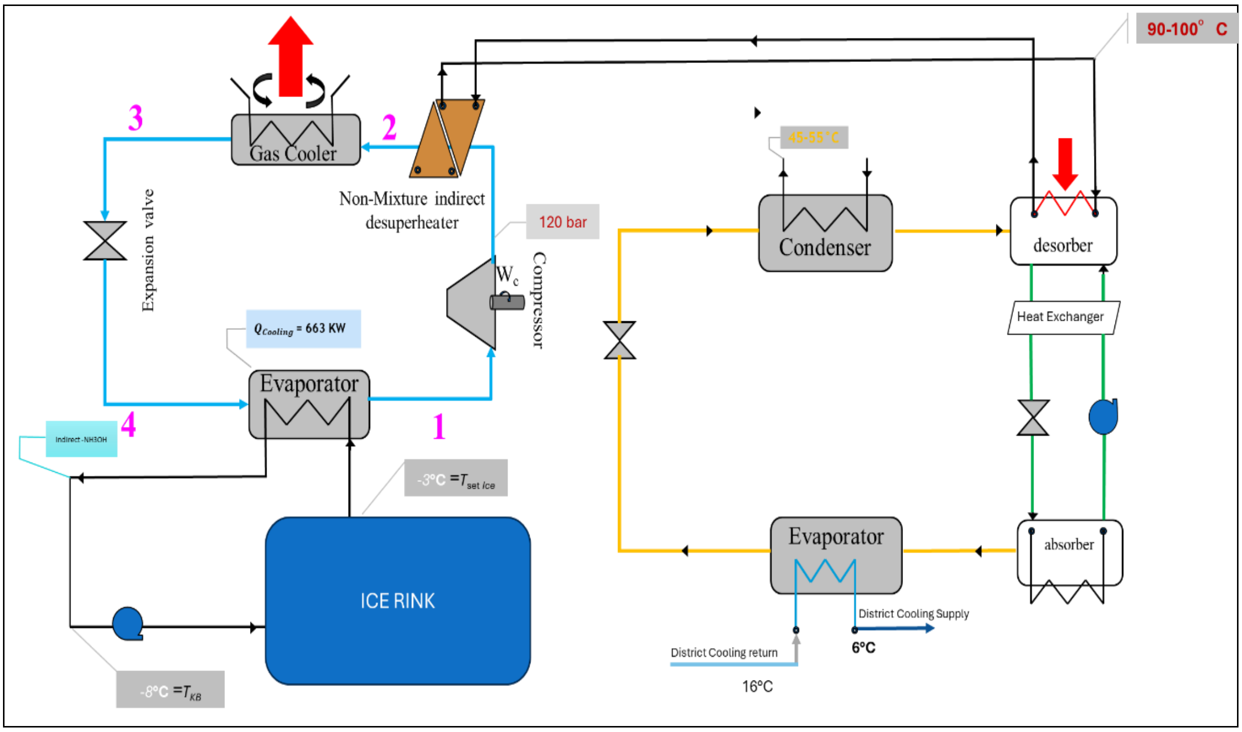


Figure 20- schematic Diagram Scenario I

I. Technical implications:

- **CO_2 System adjustment**: Raising gas cooler pressure to boost heat output (e.g., 120 bar for 90°C) reduces COP by 10–15% but enables waste heat reuse at 100–110°C under optimized pressure (90–120 bar), see Figure 19 , while LiBr-H_2O Chiller's Desorber requires ≥75°C heat input for viable COP (0.6–0.8) can drive a 720–900 kW unit absorption chiller.
- **Buffer Tanks:** Decouple intermittent waste heat supply from steady desorber demand.
- **Heat Exchanger Design**: Use plate-and-shell heat exchangers to handle high CO_2 pressures (≤120 bar) and corrosive absorbent flows.
- **Temperature Glide Management**: Match CO_2's 5–10°C glide with the desorber's heating curve to minimize exergy loss(Cefarin, 2015)
- **Heat Transfer Fluid**: Glycol-water mixtures (e.g., 30% propylene glycol) prevent freezing in LiBr systems.

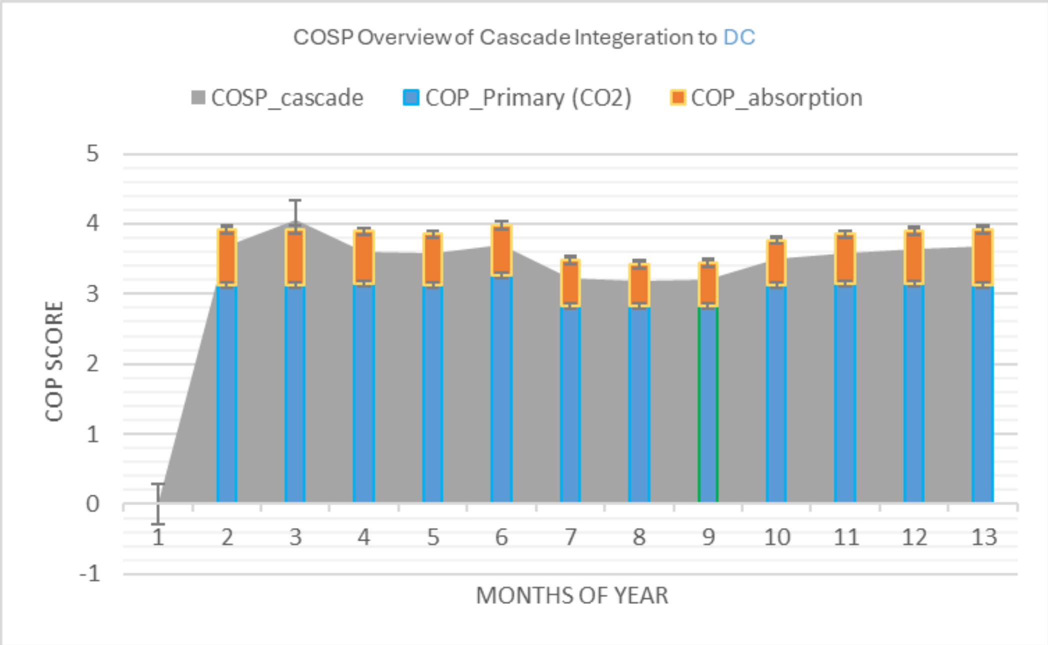


Figure 21_ Cascade system Performance

II. Outcome results

Table 9_ Cascade to DC Scenario outcomes

Parameter	Value	Unit	Alteration
$W_{input, Rink}$	258.07	kw	
$W_{pump, absorbent cycle}$	Neglected	kw	
HX $_{effectiveness}$	0.7	%	
Max Q $_{recovered}$	268.05	kw	
Max Q $_{cooling\_absorption}$	160.83	kw	
Mean $COP_{Primary\ CO2}$	3.07		
Mean COP absorption	0.71		
Mean COSP $_{\_Reference}$	2.72	Kwh/Kwh	See Figure 21
Mean COSP $_{\_Cascade}$	3.55	Kwh/Kwh	
Mean SPF $_{\_Cascade}$	3.45		See Figure 22
Mean SEER $_{\_reference}$	9.48	BTUS/Wh	See Figure 23
Mean SEER $_{\_cascade}$	12.13	BTUS/Wh	

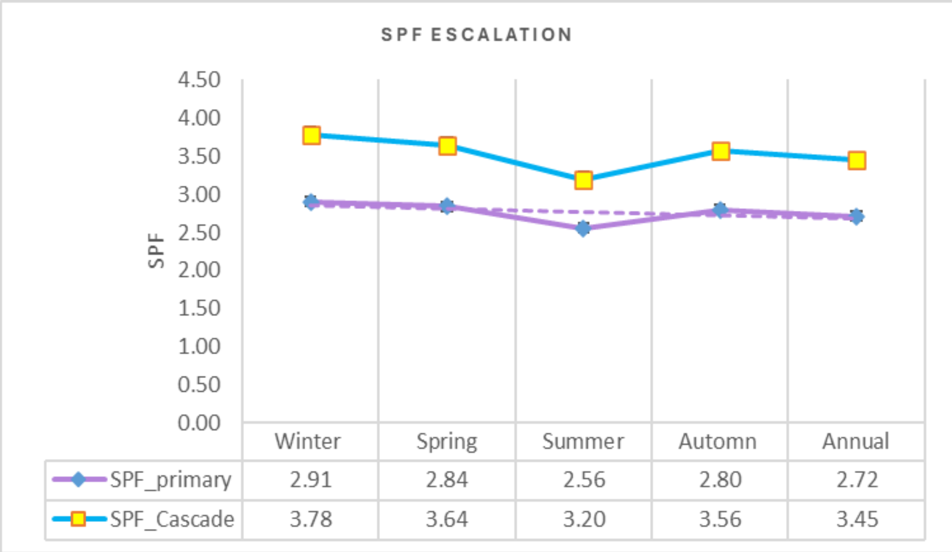


Figure 22 _ SPF Alteration for Cascade to DC

III. Energy Efficiency Trade-offs

Heat Recovery Potential: A 1 MW CO_2 compressor generates ~1.2–1.5 MW recoverable heat at 80°C (Khalid et al., 2019). Systems with absorption chillers achieve Net Efficiency Gain 20–30% overall vs. standalone CO_2 refrigeration.

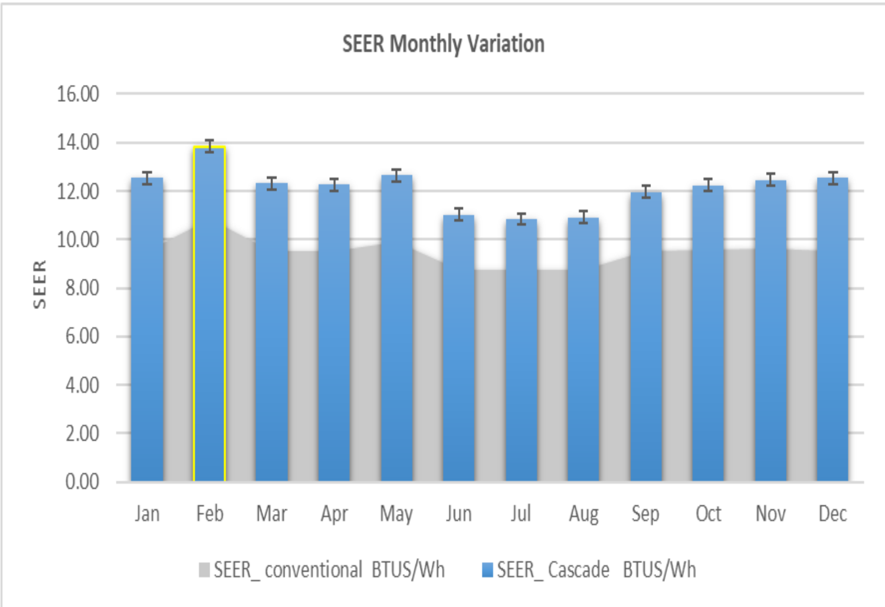


Figure 23_ SEER alteration Scenario 1

4.3 2nd Scenario: to preheat the return line of the DH network before feeding it into the absorption chiller:

Implementing waste heat recovery from a CO_2 transcritical gas cooler to preheat the return line of a district heating (DH) absorption chiller (see Figure 24) is technically feasible and energy-efficient, particularly in systems prioritizing low-carbon heat integration.

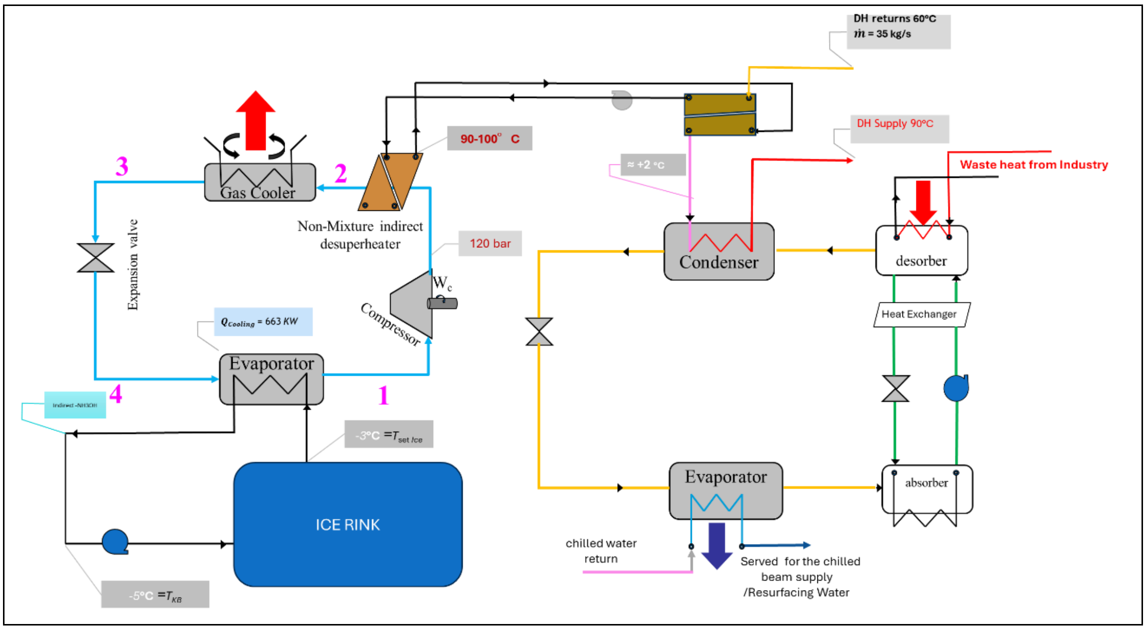


Figure 24_schematic Diagram Scenario II

I. Technical implications:

The model progressed followed the similar criteria as progressed in IV, but the heat source, instead of the desorber, would be an intermediate HX between the gas cooler outlet and DH return feed inlet. Therefore, the technical consideration that was proposed for the previous Scenario remains valid for this integration as well. But the Temperature Glide needs to be customized based on DH return line mass and Temperature, CO_2 Gas Cooler inlet temperature. Operates at 102 -108 °C under optimized discharge pressures (90–120 bar).

DH Return Line Preheating:

a) Typical DH return temperatures: 55 – 60°C (according to Göteborg Energy).
b) Target preheated temperature: 65–70°C to reduce primary heating demand.
c) Base return-line flow is 350 kg/s (GE side) under standard conditions (Nouri, 2020).
d) Conventional operational pressure in 3GDH systems is commonly designed for up to 10 bars (1,000 kPa), but actual operating pressures are typically lower, especially in the return line (Gudmundsson, 2016).

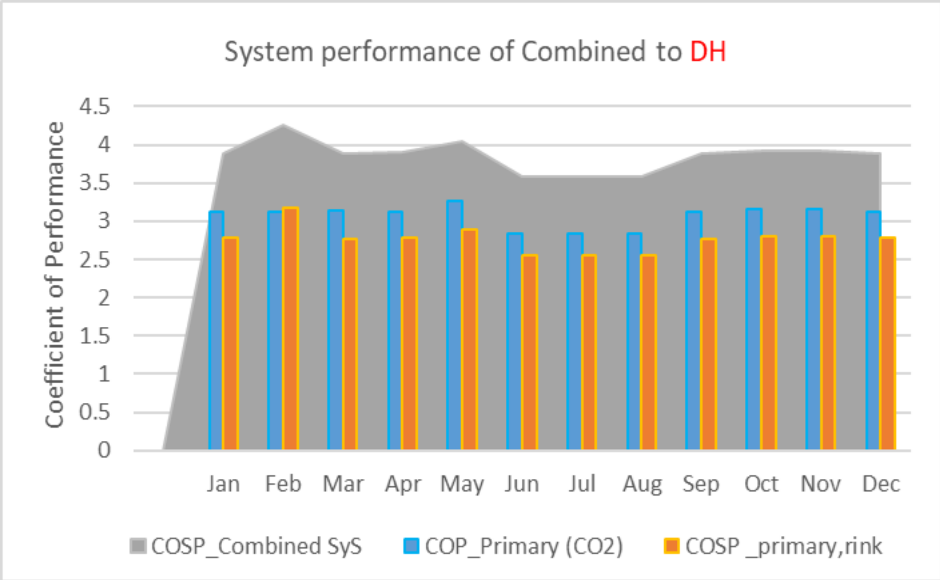


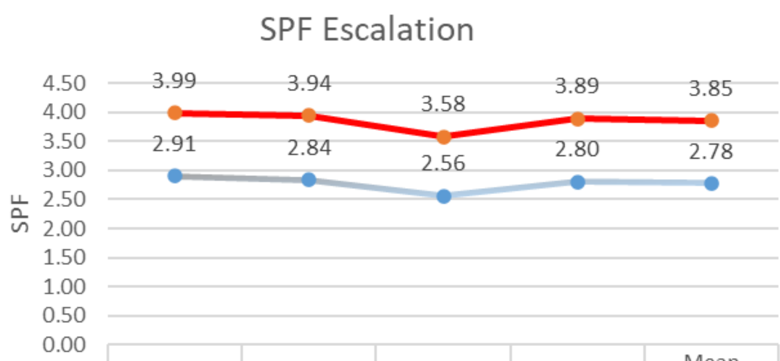
Figure 25_Combined System performance

II. Impact on the absorption of Chiller performance

Given the targeted preheated Temperature of the DH return line, the temperature lift between the feed and supply loop is expected to drop down to 15 degrees, which leads to a severe positive impact on systematic performance and efficiency, which is strictly correlated to overall system characteristics and chillers' arrangement. Without a precise layout, it's hard to justify how efficient the impact on the DH chiller is, for example, the impact on a single-effect absorption chiller would be different in comparison with a multiple-effect type, or it's crucial to understand what the arrangement of the chiller would be look like to customize the system proportional to the corresponds system setup.

The key impact could be summed up as follows:

- Reduced the Generator Heat load needed within the absorption Cycle
- Minimize the needed Heat exchanger areas
- Mitigate the Pumping loads
- Prolong the life span of equipment
- Cut down the system response time against load fluctuations



	Winter	Spring	Summer	Automn	Mean Annual
SPF_primary	2.91	2.84	2.56	2.80	2.78
SPF_Combined	3.99	3.94	3.58	3.89	3.85

Figure 26_ Seasonal Performance Factor Scenario II

III. Outcome results

Table 10_Combined with DH Scenario outcomes

Parameter	Value	Unit	Alteration
$W_{input, Rink\_design}$	259	kw	
$W_{pump, Intermediate\ loop}$	0.93	kw	
Mass flow$_{_Intermediate\ loop}$	6.14	Kg/s	
HX $_{effectiveness}$	0.7	%	
Max $Q_{, GC}$	893.5	Kw	
Max $Q_{recovered}$	268.05	kw	
Max $Q_{cooling\_absorption}$	160.83	kw	
Mean $COP_{Primary\ CO2}$	3.07		
Mean $COSP_{\_reference}$	2.72	Kwh/Kwh	See Figure 25
Mean $COSP_{\_Combined}$	3.87		
Mean $SPF_{\_Combined}$	3.78	Kwh/Kwh	See Figure 26
Mean $SEER_{\_reference}$	9.48	BTUS/Wh	See Figure 27
Mean $SEER_{\_Combined}$	13.14	BTUS/Wh	

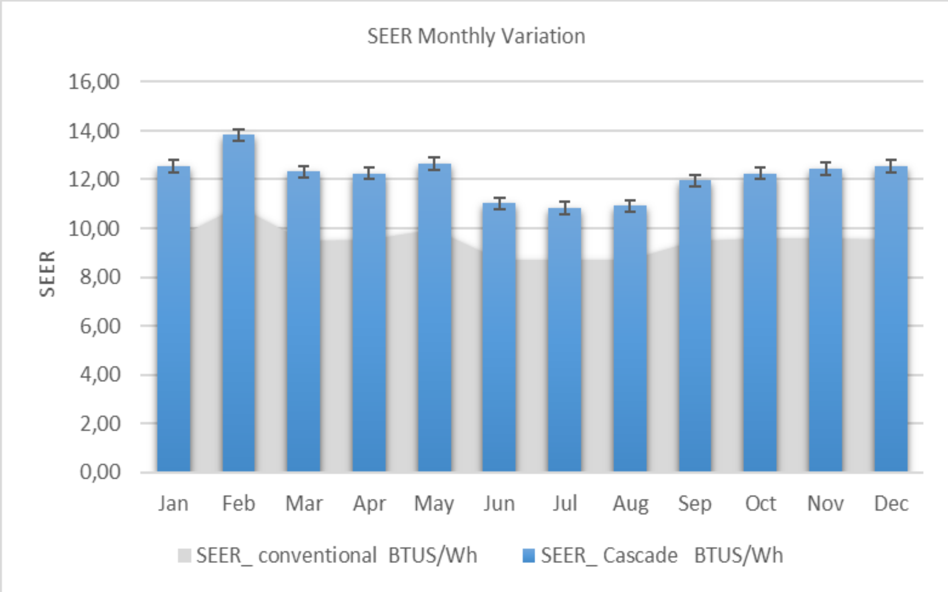


Figure 27_SEER alteration Scenario II

4.4 3rd Scenario: ORC Solution run against the gas cooler outlet

The transcritical CO_2 gas cooler provides an excellent temperature profile for ORC integration and operates with inlet temperatures around 100-110 °C, cooling down to approximately 35°C in 32°C ambient conditions (Ltd, 2025). This temperature range glide is highly advantageous for ORC systems. Moreover, the absence of phase change during heat transfer allows CO_2 to be heated to very high temperatures continuously, creating an ideal heat source for ORC integration, which is compatible with the transcritical CO_2 refrigeration cycle.

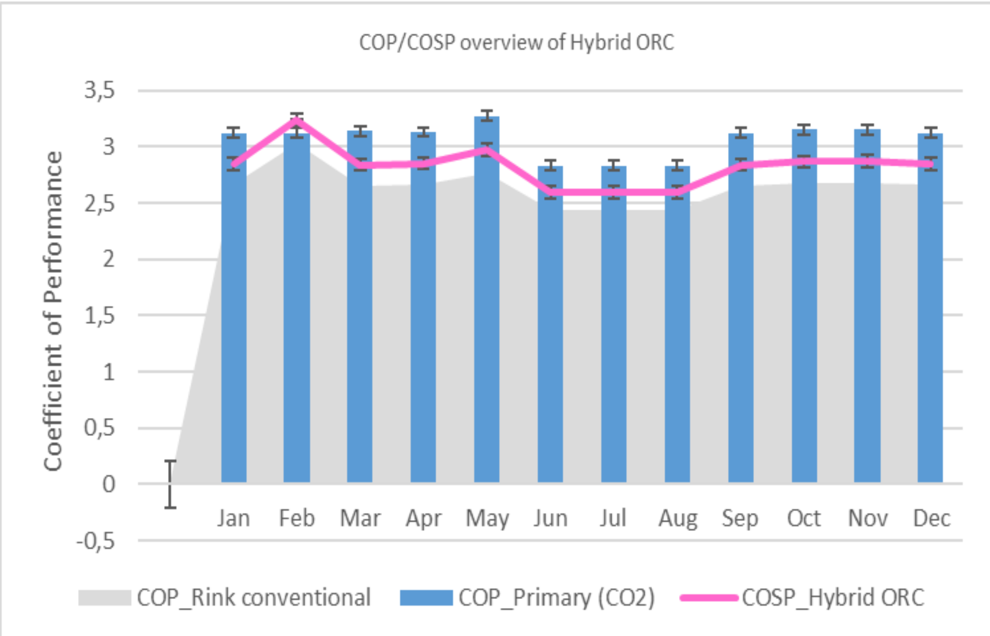


Figure 28_ 3rd Scenario Sys Performance

I. Technical consideration

- The pressure control of the transcritical CO_2 system is critical for maximizing COP and must be maintained when adding an ORC unit. (DanfossClimateSolutionsPublic&IndustryAairs, 2023)
- Heat exchanger designs must minimize exergy destruction in the heat transfer process between the two systems (Bellos and Tzivanidis, 2021).
- The ORC would likely need to be customized for this application to handle the temperature profile and pressure characteristics of the CO_2 system.

II. Outcome results

Table 11_ Hybrid ORC operation outcomes

Parameter	Value	Unit	Alteration
$W_{input,design}$	270	Kw	
Max El $_{Generation}$	16.75	Kw	
Max $Q_{, GC}$	893.5	Kw	
Mean $COP_{Primary\ CO2}$	3.07		
Mean $COSP_{\_reference}$	2.72	Kwh/Kwh	See Figure 28
Mean $COSP_{\_Hybrid,ORC}$	2.83	Kwh/Kwh	
Mean $SPF_{\_Hybrid,\ ORC}$	2.98		See Figure 29
Mean $SEER_{\_reference}$	9.48	BTUS/Wh	See Figure 29
Mean $SEER_{\_Hybrid,ORC}$	10.17	BTUS/Wh	

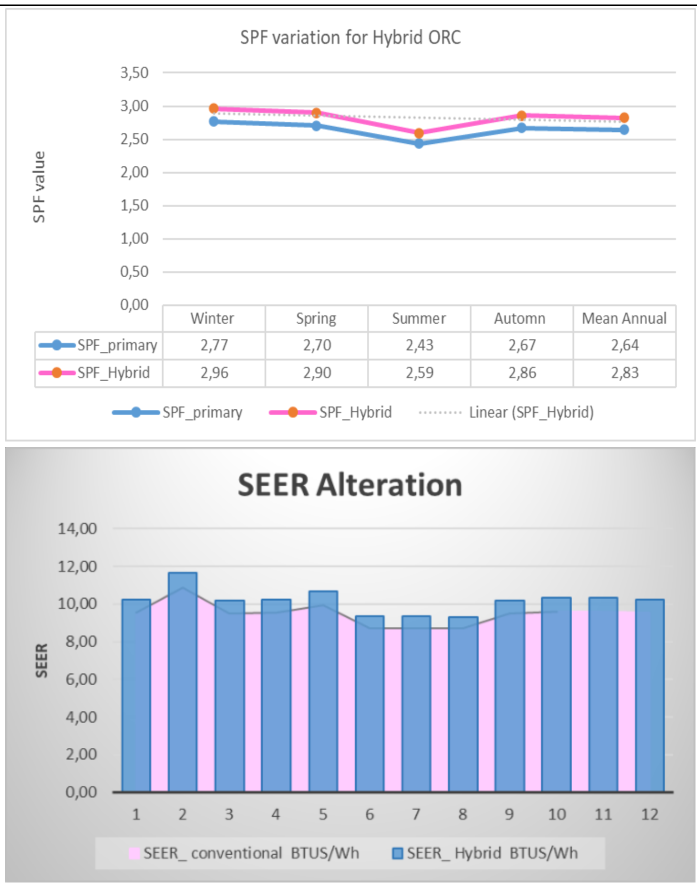


Figure 29 _ SPF & SEER values for Hybrid ORC

4.5 4th Scenario: (Direct Disposal of Waste Heat (Cooling) via Göta River)

Direct cooling disposal of waste heat through the Göta river nearby, proposed as the base option, The pump load (work) for cooling tower water or river direct cooling is not typically included in the Win (work input) for the Coefficient of Performance (COP) benchmarking of a CO2 chiller and considered auxiliary loads that are generally excluded from the chiller's COP calculation (Yamamoto et al., 2020). For evaluating the entire chilled-water system (chiller + pumps + cooling tower), auxiliary energy, a broader metric, COSP, has been taken into account (3.4.20). The needed work to discharge the heat from the system for optimal COP has been evaluated based on monthly values.

I. Heat Dissipation for reference system without WHR (Direct Cooling):

The heat dissipation system of the stand-alone CO_2 trans-critical chiller comprises a pump and a direct cooling arrangement utilizing either chiller water from the cooling tower or pumped directly from Göta Canal's water, with a volumetric flow rate up to 6,000 m^3/h. In this setup, cooling water is circulated through the closed-loop circuit of the system, passing through heat exchangers connected to the gas cooler and condenser components in a series flow configuration. The system serves to dissipate the total heat rejected during the transcritical operation of the chiller. Furthermore, the system was fully instrumented with thermocouples to measure the temperatures within the cold-water circuit as well as the temperature of the incoming lake water. This arrangement, Thanks to the lower temperature range of River water between 8-12, leads to boosting the Heat Rejection Potential and overall COP by 15 % higher than a cooling tower setup for the reference scenario. Flow rate requirement for the CO_2 system could vary between 4200-5500 m^3/h based on the load of the System.

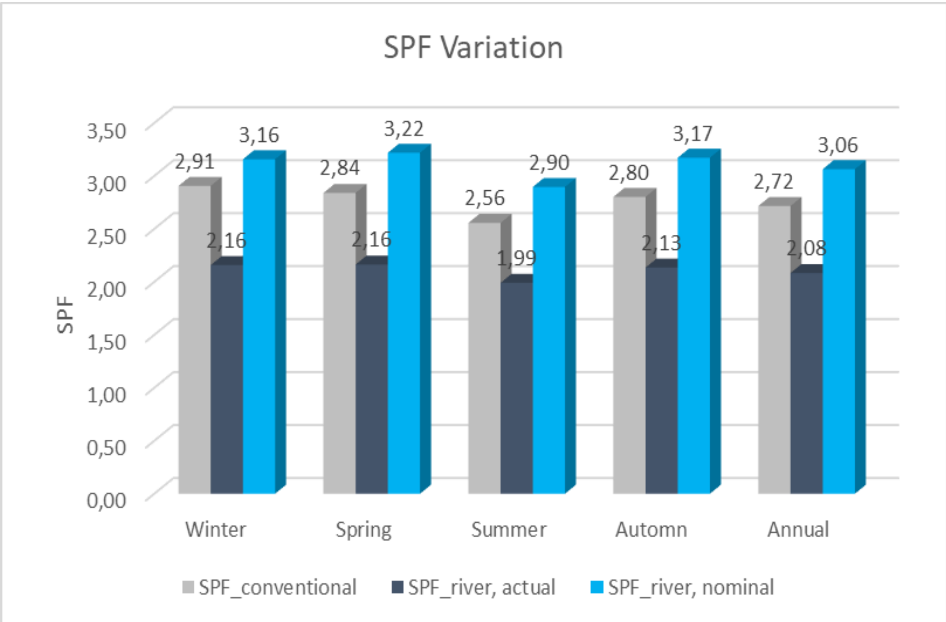


Figure 30 _SPF Variation in River Cooling

Transcritical CO_2 systems align with river cooling, as the deliverable river water to the substation is more than enough to meet the technical criteria for system requirements. The available flow rate has the potential to boost the overall COP value of the chiller up to 15 %

which minimizes the compression work up to 6.7 KW in case of running the system in full capacity.

II. Outcome results

Table 12-Direct River cooling Scenario outcomes

Parameter	Value	Unit	Alteration
$W_{input, design}$	213.18	Kw	
Anticipated $T_{, Discharge}$	W.35 , S.15	°C	
Max $Q_{, GC}$	850.97	Kw	
Mean $COP_{Primary\ CO2}$	3.07		
Mean $COP_{CO2,river\ cooling}$	3.53		
Mean $COSP_{\_reference}$	2.72		See Figure 31
Mean $COSP_{\_River\ cooling}$	3.11	Kwh/Kwh	
Mean $SPF_{\_River\ Cooling}$	3.06		See Figure 30
Mean $SEER_{\_reference}$	9.48	BTUS/Wh	See Figure 32
Mean $SEER_{\_River\ Cooling}$	10.61	BTUS/Wh	

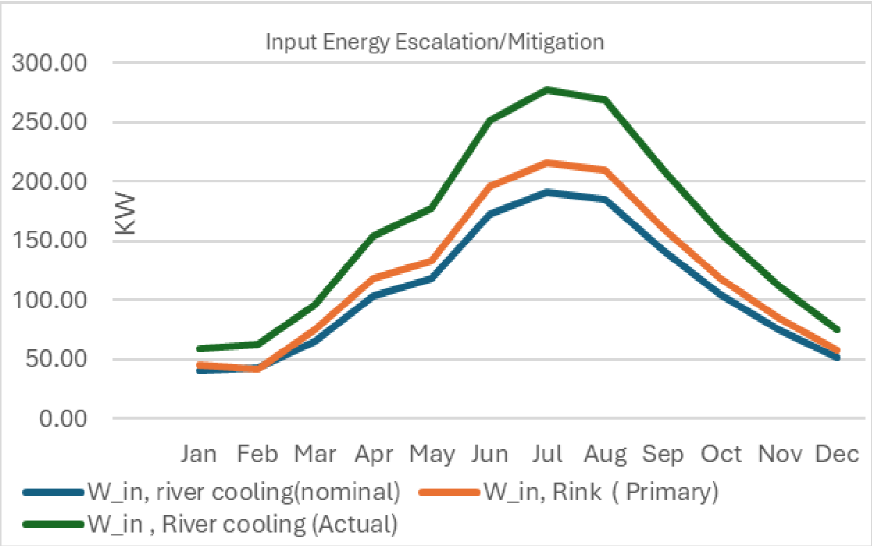


Figure 31-River cooling system Performance

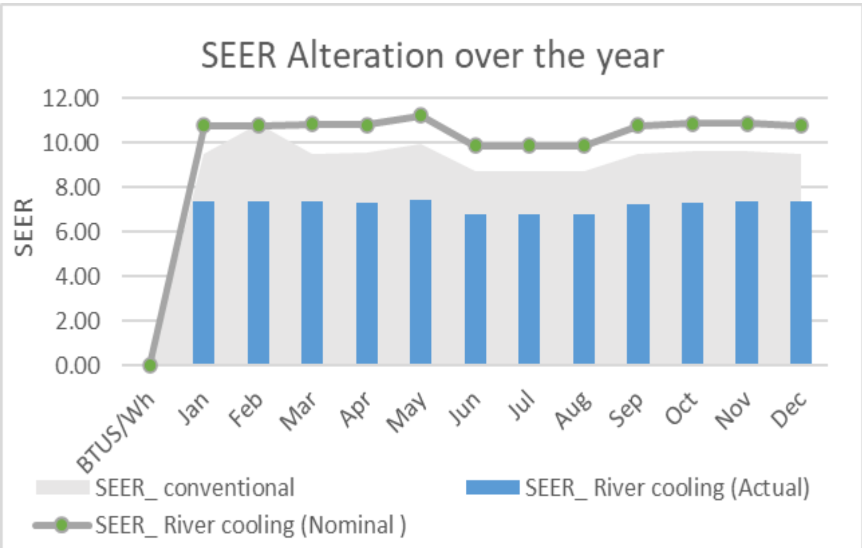


Figure 32_SEER Fluctuation for River cooling

4.6 5th & 6th Scenario, so-called COMBO and COMBO+DHW Scenario respectively

The Combo arrangement followed the similar criteria as addressed in (Figure 33)IV, cascading an ammonia-water absorption chiller (base load) with a CO_2 compression chiller (Peaker) for an indirect ICE rink system, typically integrates the absorption chiller as the main cooling supplier and the compression chiller as fast-response backup, both delivering cooling to a brine loop distributed under the ice surface.

I. *Technical Implications*

- The Peaker (CO_2) chiller provides quick temperature corrections, while the base (ammonia absorption) handles steady loads with thermal inertia. The cascade arrangement allows seamless switching or engagement between chillers.
- Hydronic Connections: The CO_2 Chiller is connected in parallel (cascaded) to the brine circuit via shared or dedicated heat exchangers and starts on demand to supplement the cooling effect. Both the absorption and compressor chillers should connect in parallel to a common chilled water supply and return loop, with isolation valves for maintenance and individual flow control into the receiver header.
- Systems may include buffer tanks to smooth transitions, and control logic (PLC or BMS) to optimize energy use as well as act as a thermal reservoir, absorbing short-term load fluctuations, preventing rapid on/off cycling of the chillers, and helping maintain a stable supply temperature to the ice rink loop.
- Sensors and controls monitor brine temperature and cooling demand, activating the CO_2 chiller during peaks.
- The Temperature range in the discharge of the combo arrangement is less than solely a Transcritical CO_2 chiller, around 45-55, which makes it suitable for local water or space heating.

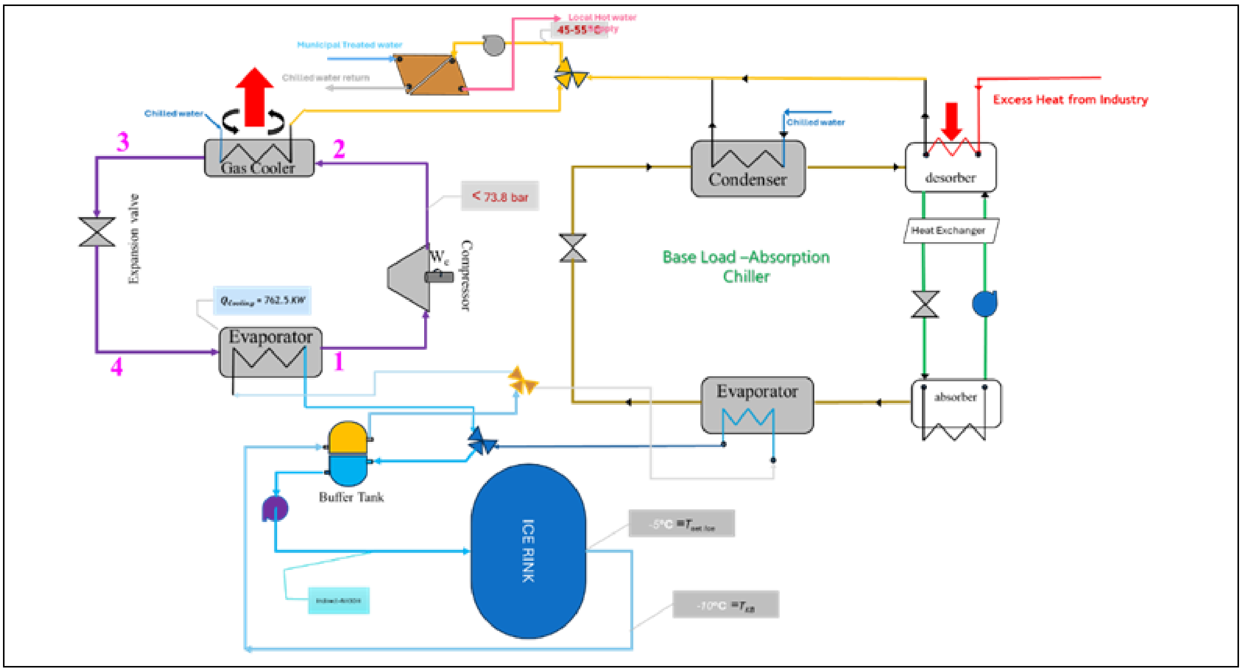


Figure 33_schematic Diagram Scenario 5th & 6th

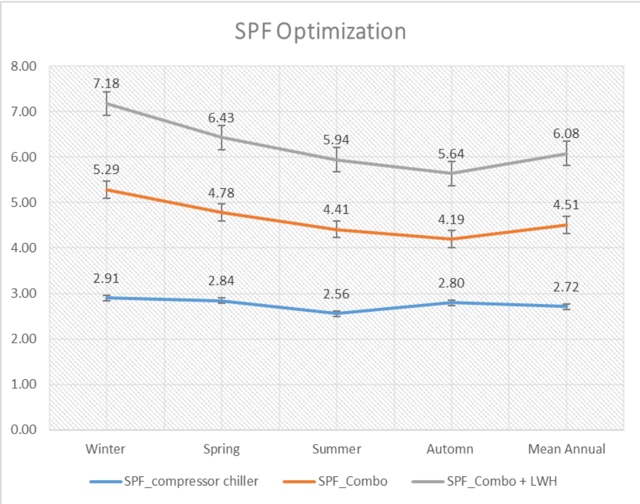


Figure 34_SFP Alteration for Combo Scenarios

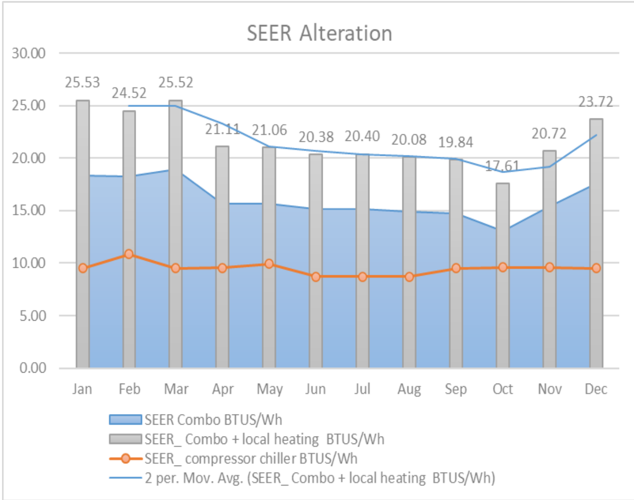


Figure 35_SEER Alteration for Combo Scenarios

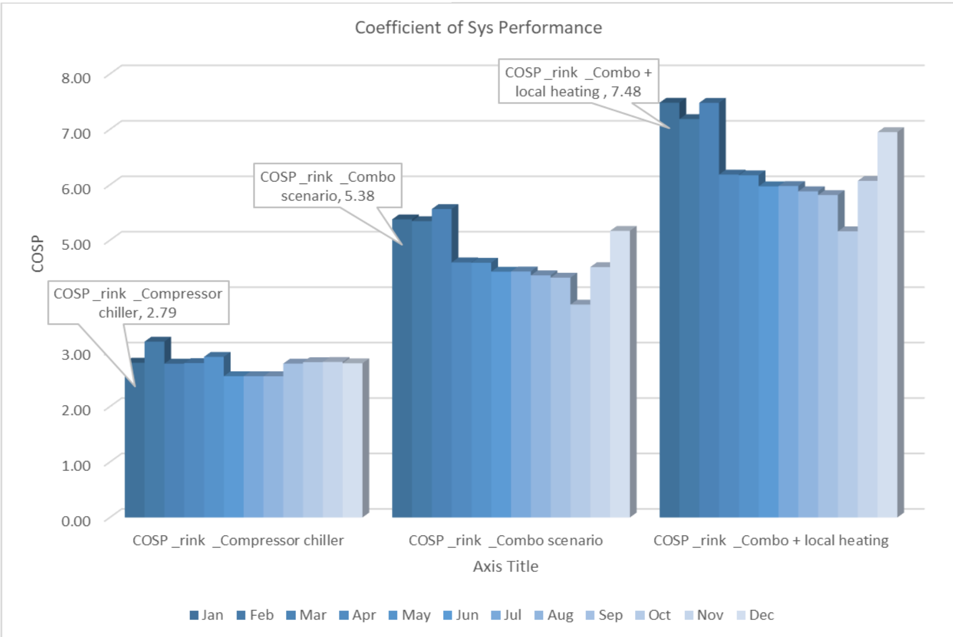


Figure 36_ Electrical COSP escalation

Table 13-COMBO Scenario's Results

Parameter	Value	Unit	Alteration
$W_{input, design}$	158.52	Kw	
$W_{pump\ DWH}$	20.44	Kw	
$Q_{\_evap\_CO2}$	228.74	Kw	
$Q_{\_eavp\_abs}$	533.73	Kw	
$Q_{\_input\ Generator}$	480.36	Kw	
Max $Q_{, GC}$	1317.40	Kw	
Max $Q_{\_DWH}$	395.22	Kw	
Mean $COP_{Primary\ CO2}$	3.07		
Mean $COP_{abs\ (El)}$	13.94		
Mean $COSP_{\_reference}$	2.77	Kwh/Kwh	
Mean $COSP_{\_COMBO}$	4.71	Kwh/Kwh	See Figure 36
Mean $COSP_{\_COMBO+DWH}$	6.36	Kwh/Kwh	

Mean SPF_reference	2.72		See Figure 34
Mean SPF_COMBO	4.51		
Mean SPF_COMBO+DWH	6.08		
Mean SEER_reference	9.48	BTUS/Wh	See Figure 35
Mean SEER_COMBO	16.09	BTUS/Wh	
Mean SEER_COMBO+DWH	21.71	BTUs/Wh	

4.7 Conceptual Design Refinement and Performance Analysis

The conceptual design has undergone iterative refinement based on performance modeling results and energy benchmarking by conducting a comparison analysis between Scenarios (Figure 41). All the findings will be discussed comprehensively in Chapter 5.

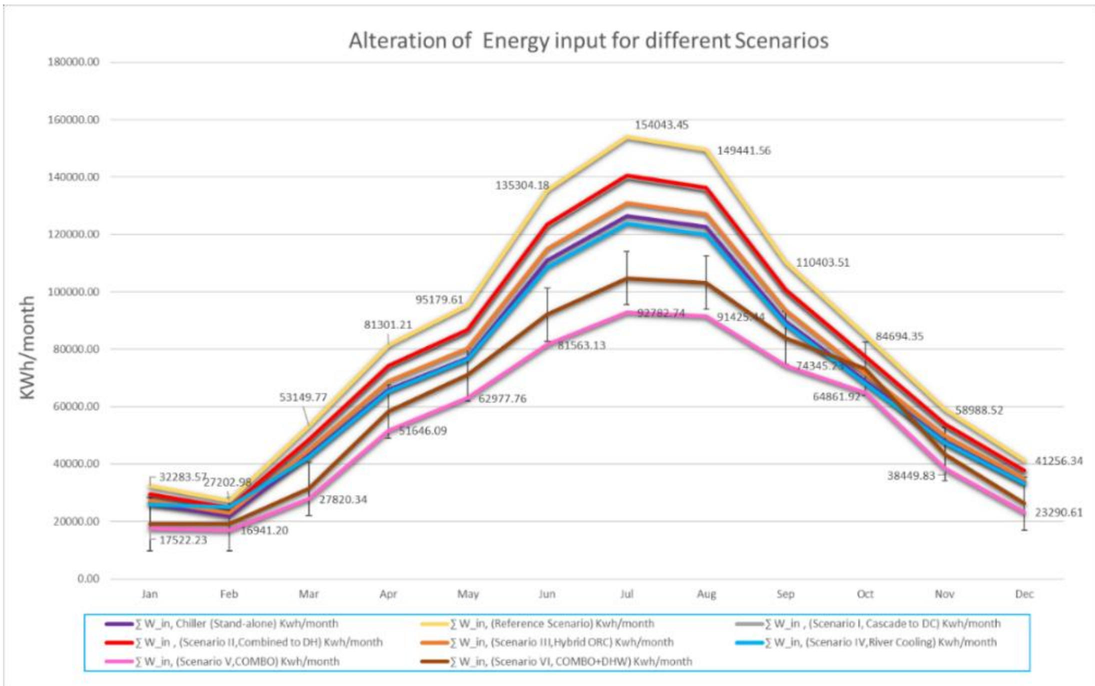


Figure 37__Energy Consumption Projection

The graphs illustrate Annual energy input (Figure 38) and monthly energy consumption projection (Figure 37) across all scenarios, with reference and DH-combined systems showing the highest energy use, especially in summer. Advanced options combo systems consistently require less energy, demonstrating their potential for large-scale energy savings and improved operational efficiency, especially when cooling demands.

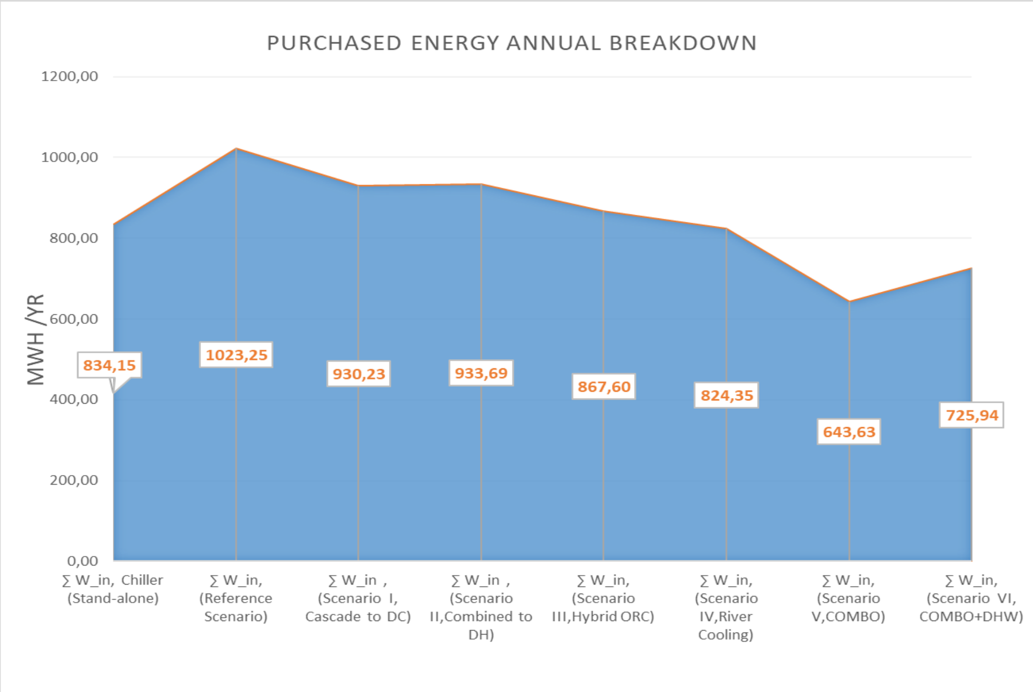


Figure 38_Annual Energy breakdown

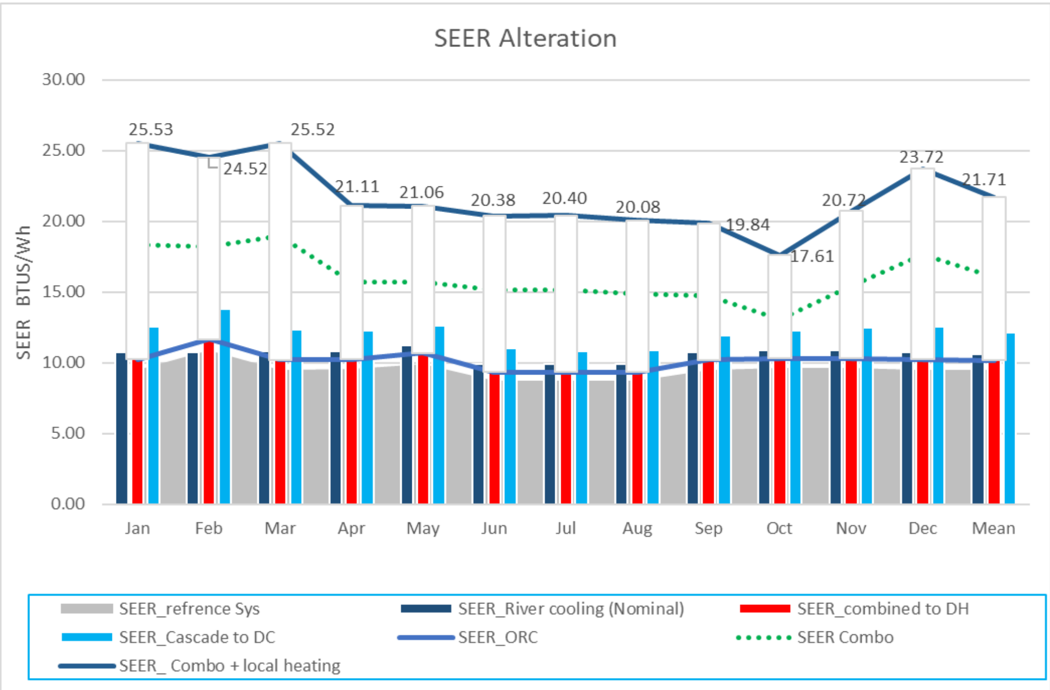


Figure 39_SEER Sensitivity illustration

The SEER chart (Figure 39) shows seasonal efficiency, highlighting that reference system and ORC solutions consistently outperform baseline. The combo solution also delivers strong, stable performance across all months. The data reinforce that system integration boosts efficiency, with notable gains in winter and a higher annual average SEER.

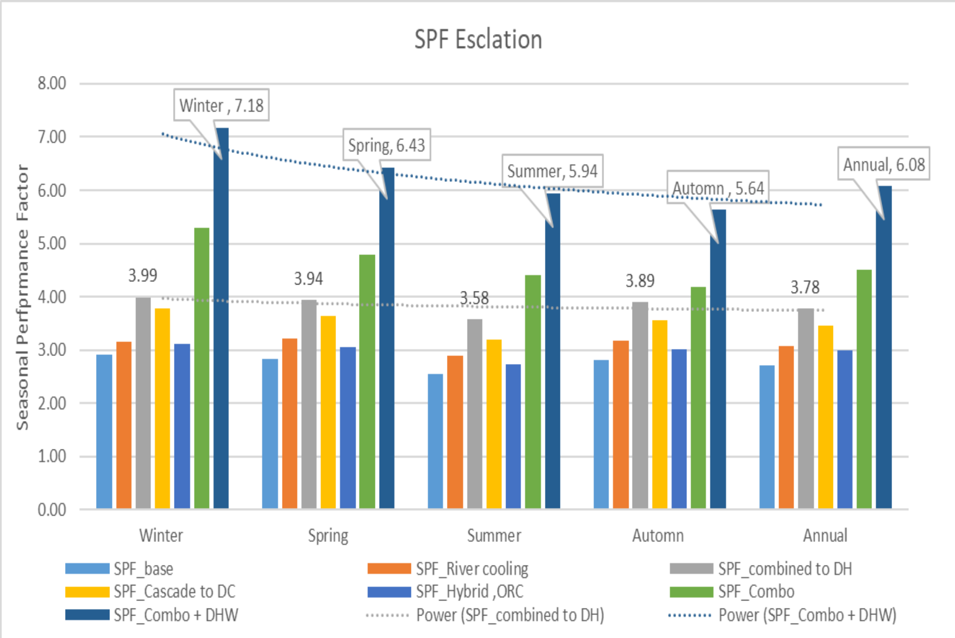


Figure 40_Seasonal Performance Factor benchmarking for Scenarios

Reaching a seasonal performance factor (SPF) of 7.2 or higher in practical, real-world engineering designs is extremely challenging. Normally, an SPF value higher than 5.2 is at the upper edge of practical, real-life achievement, and typically only possible in highly integrated, optimal scenarios with significant heat recovery or where much of the input "energy" is nearly free waste heat, exactly what is applied to this design due to utilizing the waste heat from industry (See Figure 40).

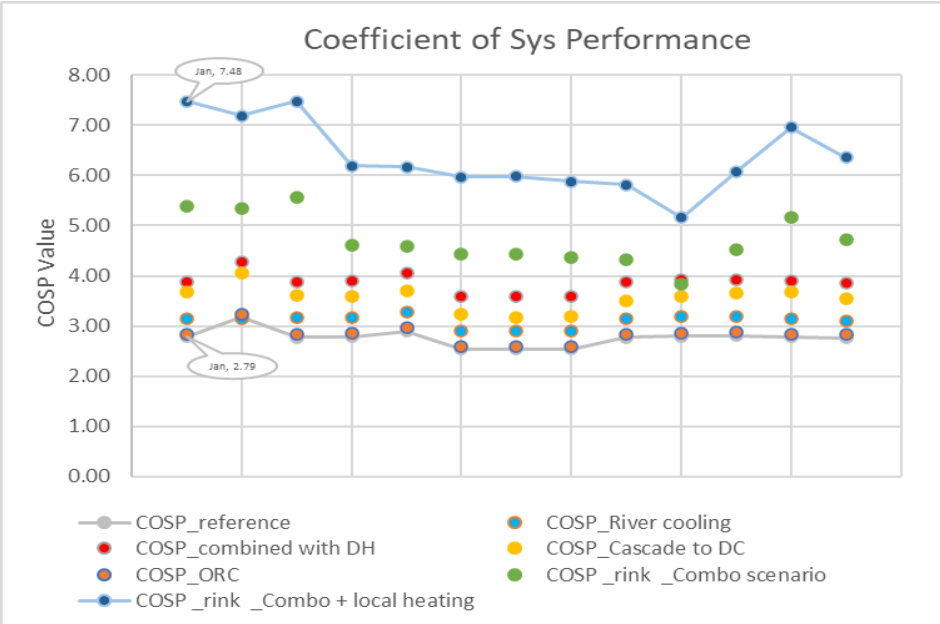


Figure 41_Coefficient of System Performance across the Scenarios

5 Economic Analysis

The economic analysis focuses on assessing the costs, benefits, and financial viability of integrating refrigeration system alternatives for the proposed ice rink at Rosenlundsverket. This evaluation considers the unique context of the Swedish energy market and the specific characteristics of Gothenburg's district energy infrastructure. The analysis prioritizes CO_2-based refrigeration systems as the main system, while integration to absorption chillers and district energy substation is considered as an alternative to investigate their economic feasibility through conducting LCC. It was assumed that the absorption chillers were already in place for district energy serving. The same hypothesis has been taken into account for the Ice rink reference refrigeration system, as it should be there to serve the ice rink facility by itself, which means no more capital cost is considered for purchasing the chillers. The initial cost of the CO_2 chiller is considered zero within the reference scenario to keep the main focus on projecting the variable cost of operation (Energy cost) to explore how it is possible to partially offset that. The main reference of energy price for different delivered energy (El, DC, DH) section 0.4, in Gothenburg has been taken as reference base on official seasonal pricing criteria followed by Goteborg energy reveals as well as the spot price of Electricity in the district where Resenlundsverket is located, in terms of share of renewable energy in supplied energy there. Despite forecasting the needed capital cost of main refrigeration skids for ice rink it has not been taken into LCC assessment explicitly, as this belief exists that the main source of payback for the initial capital fund for the Ice rink facility should be forecasted from future venue's activities, hosting events and competitions, sponsorship from local hockey clubs or either membership or ticket sales revenue. Except for the capital cost for PV films over shading and the anticipated initial marginal expense corresponding to integration, like repining or retrofitting, no additional capital cost has been considered within the LCC assessment.

5.1 Capital Cost Estimation

Capital cost forecasting has been conducted with primary focus on initial investment for Purchase, procurement, commissioning the main Chiller skid and subsidiary infrastructure for the Ice rink facility.

Main Refrigeration System

Transcritical CO_2Chiller: Includes costs for trans-critical CO_2 compressors, high-pressure piping, and heat recovery units, has been anticipated on rough estimation around 5,3 MSEK, exclude labor cost and installation.

ICE Rink Secondary subsystem (Indirect cooling): Based on the requested quote from potential vendors and previous accomplished project (Nilsson, 2019), the Capex needed for the secondary indirect System is forecasted to be roughly around 7 MSEK, anticipated to be higher than ground-level construction with respect to the sub-system size and extent with respect to the Rooftop design.

Invest in energy efficiency Practice:

Integration Costs: Account for expenses related to connecting the refrigeration system to Rosenlundsverket district heating and cooling networks (e.g., heat exchangers, control systems).

Absorption Chillers retrofit: Estimate expenses for equipment utilizing waste heat interconnected to district energy subsystems.

Table 14-CAPEX of Sys Extension

Item	Calculation	Cost (SEK)
Pipe Installation, Procurement & Retrofit	268 kW × 1,500 SEK/kW = 402000 SEK	402,000
Retrofit Complexity Premium (25%)	402,000 × 1.25 = 502,500 SEK	100,500
Commissioning & Ancillary Procurement (15%)	402,000 × 0.15 = 60,300 SEK	60,300
Sum	502,500 + 60,300	562,800

5.1.1 Capital Cost Summary for Different Scenarios

Below is a comparative summary of the estimated capital costs for key refrigeration system alternatives and integration scenarios relevant to a rooftop ice rink project. **Error! Reference source not found.** synthesizes findings from the provided report and supporting literature as well as consultation with potential vendors and supplier , focusing on the refrigeration system and its integration with district energy networks, excluding building construction and structural costs (Girip et al., 2023b).

Table 15-Capital cost breakdown of Refrigeration Sys: Components, Price & Key Drivers

Scenario/System Type	Main Components Included	Capital Cost Range Excl labor & installation cost	Key Cost Drivers & Notes
CO_2 Transcritical Chiller - Stand-alone with direct river cooling	Compressors, high-pressure piping, heat exchangers, control systems, and heat recovery units	SEK 5.5 – 6 M SEK Excluding the cost of the river cooling infrastructure	Choosing the right Vendor
Absorption Chiller	Desorber, Generator, Evaporator, Condenser, pumps, Intermediate HX, economizer	SEK 7.5 - 10 M SEK	Choosing the right Vendor 30-50 % higher than the same size compressor chiller
Indirect cooling Ice rink subsystem	Secondly, the coolant loop, track pipes, pumps, sensors, instrumentation, and interchange HX	SEK 7-8 MSEK	Rooftop design increases the CAPEX for the Rink subsystem

COMBO Scenario	Absorption chiller runs as a base load and an auxiliary compressor chiller as a Peaker to deal with dynamic load response	+ SEK 526 K SEK Excluded the capital cost of the absorption chiller	Higher initial cost due to specialized components; often offset by lower energy/maintenance costs
COMBO + Local heating	An additional heat recovery feature is considered for the DHW supply.	+SEK1.12 MSEK	utilize waste heat for local heating; offset the local heating cost
CO_2 Chiller + District Heating Integration	Above CO_2 system plus integration (heat exchangers, controls)	+SEK 562 K SEK	The cost of connecting to the DH network enables heat sales and greater efficiency.
CO_2 System + District Cooling Integration	Above CO_2 system plus integration (heat exchangers, controls)	+SEK 562 K SEK	Enables efficient heat rejection and absorption chiller synergy.
CO_2 System + ORC (Organic Rankine Cycle) Unit	CO_2 system plus ORC module, heat exchangers, generator	+SEK 0.5 – MSEK	Converts waste heat to electricity; payback rely on incentives.

All costs are indicative for a medium-sized (1,100–1,800 m^2) ice rink in Sweden, based on recent market data, literature, and case studies. Costs exclude VAT, site-specific construction, and building/structural works.

5.1.2 Invest in Green Energy - Photovoltaic Panels

Financial Performance Analysis

The integrated photovoltaic system represents a critical component of the overall project economics, providing substantial long-term value creation through renewable energy generation according to the System Specifications in Figure 42 and Appendix VI :

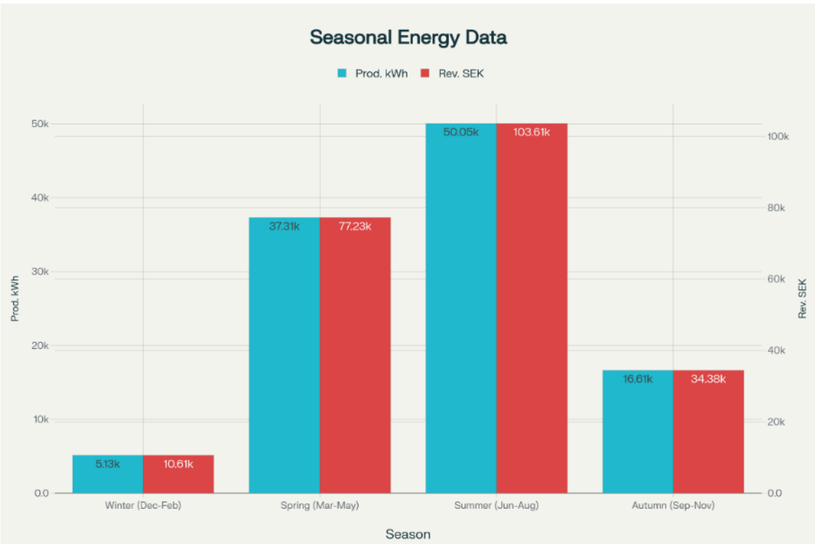


Figure 42_Monthly PV Production and Revenue Analysis

Long-term Financial Projections

Investment Economics (25-year analysis):

- Total capital investment: 2,567,600 SEK
- Annual revenue (2025 prices): 225,833 SEK at 2.07 SEK/kWh
- Annual O&M costs: 12,838 SEK (0.5% of investment)
- Net annual benefit: 212,995 SEK
- Net Present Value: 2,031,656 SEK
- Internal Rate of Return: 6.6%
- Simple payback period: 12.1 years
- Highest Revenue Month: June (36,767 SEK - 16.3% of annual revenue)
- Lowest Revenue Month: December (2,111 SEK - 0.9% of annual revenue)
- Average Monthly Revenue: 18,819 SEK
- Peak/Minimum Production Ratio: 17.4:1

The financial projections incorporate a 4% discount rate, 5% annual electricity price escalation, and 2% O&M cost inflation, reflecting Swedish energy market conditions and renewable energy incentives.

5.2 Variable Cost Projections

Variable costs were projected dominantly based on the cost of consumed energy against generated energy according to the modeling outcome for each Scenario:

- **Energy Consumption**: Calculate expected electricity usage for compressor, CO_2 Chiller, and secondary coolant loop for Ice rink indirect cooling, absorbent and brine pumps, either cooling tower or river cooling pump, considering seasonal variations in cooling demand as well as cooling and heating effect generation via integration.

- **Maintenance Costs**: Estimate maintenance expenses specific to refrigeration systems, such as servicing compressors, etc., in accordance with ASHRAE guidelines for estimation annual maintenance costs based on the rule of thumb.it was assumed that the compression chillers are electric centrifugal (single compressor industrial-field erected) type with a maintenance cost of 2.7 SEK per KW per year, and a hot water absorption chillers (single effect) type were assumed for the absorption chiller cooling system with the maintenance cost of 4.6 SEK per KW per year, but it has not been excluded in our LCC assessment (Snow, 1982).

- **Carbon Tax Implications**: due to limited scope of this assessment and needs of precise result from and external LCA analysis to predict the potential emission or mitigation of the system the carbon tax incentives or penalties has not been taken into account explicitly, as the main energy source supply would be electrical, the impact of Sweden's carbon tax on operational costs, considered to be included in the delivered electricity price, it has been regulated in addition to the inflation rate through considering the discount rate of 4% and energy escalation 5% within LCC assessment.
- **Manhours wage** is excluded from this economic assessment.

5.3 Revenue Forecasts

Revenue forecasts will focus on prospective income sources associated with refrigeration system integration alternatives:

5.3.1 Heat Recovery Revenue:

Estimate the value of waste heat sold to Gothenburg's district heating network, taking into account seasonal demand and pricing structures. Calculate the cost savings from greater energy efficiency and decreased carbon tax liabilities for CO_2 systems compared to the reference Scenario.

5.3.2 ORC revenue

The economic justification for implementing an ORC system lies in its capability to convert waste heat into power, indirectly decreasing energy expenses and emissions. The typical payback period ranges from 2 to 5 years, particularly when incentives are considered. This enhances efficiency and sustainability, rendering it a prudent investment for the sector. The economic feasibility of ORC systems is contingent upon variables such as energy prices, the availability of waste heat, and structured system cost. A precise analysis without optimism is required to determine the optimal implementation.

5.3.3 Energy Saving Revenue

By offsetting operational costs primarily electricity through the implementation of energy efficiency measures, the overall economics of the facility can be transformed by minimizing variable costs and generating additional benefits.

5.4 LCC Analysis

The extensive LCC analysis examines six energy scenarios excluding the reference scenario, omitting the baseline reference scenario for the rooftop ice rink in Gothenburg, and includes thorough financial modeling, sensitivity analysis, and risk assessment over 25 25-year lifecycle. The research identifies substantial prospects for economic enhancement, with the leading three scenarios presenting improvements of 40.4, 30.2, and 22.8 million SEK, respectively, compared to the reference case. The analysis delineates a more defined strategic environment, with COMBO and COMBO+DHW emerging as the undisputed economic privilege, while preserving substantial analytical capabilities across financial forecast, sensitivity analysis, and multi-criteria decision frameworks (see Figure 43).

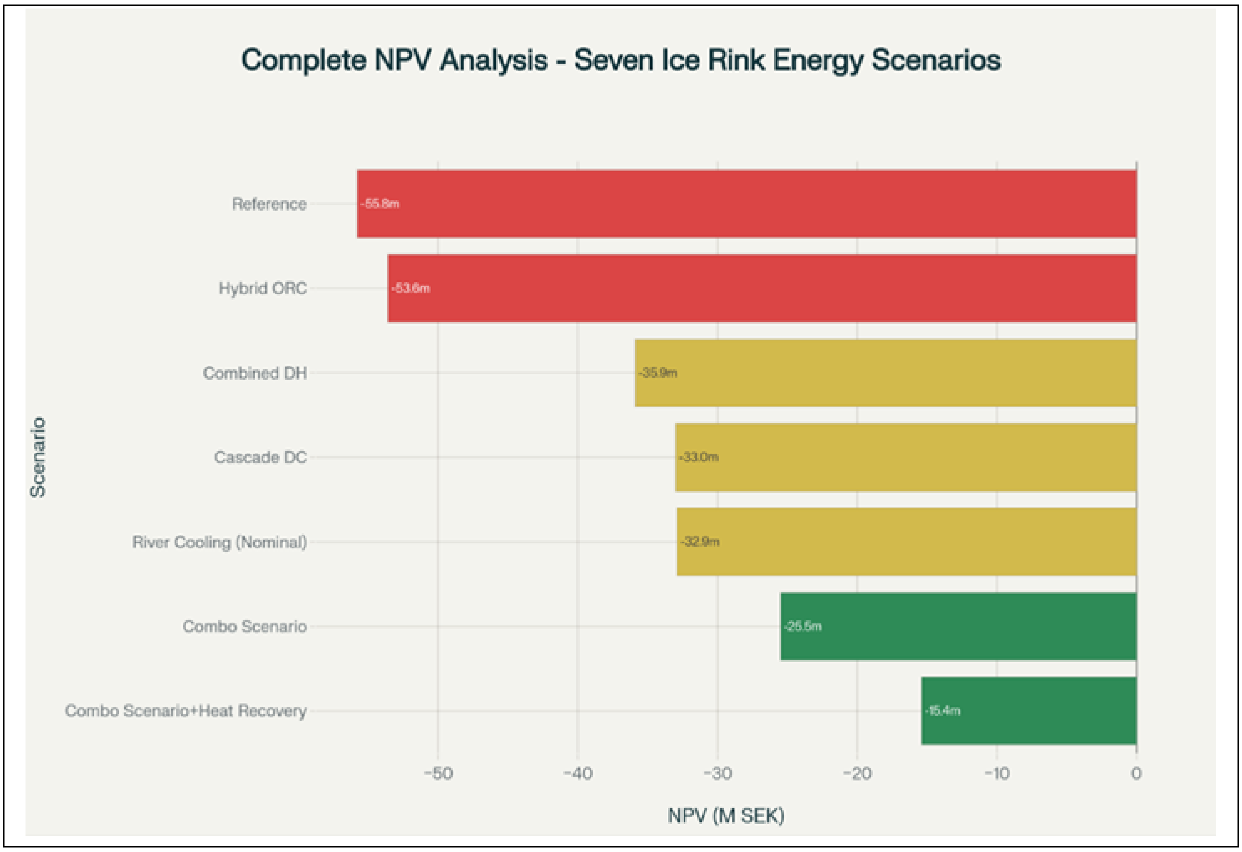


Figure 43_ NPV Analysis

5.4.1 Financial Performance Overview

The Combo Scenario+Heat Recovery results in a 40.4 MSEK enhancement over the prior optimal alternative, signifying a 72% overall cost reduction relative to the reference scenario. The research indicates a substantial potential enhancement between the optimal and suboptimal scenarios, with the leading three scenarios grouped within 3.0 MSEK of one another, suggesting several feasible investment pathways.

Table 16-Scenario classification

Tier	Technology/Scenario	NPV (MSEK)	Description
Tier 1 **Breakthrough Technology**	Combo Scenario + Heat Recovery	-15.4	Revolutionary Leader
	Combo Scenario	-25.5	Advanced Innovation
	River Cooling (Nominal)	-32.9	Proven Champion

Tier 2 **Legacy High Performers**	Cascade DC	-33.0	Balanced Option
	Combined DH	-35.9	Efficiency Leader
Tier 3 **Conventional options**	Hybrid ORC	-53.6	—
	Reference	-55.8	Baseline

Each scenario's overall efficiency was computed as the average of its normalized electrical and thermal efficiency gains, allowing for direct comparison of investment value against holistic site performance across both energy domains. Combining normalized electrical and thermal gains gives a holistic metric aligned with industry standards for energy and economic analysis (see Equation 36).

Equation 36

$$\eta_{El,norm} = \frac{\text{El.efficiency.Gain}}{\text{Max El.efficiency.Gain}} \quad , \quad \eta_{Thermal,norm} = \frac{\text{Thermal.efficiency.Gain}}{\text{Max Thermal efficiency.Gain}}$$

$$\eta_{(Overall\,,norm)} = [\![\eta_{(El,norm)} + \eta_{(Thermal,norm)}]\!] \;/2$$

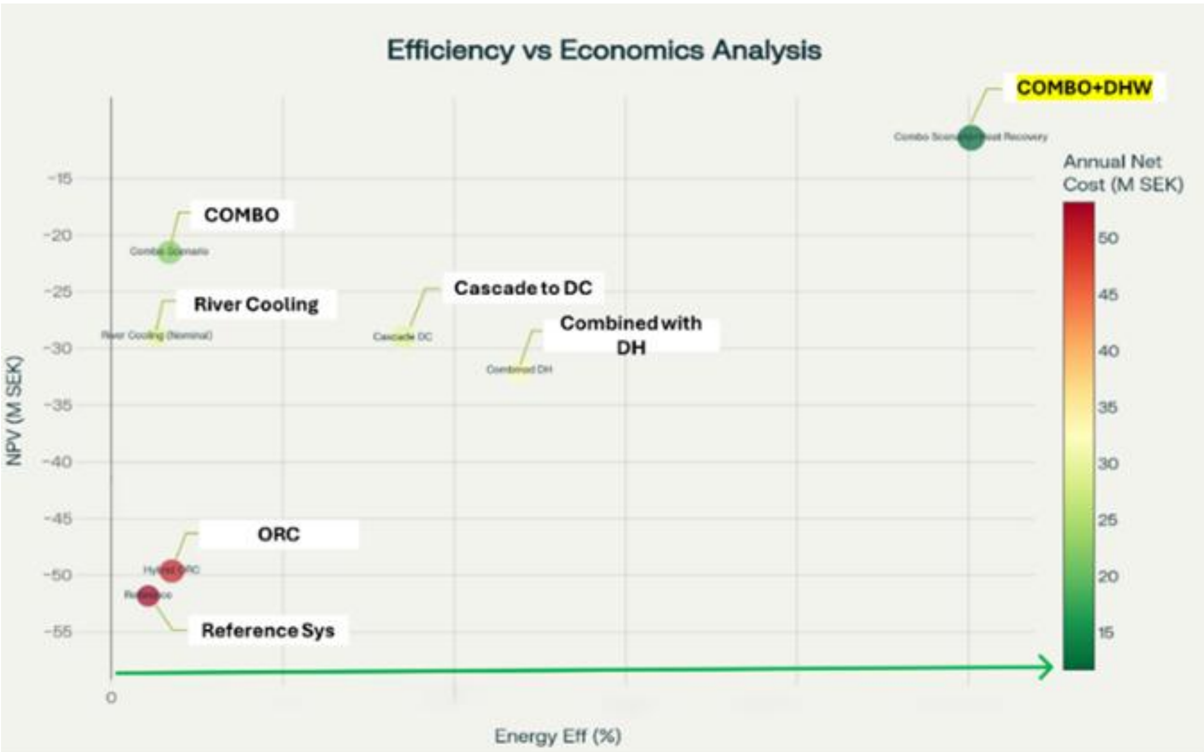


Figure 44_Multi-Dimensional Performance Analysis

Multi-dimensional analysis (Figure 44) showing clear performance clusters with the COMBO+DHW scenario not only achieving the highest efficiency but also offering the best economics

The scatter plot analysis reveals distinct performance clusters:

- ***Efficient***: Combo+ DHW (+ 88 % overall Normalized efficiency) and Combined DH with (+53.4% overall normalized efficiency) demonstrate exceptional energy generation capabilities
- ***Most economical***: COMBO+ DHW and COMBO scenarios respectively achieve the best NPV improvement.
- ***Clear Underperformers***: Reference and Hybrid ORC scenarios show poor economic returns.
- ***Performance Leap***: Orders of magnitude improvement over all conventional approaches

5.4.2 Cost Structure Analysis

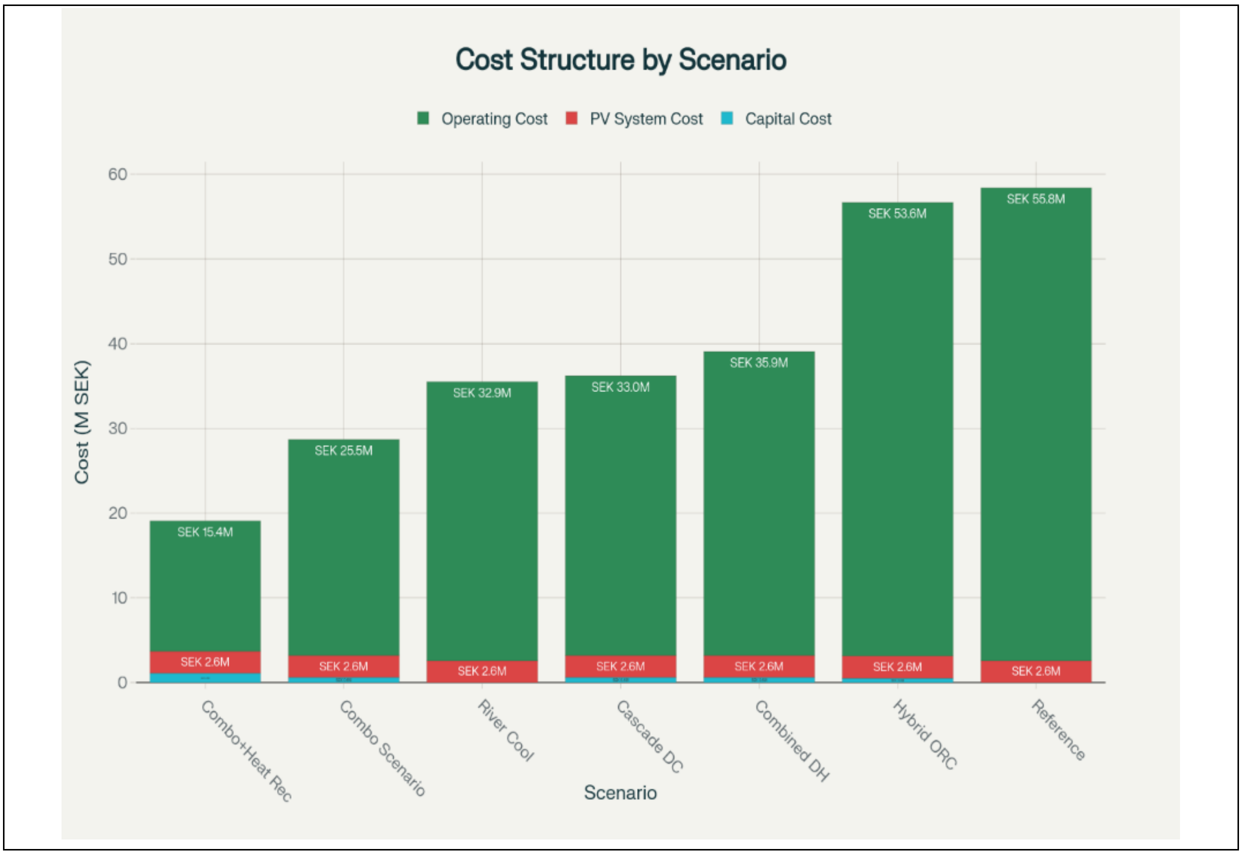


Figure 45_Cost Structure

Lifecycle cost breakdown (Figure 45) confirming operating costs dominate all scenarios, with COMBO scenario having lowest total cost at 18MSEK.

The lifecycle cost breakdown demonstrates that operating expenses dominate total costs across all scenarios:

Table 17-Cost breakdown

Scenario	Capital (MSEK)	PV System (MSEK)	Operating (MSEK)	Total (MSEK)
COMBO+DHW	1.2	2.6	15.4	19.2
COMBO	0.6	2.6	25.5	28.7
River Cooling	0.0	2.6	32.9	35.5
Cascade DC	0.6	2.6	33.0	36.2
Combined DH	0.6	2.6	35.9	39.1
Hybrid ORC	0.5	2.6	53.6	56.7
Reference	0.0	2.6	55.8	58.4

Operating costs represent 93-96% of total lifecycle expenses across viable scenarios, highlighting the critical importance of energy efficiency optimization.

5.4.3 Sensitivity Analysis

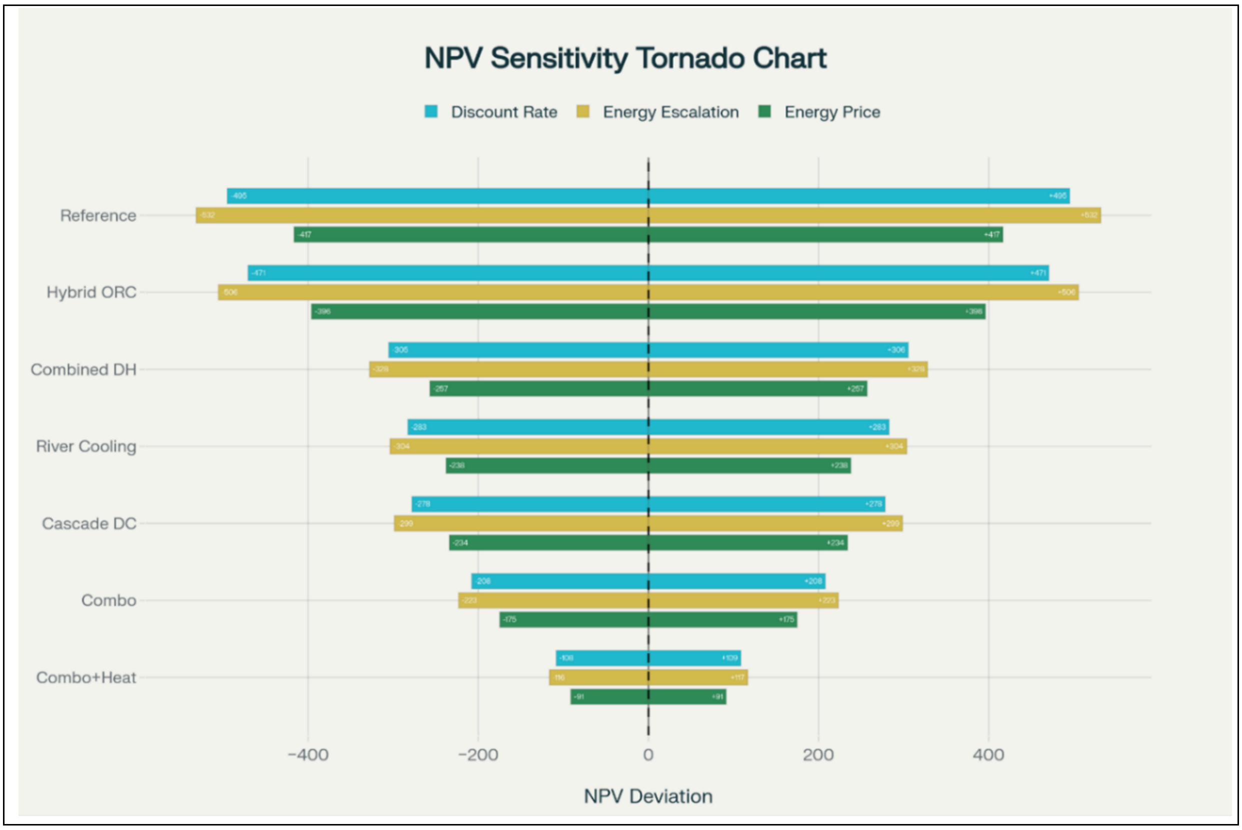


Figure 46_The risk margin depicted by NPV Tornado Chart

5.4.4 Risk-Adjusted Performance Analysis:

Among the Scenarios is one that solely runs on a compressor chiller. The risk adjustment favors Cascade DC due to its low parameter sensitivity, despite having slightly lower raw NPV than River Cooling. However, overall, the tornado chart (Figure 46) shows Reference and Hybrid ORC as most sensitive to parameter changes, while revealing superior risk profiles for the Combo scenarios.

Risk Classification:

- Very Low Risk (±200-250 MSEK): Combo Scenario+DHW (±211 MSEK)
- Low-Medium Risk (±400-600 MSEK): Combo Scenario (±404 MSEK), Cascade DC, Combined DH, River Cooling Nominal
- High Risk (±800-1000 MSEK): Reference, Hybrid ORC

Critical Insight: Combo Scenario+Heat Recovery shows the lowest sensitivity to parameter changes, making it the most robust investment despite being the most advanced technology.

5.4.5 Multi-Criteria Decision Analysis & Strategic Positioning

Figure 47_Decision matrix heat map

The decision-making matrix followed AHP logic so-called 80/20 principle (Figure 47): the two most critical factors (Economics + Efficiency = 60%) receive primary emphasis, while supporting factors (Risk + Speed = 40%). This allocation derives normalized weights that reflect their relative importance(Saaty, 2013). Risk counts as a positive factor here, but as inverse proportion to be adapted to the whole decision-making agenda. The weighted decision matrix (Economic 35%, Efficiency 25%, Risk 20%, Speed 20%) Decision matrix illustrates Combo Scenario+Heat Recovery as the clear winner with revolutionary performance across all criteria. Decision matrix confirming Combined COMBO+ Heat as top overall choice, with a clear top 3 tier emerging from analysis demonstrates a scores 30% higher than the most optimal scenario solely run by the compression chiller, establishing clear technological leadership.

Key Sensitivity Breakthroughs:

1. Energy Price Immunity: Combo+Heat Recovery's net producer status provides a natural hedge
2. Future-Proofed: Lowest sensitivity to energy escalation rates
3. Market Leadership: Best risk-adjusted returns across all scenarios

Strategic Insight:

All the research outcomes, both technically and economically, clearly reflect the COMBO +DHW as the ultimate leading option, with distinguished:

- The Combo Scenario+Heat Recovery scores 30% higher than the most optimal scenario solely run by the compression chiller, establishing clear technological leadership.
- Game-Changing Impact: The Combo Scenario+Heat Recovery delivers a 40.4 MSEK improvement over the previous best option, signifying a 72% overall cost reduction relative to the reference scenario.
- Revolutionary Impact: All Combo scenarios deliver exceptional returns with an extraordinary rate of return on investment, fundamentally changing ice rink investment economics.
- Risk Excellence: Lowest sensitivity despite the most advanced technology
- The Combo Scenario+Heat Recovery stands as proof that ambitious sustainability goals and superior economic performance are not only compatible but mutually reinforcing.
- Providing a pathway for ice rink facilities to achieve carbon-negative operations while generating substantial.
- This ranking provides clear guidance for strategic selection depending on whether a site most prioritizes economic return, overall efficiency, savings, or minimal capital requirement, with Combo Scenario+DHW generally leading on most impact measures.

5.4.6 Investment Comparison Analysis

The capital cost of the main refrigeration system has not been taken into assessment explicitly as the main objective was to offset the operational cost dominantly in this assessment. Beyond the additional capital for standard PV system installation and partial investment for extension, the System is proportional to integration Scenarios. No more CAPEX has been considered. Based on these criteria, the top scenarios deliver exceptional returns on investment, with payback periods under 6 months, excluding the CAPEX for main Chillers.

Table 18-Economical overview

Scenario	NPV Improvement MSEK	Overall Efficiency (%) Normalized	Annual Savings MSEK/year	Additional CapEx MSEK	25-Year ROI
Combo Scenario+DHW	+40.4	88.15	41.5	1.13	3.576 %
Combo Scenario	+30.3	18.55	30.9	0.56	5.411 %
River Cooling	+22.8	9.7	22.8	0.0	N/A*
Cascade DC	+22.7	37.85	23.3	0.6	4.042 %
Combined DH	+19.8	53.4	20.4	0.6	3.521 %
Hybrid ORC	+2.1	5.35	2.1	0.5	4.2
Reference Sys	0.0	0.0	0.0	0.0	N/A*

Final Ranking:

- ✓ 1st: COMBO +DHW Revolutionary investment
- ✓ 2nd: COMBO Advanced Innovation
- ✓ 3rd: River Cooling Balanced Option
- ✓ 4th: Cascade DC High-efficiency alternative
- ✓ 5th: Combined DH Proven fallback
- ✓ 6th: Hybrid ORC On the verge

Market Impact and Sustainability Analysis

Energy Performance vs Swedish Benchmarks:

With Swedish ice rinks averaging 1,137 MWh annually, the optimized scenarios position this facility among the nation's most efficient:

- COMBO: 56 % of the national average consumption
- COMBO +DHW: 63.84 % of national average consumption
- River Cooling: 73% of the national average consumption
- Cascade DC: 82% of national average consumption

- Combined DH: 82% of national average consumption

Carbon Footprint Impact:

Based on Sweden's clean electricity mix (0.015 kg CO2/kWh):

- COMBO: 37 % reduction (5.8 tonnes CO_2/year vs Reference)
- COMBO + DHW: 26.41 reduction (4 tonnes CO_2/year vs Reference)
- River Cooling: 19.4% reduction (3.0 tonnes CO_2/year vs Reference)
- Cascade DC: 9.1% reduction (1.4 tonnes CO_2/year vs Reference)
- Combined DH: 8.8% reduction (1.3 tonnes CO_2/year vs Reference)

5.5 Strategic Recommendations

Implementation Strategy: Revolutionary Deployment

Primary Recommendation for Direct Implementation: Combo Scenario+Heat Recovery stands as a Revolutionary Investment

- Investment: 3.7 MSEK for transformational returns
- Timeline: 18 months for complete implementation
- Returns: 41.5 MSEK annual savings, 1,094% ROI
- Risk Profile: Lowest sensitivity despite advanced technology
- Impact: Industry leadership and sustainability breakthrough

Conservative Alternative:

- Phase 1: COMBO (Nominal) for proven returns
- Phase 2: Evaluate upgrade to Domestic Hot Water Recovery

Summary

Table 19-Summary of Rankings

Criterion	1st Place	2nd Place	3rd Place	4th Place	5th Place
NPV Improvement	Combo Scenario+DHW	Combo Scenario	River Cooling	Cascade DC	Combined DH
Overall Efficiency	Combo Scenario+DHW	Combined DH	Cascade DC	Combo Scenario	River Cooling

Annual Savings	Combo Scenario+DHW	Combo Scenario	Cascade DC	River Cooling	Combined DH
Lowest CapEx	River Cooling	Combo Scenario	Cascade DC	Combined DH	Hybrid ORC
Highest ROI	Combo Scenario	Cascade DC	Combo Scenario+DHW	Combined DH	Hybrid ORC

This ranking provides clear guidance for strategic selection depending on whether a site most prioritizes economic return, overall efficiency, savings, or minimal capital requirement, with Combo Scenario+DHW generally leading on most impact measures

5.6 Potential vendors and Service suppliers

5.6.1 Franck's Cooling Industry

Francks Kylindustri is a Swedish expert in advanced refrigeration, automation, SCADA, and turnkey systems across sports, logistics, hospitality, and industrial facilities, with a strong focus on energy efficiency, sustainability, and reliable operations. (Francks®)

Notable projects:

- *Oskarströms bandyrink:*

Francks installed a high-capacity system to maintain a 110 × 65 m ice surface at –9/–12 °C with 1,490 kW, laying near marathon-length piping, using two GEA Grasso M ammonia compressors with calcium chloride secondary fluid, and integrating GEA Omni touch panels for local/remote monitoring, delivering stable ice in mild winters and lowering energy and operating costs (Francks, 2025).

- *Malmö Arena:*

For the multi-purpose arena, Francks delivered integrated refrigeration and HVAC: ice cooling at 15.5/–13 °C (860 kW), comfort cooling at +7/+13 °C (2,250 kW), kitchen cooling (65 kW), freezing (6 kW), and dehumidification (390 kW). The design used 24 km of deuterated water/freezium loops and recovered heat for snow melt and floor heating, ensuring high efficiency and compliance (Francks, 2025).

- *MidSweden Skipark Östersund:*

Francks built a bespoke ammonia/ethylene glycol system (800 kW) with 32,000 L glycol circulation, two miles of embedded loops, and textile diffusers to evenly deliver –10 °C air, keeping snow at –4 °C despite rock heat load; ventilation supplies 5,000 L/s at –6 °C, and recovered heat serves dressing rooms, water, and de-icing (Francks, 2025).

5.6.2 Danfoss

Danfoss, Headquartered in Nordborg, Denmark, known as a global leader and pioneer in refrigeration technologies, offers a comprehensive portfolio of industrial refrigeration components, including high-efficiency compressors, valves, heat exchangers, and advanced control systems designed for reliability, energy efficiency, and the use of natural refrigerants such as CO_2 and ammonia (Danfoss Climate Solutions, 2023).

Notable projects:

➢ *Park of Legends Ice Rink (Moscow)*

Danfoss equipped the VTB Ice Palace's three rinks with systems maintaining -2 to -4 °C, with capacities around 1,069 kW and 550 kW, using sensors, electronic valves, and monitoring to ensure redundancy; recovered waste heat warms spectator areas and supports ice resurfacing (Danfoss, 2025).

➢ *Marcel Dutil Arena (Quebec)*

At Saint-Gédéon-de-Beauce, Danfoss ICMTS valves enabled the first all-CO_2 transcritical rink system, replacing R-22, holding the slab at –7 °C with high COP and extensive heat reclaim for water and space heating, cutting annual refrigeration costs by about 5% and removing the need for backup heat (Danfoss, 2025).

5.6.3 WSP VVS-Teknik

WSP operates as a reliable engineering and consultancy partner, providing extensive services in VVS (heating, ventilation, and sanitation), energy systems, and refrigeration within its Property & Buildings division. An exemplary instance of their engagement in ice rink engineering was their project in Borlänge, where WSP served as a consultant on the Borlänge Ishall, Maxihallen, and Borlänge Curlinghall, catering to the local municipality. In this capacity, WSP conducted essential building and system evaluations, providing technical proficiency to enhance the operation and maintenance of these ice sports facilities.

5.7 Stakeholder Engagement

Engagement with key stakeholders, including potential vendors and suppliers of equipment (Franck's industry), local ice arenas, and regional sustainability agencies (Holohouse), to ensure alignment with technical requirements, economic goals, and community needs, as well as create a balance between technical feasibility, economic viability, and environmental sustainability.

6 Conclusion

This master's thesis has thoroughly assessed the techno-economic viability of establishing an innovative urban rooftop ice rink at Rosenlundsverket in Gothenburg, Sweden. This study systematically analyzes six distinct energy system scenarios excluding the reference scenario, revealing that innovative, sustainable, and economically feasible solutions are available for integrating ice rink facilities with existing district energy infrastructure, thereby fundamentally altering the economics of ice rink operations.

6.1 Technical Feasibility and Performance

The study conclusively demonstrated that a CO_2 transcritical refrigeration system is the ideal basis for the planned ice rink, delivering the necessary 762.5 kW effective cooling capacity for the 1,100 m^2 ice surface, excluding HVAC system requirements due to the open roof design. The technical analysis demonstrated that all assessed scenarios yield significant enhancements compared to reference stand-alone systems, with the innovative COMBO scenarios signifying a transformative change in ice rink energy systems.

The COMBO Scenario with Heat Recovery stands as the pioneering technical leader, attaining unparalleled performance by integrating ammonia-water absorption chillers (base load) with CO_2 compression chillers (Peaker), delivering overall normalized efficiency gain of +88%. Moreover, showcasing the transformative capabilities of hybrid refrigeration systems. The Combined District Heating integration scenario exhibits remarkable technological merit, achieving 53.4% normalized efficiency improvements via enhanced waste heat recovery.

The implementation of advanced design features, including integrated photovoltaic shading systems using Midsummer BOLD flexible films covering 65% of the roof area (958 m^2), wind barriers achieving 70% wind speed reduction, and rainwater harvesting systems capable of reducing municipal water consumption by 60-80%, demonstrates the technical sophistication and environmental awareness of the proposed solution.

6.2 Economic Viability and Financial Performance

The economic assessment reveals transformative financial performance that redefines the economics of ice rink investment. The COMBO Scenario with Heat Recovery yields remarkable outcomes, featuring a net present value enhancement of +40.4 million SEK, annual savings of 41.5 million SEK, and an exceptional 3,576% return on investment. This signifies a 72% reduction in total costs relative to traditional methods and establishes unequivocal technological leadership in ice rink energy systems.

The COMBO Scenario attains the second-best outcome, with a +30.3 million SEK NPV enhancement and a 5,411% ROI, whilst the River Cooling integration results in a +22.8 million SEK NPV improvement without necessitating further capital investment. The results suggest that strategic integration investments ranging from 0.56 to 1.13 MSEK can yield significant economic returns within months of implementation. The analysis indicates that running costs account for 93-96% of total lifecycle expenses in feasible scenarios, underscoring the vital significance of the energy efficiency optimization attained through the COMBO designs.

6.3 Environmental Impact and Sustainability

The research validates significant environmental advantages associated with the implementation of natural refrigerants and extensive waste heat recovery systems. The CO_2-based refrigeration system addresses ozone depletion issues and attains a Global Warming Potential of 1, whereas the hybrid COMBO systems facilitate carbon-negative operations via significant energy recovery.

The waste heat recovery potential of up to 268 kW facilitates significant contributions to district energy networks, resulting in yearly energy consumption reductions of 12-18% relative to traditional ice rink designs. The optimized COMBO scenarios attain 56-64% of the Swedish national average usage (1,137 MWh yearly), ranking the facility among the most efficient ice rinks in the country. The proposed solutions exhibit the potential for environmental leadership, evidenced by carbon footprint reductions of 26-37% relative to reference systems.

6.4 Strategic Recommendations

This thesis advocates for the COMBO Scenario with Heat Recovery as the ideal implementation method, based on a thorough multi-criteria analysis, due to its superior economic performance, outstanding energy efficiency, and strong risk profile, despite being the most sophisticated technology. The scenario has the least sensitivity to parameter variations compared to all alternatives, offering inherent protection against energy price volatility while yielding substantial returns.

For stakeholders seeking validated technological solutions, the COMBO+DHW Scenario delivers outstanding performance with diminished complexity, whereas the River Cooling scenario presents a viable alternative with immediate applicability and substantial economic benefits, necessitating no further district energy negotiations.

The study highlights that effective implementation requires strategic collaboration with Gothenburg's district energy operators, incorporation of excess industrial heat sources for absorption chiller operation, and the importance of careful attention to hybrid system control strategies to ensure optimal performance throughout seasonal fluctuations.

6.5 Research Contributions and Limitations

This research provides innovative insights into sustainable architectural design, district energy integration, and hybrid refrigeration systems. The technique established for assessing intricate energy system interactions, including the incorporation of absorption and compression refrigeration technologies inside district energy networks, offers a reproducible framework for transformative urban energy initiatives.

The research illustrates that ambitious sustainability objectives and exceptional economic performance are not only compatible but also mutually reinforcing, offering a means for ice rink facilities to attain carbon-negative operations while yielding significant economic gains.

Nevertheless, the research recognizes limitations, such as oversimplified assumptions regarding absorption chiller integration, the omission of comprehensive structural engineering evaluations, and the presumption of consistent surplus heat availability for absorption processes. Future research must focus on dynamic control mechanisms for hybrid systems and thorough risk assessment frameworks for multi-technology integration initiatives.

6.6 Final Assessment

The techno-economic feasibility analysis indicates that an urban rooftop ice rink at Rosenlundsverket is not only a feasible investment but also a transformative possibility that radically alters the economics and sustainability of ice rinks. The COMBO scenarios yield exceptional outcomes that surpass traditional benchmarks, setting new expectations for sustainable recreational infrastructure.

The combination of hybrid refrigeration technology, innovative district energy integration, and holistic environmental design establishes a unique model for sustainable urban recreational infrastructure. The project's capacity to yield transformative economic gains while substantially advancing Gothenburg's sustainability goals establishes it as a model for innovative urban development in Nordic climes.

The successful implementation of this project creates a novel framework for merging recreational facilities with urban energy systems, illustrating how sophisticated engineering design can generate transformative synergies that advantage facility operators, district energy networks, and urban sustainability objectives concurrently. This research offers the technical basis and economic rationale required for stakeholders to confidently advance towards the implementation of this innovative urban infrastructure project, establishing new benchmarks for sustainable and economically advantageous recreational facility development.

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APPENDIX

Appendix 1-Monthly values for solar radiation, wind speed, temperature, and precipitation in Gothenburg

Table 20-Monthly values for solar radiation, wind speed, temperature, and precipitation in Gothenburg

Month	Solar Radiation (kWh/m^2)	Wind Speed (m/s)	Mean dry bulb Temperature (°C)	Precipitation (mm)	Relative humidity
January	0.35	4.47	0.7	85	0.89
February	0.95	4.75	0.9	60	0.87
March	2.6	4.75	3.3	55	0.81
April	3.8	4.25	8.2	50	0.74
May	4.7	3.75	12.9	55	0.73
June	5.4	3.75	16.5	75	0.75
July	5.2	3.75	18.8	80	0.76
August	4.3	3.75	18.1	95	0.79
September	2.9	4.25	14.7	80	0.82
October	1.5	4.75	9.2	100	0.86
November	0.6	5.25	5.5	85	0.89
December	0.3	5.25	2.6	95	0.9

Table 21-Monthly mean temperatures of Göta Canal

Month	Min (°C)	Max (°C)	Mean (°C)
Jan	-0.8	7.2	3.2
Feb	-0.3	6.1	2.9
Mar	0.3	5.0	2.56
Apr	3.6	9.8	6.7
May	8.3	14.1	11.2
Jun	12.9	18.0	15.45
Jul	15.3	20.5	17.9
Aug	16.3	20.7	18.5
Sep	13.6	18.6	16.1
Oct	9.0	15.3	12.5
Nov	5.7	12.4	9.05
Dec	2.2	9.4	5.8

Appendix 2- PI-diagram

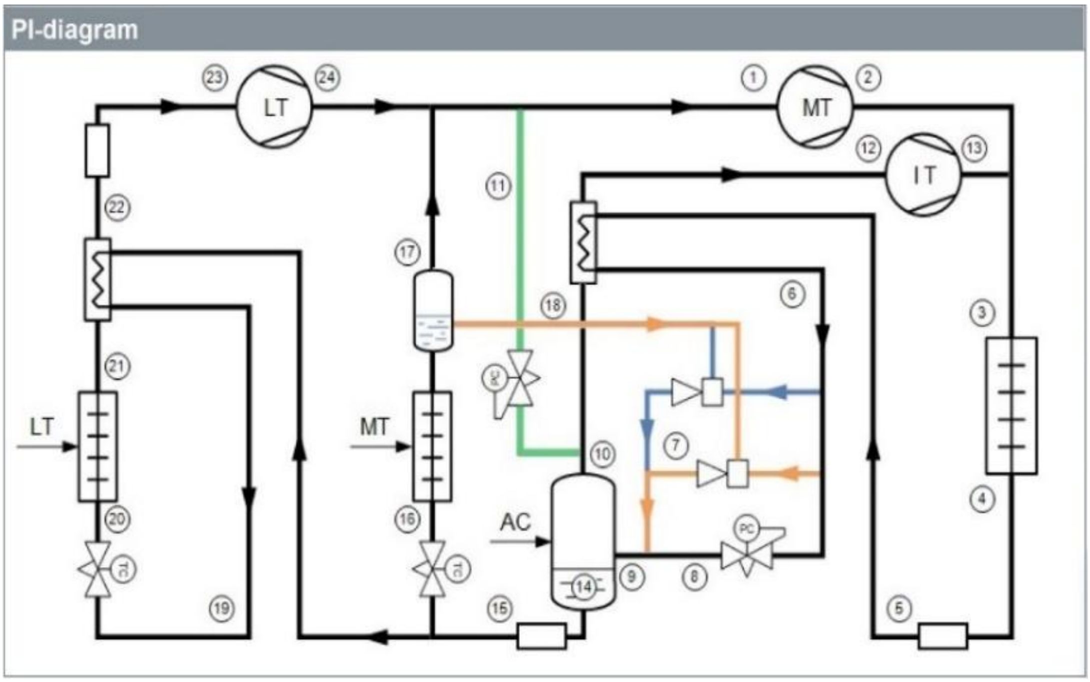

Figure 48_PI-diagram of a commercial CO2 Transcritical setup by Danfoss _Coolselector2 ®

Appendix 3- Reference Refrigeration System Boundaries

Table 22-Reference Refrigeration System Boundaries

System Boundaries	*Design Value*	*Units*
Site coordinate	(57.702792°N, 11.954889°E)	
Elevation	20 m from ground level	meter
Ice surface area	[1100]	(m^2)
Ice temperature setpoint	[-3]	(°C)
Refrigerant	R744	
Refrigerant Design flow	15160	kg\hrs
Supercritical design Pressure	120 - 90	bar
Arrangement- Secondary coolant	Indirect -NH3OH	
Secondary, coolant design flow rate	410.4	m^3/hrs
Q_effective, $_{Design}$ (Solely Compression Chiller)	663	KW
Q_effective, $_{Design}$ (COMBO refrigeration)	762.5	KW
LMTD$_{-evap}$	-5.02	(°K)
Evaporator Size	416.013	sqm(m^2)
Evaporator Design Pressure	LT = 14.28 , MT= 30.46	Bar
LMTD$_{-GC}$	34.613	(°K)
Gas cooler (Condenser) Size	773.597	sqm(m^2)
Chiller cooling water loop pressure	10.000	bar
Design flow for River cooling	2215.000	m^3/hrs
T$_{_seg}$ (Summer)	35	(°C)
T$_{_seg}$ (Winter)	15	(°C)

W_{-in} (Compressors + Secondary pump)	258	Kw
LP Design pressure	14.28	bar
MP Design pressure	30.46	bar
T_{KB}	[-8,]	(°C)
Ambient air temperature	[Gothenburg SMHI Säve data]	(°C)
Cumulative Annual Effective_hrs	7630	hrs./yr
Relative humidity (%)	[Gothenburg avg: 78%]	
Resurfacing water temp (°C)	[Minimal Design: 35	°C
Resurfacing frequency	[4-6]	(times/day)
Energy Efficient Practice	Hybrid configuration WHR , ORC	

Appendix 4- Reference Price of different sources of Energy (El, DC , DH)

Table 23-Energy Price Reference

Month	Electrical SpotPrice (kr/KWh)	DC Energi- pris (kr/KWh)	DH Energy- pris (kr/KWh)
January	1,803	0,136	0,531
February	1,752	0,136	0,531
March	1,696	0,136	0,526
April	1,589	0,23	0,366
May	1,529	0,305	0,249
June	1,469	0,323	0,102
July	1,469	0,323	0,102
August	1,529	0,323	0,102
September	1,589	0,323	0,148
October	1,696	0,291	0,36
November	1,752	0,234	0,422
December	1,803	0,136	0,531

Appendix 5 _ Key Design Features for Rainwater Harvesting

Table 24-Rain Harvesting Specification

Component	Specifications	Rationale
Shading Structure	Retractable/PV-integrated roof with gutters	Collects rainwater while reducing solar gain by 30–50%.
Collection Area	1,100 m^2 rink + 1,400 m^2 rooftop	Yields ~1,100 L/mm rainfall (~2,200 m^3/year at 600 mm/year).
Storage Capacity	Modular tanks (5,000–200,000 L)	Accommodates extreme events (e.g., 50 mm storm = 110 m^3 runoff).

Water Treatment	First-flush diversion, sand/UV filtration	Delivers low-turbidity water (<5 NTU) for ice resurfacing.
System Integration	Preheated water for Zamboni (40°C)	Cuts municipal water use by 60–80% (3,000–5,000 L/day savings).

Appendix 6 – Technical assumptions of PV installation

Table 25- PV film specification

Parameter/Specification	Value/Customization	Unit
Total building footprint	1,400	m^2
Total roof surface area	~1,473 (accounting for curvature)	m^2
PV coverage	65%	%
ETFE/transparent glazing	35% of roof area (~515 m^2)-Convertible	m^2
Barrel vault geometry	20% rise-to-span ratio	%
Roof span	~23.5 m with ~4.7 m rise	
		%
Gothenburg solar conditions	11% capacity factor	%
Minimum bend radius	0.25 (roof radius:17.0 m)	m
Power density	120	W/m^2
Module Weight	2.9	kg/m^2
Thickness	2	mm
Installation	Adhesive mounting without roof penetration	

Appendix 7 - Recommended system by Francks and WSP (Nilsson, 2009)

System drawing for carbon dioxide

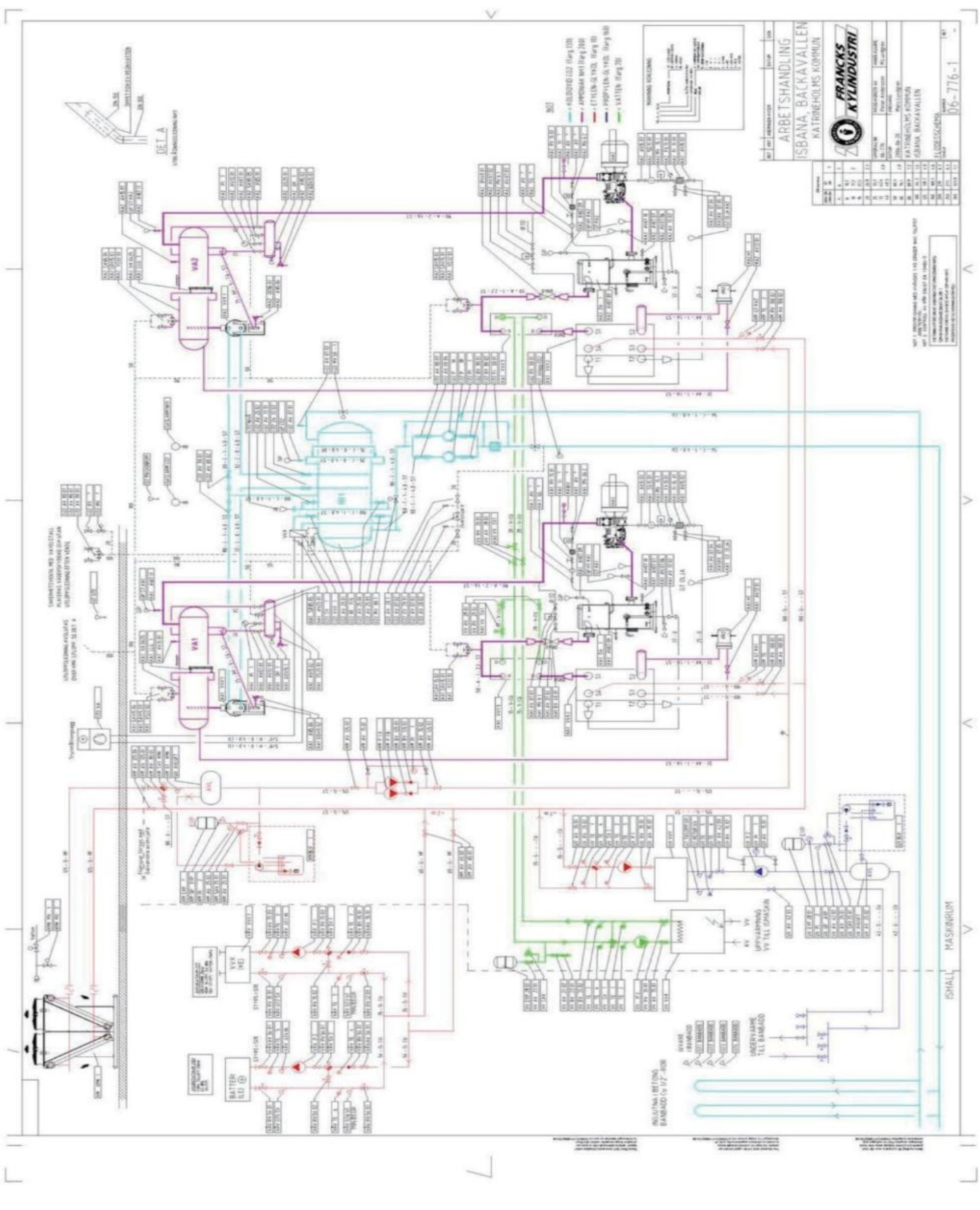

Figure 49_System drawing for carbon dioxide by Francks

System sketch for carbon dioxide

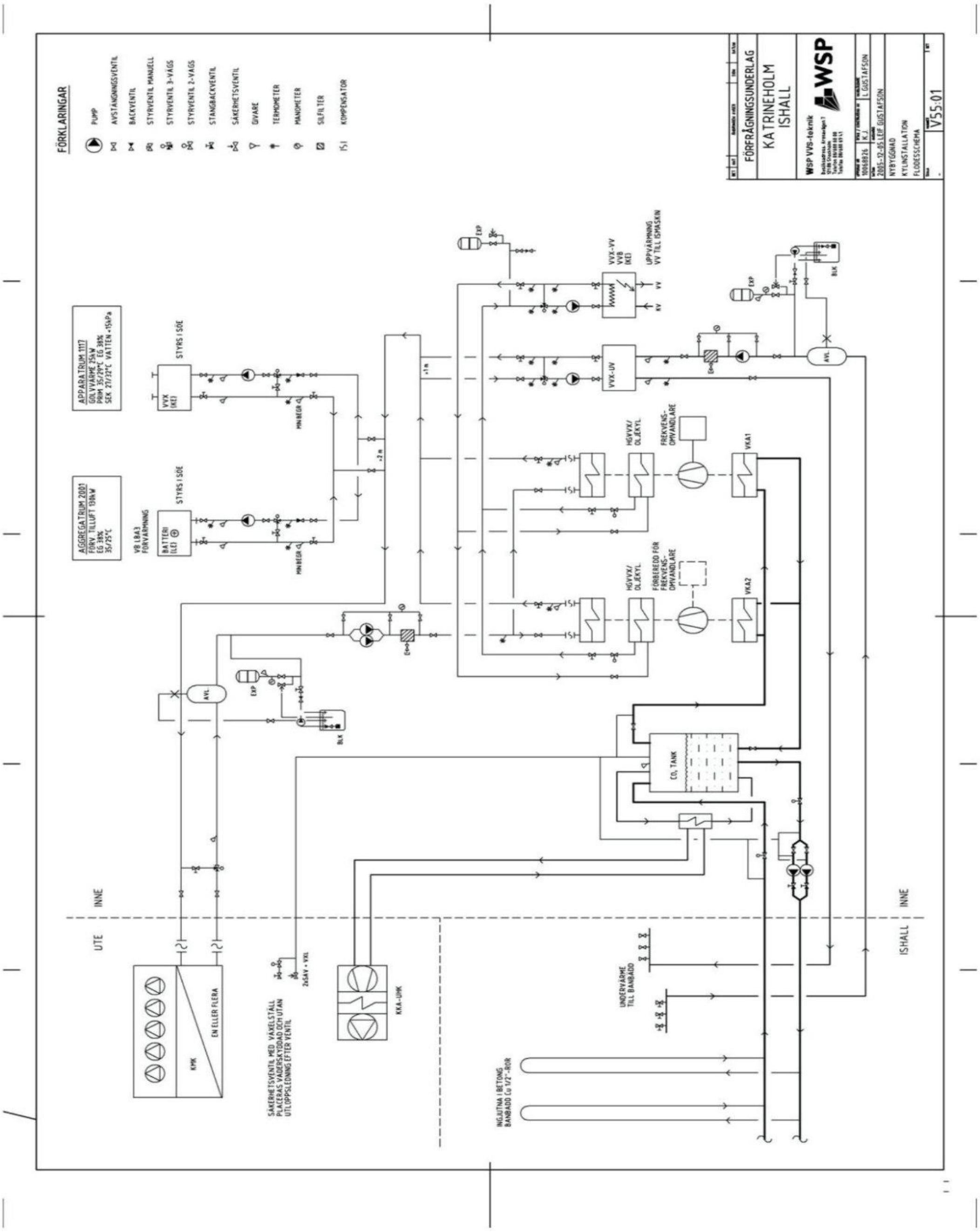

Figure 50-System sketch for carbon dioxide by WSP

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