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Report

Heat-driven snow production

A review of potential refrigeration technologies, heat sources and winter sports locations for temperature independent snow production

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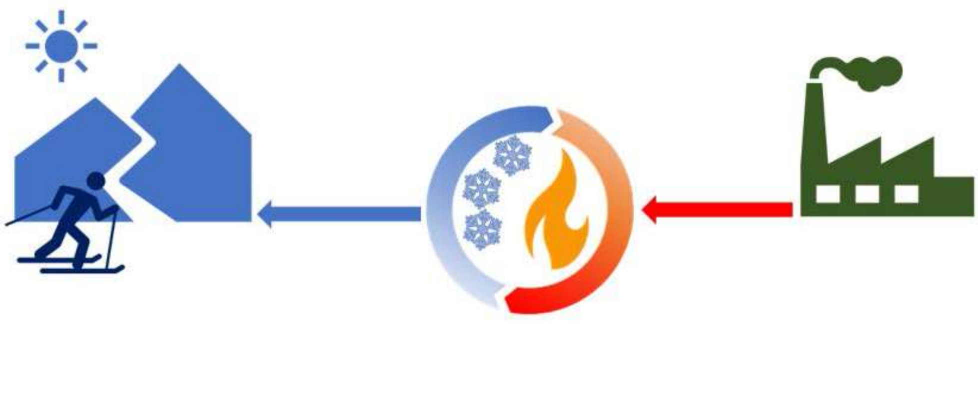

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ABSTRACT

Global warming is causing increased temperatures and reduced snow cover, which threatens the possibility to exercise winter sports activities near populated areas in the future. Temperature Independent Snow (TIS) production can contribute to increase the snow reliability for winter sport. However, this method is very energy intensive and to reduce operational costs the use of heat instead of electricity as energy source is a potential solution.

Absorption appears to be most promising among the possible heat driven refrigeration technologies. Heat supply temperatures can be as low as 85-90°C, which opens the possibility for using district heating. Calculations showed that a thermally driven refrigeration system for TIS production is more than twice as energy intensive compared to using electricity and will depend on low energy costs to be competitive.

The review of potential heat sources shows that there is a potential for utilization of district heating and industrial waste heat. District heating is the most prevalent and about 40 % of the identified winter sport facilities are in municipalities where this is offered. However, low temperatures and lack of available heat are potential challenges. Industrial waste heat exists in large quantities at sufficient temperature levels but is less co-located with the winter sports facilities. Furthermore, there are important factors related to cost and technology, which can limit the potential for utilization.

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Table of contents

1 Introduction

This report is part of the "Snow for the future" project and discusses the potential for utilizing heat driven refrigeration technologies for temperature independent snow production (TIS). The goal of this report is to map potential locations where heat, e.g. industrial waste heat, district heat or other types of heat sources are available and can be utilized for TIS to nearby winter sport locations.

Global warming is causing increased temperatures and reduced snow cover, which threatens the possibility to exercise winter sports activities near populated areas in the future. One of the strategies that can contribute to increase the snow reliability for winter sport is the use of Temperature Independent Snow (TIS) production. This is a method which is not restricted by the ambient temperature, unlike traditional snow production methods which require temperatures below freezing. In previous work conducted in the "Snow for the future project" TIS production has been found to be able to increase the length of the season (Trædal, 2018a), (Tveten, 2020). However, both operational and investment costs have been found to be significantly higher than traditional Temperature Dependent Snow (TDS) production.

The high operational costs of TIS production are mainly due to its high energy (electricity) consumption. Commercial systems used for snow production have a specific energy consumption in the range of 16 – 90 kWh/m^3 of snow produced (Trædal, 2017). In comparison TDS production systems are in the range of 0.5 – 6 kWh/m^3 (*snøkompetanse.no*, 2020b).

Reduced operational costs of TIS production machines may increase their attractiveness and lead to more extensive use in winter sports locations in Norway. One method to reduce the costs is to replace electricity with heat as the energy source. This may also lead to reduced CO_2 emissions compared to using electricity. Heat driven refrigeration systems are used for various industrial and domestic applications, and commercial large-scale systems for heat driven ice production are in existence today.

There have been several studies (Sollesnes and Helgerud, 2009), (Oslo Economics / Asplan Viak, 2020) which have identified that there is a large potential for utilization of energy from heat in Norway, e.g. in the form of industrial waste heat or district heating.

An important aspect of using heat to produce snow is that the number of applicable winter sports locations is limited compared to electrically driven TIS production, since the distance between the heat source and the location where the snow is needed is restrained by both cost and technical challenges of transporting either heat or snow over large distances.

In the next chapters, potential heat driven cooling technologies applicable for snow production are reviewed and compared. Moreover, based on the results from previous work with TIS production in "Snow for the future" a calculation of the heat requirement for a heat driven TIS production system is performed. Then, a review of available heat sources is presented. Finally, potential winter sports locations in Norway are mapped, along with nearby sources of heat in order to identify suitable locations for heat driven TIS production.

2 Overview of heat driven refrigeration technologies

In this chapter a review of the current existing technologies for thermally driven refrigeration systems is presented.

Thermally driven refrigeration systems have been in existence since the 18th century, and were frequently adopted by the 20th century. However, after the development of cheap, reliable and efficient electrically driven mechanical vapor refrigeration systems, thermally driven refrigeration systems were mostly outcompeted, and mainly being preferred for industrial processes when large amounts of waste heat has been available, or in niche product segments such as absorption refrigerators for minibars in hotels due to its silent operation (Best and Rivera, 2015).

However, in the last few decades thermal refrigeration systems have gained attention due to the increased focus on the use of waste heat and renewable energy. At the same time, the cooling and refrigeration demands have increased, both for industrial and household use. This has led to extensive research and development of efficient alternatives to electrically driven heat pumps, which instead can utilize industrial waste heat, district heating or solar energy. (Deng, Wang and Han, 2011), (Nikbakhti *et al.*, 2020).

Commercially available thermally driven refrigeration systems have been dominated by sorption technologies, of which absorption is the most dominant (Deng, Wang and Han, 2011). Other technologies are also currently being researched and developed such as adsorption refrigeration and thermo-mechanical cooling systems. Based on the literature review, four heat driven refrigeration technologies presently stand out as the most promising:

- Absorption refrigeration systems
- Adsorption refrigeration systems
- Ejector based refrigeration systems
- Rankine based refrigeration systems

A further review of these technologies is conducted in the following sections.

2.1 Coefficient of performance and temperature requirements

Compared to compressor driven refrigeration systems where pure exergy in the form of electricity is supplied to produce cooling, thermally driven refrigeration systems are limited by the second law of thermodynamics, causing only the exergy share of the heat available to supply cooling power, which greatly reduces the theoretical Coefficient of Performance (COP).

This principle is easier understood if the thermally driven cooling cycle is divided into two thermodynamic engines: A Carnot heat engine and a Carnot refrigeration engine, given in Figure 1:

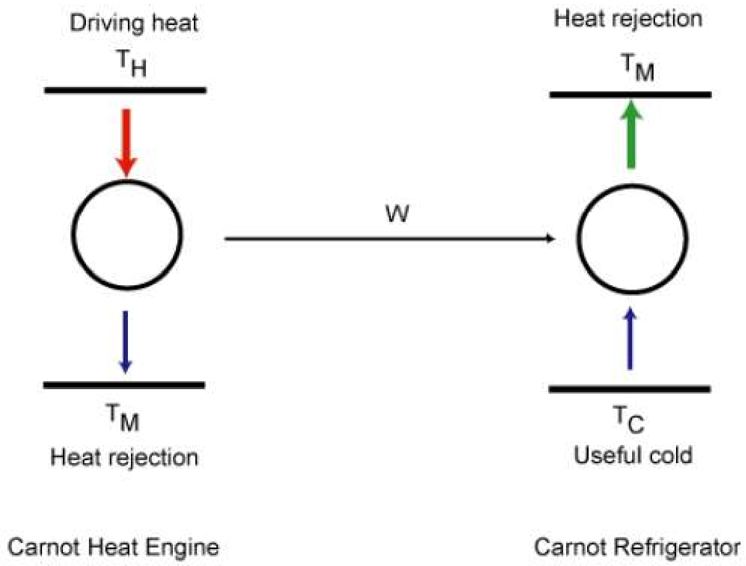

Figure 1: Thermodynamic principle of the thermally driven refrigeration system (Thomas, 2013)

As seen in Figure 1 the thermally driven refrigeration system is governed by three temperature levels, the high temperature T_H, which is the temperature where heat is supplied, T_M, which is the temperature level where heat is rejected to the surroundings, typically the ambient temperature level, and T_C, which is the cooling output temperature level. In this case, the temperature required to produce snow.

In the heat engine, the heat supplied can produce useful work (exergy), which can be used to drive the refrigeration engine to produce cooling. The COP for the thermal refrigeration machine is based on the ratio between the cooling output and the heat input:

$$COP = \frac{Q_C}{Q_H}$$

Q_c: Cooling output
Q_H: Heat input

The theoretical maximum COP can be found by combining the maximum efficiencies for the Carnot heat engine and the Carnot refrigerator. The maximum efficiency for a heat engine is limited by the Carnot efficiency:

$$\frac{W}{Q_H} = \frac{T_H - T_M}{T_H}$$

W: Work

The maximum COP for the refrigerator is given by the Carnot COP:

$$\frac{Q_c}{W} = \frac{T_C}{T_M - T_C}$$

Combining these two equations gives the maximum COP for a thermally driven refrigeration system:

$$COP = \frac{Q_c}{Q_H} = \frac{Q_c}{W} * \frac{W}{Q_H} = \frac{T_H - T_M}{T_H} * \frac{T_C}{T_M - T_C}$$

Figure 2 shows graphically how the COP from this equation varies with different heat supply temperatures (T_H), and ambient temperatures (T_M) for a cooling temperature (T_C) of – 20 °C, which is a typical evaporation temperature for e.g. flake-ice production machines (Trædal, 2017). Compared to mechanical vapor compression refrigeration systems (VCRS) the COP is significantly lower. As an example, an ambient temperature of 10 °C and a heat supply temperature of 100 °C gives a theoretical COP of 2, while a VCRS working with the same conditions can achieve a theoretical COP of 8.4. In practice, COP values for large capacity commercial heat driven refrigeration system producing ice can be expected to be around 0.25 – 0.6 (Deng, Wang and Han, 2011).

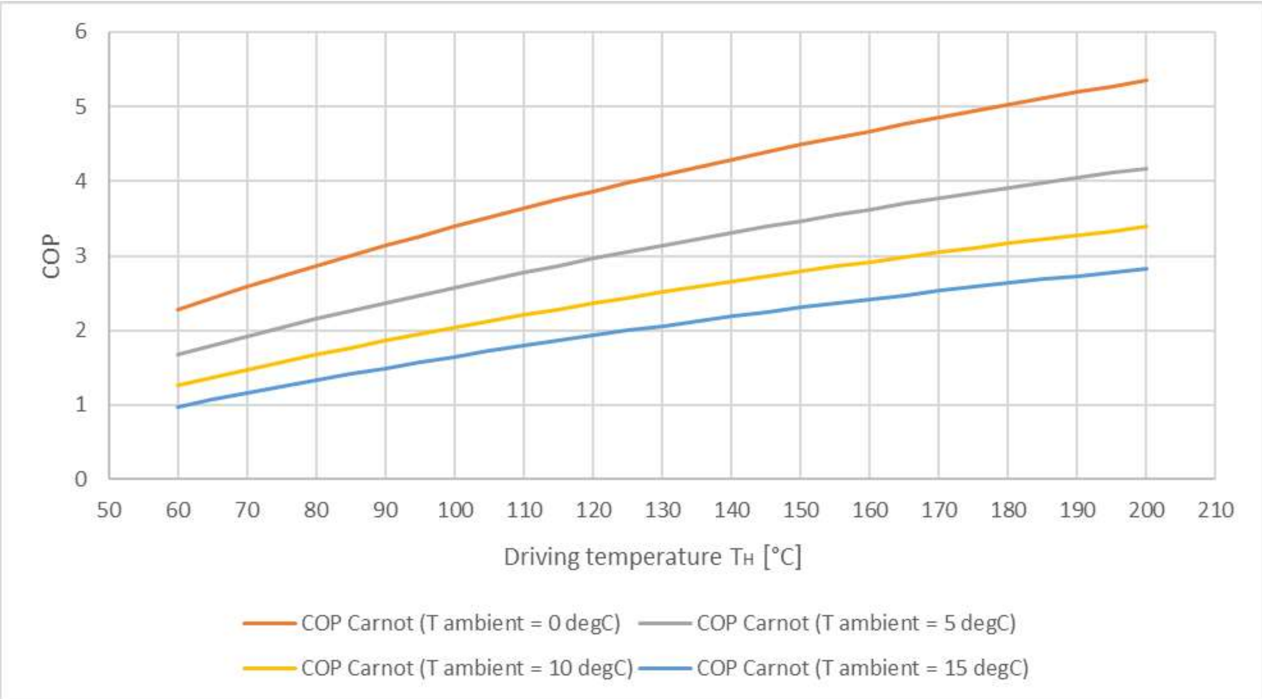

Figure 2: Carnot COP for a thermally driven refrigeration system vs driving temperature (heat supply temperature) for various ambient temperature levels, T_C = - 20 °C

An important aspect from the figure is that the ambient temperature level has a great effect on the theoretical COP for thermal refrigeration systems, even greater than for VCRS. It is therefore of importance to evaluate which ambient conditions these systems can operate within, with respect to the available heat supply and the cooling demand.

2.2 Absorption based refrigeration systems

The absorption refrigeration technology dates back to the 18th century and was frequently adopted by the early 20th century. Compared to other thermally driven refrigeration systems, absorption is the most widely adopted (*bine.info*, 2020).

Besides being able to operate based on low grade heat, absorption refrigeration systems have other benefits, such as the possibility to use both environmentally and climate friendly refrigerants, quiet operation due to almost no high-speed moving parts and high durability. Very low electricity consumption, and the ability to run on partial load with high efficiencies (Dominik Schöpfer and Alberto Caiado Falcão de Campos, 2015).

The disadvantages compared to VCRS are low COP, high investment costs, large installation size and larger required heat rejection capacity. The latter is because the overall supply of energy to run these systems are much higher than for mechanical vapor compression refrigeration systems (Dominik Schöpfer and Alberto Caiado Falcão de Campos, 2015).

2.2.1 Working principle

Absorption refers to a process where a substance in one phase is incorporated into the bulk volume of another substance of a different phase. Absorption is an exothermic process, and the solvability of the two substances depends on the temperature. This principle enables the use of heat to drive the refrigeration system since low temperatures leads to absorption, while high temperatures leads to desorption.

A simple absorption refrigeration system is shown in Figure 3. The compressor found in a typical mechanical vapor refrigeration system is replaced with a generator, an absorber, a liquid pump, and a solution expansion valve. Absorption refrigeration requires a refrigerant and an absorbent, forming a working pair. The most common working pair is water as refrigerant and lithium-bromide solution as absorbent, known to give good performance and is environmentally friendly. However to achieve cooling below 0°C, the most common working pair is ammonia-water, where ammonia is the refrigerant and water is the absorbent (Wu *et al.*, 2014).

The refrigerant is evaporated at low pressure in the evaporator producing the cooling effect from the heat of evaporation. Attraction forces between the absorbent, and refrigerant vapor sucks the refrigerant into the absorber where it is absorbed. This process releases heat and cooling is required to maintain the absorption rate.

The absorption refrigeration system can now utilize the fact that the refrigerant now is in a liquid state due to being absorbed by the absorbent and a liquid pump can therefore be used to pump the refrigerant to the high pressure side of the cycle. This almost eliminates the use of electric power compared to using a compressor.

In the regenerator thermal energy is applied to desorb the refrigerant from the absorbent. The absorbent is cycled back to the absorber via an expansion valve to repeat the absorption process, while the refrigerant is condensed in the condenser by heat rejection to the ambient. It is then expanded back into the evaporator via an expansion valve and the cycle is repeated (Nikbakhti *et al.*, 2020), (Deng, Wang and Han, 2011).

The use of an ammonia-water pair requires the use of a rectifier because some of the water absorbent is vaporized together with the ammonia in the generator. The rectifier removes the water vapor and ensures that pure ammonia enters the condenser. Without the rectifier water would build up in the evaporator and the cooling capacity would be reduced (Wu *et al.*, 2014).

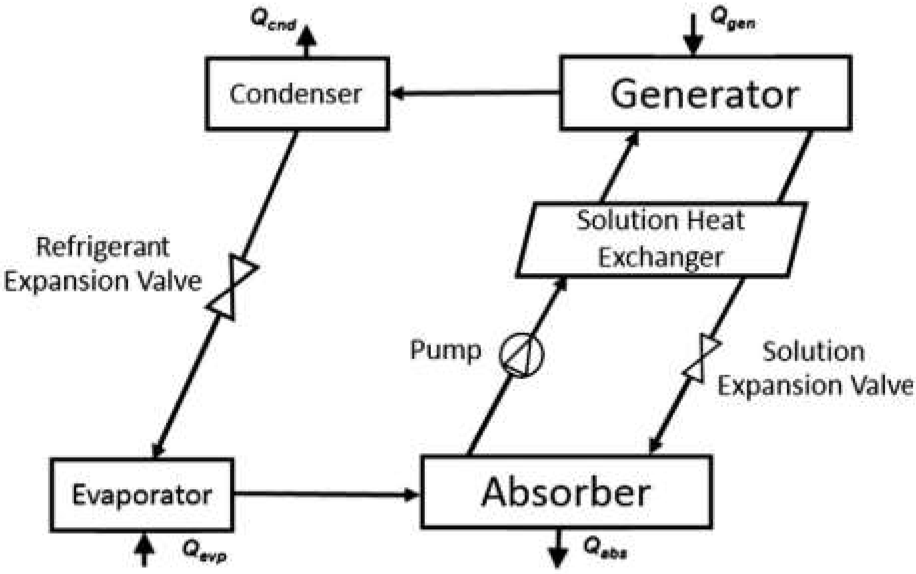

Figure 3: Schematic of a single-effect absorption refrigeration system (Nikbakhti *et al.*, 2020)

2.2.2 Technological development

The cycle shown in Figure 3 is a so-called single effect cycle, which is the simplest and most common configuration of an absorption refrigeration system. However, modified cycles with higher performance have been successfully developed and commercialized within absorption refrigeration (Henninger *et al.*, 2011). The purpose of the modifications is to increase the COP, reduce the evaporation temperature for increased cooling demands or to enable the use of other heat supply temperatures. Most modifications are employed to re-use heat from different stages in the cycle. A minor modification from the basic cycle in Figure 3 for example is the use of a solution heat exchanger to recover sensible heat from the regenerated absorbent (S. Henninger, K. Witte G. Fuldner et al).

To further increase the performance, absorption systems with multiple stages (multi-effect cycles) have been developed. Figure 4 shows a double effect cycle as an example. It has two compression steps and two generators. The idea behind the multi-effect cycles is to re-use the heat generated in the condenser in the low temperature generator. The energy-recovery increases the COP drastically and compared to single-effect cycles, the COP almost doubled when using a double-effect cycle from 0.6-0.8 to 1.0-1.3, as seen in Figure 5. The increased COP is also a consequence of increased generator temperature, made possible by introducing more stages. As seen in Figure 5, the single, double and triple effect cycles have different applicable heat source temperature ranges, with double and triple-effect cycles allowing the utilization of high temperature waste heat (Nikbakhti *et al.*, 2020). For situations where low temperature waste heat sources up to 100 °C is available, such as district heating, this is best utilized by using the single-effect cycle.

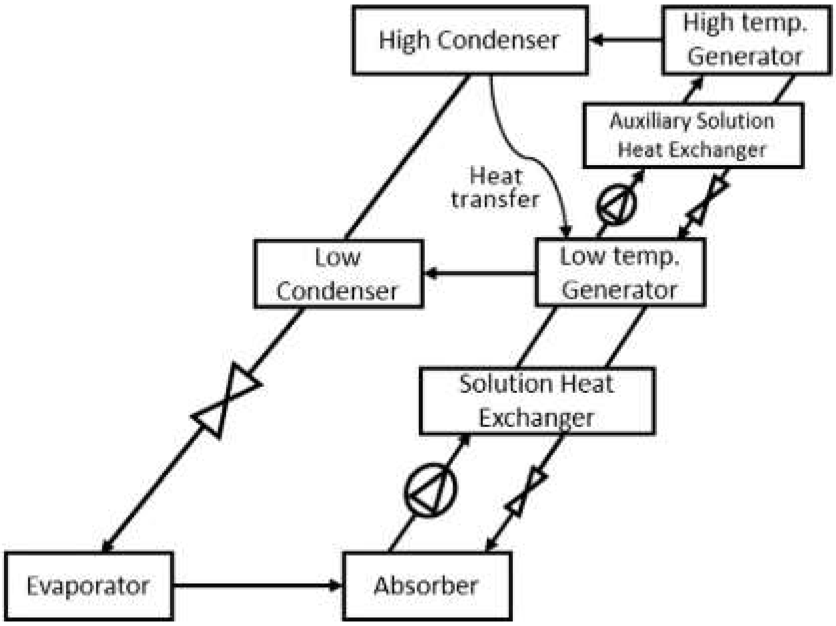

Figure 4: Schematic of a double-effect, parallel flow absorption refrigeration system (Nikbakhti *et al.*, 2020)

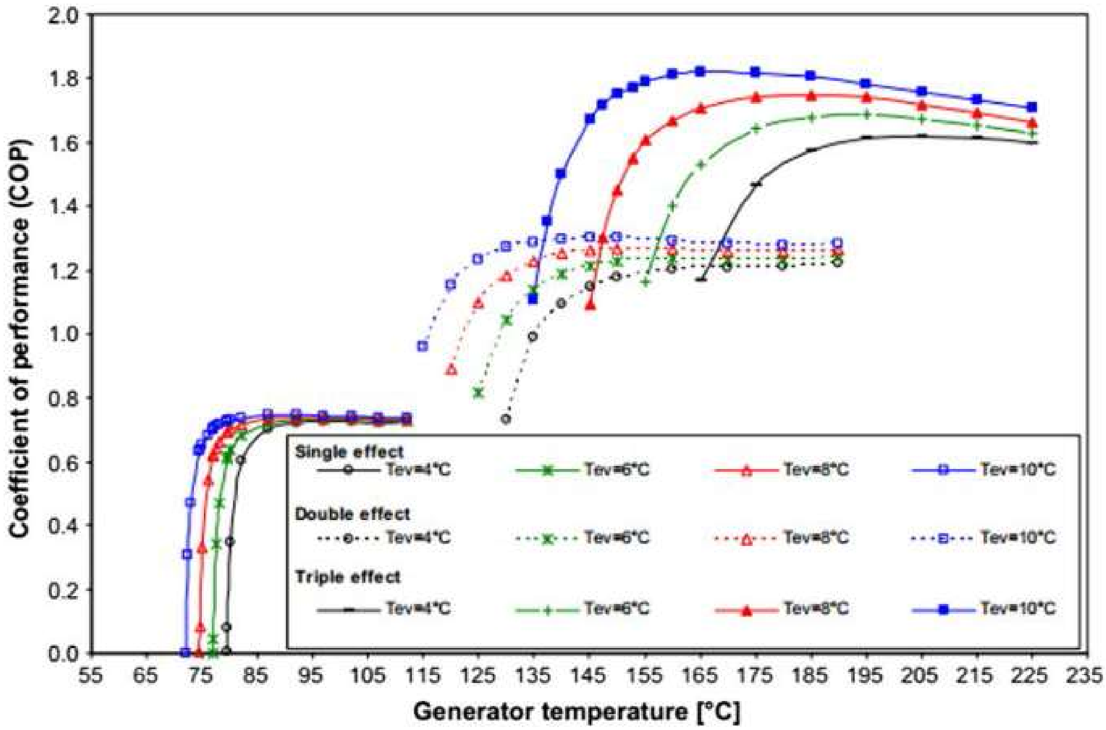

Figure 5: COP comparison between single, double and tripling effect absorption refrigeration systems (Nikbakhti *et al.*, 2020)

2.2.3 Commercial Status

Most commercial suppliers of absorption chillers base their systems on LiBr-water working pairs and delivery systems with cooling temperatures above 0°C. However, there exist several commercial suppliers of industrial absorption chillers for freezing applications. These mostly use ammonia-water based working pairs and single or dual stage absorption cycles to achieve cooling temperatures well below 0°C. A couple manufacturers have achieved cooling slightly below 0°C using LiBr-water working pairs that operate on so-called single-effect double lift cycles.

The following is a list of companies offering absorption-based refrigeration technologies capable of delivering cooling temperature below 0°C:

Colibri BV (Netherland) is a company that offers single or dual stage ammonia-water based absorption refrigeration systems capable of delivering cooling down to -60°C, with capacities up to several MW. For an evaporation temperature of -20°C, and a cooling water temperature (T_M) of 10 °C, a COP of 0.6 can be expected. A minimum heat supply temperature of approximately 90-110 °C, is expected depending on the cooling water temperature (*colibris.home.xs4all.nl*, 2020).

Ago AG (Germany) is a company that offers an ammonia-water based unit capable of delivering refrigeration outlet temperatures down to -40°C and a cooling capacity up to 1500 kW. Industrial waste heat in the form of water, steam or thermal oil can be used with supply temperatures ranging from 85-160°C (*ago.ag*, 2020).

York (United States) is a company that offers the YHAU-C-L absorption chiller, which can deliver -5°C refrigeration using LiBr-water as working pair, using a two-step evaporator and absorption cycle. The unit has a capacity up to 1800 kW and can utilize waste heat sources such as water, steam and exhaust gases. The unit is sold in Europe through Johnson Controls (*johnsoncontrols.com*, 2020).

World Energy CO. LTD. (South Korea) offers a double-lift single effect LiBr-water based absorption chiller able to supply brine at -7°C with a capacity up to 7000 kW. Applicable heat sources are hot water, steam, exhaust gas or fossil fuel with a driving heat of approximately 95°C. The COP is approximately 0.4 (*worldenergy.co.kr*, 2020).

Energy Concepts Co LLC (United States) is a company that offers an ammonia-based absorption chiller called the ThermoChiller, which has been used for ice production applications. The chiller can provide cooling down to -46 °C from low grade heat sources with units with several MW capacity. According to their estimation tool, producing ice at -20°C, requires a heat source temperature of approximately 90°C. If it can be produced at -10 °C, the heat source temperature can be reduced to around 83°C. The calculations are based on cooling water available at 10°C. The COP values for these conditions are around 0.6 (*energy-concepts.com*, 2020).

Transparent Energy Systems PVT. LTD. (India) is a company that offers ammonia-based absorption refrigeration plants able to provide cooling down to -55°C with capacities ranging from 18 – 3500 kW. Ice production is one of several applications (*tespl.com*, 2020).

2.3 Adsorption based refrigeration systems

Similar to absorption, the adsorption-based refrigeration technology is an old and well-known technology that was outcompeted by the introduction of VCRS in the 20th century. However, together with other heat driven refrigeration technologies, adsorption-based systems have gained more attention the last few decades, with several ongoing research attempts and product commercialization (Deng, Wang and Han, 2011), (Henninger *et al.*, 2011).

Adsorption-based refrigeration systems have many of the same advantages as absorption-based systems. However, they are capable of utilizing even lower heat supply temperatures, down to 60-90°C, which opens the possibility for other sources of heat. Furthermore, in contrary to absorption, there is no need for liquid pump, and there are few moving parts in general. This leads to a very low electricity consumption, further reduced maintenance and noise levels, which has made the technology attractive for utilizing solar heat in remote locations to produce cold. The disadvantages with the adsorption cooling systems are that they have even lower COP values and are relatively bulky in size compared to the cooling capacity (Deng, Wang and Han, 2011).

2.3.1 Working principle

Adsorption refers to a process where atoms, molecules or ions accumulate on the surface of a solid material. It is caused by weak van der Waal forces between the adsorbent (solid) and the adsorbate (refrigerant), and is the most common and promising category for cold production (Vingelsgård, 2019). The solid adsorbent consists of a porous media with pores, cavities and channels creating a large surface area necessary for adsorption. Like absorption, the process of adsorption releases heat, while desorption requires heat.

A refrigeration cycle utilizing this principle is shown in Figure 6, with the corresponding PT diagram shown in Figure 7. It consists of two adsorption beds filled with an adsorbent material acting as both adsorber and desorber. The system also contains a condenser and an evaporator, valves between each component and a throttling valve between the condenser and evaporator (Pal *et al.*, 2016).

Since the adsorption process shown in Figure 6 uses fixed beds this results in intermittent cycle operation, with the beds changing between adsorption and desorption operation. To achieve continuous cooling and increase COP it is therefore necessary to use two or more beds operating in opposite stages (Fan, Luo and Souyri, 2007).

There are four main steps in the adsorption refrigeration cycle: Pre-heating, desorption, pre-cooling and adsorption:

Pre-heating is performed to provide pressure and temperature conditions suitable for desorption. This is conducted in the adsorbent bed, with the valves connecting the bed to the condenser and evaporator in closed positions. The refrigerant vapor is already adsorbed in this stage, and the added heat from an external heat source combined with a fixed volume, causes the pressure and temperature to increase. The heat source may typically consist of hot water or steam.

In the desorption process, the heat input to the adsorbent bed continues, and the valve between the bed and the condenser is opened. The pressure is kept constant, while the temperature continues to increase in this process, which causes the refrigerant vapor be desorbed from the adsorbent. The vapor flows to the condenser where rejection of heat to cooling water causes it to condense. Finally, a throttling valve expands the condensed refrigerant into the evaporator.

Pre-cooling is performed to prepare pressure and temperature conditions suitable for adsorption and is conducted in the adsorbent bed with all connecting valves closed. Heat is rejected using cooling water, and the pressure is reduced to the same level as the evaporator.

In the adsorption process the valve between the adsorbent bed and the evaporator is opened. Low temperature heat from an external source, causes the refrigerant to evaporate, which produces the cooling effect, and then flow into the adsorbent bed. The vapor is then adsorbed in the pores of the adsorbent until saturation is reached. Since the adsorption process generates heat which needs to be removed, this is rejected through heat exchange with cooling water. (Demir, Mobedi and Ülkü, 2008).

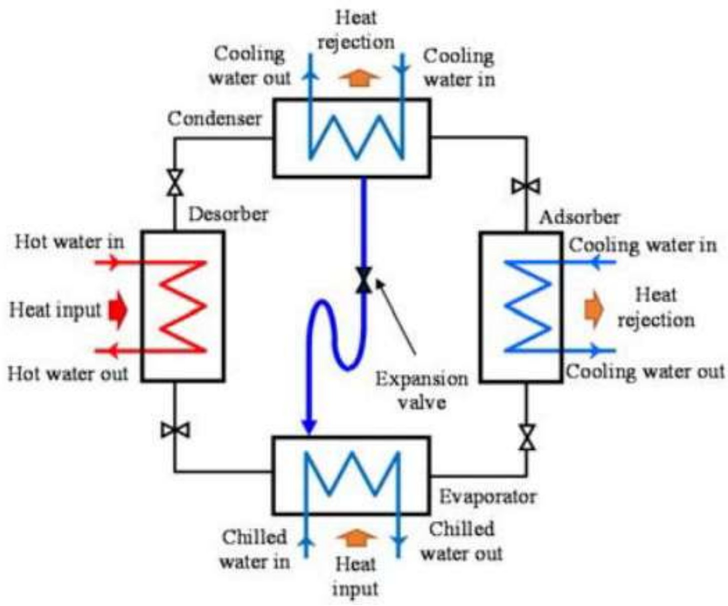

Figure 6: Schematic of an adsorption cycle, (Pal *et al.*, 2016)

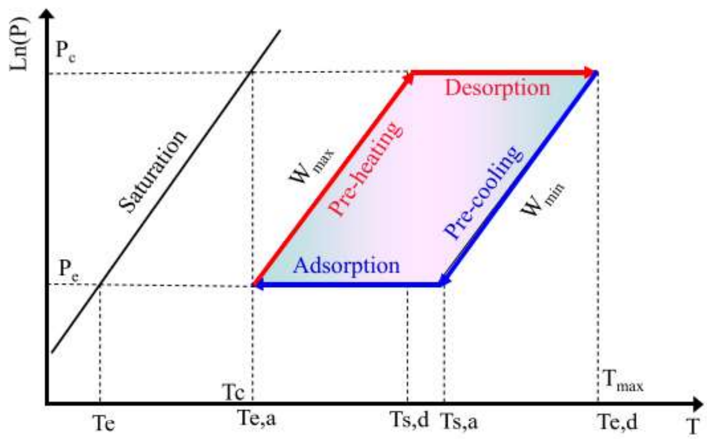

Figure 7: Thermodynamic process of an adsorption cycle, (Pal *et al.*, 2016)

2.3.2 Technological development

There have been several studies reviewing the technological development of adsorption refrigeration systems (Deng, Wang and Han, 2011), (Sah, Choudhury and Das, 2016), (Best and Rivera, 2015), (Younes *et al.*, 2017) and (Henninger *et al.*, 2011). The research seems to focus on the study of adsorbent and adsorbate working pairs, rather than modifications to the basic adsorption cycle. Thermodynamic and chemical properties cause the adsorbent to exhibit different attractions towards different adsorbate mediums and choosing a suitable adsorption pair is therefore of high importance to create an adsorption system with high performance.

The most common working pairs for cold production are zeolite-water, silica gel-water, activated carbon-ammonia and activated carbon-methanol (Henninger *et al.*, 2011). The use of silica gel-water are non-toxic, non-corrosive, can be driven by 60-90℃ waste heat and has been commercialized (Deng, Wang and Han, 2011). However, since water is the adsorbate it is not suitable cooling below 0°C.

Reviews on adsorption refrigeration systems for cooling below 0°C show that the use of activated carbon-methanol is suitable with waste heat sources down to 80°C, achieving COP values in the range of 0.1-0.2. The use of activated carbon – $CaCl_2$-NH_3, is applicable with higher temperature heat sources up to 120-140°C, with COPs reaching 0.4 (Deng, Wang and Han, 2011), (Sah, Choudhury and Das, 2016). As an example, a waste heat powered adsorption refrigeration ice maker was developed by Shanghai Jiao Tong University to provide partial cooling power for conserving fish on fishing boats (Sah, Choudhury and Das, 2016).

2.3.3 Commercial status

Although the adsorption refrigeration systems have been commercialized, there exist few manufacturers and there are currently no existing adsorption systems delivering sub-zero temperature cooling available for commercial use. Adsorption-based ice makers only exist as research prototypes. Low COP values, bulky size and high cost are limitations that currently stand in the way of commercializing adsorption chillers able to produce cooling below 0°C. There is ongoing research on developing new materials for adsorption beds, such as compounds or nano particles, which can increase heat and mass transfer. Together with optimization of advanced adsorption cycles this may increase performance and make it commercially attractive (Vingelsgård, 2019), (Sah, Choudhury and Das, 2016). Some commercial suppliers of MW-scale adsorption systems today for cooling above 0°C are Mayekawa (Japan), Bry-Air (United Arabic Emirates) and Nishiyoudo Kuchouki Co. Ltd (Japan).

2.4 Ejector based refrigeration systems

Ejector based cooling systems are categorized as thermo-mechanical cooling systems. In these systems heat is used to produce mechanical work, which then is used compress the refrigerant in a refrigeration system. The ejector itself is a form of jet activated pump typically used for pumping gases from systems to produce a vacuum or for vapor compression (Aidoun, Giguère and Scott, 2011). The ejector technology was invented in the early 20th century and has since been adopted for industrial cooling applications in the form of steam ejector chillers (Chunnanond and Aphornratana, 2004), (Liang *et al.*, 2020).

2.4.1 Working principle

The schematic view of an ejector is shown in Figure 8. The working principle of the ejector is based on the venturi effect, enabling a high-pressure motive flow to compress a low-pressure suction flow. The motive flow enters the 'primary' or motive nozzle with a with a high pressure, where reduced flow area causes it to expand and accelerate. The pressure at the outlet of the motive nozzle is lower than the pressure of the suction flow and causes it to be sucked in from a suction chamber. The two flows then mix in the mixing chamber, before entering the diffuser where the increased cross sectional area causes the pressure recovery (Grimsby Institute, no date).

Figure 8: Schematic view of an ejector (Aidoun, Giguère and Scott, 2011)

A simple thermally driven refrigeration cycle using an ejector is shown in Figure 9. It consists of two loops, the power loop, and the refrigeration loop. In the power loop, the refrigerant is heated in the generator at high pressure until it evaporates. It then flows into the ejector where it mixes with the low-pressure refrigerant from the refrigeration loop, before being condensed at an intermediate pressure in the condenser. The refrigerant is then divided into two streams, one is recirculated by a pump, which typically requires about 1 % of energy used for the heat supply (Chunnanond and Aphornratana, 2004), back to the generator in the power loop, while the other is expanded to an evaporator where the cooling effect is achieved (Grimsby Institute, no date).

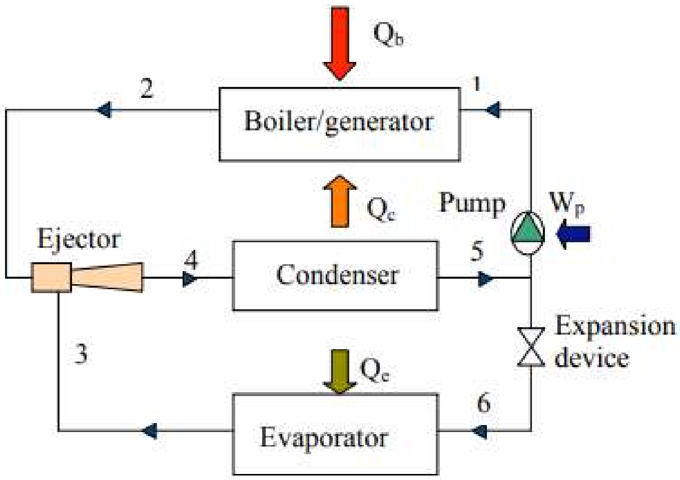

Figure 9: Schematic of a basic ejector based refrigeration cycle (Grimsby Institute, no date)

2.4.2 Technological development

The simple structure and low maintenance requirements makes the ejector refrigeration systems appealing for certain applications (Zeyghami, Goswami and Stefanakos, 2015). The COP, however, for ejector cooling systems is low (0.2-0.3) compared to other thermally driven technologies. In addition, the COP for the basic cycle drops significantly at operation away from the design point (Grimsby Institute, no date). These issues are being addressed with a series of modifications to enhance the performance.

The use of multi-stage ejector refrigeration systems is employed to maintain the performance over a range of operating conditions. It works by placing multiple ejectors in parallel before the condenser. A solenoid valve on top of each ejector can open and close the ejector depending on the needed operational capacity (Besagni, Mereu and Inzoli, 2015). Multistage ejectors are commercially employed today in CO_2 based VCRSs operating with trans-critical cycles (Gullo *et al.*, 2019).

Ejectors are also applicable to be combined with other refrigeration technologies to form hybrid systems for example together with mechanical compression, or absorption.

Compression-ejection refrigeration systems can be divided into two sub-categories. The first employs a compressor to enhance the performance of thermally driven ejector refrigeration systems, while the second employ an ejector to enhance the performance of traditional mechanical refrigeration systems. The first sub-category is most relevant for this study, and is shown to the left in Figure 10. The booster-compressor is installed downstream the evaporator compressing the suction flow before entering the ejector. The COP compared to a standard ejector refrigeration cycles is more than doubled, however it also introduces a significant power consumption compared to just using the liquid pump (Besagni, Mereu and Inzoli, 2015).

The combined ejector-absorption refrigeration system, shown to the right in Figure 10, is also a form of hybrid refrigeration system, where addition of the ejector enhances the system efficiency. The high-pressure absorbent solution entrains the refrigerant from the evaporator in the ejector. The modification is meant to increase performance by increasing the refrigerant flow from the evaporator.

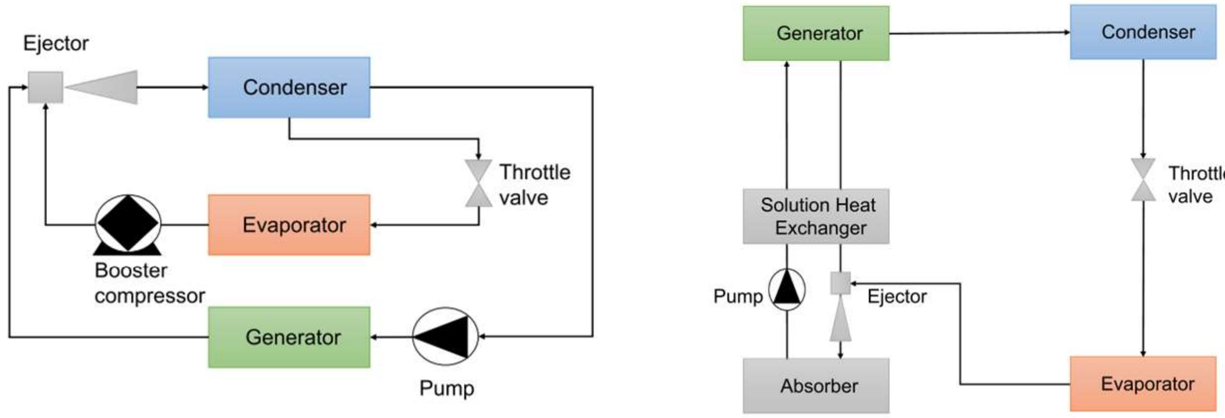

Figure 10: Hybrid ejector-refrigeration systems, to the left combined compression-ejection refriger, to the right ejector-absorption (Besagni, Mereu and Inzoli, 2015)

Regarding the use of refrigerants, water is the most widely used working fluid for commercial ejector refrigeration systems, however the typical usage is for cooling applications above 0°C, as well requiring a relatively high generator temperature (Chunnanond and Aphornratana, 2004). There is ongoing research for utilizing other working fluids such as halocarbons (CFCs and HCFCs), but also more climate friendly refrigerants such as hydrocarbons, CO_2 and ammonia. This can extend the cooling ability to below 0°C and reduce the generator temperature requirements. One reported experimental setup used methanol in a simple ejector refrigeration system and achieved evaporation temperatures of -2°C, with generator temperatures of 80-100°C, and achieved a COP of 0.2-0.4 (Besagni, Mereu and Inzoli, 2015).

2.4.3 Commercial status

A number of companies deliver thermally driven steam ejector systems using water as refrigerant for cooling applications above 0°C, with capacities up to 60 MW (Grimsby Institute, no date). When it comes to hybrid technologies, ejector-boosted mechanical vapor compression refrigeration cycles are commercially adopted and can provide cooling below 0°C, however these have electrical consumptions in the same range as traditional VCRSs.

For thermally driven ejector refrigeration cycles producing cooling below 0°C, there are few research initiatives in the available literature. However, DemacLenko (Austria) is a company that produces a snow making machine based on vacuum ice technology driven partially by heat. The system employs a thermally driven ejector, which can use heat from renewable sources such as solar or biomass and is described more detailed in the report by (Trædal, 2017). Calculations show that the system achieves a combined COP of 1.04 with the exergy from the heat taken into account, when producing snow at ambient conditions of 10°C. The system does use a vacuum pump to create the necessary pressure to form the ice slurry in the evaporator. Nevertheless, compared to the other snow production systems reviewed by Trædal (2017), the specific electrical power consumption per m^3 of snow produced was only between 11 - 31% of the consumption of the other systems. During prototype testing of the of the system thermal COP values ranging between 0.1-0.4 was achieved with evaporation temperatures close to 0°C, heat supply temperatures of 170-180°C and a cooling water temperature of 10°C (Joemann *et al.*, 2017).

2.5 Organic Rankine cycle-based refrigeration systems

Another form of thermo-mechanical cooling system is the Rankine based refrigeration system. Lately, this type of cooling system has received more attention and research lately due to recent advancements in the Organic Rankine Cycle (ORC) equipment and the introduction of environmentally friendly refrigerants (Zeyghami, Goswami and Stefanakos, 2015). An advantage of ORC-systems is that they can utilize heat sources with a large temperature range. Commercial systems exist today for heat sources in the range of at least 55 – 530 °C (Røssland, 2016).

2.5.1 Working principle

The principle behind the ORC-based cooling system is to utilize heat to produce mechanical work which drives a conventional mechanical vapor compression refrigeration cycle. Two common design configurations for this type of system is shown in Figure 11. The configuration to the left uses separate power and cooling cycles, where the ORC expander is mechanically connected to the compressor of the refrigeration cycle. The configuration to the right is an integrated cycle which uses a combined expander-compressor and a joint condenser. The integrated design is simpler and uses the same condenser and working fluid for both cycles, but this also reduces the system flexibility (Zeyghami, Goswami and Stefanakos, 2015).

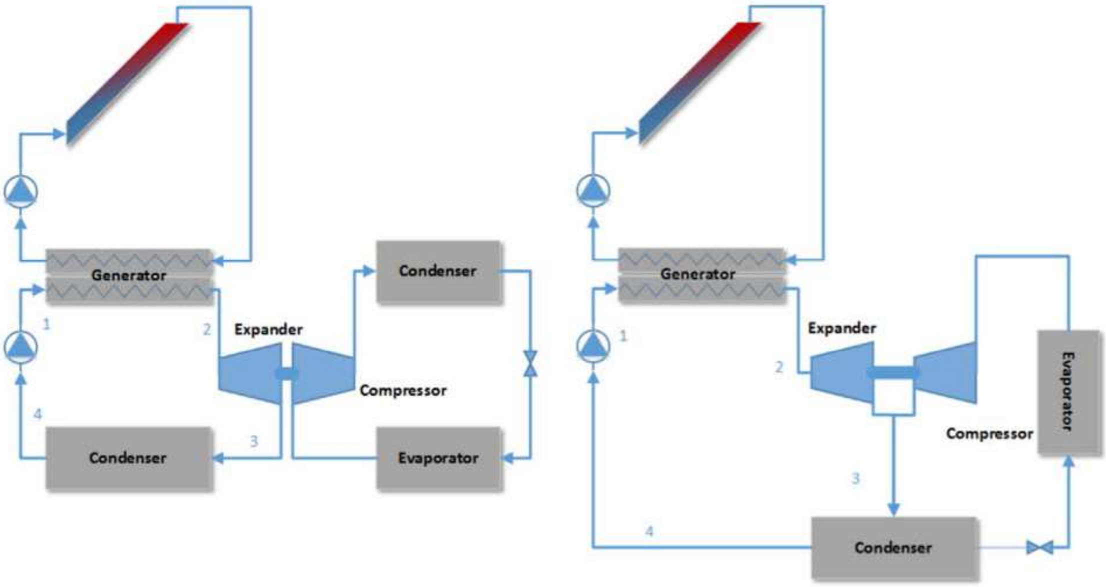

Figure 11: Schematics of two common design configurations for solar powered Rankine cooling system (Zeyghami, Goswami and Stefanakos, 2015)

2.5.2 Technological development

Apart from ORC-VCR systems, the coupling of ORC systems with other technologies such as absorption/ adsorption technologies for cooling purposes are being researched. The application area for these systems are often co-generation where cooling and, or heating are required in combination with electrical power. These systems are based on the cascade utilization of heat where condenser heat from the ORC can be used to power the cooling cycle as shown in Figure 12. Since the heat from the condenser used to drive the chiller module is at a lower temperature than the heat input to the ORC, this will necessarily require a higher heat source temperature overall compared to if the heat source was used to drive the chiller module directly.

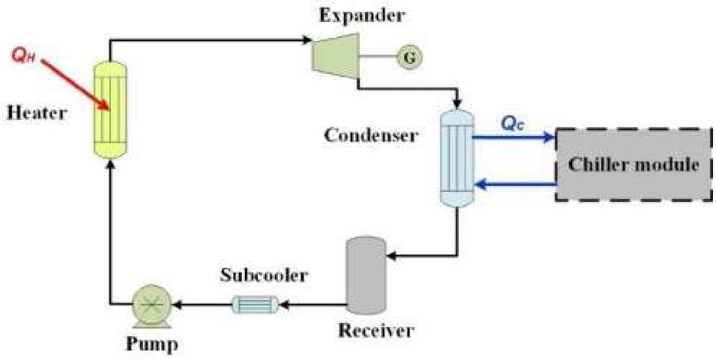

Figure 12: Schematic of an ORC module coupled with a chiller module (Roumpedakis, 2018)

As with other refrigeration technologies the selection of working fluids is one of the main design parameters. There has been performed some studies for cooling below 0°C. From a review it was found that n-butane (R600) was the best performing suitable working fluid when not considering working fluids soon to be phased out. Sub-zero cooling could be achieved with generator temperatures between 60-90 °C with COP values between 0.05-0.75 (Zeyghami, Goswami and Stefanakos, 2015).

2.5.3 Commercial status

The ORC technology has been extensively developed and commercialized over the years, and a large number of manufacturers now supply ORC technology based on a variation of heat sources such as solar, geothermal energy, and waste heat, with ORMAT as the by far largest supplier. The main application area is currently power production or combined heat and power (CHP) (Roumpedakis, 2018), (Tartière and Astolfi, 2017), and the capacities range from small size kW to MW scale plants. There is an emerging market for trigeneration systems also delivering cooling. For example, the second largest supplier Turboden (Italy), have delivered several plants for combined cooling, heating and power (CCHP), using hot water produced by the ORC units to drive an absorption chiller for cold production with the main application area being building cooling. One of the plants is installed in Heathrow airport using waste clean wood to produce power (1.8MW) and space heating/cooling (7.8MW) (*globalbioenergy.org*, 2020). The focus area for these systems, however, seems to be on space cooling with temperatures above 0 °C. There is a lack of commercial development of ORC systems targeting production of cooling below 0 °C.

2.6 Summary of thermally driven refrigeration technologies

Figure 13 classifies the refrigeration technologies that have been reviewed. There exist other technologies which have not been included in this report, as they are not applicable for cooling below 0°C, e.g. desiccant cooling, or only used for small-scale cooling applications (Stirling cycle).

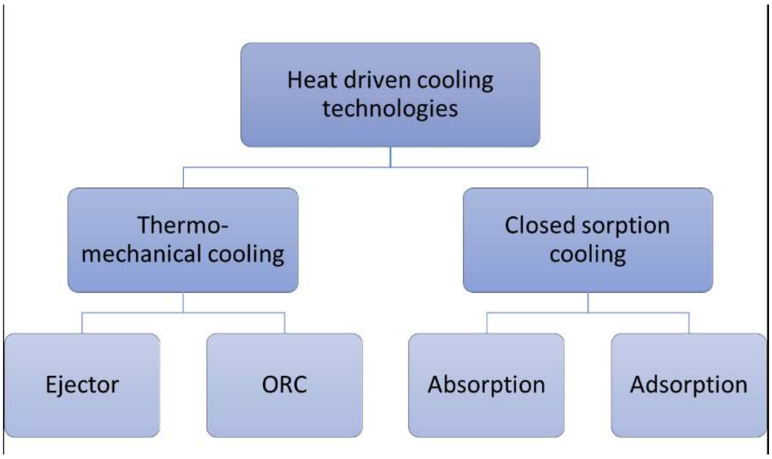

Figure 13: Classification of the reviewed heat driven cooling technologies applicable for <0°C cooling

The technologies are summarized and compared in Table 1. Note that the parameter values used in this comparison is based on systems providing cooling below 0°C. Absorption is the only technology that has widespread commercial use. It achieves relatively good COP values and there is wide range in possible waste heat sources and temperatures. Adsorption-based systems could be promising as they have the possibility of utilizing even lower heat supply temperatures, but so far this technology remains at a research level. Ejector-based cooling systems have seen widespread commercial use, but there seems to be a low attention to cooling applications below 0°C. However, there is one known commercialized unit from DemacLenko utilizing this technology for snow production. ORC-based cooling systems are commercialized for applications above 0°C, but there is limited research for sub-zero temperature refrigeration applications. An advantage for ORC-based systems is that they can utilize heat sources with very high temperatures.

Table 1: Summary and comparison of heat driven cooling technologies

Technology	COP	Heat source	Refrigeration output	Commercial status
Absorption	0.4-0.6	Hot water, steam, exhaust or direct fired (90-200°C)	Ice, brine or glycol-water (-60 - 0°C)	Several existing suppliers offering MW scale units
Adsorption	0.1-0.4	Hot water, steam, exhaust (80-120°C)	Ice, chilled water, glycol-water (-15 - 0°C)	A few manufacturers offering MW scale units for cooling. Refrigeration below 0°C only at experimental level.
Ejector	0.1-0.4	Hot water, steam (80-180°C)	Ice slurry, chilled water (0.01°C)	Several suppliers offering MW scale units for cooling above 0°C. DemacLenko is the only known supplier offering a thermally driven snowmaking machine based on ejector-assisted vacuum pump ice technology.
ORC	0.05-0.75	Hot water, steam, exhaust, exhaust gases (55-530°C)	-	A few suppliers offering MW scale units for CCHP purposes. Refrigeration below 0°C only at an experimental level.

3 Snow facility production requirements

In order to estimate the required heat supply for a thermally driven refrigeration system for snow production it is necessary to know the snow demand from a Temperature Independent Snow (TIS) production system. Previous work in the "Snow for the Future" project has resulted in a simulation tool where it is possible to calculate the annual snow production volume from a TIS production system (Trædal, 2018b).

A number of inputs were used to develop the model, amongst other climate data containing information about temperature, humidity, precipitation levels and freshly fallen snow. Another important input is information about the winter sports facility itself. Here, information about the snow-covered area, required snow depth, snow storage capabilities, Temperature Dependent Snow (TDS) production capacities and duration of the season are amongst the input parameters.

3.1 Case Description

To gather climatic and facility specific information for each individual winter sports facility that has been mapped in this report is a large task and outside the scope of this work. The report will instead rely on the results produced earlier in the "Snow for the future" project. Trædal (2018a) and Tveten (2020) used the developed simulation tool to estimate the TIS production requirements for the Granåsen arena, which is one of the main facilities for winter sports in Norway and holds competitions on international, national and regional levels.

In the case descriptions for Granåsen, Case 3 by Trædal (2018a) is of particular interest as it takes into account a TIS machine complementing TDS production. Tveten (2020) further developed the snow model and presented two more cases (case 2 and 3) where TIS production was included. All three cases assume TDS production and snow storage, which are methods already implemented for Granåsen to reduce dependence on natural snow fall. The calculations were performed for several seasons, from the 2007/2008 season to the 2015/2016 season. The cases by Tveten also include the overall TIS snow production (m^3) in the results, which can be used to calculate the cooling demand for a thermally driven refrigeration system.

An important result from these cases was the average annual season days, which is based on sufficient snow cover for at least a 3 km ski run + 12 000 m^2 stadium area (phase I), a total of 10700 m^3. Included in the season days is a phase II where an additional 5.8 km of ski runs and the ski jumping facility is covered with snow. Case 2 by Tveten achieved the highest number of season days, on average 6 more per season than case 3. However, the average seasonal TIS production volume was 2.4 times higher for case 2 than case 3. The difference between the cases were the TIS production capacity (600 for case 2 and 300 for case 3). In addition, case 2 assumed continuous snow production throughout the season, while for case 3 TIS production dependent on the amount of snow in the runs or snow storage level. From a cost-benefit perspective case 3 appear to be the best case.

In addition to TIS production for cross-country skiing purposes it is of interest to investigate TIS production for alpine skiing. An estimation of the needed snow volume for an alpine slope is based on a 1000 meter long slope, 40 meters wide and with a snow depth of 0.7m (*snøkompetanse.no*, 2020a). This results in a total snow volume of 28 000 m^3, almost three times the size of the phase I snow volume in Granåsen. Extrapolating from the results in case 3 (14 415*28000/10700), gives an annual TIS snow production of 37721 m^3.

The cases for Granåsen (Tveten Case 3) and the alpine slope, will be used as base cases and are summarized in Table 2.

Table 2: Case Descriptions for TIS production in the Granåsen Winter Sports Facility. The Granåsen case is the result from case 3 by Tveten, while the alpine slope case is an extrapolation from the case 2 results.

Case	TDS and snow storage	TIS production capacity (m^3/day)	Continuous TIS	Average annual TIS snow production [m^3]	Average annual season days
Granåsen	Yes	300	No	14415	128
Alpine slope	Yes	300	No	37721	-

3.2 Sensitivity analysis

In addition to the base cases, a sensitivity analysis will be conducted to show a range in heat requirement based on important sensitivity variables, which are given in Table 3. The base value for the heat rejection temperature is set to 10°C. Regarding the sub cooling temperature, the base value is set to -10°C, as there will be a temperature difference in the heat exchange between the refrigeration cycle and the ice production machine.

Table 3: Sensitivity variables used in heat requirement calculations

Sensitivity variable	Variation	Base value	Motivation
Heat rejection temperature and feed water temperature [°C]	5 - 15	10	Depends on ambient wet bulb temperatures and affects the COP
Ice, sub cooling temperature [°C]	-5 – (-20)	-10	Commercial TIS machines produce snow at different temperatures, depending on technology.
Annual TIS production [m^3]	10000 - 50000	See Table 2	Depends on facility size, area climate and weather conditions
TIS daily production capacity [m^3]	300-600	See Table 2	Affects the required heat effect [MW]

3.3 TIS Production Energy Requirement

Based on the snow production model, the TIS production potential is calculated as follows (Trædal, 2018b):

TIS production potential = $\dot{V}_{TIS} * \ C_{TIS}$

$\dot{V}_{TIS}$: TIS production capacity [m^3/24 hrs]
C_{TIS}: Compression factor for the temperature independent produced snow

The snow density for groomed snow is 590 kg/m^3, while the snow density for TIS snow, $\rho_{snow,TIS}$, is 450 kg/m^3. This gives a compression factor C_{TIS} = 0.76.

The cooling effect from the refrigeration cycle Q_c equals the heat removed from water to produce snow:

$$\dot{Q}_{ice} = \frac{\dot{m}_{ice} * (C_{p,water} * \left(T_{feed} - T_{freezing}\right) + \ h_{if} + C_{p,ice} * \left(T_{freezing} - T_{ice,sub\ cooled}\right))}{3600 * 24}$$

$C_{p,water}$: Specific heat capacity for water, 4.22 kJ/kgK
$C_{p,ice}$: Specific heat capacity for ice, 2.12 kJ/kgK
T_{feed}: Feed water temperature, base case 10°C
$T_{freezing}$: Freezing temperature of water, 0°C
$T_{ice,sub\ cooled}$: Temperature of sub cooled produced ice, base case -10°C
h_{if}: Heat of fusion for water, 333.5 kJ/kg
$\dot{m}_{ice}$: mass of produced ice [kg]

$$\dot{m}_{ice} = \dot{V}_{TIS} * \rho_{snow,TIS}$$

The required heat to produce the cooling effect becomes:

$$\dot{Q}_H = \frac{\dot{Q}_{ice}}{COP}$$

COP: Coefficient of performance for the refrigeration system.

As shown in section 2.1 the COP will vary depending on the temperature conditions. As an example, one of the manufacturers (Ago AG) displays the optimum heat ratio (COP) for various working conditions as shown in Figure 14. For refrigeration temperatures between -5 and -20 °C, The COP varies between 0.55-0.65 approximately. Using the heat supply temperature and cooling water temperature to calculate the Carnot COP in the same way as in section 2.1, results in Carnot efficiencies of 0.2-0.3. As a conservative measure a Carnot efficiency of 0.2 will be used in the calculation of the heat requirements. Furthermore, the COP value in commercial and experimental absorption systems providing cooling below 0°C, typically tops out at 0.7. As an additional conservative measure, a COP cap of 0.7 will be used in cases where a Carnot efficiency of 0.2 results in COP > 0.7.

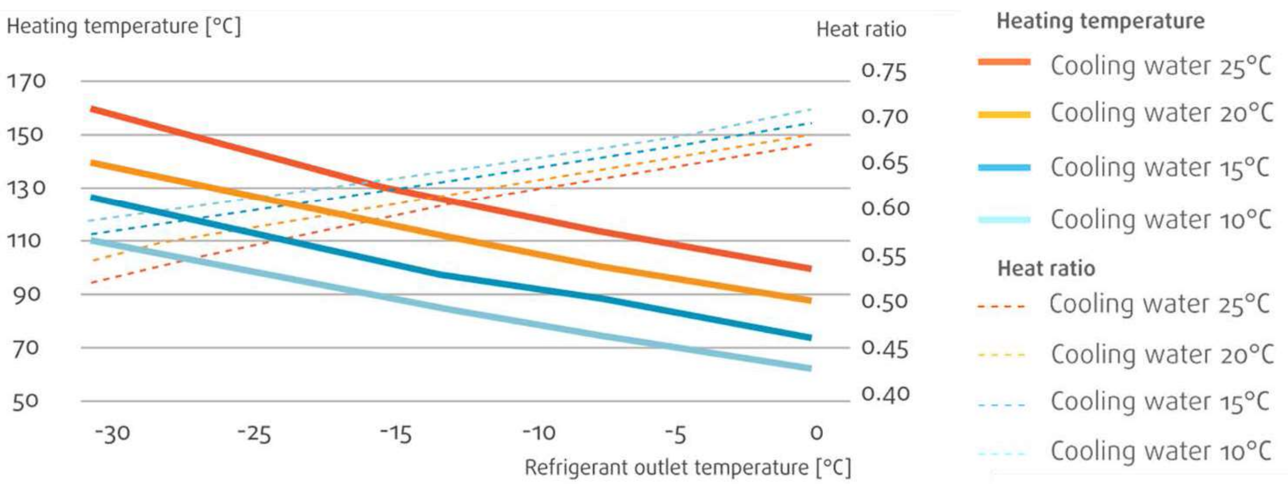

Figure 14: Expected heat ratio (COP) when varying against refrigerant outlet temperatures, cooling water temperatures and heating temperatures (*ago.ag*, 2020)

3.4 Heat requirement results

3.4.1 Base cases

The results for the base cases are summarized in Table 4. The temperature conditions for the base cases resulted in a COP of 0.61 for the refrigeration system. For the calculation of this value it has been assumed that the heat is supplied at 95°C, which is normally the highest operating temperature in a district heating network (*statkraft.no*, 2020).

The resulting heat supply per volume of groomed snow produced for the bases cases is 104 kWh/m^3. In addition to consumption of heat, some electricity consumption is needed to run the refrigeration and the ice production process. However, this is not included in the results.

As a comparison, the specific power consumption for the electrically driven TIS production in case 3 was 50 kWh/m^3 (Tveten, 2020), and for traditional TDS production 0.6 – 6 kWh/m^3 (*snøkompetanse.no*, 2020b).

Table 4: Heat requirement results for the bases cases

Case	Annual heat supply	Max heating effect	Annual cooling output	Max cooling effect
Granåsen	1.50 GWh	1.34 MW	938 MWh	0.81 MW
Alpine slope	3.93 GWh	1.34 MW	2454 MWh	0.81 MW

3.4.2 Sensitivity analysis

Figure 15 shows the heat requirements for TIS production for the Granåsen and alpine hill cases, when varying against different feed water temperatures and evaporation temperatures. The COP values for each result is listed in the second row of the x-axis. The changes in the energy requirement is greatly affected by the change in COP that occur when feed water and evaporation temperatures are changed, ranging from 1.2/3.2 GWh (Granåsen/Alpine hill) in the best case to 3.1/8.0 GWh in the worst case.

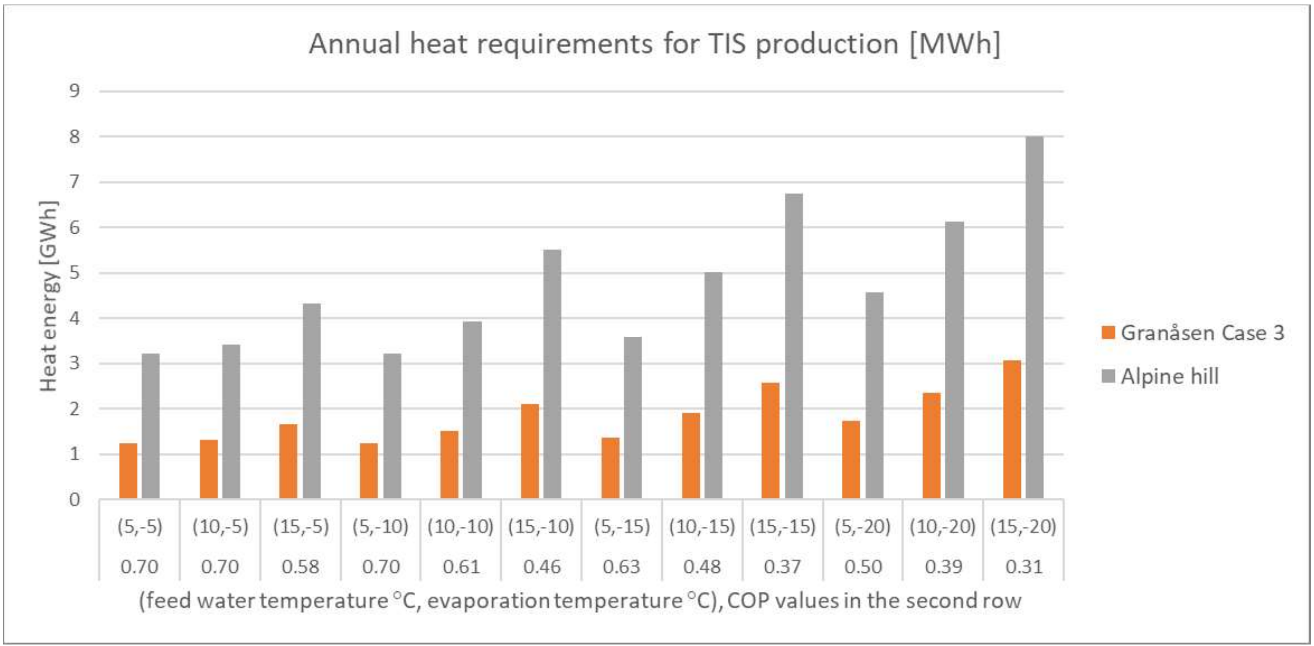

Figure 15: Sensitivity against feed water and evaporation temperature for the Granåsen and alpine hill cases

Figure 16 shows the heat effect requirements for TIS production ranging from 300 to 600 m^3 per day. The marked area in the chart mark the sensitivity range with regards to feed water and evaporation temperatures. As with the energy requirements the expected heat effect for the same production volume more than doubles when going from the best case to the worst case.

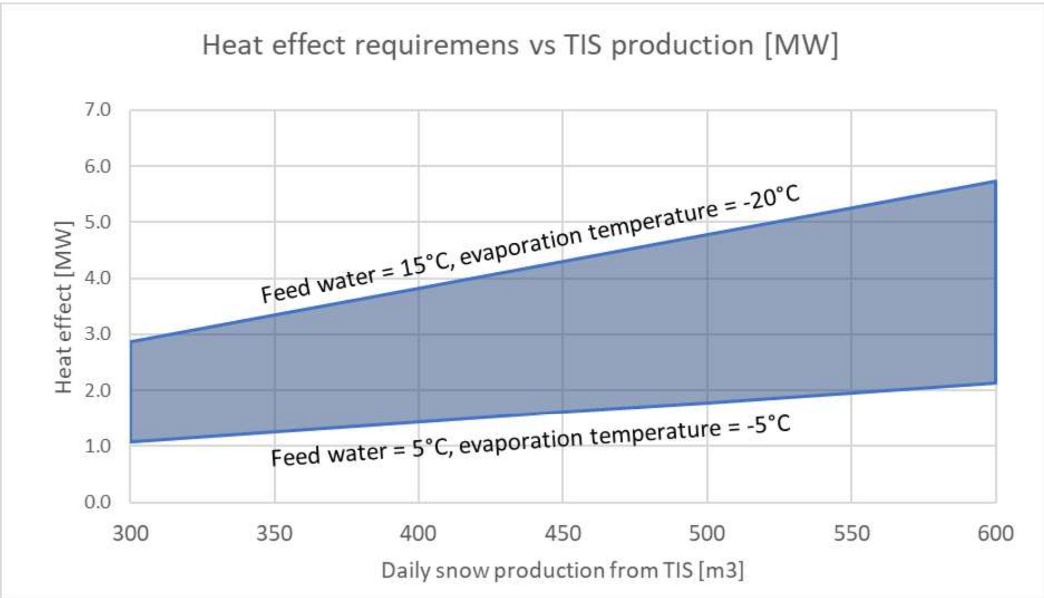

Figure 16: Heat effect requirements vs TIS production

4 Available heat sources

This chapter discusses and reviews the potential heat sources suitable for providing thermal energy for a TIS production system. The review builds on two main sources of information: In 2009 a comprehensive review on potential industrial waste heat was conducted by Norsk energi and NEPAS for Enova and Norsk Industri (Sollesnes and Helgerud, 2009). And in 2020 another comprehensive review was conducted by Oslo Economics and Asplan Viak for the Norwegian Water Resources and Energy Directorate in 2020, where the potential for further optimization of heating and cooling was reviewed (Oslo Economics / Asplan Viak, 2020).

4.1 Heat source categories

There are five main energy sources offered for heating in Norway (Oslo Economics / Asplan Viak, 2020):

- Electricity
- District heating
- Local heating systems
- Wood
- Heat pumps

Three of these are thermal; district heating, local heating systems and wood, where the latter is mostly used for household heating and is not relevant.

4.2 District heating

District heating provided 8 % (6 TWh) of the heating energy supplied to households, industry and service industry in 2018. There are more than 100 District heating companies (*fjernvarme.no*, 2020) and district heating is available in most larger cites, and the highest district heating capacities are found in the largest cities. The energy sources used in district heating consists mostly of recovered waste heat from combustion of residual waste, bioenergy and electricity (Oslo Economics / Asplan Viak, 2020). The distribution range from a district heating central can stretch over many kilometres, often with help from smaller sub-centrals to maintain the supply temperature.

The district heating supply temperatures vary between the different suppliers. The temperature is also seasonally dependent, with higher temperatures in the winter than in the summer. The largest facilities are often dimensioned for temperatures up to 120°C. Statkraft Varme states that their systems normally deliver supply temperatures up to 95°C in the winter and down to 70°C during the summer (*statkraft.no*, 2020). Based on the technology review in chapter 2, a summer supply temperature of 70°C is on the borderline for how low the supply temperature to a thermally driven TIS production system can be.

Figure 17 Shows the municipalities where district heating is currently provided. It is currently most prevalent in the south-eastern parts of Norway and other major cities. Annual production is highest in municipalities marked dark green.

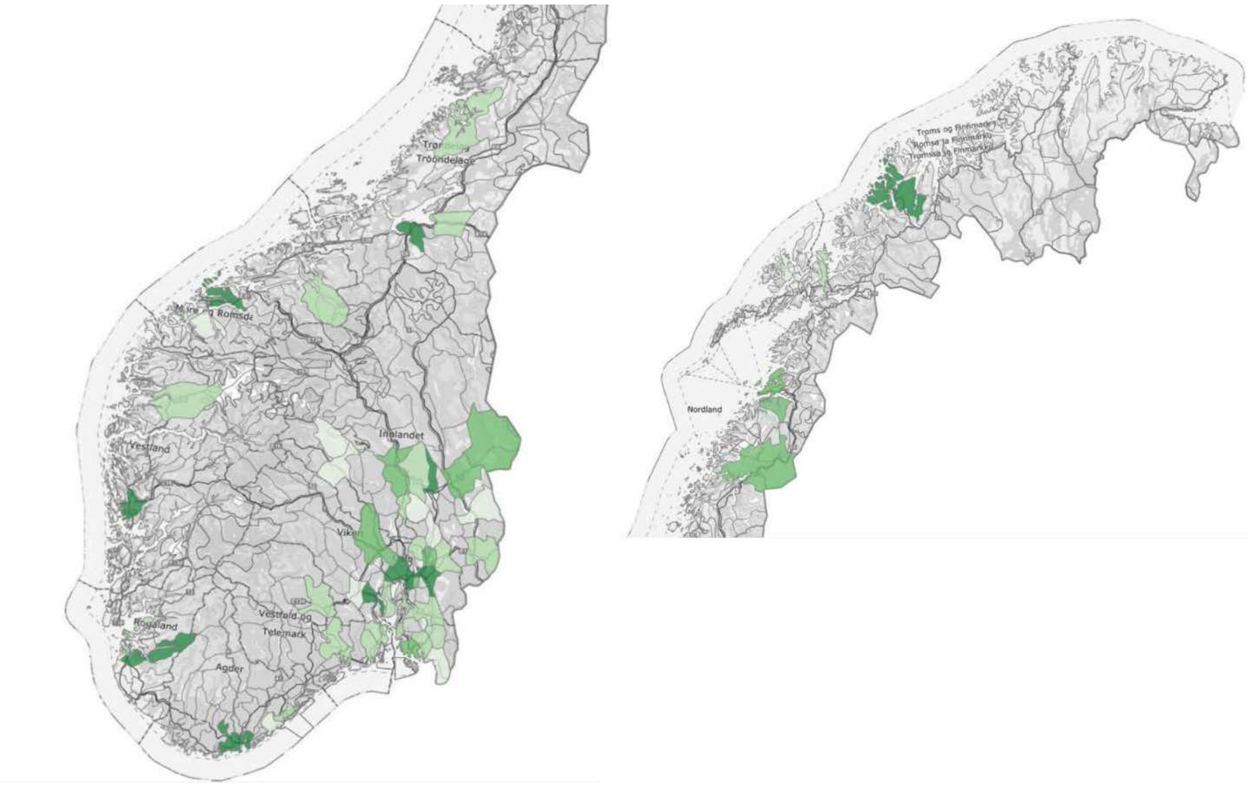

Figure 17: Municipalities where district heating is provided (2018), source: (*energi.avinet.no*, 2020)

4.3 Local heating facility

A local heating facility consists of a heating unit providing heat to buildings, mainly for industrial purposes. The energy produced from local heating facilities was 11 TWh in 2018, thus contributing to a larger share of the total heat supply than district heating. Energy sources are often non-renewable, such as oil or gas fired boilers (Oslo Economics / Asplan Viak, 2020). In most cases it will probably not be a feasible option to utilize heat from a local heating facility for other purposes, since the heat supply is dimensioned and purpose built for its original consumer, with little scalability. This heat source option is not further reviewed here.

4.4 Waste heat

An additional potential heat source apart from the energy sources listed in section 4.1, is waste heat. Waste heat can be described as unutilized heat energy in the form of air, water, steam or exhaust with higher temperatures than the surrounding (Oslo Economics / Asplan Viak, 2020).

4.4.1 Waste heat sources

The primary sources of waste heat are from the industry and from the combustion of waste. Data centres are a third source, which is expected to grow in size in the future. However, the temperature levels are too low to be utilized (35-45 °C) for the purpose of TIS production (Oslo Economics / Asplan Viak, 2020).

An overview of the available waste heat sources distributed per industry sector is shown in Figure 18, and is based on the review from Norsk Energi and NEPAS in 2009. Heat from combustion of waste is included in the overview. The waste heat potential constitutes a total of 19 198 GWh/year (2009) (Sollesnes and Helgerud, 2009), which is more than three times the amount of district heating supplied in 2018.

Potential heat from waste combustion constitutes a small fraction of the waste heat potential. Most waste combustion facilities already utilize the waste heat for district heating. Some facilities also re-use the energy from combustion in the form of new materials. Still a total of 1 TWh (2018) is not utilized due to low demand during the summer season (Oslo Economics / Asplan Viak, 2020).

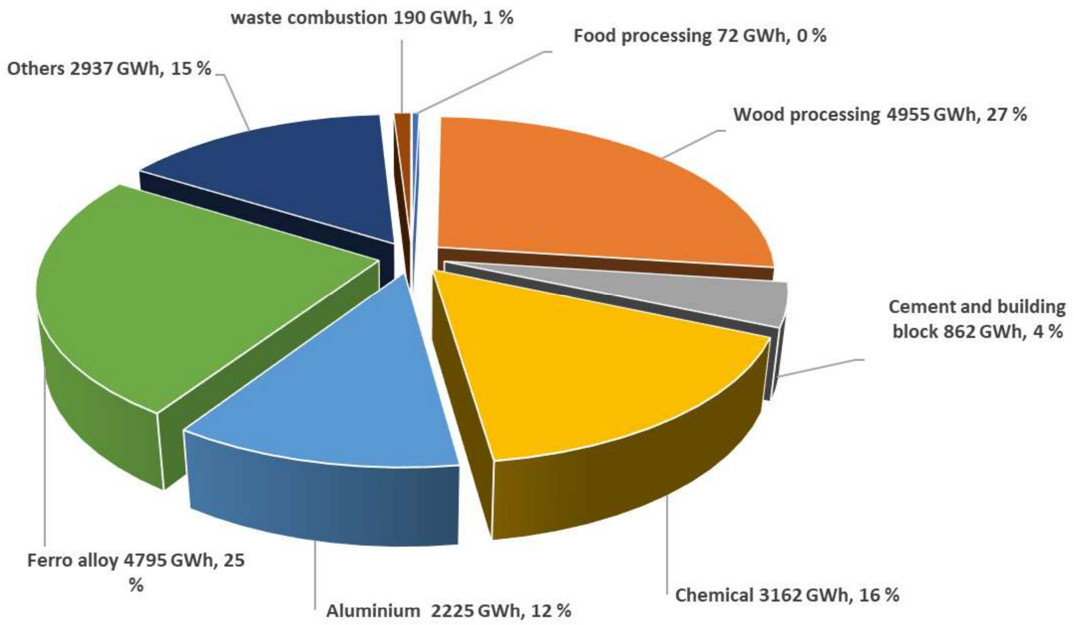

Figure 18**:** Sector distribution of waste heat sources. Reproduced from (Sollesnes and Helgerud, 2009)

Figure 19 shows the temperature distribution of waste heat for the different sectors. Waste heat sources with the temperature ranges 60-140°C and >140°C are suitable for TIS production. All sectors have potential waste heat with suitable temperature levels, however the Ferro alloy, Cement and building block, aluminium, waste combustion and "others" have a relatively higher share of the waste heat at high temperatures.

The energy carriers of waste heat for the different sectors is given in Figure 20. As shown in the technology summary in section 2.6, thermally driven refrigeration technologies are in principle capable of utilizing all the major carriers in the figure: exhaust, water, and steam. However, in the ferro alloy, aluminium, cement and building block, wood processing and "others" sectors a portion of the waste heat is contaminated with particles and dust, which may lead to technological challenges and increased equipment costs for waste heat recovery. In some cases, it will not be technically or economically feasible to utilize the waste heat.

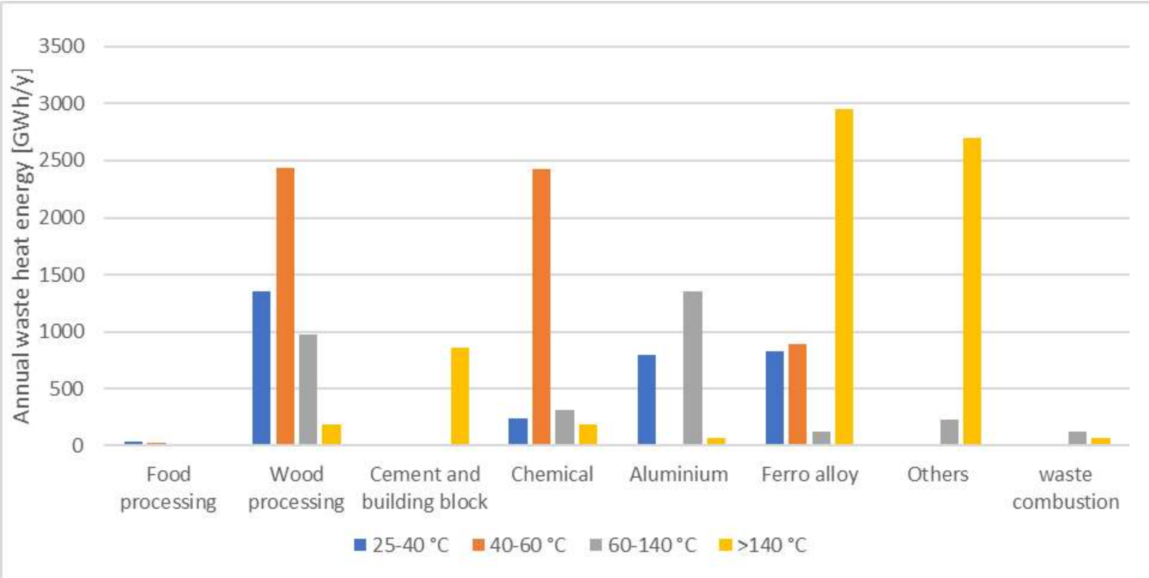

Figure 19: Temperature distribution of waste heat for the different sectors. produced with data from (Sollesnes and Helgerud, 2009)

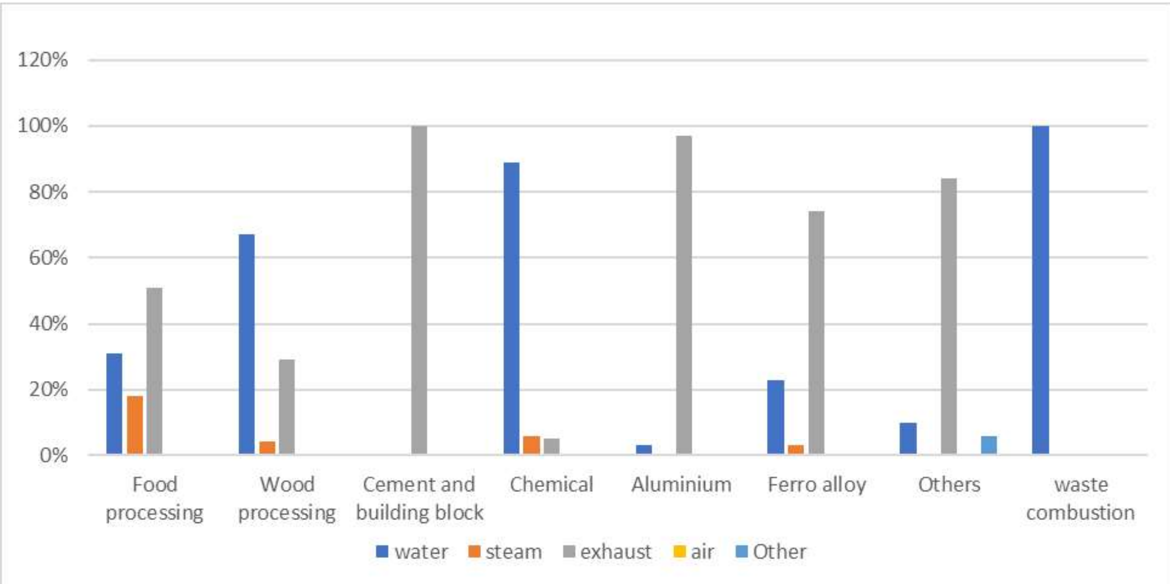

Figure 20: Waste heat energy carrier for the different sectors. Produced with data from (Sollesnes and Helgerud, 2009)

4.4.2 Industrial waste heat recovery projects

In the period 2002-2019, 192 projects with the purpose of recovering industrial waste heat have been realized with support from Enova, resulting in 3.23 TWh recovered energy (Oslo Economics / Asplan Viak, 2020). However, the purposes have typically been for internal re-use or power production, and in 2019 only 212 GWh (3%) of the total district heating supplied came from industrial waste heat sources (*fjernvarme.no*, 2020)

In many industrial locations the utilization of waste heat for district heating purposes is challenging because the local market for heat is too small to make the project viable. Other barriers are missing district heating infrastructure or lack of awareness among the industrial actors. However, an increased number of projects realizing waste heat recovery for district heating in the later years may indicate that these barriers are about to be broken (Oslo Economics / Asplan Viak, 2020).

4.4.3 Geographical locations of waste heat sources

Figure 21 shows the location of some of the potential waste heat sources from the industry, which has been mapped by (Oslo Economics / Asplan Viak, 2020). Red dots are locations of industrial sources of waste heat, while blue dots are locations of waste combustion facilities with potential waste heat. The sector distribution on the map is as follows:

- Chemical: 37
- Metal (Ferro ally and aluminium): 28
- Food processing: 13
- Wood processing: 11
- Cement: 2
- Refineries: 2
- Land based petroleum processing: 6

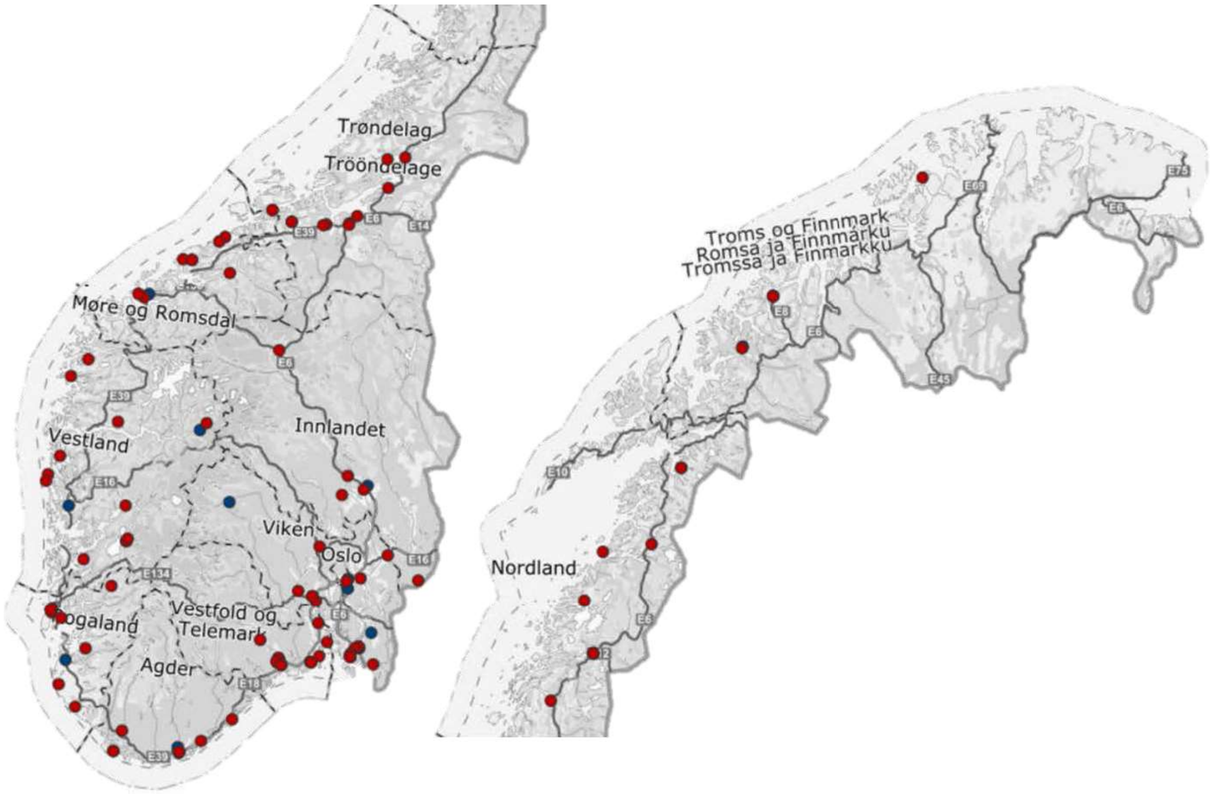

Figure 21: Locations of potential industrial waste heat sources, including waste combustion facilities, source: (*energi.avinet.no*, 2020)

5 Winter sports locations

In this chapter, winter sports arenas and their locations in Norway, along with potential nearby sources of heat are mapped in order to identify where there is a potential for heat driven TIS production.

5.1 Facility identification

Facilities and arenas within four main winter sports disciplines have been mapped:

- Cross country skiing
- Biathlon
- Alpine skiing
- Ski jumping

These sources were used to identify the winter sports locations:

- List of homologated arenas in Norway (*fis-ski.com*, 2020)
- Statistics from Skiinfo.no, a web site information about ski destinations, focused on alpine skiing (*skiinfo.no*, 2020)
- The Norwegian Biathlon Associations overview of biathlon arenas in Norway (*anlegg.skiskyting.no*, 2020)
- An overview of the 25 best alpine skiing facilities from the newspaper Dagens Næringsliv (*dn.no*, 2020)
- On overview of ski jumping hills in Norway on Wikipedia (*wikipedia.org*, 2020)

From these sources a total of 168 facilities were identified for further review.

A few remarks to the facility identification:

- Cross country skiing and biathlon are quite similar with regards to snow requirements, and many ski arenas are used for both disciplines. They have therefore been merged in the arena discipline identifier.
- All exercised disciplines have been marked in multidiscipline arenas, such as Holmenkollen
- Some facilities are primarily used for one discipline but may also facilitate other disciplines. For these only the primary discipline has been used as identifier. For example, many alpine skiing facilities have cross-country skiing tracks nearby for recreational use, but the facility has only been marked for alpine skiing.
- Storlien skisenter is located in Sweden, but it has been included in the list since it is located near the border to Meråker is a part of the Meråker-storlien cooperation (*meraker-storlien.com*, 2020)

A table with all facilities is given in Table A-1 in the appendix.

5.2 Facility potential

Since there are considerable costs related to the investment and operation of a thermally driven TIS production system, it is expected that at the time being, only a few winter sports facilities have the potential to invest in such a snow production system. Since 168 locations were identified, a few indicators were used to roughly screen and score the facility potential. Examples of positive indicators were:

- Large facility or arena size
- Proximity to populated areas
- Competitions and events on a high level

Based on these indicators, a ranking from 1-3 was given to all 168 facilities to indicate the potential, with 1 being the best score.

5.3 Screening results

Table 5 shows the overall results from the screening. 22 Facilities were given the best ranking value, while an almost equal number of facilities were ranked 2 and 3. Included in the results is the discipline count for each ranking value. Since there are many multi-disciplined facilities the sum of the count from each disciple does not equal the total number of facilities. Cross country / biathlon and alpine skiing are the disciplines with the most facilities. There are fewer facilities for ski jumping, however a higher share of these were given the highest ranking, such that the distribution between the disciplines within the facilities with the highest potential is quite even.

Table 5: Screening results of winter sports facilities and ranking

Ranking value	Total number of facilities	Cross country skiing / biathlon	Alpine skiing	Ski jumping
1	22	13	11	8
2	74	37	34	17
3	72	34	34	17

Figure 22 is a geographical overview of where the identified winter sports facilities are located. High ranked facilities are marked with a dark blue colour. Most of them are in south-eastern parts of Norway, in regions such as Innlandet, Viken and Oslo. The southern-eastern parts of Norway also have the highest number of identified facilities. This is reasonable due to the high population in these regions. In addition, all disciplines are well represented with many facilities in these regions. By comparison, there are few alpine skiing facilities in Trøndelag, and relatively few cross country / biathlon arenas in Vestland.

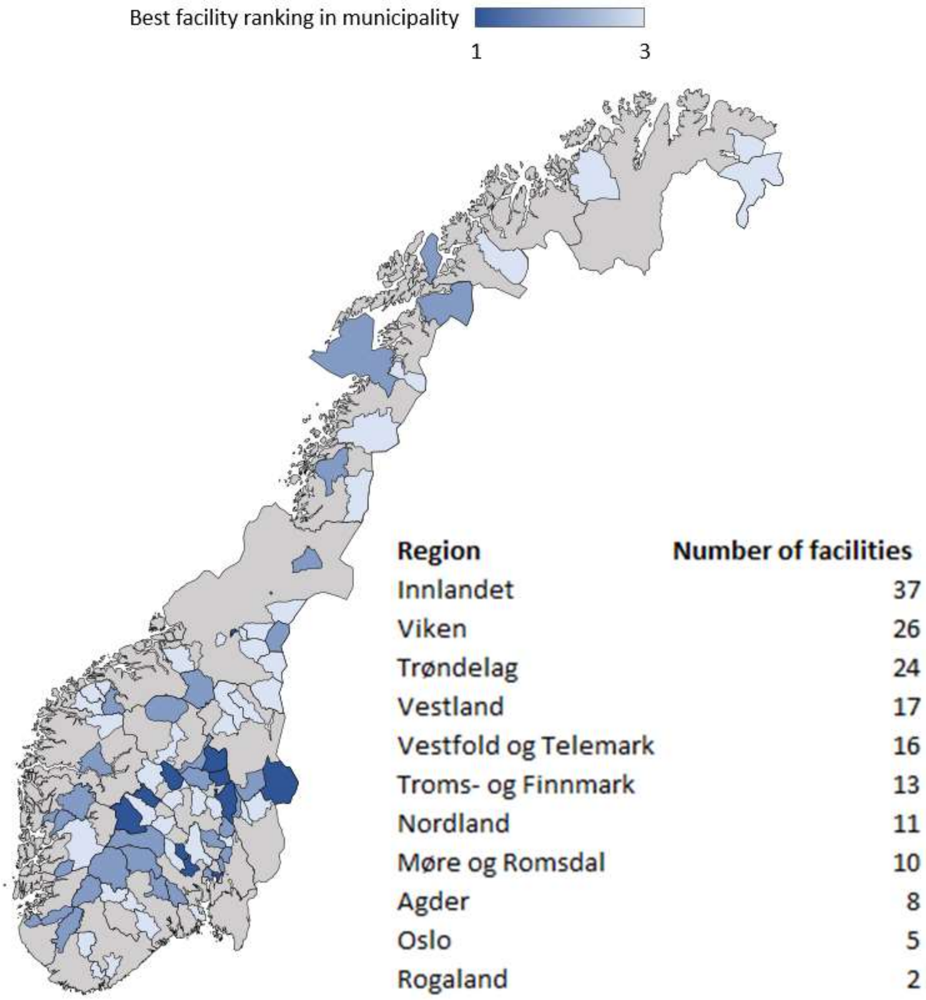

Figure 22: Graphical overview of winter sports facility location and ranking value

The 22 facilities identified with the highest potential (highest ranked) are listed in Table 6. Many of these are homologated for competition and typically hold large competitions such as world cups (WC), continental cups (COC), national championships (NM) or other major competitions.

Several of them are close to large, populated areas, which provides a potential for a high number of users. The alpine skiing destinations in the list are some of the largest in Norway, some with yearly revenues from lift pass sales alone surpassing 100 MNOK (*alpinanleggene.no*, 2020). Although many of these are in remote and relatively snow proof areas, TIS production can help to extend the season and increase revenue potential.

Some facilities were included because they are in the same municipality as another large venue and can benefit from the advantage of co-location.

Table 6: Overview of winter sports facilities with the highest rank

Facility name	Municipality	Region	Comments	CC skiing/ biathlon	Alpine skiing	Ski jumping	Competitions
Drammen	Drammen	Viken	Temporary venue used for cross country WC sprints	Yes			WC
Konnerud	Drammen	Viken		Yes		Yes	WC, NM
Drammen skisenter	Drammen	Viken	Co-location benefits		Yes		
Hemsedal skisenter	Hemsedal	Viken	53 runs, Norway's second largest alpine skiing destination	Yes	Yes		FIS, NC
Geilo	Hol	Viken	45 runs		Yes		FIS
Norefjell	Krødsherad	Viken	30 runs		Yes		FIS
Lysgårdsbakkene	Lillehammer	Innlandet				Yes	WC, COC, NM
Nye birkebeineren skistadion	Lillehammer	Innlandet		Yes		Yes	WC, NM
Vikersund hoppsenter	Modum	Viken	World's largest ski jumping hill	Yes		Yes	NM, WC
Modum skisenter	Modum	Viken	2 runs, co-location benefits with Vikersund		Yes		
Holmenkollen	Oslo	Oslo	Norway's national arena	Yes		Yes	WC, WC, NM
Linderudkollen	Oslo	Oslo		Yes		Yes	NM
Midtstubakken	Oslo	Oslo		Yes		Yes	NC, NM
Oslo vinterpark - Tryvann	Oslo	Oslo	High revenue		Yes		FIS
Kvitfjell	Ringebu	Innlandet	33 runs		Yes		WC
Sjusjøen Natrudstilen skisenter	Ringsaker	Innlandet	Lillehammer area (Co-location benefits)	Yes	Yes		
Granåsen	Trondheim	Trøndelag		Yes		Yes	NM, WC
Trysilfjellet arena	Trysil	Innlandet	Co-location benefits with Trysilfjellet skisenter	Yes			
Trysilfjellet skisenter	Trysil	Innlandet	Norway's largest alpine skiing destination		Yes		COC
Hafjell alpinsenter	Øyer	Innlandet	31 runs, includes cross-country arena on Mosetertoppen.	Yes	Yes		FIS, NJC
Beitostølen arena	Øystre Slidre	Innlandet		Yes			NC, FIS
Beitostølen skisenter	Øystre Slidre	Innlandet	Co-location benefits		Yes		

5.4 Co-location between winter sports facilities and heat sources

The next step in the screening is to identify which of the winter sports facilities are co-located in municipalities with available thermal heat.

Based on the review of available heat sources in chapter 4, the most promising heat sources are:

- district heating
- Industrial waste heat
- heat from waste combustion (not used for district heating)

Information about potential industrial waste heat and waste heat combustion (*energi.avinet.no*, 2020) (2019 values), and district heating (*Fjernkontrollen.no*, 2020) (2019 values) has been retrieved to identify in which municipalities the potential heat sources are located and compare those results against the location of the winter sports facilities.

It is important to note that while the list of municipalities where district heating is offered and waste combustion plants are located is complete, the list of potential industrial waste heat sources is not. There may therefore exist potential sources which are not mapped in this report.

5.4.1 Co-location considerations

The use of heat to produce snow limits the amount of applicable winter sports locations, since the distance between the heat source and the location where the snow is needed is restrained by the cost and technical limitation of transporting either heat or snow over large distances.

Since co-location between winter sport facilities and potential heat sources in this report is constructed within the municipal boundary and not by geographical distance, it is likely that some cases with a promising potential are lost as the heat source and recipient can be located in geographical proximity to one another, but in different municipalities. On the other hand, a case that seems promising, may not be if the source and recipient are located on the opposite sides in the same municipality with a large distance between them. More in depth investigations therefore need to be made to determine the exact co-location potential for each case.

There are two main methods to connect the heat source and winter sports facility:

- Heat is transported, snow is produced locally at the winter sports facility
- Snow is transported, and is produced at the location of the heat source

To enable local production the heat needs to be transferred through an insulated pipe e.g. district heating distribution network to the local snow production system. In the case of district heating, winter sport facilities located in urban areas with an existing district heating distribution network will have an advantage as the investment cost is limited to the branch line from the distribution and a substation located at site (Econ, 2010).

If the winter sport facility is located in an area outside existing distribution, additional investment costs related to the construction of the distribution network is expected. In the case of utilization of waste heat from an industrial plant, the infrastructure will most likely need to be built from the ground up and significant costs are expected. In such cases it is beneficial if the distance between the heat source and winter sport location is short, or if the investment costs can be shared between multiple heat recipients.

5.4.2 Results

In Table 7 possible heat sources in the same municipalities as the winter sports facilities with the highest potential are listed. The list shows that district heating is available in most municipalities. The annual heat supply is for most part well above the base case annual requirements (1.5 / 3.93 GWh) from section 3.4.1.

Table 7: Overview of the highest ranked winter sports locations and potential heat sources

Facility name	Municipality	Region	District heating potential: Supplier, annual heat supply (2019)	Industrial waste heat potential:	
				Industrial facility	sector
Drammen (by-løypa)	Drammen	Viken	Drammen fjernvarme, 114 GWh	Vajda-papir scandinavia as, Krystal as, Bim norway as, Solenis norway as avd. drammen	Wood processing, Chemical processing, Chemical processing, Chemical processing
Konnerud	Drammen	Viken	Drammen fjernvarme, 114 GWh	Vajda-papir scandinavia as, Krystal as, Bim norway as, Solenis norway as avd. drammen	Wood processing, Chemical processing, Chemical processing, Chemical processing
Drammen skisenter	Drammen	Viken	Drammen fjernvarme, 114 GWh	Vajda-papir scandinavia as, Krystal as, Bim norway as, Solenis norway as avd. drammen	Wood processing, Chemical processing, Chemical processing, Chemical processing
Hemsedal skisenter	Hemsedal	Viken			
Geilo	Hol	Viken			
Norefjell	Krødsherad	Viken			
Lysgårdsbakkene	Lillehammer	Innlandet	Eidsiva, 59 GWh		
Nye birkebeineren skistadion	Lillehammer	Innlandet	Eidsiva, 59 GWh		
Vikersund hoppsenter	Modum	Viken			
Modum skisenter	Modum	Viken			
Holmenkollen	Oslo	Oslo	Fortum, 1.8 TWh	Nordox	Chemical processing
Linderudkollen	Oslo	Oslo	Fortum, 1.8 TWh	Nordox	Chemical processing
Midtstubakken	Oslo	Oslo	Fortum, 1.8 TWh	Nordox	Chemical processing
Oslo vinterpark - Tryvann	Oslo	Oslo	Fortum, 1.8 TWh	Nordox	Chemical processing
Kvitfjell	Ringebu	Innlandet			
Sjusjøen Natrudstilen skisenter	Ringsaker	Innlandet	Eidsiva, 22 GWh	Strand unikorn as - (strand brænderi)	Chemical processing
Granåsen	Trondheim	Trøndelag	Statkraft, 663 GWh	Ranheim paper & board as	Wood processing
Trysilfjellet arena	Trysil	Innlandet	Eidsiva, 48 GWh		
Trysilfjellet skisenter	Trysil	Innlandet	Eidsiva, 48 GWh		
Hafjell alpinsenter	Øyer	Innlandet			
Beitostølen arena	Øystre Slidre	Innlandet	Stølslie Biovarme, 7 GWh		
Beitostølen skisenter	Øystre Slidre	Innlandet	Stølslie Biovarme, 7 GWh		

Figure 23 gives an overview of potential heat sources for all of the identified winter sports facilities. It is divided into pie-bar bar charts, one for each heat source. The pie shows how many facilities that are located in a municipality where the given heat source exists, while the bar shows the distribution between these facilities with respect to their ranking score.

A total of 68 identified winter sports locations are in municipalities which provide district heating. Hence, district heating is the most prevalent heat source with regards to co-location. Considering the annual heat supply there are large variations between the municipalities, ranging from less than 10 GWh to almost 2 TWh. The available heat reserves for new potential district heating recipients in the individual municipalities is not known, but it is likely to assume that there is higher availability in municipalities with already high annual supply.

There are 33 identified winter sports facilities located in municipalities where there are industrial waste heat sources. However, as discussed in 5.4, not all of the industrial waste heat sources have been mapped. Municipalities where these sources are missing are therefore marked "unknown" in the figure. Even though the heat supply potential from each source is not known, based on the findings in (Oslo Economics / Asplan Viak, 2020) and (Sollesnes and Helgerud, 2009), many companies have a waste heat potential well above the required quantity and at sufficient temperatures.

There is only 1 winter sports facility located in municipalities where there are waste combustion plants, which do not produce heat for district heating purposes. This is Narvik skisenter, and the waste combustion facility is located at Norcem's cement factory in Kjøpsvik. The heat from the waste combustion is utilized by the factory, and the remaining heat potential for other users is unknown.

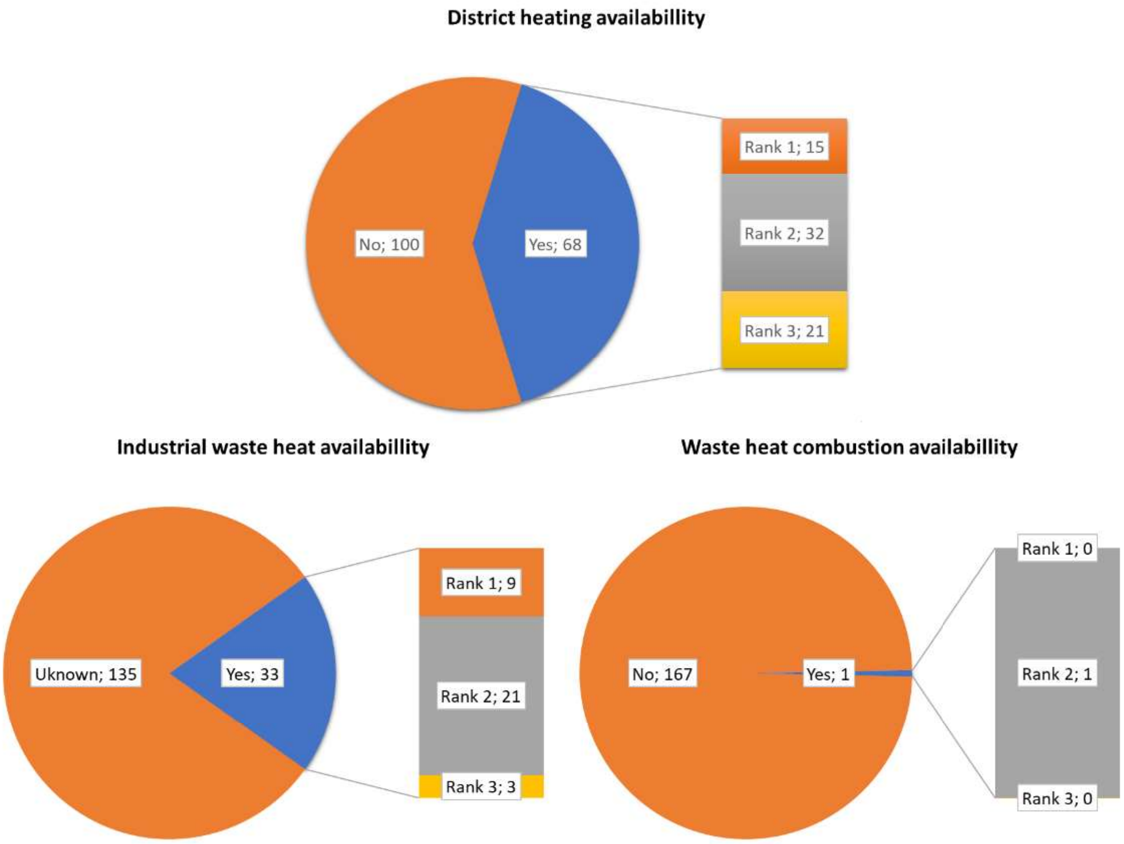

Figure 23: Municipal co-location between winter sports facilities and various waste heat sources. The pie shows the number of winter sports facilities, while the bar shows the ranking distribution.

6 Conclusion and further work

In this report the potential for utilizing heat driven refrigeration technologies for temperature independent snow production (TIS) has been investigated. The goal of this report has been to review potential thermally driven refrigeration technologies suitable for snow production and to map potential locations where heat, e.g. industrial waste heat, district heat or other types of heat sources are available and can be utilized for TIS to nearby winter sport locations.

The review of the thermally driven refrigeration technologies showed that absorption refrigeration appears to be most promising technology. It already has widespread commercial usage for purposes where cooling below 0°C is required. It achieves relatively good Coefficient of Performance (COP) values compared to the other technologies in the review and there is wide range in possible heat sources, such as hot water, steam and exhaust. The heat supply temperature can be as low as 85-90°C, which opens the possibility for using district heating. Other technologies may be promising in the future, however refrigeration systems achieving cooling output below 0°C are currently only at a research level. However, there is one known exception, which is the snow production unit by DemacLenko which utilizes a heat driven ejector.

Calculations of heat requirements for a thermally driven refrigeration system for TIS production was performed based on the results from previous work in the "Snow for the future" project by Trædal (2018b) and Tveten (2020) for the Granåsen winter sports arena. Case 3 in Tveten's report was used as a base case for the calculations, and the results were extrapolated to create a second base case for a 1 km long alpine slope. The results showed that heat driven TIS production is very energy intensive with a specific energy requirement of 104 kWh/m^3 groomed snow produced, which more than twice the specific energy consumption of 50 kWh/m^3 for the electrically driven TIS system. Since the operational costs for current TIS production is already at a high level, mainly due to the electricity consumption, the cost of energy input needed for a thermally driven TIS production system would need to be correspondingly lower to be competitive.

The review of potential heat sources revealed that there is a potential for utilization of district heating and industrial waste heat. District heating supplied 6 TWh of heat energy in 2018, and 1 TWh from waste combustion was unutilized due to low demands, primarily during the summer. A potential challenge is that during the summer, when availability is at its highest, the supply temperature is possibly too low to be utilized for snow production. Furthermore, based on the heat requirement calculations and annual figures for the energy supply, not all municipalities where district heating exists will have enough available heat.

Potential Industrial waste heat sources are available in many industrial sectors, and a large fraction is available at suitable temperatures. However, there has so far been few projects where heat recovery has been used supply heat to external recipients, compared to power generation or internal re-use. A lack of local market for heating, missing heat transport infrastructure and technological challenges with heat recovery are among the limiting factors for utilization of industrial waste heat. Experience from projects where industrial waste heat has successfully been used for district heating, or other external recipients may provide further insight into the subject.

In the review of potential winter sports locations a total of 168 facilities and arenas were mapped within the disciplines of cross country skiing, biathlon, alpine skiing and ski jumping. The mapping showed that a high proportion of the facilities are located in south-eastern parts of Norway. A rough assessment based on factors such as facility size, competition level and proximity to large, populated areas, narrowed the list down to a total of 22 high potential facilities, of which 15 were located in a municipality with either district heating or industry with a waste heat potential. Furthermore, the investigation of co-location between all identified winter sports facilities and potential heat sources showed that district heating is the most prevalent heat source, with 40 % of the facilities located in municipalities with district heating. For industrial waste heat the number was 20 %. Even though in-depth investigations are needed to determine the actual feasibility for each case, these results provide a promising outlook.

The recommended further work is to perform case specific evaluations for a few of the facilities, for example the 22 high potential facilities mapped in this report. This may include:

- Detailed assessment of the individual winter sports locations, current snow production and conservation capabilities and potential needs and requirements for a TIS production system. Given sufficient input data the snow simulation tool can be used to make case specific assessments.
- The nearby heat sources should be evaluated in terms of availability and quality of heat and existing infrastructure (district heating). Compare the two options of connecting the heat source and winter sport facility: Transport of heat or snow.
- Investment and operational costs of a thermally driven TIS production system considering the investment costs of the TIS production unit, heat recovery and transport infrastructure, and the operational costs of energy, maintenance, and in-situ transportation of snow.

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Appendix

A Winter sports locations and potential heat sources

Table A-1: Overview of all mapped winter sports facilities and corresponding potential heat sources. Table abbreviations: CC/B: cross country / biathlon, A: Alpine, S: Ski jumping, NM: Norwegian championship, NC: Norwegian cup, NJC: Norwegian junior championship, WC: world cup, SC: Scandinavian cup, COC: continental cup, FIS: International Ski Federation.

			Discipline					Available District heating		Available Industrial waste heat			
Facility name	**Municipality**	**Region**	**CC/B**	**A**	**S**	**Events**	**Rank**		**Supplier**		**Companies**	**Sectors**	**Waste heat combustion in municipality?**
Altabakken	Alta	Troms- og Finnmark			Yes		3	Yes	Alta Fjernvarme, 7 GWh				
SarvesAlta	Alta	Troms- og Finnmark		Yes			3	Yes	Alta Fjernvarme, 7 GWh				
Kaiskuru Skistadion	Alta	Troms- og Finnmark	Yes			NM, NC	3	Yes	Alta Fjernvarme, 7 GWh				
Sandeggbakken	Alvdal	Innlandet			Yes		3						
Bardufoss	Bardufoss	Troms- og Finnmark	Yes	Yes			3						
Birkenesparken idrettsanlegg	Birkenes	Agder	Yes				3						
Skarmoen alpinpark	Bodø	Nordland		Yes			2	Yes	Bodø Energi, 56 GWh	Yes	Bodø sildoljefabrikk a.s	Food processing	
Vestvatn alpinanlegg	Bodø	Nordland		Yes		FIS	2	Yes	Bodø Energi, 56 GWh	Yes	Bodø sildoljefabrikk a.s	Food processing	
Bestmorenga fritidspark	Bodø	Nordland	Yes				2	Yes	Bodø Energi, 56 GWh	Yes	Bodø sildoljefabrikk a.s	Food processing	
Hovden alpinsenter	Bykle	Agder		Yes		FIS, NJC	2						
Kirkerudbakken	Bærum	Viken		Yes		FIS, NJC	2	Yes	Oslofjord varme, 89 GWh				
Fossum idrettsanlegg	Bærum	Viken	Yes				2	Yes	Oslofjord varme, 89 GWh				

Vestmarka ski- og skiskytteranlegg	Bærum	Viken	Yes				2	Yes	Oslofjord varme, 89 GWh				
Drammen	Drammen	Viken	Yes			WC	1	Yes	Drammen fjernvarme, 114 GWh	Yes	Vajda-papir scandinavia as, Krystal as, Bim norway as, Solenis norway as avd. drammen	Wood processing, Chemical processing, Chemical processing, Chemical processing	
Konnerud	Drammen	Viken	Yes		Yes	WC, NM	1	Yes	Drammen fjernvarme, 114 GWh	Yes	Vajda-papir scandinavia as, Krystal as, Bim norway as, Solenis norway as avd. drammen	Wood processing, Chemical processing, Chemical processing, Chemical processing	
Drammen skisenter	Drammen	Viken		Yes			1	Yes	Drammen fjernvarme, 114 GWh	Yes	Vajda-papir scandinavia as, Krystal as, Bim norway as, Solenis norway as avd. drammen	Wood processing, Chemical processing, Chemical processing, Chemical processing	
Gautefall biathlon stadion	Drangedal	Vestfold og Telemark	Yes			Gautefall sportsfestival (Sommer)	3						
Harpefossen	Eid	Vestland		Yes			3						
Ydalir hoppbakke	Elverum	Innlandet			Yes		3	Yes	Eidsiva 64 GWh				
Hermanshulua hoppbakker	Elverum	Innlandet			Yes		3	Yes	Eidsiva 64 GWh				
Holtanlia snøpark	Fauske	Nordland		Yes			3						
Klungsetmarka	Fauske	Nordland	Yes			NM	3						
Skeikampen	Gausdal	Innlandet	Yes	Yes			2						
Kremmerlia hoppbakker	Gausdal	Innlandet			Yes	District championship	3						
Vind skistadion	Gjøvik	Innlandet	Yes				3	Yes	Eidsiva 53 GWh				
Lygna	Gran	Innlandet	Yes		Yes	NM	2						
Bjørgan skianlegg	Grong	Trøndelag	Yes	Yes			2						
Gåsbu	Hamar	Innlandet	Yes		Yes	NM, NC	2	Yes	Eidsiva 159 GWh	Yes	K. a. rasmussen a.s	Ferro alloy and aluminium	
Lierberget Hoppsenter	Hamar	Innlandet			Yes	Youth NM	2	Yes	Eidsiva 159 GWh	Yes	K. a. rasmussen a.s	Ferro alloy and aluminium	
Harstad	Harstad	Troms- og Finnmark	Yes		Yes	NM, SC	2	Yes	Statkraft, 46 GWh				

Sollifjellet alpinsenter	Harstad	Troms- og Finnmark		Yes			2	Yes	Statkraft, 46 GWh				
Hattfjeldal skiskyting	Hattfjelldal	Nordland	Yes				3						
Hemsedal skisenter	Hemsedal	Viken	Yes	Yes		FIS, NC	1						
Geilo	Hol	Viken		Yes		FIS	1						
Hovden langrennsarena	Hovden	Agder	Yes				2						
Hurdal	Hurdal	Viken		Yes			3						
Lundbybakken	Hurdal	Viken			Yes		3						
Heistadmoen skistadion	Kongsberg	Viken	Yes			NM	2						
Kongsberg skisenter	Kongsberg	Viken		Yes		FIS	2						
Strai lysløypes skøytebane	Kristiansand	Agder	Yes				2	Yes	Agder energi 136 GWh	Yes	Rec solar norway as avd kristiansand, Glencore nikkelverk	Chemical processing, Ferro alloy and aluminium	
Norefjell	Krødsherad	Viken		Yes		FIS	1						
Furedalen - Kvamskogen	Kvam	Vestland		Yes			2			Yes	Elkem asa avd bjølvefossen	Ferro alloy and aluminium	
Vårdal Panorama skisenter	Kviteseid	Vestfold og Telemark		Yes			3						
Bjorli skisenter	Lesja	Innlandet		Yes		FIS	2						
Lysgårdsbakkene	Lillehammer	Innlandet			Yes	WC, COC, NM	1	Yes	Eidsiva 59 GWh				
Nye birkebeineren skistadion	Lillehammer	Innlandet	Yes		Yes	WC,NM	1	Yes	Eidsiva 59 GWh				
Hvaltjern skistadion	Lillestrøm	Viken	Yes				2	Yes	Akershus Energi, 123 GWh				
Bjønndalen skianlegg	Lillestrøm	Viken	Yes				3	Yes	Akershus Energi, 123 GWh				
Budor skistadion	Løten	Innlandet	Yes		Yes	NM	3						
Jervskogen skisenter	Malvik	Trøndelag	Yes				3						
Meråker vgsk skiskytingsanlegg	Meråker	Trøndelag	Yes				2						
Grova	Meråker	Trøndelag	Yes		Yes	NM, WC	2						

Meråker Alpinsenter	Meråker	Trøndelag		Yes			2						
Storlien skisenter	Meråker	Trøndelag		Yes			2						
Lifjell	Midt-Telemark	Vestfold og Telemark		Yes			3	Yes	Bø fjernvarme, 7 GWh				
Vikersund hoppsenter	Modum	Viken	Yes		Yes	NM, WC	1						
Modum skisenter	Modum	Viken		Yes			1						
Simostranda skianlegg	Modum	Viken	Yes			NC, NM	2						
Skaret skisenter	Molde	Møre og Romsdal	Yes			NM	2	Yes	Istadkraft, 11 GWh				
Molde	Molde	Møre og Romsdal			Yes		2	Yes	Istadkraft, 11 GWh				
Tusten skisenter	Molde	Møre og Romsdal		Yes			2	Yes	Istadkraft, 11 GWh				
Skarbakkene	Molde	Møre og Romsdal			Yes	NM	2	Yes	Istadkraft, 11 GWh				
Målselv fjelllandsby	Målselv	Troms- og Finnmark		Yes			3						
Åsen stadion	Nannestad	Viken	Yes			NM, NC, NJC	2	Yes	Statkraft, 10 GWh				
Narvik skisenter	Narvik	Nordland	Yes	Yes		FIS	2						Yes, Norcem Kjøpsvik (sementfabrikk) Foredlet avfallsbrensel
Nesbyen Alpinsenter	Nes	Viken		Yes			3						
Nes skianlegg	Nes	Viken	Yes		Yes	SC, NC, FIS	3	Yes	Akershus Energi, 15 GWh				
Varingskollen alpinsenter	Nittedal	Viken	Yes	Yes		FIS	2						
Valdres alpinsenter	Nord-Aurdal	Innlandet		Yes			3	Yes	Valdres biovarme, 6GWh				
Spåtind skisenter	Nordre Land	Innlandet		Yes			3						
Uvdal skisenter	Nore og Uvdal	Viken		Yes			2						

Grønkjær Skisenter	Notodden	Vestfold og Telemark	Yes				2	Yes	Thermokraft, 16 GWh				
Tveitanbakken	Notodden	Vestfold og Telemark			Yes	COC, FIS	2	Yes	Thermokraft, 16 GWh				
Høgås skiskytterarena	Notodden	Vestfold og Telemark	Yes				3	Yes	Thermokraft, 16 GWh				
Oppdal skisenter	Oppdal	Trøndelag		Yes		FIS	2						
Knyken skisenter	Orkland	Trøndelag	Yes	Yes	Yes	NC, Youth NM, Veteran NM	2	Yes	Orkland Energi, 16 GWh				
Kløvsteinbakken	Orkland	Trøndelag			Yes		3	Yes	Orkland Energi, 16 GWh				
Holmenkollen	Oslo	Oslo	Yes		Yes	WC, WC, NM	1	Yes	Fortum, 1.8 TWh	Yes	Nordox	Chemical processing	
Linderudkollen	Oslo	Oslo	Yes		Yes	NM	1	Yes	Fortum, 1.8 TWh	Yes	Nordox	Chemical processing	
Midtstubakken	Oslo	Oslo	Yes		Yes	NC, NM	1	Yes	Fortum, 1.8 TWh	Yes	Nordox	Chemical processing	
Oslo vinterpark - Tryvann	Oslo	Oslo		Yes		FIS	1	Yes	Fortum, 1.8 TWh	Yes	Nordox	Chemical processing	
Lillomarka skiarena	Oslo	Oslo	Yes				2	Yes	Fortum, 1.8 TWh	Yes	Nordox	Chemical processing	
Fageråsbakken	Rana	Nordland			Yes		3	Yes	Mo Fjernvarme, 89 GWh	Yes	Celsa armeringsstål as, Ferroglobe mangan norge as, Elkem rana as	Ferro alloy and aluminium, Ferro alloy and aluminium, Ferro alloy and aluminium	
Skillevollen Alpinsenter	Rana	Nordland	Yes	Yes	Yes		3	Yes	Mo Fjernvarme, 89 GWh	Yes	Celsa armeringsstål as, Ferroglobe mangan norge as, Elkem rana as	Ferro alloy and aluminium, Ferro alloy and aluminium, Ferro alloy and aluminium	
Kvitfjell	Ringebu	Innlandet		Yes		WC	1						
Hovsmarka skiskytteranlegg	Ringerike	Viken	Yes				3	Yes	Vardar Varme, 54 GWh	Yes	Smurfit kappa norpapp as avd. hønefoss	Wood processing	
Sjusjøen Natrudstilen skisenter	Ringsaker	Innlandet	Yes	Yes			1	Yes	Eidsiva 22 GWh	Yes	Strand unikorn as - (strand brænderi)	Chemical processing	
Veldre Sag	Ringsaker	Innlandet	Yes			NC	2	Yes	Eidsiva 22 GWh	Yes	Strand unikorn as - (strand brænderi)	Chemical processing	
Røros Alpinsenter hummelfjell	Røros	Trøndelag		Yes			3						

Røros	Røros	Trøndelag	Yes			NM	3						
Eikedalen	Samnanger	Vestland		Yes			2						
Sandnes sparebank arena	Sandnes	Rogaland	Yes				2	Yes	Lyse, 11 GWh				
Sauda	Sauda	Rogaland		Yes			2			Yes	Eramet norway as, sauda	Ferro alloy and aluminium	
Selbuskogen skisenter	Selbu	Trøndelag	Yes				3						
Haglebu skisenter	Sigdal	Viken		Yes			3						
Sirdal skisenter	Sirdal	Agder		Yes			2						
Feed skiarena	Sirdal	Agder	Yes				3						
Ramsjøen skiskytterarena	Skaun	Trøndelag	Yes				3						
Jarseng idrettsanlegg	Skien	Vestfold og Telemark	Yes		Yes		2	Yes	Skien fjernvarme, 47 GWh				
Sogndal Skisenter Hodlekve	Sogndal	Vestland		Yes			2						
Steinkjer skistadion	Steinkjer	Trøndelag	Yes			NM	2			Yes	Nortura sa steinkjer	Food processing	
Steinfjellbakken	Steinkjer	Trøndelag			Yes	NC, NM	2			Yes	Nortura sa steinkjer	Food processing	
Steinkjerbakken	Steinkjer	Trøndelag			Yes	NC	2			Yes	Nortura sa steinkjer	Food processing	
Bjørkbakken	Stjørdal	Trøndelag			Yes	NC	3	Yes	Statkraft, 27 GWh				
Klempen skianlegg	Stjørdal	Trøndelag	Yes		Yes		3	Yes	Statkraft, 27 GWh				
Storås langrennsarena	Stokke	Vestfold og Telemark	Yes				2						
Stranda - skisenter	Stranda	Møre og Romsdal		Yes			2						
Bjørkelibakken	Stryn	Vestland			Yes	NM	3						
Stryn Vinterski	Stryn	Vestland		Yes			3						
Ullsheim skianlegg	Stryn	Vestland	Yes				3						
Nordfjord fritidssenter	Stryn	Vestland	Yes	Yes			3						
Jølster skisenter	Sunnfjord	Vestland	Yes	Yes		FIS	3	Yes	Førdefjorden energi, 14 GWh				
Langeland Skisenter,	Sunnfjord	Vestland	Yes				3	Yes	Førdefjorden energi, 14 GWh				
Surnadal Alpinsenter	Surnadal	Møre og Romsdal		Yes			3						

Sunnmørsalpene skiarena	Sykkylven	Møre og Romsdal		Yes			3						
Gålå skianlegg	Sør-Fron	Innlandet	Yes	Yes		NM, NC, FIS	2						
Jordalen hoppanlegg	Sør-Fron	Innlandet			Yes	Youth NM	2						
Sandnes Alpinsenter	Sør-Varanger	Troms- og Finnmark		Yes			3						
Gaustablikk	Tinn	Vestfold og Telemark	Yes		Yes		2						
Huka Høydalsmo	Tokke	Vestfold og Telemark	Yes		Yes	NC	3						
Hallbjønn høyfjellsenter	Tokke	Vestfold og Telemark		Yes			3						
Hamran	Tolga	Innlandet			Yes	NM	3						
Sætersgård stadion	Tolga	Innlandet	Yes			NM	3						
Storelva skistadion	Tromsø	Troms- og Finnmark	Yes			NM	2	Yes	Kvitebjørn fjernvarme, 149 GWh	Yes	Cognis scandinavia produksjon	Chemical processing	
Tromsø skistadion	Tromsø	Troms- og Finnmark	Yes			NM	2	Yes	Kvitebjørn fjernvarme, 149 GWh	Yes	Cognis scandinavia produksjon	Chemical processing	
Grønnåsen Hoppsenter	Tromsø	Troms- og Finnmark			Yes		2	Yes	Kvitebjørn fjernvarme, 149 GWh	Yes	Cognis scandinavia produksjon	Chemical processing	
Tromsø alpinpark	Tromsø	Troms- og Finnmark		Yes			2	Yes	Kvitebjørn fjernvarme, 149 GWh	Yes	Cognis scandinavia produksjon	Chemical processing	
Granåsen	Trondheim	Trøndelag	Yes		Yes	NM, WC	1	Yes	Statkraft, 663 GWh	Yes	Ranheim paper & board as	Wood processing	
Saupstad skiskytteranlegg	Trondheim	Trøndelag	Yes				2	Yes	Statkraft, 663 GWh	Yes	Ranheim paper & board as	Wood processing	
Vassfjellet Vinterpark	Trondheim	Trøndelag		Yes		FIS	2	Yes	Statkraft, 663 GWh	Yes	Ranheim paper & board as	Wood processing	
Trysilfjellet arena	Trysil	Innlandet	Yes				1	Yes	Eidsiva, 48 GWh				
Trysilfjellet skisenter	Trysil	Innlandet		Yes		COC	1	Yes	Eidsiva, 48 GWh				
Fulufjellet alpinsenter	Trysil	Innlandet		Yes			2	Yes	Eidsiva, 48 GWh				
Tydal skisenter	Tydal	Trøndelag		Yes			3						
Savalen Skiarena	Tynset	Innlandet	Yes	Yes			3						
Bibo idrettspark	Tønsberg	Vestfold og Telemark	Yes				3	Yes	Skagerak varme 48 GWh				

Røldal skisenter	Ullensvang	Vestland		Yes			3						
Fonna Glacier ski resort	Ullensvang	Vestland	Yes	Yes			3						
Korlevoll skistadion	Ullensvang	Vestland	Yes				3						
Vadsø skistadion	Vadsø	Troms- og Finnmark	Yes				3						
Tyin-filefjell skisenter	Vang	Innlandet		Yes			3						
Raudalen alpinsenter	Vang	Innlandet		Yes		World championship (Telemark)	3						
Sjåmoen Skistadion	Vefsn	Nordland	Yes			Regional race	2			Yes	Alcoa mosjøen	Ferro alloy and aluminium	
Kjemsåsen Alpinsenter	Vefsn	Nordland		Yes			2			Yes	Alcoa mosjøen	Ferro alloy and aluminium	
Henningvola	Vekre	Trøndelag	Yes				3						
Sandripheia idrettspark	Vennesla	Agder	Yes		Yes		3						
Vinne skiskytteranlegg	Verdal	Trøndelag	Yes				3						
Haukeli	Vinje	Vestfold og Telemark		Yes			2						
Rauland - Raulandsfjell	Vinje	Vestfold og Telemark		Yes			2						
Rauland - vierli	Vinje	Vestfold og Telemark	Yes	Yes			2						
Rauland skisenter	Vinje	Vestfold og Telemark		Yes			2						
Volda skisenter	Volda	Møre og Romsdal	Yes	Yes			3						
Hornindal skisenter	Volda	Møre og Romsdal		Yes			3						
Voss ski og tursenter	Voss	Vestland	Yes			NM, FIS	2						
Bavallsbakken	Voss	Vestland			Yes	NM, COC	2						

Voss resort	Voss	Vestland		Yes		NM jr.	2						
Myrkdalen	Voss	Vestland		Yes			2						
Lemonsjøen skisenter	Vågå	Innlandet		Yes			3						
Ørsta skisenter	Ørsta	Møre og Romsdal		Yes			3	Yes	Tussa Energi, 10 GWh				
ØTS skisenter karidalen	Østre Toten	Innlandet	Yes			Youth NM	2	Yes	Eidsiva 8 GWh				
Hafjell alpinsenter	Øyer	Innlandet	Yes	Yes		FIS, NJC	1						
Beitostølen arena	Øystre Slidre	Innlandet	Yes			NC, FIS	1	Yes	Stølslie Biovarme, 7 GWh				
Beitostølen skisenter	Øystre Slidre	Innlandet		Yes			1	Yes	Stølslie Biovarme, 7 GWh				
Ål skisenter	Ål	Viken		Yes		FIS	3						
Rena	Åmot	Innlandet	Yes			NC	2						
Renabakken	Åmot	Innlandet			Yes	NC, COC, NM	2						
Rena Alpin- og skisenter	Åmot	Innlandet		Yes			3						
Ljosland skisenter	Åseral	Agder		Yes			3						

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