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Utilization of surplus heat from snow producing machines

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MASTER THESIS

for

Student Bernhard Haver Vagle

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Utilization of surplus heat from snow producing machines

Utnyttelse av spillvarme fra snøproduserende maskineri

Background and objective

In the perspective of increased global warming, there is a challenge having available snow close to the cities and villages in the mountain for a reasonable winter activity and sport season. The periods of natural snow is shorter and in areas, the snow in the winter is disappearing. In Europe the facilities is moving to higher locations to be able to arrange winter games.

In the Nordic countries, it is also a tradition for doing winter activities in snow like kinder gardens, schools and for the families to go skiing in weekends and holidays. If the trend with milder winters is going on, the travel distance to areas with snow will increase. To be able to maintain the snow activity close to the cities, it will be of importance to produce snow at temperatures above 0°C. Since this is an energy consuming process, the energy efficiency of the equipment is of importance and the possibility to utilize the heat for space heating or hot tap water, with emphasise on reducing the operational costs.

This master thesis work is a cooperation with Trondheim Kommune and the Norwegian Ski Assosiation within the scope of "Snow for the future". Trondheim Kommune will in the near future build a centre in Granåsen that will give possibilities for future winter games and give the people possibility to enjoy the winter activity in the period from November to March.

To be able to minimize the cost of operation it would be of interest to investigate when the snow production should be done (during the year), given electicity price, storage possibilities and utilization of the surplus energy. The investigation should include the costs of truck loading and transport.

The following tasks are to be consider:

1. Literature review of utilization of surplus heat from snow making machines

2. Define and investigate different strategies/cases of snow production in relation to heat recovery, electric energy costs/price, storage and transport
3. Make a calculation tool/model to investigate the potential for heat utilization and the economical aspects based on the different strategies
4. Based on the calculation tool it should be made a comparison of the different alternatives
5. Make a draft scientific paper from the main results of the master thesis
6. Make proposal for further work

- " --

Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. The final report in Word and PDF format, scientific paper and all other material and documents should be given to the academic supervisor in digital format on a DVD/CD-rom or a memory stick at the time of delivery of the Master Thesis.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, January 18th 2016

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Preface

This is a master thesis written at the Norwegian University of Science and Technology (NTNU).

As an active cross-country skier, the project has been truly interesting for me from the beginning. I have grown up in western Norway, where snow is a rarity even in the winter. As the weather seems to be moving in the wrong direction, it will be important to come up with solutions to ensure proper ski conditions for the future.

Finding a strategy for sustainable snow supply is a challenging task. In this thesis, heat recovery has been emphasized as a way of making temperature independent snowmaking sustainable. Working with the thesis has been demanding, as a large amount of information had to be collected. A system for snow supply is complex, and a thorough analysis should cover a wide range of areas, from the approach temperature in heat exchangers to the fuel consumption of snow blowers.

I would like to thank my supervisor, Trygve Magne Eikevik, and my co-supervisor, Bjørn Aas, for guidance. Also, I have appreciated the cooperation with fellow student Jon-Brede Rykkje Dieseth during the thesis.

I personally hope that the effort put down in the candidacy to host the Nordic World Ski Championships in 2021 will be brought on to a following campaign, and that Trondheim and Granåsen will be selected to host the Championships in 2023.

For the white winters to come.

Trondheim, June 2016

Bernhard Haver Vagle

Abstract

In this master thesis, the performance of four cases of snow supply have been evaluated under the condition that a 5 km track of ski conditions can be guaranteed from November to the end of April at a nordic ski arena in Granåsen, outside of Trondheim, Norway. The four cases are:

- Case A: snow storage
- Case B: temperature independent snowmaking with direct heat recovery
- Case C: indoor snowmaking with direct heat recovery
- Case D: temperature independent snowmaking with indirect heat recovery

The evaluation of the cases is based on costs and energy consumptions, while ecological impacts, maintenance and interest rates are not covered in the analysis. Heat recovery is implemented using a CO_2-heat pump to deliver 1,5 GWh of water at 70 °C to three planned buildings at the ski arena, either directly or indirectly, through a borehole thermal energy storage (BTES) system. Multiple methods have been applied to compare the cases. A literature study on snowmaking, ice production, snow storage and heat recovery is performed. Moreover, calculations and simulations based on the theory of refrigeration technology, heat transfer and fluid dynamics are conducted. Finally, public price lists and conversations with sources and suppliers are used to estimate investment costs and electricity prices.

The estimated investment costs of case A are 2,1 MNOK, which is lowest by far, among the cases, as the rest are in the range of 17,2-32 MNOK. None of the cases obtain operating costs below 0 NOK per m^3 snow (NOK/m^3), due to a highly cost demanding process of distributing the snow to the ski tracks. This process is based on a single example from Granåsen in 2015 with estimated operating costs of 54,43 NOK/m^3. From other examples, these costs can be expected to be decreased to 23,5 NOK/m^3, which would result in total operating costs below 0 NOK/m^3 for the cases involving heat recovery. Thus, a continuous operation of these cases to fully utilize the investment costs would be desired, if the demand for heat was present.

<u>Based on the findings in this thesis, snow storage is best suited in Granåsen among the cases considered.</u> This is because the estimated accessible heat demand in Granåsen is low, not allowing the other cases to fully utilize their potential. <u>The operating costs of case A is estimated to be 59,48 NOK/m^3, and it would take 36 years before case B would equalize the total costs</u> at an average electricity price of 0,8 NOK/kWh. The focus in Granåsen should be on automation of snowmaking and methods for distribution of snow. However, for a general ski arena/resort, a continuous operation would improve the effectiveness of the cases with heat recovery if the demand for heat was present, and if the costs related to distribution were decreased. Case C, possibly in combination with case D, is the most promising option as such. This would give 12 GWh/yr of surplus heat, nearly 200.000 m^3/yr of snow and savings of 7,1 MNOK/yr. If the investment costs were held fixed, the payback period would be less than 5 years. <u>Thus, future ski arenas/resorts should be considered located nearby heat demanding industry, shopping malls or similar, which in turn would move the ski tracks closer to populous areas.</u>

Sammendrag

I denne masteroppgaven har fire case på snøforsyning blitt evaluert med forutsetning om at det skal kunne tilbys skiforhold i en løype på 5 km fra november til slutten av april ved Granåsen skisenter, utenfor Trondheim. De fire casene er:

- Case A: snølagring
- Case B: temperaturuavhengig snøproduksjon med direkte varmegjenvinning
- Case C: innendørs snøproduksjon med direkte varmegjenvinning
- Case D: temperaturuavhengig snøproduksjon med indirekte varmegjenvinning

Evalueringen er basert på kostnader og energiforbruk, mens ytterligere miljøpåvirkninger i tillegg til vedlikehold og renter, ikke er diskutert. Varmegjenvinning er implementert ved å utnytte en CO_2-varmepumpe til å levere 1,5 GWh av vann ved 70 °C til tre planlagte bygninger på skisenteret, enten direkte eller indirekte, gjennom et termisk energilager med borehull. Flere metoder er brukt for å sammenligne casene. Et litteraturstudie på snøproduksjon, isproduksjon, snølagring og varmegjenvinning er gjennomført. Beregninger og simuleringer er i tillegg utført, basert på teori rundt kuldeteknikk, varmeoverføring og fluiddynamikk. Offentlige prislister og samtaler med kilder og leverandører er videre brukt til å estimere kostnader og elektrisitetspriser.

Estimert investeringskostnad for case A er 2,1 MNOK, som er klart lavest blant casene, da resten ligger på 17,2-32 MNOK. Ingen av casene har driftskostnader på under 0 NOK per m^3 snø (NOK/m^3), på grunn av en svært kostnadskrevende prosess med å distribuere snø ut i løypene. Denne prosessen er basert på et enkelt eksempel fra Granåsen i 2015, med estimerte driftskostnader på 54,43 NOK/m^3. Fra andre eksempler kan det tyde på at en senking av disse kostnadene til 23,5 NOK/m^3 er realistisk. Dette ville ført til totale driftskostnader på under 0 NOK/m^3 for case B-D. Dermed ville en kontinuerlig drift av disse casene vært ønskelig for å utnytte investeringskostnadene maksimalt, dersom det fantes et tilstrekkelig behov for varme.

Basert på resultatene, er snølagring best egnet i Granåsen blant de analyserte casene. Dette fordi det estimerte tilgjengelige varmebehovet i Granåsen er for lavt til at de andre casene får utnyttet sitt potensiale. Driftskostnadene til case A er estimert til 59,48 NOK/m^3, og det vil ta 36 år før de totale kostnadene blir utlignet av case B ved en elektrisitetspris på 0,8 NOK/kWh. Fokuset i Granåsen bør følgelig ligge på automatisering av snøproduksjon, samt utvikling av en mer effektiv distribusjonsprosess. Samtidig, for et vilkårlig skianlegg vil en kontinuerlig drift kunne forbedre ytelsen til case B-D dersom kostnadene relatert til distribusjon kan senkes, og dersom et tilstrekkelig varmebehov er til stede. Case C, eventuelt i kombinasjon med case D, er den mest lovende løsningen i så måte. Dette vil gi 12 GWh/år med overskuddsvarme og nesten 200.000 m^3/år med snø, som betyr besparelser på 7,1 MNOK/år. Dersom investeringskostnadene antas å være uforandret vil nedbetalingstiden til en slik løsning være på under 5 år. Det bør derfor vurderes å plassere fremtidige skianlegg i nærheten av varmekrevende industri, kjøpesentre eller lignende, noe som dessuten vil flytte skiløypene nærmere folkerike områder

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Symbols and abbreviations

A	Area	[m^2]
C	Degree-day coefficient	
c	constant	[kW/°C]
cp	Specific heat capacity	[kJ/kg·K]
D	Diameter	[m]
f	Darcy friction factor	
ΔH	Specific enthalpy difference	[kJ/kg]
H	Height	[m]
L	Length	[m]
lf	Specific latent heat of fusion	[kJ/kg]
lv	Specific latent heat of vaporization	[kJ/kg]
M	Melting rate	[m^3/day]
m	Mass	[kg]
$\dot{m}$	Mass flow rate	[kg/s]
η	Efficiency	
ΔP	Pressure difference	[bar]
ρ	Density	[kg/m^3]
Q	Heat	[kW]
q	Specific heat	[kJ/kg]
q_B	Specific heat extraction rate	[W/m]
ΔT	Temperature difference	[K]
ΔT_{LM}	Logarithmic mean temperature difference	[K]
T_w	Wet-bulb temperature	[°C]
U	Overall heat transfer coefficient	[W/m^2·K]
V	Volume	[m^3]
v	Velocity	[m/s]
$\dot{V}$	Volumetric flow rate	[m^3/s]
W	Work	[kW]
W	Width	[m]
π	Pressure ratio	

BTES	Borehole thermal energy storage	
COP	Coefficient of performance	
EVR	Energy-volume ratio	[kWh/m^3]
EED	Earth Energy Designer	
CVR	Cost-volume ratio	[NOK/m^3]
GSHP	Ground source heat pump	
HFC	Hydrofluorocarbon	
MET	Monoethylene Glycol	
PPF	Production potential, fan gun	[m^3/hr]
PPL	Production potential, lance	[m^3/hr]
RH	Relative humidity, air	
SEER	Specific energy extraction rate	[kWh/m·yr]
SF220	the Snowfactory model 220 from TechnoAlpin	
SPF	Seasonal performance factor	
TDS	Temperature dependent snowmaker	
TIS	Temperature independent snowmaker	

1. Introduction

The effects of global warming have already made its mark on the length of the winter season across the world. The average temperature on earth has increased by 0,74 °C during the 20th century, and are expected to increase further at a higher rate than previously [1]. If the temperatures continues to rise as predicted, the natural snow will gradually disappear from the ski tracks, and the *production potential* of snow from temperature dependent snowmakers will decline. Some examples follow to clarify the situation:

- The number of days with ski conditions in Oslo, Norway, has been reduced by 1-2 months over the last century.
- In 2050, it is assumed that the length of the winter in Oslo will be halved compared to 1980 [2].
- During the winter of 2014/2015, 24 out of 66 nordic skiing competitions in Nord-Trøndelag county, Norway, were cancelled due to a lack of snow [3].

The focus of this thesis will be on a nordic ski arena located in Granåsen, outside of Trondheim, Norway. Trondheim hosted the FIS Nordic World Ski Championships in Granåsen in 1997, and several World Cup Events in ski jumping, nordic combined, biathlon and cross-country skiing have been held here. Trondheim was a host candidate for the FIS Nordic World Ski Championships in 2021, but was not selected. However, the facilities in Granåsen are to be expanded for 800 MNOK [4], and it is likely that a new application will be submitted for the Championships in 2023. Based on a series of winters with poor snow conditions, not to mention the climate predictions, a strategy for keeping snow in the ski tracks during the winter at a reasonable energy consumption and cost has to be determined.

In the following, four cases will be evaluated, given the condition that a 5 km track of snow can be guaranteed in Granåsen from November 1 to April 30. A 5 km track of snow, 6 m wide and 0,4 m deep will require 12.000 m^3 of snow. The objective is to obtain the *energy-volume ratio (*EVR), *cost-volume ratio* (CVR), investment costs and operating costs of the four cases. The EVR and CVR are defined as the energy consumption and cost per m^3 of snow produced, in kWh/m^3 and NOK/m^3 respectively. The four cases are:

- Case A: snow storage
- Case B: temperature independent snowmaking with direct heat recovery
- Case C: indoor snowmaking with direct heat recovery
- Case D: temperature independent snowmaking with indirect heat recovery

The cases are picked to cover a wide range of methods, but the focus will be on outdoor ski tracks of snow. Other alternatives such as dry ski slopes and ski tunnels are not considered. Temperature independent snowmaking and snow storage are techniques that recently have been applied in order to meet the growing demand for snow, while indoor snowmaking for outdoor ski tracks is not widespread. Heat recovery, implemented either directly or indirectly, through energy wells, is emphasized as a way of making the last three cases sustainable.

Figure 1: Future Granåsen [5].

1.1 Methods

The methods used in the analysis are:

- A literature study on snowmaking, ice production, snow storage and heat recovery, including comparable examples,
- Calculations based on the theory of refrigeration technology, heat transfer and fluid dynamics.
- Simulations made with the software Earth Energy Designer (EED) and CoolPack.
- Public price lists, and conversations with sources and suppliers to estimate investment costs and electricity prices.

1.2 Limitations

Any assumption made to simplify the analysis will be stated. It should be stressed that the results obtained are not fixed, but serves as a tool for comparison between the cases. However, the results will be thoroughly discussed to enlighten possible improvements or errors. The goal of this thesis is not to come up with a definitive answer, but to draw a picture of the characteristics of the cases. Costs and energy consumptions related to maintenance are neglected. Moreover, ecological impacts such as CO_2-emissions are not covered in the analysis, apart from energy consumptions. Still, the power sources are assumed to be renewable. Finally, the economic analysis does not include the time value of money.

1.3 Thesis structure

After the introduction, a literature study will follow, before the four cases are presented and evaluated in turn. The literature study is rather comprehensive, meant to support the results. Possible sources of error will be listed prior to the conclusion, followed by suggestions for further research. Symbols and abbreviations are listed in the previous chapter, and abbreviated subscripts will be explained during the thesis. Additional details are given in the Appendix.

2. Literature study

2.1 Temperature dependent snowmaking

The simple working principle of a temperature dependent snowmaker (TDS) is that tiny water droplets are sprayed into the air, where they are supposed to freeze and turn to snow. Artificial snow was first produced without intention by Dr. Ray Ringer. He was simulating natural weather to study how jet engines would react to rime ice. By spraying water into a cold wind tunnel, he created snow instead of ice. No patent was made, as this was not the objective of the research. However, a scientific report was published regarding the jet engines [6].

Tey Manufacturing Company had remarked how the bad winters were affecting their sales of skis. Inspired by Dr. Ringer, the first snowmaker was invented in 1950 by Art Hunt, Dave Richey and Wayne Pierce [6]. The machine was supplied with compressed air and water, a so called air/water snow gun. Alden Hanson later made the first patent of a so called airless snow gun in 1958. This type of snowmaker had a fan and a built-in compressor. Most TDSs today are developed around one of these two types of snowmakers.

Since the start in 1950, the use of snowmakers escalated during the 1970s, especially for alpine ski resorts. Today, 90% of all ski resorts rely on snowmakers to satisfy their demand for snow [7].

2.1.1 Factors affecting snowmaking

There are a couple of factors to consider regarding snowmaking, the most important being the wet-bulb temperature (T_W), humidity and water temperature. Others include wind speed, frost, water pollution as well as the droplet size of the water leaving the snowmaker [8].

The T_W is a combination of the dry-bulb temperature and the relative humidity of the air (RH). The dry-bulb temperature is normally called air temperature, and is simply the temperature to be read on a thermometer. The RH is defined as the ratio of the water content of the air to the maximum water content at a certain temperature and pressure. As the temperature and pressure decreases, less water can be contained by the air. If the RH is 100%, the air is saturated and cannot absorb more water. This point is also called the dew point temperature, and marks the distinction where the air starts to condensate water. At this temperature T_W equals the dry-bulb temperature. The T_W decreases with a reduced RH, and is always lower than or equal to the dry-bulb temperature.

Furthermore, the T_W is the temperature one will read, if a wet cloth is put around the mercury bulb of a thermometer, hence the name. The reason for the different readings originates from evaporation of water in the wet cloth to the surrounding air. When water evaporates, energy in the form of heat is released, leading to the lower readings on the thermometer. This phenomenon is called evaporative cooling. Water droplets will freeze faster in dry air due to evaporation, which is why the T_W is so important when it comes to snowmaking [9].

For water droplets to freeze, heat has to be removed until the temperature of the water is below 0 °C, but this is not sufficient to ensure freezing. Although the freezing process can begin at 0 °C, water can be supercooled down to its crystal homogenous nucleation at -48 °C without freezing. For this to happen, the water has to be 100% pure. This is because ice crystals will grow around small particles in the water, in a heterogeneous nucleation. For this reason, natural water sources such as dams and rivers are better suited for snowmaking than tap water, which is often too pure. In addition, particles are often added to the water to enhance the probability of freezing to occur. A widely used additive is a natural protein called Snomax. Snomax is a nucleating agent, able to create cores at temperatures up to -3 °C, which is beneficial for snowmaking in marginal temperatures [10]. The amount of Snomax added to the water is approximately 0,9 g per 1.000 l [11], which can increase the amount of snow produced by up to 40% [10]. There are detected no negative environmental consequences from usage of Snomax [12].

Among other conditions for snowmaking, the temperature and pressure of the air fed to the snowmaker is of importance. Pressurized air will be expanded in the ambient air, which will lead to a drop of temperature, according to the Joule-Thompson effect. Furthermore, the snowflakes will melt rapidly without frost in the ground. Finally, the water droplets should be as small as possible to increase the surface area to volume ratio, but too small droplets will drift away. A droplet diameter of 200-700 μm has shown to be a good trade-off [8].

Figure 2 displays the snowmaking conditions at different wet-bulb temperatures. Heat and mass transfer during snowmaking is in the form of convection and evaporation. The cold surrounding air cools the droplets with convection, while evaporation of water leads to evaporative cooling. Convection dominates for air temperatures below -7 °C, and evaporation is more important for air temperatures above -7 °C [13]. Note that convection requires wind, and evaporative cooling requires a RH below 100% to contribute in the cooling process.

Good Snow Quality | Poor Snow Quality | No Snowmaking

Temp C / Humidity	10%	15%	20%	25%	30%	35%	40%	45%	50%	55%	60%	65%	70%	75%	80%	85%	90%	95%	100%
-9	-12	-12	-12	-12	-12	-12	-12	-11	-11	-11	-11	-11	-11	-10	-10	-10	-10	-9	-9
-8	-12	-11	-11	-11	-11	-11	-11	-10	-10	-10	-10	-9	-9	-9	-9	-9	-9	-8	-8
-7	-10	-10	-10	-9	-9	-9	-9	-9	-9	-8	-8	-8	-8	-8	-7	-7	-7	-7	-7
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-5	-9	-9	-8	-8	-8	-8	-8	-7	-7	-7	-7	-7	-6	-6	-6	-6	-6	-6	-5
-4	-8	-8	-8	-8	-8	-7	-7	-7	-7	-7	-6	-6	-6	-6	-6	-5	-5	-5	-4
-3	-7	-7	-7	-7	-6	-6	-6	-6	-5	-5	-5	-4	-4	-4	-4	-3	-3	-3	-3
-2	-7	-7	-6	-6	-6	-6	-5	-5	-5	-4	-4	-4	-4	-3	-3	-3	-3	-3	-2
-1	-6	-6	-5	-5	-4	-4	-4	-3	-3	-3	-3	-2	-2	-2	-2	-2	-1	-1	-1
0	-5	-5	-4	-4	-4	-4	-3	-3	-3	-3	-2	-2	-2	-2	-1	-1	-1	-1	0
1	-5	-4	-4	-4	-3	-3	-3	-3	-2	-2	-2	-2	-1	-1	-1	-1	0	0	1
2	-4	-3	-3	-3	-2	-2	-2	-1	-1	-1	-1	0	1	1	1	1	2	2	2
3	-3	-3	-3	-2	-2	-2	-1	-1	-1	0	0	1	1	1	2	2	2	3	3
4	-2	-2	-1	-1	-1	0	0	1	1	1	2	2	2	3	3	3	4	4	4

Figure 2: Snowmaking chart. All temperatures are wet-bulb temperatures, except from the column to the left which are dry-bulb temperatures [14]

The lower limit for snowmaking in Figure 2 is a wet-bulb temperature of -3 °C, however a limit of -2 °C is also stated [11]. In practice, snowmaking starts at colder temperatures, and a questionnaire given to many ski arenas/resorts in Sweden stated that -5,5 °C was a normal start-up temperature [15]. Generally, the lower the *Tw*, the more snow can be produced and at a better quality. The snowmaking chart can be further explained by the Mollier diagram in Figure 3. The Mollier diagram shows the relationship between the air temperature, the RH and the specific enthalpy of the air. The enthalpy lines for good snow quality and poor snow quality, as defined in Figure 2, is marked, and the difference between them is 6,2 kJ/kg_{Air}.

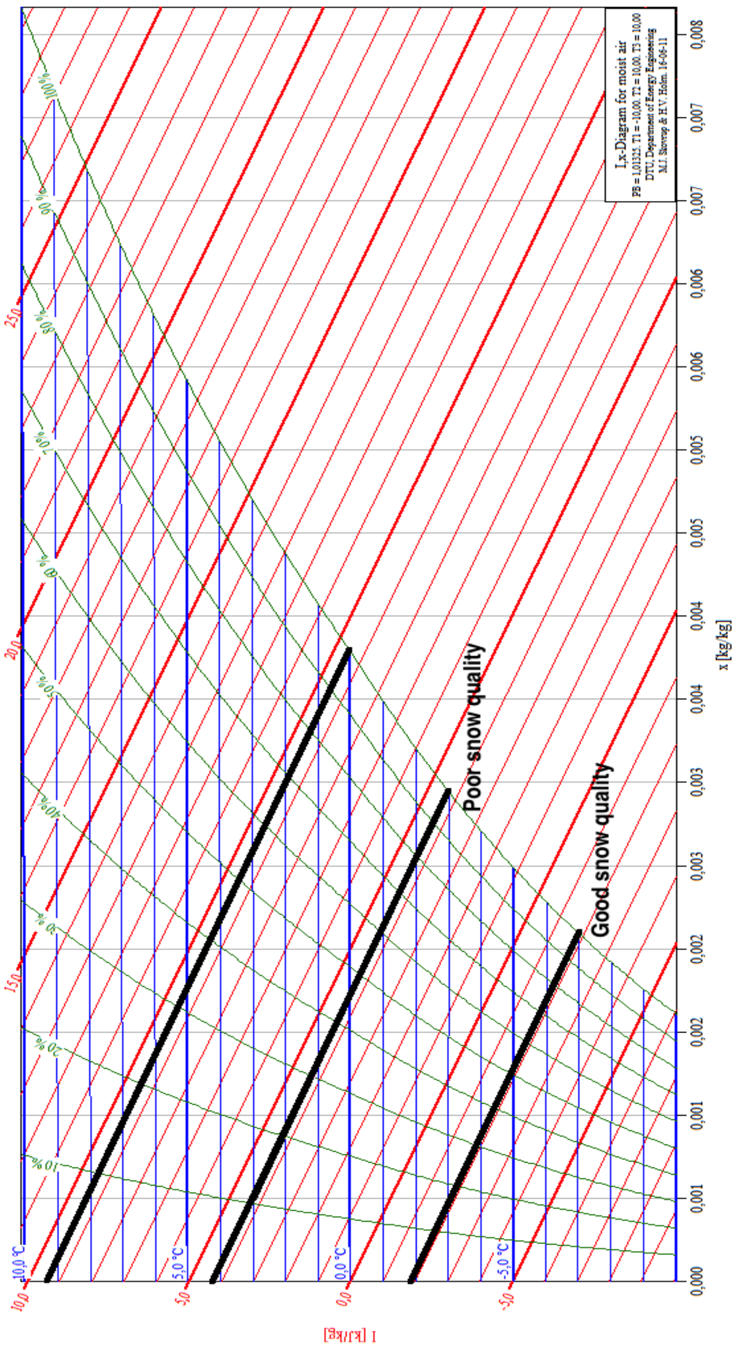


Figure 3: Mollier diagram with air temperature (blue), RH (green) and specific enthalpy (red). The x-axis shows the moisture content of the air in kg_{Water}/kg_{Air}. The black lines show the specific enthalpy of the air at T_W= 0 ℃, T_W= -3 ℃ (poor snow quality) and T_W =-7 ℃ (good snow quality).

2.1.2 Natural snow and artificial snow

The most important difference between natural snow and artificial snow is the shape of the snowflakes. While natural snow has plenty of time on its way down to form, the artificial snow only has a couple of seconds before it hits the ground. Natural snow forms from water molecules that freeze from the outside and inwards, creating hexagonal patterns as can be seen in Figure 4. Artificial snow nucleates around a core, forming spherical snowflakes. As a result, artificial snow is more compact than natural snow, and it resists wind, water and temperature impacts to a greater extent [11]. The density of freshly fallen natural snow is around 100 kg/m^3, while artificial snow has a density of 400-500 kg/m^3. As the density of water is 1000 kg/m^3, 1 m^3 of water will produce 2-2,5 m^3 of snow. Old natural snow has a density of around 500 kg/m^3 [16]. Both natural snow and artificial snow made with TDS are great skiing surfaces.

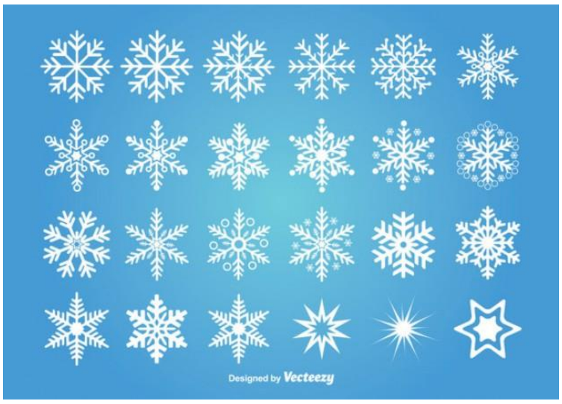


Figure 4: Hexagonal patterns of natural snowflakes [17].

2.1.3 Types of TDSs

The main types of TDSs can be divided into four groups [18]:

- **Air/water snowmakers**
  These are the traditional snowmakers, supplied with compressed air and water and mounted on a sled or a tower. The production rate from these snowmakers are quite good, but the energy consumption is very high, due to a high consumption of compressed air. In addition, these snowmakers have a high noise level. These snowmakers work well in marginal temperatures, but are nevertheless being phased out of the market, due to a high energy consumption [11].

- **Lances**
  These are modified air/water snowmakers with a 70-80% reduction in the use of compressed air [19]. This results in a snowmaker with a low energy consumption and a low noise level. The reduced amount of compressed air, leads to a lower droplet speed out of the nozzles, meaning that these snowmakers should be mounted in towers to obtain enough air time for the droplets to freeze. It also means that cold temperatures are required before snowmaking can start, and the production rate is not the highest. These

snowmakers are popular today, and they work especially well in cold temperatures and for narrow tracks, due to a short throw distance.

- **Watersticks**
  Watersticks are not supplied with compressed air, and this results in a very low energy consumption. However, a waterstick requires low temperatures to produce snow, and the production rate is very low. These snowmakers have a low noise level and are mounted in towers.

- **Fan guns (airless snow guns)**
  Fan guns differs from the other types in that they use a fan to blow the snow, leading to a very high production rate of snow. Fan guns are formed as a cylinder, with a fan on the back and nozzles in a ring on the front side. Fan guns does not require the supply of compressed air, but has their own built-in piston compressor instead. The energy consumption is high, and a supply of electrical power is required to run the fan and the compressor. These snowmakers are quite heavy, and not convenient to move, but can be rotated 360° and has a long throw. Fan guns are very popular today, especially for alpine ski resorts.

Figure 5: An example of a lance [20].

Figure 6: An example a fan gun [20].

Although cryogenic snowmaking and hoar frost growth are other methods to produce snow artificially, their production rate and energy consumption make them inapplicable for skiing purposes. A summary of the four types can be seen in Table 1.

Table 1: Summary of the different types of TDSs.

Type	Requirements	Advantages	Disadvantages
Air/water snowmaker	Compressed air and water	High production rate.	High energy consumption and noise level.
Lance	Compressed air and water	Low energy consumption. Low noise level.	Not the best production rate. Sensitive to wind.
Waterstick	Compressed water	Very low energy consumption. Very low noise level.	Only for cold temperatures. Low production rate. Sensitive to wind.
Fan gun	Compressed water and electricity.	Very high production rate. Good in marginal temperatures.	High energy consumption. Heavy, not easy to move.

The rest of this thesis will focus on fan guns and lances, which dominate the market of TDSs today.

2.1.4 Cooling towers

Among the main factors affecting snowmaking conditions, the water temperature is easiest to modify. The supply water should be as cold as possible, without freezing, to enhance the production rate of snow. Water at temperatures above 3 °C should be considered cooled [21]. Cooling towers are the most common way to cool the water in a snowmaking system. In Figure 7, a typical cooling tower is illustrated. *Warm water* is supplied and sprayed over a lattice leading down to a cold basin at the bottom. Cold air is blown by a fan through the tower in the opposite direction, contributing in cooling the water. As the water passes through the tower, some of the water evaporates leading to the cooling of the remaining water. It is important that the water does not freeze, and the fan speed is used to regulate the amount of cooling. The *cold water* is eventually supplied to the snowmakers [11].

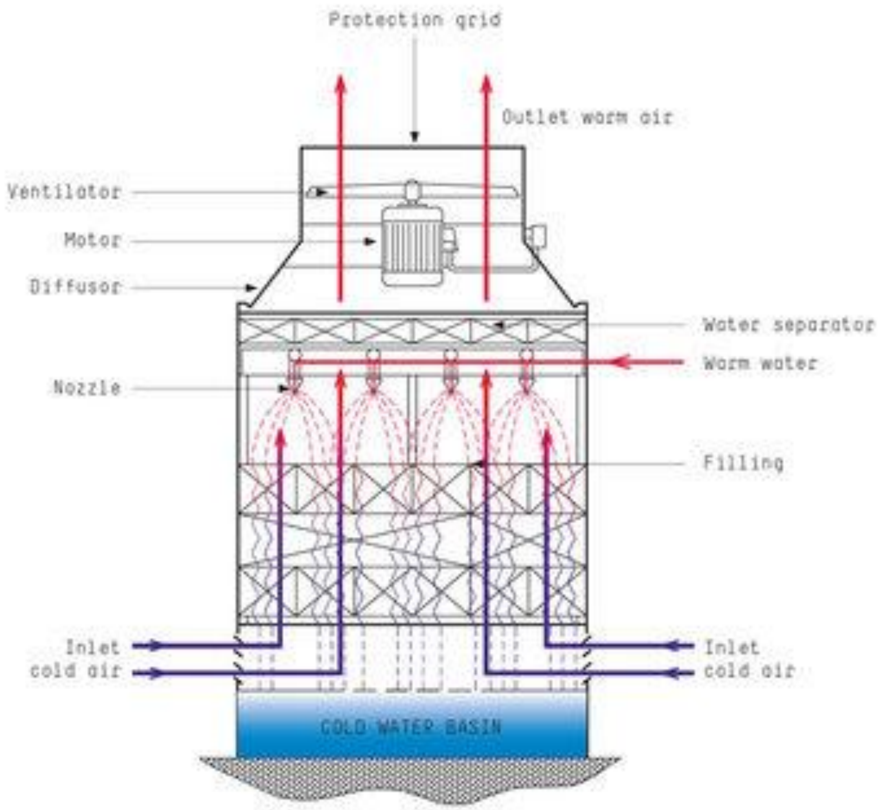


Figure 7: A cross-section view of a cooling tower [22].

2.1.5 Modern snowmaking systems

More than 1.500 snowmaking systems are in operation around the world today, approximately 500 of them in Europe [7]. A modern snowmaking system consists of snowmakers, water supply, water pumps, air compressors as well as connecting infrastructure and sensors and possibly cooling towers. The system is fully automated, and controlled through a software to start the production once the weather conditions allow it, such that the pockets of cold weather are utilized. Furthermore, automation leads to a reduced demand for labor.

The water supply can come from either a dam, a river or tap water if no natural source exists. The water infrastructure is divided into a low-pressure side and a high-pressure side. The low-pressure side consists of pumps for water extraction from the water source, followed by a centralized pumping station that boosts the pressure to the snowmakers on the high-pressure side. It is an advantage, especially for alpine ski resorts, to extract the water and place pumping stations as high in the mountain as possible. For every 10 m of altitude, the water pressure will decrease with approximately 1 bar. Most snowmakers require a water pressure of at least 40 bar to utilize their potential. It is not unusual to boost the pressure up to 100 bar at the pumping station to take into account friction losses in the pipes and altitude differences [7]. Typical booster pumps used are horizontal centrifugal pumps, which can be connected in parallel to increase the flow rate, or in series to increase the pressure.

The compressor capacity is often the bottleneck of the snowmaking system. If lances are used, a centralized compressor station is preferred. The alternative is a stand-alone piston compressor on each lance.

2.1.6 Comparison between fan guns and lances

Lances are usually 6-9 m high to ensure sufficient air time for the droplets to freeze. Due to a short throw distance, lances are sensitive to wind. Fan guns have a longer throw distance, and are less sensitive to wind, but they have a higher noise level than lances. The head of the lance consists of nozzles, usually between 2-20. Most of them *water nozzles* and some of them *nucleation nozzles*. The nucleation nozzles spray very small water droplets which will freeze immediately in the air to form a seed. The water nozzles spray slightly larger water droplets which will freeze around the seeds. A typical configuration for a lance is 10 water nozzles and 2 nucleation nozzles. The nucleation nozzles should be placed below the other nozzles, as the heavier droplets will fall quicker. Fan guns can have more than 300 nozzles, with a typical configuration of 300 water nozzles and 45 nucleation nozzles, but the amount of nozzles varies greatly. With more nozzles, more regulating options are available. This is an advantage, especially if the snowmaker is automatic, which means that it can regulate the flow through the nozzles based on the weather conditions. In marginal temperatures it is beneficial to lower the water flow to ensure good snow quality, while the water flow can be increased in colder temperatures to increase the production rate. Small amounts of heat are used to prevent the nozzles from freezing, and for this reason also lances are normally supplied with electricity.

Figure 8: Examples of lance heads. Nucleation nozzles are at the bottom row [23]

Figure 9: A fan gun seen from up front. The fan can be seen in the back, and the nozzles in the front. The outer ring of nozzles are the nucleation nozzles [24].

Technical data provided by some of the manufacturers of fans guns and lances are given in Table 2 and can be used to compare the two types. The most significant difference is that the production rate is approximately twice as high for fan guns than it is for lances. The price of a fan gun starts at 190.000 NOK, while a lance can be bought at around 20.000 NOK [25].

*Table 2: Technical specifications for some existing snowmakers. *The power supply excludes pumps and centralized compressor stations [23] [26] [27].*

Model	**Titan 2.0, Demaclenko**	**Visup 4, Demaclenko**	**Peak, Sufag**	**Taurus 2.0, Sufag**
Type	Fan gun	Lance	Fan gun	Lance
Production rate	2.520 m^3/day	1.296 m^3/day	2.304 m^3/day	1.344 m^3/day
Power supply*	24,5 kW	-	23,4 kW	-
Start-up T_W	-	-	-	-2 °C
Water flow rate	11 l/s	5,4 l/s	-	-
Water pressure	8-50 bar	15-50 bar	8-40 bar	15-60 bar
Air consumption	-	245 l/min	-	400-600 l/min
Throw	75 m	-	40 m	-
Weight	763 kg	212-280 kg	690 kg	75-150 kg
Mode	Automatic	Automatic	Automatic/manual	Automatic
Water nozzles	80	15	310	8
Nucleation nozzles	12	5	45	4

A model of the average production potential for fan guns and lances as function of the T_W is developed in equation (1) and (2) [28]. The model is based on various snowmakers, and is valid at a water temperature of 2 °C and a water pressure of 25 bar.

$$PPF = -4{,}83T_W + 3{,}94 \qquad (1)$$

$$PPL = -3{,}94T_W - 4{,}24 \qquad (2)$$

Where PPF and PPL is the production potential in m^3/hr for fan guns and lances respectively. Equation (1) and (2) are valid in the range:

$$-13\,°C \leq T_W \leq -2\,°C$$

At wet-bulb temperatures colder than -13 °C, the production potential is assumed to remain constant. The production potential is visualized in Figure 10. The average production potential for fans guns and lances in Figure 10 is 1.091 m^3/day and 712 m^3/day respectively.

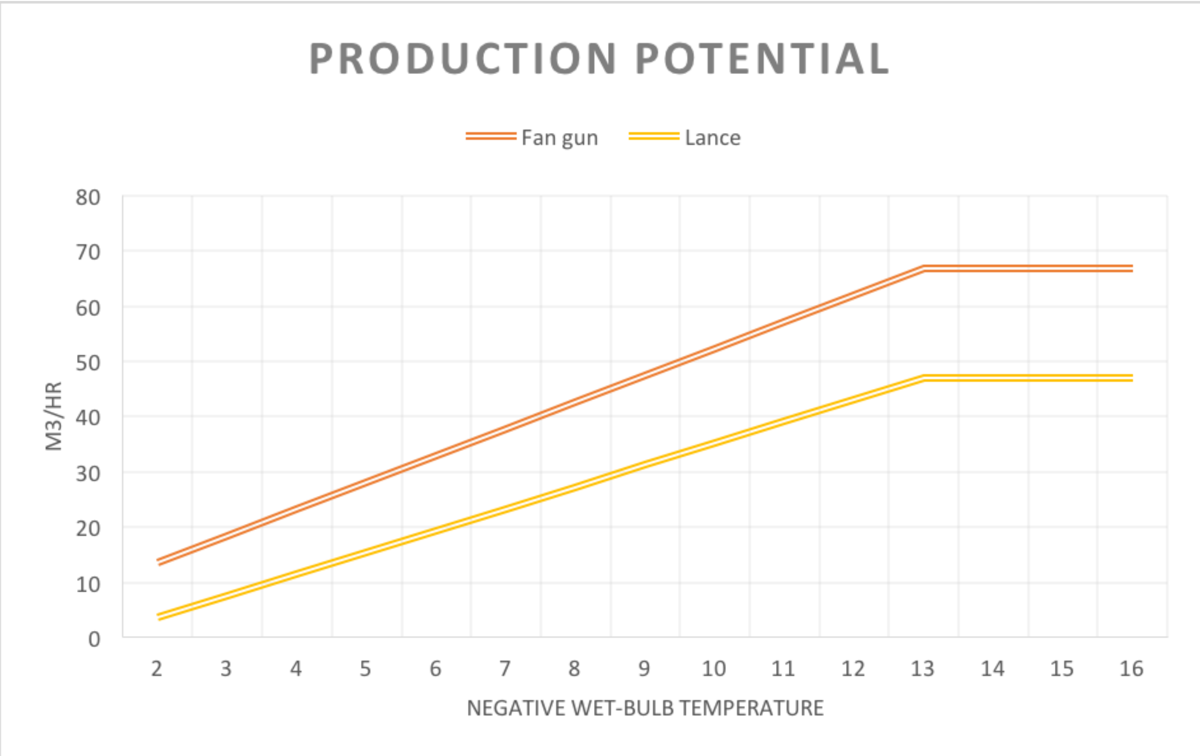


Figure 10: Production potential for fan guns and lances as a function of the T_W.

Recall that these relations are based average values of various snowmakers, and deviations occur. For example, the maximum production rate for the fan gun Titan 2.0 is 2.520 m^3/day (Table 2), while the production potential for a fan gun in Figure 10 is 1.600 m^3/day.
Next, the EVR is examined. The total power supply to fan guns, P_F, and lances, P_L, is the sum of the following:

$$P_F = P_{Compressor} + P_{Fan} + P_{Heating} + P_{Water\ pump} \quad (3)$$

$$P_L = P_{Compressor} + P_{Heating} + P_{Water\ pump} \quad (4)$$

Apart from $P_{Water\ pump}$, the components are assumed to be constant and are listed in Table 3.

Table 3: Power supply for typical fan guns and lances. The numbers are based on average values from technical data given by various manufacturers.

	Fan gun	**Lance**
$\mathbf{P_{Compressor}}$	4 kW	3 kW
$\mathbf{P_{Heating}}$	2 kW	0,5 kW
$\mathbf{P_{Fan}}$	17 kW	-

The power supply to the water pump is given by:

$$P_{Water\ pump} = \frac{\dot{V}\Delta P}{\eta} \quad (5)$$

Where the volumetric flow rate depends on the production rate. The pressure difference through the pump, is set to 40 bar, and the pump efficiency which accounts for pressure drops in the pipes as well as a temperature rise of water, is set to 0,65 [7]. Based on equation (1)-(5), the EVR of fan guns and lances can be calculated as a function of the T_w and the results are displayed in Figure 11. The average EVR is 1,42 and 0,98 kWh/m^3, for fan guns and lances respectively.

When selecting a snowmaker, it is important to look at the conditions it is to be used at. A higher lance will give more air time, but it will also be more sensitive to the wind. A longer barrel and a more powerful fan will give a longer throw, but it will also have a higher noise level and be more energy demanding. With more nozzles, more regulating options exist, and the snowmaker will run better in marginal temperatures. Generally, a fan gun is preferred in marginal temperatures, while a lance is preferred in colder climates with narrow tracks or slopes [9].

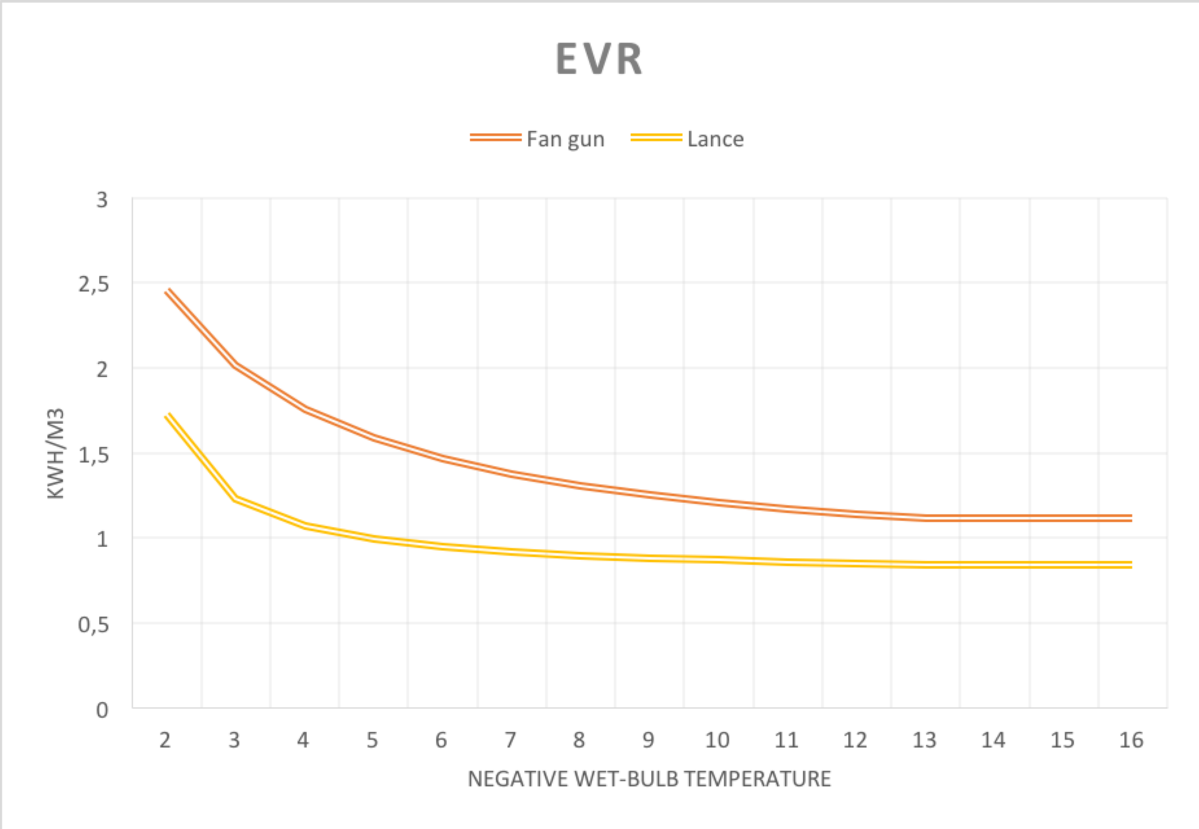


Figure 11: EVR of fan guns and lances as a function of T_W.

2.1.7 Cost examples

A typical distribution of the costs associated with a snowmaking system is given in Table 4.

Table 4: Typical distribution of investment costs [19] and operating costs [18] in a snowmaking system.

Investment costs		**Operating costs**	
Pipes and fittings	40%	Electricity	48%
Air compressors	30%	Personnel costs	33%
Water pumps	15%	Maintenance and transport	19%
Snow machines and hoses	5%		
Hydrants, regulators and valves	5%		
System engineering	5%		

Depending on the demand for for snow, the snowmaking costs are typically in the range of 15-25% of the total expenses in a ski arena/resort [19]. The distribution of the operating costs in Table 4 is based on numbers from Idrefjäll, Sweden, which is a large ski resort. The investment costs of a snowmaking system at a nordic ski arena in Tonstad, Norway, were estimated to be around 3,2 MNOK [29]. In this case, a 2 km track was to be filled with snow

2.2 Temperature independent snowmaking

A temperature independent snowmaker (TIS), is a snowmaker that can produce snow at above 0 °C. The first TIS was manufactured in 1993, and today there are at least four known manufacturers of such snowmakers. However, only a few ski arenas/resorts have invested in these products, which can be both stationary and mobile. The snow from these machines is not snow in its natural form, but actually small ice particles. To explain temperature independent snowmaking, traditional ice production has to be described first.

Ice has been used for cooling purposes for several millenniums. Natural ice formed on lakes and dams was cut into 200-300 kg blocks and stored in insulated warehouses [30]. Today, ice production has outdistanced the harvesting of natural ice. Block ice, flake ice, tube ice, plate ice, cube ice and ice slurry are examples of ice produced by ice machines. These types of ice can be divided into two subgroups, dry subcooled ice or wet ice [31]. Generally, subcooled ice is produced in machines that mechanically remove the ice from the cooling surface. Wet ice is usually made in machines that use a defrost procedure to release the ice. The defroster partially melts the ice at the cooling surface, allowing the ice to loosen. In some machines, the ice is formed and collected to produce an ice slurry, which contains much more water than other types of wet ice. Flake ice, plate ice and ice slurry and are the types of ice in existing TISs.

Flake ice is collected as dry subcooled flakes that typically can be up to 3 mm thick and between 100 and 1.000 mm^2 in size [31]. A schematic diagram of a flake ice machine can be seen in Figure 12. Water is sprayed down from several sprinklers onto the inside surface of a cylindrical container, which operates as the evaporator in the system, resulting in ice on the surface. In some models, a drum-shaped cylinder rotates and the scraper on the outer surface remains stationary, as is the case in Figure 12. In others, the scraper rotates and removes the ice from the inner surface of a stationary drum. Usually, the rotation is in the vertical plane, but some models have rotation in the horizontal plane. Immediately before the scraper, no water is added and the temperature of the ice reduces to a subcooled temperature [32]. This ensures that only dry subcooled ice falls into the storage space below the scraper.

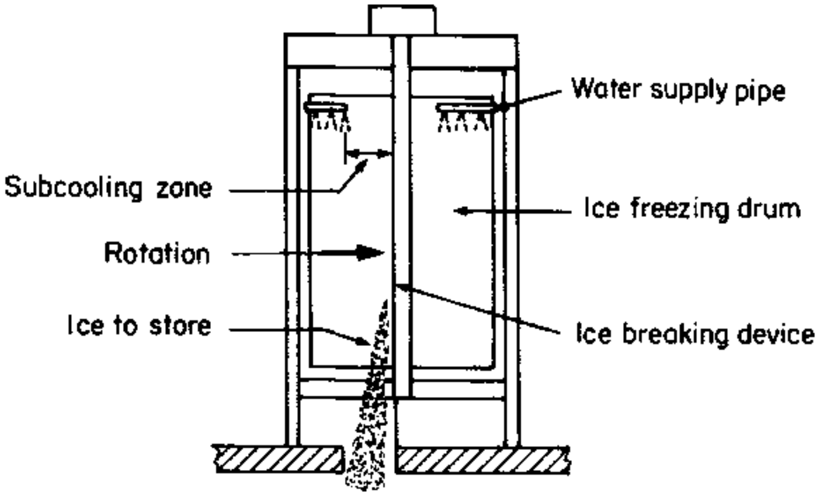


Figure 12: A schematic diagram of a flake ice machine [31].

One advantage of the rotating drum method is that the ice-forming surface and the ice scraper are exposed, so the operator can easily observe if the plant is operating as designed. In contrast to machines with rotating scraper, these machines require a rotating seal on the refrigerant supply and return pipes. This can be a weak spot, but in modern machines, the seal has a high degree of reliability.

Plate ice is formed by spraying water over the face of a refrigerated vertical plate. The plates are released by running water on the other faces of the plates to defrost them. Other types of machines form ice on both surfaces and use an internal defrost process, illustrated in Figure 13. The harvesting time of the ice may vary, depending on the operating conditions, but is normally between 8-10% of the total cycle time. Multiple vertical plates are arranged to form the plate ice machine, and the capacity of the machine can be adjusted by adding or removing more plates. An ice crusher is required to break the ice into a suitable size for storage and usage [31].

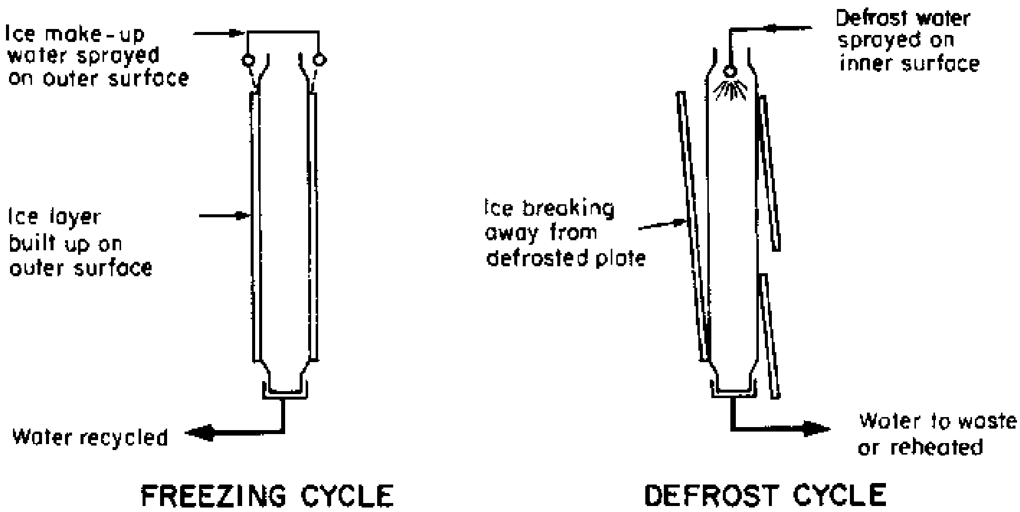


Figure 13: The working principle of a plate ice machine [31].

Ice slurry is a mixture of ice particles and a liquid. The size of the ice particles varies between 0.1-1 mm in diameter [33]. The liquid can be pure water or freezing point depressants widely used in the industry today, such as sodium chloride, ethanol, ethylene glycol and propylene glycol [34]. Compared to other types of produced ice, ice slurry has good heat transfer performance when releasing the latent heat of fusion, because of the large heat transfer surface area created by the numerous small ice particles [34]. There are many methods of produce ice slurry, and some of them are described in the following sections.

2.2.1 Vacuum ice machine

Figure 15 shows a schematic diagram of a vacuum ice machine. The system consists of a vacuum freeze evaporator, a compressor, a vacuum pump and a condenser [35]. The operating principle of the vacuum ice machine is to bring water close to its triple point conditions, where the vapor pressure is 611 Pa, which is less than 1% of the atmospheric pressure [36], and the temperature is just above 0 °C. At the triple point, water can exist as gas, liquid and solid. Figure 14 shows the different phases of water in a PT-diagram. When water is brought to triple point conditions, some of the water will evaporate. The energy used to evaporate the water causes the temperature in the water to decrease. Eventually it will freeze, and create an ice slurry. The latent heat of fusion and vaporization of water is 334 kJ/kg and 2.500 kJ/kg [36] respectively, resulting in the mass of ice produced being approximately 7,5 times the mass of water vapor.

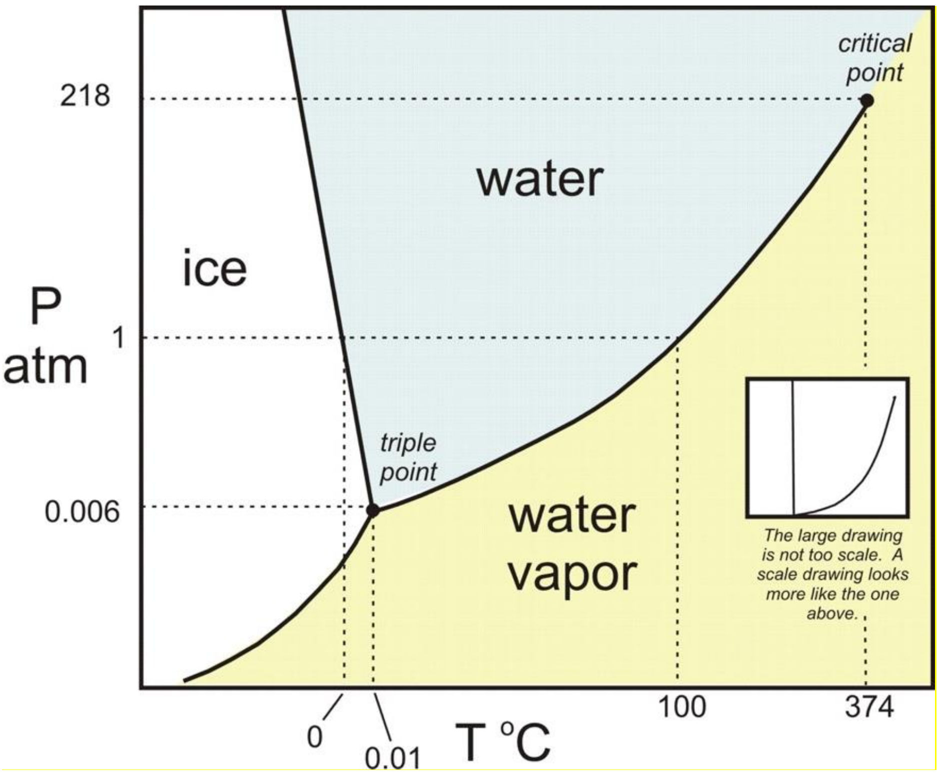


Figure 14: Phase diagram for water [37].

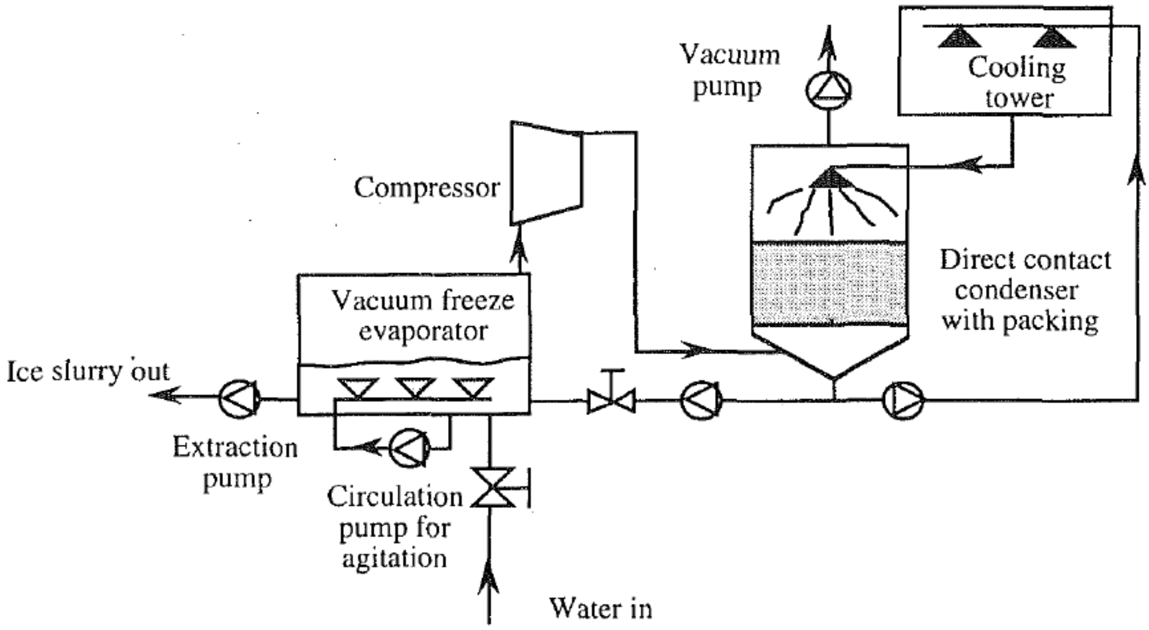


Figure 15: A Schematic diagram of a vacuum ice machine [36].

The low operating pressure results in a very large specific volume of the gas. Therefore, the compressor needs to handle a large amount of gas, which will influence the size of the compressor. In addition, the need for large pressure ratios, makes the application of standard compressors expensive [36]. The low operating pressure of the evaporator results in small aerodynamic forces on the internal compressor components, which makes it possible to build a lightweight construction of composites, which can lower the investment costs. An example of a centrifugal compressor successfully implemented in a vacuum ice machine is 2,6 m in diameter, and have titanium alloy blades that are only 1,5 mm thick [36] [38]. Next, the compressed vapor condenses, after which it is being injected back into the evaporator. A circulation pump is installed in the evaporator in order to agitate the slurry. Without agitation, the specific freezing capacity and crystal quality will be poor [39]. The ice slurry is continuously removed from evaporator and collected in a tank, where ice and water can be separated.

2.2.2 Direct heat exchange

In direct heat exchange ice slurry generators, a refrigerant is injected directly into water. The refrigerant is sprayed through a nozzle and starts to evaporate. This will cool the water until ice is formed [40]. The evaporated refrigerant is collected above the water surface, led into a compressor and expanded, in order to be sprayed into the water column again [41]. Due to buoyancy, the ice formed tends to move to the top of the tank. This ice fraction can be up to 40% [40]. When designing a direct heat exchange ice slurry generator there are several practical and operational aspects that has to be taken into account. For example, the refrigerant droplets should not be trapped in the ice slurry, as it can pollute the ice slurry. Thus, the ice slurry formed must separate easily from the refrigerant. Also, no water should enter the cooling circuit due to the risk of clogging [40].

The position of the nozzle injecting the refrigerant is important for the quality of the ice slurry. If the nozzle is placed in position (i) in Figure 16, there is no contact between the water and the nozzle, which removes the risk of water freezing around the nozzle. However, the spray of refrigerant has to penetrate the slurry to reach the bottom of the tank, which may break the slurry apart. In position (ii), the nozzle is placed below the slurry, and the direct contact between the nozzle and the water may lead to blockage of the nozzle due water freezing. In the last position (iii), at the bottom of the tank, the slurry and the refrigerant separate easily because they move in opposite directions. Here, the risk of blockage of the nozzle is minimized, as there is only intermittent contact between the water and the nozzle [40]. The nozzle is therefore normally placed at the bottom of the tank [41].

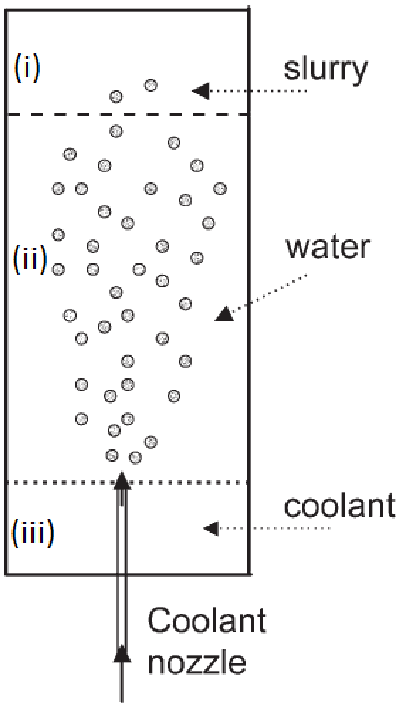


Figure 16: Different nozzle positions in a direct heat exchange ice slurry generator [40]. (Coolant = refrigerant)

2.2.3 Scraped surface ice slurry generator

The scraped surface ice slurry generator is currently the most technologically developed and widely accepted ice slurry generation method over the last 20 years [42]. Typically, the scraped surface ice slurry generator is a circular shell-and-tube heat exchanger through which an evaporating refrigerant flows, as can be seen in Figure 17. Water flows through the inside space, bounded by the inner cylinder. The ice is created on the walls of this inner cylinder, and are removed by rotating scrapers or knives, so that the ice falls into an accumulator. The scraped surface generator has a large surface for ice creation, and is therefore used when high ice production rates are required. It is possible to use scrapers made of both metals and polymers.

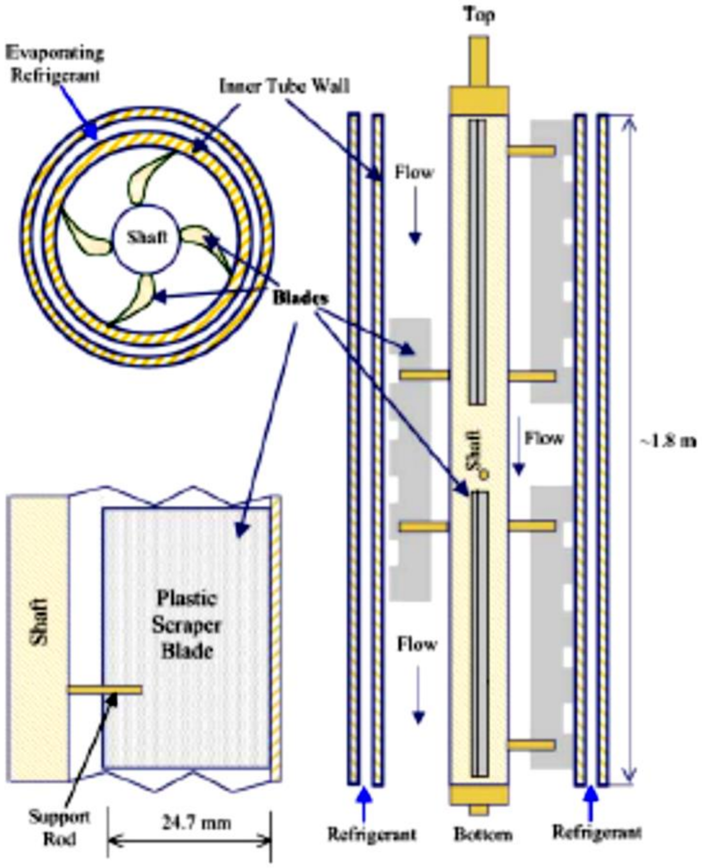


Figure 17: A schematic diagram of a scraped surface ice slurry generator [42].

In other types of scraped surface generators, the crystals are produced in tubes and removed by turning screws [41] [43]. Scraped surface generators are quite expensive and have high maintenance costs [44]. The removal of ice is required to prevent formation of a thick ice layer on the walls. If the thickness grows too big, an additional thermal resistance will be introduced, lowering the heat transfer rate. Because of continuous accumulation of ice on the inner walls, the scraper blades will eventually be blocked, causing the ice generator to freeze. To prevent this, solutes are added to lower the freezing point of the solution. In order to increase the heat transfer rates, a turbulence in the ice slurry flow is mechanically induced. This is done by the rotating scraper blades, which facilitates the production of a homogeneous ice slurry mixture [42].

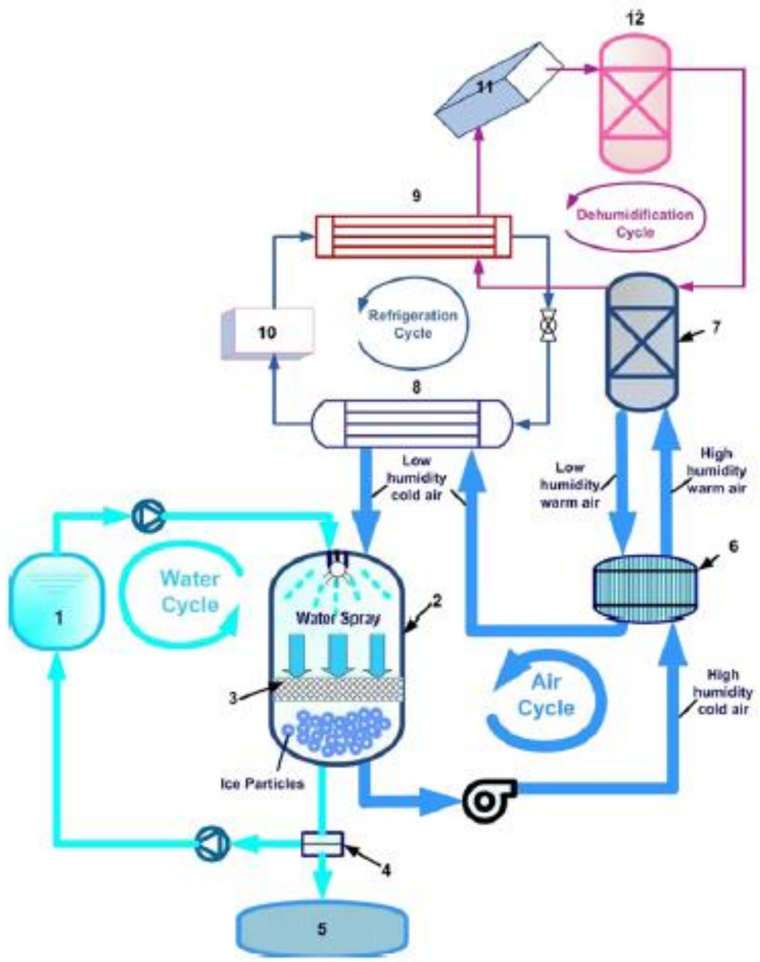
2.2.4 Supercooling

Ice slurry production by supercooled water is a concept where water flows into an evaporator, typically designed as a shell-and-tube heat exchanger [43], which cools the water to below the freezing point without crystallization [44]. After leaving the evaporator, the water is disturbed to initiate crystallization. It is important to control when the crystallization starts, in order to adjust the concentration of the produced ice. If the freezing starts too early, the system may clog [45]. There exist some methods to prevent ice blockage. If the water is supercooled in an open space, the risk of blockage reduces [45]. By experiments, it has been shown that it is impossible to avoid crystallization inside the evaporator having a degree of supercooling of more than 2 K, and it is therefore necessary to precisely control the evaporation temperature. To ensure that no ice is created inside the evaporator, the system should include a defrosting system. When implementing this type of ice slurry generator in a real installation, the system requires a geometry that does not initiate crystallization [44]. These types of generators produce ice slurry of low ice fraction, so ice concentrators must be applied in addition. They usually contain a sieve and a pump to remove the liquid, resulting in an increased ice fraction. Due to the difference in density between the fluid and the solid phases, a centrifugal technique can also be used for the separation process [41].

2.2.5 Dehumidification

Water always strive to be in equilibrium between the phases: liquid-vapor, liquid-solid or solid-vapor. For every liquid temperature, there is a corresponding equilibrium vapor pressure. If the vapor pressure is less than this, water will continue to evaporate to establish this pressure. Just as a temperature difference will cause a heat flux to bring the temperature to an equilibrium, so will water evaporate in order to match the vapor pressure. A low water vapor pressure could be obtained by decreasing the RH. This shows that it is not necessary with a vacuum to create a proper environment of low vapor pressure for evaporative freezing [45].

Water just above 0 °C sprays and evaporates in a low RH, and thus low vapor pressure, chamber. The evaporative cooling leads to lower temperature in the remaining water, resulting in supercooled water. The supercooled water is then physically disturbed, and some of the water crystalizes to produce an ice slurry. After the evaporation, the RH of the air increases, and a dehumidification cycle reproduces air of low RH. The refrigeration cycle cools the dry air, regulating the temperature of the air, while the waste heat from the condenser is reutilized in the dehumidification cycle. Therefore, the refrigeration system has a double effect so that electric power can be saved [46]. The system is illustrated in Figure 18.



1. Water Tank; 2. Ice-producing Chamber; 3. Supercooled state Releaser;
4. Ice-water separator; 5. Ice Storage Tank; 6. Heat Exchanger; 7. Dehumidifier;
8. Evaporator; 9. Condenser; 10. Energy Input Module;
11. Energy Supplement Module; 12. Regenerator.

Figure 18: A schematic diagram of the dehumidification ice making system [45].

2.3 Manufacturers of TISs

In the following sections comes an overview of the four known manufacturers of TISs. The most significant differences between the machines is the technique used for ice production and whether the machine is stationary or mobile. The only requirements are supply of water and electricity.

2.3.1 Flake ice based TISs

Flake ice is the most commonly used snow substitute [47]. TechnoAlpin AG from Italy released *the Snowfactory* in 2014. The Snowfactory produces flake ice, which is crushed to finer ice particles and delivered to the ambient by a conveyor or a fan. The Snowfactory was for example installed at Sjusjøen and Geilo, Norway, in the summer of 2015, where the ski tracks opened in the end of September 2015. There are different versions of the Snowfactory, both mobile and stationary, with different capacity and size.

Figure 19: The Snowfactory model SF220 from TechnoAlpin [48].

2.3.2 Plate ice based TISs

SnowMagic Inc. from USA was the first manufacturer to offer a TIS in 1993. SnowMagic applies a plate ice machine as the source of snow. After the ice is released from the plates, it is sent to an ice crusher to create smaller particles. SnowMagic use a patented technology to make even finer particles of ice after the ice crusher, seen in Figure 20. Four models exist with different capacity.

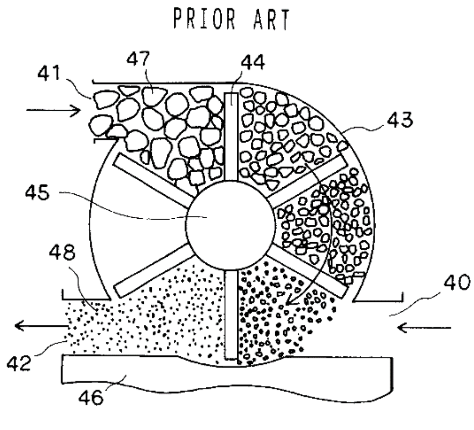


FIG. 8

Figure 20: A sketch of the ice crusher from SnowMagic. Ice is fed into an apparatus with a high-speed rotor blade that beats and crushes the ice [49].

2.3.3 Ice slurry based TISs

The Israeli company IDE Technologies Ltd. developed their first stationary TIS in 2005, and in 2013, they released the mobile *VIM100 Snowmaker2go.* IDE technologies utilize ice slurry produced in a vacuum ice machine. The slurry is then separated into ice and water is in a snow concentrator. A schematic diagram of the VIM100 Snowmaker2go can be seen in Figure 21.

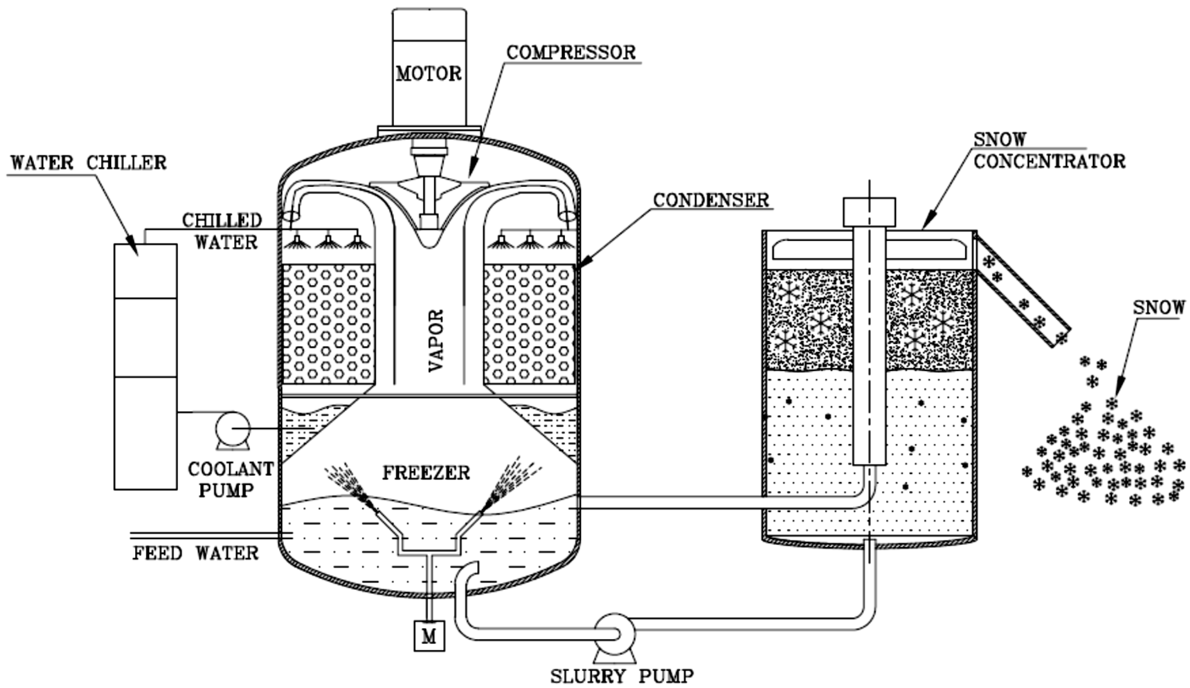


Figure 21: A schematic diagram of the vacuum ice machine from IDE technologies [50].

Another manufacturer that applies ice slurry is SnowTek from Finland. Their machine, *SnowGen*, utilizes a scraped surface ice slurry generator to produce the slurry, before the ice and water is separated. The SnowGen, which is stationary, produced snow for the disciplines ski jumping and nordic combined in the 2014 Olympic Games in Sochi, and can be seen in Figure 22.

Figure 22: SnowGen from SnowTek [51].

2.3.4 Overview and comparison

The following tables contain technical data of TISs:

Table 5: Technical data of TechnoAlpin's TISs, assumed a water temperature of 5 ℃ and an air temperature of 15 ℃. Higher temperatures will increase the power supply and decrease the amount of snow produced [52].

TechnoAlpin	**SF100 (2014)**	**SF220 (2014)**
Principle	Flake ice machine	Flake ice machine
Type	Mobile	Stationary
Power supply	130 kW	227 kW
Snow density	450 kg/m^3	450 kg/m^3
Production rate	100 m^3/day	220 m^3/day
Water flow rate	0,8 l/s	1,5 l/s (129.600 kg/day)
Refrigerant	R404A (HFC)	R717 (Ammonia)
Size	1 x 40' container	2 x 40' containers
EVR	31,2 kWh/m^3	24,76 kWh/m^3
Water pressure	2-4 bar	2-4 bar
Customers	The German Ski Federation, Geilo, Idrefjäll	Winterberg, Sjusjøen

Table 6: Technical data of SnowMagic's TISs at 21,7 ℃ [49] .

SnowMagic	SnowMagic 50	SnowMagic 100	SnowMagic 150	SnowMagic 200
Principle	Plate ice machine	Plate ice machine	Plate ice machine	Plate ice machine
Type	Mobile	Mobile	Stationary	Stationary
Power supply	151 kW	248 kW	362 kW	545 kW
Production rate	100 m^3/day	200 m^3/day	300 m^3/day	400 m^3/day
Water flow rate	0,6 l/s	1,2 l/s	1,7 l/s	2,3 l/s
Size	40' container	40' container	40' container	40' container
EVR	36,24 kWh/m^3	29,76 kWh/m^3	28,96 kWh/m^3	32,7 kWh/m^3

*Table 7: Technical data of IDE technologies' TISs, assumed a water temperature of 4,5 ℃. An increase in the water temperature with 1 ℃ will decrease the amount of snow produced by approximately 1,5% [50] [53]. *The power supply does not include the cooling system.*

IDE Technologies	VIM850 all weather snowmaker (2005)	VIM400 all weather snowmaker (2009)	VIM100 Snowmaker2go (2013)
Principle	Vacuum ice machine (ice slurry)	Vacuum ice machine (ice slurry)	Vacuum ice machine (ice slurry)
Type	Stationary	Stationary	Mobile
Power supply	397 kW*	235 kW*	190 kW
Snow quality	Spring snow	Spring snow	High quality snow
Snow density	650 kg/m^3	650 kg/m^3	560 kg/m^3
Production rate	1.720 m^3/day	860 m^3/day	200 m^3/day
Water flow rate	12,9 l/s	6,5 l/s	1,3 l/s
Refrigerant	Water	Water	Water
Size	-	-	1 x 40' and 1 x 20' containers
EVR	-	-	22,8 kWh/m^3
Customers	-	Pitzal, Zermatt	-

Table 8: Technical data of SnowTek's TIS SnowGen [3].

SnowTek	**SnowGen (2014)**
Principle	Scraped surface ice slurry generator
Type	Stationary
Power supply	280 kW
Production rate	220 m^3/day
Water flow rate	1,4 l/s
Refrigerant	R717 (Ammonia)
EVR	30,55 kWh/m^3
Customers	Sochi

The different TISs are seen to have similar characteristics. Traditional ice machines have slightly lower EVRs than the TISs. However, ice crushers or air blowers for distribution of the ice are not included for these machines, which can explain the small difference.

The TISs can be compared with TDSs. The TIS with the lowest EVR, the VIM100, uses 22,8 kWh/m^3, while the average fan gun uses 1,42 kWh/m^3 (see section 2.1.6). Furthermore, producing 12.000 m^3 of snow to a 5 km track would take 60 days with the VIM100, while the TDS would need 11 days (at an average production rate of 1.091 m^3/day, see section 2.1.6). <u>Thus, the TIS uses 16 times more energy per m^3 of snow produced than the average TDS and the production rate is 5,45 times lower.</u> Note that the technical data of the TISs does not take into consideration the energy required for water supply, which makes the EVR of the TISs even worse. Moreover, the snow quality from a TIS is not as good as from a TDS. While snow from a TDS can remind of freshly fallen natural snow, snow from a TIS reminds more of a coarse wet snow which is common at the end of the season, see Figure 23. These remarks imply that a TIS should only be used when a TDS cannot be used as a consequence of the weather conditions, unless the residual heat can be utilized in an effective manner.

Figure 23: The picture on the left shows fresh snow out of the SF100. The picture on the right shows the snow after a few days in the tracks. Both pictures are taken at Idrefjäll, Sweden.

Another remark is that at least one of the TISs, the SF100, is using the refrigerant R404A which is a hydrofluorocarbon (HFC). HFCs are expected to be phased out by 2020 because of new legislations regarding the use of refrigerants with high GWP (Global Warming Potential) in refrigeration systems. Natural refrigerants, such as CO_2 and ammonia are more promising choices of refrigerants [54].

2.4 Indoor snowmaking

In the last 20 years, indoor ski halls have been built several places in Europe and Asia. Ski tunnels for nordic skiing exists in Finland, Sweden and Germany, and are planned in Norway, while halls for alpine skiing exist in many countries, even in hot regions such as Dubai. The ski halls offer shorter tracks with snow throughout the year. The snow is supplied from flake ice machines, vacuum ice machines or TDSs [47]. The following analysis will focus on TDSs as the source of snow.

The total refrigeration load associated with a ski hall can be divided into the following parts [55]:

- **Transmission load**
  Heat leakage through walls, floors and ceilings.
- **Internal load**
  Heat from snowmaking, people, machines, lighting, etc.
- **Infiltration load**
  Heat from air infiltration.
- **Equipment load**
  Heat from the refrigeration system, for example the evaporator fans and defrosting.

The snowmaking load is by far the largest contributor to the refrigeration load. Smaller snowmakers with a maximum water flow rate of 9 l/min are usually used indoor [16]. The snowmaking happens during the night, when the air temperature in the hall is cooled to below -6 °C. At daytime, the hall is kept at -1,5 °C [16]. Smaller droplet diameters, 50-100 μm, can be used when producing snow indoor compared to outdoor, due to the absence of wind for droplets to drift away with [16]. This results in a quicker freezing process.

Ski halls are heavily insulated and consists of air coolers, floor coolers and snowmakers. The floor coolers are made of an extensive pipework in the floor filled with glycol. The purpose of the floor coolers is to prevent the snow from melting, as well as creating a constructive metamorphism in the snow layer. As the floor temperature is colder than the air temperature, water vapor will be drawn from the air into the snow, where it will freeze and create a constructive metamorphism. Constructive metamorphism occurs in reverse in nature if the ambient air is colder than the ground.

An important feature of the air coolers, which are located in the ceiling, is to dehumidify the air. Air leaving conventional air coolers used in cold stores have a humidity close to saturation. This will cause a problem in a ski hall as evaporative cooling, which accounts for a large portion of the heat and mass transfer during snowmaking, cannot occur in saturated air. Ski halls have solved this with wider spacing between the fins on the cooling coils of the air coolers, which gives the leaving air a lower RH [16]. The explanation to this can be seen in Figure 24. The air between the fins can be divided in zones, where the air closest to the fins is cooled to the fin temperature, while the air furthest away from the fins will be the warmest.

This means that the air closest to the fins will have a lower dew point, and the air in the adjacent zones will give up moisture due to the vapor pressure difference between the zones. Hence, there will be a cold flow of air close to the fins, which is saturated, and a warmer flow of air in between the fins, which is dry. When these air streams mixes at the outlet, the result is a cold and relatively dry air.

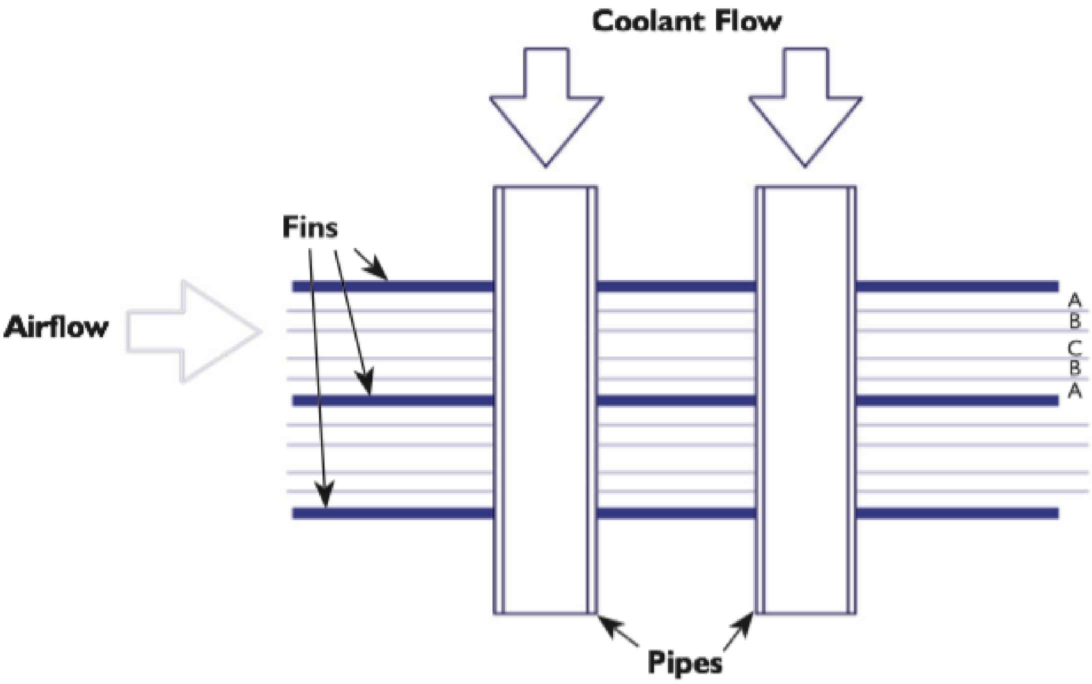


Figure 24: The airflow through a cooling coil is divided in zones between the fin. Zone A is colder than zone B, which is colder than zone C [16].

Another challenge with indoor snowmaking is that the air coolers have to be defrosted. As the air is close to saturation, some water will condensate and freeze at the cold surfaces of the air coolers. The air coolers should be defrosted for every 4-5 hours of operation [16].

2.4.1 Examples

A ski hall in Bottrop, Germany, has a snow surface of 19.200 m^2 and a total indoor volume of 150.000 m^3. The air temperature is held at -5 °C and the refrigeration plant has a total cooling capacity of 1.400 kW, driven by two rotary screw compressors [56].

A ski tunnel in Oberhof, Germany, has a total snow surface of 1.100 m^2. The air temperature is held at -3 to -4 °C, and the RH is between 80-100%. The refrigeration plant is driven by two rotary screw compressors with a total power of 620 kW. Ten air coolers and four snowmakers are located in the ceiling, which is 6-8 m high [57].

Calculations on a planned ski hall in Wittenburg, Germany, with a snow surface of 30.000 m^2 and an indoor volume of 1.800.000 m^3 estimated a required refrigeration capacity of 2.000 kW [47].

2.5 Snow storage

Storage of snow, or *snow farming*, is another approach to solve the lack of snow issue. Although it has been used for cooling purposes for a while [58], it was first tested for skiing purposes in Vuokatti, Finland, in 2004 [59]. Lately, the technique has been applied in several places well known for nordic skiing, such as Beitostølen, Østersund Seefeld, Ramsau, Canmore and Davos. There are four different ways to store snow: indoor storage, ground storage, pond storage and underground storage. In the latter case, no insulation is required. For ground and pond storage, a thermally insulating layer is placed on top of the snow pile. Other methods like water injection and compaction without insulation have also been tested, but gave no significant reduction of the melting rate [60]. A sketch of the different methods can be seen in Figure 25

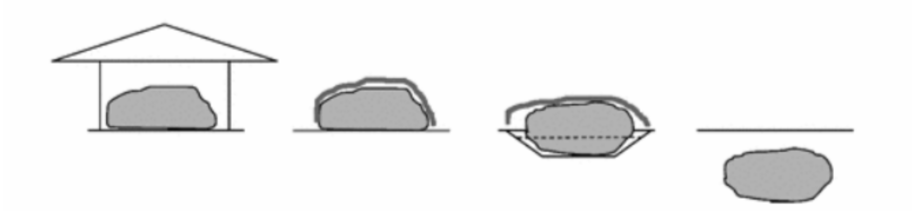


Figure 25: Snow storage methods from left to right: indoor storage, ground storage, pond storage and underground storage [58].

2.5.1 Snowmelt

The snowmelt can be divided into rain melt, ground melt and surface melt. The source of rain melt is the precipitation, while the ground melt springs from heat conducted from the ground. Regarding the ground melt, it is important that the ground has sufficient drainage, as well as a low thermal conductivity. Gravel is a good example. In addition, the ground can be thermally insulated to further decrease the ground melt. The surface melt originates from heat transfer through the insulating surface layer on top of the snow pile [61]. Calculations with wood chips, showed that the rain and ground melt contributed to less than 20% of the total losses from a 30.000 m^3 pile of snow [61]. This result implies that it is important to focus on the surface melt. The heat and mass transfer mechanisms through the insulating layer is seen in Figure 26.

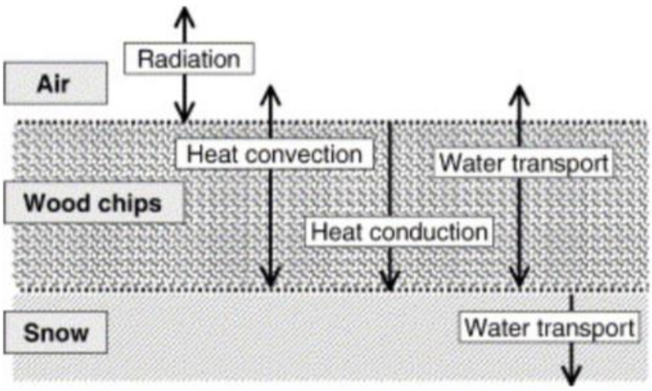


Figure 26: Heat and mass transfer through a layer of wood chips [61]

2.5.2 Cover materials

The key parameters identified to be important for a cover material is listed below [60]:

- **Radiative properties**
  The radiative properties can be divided between reflectivity (Albedo) and emissivity, both should be high to reflect incoming radiation.
- **Thermal conductivity**
  The thermal conductivity should be low to slow down the rate of heat transfer.
- **Permeability**
  Impermeable materials are not suited, as a lot of water will accumulate under the cover material. Entirely permeable materials are not suited either, as it will give a poor protection against the rain melt. Hence, semipermeable materials are a good trade-off.
- **Tensile strength**
  The tensile strength is of importance to withstand heavy wind and weather.
- **Surface roughness**
  If the surface roughness is too low, it will not stick to the snow and easily slip off. On the other hand, a very high surface roughness can lead to collection of dirt which might worsen the albedo.
- **Thickness**
  The thickness of the cover material will influence the insulating ability. Generally, the thicker the better, but the performance seems to stagnate, especially for geotextiles [60].

With respect to the key parameters listed above, wooden materials such as sawdust, cutter shavings, wood chips and bark, as well as geotextiles are examples of well suited cover materials, which all have been tested. The main advantages with wooden materials, is the evaporative cooling effect and the fact that they move with the snow. Water in soaked wood will evaporate, leading to evaporative cooling, and as the latent heat of vaporization of water is around 7,5 times the latent heat of fusion, a small portion of evaporation will lead to large savings in terms of melting losses. Furthermore, wooden materials, except from bark, have a high albedo that will reflect most of the incoming radiation. Sawdust or cutter shavings are considered to be the best cover materials, due to the dense structure with little room for air gaps. Results from a study showed that 0,1 m of cutter shavings was as effective as 0,2 m of sawdust [13]. The drawback with wooden materials is that they are expensive and that the materials should be replaced or sent for drying every three years of operation for maximum performance [61]. Geotextiles are cheaper, easier to transport and easier to handle, but experiments have shown that wooden materials are more effective cover materials.

2.5.3 Examples
All of the following examples have used ground storage, unless otherwise stated.

Experiments with geotextiles as the cover material were conducted at a glacier in the Alps in 2004-2005. The natural melting rate was decreased by 60%, using a 4 mm layer of geotextile. The experiments also showed that a double layer of geotextile further decreased the melting rate with around 10%, whereas a triple layer gave no further reduction [60].

Östersund, Sweden, has stored snow for a couple of seasons. In 2006, 20.000 m^3 of snow was stored. The snow was stored under a 0,7-0,8 m layer of sawdust, and 40% of the snow melted during the summer [13].

In 2008, 2.500 m^3 of snow was stored in two piles in Davos, Switzerland. One pile was covered with geotextile, and the other was covered with sawdust. The results were significant: 25% of the snow melted under the sawdust, but as much as 80% melted under the geotextile [62]. The costs of the storage project were estimated to be 130 NOK/m^3, 12% from snowmaking, 35% from cover material and transport, 35% from covering and decovering and 18% for distribution and preparation [63].

In Beitostølen, Norway, snow storage covered by a 0,4 m layer of sawdust has been done since 2012. 13.000 m^3 of snow was stored after the winter in 2012, and less than 25% melted during the summer.

Figure 27: Snow storage under sawdust at Beitostølen, Norway, 2013 [59].

Granåsen tested snow storage through the summer of 2015. 8.000 m^3 of snow was harvested and stored under 0,4 m layer of sawdust from April to November, with a total loss of 22% [64]. The result was a ski track of 1,7 km. The melting rate can be seen in Figure 28.

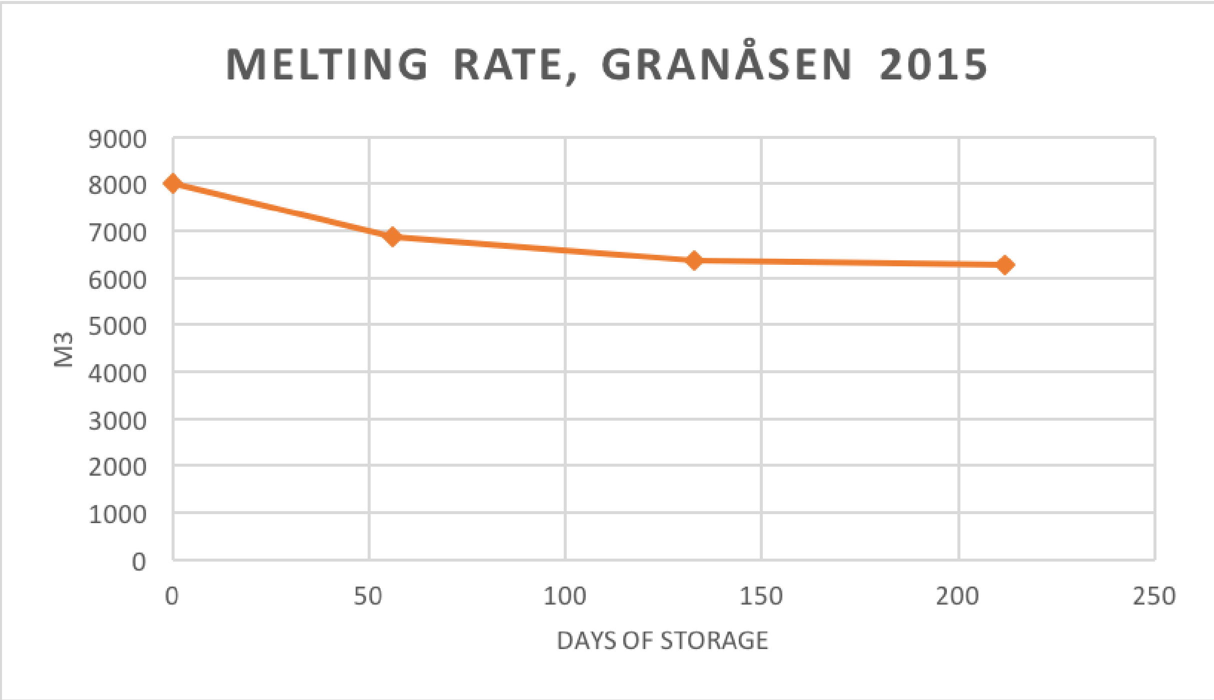


Figure 28: Snow storage in Granåsen from April 22 to November 20, 2015.

Storage of snow is also gaining attention in the field of cooling. A hospital in Sundsvall, Sweden, has stored 70.000 m^3 of snow each year since year 2000 for cooling purposes, reducing its energy consumption with 90% [59]. A similar project is planned at Oslo Airport Gardermoen, where the snow will be covered by a 10 cm layer of sawdust [65] [66]. These two projects are applying pond storage, with sawdust as the cover material.

2.6 Heat recovery from snowmaking

At snowmaking, there will always be a certain amount of surplus heat available. For snowmaking at below 0 °C this amount of heat is negligible, but above 0 °C, it can be remarkable. Existing TISs release this heat into the ambient air, for example through air cooled condensers. Instead of wasting this heat, it could be applied in a heat recovery system, either directly or indirectly. A direct heat recovery system would transfer the surplus heat directly (via a heat pump) to a heat demand. The alternative would be to store the heat in energy wells, or in a borehole thermal energy storage (BTES) system. This would be an indirect heat recovery system. An indirect system is necessary if the heat demand does not coincide with the production of surplus heat. For example, if the snow is produced during the autumn to prepare for the winter, while the heat demand is largest in the winter.

2.6.1 Borehole thermal energy storage

At depths of 10-15 m below the earth's surface, the temperature will not be affected by the air temperature, and from 20 m and downwards, the temperature is assumed to be constant during the year, see Figure 29. Further downwards, the temperature will increase by 1-3 °C for every 100 m. A BTES system is based on this principle.

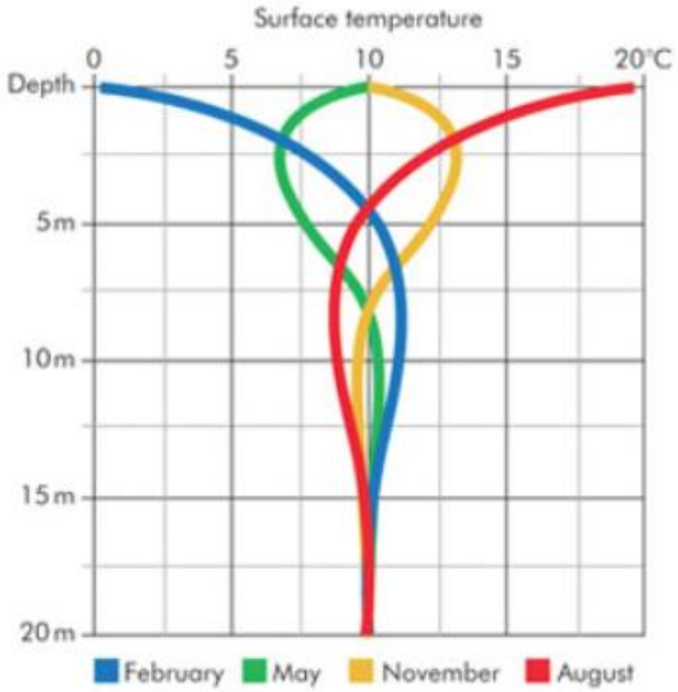


Figure 29: Temperature profile in the ground at different depths and seasons [67].

As opposed to a traditional ground source heat pump (GSHP), which only exchanges heat with the ground, a BTES system stores heat in the ground. A heat source, for example a TIS, can transfer surplus heat down in boreholes for storage in the ground until there is a demand for heating. Compared to a GSHP, which have a coefficient of performance (COP) of around 3,5, a BTES system can have a COP from 4-8 [68]. Typically, boreholes are drilled in a circular formation, seen from above (Figure 30). The inner part of this circle will be the warmest, as the temperature gradually decreases towards the periphery of the circle.

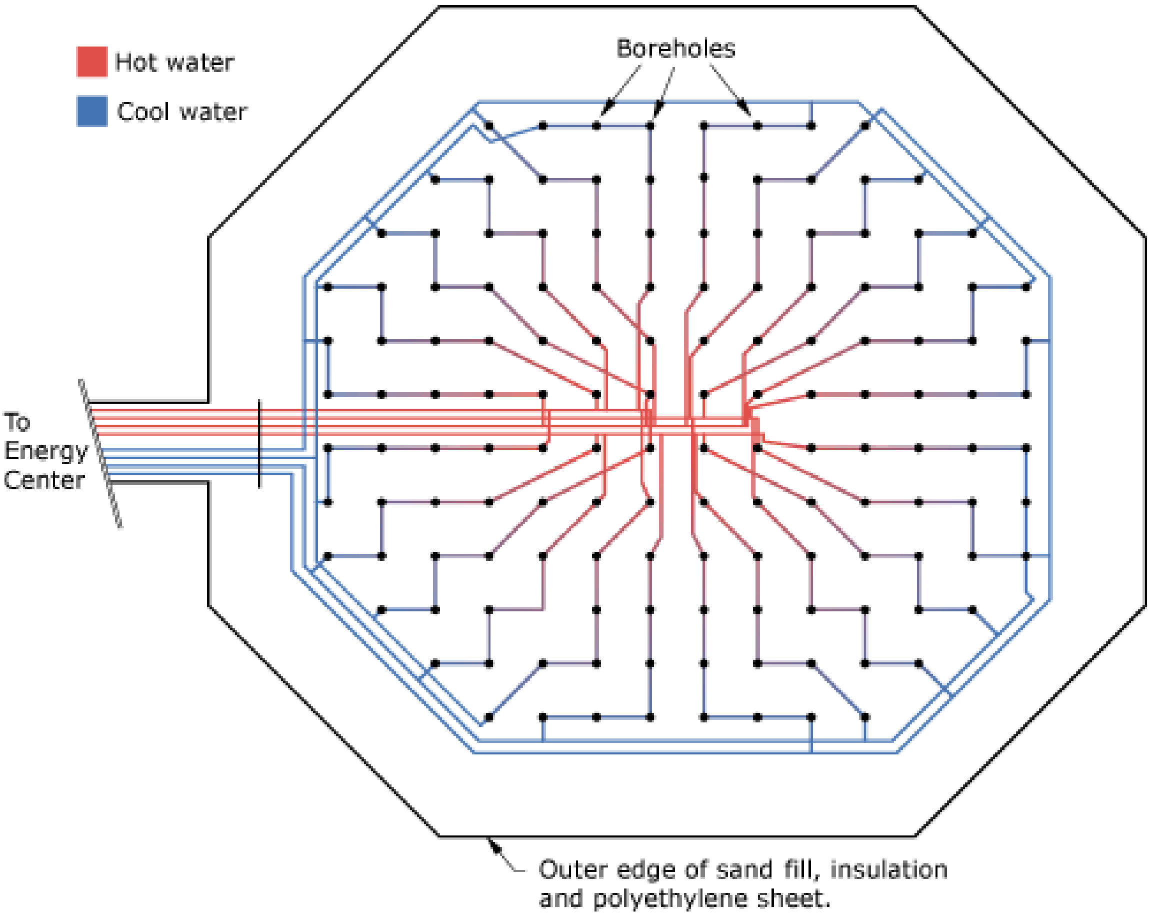


Figure 30: BTES working principle [69]

A plastic, usually U-shaped, pipe functioning as a borehole heat exchanger (BHE), is inserted in each borehole, and the hole is filled with a material that ensures good thermal contact with the ground [69]. In Scandinavia, it is normal to have air in the upper layer of the borehole, to minimize heat exchange with the ground near the surface [70]. The heat carrier fluid flowing in the U-pipes in the boreholes are usually water in combination with ethanol to avoid freezing.

The diameter of the boreholes varies from 100 to 140 mm, while the diameter of the U-pipes lay between 32-40 mm [70]. The number of boreholes, the distance between them and the depths have to be dimensioned such that the produced amount of heat can be stored without significant losses, in combination with keeping the investment costs at a minimum. A short distance between the boreholes, defined as 5-7 m, is obtainable in a balanced system, where the amount of heat ejected equals the amount of heat injected from the heat source. Otherwise, the distance between the holes have to be 15-20 m to avoid discharging. Hence, the area requirements of a BTES system is lower than for a GSHP. This fact also reduces the pump capacity required to pump the heat carrier fluid. Storage temperatures from 0-40 °C is called a low temperature storage system, and higher temperatures will make a high temperature storage system. A low temperature storage system has a lower temperature gradient, and hence lower losses, but a heat pump will be required to increase the temperature before utilizing the heat.

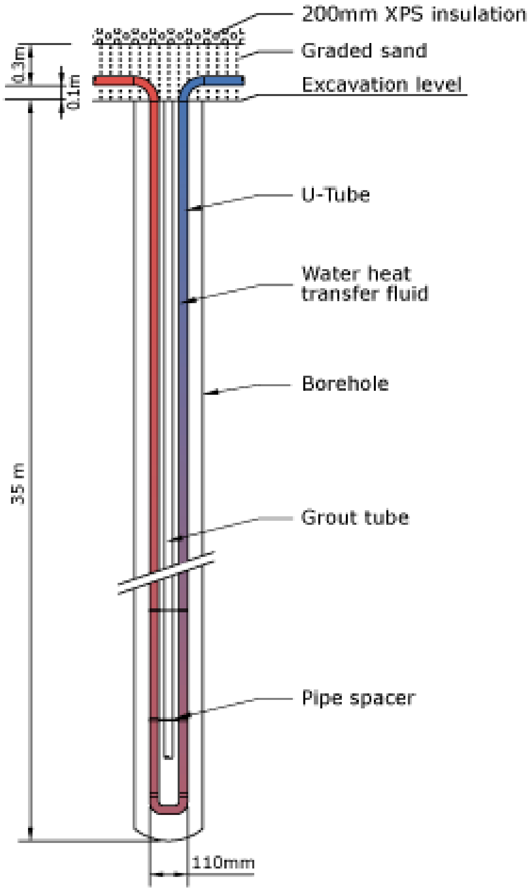


Figure 31: Example of a borehole, seen from the side [69].

A BTES system will introduce a significant investment cost due to drilling of boreholes and a large network of pipes. To justify these investment costs, the system should be able to store a large amount of surplus heat. To provide a few examples, details of some recent BTES-projects in Norway are given in Table 9.

Table 9: BTES projects in Norway [71] [72] [73].

Project	**Number of wells**	**Depth of wells**	**Borehole length**	**Capacity**	**Price**	**Price/borehole length**
Ahus	390	200 m	78.000 m	8 MW	100 MNOK	1.282 NOK/m
Nydalen	180	200 m	36.000 m	6 MW	60 MNOK	1.667 NOK/m
Arcus	91	300 m	27.300 m	4 MW	38 MNOK	1.392 NOK/m
Average	-	-	-	-	-	**1.447 K/m**

3. Case A: Snow storage

In the following chapters, the four cases will be presented and evaluated.

Case A examines snow storage and is divided into three parts: cover material, source of snow and distribution. The first part concerns purchase and transport of the cover material. Next, snow harvesting and temperature dependent snowmaking will be evaluated against each other as sources of snow. Finally, the distribution of snow to the ski tracks will be examined, before the results are discussed in a summary.

3.1 Cover material

The cover material used in this case is sawdust, because it has been used several times before showing good results. Based on the literature study, a 30% melting loss is accounted for during the storage period. As 12.000 m^3 of snow is required in the end of October, at least 17.143 m^3 should be stored. A suitable storage site is assumed to exist near the ski tracks, and all the snow will be stored in one pile (The effects of splitting the pile into several smaller piles will be discussed in the summary). Assumed that the pile has the shape of a hemisphere, 2.552 m^3 of sawdust is required to cover the snow, when the thickness of the layer is set to 0,4 m. Granåsen has a supplier who sells sawdust for total price of 57 NOK/m^3, which includes transportation to Granåsen [64]. The transportation of sawdust is done by lorries, and the loading capacity of the lorries is assumed to be of 25 m^3. As the distance is 34 km each way, the total distance becomes 6.941 km. The fuel consumption of the lorries is assumed to be 0,47 l/km, at an average speed of 60 km/hr [74]. The total fuel consumption becomes 3.262 l/yr, which is equivalent with 34.909 kWh/yr (1 l diesel = 10,7 kWh). Given that the sawdust should be fully replaced every three years [61], the operating costs associated with purchase and transport of sawdust are 48.488 NOK/yr, while the investment costs are 145.464 NOK.

3.2 Source of snow

3.2.1 Snow harvesting

Snow harvesting for skiing purposes is not a widely used technique. The procedure involves collection of snow from roads, football fields, parking spaces, lakes, airports or similar. This snow can be stored, or used directly in the ski tracks. However, if the snow is harvested near the ski arena, the need for plowing implies that there is already snow in the ski tracks, so storage will be the obvious alternative.

The first question regarding snow harvesting is where the snow should be collected from. The best option is to take the snow from a place where is has to be removed anyway, to obtain a dual effect. However, due to traffic, short time limits for plowing and the amount of dirt in the snow, roads and parking spaces are not optimal sources. Football fields might be a better option if they can be accessed by heavy vehicles. Lakes are probably the best source, due to the clean surface layer. Granåsen has a couple of lakes with road access within a 5 km range. The problem with lakes is that the ice has to be thick enough to carry heavy vehicles. Moreover, wheeled vehicles can easily lose the grip on the ice, or get jammed. Airports can also be good sources, because they have enormous areas of paved surface. Again, the traffic and the time limit is problematic, but there is a time window at night when there is no air traffic. The nearest airport to Granåsen is Trondheim Airport, Værnes, which is located 40 km away.

The 8.000 m^3 of snow stored in Granåsen in 2015 was collected from the ski tracks, and will be used as an example. The work took 8 days and 525 working hours in total with several vehicles: excavators, lorries, tractors with supporters, wheel loaders, snow groomers and snow blowers. For more details, see Appendix D. The average wage is set to 915 NOK/hr including fuel costs, and the average fuel consumption of the vehicles is set to 17,4 l/hr. For details regarding these values, see Appendix E. The resulting operating costs are <u>480.375 NOK/yr</u> and the fuel consumption is 9.135 l or <u>97.745 kWh/yr.</u>

The problem with harvesting snow from a ski arena is the infrastructure. The network of tracks is simply not dimensioned for heavy vehicles. An alternative method of snow harvesting could be done with less vehicles. A promising way would be to use a large self-propelled snow blower to blow snow into trailing lorries. The lorries would transport the snow to the storage site, where an excavator would build a pile. An example of a suited snow blower is the TV2200 from Øveraasen AS, which is made for snow clearing at airports, and can handle as much as 48.000 m^3/hr [75]. The costs and energy consumption of snow harvesting depend on the distance between the collection and storage. For now, it is assumed that there is a suitable place for snow harvesting, 1 km from Granåsen.

It is assumed further, that two lorries with a load capacity of 25 m^3 could transport the snow effectively, with a filling time of 3 minutes, and an equal time for driving and dumping, at an average speed of 50 km/hr. In a working day of 7,5 hours, 3.750 m^3 could be harvested, which means that 17.143 m^3 could be stored in 4,57 days or 137,14 working hours, divided by four vehicles. The TV2200 has a high fuel consumption, so the average wage including fuel costs are set to 1.390 NOK/hr for this method. The average fuel consumption is set to 66 l/hr, see details in Appendix E. The total operating costs would be <u>190.625 NOK/yr</u>, while the fuel consumption would be 9.051 l/yr or <u>96.849 kWh/yr</u>.

3.2.2 Temperature dependent snowmaking

Today, the snowmaking system in Granåsen consists of 40 non-automatic TDSs, 30 of which are lances and the rest are fan guns [76]. The fan guns are of the type Lenko NW450, which have a maximum production rate of 1.600 m^3/day at a density of 400 kg/m^3 [77]. The maximum water capacity in Granåsen is 300 m^3/hr from a nearby lake [64]. This means that it is possible to produce 18.000 m^3 of snow with a density 400 kg/m^3 per day.

The production potential is evaluated from TDSs in Granåsen. For the last ten years, daily weather data (air temperature and RH) for the winter months December, January and February is collected from the Norwegian Meteorological Institute. The weather station used is Trondheim, Voll, which is located 8 km from Granåsen. Equation (6) is used to calculate the T_W from the air temperature, T, and the RH [78]:

$$T_W = T\tan^{-1}[0{,}15(RH + 8{,}31)^{0,5}] + \tan^{-1}[T + RH] - \tan^{-1}[RH - 1{,}68] + 0{,}0039[RH^{1,5}]\tan^{-1}[0{,}023RH] - 4{,}69 \qquad (6)$$

Given the daily T_W, the production potential and corresponding EVR for fan guns and lances in Granåsen can be evaluated from equation (1)-(5). Table 10 and Table 11 summarizes the most important results from temperature dependent snowmaking in Granåsen. The electricity costs are based on an average electricity price of 0,8 NOK/kWh in Trondheim since 2005, including all charges, such as grid rental etc. The data is collected from Nord Pool, see details in Appendix C.

Table 10: Estimated average production potential per season at different start-up temperatures (T_W start) for a fan gun in Granåsen in the last ten winters. The worst year (2008) is also included. Weather data is collected from www.eklima.met.no.

***Tw* start**	**Avg. production potential**	**Production potential, worst year**	**Avg. snowmaking days**	**Avg. EVR, kWh/m³**	**Avg. operating costs, NOK**	**Avg. CVR, NOK/m³**
-2	39.052	19.072	45,4	1,45	44.358	1,16
-5	30.497	5.087	27,4	1,29	30.868	1,03
-7	23.865	2119	18,9	1,22	23.033	0,98

Table 11: Estimated average production potential per season at different start-up temperatures (T_W start) for a lance in Granåsen in the last ten winters. The worst year (2008) is also included. Weather data is collected from www.eklima.met.no.

***Tw* start**	**Avg. production potential**	**Production potential, worst year**	**Avg. snowmaking days**	**Avg. EVR, kWh/m³**	**Avg. operating costs, NOK**	**Avg. CVR, NOK/m³**
-2	24.118	9.483	45,4	0,95	17.893	0,76
-5	19.988	3.077	27,4	0,89	14.141	0,71
-7	16.097	1.371	18,9	0,87	11.176	0,70

The snowmaking system has to be dimensioned for the worst year, which was 2008. Lances are chosen as they have a lower CVR and EVR. A T_W start at -5 °C, seems to be sufficient. This would require six lances to produce at least 17.143 m^3. These lances are assumed to be fully automated, and mounted in a circle around the storage site. Assuming that six of the existing lances in Granåsen can be used, there will be no investment costs of new snow producing equipment. However, the system has to be automated. These costs are estimated to be 2 MNOK from the plant in Tonstad, Norway, in section 2.1.7 [29]. It should be noted that this investment should be made in any case, in order to fully utilize the available snowmakers.

The different techniques to provide snow are compared in Table 12. The harvesting technique used in Granåsen in 2015 is not sustainable, as the CVR is more than 100 times higher than temperature dependent snowmaking. Although the alternative method of snow harvesting has an improved effectiveness, temperature dependent snowmaking is by far the most effective method. Note that the alternative method of snow harvesting would have the lowest total costs for 11,2 years, but this process is based on optimal conditions. Hence, temperature dependent snowmaking will be chosen as the source of snow.

Table 12: Snow sources compared. Melting losses are not included

	Harvesting, Granåsen 2015	**Harvesting, Alternative**	**Snowmaking**
Investment costs	-	-	2 MNOK
Operating costs	480.375 NOK/yr	190.625 NOK/yr	12.172 NOK/yr
EVR	15,59 kWh/m^3	8,07 kWh/m^3	0,89 kWh/m^3
CVR	76,61 NOK/m^3	15,89 NOK/m^3	0,71 NOK/m^3

3.3 Distribution

The distribution of the snow to the ski tracks is evaluated with data from Granåsen in 2015. The procedure used in Granåsen depends on the same vehicles as the harvesting procedure, except from the snow blowers. The work took 4 days and 373 working hours in total, see Appendix D. The operating costs were 341.295 NOK/yr, and the fuel consumption was 6.490 l/yr or 69.445 kWh/yr, based on the same wage and fuel consumption as for the harvesting procedure in Granåsen.

3.4 Summary and discussion

A summary of case A presented in Table 13.

Table 13: Summary of snow storage and its parts, based on a final snow volume at the end of the storage period of 12.000 m^3, assumed 30% melting losses.

Case A	**Cover material**	**Snowmaking**	**Distribution**	**SUM**
Investment costs	145.464 NOK	2 MNOK	-	2,1 MNOK (estimated)
Operating costs	48.488 NOK/yr	12.172 NOK/yr	653.160 NOK/yr	713.760 NOK/yr
EVR	2,91 kWh/m^3	1,27 kWh/m^3	11,08 kWh/m^3	15,26 kWh/m^3
CVR	4,04 NOK/m^3	1,01 NOK/m^3	54,43 NOK/m^3	59,48 NOK/m^3

Case A has relatively low investment costs, but the operating costs are high. Table 13 shows clearly how the major part of the energy and costs goes to the distribution of the snow. It should be remembered that some distribution is necessary, even for temperature dependent snowmaking without snow storage, meant for immediate distribution in the ski tracks. Note that the distribution process is based on a single example (Granåsen, 2015), and this will be discussed further in the conclusion.

A way to minimize the distribution costs can be to store the snow in several smaller piles at the ski arena, to shorten the total distance of transport. Assume that fully automated lances or fan guns were spread along the ski tracks at a certain interval. At times with urgent demand for supplementary snow, the snowmakers could blow snow on the ski tracks, and at times with certain snowmaking conditions, some of the snowmakers could be rotated to produce piles for storage. This would require more cover material, but shorten the total length of transport of the snow. Another advantage with this solution is that the distance between the snowmakers would reduce the risk of a possible problem: micro-climate. When six snowmakers run simultaneously in a relatively small circle, a micro-climate can occur within a large plume in the middle of the circle. In this plume, saturated conditions occur quickly when several snowmakers are used simultaneously. If the snowmakers are spread out, they will create individual plumes instead. Also, although the system has to be dimensioned for the worst year, the average year requires only 1-2 lances to meet the demand for snow for storage. This makes the rest of the snowmakers redundant, and they have to be moved to the tracks before they can be used, which makes the system more complicated. A good trade-off may be to mount 1-2 snowmakers permanently at a storage site, and the rest along the ski tracks. This way, the snowmakers along the tracks could create additional piles next to the tracks during poor winters.

The drawback with snow storage is the lack of possibility to produce supplementary snow at warm temperatures. The stored snow in Granåsen in 2015 was distributed in late November, but after a short period of rain and warm weather, the tracks were icy, nearly melted and not suited for skiing by early December. In the worst year (2007), conditions for snowmaking from TDSs were not present until mid-January, based on weather data for the last ten years. A possible outcome is therefore that all of the stored snow is melted by December, and 1,5 months goes by without ski conditions in the middle of the winter. As the temperatures are excepted rise in the years to come, case A’s viability is further threatened. A warmer climate would mean less days with temperature dependent snowmaking, and a higher melting rate. Thus, an extra volume of snow should be stored as a precaution for the unpredictability of the weather, decreasing the effectiveness of the case.

Snow harvesting seems unsustainable from the results in Table 12. With existing machinery, the alternative method presented here can be considered as the optimal method of snow harvesting. Even with the assumption that a harvesting site exists 1 km from the storage site, the method cannot compete with temperature dependent snowmaking. The only way for snow harvesting to be viable, is if it can be conducted in combination with the necessary task of plowing roads, airports, parking spaces, football fields or similar. With complex logistics and cooperation with the other plowing-operators, such a snow harvesting system is possible, especially if vehicles designed for the purpose were developed. Eventually, it could eliminate the costs associated with snowmaking, and create an income instead. The question is if the drive for such a development is present, as the costs associated with temperature dependent snowmaking is minor.

A possible improvement could be to thermally insulate, or even cool the ground beneath the ski tracks. A recent report calculated that 500 W/m^2 could keep a 10 cm layer of snow stable in temperatures just above 0 °C [79]. In a 5 km track, 6 m wide, this would take 15.000 kW, which equals 10,8 GWh or 8,64 MNOK in one month (0,8 NOK/kWh). Even one day with cooling of a 1 km track would cost 57.600 NOK. Hence, cooling of the ground is rejected, unless a heat recovery system could be implemented.

4. Case B: Temperature independent snowmaking with direct heat recovery

4.1 Heat recovery in Granåsen

To make comparison between the cases easier, a transcritical one stage CO_2-heat pump will be used in the cases involving heat recovery. CO_2 is chosen as the refrigerant in order to obtain high temperatures. In the cases involving TISs, the SF220 will be used as the example.

If a heat recovery system is to be implemented, there has to be a nearby demand for heat. Trondheim has a large district heating network, which covers more than 30% of the city's total heat demand [80]. Quite conveniently, the network passes next to the facilities in Granåsen (see map in Appendix B), and it would be a perfect receiver of surplus heat from snowmaking. However, the required connection temperature to the network is 115 °C [81], which is high, coming from a snowmaking system. Especially since the limitation of a conventional CO_2-heat pump is to heat water at atmospheric pressure up to 100 °C [82]. The bottleneck is the high pressure in the *gas cooler*.

An alternative is to create a local district heating network. A swimming pool with a heat demand of 1,8 GWh in 2014 [81] is located 2,3 km away. In addition, a large shopping mall is located 4 km away. However, both already get their heat demands filled from the mentioned district heating network. Also, the distances would introduce large investment costs due to piping.

As other options are unsuitable, the surplus heat from snowmaking will be used to meet the heat demand of three planned buildings at the ski arena in Granåsen. A map of these buildings can be seen in Figure 32. The heat pump will be utilized to heat water from 15-70 °C, which is sufficient to deliver heat and hot tap water to the buildings. The focus will be on heat supply to the building structures, and the arrangements inside the buildings are not covered.

Unless otherwise stated, the approach temperature in all heat exchangers is set to at least 10 K. The approach temperature is the minimum temperature difference between two streams in a heat exchanger, creating a driving force for the heat exchange. Furthermore, pure countercurrent flow is assumed in all heat exchangers, so the heat exchange follows these relations:

$$Q = UA\Delta T_{LM} \tag{7}$$

$$Q = \dot{m}cp\Delta T \tag{8}$$

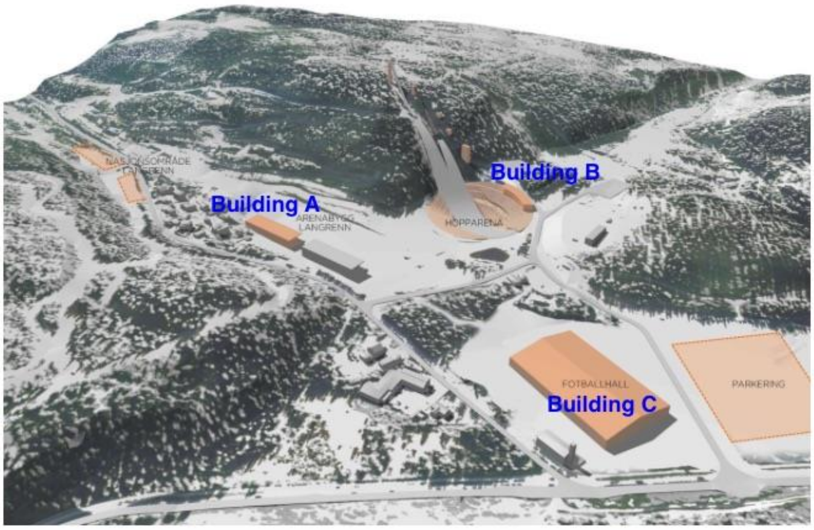


Figure 32: Granåsen and the planned buildings. The illustration is collected from the Norwegian Ski Federation and modified.

4.2 Overview

In case B, the SF220 will run throughout the year, sending residual heat directly to the buildings. A schematic diagram of the direct heat recovery system can be seen in Figure 33.

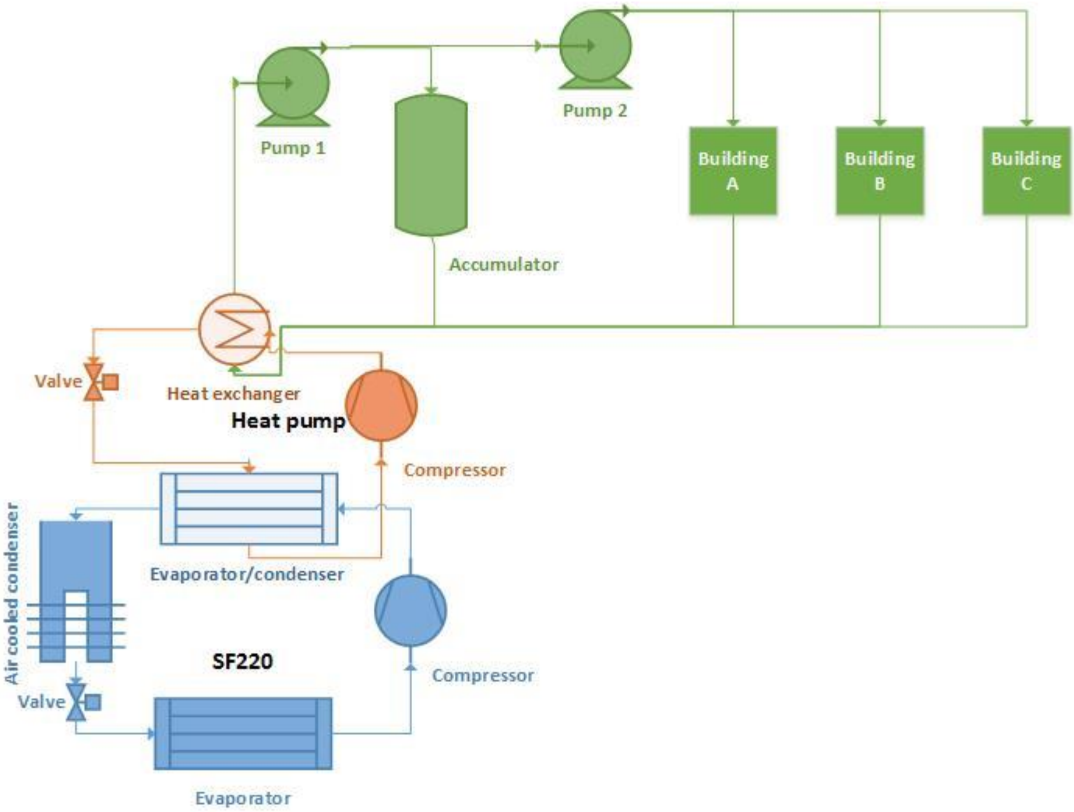


Figure 33: A schematic diagram of the direct heat recovery system in case B. The CO_2-heat pump (red) is connected to the SF220-circuit (blue), creating a cascade cycle. The heat pump delivers heat to the buildings through water (green).

The SF220 will be connected directly to a transcritical CO_2-heat pump, creating a cascade cycle so that the condenser in the SF220-circuit will be the evaporator of the heat pump. After the heat pump follow accumulator tanks of water for short term storage of heat. This is necessary because the heat output from the heat pump will be higher than the heat load to the buildings, even during the coldest days. From the accumulators, hot water at 70 °C is supplied to the buildings, and returned at 15 °C. An air cooler is added to the SF220-circuit in order to have the possibility to run the SF220 without heat recovery.

An overview of case B is depicted in Figure 34. The cascade cycle will be placed in a small building next to building A, with the following dimensions: 12 m x 5 m x 5 m (L, W, H; the size of two 40' containers). This is done for two reasons. Firstly, the accumulators should be placed inside one of the buildings (building A), and secondly, the snow should be produced near the ski tracks. The water is then distributed to the other buildings, and the total distance of ditches between the buildings is measured to be 550 m.

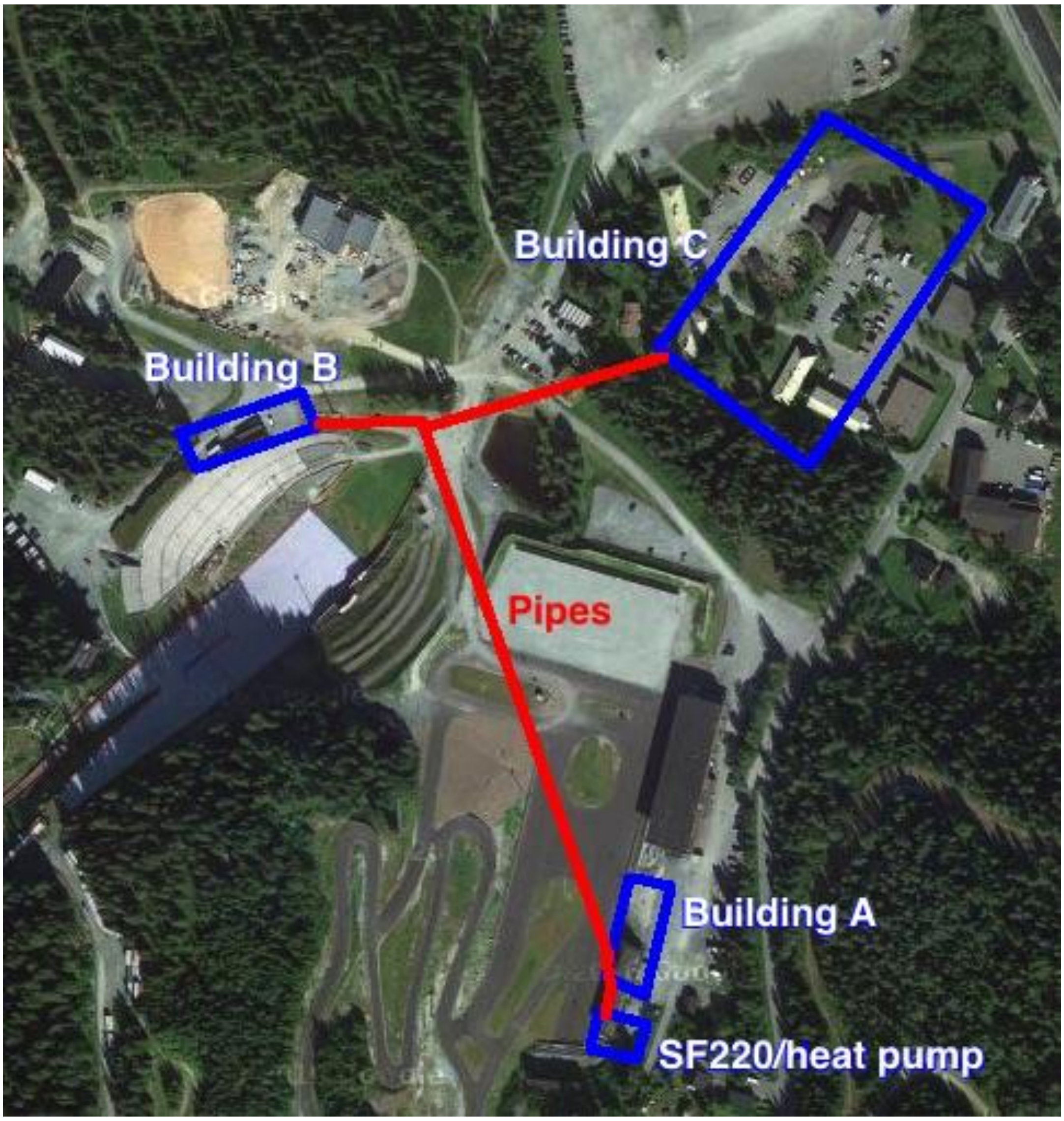


Figure 34: Case B, seen from above. Note the stored pile of snow covered with sawdust in the upper left corner. The satellite photo is collected from Google Maps, 2015.

4.3 Available heat from the SF220

First, the amount of heat available from the SF220 is calculated. The water flow rate is 129.600 kg/day (Table 5), with inlet and outlet temperatures of 5 °C and -8 °C respectively [52]. The specific heat which has to be removed from the water can be calculated as follows:

$$q = (cp_{Water}\Delta T_{Water}) + lf_{Water} + (cp_{Ice}\Delta T_{Ice}) \qquad (9)$$

$$q = (4{,}19\left[\frac{kJ}{kg \cdot K}\right] * 5[K]) + 334\left[\frac{KJ}{kg}\right] + (2{,}11\left[\frac{kJ}{kg \cdot K}\right] * 8[K]) = 353{,}83\left[\frac{kJ}{kg}\right]$$

From this, the cooling demand in the evaporator, Q_E, can be calculated:

$$Q_E = q\dot{m} = 353{,}83\left[\frac{kJ}{kg}\right] * 129.600\left[\frac{kg}{day}\right] = 45.748.800\left[\frac{kJ}{day}\right] \qquad (10)$$

$$Q_E = 45.748.800\left[\frac{kJ}{day}\right] * 0{,}00027\left[\frac{kWh}{kJ}\right] = 12.352\left[\frac{kWh}{day}\right] = 514{,}68\,[kW]$$

Given that the evaporator and condenser temperatures are -30 °C and 25 °C respectively [52], details at the state points of the isentropic one stage SF220-cycle with Ammonia as refrigerant are given in Table 14. The locations of the state points can be seen in Figure 35.

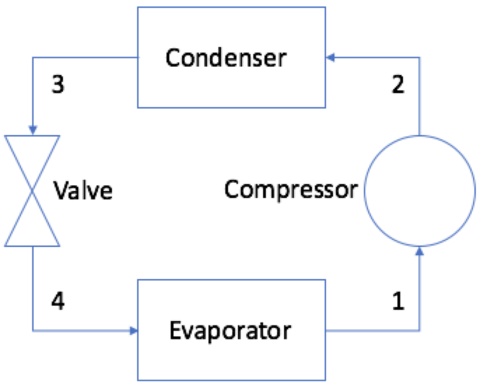


Figure 35: A one stage vapor compression cycle.

Table 14: Pressure, temperature and enthalpy difference at the state points of the SF220-cycle.

State point	Pressure	Temperature	ΔH to the next point
1	1,2 bar	-30 °C	315,67 kJ/kg
2	10 bar	25 °C	1422,6 kJ/kg
3	10 bar	25 °C	-
4	1,2 bar	-30 °C	1106,92 kJ/kg

Next, the mass flow rate of the refrigerant, $\dot{m}_{R717}$, can be calculated:

$$\dot{m}_{R717} = \frac{Q_E}{\Delta H_{4-1}} = \frac{514{,}68\ [kW]}{1106{,}92\ \left[\frac{kJ}{kg}\right]} = 0{,}46\ [\frac{kg}{s}] \tag{11}$$

Where ΔH_{4-1} is the enthalpy difference from state point 4-1. With the mass flow rate given, the isentropic work input to the compressor, W_{Is}, can be obtained:

$$W_{Is} = \dot{m}_{R717}\Delta H_{1-2} = 0{,}46\ \left[\frac{kg}{s}\right] * 315{,}67\ \left[\frac{KJ}{kg}\right] = 145{,}21\ [kW] \tag{12}$$

Which in turn is used to find the isentropic efficiency, η_{Is}:

$$\eta_{Is} = \frac{W_{Is}}{W_{SF220}} = \frac{145{,}21\ [kW]}{227\ [kW]} = 0{,}64 \tag{13}$$

Where the actual work input, W_{SF220}, is given in Table 5. The pressure ratio, π, of the cycle is:

$$\pi = \frac{P_H}{P_L} = \frac{10\ [Bar]}{1{,}2\ [Bar]} = 8{,}33 \tag{14}$$

Where P_H and P_L is the pressure at the high-pressure side and low-pressure side respectively. The actual COP for cooling, COP_C, follows this relation:

$$COP_C = \frac{Q_E}{W_{SF220}} = \frac{514{,}68\ [kW]}{227\ [kW]} = 2{,}28 \tag{15}$$

By comparison, the maximum obtainable COP, COP_{Carnot}, between two temperature levels, T_L and T_H, is given by:

$$\mathrm{COP_{Carnot}} = \frac{\mathrm{T_L}}{\mathrm{T_H - T_L}} = \frac{243\ [\mathrm{K}]}{(298 - 243)\ [\mathrm{K}]} = 4{,}41 \tag{16}$$

Which allows the thermal efficiency, η_{Th}, to be found:

$$\eta_{Th} = \frac{COP_C}{COP_{Carnot}} = \frac{2{,}28}{4{,}41} = 0{,}52 \tag{17}$$

Next, the COP for heating, COP_H, can be calculated:

$$COP_H = COP_C + 1 = 2{,}28 + 1 = 3{,}28 \tag{18}$$

Finally, the available heat in the condenser, Q_C, is found:

$$Q_C = COP_H \frac{Q_E}{COP_C} = \frac{3,28}{2,28} * 12.352 \left[\frac{kWh}{day}\right] = 17.770 \left[\frac{\text{kWh}}{\text{day}}\right] = 740,5 \, [kW] \qquad (19)$$

As a verification, the isentropic efficiency from equation (13) corresponds well with Figure 36, at the given pressure ratio. A thermal efficiency in the range between 0,4-0,6 is also within normal limits. This strengthens the reliability of the calculations.

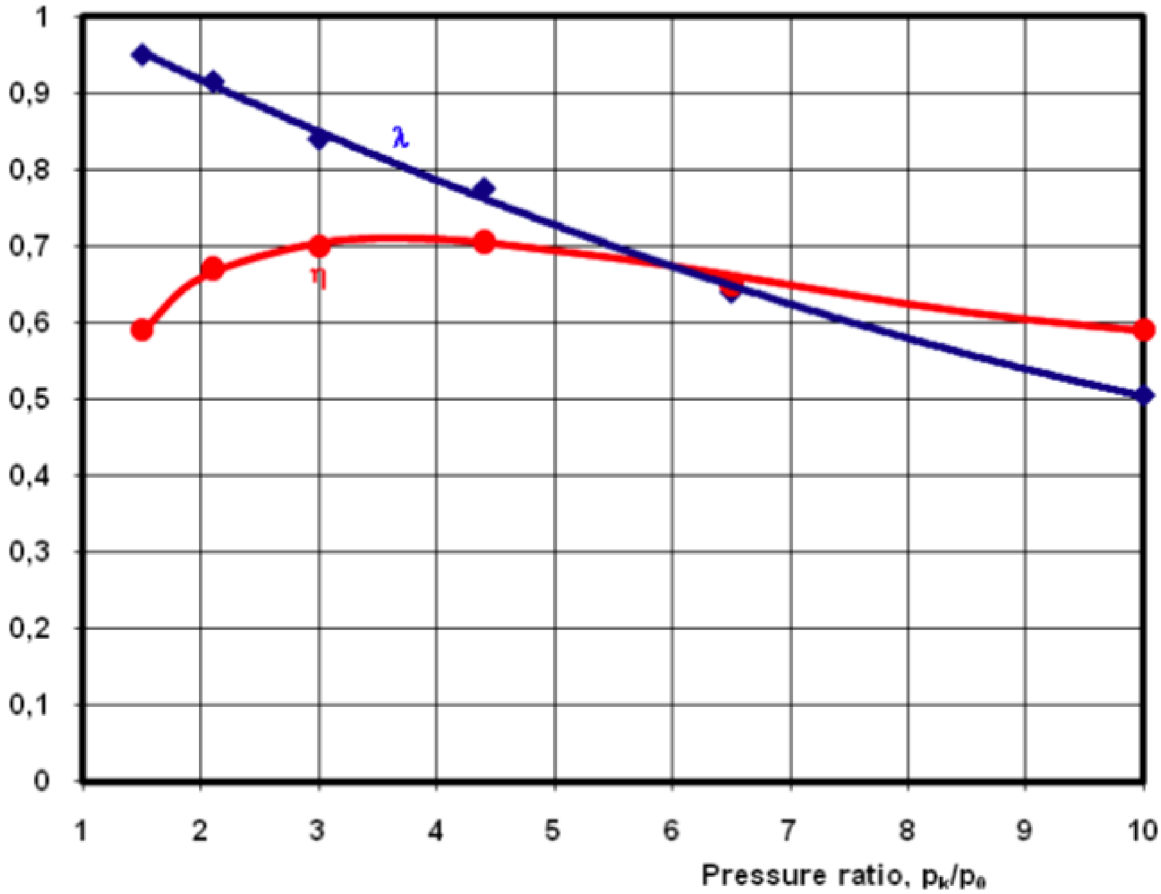


Figure 36: Isentropic efficiency in red, as a function of the pressure ratio for a typical large piston compressor [83].

4.4 Heat pump

The heat pump cycle is drawn in a temperature-entropy (TS) diagram in Figure 37, and details at the state points are given in Table 15. Note that as the heat pump cycle is transcritical, the heat is delivered at gliding temperatures. This means that the heat pump operates both below and above the critical point of CO_2.

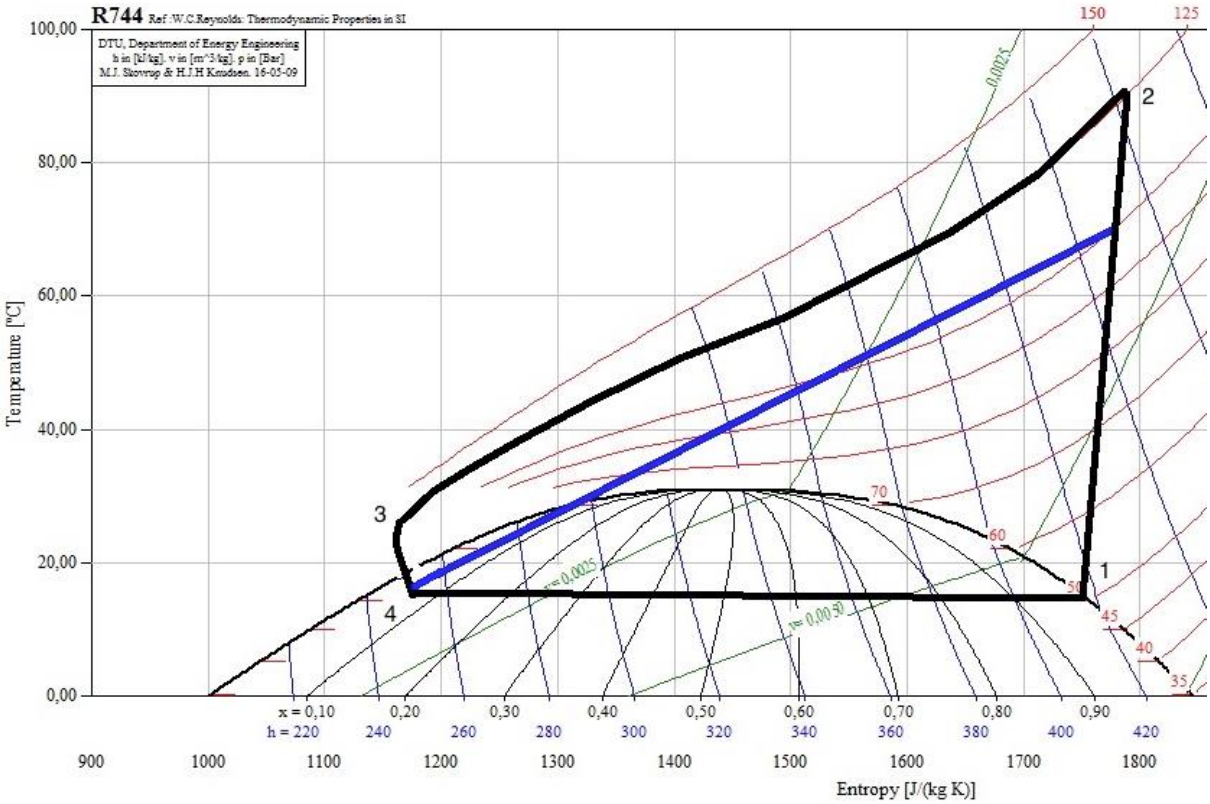


Figure 37: TS diagram of the heat pump cycle in case B. The blue line shows the water temperature through the heat exchanger at the high-pressure side.

Table 15: Details of the heat pump cycle in case B.

State point	Pressure	Temperature	ΔH to the next point
1	50,1 bar	15 °C	45,6 kJ/kg
2	125 bar	90,4 °C	212,53 kJ/kg
3	125 bar	25 °C	-
4	50,1 bar	15 °C	166,2 kJ/kg

From the approach temperature of 10 K, evaporation of CO_2 will happen at 15 °C. The pressure after the compressor is set to 125 bar, such that the *pinch point*, or the lowest temperature difference between the stream of water and CO_2 will happen at state point 3. A lower pressure would move the pinch point dramatically towards state point 2. This gives a pressure ratio of 2,5, and the isentropic efficiency is set to 0,7 according to Figure 36. Note that the water temperatures shown in blue in Figure 37 is a flat line, while the temperatures of the CO_2 is more curved. This is because the specific heat of water is nearly constant, while the specific heat of CO_2 varies with temperature.

The available heat from the SF220 is already known to be 740,5 kW, so the required mass flow rate of CO_2 is found from equation (11):

$$\dot{m}_{CO2} = \frac{Q_E}{\Delta H_{4-1}} = \frac{740{,}5\ [kW]}{166{,}2\ \left[\frac{kJ}{kg}\right]} = 4{,}46\ \left[\frac{kg}{s}\right]$$

With the mass flow rate given, the heat output from the gas cooler, Q_{GC}, can be calculated:

$$Q_{GC} = \dot{m}_{CO2}\Delta H_{2-3} = 4{,}46\ \left[\frac{kg}{s}\right] * 212{,}53\ \left[\frac{kJ}{kg}\right] = 946{,}92\ [kW] \tag{20}$$

And the work input to the heat pump, W_{HP}, can be determined from equation (12):

$$W_{HP} = \dot{m}_{CO2}\Delta H_{1-2} = 4{,}46\ \left[\frac{kg}{s}\right] * 45{,}6\ \left[\frac{kJ}{kg}\right] = 203{,}17\ [kW]$$

4.5 Heat demand and corresponding snow volume

The sizes of the new buildings in Granåsen are roughly estimated [84], and based on these, the heat demands are estimated in Table 16.

Table 16: List of future buildings in Granåsen with estimated sizes and heat demands.

Building	Purpose	Estimated size	Estimated heat demand
A	Arena building, cross-country skiing	1.000 m^2 x 3 floors	355.500 kWh/yr
B	Arena building, ski jumping	2.000 m^2 x 2 floors	474.000 kWh/yr
C	Multipurpose hall	9.000 m^2	675.000 kWh/yr
SUM	-	-	**1.504.500 kWh/yr**

The sizes of the buildings are only roughly estimated, so the heat demands are not simulated with the highest accuracy. Instead, they are approximated from the Norwegian building regulations TEK10 [85]. TEK10 requires the specific energy use of office buildings to be less than 150 kWh/m^2. Using the assumption that 79% of a building's energy demand is for heating purposes [86], the approximated heat demands can be calculated for the two arena buildings. The specific heat demand from the multipurpose hall is assumed to be 75 kWh/m^2, from a similar multipurpose hall [87].

Next, the total heat demand of the buildings is distributed on a monthly basis, from the assumption that the heat demand is proportional to the ambient temperature:

$$Q = c\Delta T = \frac{1.504.500 \left[\frac{kWh}{yr}\right]}{8.760 \left[\frac{hrs}{yr}\right]} = 171{,}75\ [kW] \qquad (21)$$

The indoor temperature is set to 19 °C, while the average temperature in Trondheim is 4,75 °C, see Appendix A. Next, *c* can be estimated from equation (21):

$$c = \frac{Q}{\Delta T} = \frac{171{,}75\ [kW]}{(19 - 4{,}75)[°C]} = 12{,}05 \left[\frac{kW}{°C}\right]$$

The monthly heat demands, Q_M, are found from the monthly temperature differences, ΔT_M:

$$Q_M = c\Delta T_M * 24 \left[\frac{hrs}{day}\right] * \frac{365}{12}\left[\frac{days}{month}\right] \qquad (22)$$

The time of snowmaking to meet this heat demand is calculated as follows:

$$t = \frac{Q_M}{Q_{GC}} = \frac{Q_M[kWh]}{946{,}92\ [kW]} \qquad (23)$$

Finally, the produced snow volume is found, based on the daily production rate of 220 m^3/day (Table 5). The monthly heat demands and corresponding snow volume is listed in Table 17.

Table 17: Monthly heat demands and average heat loads from the buildings, as well as the corresponding amount of snow produced to meet these demands.

Month	**Heat demand, kWh**	**Heat load, kW**	**Production days**	**Snow volume, m^3**
January	193.561	265	8,5	1.874
February	189.162	259	8,3	1.831
March	167.167	229	7,3	1.618
April	140.772	193	6,2	1.363
May	87.982	121	3,9	852
June	61.587	84	2,7	596
July	52.789	72	2,3	511
August	57.188	78	2,5	554
September	87.982	121	3,9	852
October	118.776	163	5,2	1.150
November	162.768	223	7,2	1.576
December	184.763	253	8,1	1.789
SUM	**1.504.500**	**-**	**66,2**	**14.564**

The maximum heat load from the buildings, Q_{Max}, can be estimated with equation (21) at an ambient temperature of -20 °C:

$$Q_{Max} = c\Delta T = 12{,}05\ \left[\frac{kW}{°C}\right] * \big(19 - (-20)\big)\ [°C] = 470\ [kW]$$

This is less than half of the heat supply from the heat pump. Hence, accumulators are needed for short term storage of heat.

4.6 Accumulators

Next, the required accumulating volume, V_A, is determined. The cascade cycle is assumed to have to run for at least 20 minutes each time it is turned on, to not reduce the lifetime of the compressors [88]. The maximum amount of heat produced by the heat pump, Q_A, is then:

$$Q_A = Q_{GC}t = \ 946{,}92\ [kW] * \frac{1}{3}[hr] = \ 315{,}64\ [kWh] \tag{24}$$

While the specific heat capacity for an accumulator tank, cp_A, is:

$$cp_A = \frac{cp_{Water}\rho_{Water}}{3600} = \frac{4{,}19\ \left[\frac{kJ}{kg \cdot K}\right] * 1\ \left[\frac{kg}{l}\right]}{3600\ \left[\frac{s}{hr}\right]} = 1{,}16 * 10^{-3}[\frac{kWh}{l \cdot K}] \tag{25}$$

The total accumulating volume becomes:

$$V_A = \frac{Q_A}{cp_A\Delta T} = \frac{315{,}64\ [kWh]}{1{,}16 * 10^{-3}\ \left[\frac{kWh}{l \cdot K}\right] * 55[K]} = 4947{,}34\ [l] = 4{,}95\ [m^3] \tag{26}$$

Where ΔT is the temperature difference of water in and out of the accumulators (70-15 °C). The accumulators are, as already mentioned, to be placed inside of building A.

4.7 Melting

A drawback with case B is the melting losses associated with the uncovered snow pile. To estimate the melting rate, the degree-day method is used. The degree-day method is a temperature index approach, where the melting rate is assumed to be proportional to the ambient temperature times the surface area of the snow pile:

$$M = AC(T - T_B)\ \left[\frac{m^3}{day}\right] \tag{27}$$

Where T_B is the boundary temperature for which the snow will melt: 0 °C. The degree-day coefficient, C, is in the range between 0,0016-0,006 m/°C·day, with a typical value of 0,0027 m/°C·day [89]. *C* is set to the middle value of 0,0038 m/°C·day. The total melting losses are estimated based on the assumption that the pile always has the shape of a hemisphere. The average daily temperatures in Trondheim since 1961 are collected from the Norwegian Meteorological Institute. Figure 38 shows the remaining volume of the snow pile during the year, with a production rate as given in Table 17, as well as a doubled production rate.

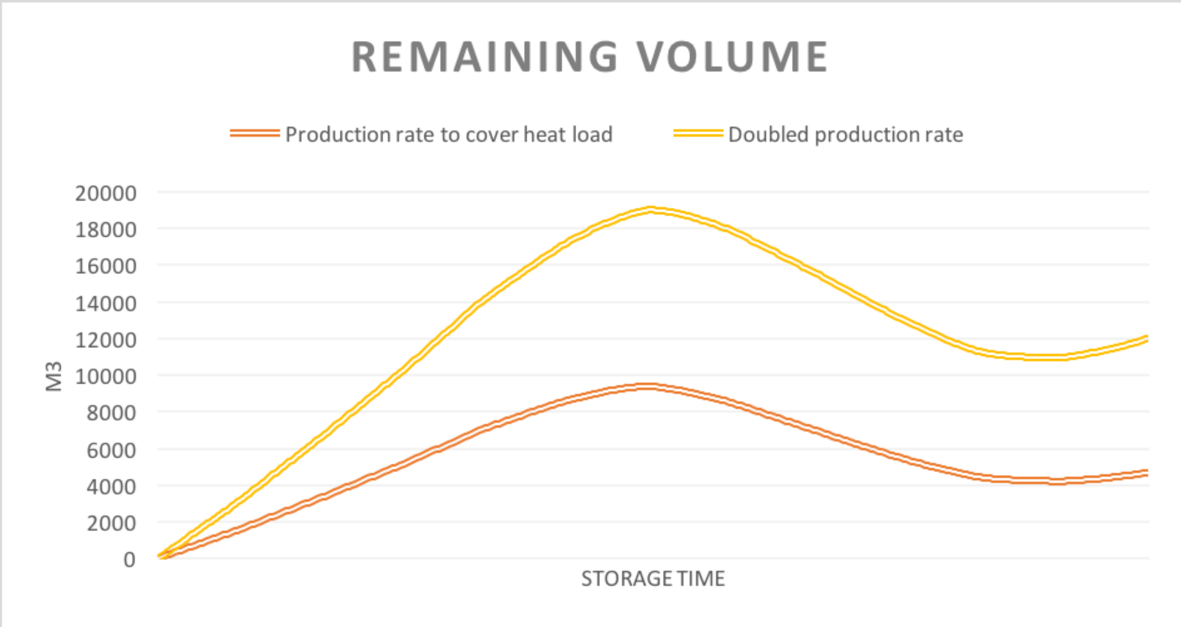


Figure 38: Estimated volume of snow during the year with a starting date at November 1. The production rate to cover the heat demand is seen in orange, while a doubled production rate is seen in yellow. The total melting losses from the two are 67,6% and 56,5% respectively.

According to Figure 38, a doubling of the production rate required to meet the heat demand of the buildings will leave a little more than 12.000 m^3 of snow at November 1. This means that half of the available heat from the SF220 will be wasted in the air cooler, and the total melting losses are 56,5%.

4.8 Heat losses

Some heat losses will occur from the pipes and the accumulators. The district heating network in Norway has 10% losses out of 1.200 km (2013) of pipework [90]. As the length of the pipes in this case is only 550 m (DN65, twin pipes), the corresponding losses would be 0,55%. The losses depend on temperature differences and the insulation of the pipes. To add a safety factor, the losses are estimated to 1%. The accumulators will also have some minor heat losses, but this heat is delivered to building A, which in turn reduces the heat demand of the building. Hence, these heat losses are neglected. The new heat demand from the gas cooler is 1.519.697 kWh.

4.9 Dimensioning the pumps

The constant mass flow rate of the water through the heat pump is found from equation (8):

$$\dot{m}_{Water} = \frac{Q_{GC}}{cp_{Water}\Delta T}\frac{946{,}92\ [kW]}{4{,}19\ \left[\frac{kJ}{kg}\right] * 55[K]} = 4{,}11\ \left[\frac{kg}{s}\right]$$

From the same equation, the maximum flow rate of water to the buildings at the maximum heat load becomes 2,04 kg/s.

Two separate water-circuits with individual pumps, are designed to avoid problems associated with mismatch of the mass flow rates (see Figure 33). Pump 1 transfers heat from the gas cooler to the accumulators, and Pump 2 transfers heat from the accumulators to the buildings. The energy consumption of these pumps are determined by the pressure drops in the pipes, which are calculated with the Darcy-Weisbach equation:

$$\Delta P = \frac{f\rho L v^2}{2D} \tag{28}$$

Where the Darcy friction factor, *f*, is found in the Moody diagram. The power requirements are calculated from equation (5), where the overall pump efficiency is set to 0,6. To perform the calculations, a MATLAB script is written, see Appendix F. Pipe diameters of 65 mm are found to be sufficient for both circuits. The length of the circuits is set to 40 m and 1.100 m for Pump 1 and Pump 2 respectively. The total energy consumption of the pumps are given in Table 18:

Table 18: Energy consumption of pumps. W_{Dim} is the power at average load, and W_{Max} is the power at maximum load.

	Pump 1	Pump 2	SUM
W_{Dim}	0,1 kW	0,07 kW	0,17 kW
W_{Max}	0,1 kW	0,3 kW	0,4 kW
Annual operation hours	1.589	8.760	-
Energy consumption	159 kWh/yr	613 kWh/yr	772 kWh/yr

Pump 1 will have a constant load, and the operation time is determined by the hours of operation of the SF220 with heat recovery from Table 17. Pump 2 is assumed to run constantly.

4.10 Costs

The investment costs of case B are estimated in Table 19:

Table 19: Estimated investment costs of case B.

Building	480.000 NOK
SF220	7.875.000 NOK
Water purification plant	1.000.000 NOK
Heat pump	2.500.000 NOK
Ditches and pipes	2.280.000 NOK
Pumps	15.500 NOK
Accumulators	61.340 NOK
Building centrals	375.000 NOK
Engineering	2.574.060 NOK
SUM	**17.160.400 NOK**
Estimated value	**17,2 MNOK**

Grounds of the estimations are given below. Unless public available, the sources or suppliers will be held anonymous:

- **Building**: Phone conversation with a supplier: 8000 NOK/m^2 x 60 m^2 floor area, including installation of the electrical system.
- **SF220**: Phone conversation with a supplier: 5,5 MNOK plus 25% taxes.
- **Water purification plant**: Conversation at a plant visit, verified by a supplier: 1 MNOK. The water purification plant is due to experiences from both Sjusjøen and Idrefjäll, regarding the Snowfactory. Dirty water from nearby lakes has caused a layer of dirt to stick to the cylindrical inside of the ice machine, causing problems. Hence, a water purification plant is required.
- **Heat pump:** Phone conversation with supplier: 2.000 NOK/kW x 1.000 kW, plus 25% taxes.
- **Ditches and pipes**: From a report, DN65 twin pipes at 4.000 NOK/m x 570 m [91].
- **Pumps**: Estimated costs of 2.500 NOK/kg/s based on prices from www.vvskupp.no and www.amazon.com: Pump 1 = 5.000 NOK and Pump 2 = 10.500 NOK.
- **Accumulators**: Price list from supplier: 2 x 2.500 l tanks at 30.670 NOK/tank [92].
- **Building centrals**: From a report: 3 x 100 kW centrals, one for each building, at 125.000 NOK/central [91].
- **Engineering**: Design, freight and mounting are set to 15% of the total costs, based on a report [91].

4.11 Summary and discussion

The seasonal performance factor (SPF) of case B, with respect to the useful heat output, is found from equation (29):

$$SPF = \frac{Q_{Output}}{W_{Input}} \tag{29}$$

$$SPF = \frac{Q_{Demand}}{W_{SF220} + W_{HP} + W_{Pumps}} = \frac{1.504.500\ [kWh]}{721.315 + 326.063 + 772[kWh]} = 1{,}44$$

Thus, for every kWh of work into the system, the heat output is 1,44 kWh. This SPF is rather low, because half of the available heat from the SF220 is wasted, due to a relatively low heat demand in Granåsen. The energy gain becomes 456.322 kWh/yr, which means savings of 365.057 NOK/yr, at 0,8 NOK/kWh. The EVR and CVR related to distribution is set equal to what was found in case A, although this case does not involve removal and disposal of sawdust prior to distribution. More on this in the conclusion. Case B is summarized in Table 20.

Table 20: Summary of case B, based on a final snow volume of 12.000 m^3.

Case B	**Snowmaking system**	**Distribution**	**SUM**
Investment costs	17,2 MNOK	-	17,2 MNOK
Operating costs	-365.057 NOK/yr	653.160 NOK/yr	288.103 NOK/yr
EVR	-38,03 kWh/m^3	11,08 kWh/m^3	-26,95 kWh/m^3
CVR	-30,42 NOK/m^3	54,43 NOK/m^3	24,01 NOK/m^3

Although the EVR of case B is negative, the investment costs are high compared to the CVR; which is positive. The reason for the positive CVR is the distribution process. It is, however, a good chance that these costs could be reduced, especially since there is no sawdust involved in this case.

The melting model implies that an improved method of storage should be considered. Now, nearly 60% of the snow produced is estimated to be lost during storage, and between April and October the total snow volume is actually decreasing. A solution could be to cover the snow pile with sawdust (or similar) in the middle of April. Another strategy could be to store the snow indoor. Indoor storage is beneficial because the losses from solar radiation, wind and precipitation will vanish.

As in case A, rapid melting can occur after distribution, if the weather is warm. Although the SF220 has the possibility to produce supplementary snow independent of temperature, the production rate is low. Hence, production of supplementary snow cannot happen on short notice and has to be planned. Thus, as in case A, an extra volume of snow should be produced

as a precaution for the unpredictability of the weather. Optionally, thermal insulation or cooling of the ground under the tracks could be considered, possibly with heat recovery.
The size requirements of the snow pile and cascade system can cause problems. The SF220 will create the snow pile in the middle of the ski arena, blocking parts of the ski tracks. One solution can be to construct a pond where the snow can be stored. This would also simplify an eventual process of covering the snow, and require less cover material.

Possible improvements of the system include an internal heat exchanger in the heat pump to exchange heat with the streams out of the evaporator and out of the gas cooler. This would preheat the gas into the compressor and subcool the liquid out of the condenser, increasing the enthalpy difference and thus the thermal efficiency of the cycle. Another benefit is that it would handle the possible problem of liquid in the compressor.

5. Case C: Indoor snowmaking with direct heat recovery

Case C involves an indoor snowmaking hall where snow is produced for extraction to the outdoor ski tracks. The refrigerated hall has a direct heat recovery system as in case B, which means that snow is produced throughout the year to meet the heat demands of the buildings. A combined heating and cooling cycle is utilized to remove heat from the refrigerated hall, and heat water from 15-70 °C. See the schematic diagram of the system in Figure 39. The idea behind this case is that the snow could be produced with a TDS, such that the production rate of snow could be higher than for the SF220. Furthermore, the melting losses could be decreased compared to case A and case B.

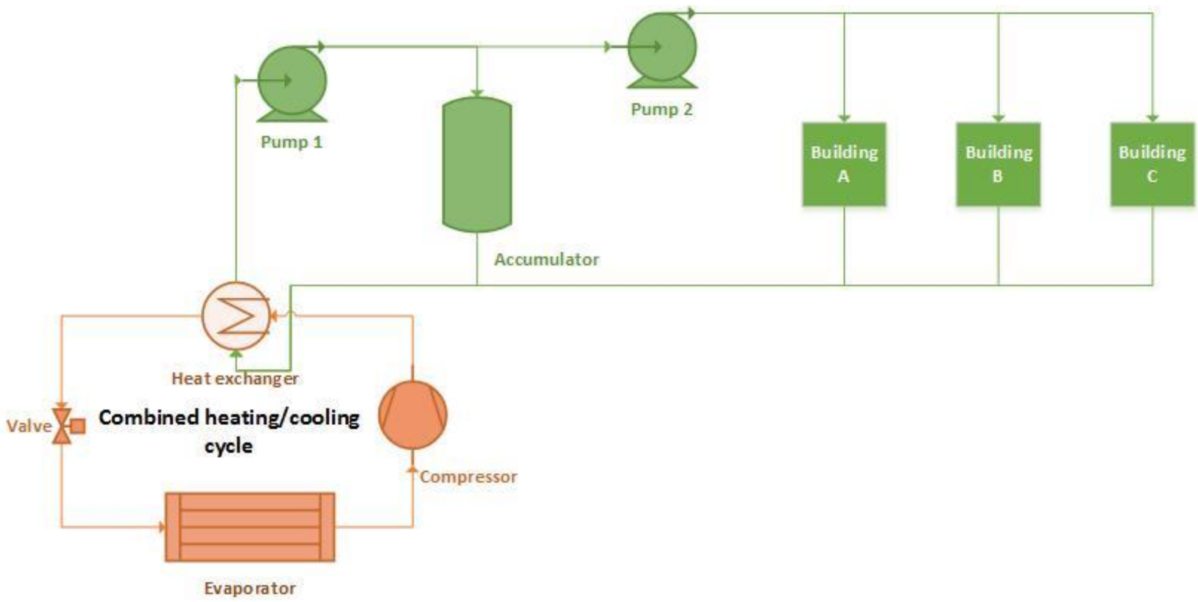


Figure 39: A schematic diagram of the direct heat recovery system in case C, with CO_2 (red) and water (green).

5.1 Dimensioning the hall

An important feature of the hall is that the water droplets have sufficient air time to ensure nucleation, so the minimum height of the TDS is set to 6 m. Moreover, the hall has to be large enough to store 12.000 m^3 of snow. Finally, the hall must have a system of extracting the snow, as well as a drainage system.

The hall is dimensioned as 50 m x 50 m x 14 m (L, W, H), such that the total volume is 35.000 m^3. The floor area requirements of 2.500 m^2 can be fulfilled by locating the building as shown in Figure 40. This location is in close distance to the three new buildings, and the total distance of ditches between the buildings is set equal to that in case B. Accumulators will be placed in building C for this case, due to the close distance from the snowmaking hall. A lance is chosen as the source of snow, assuming that one of the existing lances in Granåsen can be used, to avoid extra investment costs. The lance will be mounted on rails in the ceiling, to be able spread the snow on the floor. The extraction and distribution of snow is assumed to be done by vehicles.

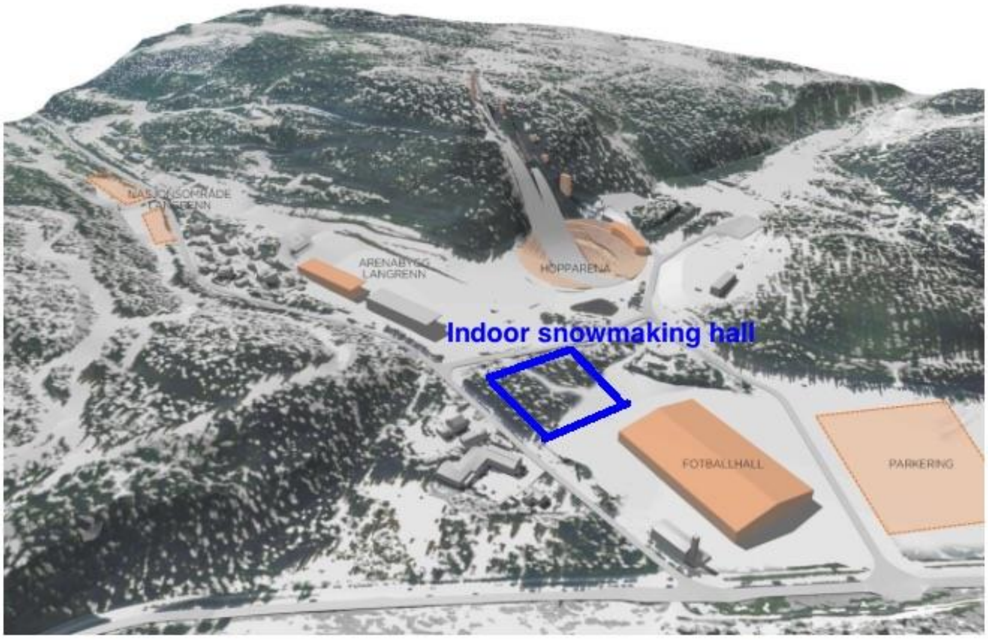


Figure 40: Proposed location of the snowmaking hall, in between the other buildings.

To ensure good snowmaking conditions, the hall is kept at a T_W of -7 °C (Figure 2) during snowmaking, obtained by an air temperature of -6 °C, and a RH of 90%. From Figure 10, the production potential from a lance at this temperature is 560,4 m^3/day. Air coolers are placed in the ceiling, drawing air from below in a draw-through configuration, such that the warmest air under the ceiling will be cooled, and a good circulation of the air in the hall can be obtained. The amount of snow produced will depend on the heat demand, but some snow will be produced daily, as the accumulators can only store heat on a short term basis. After snowmaking, the snow will keep the temperature in the hall from rising to the ambient temperature. This is beneficial because it will lead to a short the start-up time to bring the hall to a T_W of -7 °C.

5.2 Refrigeration load

The estimated refrigeration load of the hall is presented in Table 21 at a dimensioning ambient temperature of 15 °C. Details of the estimations follow after the table.

Table 21: Estimated refrigeration load for the snowmaking hall.

Snowmaking load	1073,6 kW
Transmission load	25,32 kW
Infiltration load	7,21 kW
Equipment load	12,8 kW
SUM	1118,93 kW
Estimated value	**1120 kW**

The snowmaking load is found from equation (9) and (10) With a production potential of 560,4 m^3/day and a snow density of 450 kg/m^3, the mass flow rate of water is 2,92 kg/s. The injected water temperature is assumed to be 5 °C, and the snow is cooled to -6 °C.

The transmission load, Q_T, is found from:

$$Q_T = UA_S\Delta T \qquad (30)$$

Where A_S is the surface area of the hall, and ΔT is the temperature difference between the ambient and inside the hall. U is based on the following k-values from TEK10 [85]: 0,18; 0,15 and 0,13 W/m^2K for walls, floors and ceilings respectively.

The infiltration load is calculated with the software CoolPack, based on a recommended air change factor of 0,4, which means that 14.000 m^3 of air will be exchanged per day. As the hall is designed for very little occupancy, not much air change is required.

The equipment load is estimated from heat gains of the evaporator fans. A specific air cooler (GEA/Goedhart LLK.s-481m^2) has four fans, each of 4 kW. The refrigeration capacity is 140 kW per unit, so eight units will be sufficient to deliver the required refrigeration capacity. The total power supply to the fans is then 128 kW. At this power, with the motor in the hall, and the fans in the outlet of the air flow, the estimated equipment load is found to be 12,8 kW from table values [93].

Any internal load from lighting, vehicles, people and door openings is neglected. However, during distribution in late October, the system will be turned off for five days.

At the average temperature in Trondheim of 4,75 °C, the refrigeration load is estimated to 1.096 kW, which is only 2% lower than the dimensioned load. Thus, the refrigeration cycle can easily be dimensioned for 15 °C and regulated by decreasing the mass flow rate.

5.3 Combined heating and cooling cycle

The combined heating and cooling cycle is drawn in a TS diagram in Figure 41 and details at the state points are given in Table 22.

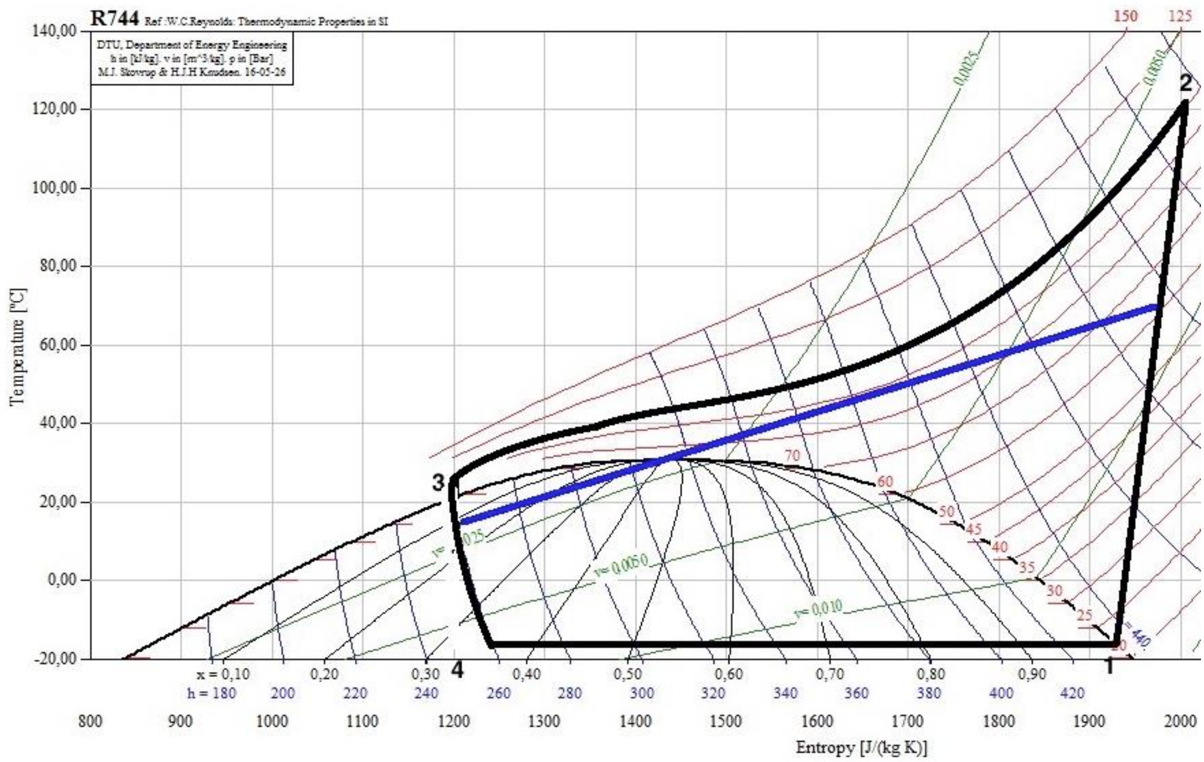


Figure 41: TS diagram of the combined heating and cooling cycle in case C. Note that the pinch point is located in the middle of the heat exchange in this case.

Table 22: Details of the combined heating and cooling cycle in case C.

State point	Pressure	Temperature	ΔH to the next point
1	22,3 bar	-16 °C	97,28 kJ/kg
2	100 bar	121 °C	277,66 kJ/kg
3	100 bar	25 °C	-
4	22,3 bar	-16 °C	180,38 kJ/kg

The pressure ratio of the cycle is 4,5, and the isentropic efficiency is set to 0,7 according to Figure 36. As the approach temperature in the heat exchangers is 10 K, the evaporating temperature will be -16 °C and the pressure at the high-pressure side is set to 100 bar, to be able perform sufficient heating of the water. A mass flow rate of 6,08 kg/s covers the average refrigeration load while the maximum load (1.120 kW) is covered by a mass flow rate of 6,21 kg/s. The average and maximum amount of heat delivered from the gas cooler is then 1.688 kW, and 1.724 kW, respectively. The total heat demand of 1.519.545 kWh/yr can be filled with 37,53 days of snowmaking, which would produce 21.035 m^3 of snow at a work input of 532.742 kWh/yr. Here lies the reason for the large dimensions of the hall: 21.035 m^3 of snow would fill the hall up to 8,4 m, leaving less than the required 6 m of height for the lance. All the values above are calculated with data from Table 22 and equation (11), (12) and (20).

The combined heating and cooling cycle is assumed to run for an average of 5 minutes daily, prior to snowmaking, to bring the hall to a T_W of -7°C. This will reduce the total amount of snow produced, such that the final volume of snow will be 20.335 m^3. Note that a change in the daily production rate would give approximately the same result, as the refrigeration load would be adjusted equivalently.

Melting losses of case C are neglected. This is because melting losses of more than 40% would still leave more than the required 12.000 m^3 at the start of the season. The melting losses are expected to be much lower than this, because the temperature in the hall will not rise to the ambient temperature, due to the daily snowmaking. Furthermore, contributions from solar radiation, wind and precipitation will vanish.

5.4 Pumps, accumulators and snowmaker

The mass flow rate of water through the gas cooler at average refrigeration load is 7,48 kg/s from equation (8). Based on the script in Appendix F, 85 mm pipes are chosen in this circuit, which length is set to 40 m. The total energy consumption to drive Pump 1 becomes 196 kWh/yr, from the same script. Pump 2 will be equal to what was found in case B, with a total energy consumption of 639 kWh/yr.

The required accumulating volume is 9.007 l (9.000 l is assumed to be sufficient), found from equation (24)-(26) at the maximum heat load. On top comes the power required to run the TDS, set to 21,5 kW from equation (4) and (5) at the given production rate of 560,4 m^3/day. With a running time of 36,43 days/yr, the energy consumption of the snowmaker becomes 18.798 kWh/yr.

5.5 Costs

The investment costs of case C are estimated in Table 23.

Table 23: Estimated investment costs of case C.

Combined heating and cooling plant	4.375.000 NOK
Building	20.000.000 NOK
Ditches and pipes	2.288.000 NOK
Pumps	23.700 NOK
Accumulators	104.700 NOK
Building centrals	375.000 NOK
Engineering	4.794.071 NOK
SUM	**31.960.471 NOK**
Estimated costs	**32 MNOK**

Grounds of the estimations are given below. Unless public available, the sources or suppliers will be held anonymous:

- **Combined heating and cooling plant**: Phone conversation with supplier: 2.000 NOK/kW x 1.750 kW, plus 25% taxes.
- **Building:** Phone conversation with a supplier: 8000 NOK/m^2 x 2.500 m^2 floor area, including installation of the electrical system.
- **Ditches and pipes**: From a report, DN65 twin pipes at 4.000 NOK/m x 550 m and DN80 pipes at 2.200 NOK/m x 40 m [91].
- **Pumps**: Estimated costs of 2.500 NOK/kg/s based on prices from www.vvskupp.no and www.amazon.com: Pump 1 = 5.000 NOK and Pump 2 = 18.700 NOK.
- **Accumulators**: Price list from supplier: 3 x 3.000 l tanks at 34.900 NOK/tank [92].
- **Building centrals**: From a report: 3 x 100 kW centrals, one for each building, at 125.000 NOK/central [91].
- **Engineering**: Design, freight and mounting are set to 15% of the total costs, based on a report [91].

This gives a cost of 12.800 NOK per m^2 floor area of the hall. As a comparison, the average investment costs of five ski tunnels built after 2004 were 7.781 NOK/m^2, ranging from 4.642 NOK/m^2 to 14.870 NOK/m^3 [94]. Note that an investment in case C without heat recovery would not be justifiable, as it would only decease the investment costs by 6%.

5.6 Summary and discussion

The SPF of case C, with respect to the useful heat output, is found from equation (29):

$$SPF = \frac{Q_{Demand}}{W_{CH} + W_{Pumps} + W_{TDS}} = \frac{1.504.500\ [kWh]}{532.397 + 835 + 18.798[kWh]} = 2{,}73$$

This SPF is higher than for case B, partly because the surplus heat only has to pass through one heat exchanger to enter the water-circuit, compared to two in case B. For every heat exchanger, temperature drops will occur, due to the approach temperature. Furthermore, although the final amount of snow is more than required, no energy is wasted as in case B. The energy gain becomes 952.470 kWh/yr, which means savings of 761.976 NOK/yr, at 0,8 NOK/kWh. The EVR and CVR related to distribution is set equal to what was found in case A, although this case (as case B) does not involve removal and disposal of sawdust prior to distribution. More on this in the conclusion. Case C is summarized in Table 24.

Table 24: Summary of case C, based on a final snow volume of 20.335 m^3.

Case C	**Snowmaking system**	**Distribution**	**SUM**
Investment costs	32 MNOK	-	32 MNOK
Operating costs	-761.976 NOK/yr	1.106.834 NOK/yr	344.858 NOK/yr
EVR	-46,84 kWh/m^3	11,08 kWh/m^3	-35,76kWh/m^3
CVR	-37,47 NOK/m^3	54,43 NOK/m^3	16,96 NOK/m^3

As for case B, the investment costs are high compared to the CVR, which is positive due to distribution. Note that the CVR would be negative if only the required 12.000 m^3 were distributed. The EVR is better than in case B, due to a higher evaporation temperature, no water losses assumed and small melting losses. This hints at case C being too optimistic, and especially the water losses can be of importance. It must be noted, however, that a higher refrigeration load would not serve as a problem if the surplus heat could be utilized in a heat recovery system. After all, the amount of snow produced in this case is 60% more than required.

As for the other cases, rapid melting can occur after distribution, but case C has the opportunity to produce supplementary snow at a high production rate. Hence, supplementary snow can be produced on short notice if required, not to mention the 8.000 m^3 of surplus snow produced.

The difference in snow quality from the other cases is questionable. Initially, the snow quality should be good, but after continuous melting and refreezing, it will degrade. Also, after the snow is distributed, the quality of the snow can rapidly degrade at warm temperatures, as happened in Granåsen in 2015.

A possible problem is that snow can be clogged in the fans of the air coolers. Also, the defrosting may cause problems. As such large volume flow rates of water is injected into a cold and humid hall, ice formation on the air coolers could be a challenge, leading to frequent defrosting and a lower SPF. However, ski halls report that defrosting is only necessary for every 4-5 hours (see section 2.4) and the maximum time of operation will not be that long. If problems with defrosting were more severe, two individual rooms could be constructed, such that snowmaking could happen in one room, while the other was defrosted.

The shape of the snowmaking hall should also be considered. For example, a cylindrical tower would provide a long air time for the droplets and ensure good contact with a cold air stream. After the snow would fall from the tower, it would form a pile of snow with the shape of a hemisphere. Hence, a dome with a tower on top could make up a good building shape, but it would be a massive building, nearly 30 m high for storage of 12.000 m^3 of snow.

A possible improvement, as in case B, is and an internal heat exchanger to improve the thermal efficiency of the combined heating and cooling cycle.

6. Case D: Temperature independent snowmaking with indirect heat recovery

In Case D, the SF220 is analyzed in an indirect heat recovery system with BTES. The SF220 will start to run late in the summer, and is supposed to be turned off when the ski tracks are opened in November. Surplus heat is pumped down in boreholes, and will be drawn to the buildings when the demand for heat is present. A heat pump will be utilized to heat water from 15-70 °C, as in case B and case C. The idea behind this case is that the melting losses would be decreased, compared to case B.

A schematic diagram of the indirect heat recovery system can be seen in Figure 42. The additional air cooler after the upper heat exchanger of the heat pump is to ensure sufficient cooling. The heat carrier fluid flowing in the boreholes is chosen to be water with 25% Monoethylene Glycol (MET). Water is a good heat carrier fluid when such large volumes are required, and the MET is added to depress the freezing point.

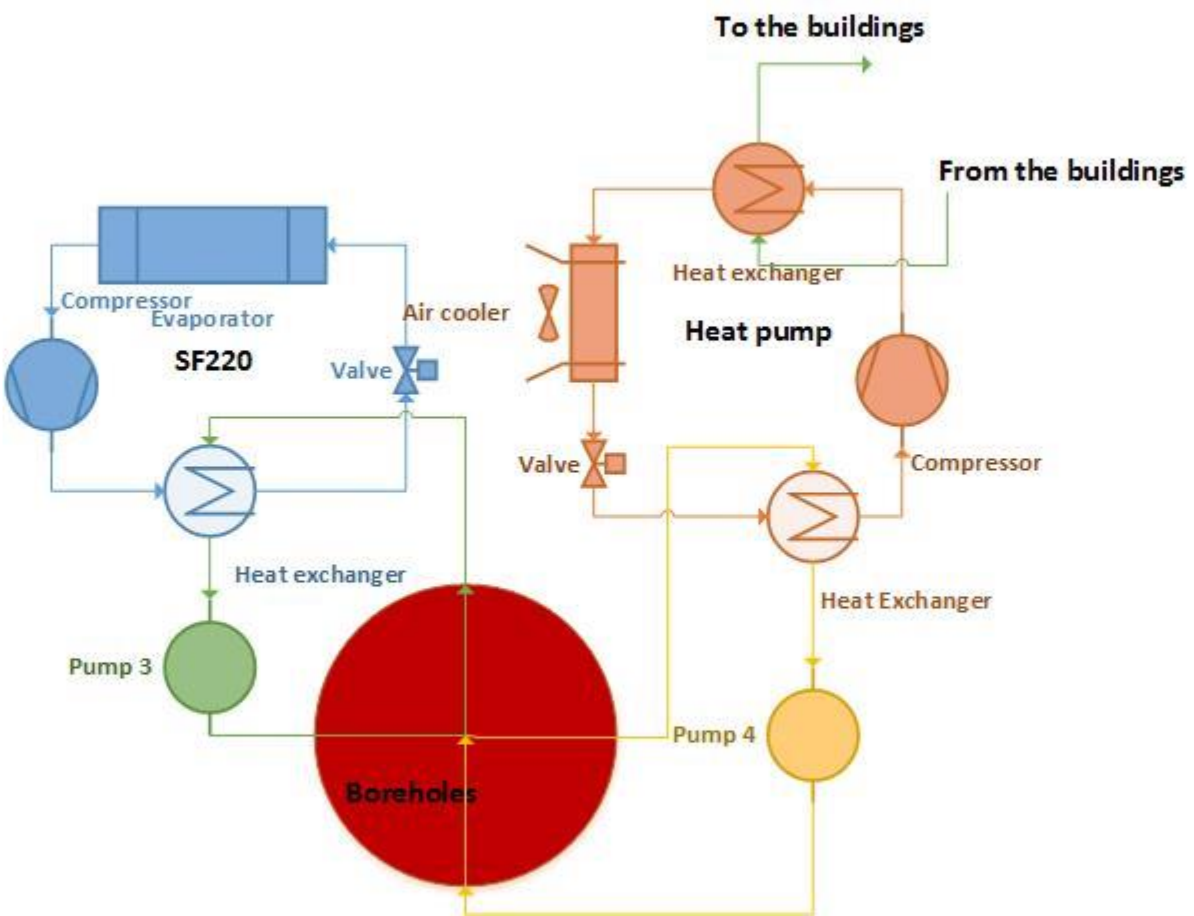


Figure 42: A schematic diagram of the indirect heat recovery system in case D, with the SF220-circuit (blue) and the CO_2-heat pump (red). The water-circuit to the buildings is as in case B and case C, and the fluid through Pump 3 and Pump 4 is MET.

A model of the indirect heat recovery system in Granåsen is depicted in Figure 43. The boreholes will be placed on a flat spot in middle of the buildings to minimize the total distance of ditches, which is estimated to 550 m, as in case B and case C. The SF220 and the heat pump will be placed in a small building next to building A with the same dimensions as in case B.

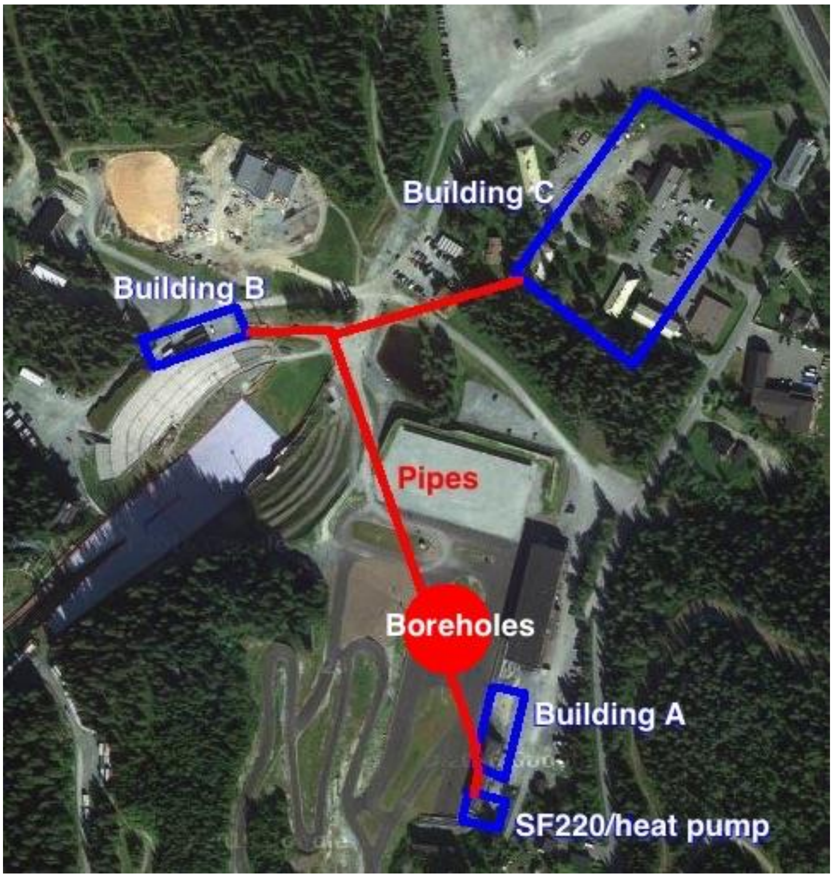


Figure 43: Case D, seen from above. Heat is transferred from the SF220 to the boreholes, and from there to the buildings.

6.1 Snow volume

As the SF220 can produce 220 m^3 of snow per day, it would take 54,5 days to produce 12.000 m^3. However, some of this snow would melt during the storage period. Using the same melting model as in case B, a start of production at August 29 seems to be a good solution. The SF220 will then run continuously for 64 days, and the total melting losses are estimated to 14,1%, such that a little more than 12.000 m^3 of the produced 14.080 m^3 of snow will remain at November 1.

The total surplus heat available from the SF220 is then 1.137.408 kWh/yr, and the work input to the compressor over the same period is 348.672 kWh/yr. The monthly amounts of surplus heat do not coincide well with heat demands of the buildings, as seen in Figure 44. Thus, a BTES system is appropriate for long term storage of heat. The net surplus heat to be stored in the ground when the SF220 is turned off in late October is 937.862 kWh/yr.

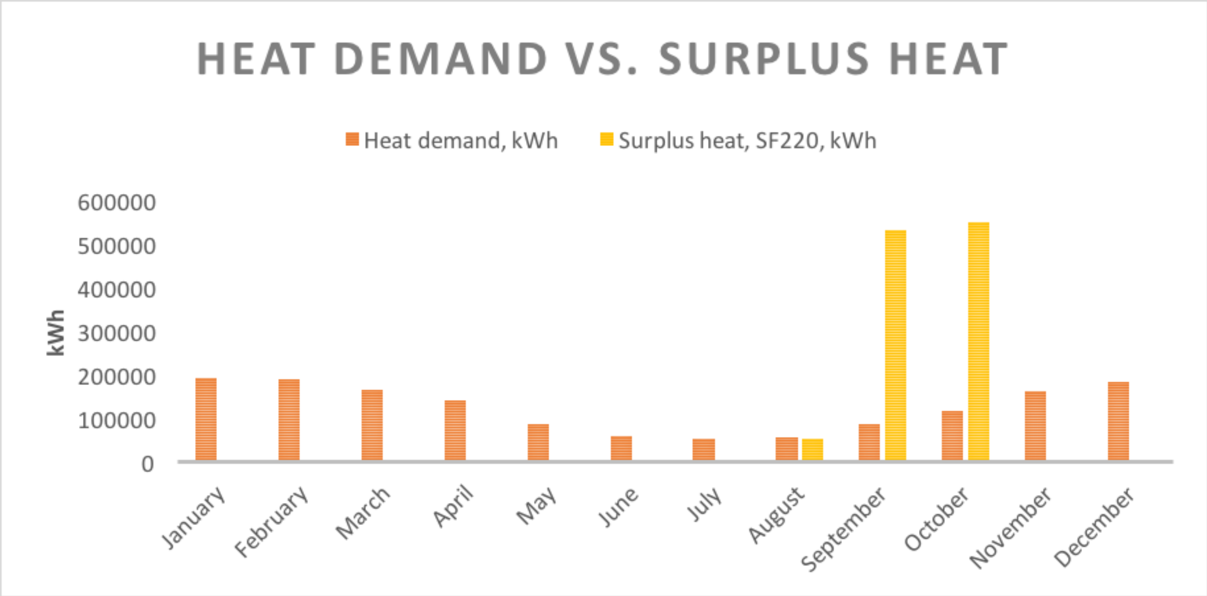


Figure 44: Monthly heat demands and surplus heat from the SF220 in case D.

6.2 BTES system

To simulate the amount of boreholes required in the BTES system, the software Earth Energy Designer (EED) is used. The most important input data to the model is the monthly heating and cooling loads, the thermal properties of the heat carrier fluid and the thermal properties of the ground, such as the volumetric heat capacity. All of the input data is listed in Appendix H.

The bedrock in Granåsen is classified as Greenstone. The sediments at the site varies between a thick layer of peat and bog and a thin layer of weathered rock. No exact examination of the ground conditions is made exactly at the planned site of the wells, but the thickness of the sediments is measured to be up to 8,3 m at the ski arena’s parking space (see maps of the ground conditions in Granåsen in Appendix G). However, the thermal properties of the ground are assumed to be constant. Effects of groundwater movements is also neglected. This is a crucial assumption to check. A thermal response test therefore has to be performed prior to an eventual construction.

The challenge with this BTES system is that the temperature of the MET cannot have large variations, as it has to be able to both cool the SF220 and heat the buildings. A lower temperature limit of the MET of more than 5 °C will require an unacceptable amount of boreholes, and as natural water sources can hold 4 °C throughout the year, this temperature level would be of little value. Hence, the lower temperature limit of the MET is set to 5 °C. To avoid large mass flow rates of MET, the approach temperature in the heat exchangers involving MET is set to 5 K. Also, the minimum temperature change of MET through the heat exchangers is set to 5 K. Thus, the evaporator temperature of the heat pump becomes -5 °C, and the MET has to lay in the range between 5-15 °C.

The optimal solution is found in EED to be 90 boreholes, in a 9 x 10 shaped rectangle, with 6 m spacing between the boreholes. This will require an area of 48 x 54 m^2. As the available space on the ski arena is more than 100 x 100 m^2, the area requirements are fulfilled. The depths of the boreholes are 300 m, which gives a total borehole length of 27.000 m. The monthly mean temperatures of the MET after 25 years of operation can be seen in Figure 45.

Figure 45: Monthly mean temperatures of the MET in year 25 of operation.

For a balanced system like this, where the amount of heat ejected equals the amount of heat injected, the temperature curve will be constant over the years. The mean temperature of the MET, T_{Mean}, is the average of the temperature in and out of the wells, T_{In} and T_{Out} when the temperature in each well is assumed to be equal:

$$T_{Mean} = \frac{T_{In} - T_{Out}}{2} \tag{31}$$

The difference between T_{In} and T_{Out} will vary from 0-4 °C, depending on:

$$\Delta T = \frac{q_B H}{2\dot{m}cp_{MET}} \tag{32}$$

Where H is the depth of the well and q_B is the specific heat extraction rate in W/m. The maximum and minimum temperature of the MET is 13,6 °C and 5,1 °C at the end of October and March respectively. At these critical operation points, ΔT is close to zero. Thus, the temperature of the MET is assumed to be given by T_{Mean} in Figure 45.

6.3 Heat pump

The heat pump cycle is drawn in a TS diagram in Figure 46 and details at the state points are given in Table 25.

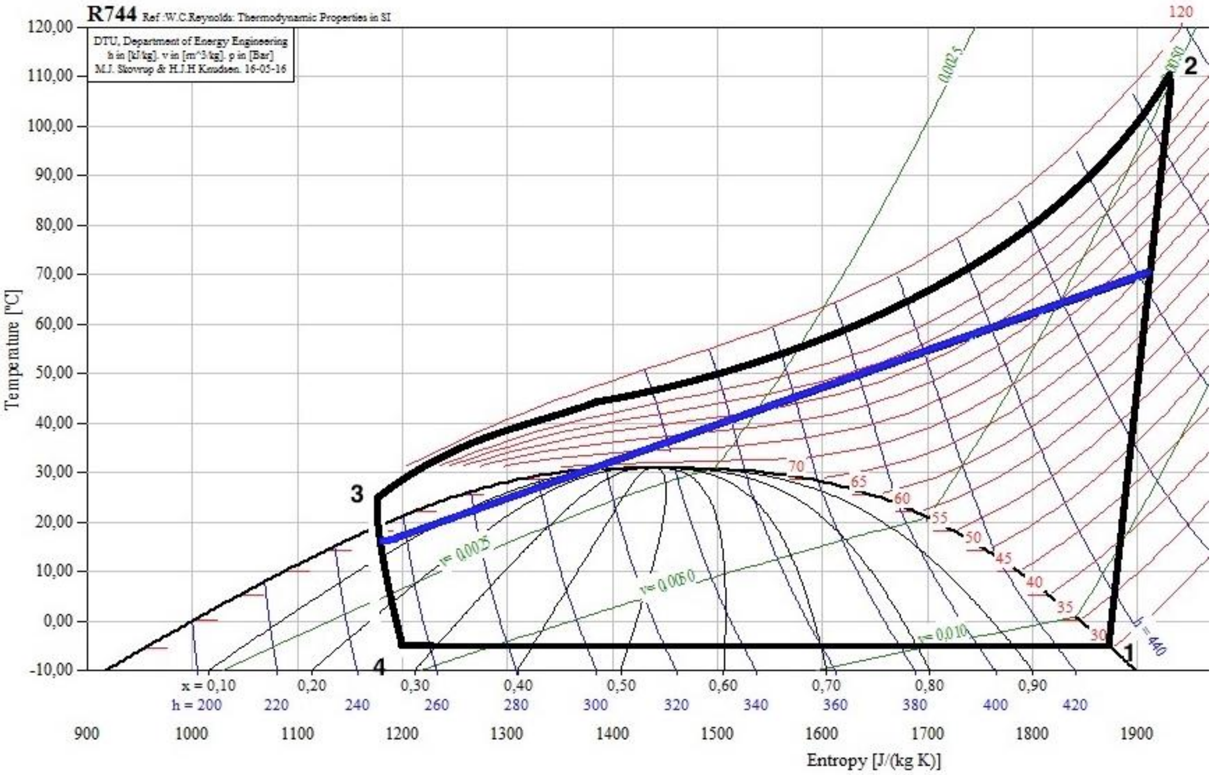


Figure 46: TS diagram for the heat pump cycle in case D.

Table 25: Details of the heat pump cycle in case D.

State point	Pressure	Temperature	ΔH to the next point
1	30,5 bar	-5 °C	76 kJ/kg
2	108 bar	108 °C	-255,3 kJ/kg
3	108 bar	25 °C	-
4	30,5 bar	-5 °C	179,3 kJ/kg

The pressure ratio of the cycle is 3,54, and the isentropic efficiency is set to 0,7 according to Figure 36. The evaporation temperature is already discussed, and the pressure of the gas cooler is set to 108 bar, to be able perform sufficient heating of the water. As 1.137.408 kWh/yr is sent into the heat pump's evaporator, 1.615.119 kWh/yr comes out of the gas cooler. This is about 7% above the estimated heat demand of the buildings. Some of this heat is assumed to be lost in the the distribution, and the rest will be cooled by the air cooler. The work required to the heat pump is 482.114 kWh/yr. As the maximum heat load to the buildings are 470 kW, the maximum refrigeration capacity of the evaporator is 330 kW. All the values above are calculated with data from Table 25 and equation (11), (12) and (20).

Due to the new approach temperature in the heat exchangers involving MET, these heat exchangers are examined by equation (7) and (8), in Table 26.

Table 26: Examination of the heat exchangers involving MET in case D.

	SF220 condenser	**Heat pump evaporator**
Q_{Max}	740,5 kW	330 kW
T_{In}**, critical**	13,6	5,1
T_{Out}**, critical**	20	0
ΔT_{LM}	7,77	7,25
Mass flow rate, critical	30,49 kg/s	17,05 kg/s
UA-value	95,36 kW/m^{2}K	45,49 kW/m^{2}K

For a plate heat exchanger, with a typical U-value of 5 kW/m^{2}K and an area density of 800 m^2/m^3 [95], the maximum volume of the SF220-condenser would be 0,024 m^3 or 24 l, which is unproblematic.

6.4 Pumps and accumulator

If the SF220 is running at a time with heat demand, some of the heat surplus will be pumped directly to the heat pump, and not through the boreholes. Thus, the maximum mass flow rate to the boreholes is 30,49 kg/s (Table 26). The pipes will then be split such that each borehole is connected in parallel to the network. The pipe dimensions and the power requirements to Pump 3 and Pump 4 are found from the script in Appendix F, and presented in Table 27.

Table 27: Dimensions of pipes and pumps, case D.

	SF220-boreholes	**HP-boreholes**	**Borehole-borehole**	**Boreholes**
Pipe diameter	150 mm	100 mm	65 mm	40 mm
Length	40 m	40 m	966 m	54.000 m
Maximum mass flow rate,	30,49 kg/s	17,05 kg/s	3,05 kg/s	0,34 kg/s
Power supply, Pump 3	0,6 kW	-	0,95 kW	0,83 kW
Power supply, Pump 4	-	0,8 kW	0,17 kW	0,14 kW

Pump 3 will run for 64 days, equal to the SF220, while Pump 4 is assumed to run for 60% of the time. Thus, the total energy consumption of these pumps are <u>9.490 kWh/yr</u>. The work to the water pumps, Pump 1 and Pump 2, are set equal to what was found in case B for simplicity, due to their minor contribution to the final result. The final energy consumption of all the pumps in case D becomes <u>10.289 kWh/yr.</u> For insurance, a small accumulator tank of <u>320 l</u> is mounted in building A, although the heat pump is equipped with multiple compressors for load regulation.

6.5 Costs

The investment costs of case D are estimated in Table 28.

Table 28: Estimated investment costs of case D.

Building	480.000 NOK
SF220	6.875.000 NOK
Water purification plant	1.000.000 NOK
Heat pump	1.250.000 NOK
Wells	9.000.000 NOK
Ditches and pipes	4.327.400 NOK
Pumps	134.350 NOK
Accumulator	5.900 NOK
Building centrals	375.000 NOK
Engineering	4.131.309 NOK
SUM	**27.542.059 NOK**
Estimated costs	**27,5 MNOK**

Grounds of the estimations are given below. Unless public available, the sources or suppliers will be held anonymous:

- **Building**: Phone conversation with a supplier: 8000 NOK/m^2 x 60 m^2 floor area, including installation of the electrical system.
- **SF220**: Phone conversation with a supplier: 5,5 MNOK plus 25% taxes.
- **Water purification plant**: Conversation at a plant visit, verified by a supplier: 1 MNOK. The water purification plant is due to experiences from both Sjusjøen and Idrefjäll, regarding the Snowfactory. Dirty water from nearby lakes has caused a layer of dirt to stick to the cylindrical inside of the ice machine, causing problems. Hence, a water purification plant is required.
- **Heat pump:** Phone conversation with a supplier: 2.000 NOK/kW x 500 kW, plus 25% taxes.
- **Wells:** Phone conversation with a supplier: 80.000 NOK/well including pipes and fluid, plus 25% taxes.
- **Ditches and pipes**: From a report [91]: DN65 twin pipes at 4.000 NOK/m x 550 m, DN65 pipes at 1.900 NOK/m x 966 m, DN100 pipes at 3.000 NOK/m x 40 m and DN150 pipes: 4.300 NOK/m x 40 m.
- **Pumps**: Estimated costs of 2.500 NOK/kg/s based on prices from www.vvskupp.no and www.amazon.com: Pump 1 = 5.000 NOK, Pump 2 = 10.500 NOK, Pump 3 = 76.225 NOK and Pump 4 = 42.625 NOK.
- **Accumulator:** Price list from supplier: one 320 l tank at 5.900 NOK [92].
- **Building centrals**: From a report: 3 x 100 kW centrals, one for each building, at 125.000 NOK/central [91].

The cost per m length of boreholes becomes 1.004 NOK/m. As a comparison, the average price per m length of boreholes in Table 9 was 1.447 NOK/m, which would have given a total cost of 39 MNOK for this case. A reason for the differences can be that the design within the buildings is not included in these estimations, apart from the accumulator tank and the building centrals.

6.6 Summary and discussion

The seasonal performance factor (SPF) of case D, with respect to the useful heat output, is found from equation (29):

$$SPF = \frac{Q_{Demand}}{W_{SF220} + W_{HP} + W_{Pumps}} = \frac{1.504.500 [kWh]}{348.672 + 482.114 + 10.289 \; [kWh]} = \; 1,79$$

This SPF lays between the SPF of case B and case C. This is because the assumed heat losses are 7%, compared to 1% in case B, and over 50% in case B. Furthermore, the heat from the SF220 has to pass through 3-4 heat exchangers before entering the water-circuit, each of which adds temperature drops. The energy gain is 663.425 kWh/yr, which means savings of 530.740 NOK/yr, at 0,8 NOK/kWh. The EVR and CVR related to the distribution of the snow is set equal to what was found in case A, although this case (as case B and case C) does not include removal and disposal of sawdust prior to distribution. More on this in the conclusion. Case D is summarized in Table 29.

Table 29: Summary of case D, based on a final snow volume of 12.000 m^3.

Case D	**Snowmaking system**	**Distribution**	**SUM**
Investment costs	27,5 MNOK	-	27,1 MNOK
Operating costs	-530.740 NOK/yr	653.160 NOK/yr	122.420 NOK/yr
EVR	-55,29 kWh/m^3	11,08 kWh/m^3	-44,21 kWh/m^3
CVR	-44,23 NOK/m^3	54,43 NOK/m^3	10,2 NOK/m^3

As for case B and case C, the CVR is positive due to distribution, and although the EVR is negative, a realization of the case is not likely to happen due to the investment costs.

The specific energy extraction rate (SEER) of the BTES system is:

$$SEER = \frac{Net\ surplus\ heat}{Total\ length\ of\ boreholes} = \frac{937.862 \left[\frac{kWh}{yr}\right]}{27.000 \; [m]} = 36,22 \; \left[\frac{kWh}{m \cdot yr}\right] \qquad (33)$$

Normally, the SEER lies in the range of 100-140 kWh/m·yr [96]. The low SEER of case D comes from the fact that the demand for heat and the production of heat does not coincide particularly, as well as strict temperature restrictions of the MET. Thus, the investment costs of the wells will be high, compared to the amount of energy gained. Note that a raised condenser temperature of the SF220 would not help, although a lower mass flow rate could be obtained. The bottleneck is the minimum temperature limit of the MET. A decreased temperature of the heat pump evaporator would make no sense either, as natural water sources can hold 4 °C throughout the year.

The assumption that groundwater movements are neglected has to be confirmed by a thermal response test. If ground water movements are present, heat storage is not feasible, as the heat will flow with the ground water out of the storage site. Such a test will also provide accurate numbers of the ground conditions in Granåsen. In the current calculations, constant ground conditions are assumed.

The melting losses are decreased compared to case B to only 14%. However, the problem with melting after distribution applies also for this case, so an extra volume of snow should be produced as a precaution for the unpredictability of the weather. Optionally, thermal insulation or cooling of the ground under the tracks could be considered, possibly with heat recovery.

The size requirements do not serve as a problem in this case. In case B, a permanent pile of snow would be located in the middle of the ski arena, blocking parts the ski tracks. In this case, the pile will be located at the same spot, but not during the ski season.

A possible improvement, as in case B and case C, is and an internal heat exchanger to improve the thermal efficiency of the heat pump.

7. Sources of error

Before the cases are compared in the conclusion, it has to be clarified that the results obtained here are subject to insecurity. Particularly, it has the be stressed that <u>the investment costs obtained in this thesis are not fixed, and should not be taken as facts</u>. Price offers from contractors on detailed total solutions can provide more information, but the true costs of the systems can only be found after they are built and tested. Coming back to the goal of this thesis: <u>not to come up with a definitive answer, but to draw a picture of the characteristics of the cases.</u>

Although the calculations performed in this thesis are based on accepted theory of heat transfer, refrigeration technology and fluid dynamics; assumptions and simplified models are diluting the accuracy of the results. Thus, the values obtained in this thesis are not to be accepted without questions. Sources of error are listed below:

- **Heat demand**
  A heat recovery system in Granåsen would be of little value if the total heat demand of the buildings turned out to be a fraction of what is estimated here. This can be simulated more precisely when the actual sizes of the buildings are determined.
- **Melting models**
  The degree-day method is widely used, due to its simplicity, and the degree-day coefficient is based on average experimental measures of snowmelt. However, more accurate results can only be obtained from a complete energy balance. The melting losses in case A is based on experience, and the melting losses in case C are neglected.
- **Case A**
  Distribution, covering and harvesting are processes which are hard to calculate, so they are based on experience. Furthermore, the fuel consumption of vehicles is based on average values, from average speed and type of vehicle. The wages are based on average wages associated with the vehicle types.
- **Maintenance**
  The costs and consequences of operation downtimes due to maintenance are neglected.
- **Refrigeration load**
  Heat leakage through doors and windows is not included in the estimated refrigeration load.
- **Snow volume**
  The required amount of snow in Granåsen has been set to 12.000 m^3. A larger demand might be the case if the snow shall cover the ski jumping hill as well, or if the tracks are made wider and longer.
- **Pumps and compressors**
  Two different pump efficiencies have been used in this thesis 0,6 (snowmaking pumps) and 0,65 (all other pumps). This minor inconsistency was not made with intention, but it was discovered too late to rectify.
  Furthermore, the efficiency of the compressors and pumps will vary. The most favorable condition is a relatively constant load, and few start-ups/stops. The compressors, Pump 1

and Pump 3 can have as much as three start-ups/stops per hour, but the load will be relatively constant. Pump 2 and Pump 4 will have more varying load conditions, but few start-ups/stops. At these load conditions, the average efficiencies may be lower than assumed.

- **Manpower during snowmaking**

The costs the manpower during snowmaking is not yet discussed. With a fully automated snowmaking system, the goal is that no manpower is required for the task of snowmaking, apart from surveillance and maintenance.

- **The SF220**

Data for the SF220 is given at an ambient temperature of 15 °C, and the calculations depend only on the incoming water temperature, while the indoor snowmaking hall also considers the ambient temperatures. Furthermore, it is assumed that a reconstruction of the SF220 with heat recovery is possible. Other TISs than the SF220 are not discussed, and it is possible that the SF220 could be outperformed to some degree.

- **Weather**

Weather data is collected from the weather station Trondheim, Voll, which is located 8 km from Granåsen at 127 masl., while Granåsen is located at 170 masl. Thus, deviations may be introduced from the weather statistics used.

- **Heat Losses**

Heat losses during heat recovery is based on losses from the district heating network in Norway, and not simulated accurately.

- **Fouling of heat exchangers**

Heat exchangers are subject to fouling, which will decrease the amount of heat transferred by a correction factor, *F*. This springs from accumulation of materials on the heat exchange surfaces, which reduces the rate of the heat transfer. The fouling factor increases with time. Fouling is neglected in the calculations.

- **Ground properties**

The ground properties in Granåsen are assumed to be constant. A normal procedure, according to a supplier, is to take the upper 10 m of the boreholes to be sediments, and thus exclude them in the calculations.

8. Conclusion

8.1 Granåsen

The four cases, along with the scenario of running the SF220 without heat recovery, are compared in Table 30 and Figure 47. The time value of money is disregarded in all of the following comparisons.

Table 30: Comparison between the cases, along with the SF220 without heat recovery (HR).

	Case A	Case B	Case C	Case D	SF220, no HR
Investment costs	2,1 MNOK	17,2 MNOK	32 MNOK	27,5 MNOK	7,9 MNOK
Operating costs	713.760 NOK/yr	288.103 NOK/yr	344.858 NOK/yr	122.420 NOK/yr	932.040 NOK/yr
EVR	15,26 kWh/m^3	-26,95 kWh/m^3	35,76 kWh/m^3	-44,21 kWh/m^3	40,14 kWh/m^3
CVR	59,48 NOK/m^3	24,01 NOK/m^3	16,96 NOK/m^3	10,2 NOK/m^3	77,67 NOK/m^3
SPF	-	1,44	2,73	1,79	-
Snow volume	12.000 m^3	12.000 m^3	20.335 m^3	12.000 m^3	12.000 m^3

Obviously, case A has the lowest investment costs, followed by the SF220 without heat recovery, but these two cases also have the highest operating costs. All the cases involving heat recovery has a negative EVR, but due to the distribution process, none of the cases have a negative CVR. It would take 36 years before case B would equalize case A in terms of the total costs. Comparing the SF220 without heat recovery with case B, 15 years of operation would equalize the total costs. The lifespan of the solutions can be estimated to be 40 years for energy wells and 15 years for heat pumps [97]. Note that case B would perform much better if half of the surplus heat was not wasted.

The distribution process, common for all the cases, seems to have a potential for improvement. For example, the CVR of distribution in Davos (2008) was 23,4 NOK/m^3 (see section 2.5.3), and Beitostølen (2012 and 2013) had an average CVR of distribution of 23,6 NOK/m^3 [98]. The average of these two is <u>23,5 NOK/m^3</u>, and the cases are compared in Figure 48 with this updated CVR of distribution. It can be seen that all of the cases involving heat recovery now obtain negative CVRs. Still, the payback period for the case with the lowest CVR, case C, would be more than 40 years. Also, as all the cases would benefit from the improved distribution process, the time before equalization would be the same as before, 36 years. Based on this, case A is the best alternative in Granåsen among the cases considered. However, with a negative CVR it would be beneficial to produce as much snow as possible in a continuous operation, given that a large heat demand was present. This will be analyzed further in the next section.

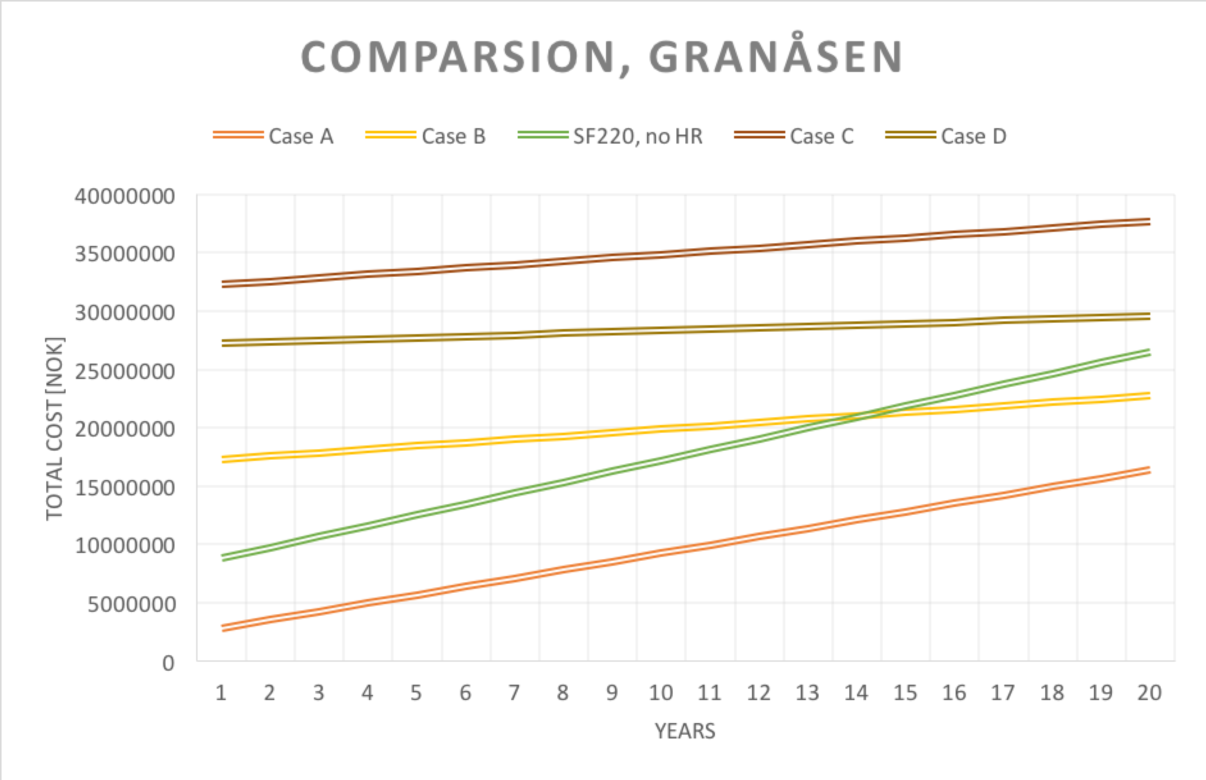


Figure 47: Cost development of the four cases, along with the SF220 without heat recovery.

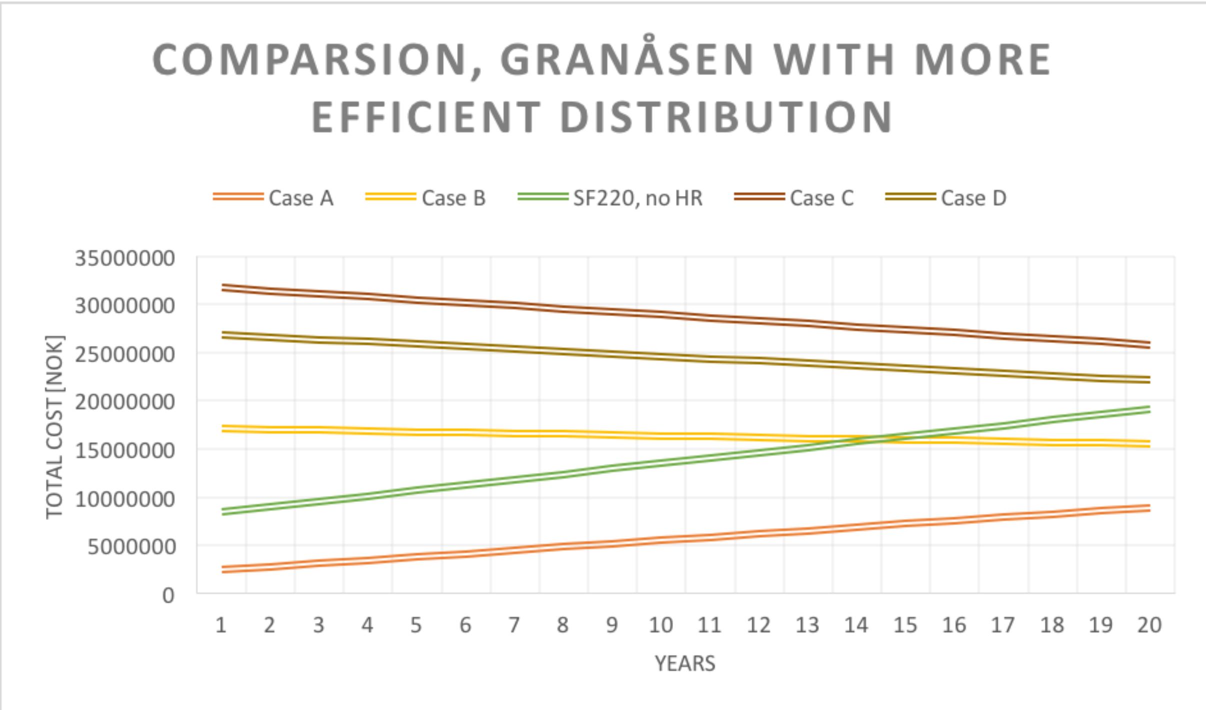


Figure 48: Cost development of the four cases, and the SF220 without heat recovery at a decreased CVR of distribution of 23,5 NOK/m^3.

8.2 Continuous operation

None of the cases in Granåsen has a perfect match between the demand for snow and heat. Case B and case D waste heat to produce enough snow, and case C produces extra snow to meet the heat demand. With a positive CVR, it would not make sense to produce more snow than the required 12.000 m^3. However, at a negative CVR, a continuous operation of a heat recovery system would be beneficial if the surplus heat could be utilized. To decide between indoor snowmaking or snowmaking from a TIS (SF220), a comparison based on case B and case C in continuous operation is made in Table 31. The indoor conditions of the hall are assumed to be as in case C, with 10 minutes of defrosting for every 4 hours and thus a production time of 350 days/yr. The indoor melting losses are neglected, and the melting losses of the SF220 are found from the melting model is case B.

Table 31: Comparison between the SF220 and indoor snowmaking at continuous operation.

	SF220	**Indoor snowmaking**
Snow capacity	80.300 m^3/yr	196.140 m^3/yr
Snow after melting	46.239 m^3/yr	196.140 m^3/yr
Surplus heat	8,21 GWh/yr	12,03 GWh/yr
Savings	3,55 MNOK/yr	7,12 MNOK/yr
SPF	2,19	2,73

The SPF of the SF220 is increased compared to case B, as no surplus heat is wasted. It could also be increased further by covering the pile of snow. If the investment costs are held fixed, the payback period will be less than 5 years for both alternatives, but indoor snowmaking has the best overall performance. Indoor snowmaking comes with more options for regulating the production rate of snow, and thus more control is obtained. For example, a lower temperature in the hall would increase the production potential, as would a higher refrigeration capacity. Furthermore, the minor melting losses is an advantage. The only limit with indoor snowmaking is the size of the hall. Such large amounts of surplus heat as shown in Table 31 would be necessary to justify the investment costs. The question is: how much surplus heat can be utilized, and what is the demand for snow?

Due to the benefits of low melting losses and regulating options, indoor snowmaking is recommended in a continuous operation with heat recovery. Furthermore, a continuous operation would probably require long term storage of heat, possibly in a BTES system. Hence, a combination of case C and case D could be an excellent solution. Required though, would be a large heat demand in close range.

8.3 Recommendations

To create or store snow at temperatures above 0 °C is not for free, but if the sport of skiing is to survive, it is a necessary strategy. The effects of global warming are already highly noticeable at ski arenas/resorts where the climate predictions leave little doubt that the days of winter will continue to decrease.

In Granåsen, an accessible heat demand for a continuous operation of the cases is not present, so case A is recommended. The focus in the design of the system for snow supply in Granåsen should be on the logistics. Automation of snow production and a system for distributing the snow seems highly important. From the average monthly electricity prices in Trondheim in Appendix C, there is little to gain on basing the production rate on the expected fluctuations of the electricity prices.

Generally, case C in a continuous operation, possibly in combination with case D, is the most attractive solution if a large heat demand is present. The required heat demand depends on the accepted payback period and the demand for snow. Locating future ski arenas/resorts near heat demanding industry, shopping malls or similar could be a good idea, which in turn would move the ski tracks closer to populous areas. Moreover, locations without snowmaking conditions for TDSs have to rely on other methods than case A.

Note that ecological impacts such as CO_2-emissions are not covered in the analysis, apart from energy consumptions.

9. Suggestions for further research

To develop the optimal system for snow supply, there are several interesting areas which should be gathered more information about:

- **Logistics**
  The logistics of covering and distribution of snow should be assessed. How can the perfect distribution system be implemented? What about using conveyors?
- **Ecological analysis**
  A comprehensive ecological analysis should be performed, with regard to various systems for snow supply.
- **A small scale indoor snowmaking hall**
  A small scale test facility in an indoor environment would be beneficial to build, to assess various aspects of snowmaking and snow storage. Production potentials, refrigeration loads, defrosting periods and melting rates are examples of measurements that could be performed.
- **TIS prototype**
  A TIS prototype with integrated heat recovery should be designed. How can a TIS be implemented with heat recovery? What about a CO_2-cycle?
- **Cover materials**
  Tests should be made on cover materials for comparison with sawdust.
- **Melting model**
  A melting model based on energy balance should be made for accurate simulations of snow melt.
- **Snow harvesting**
  Snow harvesting and the related logistics should be evaluated in depth. How could the optimal vehicle for snow harvesting for skiing purposes be designed?
- **Thermal insulation and cooling of the ground**
  Thermal insulation of the ground under the ski tracks should be evaluated, along with the possibilities for a heat recovery system from ground cooling system.

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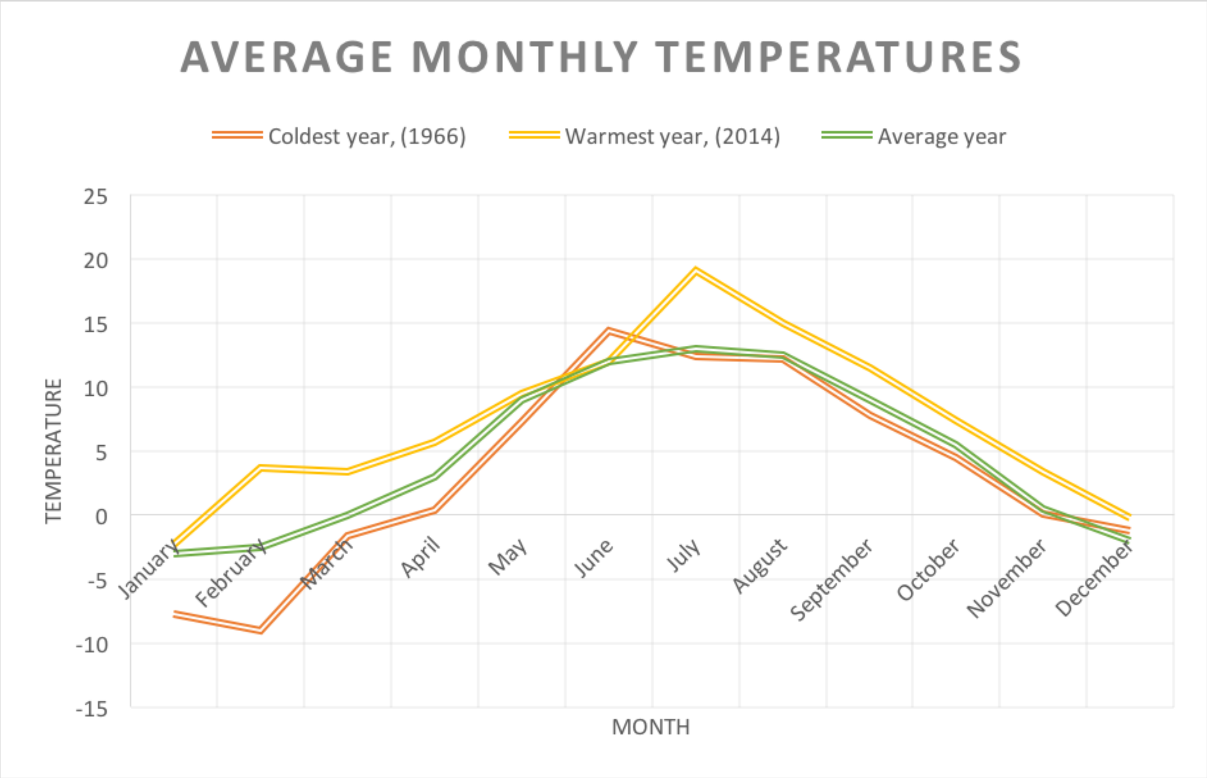
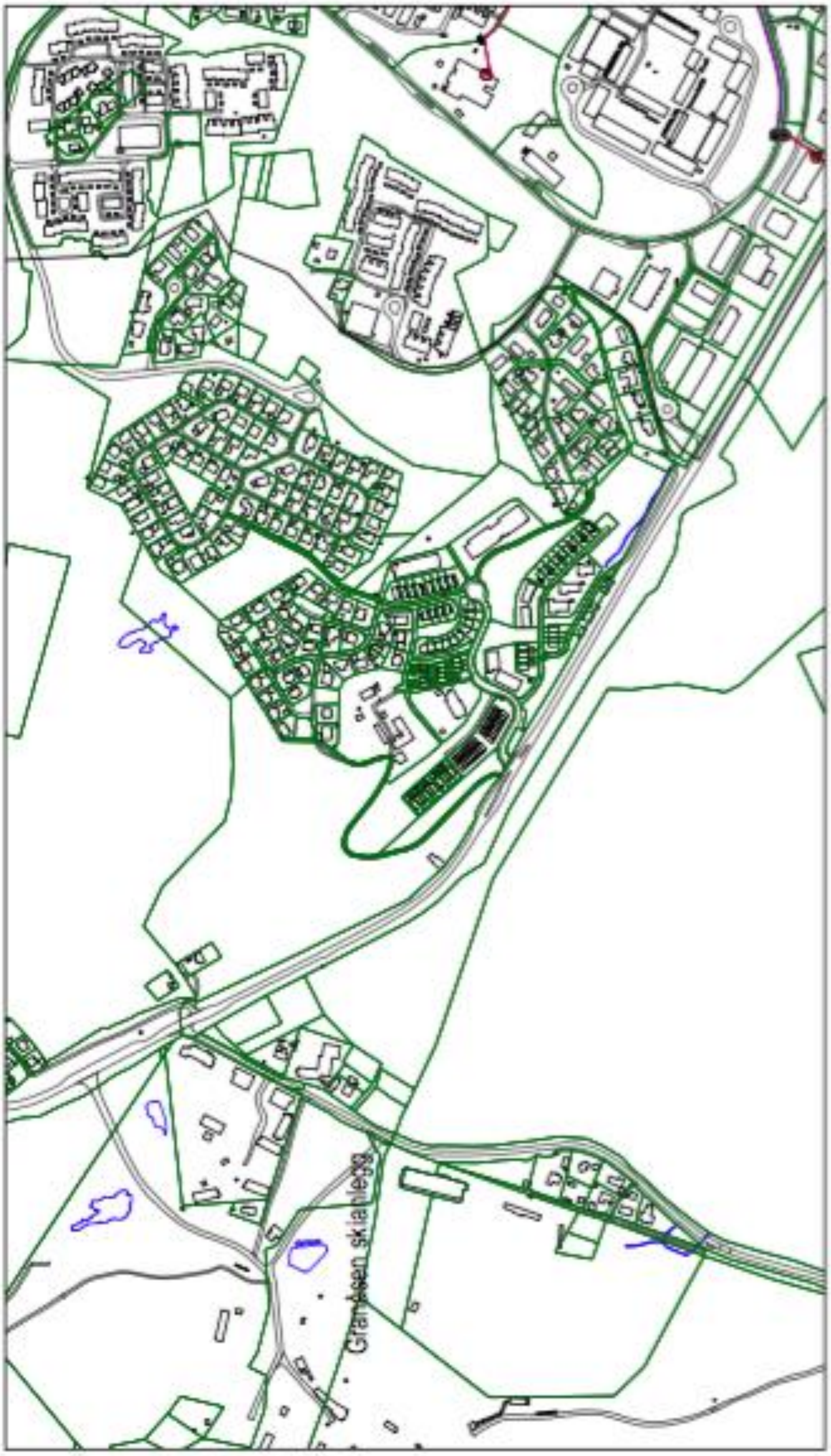
Appendix

A Temperature data



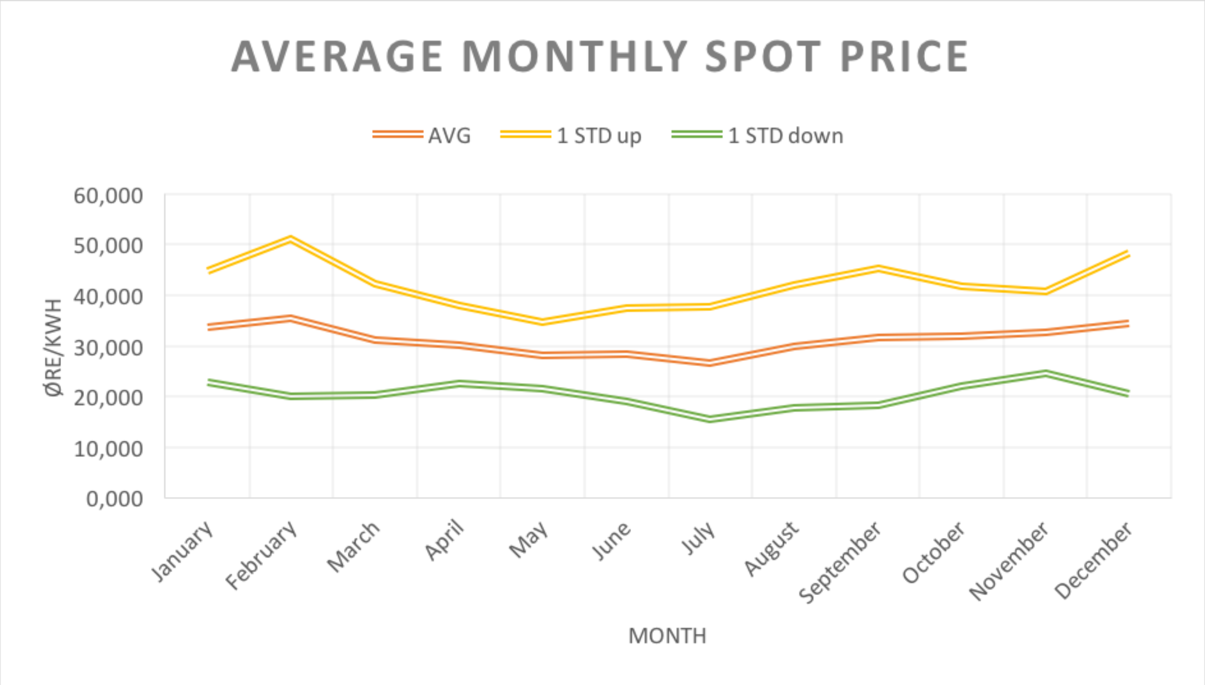
Average monthly temperatures [°C] in Trondheim since 1950. The three curves represent the warmest year (2014), the coldest year (1966) and the average year. The weather station is Trondheim, Voll, located 8 km from Granåsen. The data is collected from the Norwegian Meteorological Institute: www.eklima.met.no

B Map of district heating network near Granåsen



District heating network in green. Source: Statkraft Varme AS.

C Electricity price



Average monthly electricity spot price in Trondheim since 2005. Source: Nord Pool, www.nordpoolspot.com/historical-market-data/. The average price in red, along with one standard deviation up and down in yellow and green respectively.

The average price is 31,19 øre/kWh.

Total price:		
Grid rental	16,50	øre/kWh
User fee	16,00	øre/kWh
Spot price	31,19	øre/kWh
Taxes	15,92	øre/kWh
SUM	79,61	øre/kWh

1 NOK = 100 øre.
Grid owner: TrønderEnergi Nett AS.
Taxes: 25%.
On top comes annual grid rental costs: 4.800 NOK/yr
The average electricity price in Trondheim is set to 80 øre/NOK.

D Harvesting and distribution, Granåsen 2015

A detailed overview from the harvesting and distribution in Granåsen in 2015 follows below in Norwegian. Source: Trondheim bydrift.

D.1 Harvesting

Dato	Utstyr	Arbeidstimer
Tirsdag 14. april 2015		
3 timer	2 løypemaskiner - dosing	6
	2 kjøretøy	
Onsdag 15. april 2015		
14 timer	2 trippelaks. Lastebil (15-18 m3)	
	2 hjullastere	
	2 løypemaskiner	
	2 traktorer med hengere	
	2 gravemaskiner, 8 tonn	
	1 Wille med frontfres	154
	11 kjøretøy	
Torsdag 16. april 2015		
13 timer	2 trippelaks. Lastebil (15-18 m3)	
	2 hjullastere	
	2 løypemaskiner	
	2 traktorer med hengere	
	2 gravemaskiner, 8 tonn	
	1 Wille med frontfres	
	1 Aibi med frontfres	156
	12 kjøretøy	
Fredag 17. april 2015		
8 timer	2 trippelaks. Lastebil (15-18 m3)	
	2 hjullastere	
	2 løypemaskiner	
	2 traktorer med hengere	
	2 gravemaskiner, 8 tonn	
	1 Wille med frontfres	
	1 Aibi med frontfres	
	1 gravemaskin, 30T	
	samt 1 lastebil og 1 hjullaster i 4 timer	112
	14 kjøretøy	
Mandag 20. april 2015		
8 timer	3 lastebiler i 2 timer, deretter 2 lastebiler	
	Wille + Aibi	
	3 hjullastere i 3 timer, deretter 2 tk.	53
	7 kjøretøy	
Tirsdag 21.april 2015		
8 timer	1 hjullaster	

	1 gravemaskin, 30T	16
	2 kjøretøy	
Onsdag 22. april 2015		
8 timer	1 hjullaster	
	1 gravemaskin, 30T	16
	2 kjøretøy	
Torsdag 23. april 2015		
6 timer	1 hjullaster	
	1 gravemaskin, 30T	12
	2 kjøretøy	
		SUM: 525

D.2 Distribution

Dato	**Utstyr**	**Arbeidstimer**
Mandag 23. november 2015		
8 timer	1 gravemaskin, 30T	
	2 lastebiler	
	2 traktorer med hengere	
	1 hjullaster (lånt fra Saupstad)	48
	6 kjøretøy	
Tirsdag 24. november 2015		
13 timer	1 gravemaskin, 30T	
	2 løypemaskiner	
	1 hjullaster	
	3 lastebiler	
	4 traktorer med hengere	143
	10 kjøretøy	
Onsdag 25. november 2015		
14 timer	1 gravemaskin, 30T	
	2 løypemaskiner	
	1 hjullaster	
	2 lastebiler	
	4 traktorer med hengere	140
	10 kjøretøy	
Torsdag 26. november 2015		
8 timer	1 gravemaskin, 30T	
	3 traktorer i 2 timer, deretter 2 stk.	
	1 hjullaster	
	1 lastebiler	42
	5 kjøretøy	
		SUM: 373

E Fuel consumption and wages

Grounds for average fuel consumption and wages used in the thesis follow.

Fuel consumption, harvesting and distribution:

Vehicle	**Fuel [l/hr]**
Tractor	15
Wheel loader	21
Excavator	15
Snow groomer	25
Lorry (20 km/hr)	11
AVG	17,4

Fuel consumption, alternative harvesting method:

Vehicle	**Fuel [l/hr]**
TV2200	200
Excavator	15
Lorry x 2 (50 km/hr)	49
AVG	66

Wages, harvesting and distribution

Vehicle	**Wage [NOK/hr]**
Construction machinery	850
Snow groomer	1.250
Lorry	650
Rounded AVG	915

Wages, alternative harvesting method

Vehicle	**Wage [NOK/hr]**
TV2200	3.400
Excavator	850
Lorry x 2 (50 km/hr)	1.300
Rounded AVG	1.390

*Contributions to the TV220 is divided between fuel costs (2.550) and wages (850).

<u>The values are based on:</u>

Øveraasen AS	Fuel consumption, TV2200.
Statistics Norway (SSB)	Diesel price.
Vestlandsforskning	Fuel consumption, lorries.
Väg och transport-forskningsinstitutet	Fuel consumption, lorries and tractors
Regionale forskningsfond, Innlandet	Fuel consumption, construction machinery, snow groomers and wages, except for the TV2200.

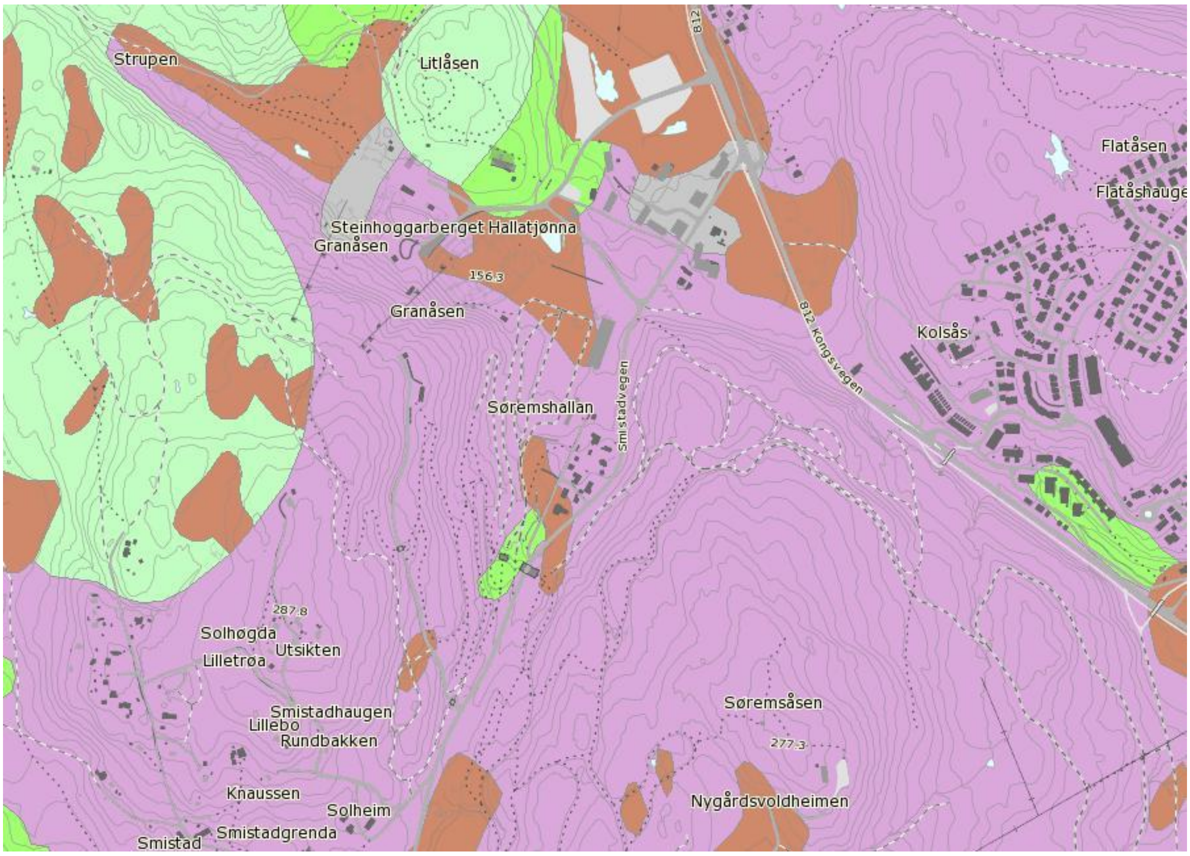
F MATLAB script

The following script were used in dimensioning the pipes and pumps.

```
%% Determination of power to Pump
clear all
clc


%% Reynold
m=1.22; %mass flow in kg/s
my=0.00653; % Dynamic viscosity, Pa*s (from engineering toolbox)
D = 0.065; % Pipe diameter, mm
ro = 992; % density, kg/m3 (from engineering toolbox, at 40 C)
V=m/ro; % Volume flow, m3/s
A=pi*((D/2)^2); % Area of pipe, m2
v=V/A; % Average velocity, m/s
Re=(ro*v*D/my) % Reynolds number


%% Darcy friction facor and pressure drop
eps=0.015; % Absolute roughness, mm,  of stainless steel pipes (from
engineering toolbox)
ed=eps/(D*1000) % Relative roughness
L=1100; % Length of pipes, m
f =0.031; % Darcy friction factor, from Moody diagram.
pd=(f*ro*(v^2)*L)/(2*D) % Pressure drop, Pa (Darcy-Weisbach
equation)


%% Pump Power
n=0.6; % Overall pump efficiency
W=(pd*V)/n % Pump power, W
pm=pd/L % Pa/m.
```

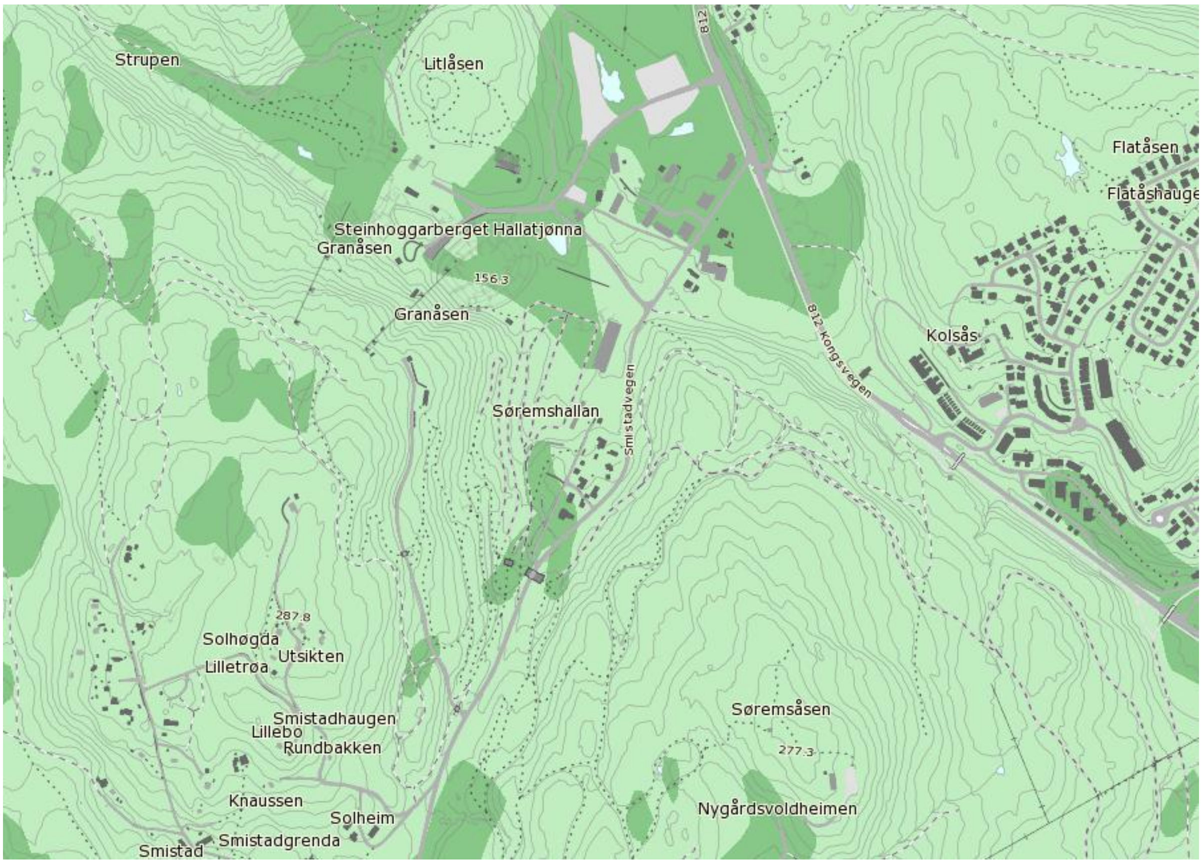
G Maps of ground conditions in Granåsen

G.1 Sediments



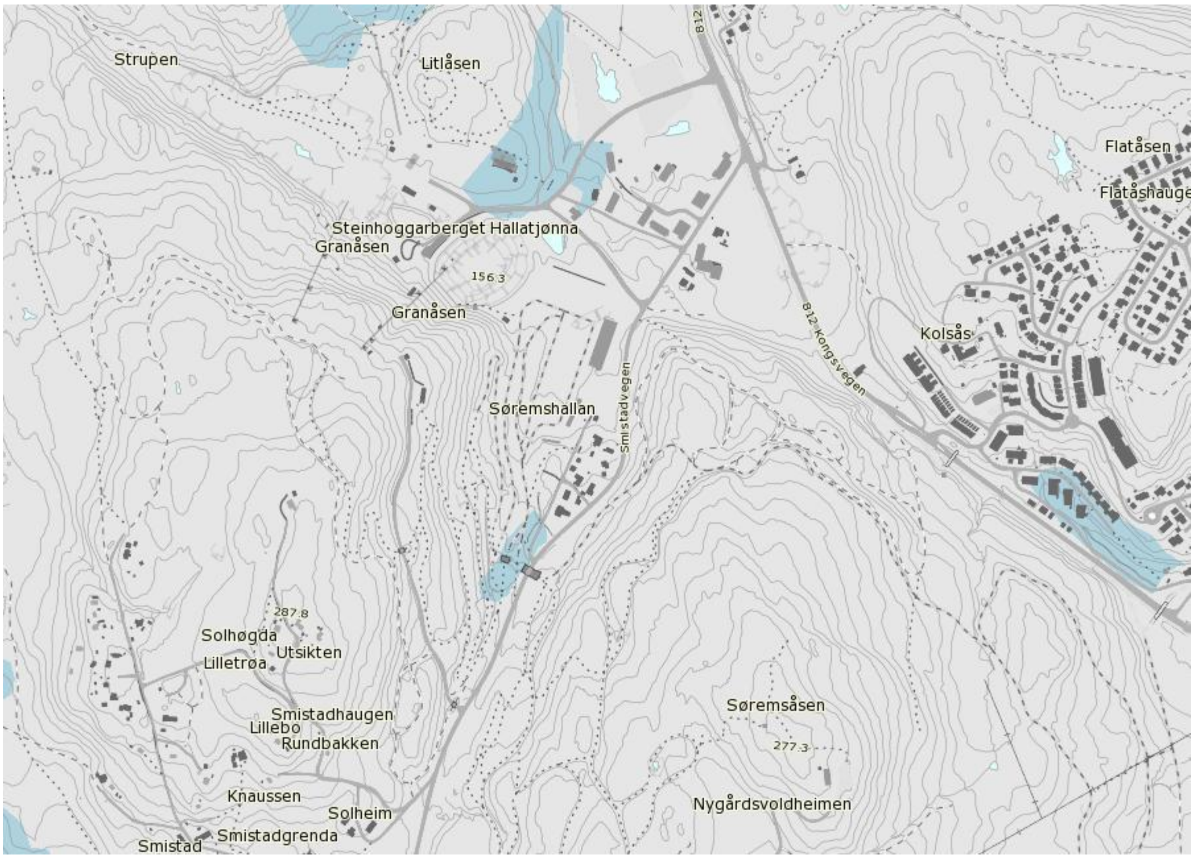
Purple: weathered rock, brown: peat and bog and green: thick moraine. Maps G.1 to G.3 are collected from the Geological Survey of Norway (NGU), www.ngu.no/kart-og-data/kartinnsyn.

G.2 Thickness of sediments



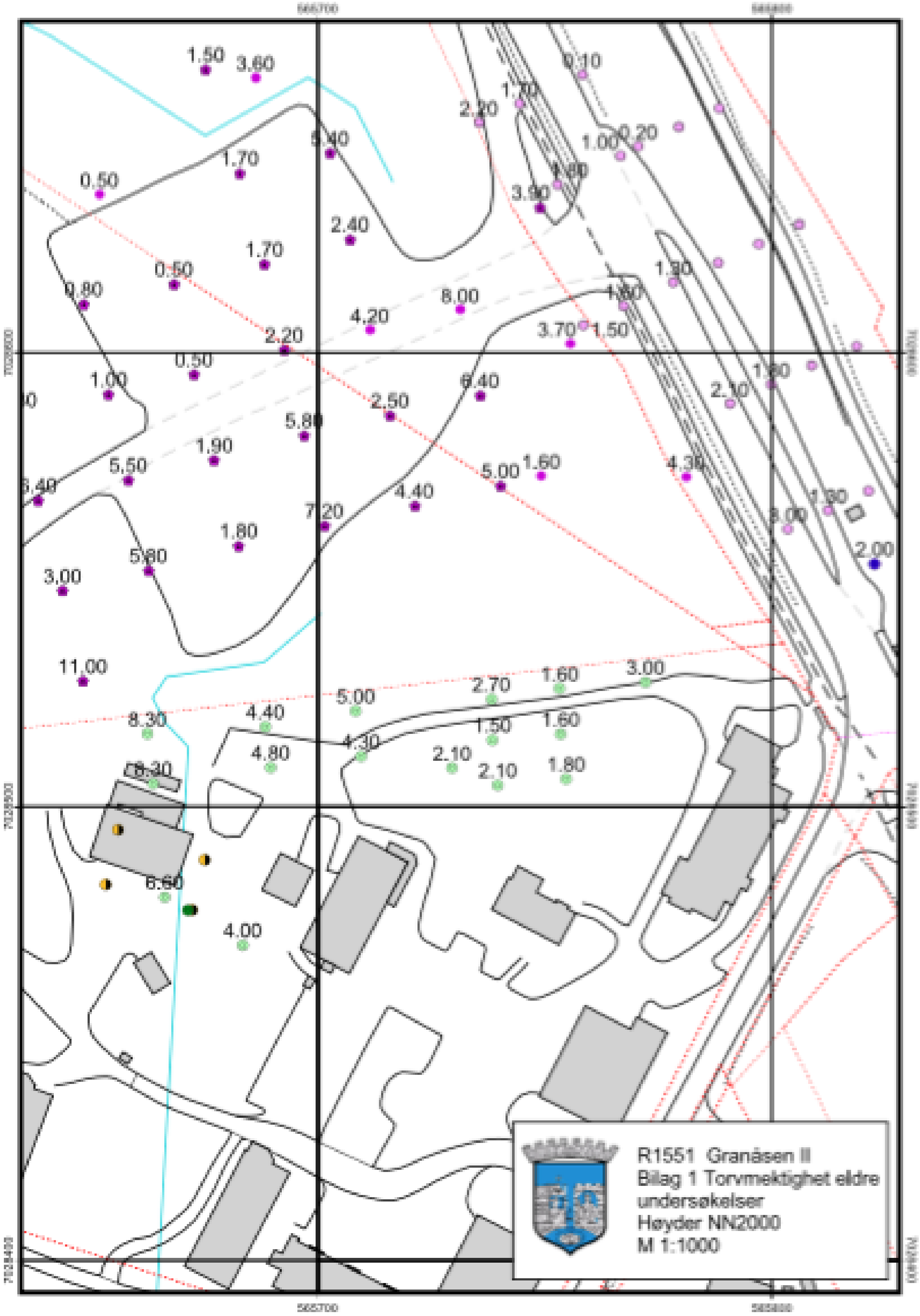
Green: thin cover and light green: thick cover

G.3 Ground water potential



Light blue: limited ground water potential and grey: no ground water potential in the sediments.

G.4 Sediment depths



Up to 8,3 m deep, just north of the ski arena. Source: Trondheim Municipality, report R.1551 Granåsen II, 2014. www.trondheim.kommune.no/content/1117741281/Geotekniske-rapporter-R.1500--.

H EED simulation data

EED 3.21 - www.buildingphysics.com - license for wenche.w.finseth@ntnu.no

MEMORY NOTES FOR PROJECT

QUICK FACTS

Number of boreholes	90
Borehole depth	300 m
Total borehole length	2,7E4 m

D E S I G N   D A T A

=====================

GROUND

Ground thermal conductivity	3,5 W/(m·K)
Ground heat capacity	2,16 MJ/(m³·K)
Ground surface temperature	4,7 °C
Geothermal heat flux	0,05 W/m²

BOREHOLE

Configuration:	556 ("90 : 9 x 10 rectangle")
Borehole depth	300 m
Borehole spacing	6 m
Borehole installation	Single-U
Borehole diameter	139,7 mm
U-pipe diameter	40 mm
U-pipe thickness	2,4 mm
U-pipe thermal conductivity	0,42 W/(m·K)
U-pipe shank spacing	99,7 mm
Filling thermal conductivity	0,6 W/(m·K)
Contact resistance pipe/filling	0 (m·K)/W

THERMAL RESISTANCES

Borehole therm. res. fluid/ground	0,1(m·K)/W
Borehole therm. res. internal	0,5 (m·K)/W

Internal heat transfer between upward and downward channel(s) is considered.

HEAT CARRIER FLUID

Thermal conductivity	0,48 W/(m·K)
Specific heat capacity	3795 J/(Kg·K)
Density	1052 Kg/m³
Viscosity	0,0052 Kg/(m·s)
Freezing point	-14 °C

Flow rate per borehole 0,5 l/s

BASE LOAD

Monthly energy values [MWh]

Month	Heat load	Cool load	Ground load
JAN	146,3	0	146,3
FEB	143	0	143
MAR	126,4	0	126,4
APR	106,4	0	106,4
MAY	66,51	0	66,51
JUN	46,56	0	46,56
JUL	39,91	0	39,91
AUG	0	10,08	-10,08
SEP	0	466,7	-466,7
OCT	0	461,1	-461,1
NOV	123,1	0	123,1
DEC	139,7	0	139,7
	-------	------	-------
Total	937,9	937,9	-0,035

Number of simulation years 25

First month of operation AUG

CALCULATED VALUES

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Total borehole length 2,7E4 m

THERMAL RESISTANCES

Effective borehole thermal res. 0,1 (m·K)/W

SPECIFIC HEAT EXTRACTION RATE [W/m]

Month	Base load	Peak heat	Peak cool
JAN	7,42	17,41	0
FEB	7,26	16,67	0
MAR	6,41	11,11	0
APR	5,4	10,37	0
MAY	3,37	5,19	0
JUN	2,36	3,7	0
JUL	2,02	3,7	0
AUG	-0,51	3,7	-27,43
SEP	-23,68	5,19	-27,43
OCT	-23,4	10,37	-27,43
NOV	6,24	11,11	0

DEC	7,09	16,67	0

BASE LOAD: MEAN FLUID TEMPERATURES (at end of month) [°C]

Year	1	2	5	10	25
JAN	6,84	5,79	5,8	5,64	5,56
FEB	6,84	5,48	5,39	5,26	5,17
MAR	6,84	5,46	5,28	5,18	5,1
APR	6,84	5,73	5,46	5,37	5,27
MAY	6,84	6,1	5,81	5,7	5,61
JUN	6,84	6,13	5,91	5,8	5,69
JUL	6,84	6,05	5,87	5,75	5,65
AUG	6,98	6,63	6,41	6,28	6,18
SEP	13,43	13,07	12,83	12,69	12,6
OCT	14,17	14,07	13,82	13,68	13,59
NOV	6,72	6,86	6,75	6,6	6,52
DEC	6,17	6,24	6,27	6,12	6,06

BASE LOAD: YEAR 25
Minimum mean fluid temperature 5,1 °C at end of MAR
Maximum mean fluid temperature 13,59 °C at end of OCT