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Heat-driven snow production applying ejector and natural refrigerant

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ABSTRACT

An effect of climate change is fewer cold days and less natural snow at lower elevations. This has spurred the interest in temperature independent snow (TIS) production, i.e., refrigeration technologies that can produce snow at ambient temperatures above zero. Commercially available TIS systems require a higher power consumption than conventional systems, i.e., snow lances and guns. Thus, to ensure that future snow-making sites are sustainable, it is necessary to develop solutions with a minimal environmental footprint. One possibility is to utilize surplus heat from industrial processes or from a district heating network to drive snow-making systems. Examples of heat driven refrigeration technologies fit for this purpose are absorption cooling and ejector cooling, both applying natural refrigerants.

This paper evaluates a solution for heat driven ejector-based snow making systems: a vacuum ice slurry system using water (R718) as refrigerant. The required amount of driving heat and its required minimum temperature level largely depend on the ejector characteristics. Thus, to enable a proper evaluation, detailed numerical simulations of the ejector design and its efficiency were performed, at different temperature levels of driving heat and ambient temperatures. Results were used as input to estimate the overall performance, in terms of specific energy consumption (per m^3 produced snow), compared to other TIS systems. The ejector-based system can be driven by low-grade heat (80 °C) and is shown to be highly efficient if cold cooling water (≤ 10°C) is available.

Keywords: Ejector, surplus heat, energy efficiency, natural refrigerants, ice, snow.

1. INTRODUCTION

Due to climate change, the reliability of natural snow and the number of potential snow production days with traditional snowmaking equipment are decreasing, especially at lower altitudes. Without snow in the proximity to cities, the foundation of many traditional winter activities is challenged. The presence of snow is also an important attraction for tourism, creating large revenues. The operation of conventional snow production technologies, such as snow guns and lances, depends on an ambient temperature below zero. To maintain conditions suitable for winter sports there is an increasing interest in snow production technologies that can provide snow at ambient temperatures above 0°C, so called temperature independent snow (TIS) production. Such compressor-driven technology exists today but has a power consumption in the range of 10 - 30 kWh/m^3 produced snow (Trædal 2017), while traditional technologies only consume 0.5 - 6 kWh/m^3 (Aalberg 2020). One way to increase the energy efficiency of compressor driven TIS systems, is to recover the surplus heat released from the condenser and use it for district heating or any nearby heating demands. Another way is to replace the compressor by using heat-driven refrigeration technologies, such as absorption refrigeration or ejector refrigeration. This study evaluates a heat-driven ejector refrigeration cycle, using water as refrigerant.

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2. THEORY

2.1. Temperature independent snow production (TIS)

TIS production is made by producing small grains of ice, with technologies originating from other areas of application, such as ice production in the fishing industry. The commercially available systems are generally based on a conventional compressor-driven refrigeration cycle, using ammonia or hydrofluorocarbons (HFCs) as refrigerant. The ice generator act as evaporator, operating between -10°C and -30°C, depending on the type of ice produced; scraped ice slurry, plate ice or flake ice. There are also a few TIS system based on vacuum ice production, originally used for mine cooling and thermal storage. A vacuum freeze evaporator operates at the triple point of water, producing an ice slurry (Trædal 2017).

2.2. Heat-driven cooling technologies

The use of thermally driven refrigeration systems dates to the 18th century, and the thermodynamic principle is illustrated in Figure 1. However, following the development of efficient electrically driven mechanical vapor refrigeration systems they were mostly outcompeted. Remaining applications were mainly for industrial processes when large amounts of waste heat are available, or in niche product segments such as absorption refrigerators for minibars due to its silent operation (Best and Rivera 2015). However, following an increased focus on waste heat recovery and renewable energy sources, such as solar heat, the heat-driven technologies are today considered for a range of different applications (Deng, Wang, and Han 2011), (Nikbakhti et al. 2020). Table 1summarizes the most common thermal technologies applicable for cooling supply below 0°C, (Moen 2021).

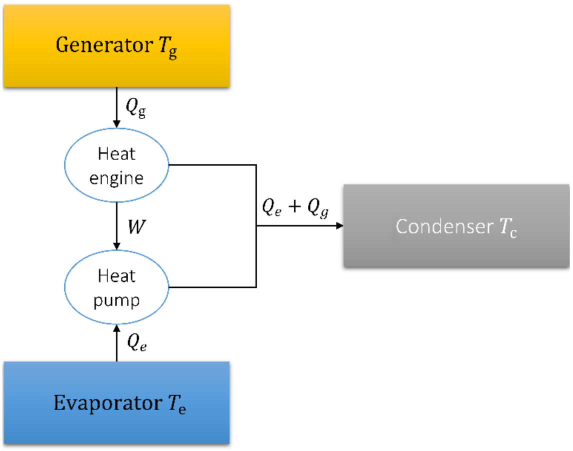

Figure 1: Basic idea of heat-driven refrigeration, showing that high-temperature energy can be used for cooling if combined with a heat engine.

Absorption based cooling is the most commercially widespread heat-driven refrigeration system (Moen 2021). In absorption systems the refrigerant is absorbed by an absorbent in a liquid phase, allowing a pump to increase the pressure of refrigerant, instead than a power-demanding compressor. Heat is used to desorb the refrigerant from the absorbent. The temperature level of the heat source should at least be 90-100°C in order to produce cooling below 0°C (Deng, Wang, and Han 2011).

Adsorption cooling systems resemble absorption systems. However, instead of being absorbed into a bulk volume of a liquid absorbent, the particles (atoms, molecules, or ions) accumulate on the surface of an adsorbate. Compared to absorption, the systems are simpler with less moving parts, and they can utilize even lower heat supply temperatures. On the downside, COP values are low (Deng, Wang, and Han 2011), and systems delivering cooling supply below 0°C are still only at an experimental level (Moen 2021).

Systems based on the Organic Rankine Cycle (ORC) have recently attracted more attention due to their use of natural refrigerants (Zeyghami, Goswami, and Stefanakos 2015) and technological development for applications such as power generation and trigeneration (combined cooling, heating and power). The principle behind ORC-based systems is to use heat to produce mechanical work, which is used to drive a conventional vapor compression refrigeration cycle. Although these systems are largely commercialized for applications such as space cooling, refrigeration below 0°C is still only at a research level (Moen 2021).

Ejector based refrigeration systems are categorized as thermo-mechanical cooling systems, where heat is used to produce the mechanical work to compress the refrigerant, as described in section 2.3 and 2.4.

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Table 1: Summary and comparison of heat driven cooling technologies (Moen 2021)

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

2.3. Ejector-driven refrigeration

Ejector technology is mostly adopted for industrial cooling above 0°C, with water as refrigerant. Advantages include simple structure, low maintenance requirements, and low installation cost (Zeyghami, Goswami, and Stefanakos 2015; Liang et al. 2020; Grazzini, Milazzo, and Mazzelli 2018). However, the COP for cooling below 0°C is generally low (0.1-0.4) compared to other thermally driven technologies (Besagni, Mereu, and Inzoli 2015), and drops significantly at operation away from design point. Still, when operating at low back pressure (condensing temperature) COP values above 1 can be achieved (Pollerberg, Ali, and Dötsch 2009). In the context of snow production, the manufacturer DemacLenko has promoted a snow making machine where the compressor is replaced with an ejector which can be driven by heat from renewable sources such as solar or biomass (Trædal 2017). This system is further described in section 2.5.

2.4. Ejectors

The ejector is a form of jet activated pump typically used for pumping gases to produce vacuum or for vapor compression (Aidoun, Giguère, and Scott 2011). Figure 2 principally shows the ejector geometry.

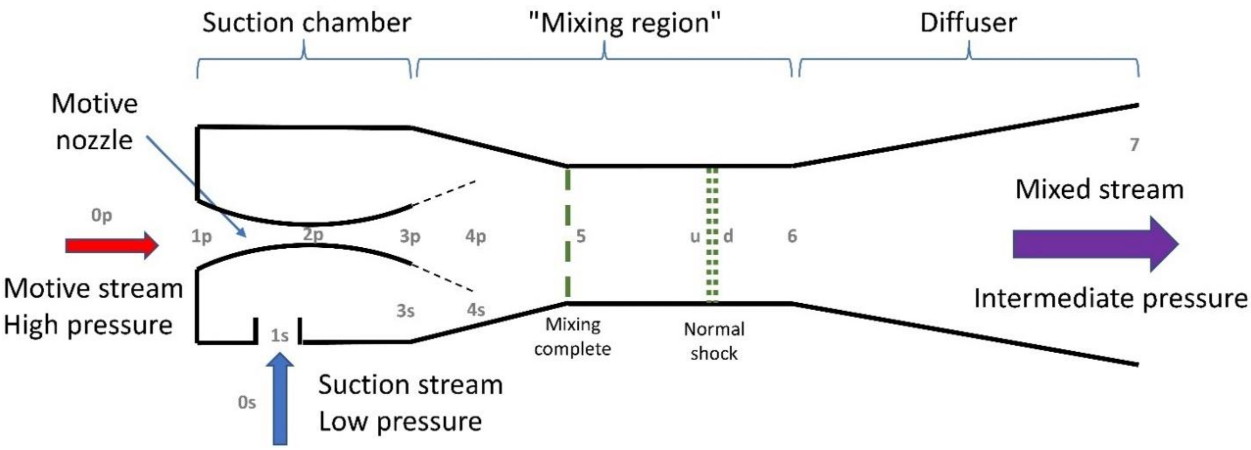

Figure 2: Ejector geometry indicating key thermodynamic states. Here, *p* and *s* refer to primary (motive) and suction stream, respectively. The motive stream chokes at *2p*. The dotted black lines are hypothetical boundaries through a region where mixing of the streams has not started, and continue until the suction stream chokes at *4s*. At state *5* mixing is assumed completed, whereas *u* and *d* represent states just upstream and downstream of a shock.

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As principally illustrated in Figure 2, a high-pressure motive stream is accelerated through the motive nozzle, creating high (supersonic) speeds and negative pressure in the suction chamber. The low-pressure suction stream is drawn in, and the two streams are mixed in the mixing region. The mixed flow then passes through a diverging nozzle (diffuser), reducing its velocity and increasing its pressure. The ejector back pressure is higher than the pressure of the suction stream, which thereby undergoes a pressure lift.The characteristics and performance of an ejector are described by the following parameters: entrainment ratio, pressure ratio, pressure lift and work recovery efficiency (Liu 2014).

Mass entrainment ratio (ER): defined in Eq. (1) as is the ratio between mass flow ($\dot{m}$) of low-pressure vapor through the suction nozzle (*SN*) and mass flow of high-pressure liquid through the motive nozzle (*MN*).

$$ER = \frac{\dot{m}_{SN}}{\dot{m}_{MN}} \qquad \text{Eq.1}$$

Pressure ratio (PR): defined in Eq. (2) as the ratio between the pressure at ejector outlet nozzle (ON) and the pressure at ejector suction nozzle. The PR can also be referred to as pressure recovery.

$$PR = \frac{p_{ON}}{p_{SN}} \qquad \text{Eq.2}$$

Pressure lift (PL): defined in Eq. (3) as the pressure difference between the ejector outlet and suction. A large pressure lift (or pressure ratio) corresponds to a low entrainment ratio, and vice versa.

$$PL = p_{ON} - p_{SN} \qquad \text{Eq.3}$$

Ejector efficiency ($\eta_{ejector}$): defined in Eq. (4) as the ratio of the recovered work (W_r) and the maximum possible (theoretical) recovered work ($W_{r,max}$). The lost work connected to expanding the motive fluid is partly recovered by the suction fluid compression through the ejector. The amount of recovered work (W_r) is estimated assuming an isentropic compression of the suction fluid, between the suction nozzle and the outlet nozzle. The theoretical amount of work recovery $W_{r,max}$ is provided by an isentropic expansion of the motive fluid, between the motive nozzle and the outlet nozzle. There is a large variation in ejector efficiencies reported in literature, ranging between 5% and 35%, but most of them are between 10-20% (Liu 2014).

$$\eta_{\text{ejector}} = \frac{W_r}{W_{r,max}} = ER \cdot \frac{h(p_{ON}, s_{SN}) - h_{SN}}{h_{MN} - h(p_{ON}, s_{MN})} \qquad \text{Eq.4}$$

The performance of an ejector refrigeration cycle depends critically on the ejector characteristics. Designing the ejector for a specific system is challenging and it must have the correct geometry for the specific application. In literature, several different mathematical models have been developed to calculate the optimal geometry and estimate the performance . For example, the ejector efficiency is largely dependent on nozzles throat area, mixing tube length and diffuser angle (Liu 2014).

Ejectors are notoriously difficult to model from first principles, as they involve several complicated physical phenomena (Grazzini, Milazzo, and Mazzelli 2018):

i. Choked flow of both the primary stream (in the motive nozzle throat), and of the suction stream (in the mixing chamber).
ii. Irreversible turbulent mixing of two streams at different temperatures and velocities. The motive stream is in the supersonic regime with Mach numbers of around 4 or 5. Before mixing, the motive stream flows more than 3000 km/h faster than the suction stream.
iii. Irreversible shock wave formation in the mixing section.
iv. Irreversible phase change processes, for example the formation of droplets as the stream accelerates through the primary nozzle.

In this work we have therefore opted for a state-based model, which for gas ejectors has proven to be rather accurate (Huang et al. 1998).

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2.5. Vacuum ice slurry production

Figure 3 shows a schematic sketch of a snow production system based on an ice slurry generator with vacuum freezing. Such a system for snow production is offered by IDE Technologies (Ide-tech.com 2020). The vacuum freezer operates at the triple point of water (0.01°C, 611 Pa). The idea is that by removing the (high-energy) vapor, the temperature in the remaining water decreases. This continues until the triple point is reached, when the water partially freezes and creates an ice slurry. The latent heat of fusion and vaporization is 333 kJ/kg and 2500 kJ/kg, respectively, which means that the mass of ice produced is about 7 times the mass of water evaporated (Van Orshoven, Klein, and Beckman 1993). To maintain the vacuum, a compressor draws out the water from the evaporator. The low water pressure results in very large volume flows and thus requires a large capacity compressor, implying a higher investment cost than other snow/ice-making systems. The vacuum ice maker production capacity depends on the feed water temperature. Each 1 K increase reduces the snow production capacity by 1,5% (Trædal 2017).

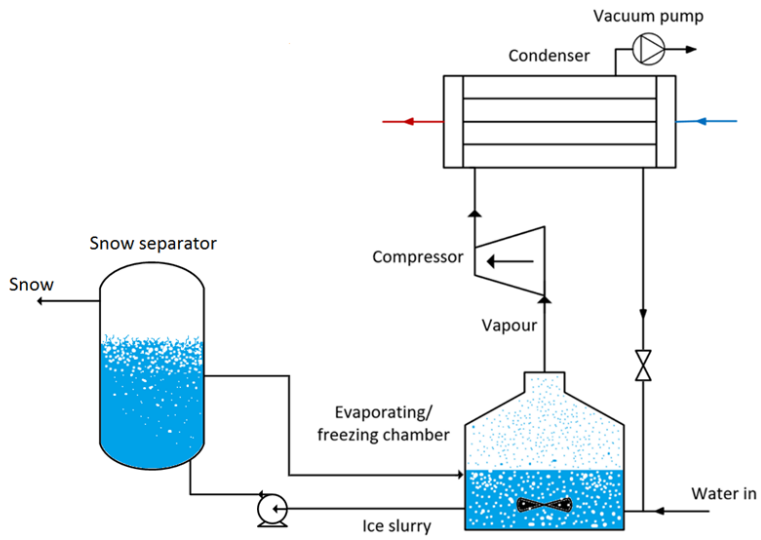

Figure 3: Principal schematic of a vacuum ice machine (Trædal 2017)

An alternative snow-making concept, based on vacuum ice slurry generation, is presented by Joemann et al. (2017). Instead of using an electrically driven compressor, a thermally driven steam ejector feeds the water vapor from evaporator to condenser. Electric power is only used for auxiliary drives such as pumps and fans. During prototype testing, where an electric boiler provides the motive steam, a thermal COP ranging between 0.1 - 0.4 was achieved with evaporation temperatures close to 0°C and steam generator temperature of around 150 °C (Joemann et al. 2017). The system, which has been demonstrated at Italian ski resorts, can produce up to 100 m^3 /day with a heating demand ranging from 45 - 90 kWh/m^3 at ambient conditions between 10 °C – 20 °C, according to the manufacturer's technical specifications (Trædal 2017). The condenser pressure represents the ejector's back pressure, meaning that a low temperature of the cooling media in the condenser decreases the back pressure, leading to reduced demand of motive steam and improved COP.

As a result of generally low COPs, to make a heat-driven system become a sustainable and economically viable option, the driving heat must be a waste heat source or a renewable energy source and be available at low price. To enable the utilization of surplus heat from district heating systems or various industries, the snow-production system should preferably operate at a lower temperature than 150°C. Generally, the supply temperature in district heating networks is between 70 °C and 120 °C and there are large amounts of industrial waste heat available at 60 °C - 140 °C (Moen 2021). Based on the system suggested by Joemann et al. (2017), the present study theoretically evaluates the system performance when operating at a lower temperature of the driving heat, with focus on the ejector design and its performance parameters.

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3. METHODOLGY

3.1. The system modelled

Figure 4 shows a simplified sketch of the investigated system, based on the heat-driven vacuum freezing prototype presented by Joemann et al. (2017).

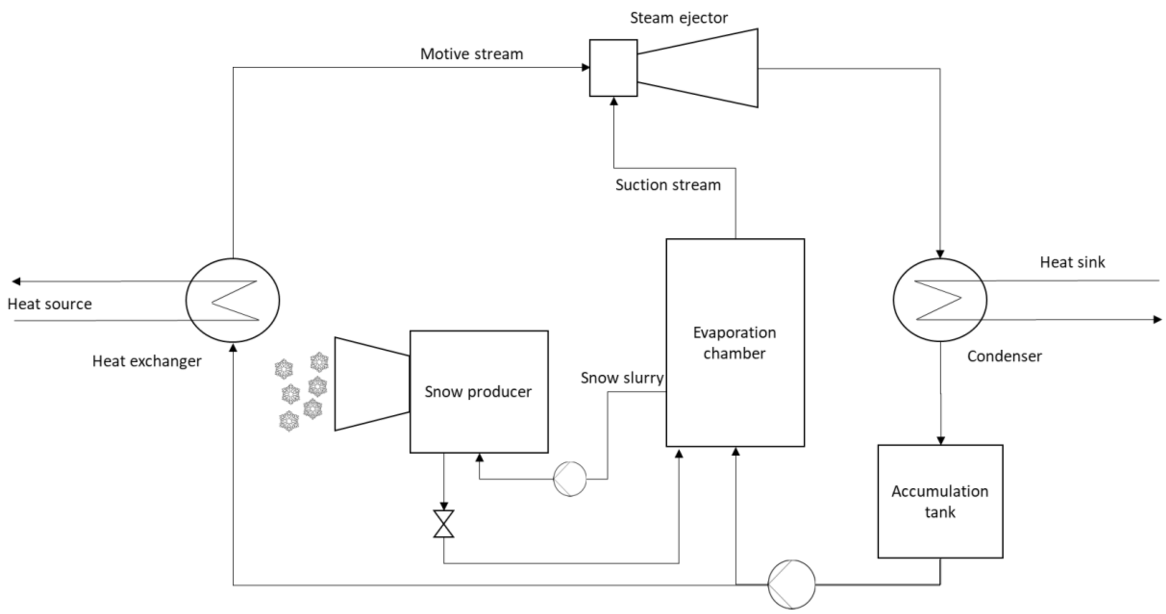

Figure 4: Schematic sketch of the investigated system, based on Joemann et al. (2017)

The following boundary conditions for the system was chosen:

- Snow production capacity: 8 m^3/h, i.e., around 200 m^3/day which is similar to several other TIS production systems (Trædal 2017)
- Temperature of the driving steam: 150°C, 110°C, 80°C, motivated by the aim of our study to investigate the possibility to use driving heat at a lower temperature than 150°C.
- Condensing temperature: 15°C and 30°C. Assuming a minimum temperature difference of 5 K in the condenser, this corresponds to a maximum cooling water temperature of 10°C and 25°C, respectively.

3.2. Energy balance calculation

Energy flow calculations for the main components in a refrigeration cycle are commonly applied to estimate the cycle performance, using assumed temperature differences in the heat exchangers and isentropic efficiencies for compressor and/or pumps. For ejector refrigeration systems, proper values of the ejector entrainment ratio or efficiency are crucial for estimating the requirements of driving heat. Based on an enthalpy balance over the ice generator together with the reported steam consumption (Joemann et al. 2017), a first estimate of the ejector performance parameters was calculated. This resulted in an entrainment ratio of 0.63 and an efficiency of 24,9%.

However, the ejector performance, and its optimal design, largely depend on the temperature of the driving heat and cooling media. Thus, to properly evaluate the required driving heat demand (i.e., steam consumption) at lower temperatures, detailed numerical simulations of the ejector design and its efficiency must be performed, as described in the next section.

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3.3. Ejector model

A simplified state-based model was implemented for the steam ejector. The model can estimate how much heat input is needed to compress a given amount of water vapor, as well as the size and shape of the various parts of the ejector. The boundary conditions defined in section 3.1, together with the energy balance calculation in section 3.2, gives the following boundary conditions for the ejector modeling:

- Suction flow: 0.16 kg/s, saturated vapor at the triple point of water.
- Motive steam temperature: 150°C, 110°C, 80°C.
- Ejector back pressure: 1.7 kPa and 4.2 kPa, corresponding to the saturation pressure at the two condensing temperatures considered, 15°C and 30°C.

As already stated, the motive steam consumption depends sensitively on the ejector characteristics, which must be calculated from a detailed model. It is also of interest to know the ejector dimensions, to estimations of investment cost and space requirements. The design procedure for steam ejectors by Samaké et al. (2016) was followed, with the following key assumptions:

- The ejector is modelled as a collection of discrete states where the flow between the states is defined by isentropic (polytropic) efficiencies. The states are shown in Table 3.
- Homogeneous thermodynamic equilibrium is valid within each stream at each state point.
- Both the motive flow and the suction flow choke at some point prior to mixing. Choking essentially means that the flow velocity is equal to the speed of sound corresponding to the local thermodynamic state. The key characteristic of choked flow is that it is not possible to increase the flow rate by decreasing the downstream pressure.
- The mixed stream transitions from super- to subsonic flow in a single, normal shock prior to the diffuser.

The model was able to accurately reproduce the motive steam flow rate of the pilot plant by Joemann et al. (2017), which had a generating temperature of 150 °C and condensing temperature of 27°C.

3.4. Comparison with other snow/ice production technologies

To enable an initial feasibility evaluation of the ejector system operating with driving heat temperatures lower temperatures than 150 °C, a simplified comparison with some existing snow/ice making systems was made, in terms of the requirements of driving energy (electric/thermal). Data for compressor-driven TIS machines was taken from the review of snow production technologies by Trædal (2017). For heat-driven absorption systems, COP values were roughly estimated for various temperatures of driving heat and cooling water. The values are based on data provided by various manufacturers for production of ice slurry at an evaporation temperature of around -10°C (Vamtec.com 2018; Colibri 2020). These COP values, presented in Figure 5as an average of the different manufacturer data, were used to estimate the energy demand for heat supply and heat rejection. Note that for a driving temperature of 80°C the cooling water must be colder than 25°C to realize cold production below 0°C.

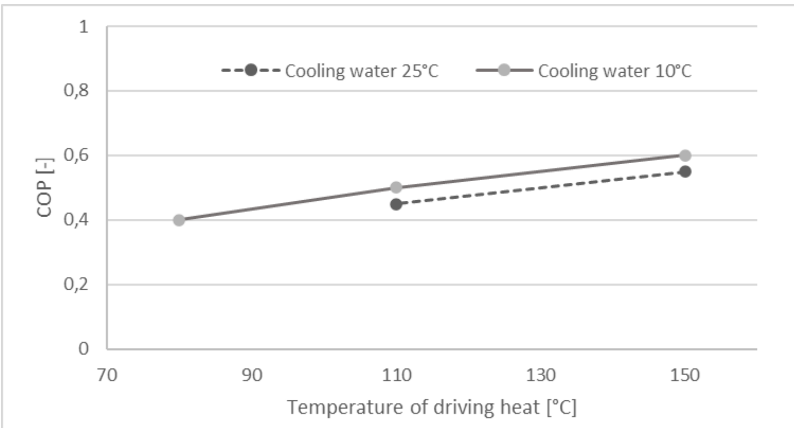

Figure 5: Assumed variation in COP for ice-producing absorption systems

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4. RESULTS AND DISCUSSION

4.1. Ejector performance

Table 2 shows the main results from the ejector simulations, in terms of the four ejector parameters defined in section 2.3, as well as the total ejector length and the required flow of motive steam.

Table 2: Results from ejector simulations at various boundary conditions

Boundary conditions		Ejector performance and design parameters					
Condensing temperature [°C]	Motive steam temperature [°C]	Pressure lift [mbar]	Pressure ratio [-]	Entrainment ratio [-]	Ejector efficiency [-]	Total length [m]	Motive flow [kg/s]
30	150	36,3	7,0	0,46	0,213	3,92	0,345
	110			0,35	0,215	4,23	0,461
	80			0,21	0,209	5,27	0,764
15	150	10,9	2,8	1,68	0,318	3,52	0,095
	110			1,42	0,344	3,64	0,113
	80			1,15	0,376	3,78	0,139

As seen, when comparing the results for the two different condenser temperatures, the results are extremely sensitive to the outlet pressure of the ejector. Much more efficient ejectors can be designed if the outlet pressure corresponds to 15 °C than for 30 °C, and considerably less high-pressure steam is needed. This can be understood from the ratio of the ejector back pressure to the suction pressure, which equals 2,8 for condensing at 15 °C and 7,0 for condensing at 30 °C.

For ejector outlet conditions corresponding to 30 °C a reduction in motive stream temperature from 150 °C to 80 °C requires a 120 % higher mass flow of motive stream to achieve the same pressure lift in the ejector. When the outlet condition corresponds to 15 °C, only a 46 % higher mass flow is required for the 80 °C case compared to the 150 °C case.

Interestingly, the ejector efficiency is almost constant at 21 % for the 30 °C outlet condition, i.e., independent of the motive steam temperature. However, for the 15 °C outlet condition, the ejector with motive steam at 80°C is significantly more efficient (37.5%) than the ejector at 150 °C (31.8%).

Table 3 shows the modelling output parameters for various ejector positions (defined in Figure 2), exemplified for the 150 °C/30 °C case. As is well-known for gas ejectors, most of the pressure increase arises across the shock (states *u* and *d*) as the flow transitions from supersonic to subsonic. The table also reveals that there is some condensation occurring along the ejector, as the vapor fraction β is less than 1, even though the ejector outlet consists of pure gas.

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Table 3 Flow states along the ejector for the case with 150°C motive steam and 30°C condenser temperature. The variables are pressure (*P*), enthalpy (*H*), speed (*v*), area (*A*), diameter (*D*), temperature (*T*), entropy (*S*), density (*ρ*), and vapor fraction (*β*). Areas and diameters with a value of zero are un-defined in the model.

	0p	*0s*	*1p*	*2p*	*3p*	*3s*	*4p*	*4s*	*5=m*	*u*	*d*	*6*	*7*
P(kPa)	476.17	0.61	476.11	274.88	14.09	0.57	0.37	0.37	0.55	0.57	3.81	3.81	4.25
H(J/g)	2746	2500.9	2745.9	2649.6	2222.2	2493.4	1844.6	2439.9	2236.1	2255.5	2649.2	2649.2	2667.9
v(m/s)	0	0	6.7	438.9	1023.5	122.9	1342.6	349.3	929.5	908.4	194.1	194.1	20
A(cm^2)	0	0	199.82	4.97	30.04	2846.4	648.81	1536.9	1108.01	1108.01	1108.01	1108.01	9905.02
D(cm)	0	0	15.95	2.52	6.18	0	28.74	0	37.56	37.56	37.56	37.56	112.3
T(C)	150	0	150	130.6	52.7	-0.1	-0.1	-0.1	-0.1	-0.1	79.4	79.4	89.3
S(J/g,K)	6.84	9.16	6.84	6.84	6.88	9.16	6.93	9.16	8.23	8.29	8.79	8.79	8.79
ρ(g/m³)	2548.1	4.85	2547.8	1572.37	111.69	4.57	3.94	2.98	4.89	5	23.41	23.41	25.41
β(-)	1	1	1	0.97	0.84	1	0.74	0.98	0.89	0.9	1	1	1

4.2. Influence of ejector design parameters

The size of the ejector depends on the specified boundary conditions. A sensitivity analysis of the ejector design was performed for the 150 °C /30 °C and 80 °C /30 °C cases. The outlet cross sectional area depends sensitively on the specified outlet speed, while the diffuser length (L6) depends on both the outlet speed and the angle that the diffuser nozzle wall makes with the horizontal. The results presented in section 4.1 were calculated using an outlet speed of 20 m/s, and a diffuser angle of 15°. In Table 4 these are compared with results from calculations made with a diffuser angle of 10° and an outlet speed of 3 m/s. As seen, the lower values of outlet speed and diffuser angel resulted in larger ejector dimensions.

Table 4: Influence of ejector outlet speed and diffuser angle on ejector outlet dimensions (length and diameter)

Boundary conditions		**Outlet speed 20 m/s Diffuser angle 15**		**Outlet speed 3 m/s Diffuser angle 10**	
Condenser temperature [°C]	Motive steam temperature [°C]	Length of outlet section (L6) [m]	Outlet diameter (D7) [m]	Length outlet section (L6) [m]	Outlet diameter (D7) [m]
30	150	1.4	0.11	7.2	0.29
	80	1.8	0.15	9.3	0.38

4.3. Energy use for snow production

Figure 6 shows the required thermal energy per volume produced snow for the heat-driven ejector system, presented at the various motive steam temperatures, and for a condensing temperature of 15 °C and 30 °C ("EJECTOR_15°C" and "EJECTOR_15°C", respectively). Included is also the estimated thermal energy demand for absorption refrigeration systems ("ABS") based on Figure 5. Note that the absorption system is not modelled, and primarily included to provide an indication of possible benefits with ejector-based systems at various operating conditions. Further, the system based on absorption refrigeration is assumed to operate

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at an evaporator temperature of -10°C, while the ejector system operates at the triple point of water (0,01 °C). This advantage of a higher effective refrigeration temperature is because the ejector system is an open system, where the refrigerant is used directly in the snow production.

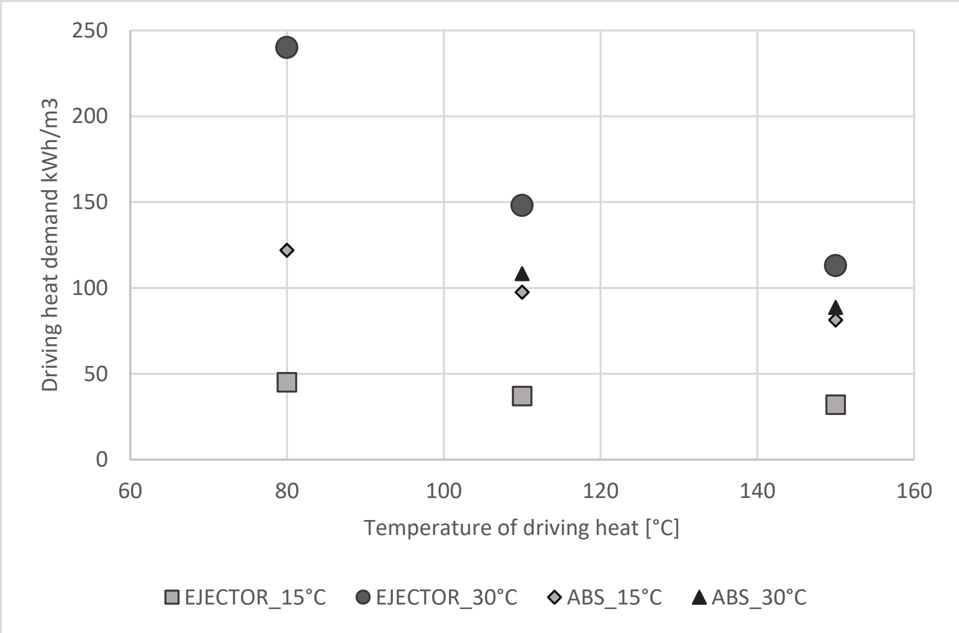

Figure 6: Required driving heat energy at different temperatures of the driving heat and at the two cooling water temperatures (15°C, 30°C). The results for the ejector are based on the modelling performed in this work, while the values for absorption cooling is primarily for comparison, and based on data from various manufactures.

An obvious observation is that the ejector system is much more sensitive to variation in operating conditions than the absorption cooling system, especially for variations in cooling water temperature. These results indicate that the ejector system outperforms absorption systems when cooling water is available at 15°C or lower. The range of thermal energy usage for the ejector system is 32 – 240 kWh/m^3, while for absorption systems the variation is only between 81 and 122 kWh/m^3.

The range of energy usage for the only commercially demonstrated ejector-based snow production system is 45 – 90 kWh/m^3, at a driving steam temperature of 150°C and ambient conditions between 10-20°C. For the more commonly installed compressor-driven TIS production systems, the electric power consumption ranges between 10 - 30 kWh/m^3, mostly depending on the ice production technology applied (flake ice, plate ice, or ice slurry). What also should be noted when comparing the energy usage between various technologies, is that a larger demand of driving energy results in a correspondingly larger cooling demand in the condenser.

5. CONCLUSIONS

In this paper a solution for heat driven ejector-based snow making systems: a vacuum ice slurry system using water (R718) as refrigerant has been evaluated. Detailed numerical simulations of the ejector design and its efficiency were performed. The results were compared against other temperature independent snow production technologies. The energy use for heat-driven snow producing is high compared to electric driven production technologies. Thus, to be a valid option in terms of energy and cost efficiency, the driving heat must be available as surplus heat, and at a low price. Heat-driven snow production based on an ejector chiller coupled with the vacuum freezing method is a promising technology. This work has shown the feasibility of driving the system with relatively low-grade heat at temperatures down to 80°C, which is often readily available as a byproduct from industrial processes, or from a district heating network. The process efficiency

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depends sensitively on the condenser temperature, with energy use decreasing significantly as the condenser temperature decreases. If cold cooling water and excess low-grade waste heat are readily available, the ejector chiller concept becomes highly attractive compared to other proposed technologies for snow production above 0°C ambient temperature. The ejector process thus becomes especially attractive in mildly cold climates such as the Nordic countries, where low-temperature cooling water is often readily available.

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