All-weather snow machine driven by solar energy

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Abstract

Technical snowmaking is the only solution to run ski slopes at the beginning and at the end of the winter season if natural snow is missing. However, technical snowmaking has several shortcomings e. g. high water and power consumption. Furthermore, with current snowmaking technology, snow can only be produced below wet-bulb temperatures of approx. -2.5 °C for compressed air and water guns or below approx. -4 °C for fan guns. This paper presents the concept of a snow machine which can be driven by renewable energy sources, able to produce high quality snow at temperatures above 0 °C, in all-weather conditions, and even in high summer season. The proprietary technology is based on a steam jet ejector chiller (SJEC) and uses the triple point of water. The thermal energy for the ejector can be provided by solar collectors or by a biomass steam boiler. Water is the only working fluid in the entire refrigeration system, which guarantees an ecologically friendly concept for the snow supply. Within the paper, the first operational experiences of the prototype as well as performance figures are presented. Furthermore, a concept for the solar circuit of a solar-assisted snow machine is presented. Subsequently, the system design and system performance of a solar-assisted machine will be evaluated for different scenarios. The evaluation is based on simulation results gathered with a model based on mass and energy balances.

Keywords: Technical snowmaking; steam jet ejector chiller; solar-assisted; all-weather conditions

1. Introduction

At the beginning and at the end of the winter season, natural snow is often missing and ski slopes are closed. The missing snow delays the opening of ski resorts, and represents a serious threat for the multibillion-dollar ski tourism industry, resulting in great losses for the economy of the mountain regions. Technical snowmaking is the only solution to run the ski slopes, but it has several shortcomings e. g. high water and energy consumption. Energy is consumed mainly to generate compressed air and to pump water. Typically, 67 % of all energy consumed in a ski resort is used for snowmaking (Smith 2010). A study, which evaluated the energy consumption of ski resorts in Finland, indicates that the average electricity consumption for slope services including lift operation, lighting and snowmaking amounts to approximately 34.6 MWh/a per hectare of snowed slope. The share of snowmaking is high with an average energy consumption of approximately 24.1 MWh/a (Timonen and Ikkunassa Oy 2010). Beside ski resorts, another sector which is dependent on technical snowmaking is indoor ski domes. The specific energy demand for snow production and air cooling in indoor ski domes is even higher as they are operated all year round even at high ambient temperatures.

1.1. Technical snow generation – State of the art

There are several physical processes to produce technical snow. The available processes split into nozzle atomization of water, cooling technology or cryotechnology (Fuhrmann 1996). Snowmaking via nozzles is the most common way. Snowmaking systems via nozzles can be divided into following subspecies: compressed air and water guns (internal and external mixing guns), and fan guns. Compressed air and water guns can be mounted on the ground or on a tower. These systems require pressurized water and compressed air from an external source which is usually supplied by a central facility via a piping network. In contrast, fan guns use electric driven axial fans to propel the water droplets to a huge distance and only require a small amount of compressed air which is nowadays usually provided by an on-board air compressor. Fan guns typically have one or more rings of nozzles which inject water into the fan air stream. Some separate nozzles are fed with a compressed air and water mixture and provide the nucleation points for the snow crystals. For both systems the production of snow works by the atomization of water into myriad water particles (droplets). A water-compressed air mixture is ejected from a

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nozzle at a pressure of 5 to 10 bar. During the expansion of the water-compressed air mixture the temperature decreases and the water droplets freeze. Aside from the cooling effect by expansion, the water droplets are also chilled by convection and evaporative cooling. The ice nucleation occurs on impurities in the water droplets. The ratio of water and compressed air has a significant influence on the consistency of the snow. By regulating this ratio, either a dry and powdery snow, a wet and heavy snow, or an ice glaze can be obtained (Pierce, JR. 1954). While a relatively low ambient temperature favors the snowmaking process and decreases the necessary compressed air volume flow, a higher snowmaking temperature increases the need for compressed air. A calm wind is advantageous in order to enable a rapid removal of the crystallization heat. The two main factors, which have influence on the effectiveness of snow production are air temperature and humidity, both combined in the wet bulb temperature. Higher air humidity results in a longer cooling time of water droplets to reach the nucleation temperature. With nozzle based systems snow can only be produced with low wet-bulb temperatures. The borderline temperature where compressed air and water guns (snow lances) can start operation is approx. -2.5 °C and the borderline temperature of fan guns is approx. -4 °C (Fuhrmann 1996). These requirements impede the use of traditional technical snowmakers in many critical situations. According to a study from 2011 assessing the energy use of mobile snowmaking at Swedish ski resorts, the energy demand to produce 1 m^3 of snow is 0.58 – 0.72 kWh_{el} for snow lances and 0.97 – 1.94 kWh_{el} for fan guns. The snow production capacity varies for the tested lances between 13 and 22 m^3/h and for the fan guns between 15 and 34 m^3/h. Thus fan guns are more energy demanding but reach a higher snow production capacity (Rogstam and Dahlberg 2011).

1.2. All-weather snow machines

At wet bulb temperatures above approx. 0 °C some kind of active cooling process is required to produce snow, either refrigeration technology or cryotechnology. Cryotechnology uses a cryogenic medium (e.g. liquid nitrogen (LN) or liquid air) which is mixed in the snow cannon with compressed air and water. The cryogen is required to cool the water droplets for snow generation at ambient temperatures above 0 °C. Alternatively, a room can be chilled with cryogen. Then ordinary snow guns can be used to generate the snow in the chilled room. Cryotechnology is complex and expensive, thus this type of snow production is only used for very special purposes (e. g. snow production in summer as part of events) and thus not suitable for large scale snow production.

In the last decades temperature independent snow machines, also called »all-weather snow machines«, based on different cooling technologies and processes have been developed. Compared to traditional snowmaking technology via nozzles, snow production at temperatures above °C is energy demanding and the systems are technologically more complex. These are the main reasons why all-weather snow machines are currently not widespread. Most of the available all-weather snow machines are electrically driven and produce ice in different forms (e.g. blocks, ice cubes, flake ice, plate ice) which is crushed afterwards to small flakes. However, the snow quality is not comparable with natural snow or snow from nozzle systems, because just big chunks of ice are crushed. The snow is not as fine as natural snow and has sharp edges. In contrast, snow machines based on ice slurry generators produce snow by separating ice crystals from an ice slurry. The snow quality of these machines is much closer to natural snow. At present several types of ice slurry generators exist (cf. (Kauffeld et al. 2005)):

- Mechanical-scraper type with rotating knifes, scrape blades, rotating brushes or screws
- Vortex-flow type with oscillatory moving cooled wall method
- Direct-injection or direct heat exchange type
- Fluidized-bed ice generator
- Vacuum freezing method
- Super-cooling water method

The main functionality of the different principles is described in (Kauffeld et al. 2005). Up to date mechanical-scraper method and vacuum freezing method are used for snow generation. Tab. 1 shows a comparison of market available all-weather snow machines.

The snow quality of snow machines based on ice slurry generators differs, too. With the mechanical-scraper method, the slurry has a slush consistence (cf. (Mogilevsky 2013)). Snow generated with the vacuum freezing method is smoother than ice, which is scraped from a surface, as the ice is generated by formation of crystals in water at the triple point conditions. However, for both technologies, main challenges are the ice separation from the slurry and to produce dry and groomable snow.

Tab. 1: On the market available all-weather snow machines (Source of Data: (Eikevik 2017), (Dieseth 2016))

Company	**SnowTek**	**TechnoAlpin / KTI**	**IDE**	**SnowMagic**
Product name	SnowGen	SF220	VIM100	SnowMagic 100
Country of origin	Finland	Italy	Israel	USA
Principle	Ice Slurry (scraper type)	Flake ice	Ice Slurry (vacuum freezing)	Flake ice
Type	Mobile	Stationary	Mobile	Mobile
Capacity	220 m³/d (approx. 9.2 m³/h)	220 m³/d (approx. 9.2 m³/h)	200 m³/d (approx. 8.3 m³/h)	200 m³/d (approx. 8.3 m³/h)
Power consumption	280 kW	227 kW	250 kW	248 kW
Water consumption	1.4 l/s (approx. 5.0 m³/h)	1.5 l/s (approx. 5.4 m³/h)	1.3 l/s (approx. 4.7 m³/h)	1.6 l/s (approx. 5.8 m³/h)
Working fluid	Ammonia	Ammonia	Water	N/A
Size	1 x 40’ container + snow separator	2 x 40’ containers	1 x 40’ and 1 x 20’ containers + snow separator	N/A
Energy per m³	30.5 kWh/m³	24.8 kWh/m³	20.4 kWh/m³	29.8 kWh/m³
References	Winter Olympics in Sochi 2014	Winterberg (GER) Sjusjøen (NO)	-	Ski resorts in Japan and USA

Fig. 1 illustrates a process scheme of an all-weather snow machine based on an ice slurry generator with vacuum freezing method. Such a system consists of an evaporator where the ice slurry is generated, a snow separator, a compressor, and a condenser. In the evaporator, water is taken to the triple point condition namely the point that occurs at 0 °C and 6.1 mbar by reducing the evaporator pressure with a compressor. In these conditions, the liquid, ice and vapor phases of water coexist in a stable equilibrium. The compressor feeds the water vapor against a higher pressure into the condenser where it is liquefied again. Some of the water in the evaporator freezes forming a water and ice mixture called ice slurry. The ice slurry is fed to the snow separator, where water and ice crystals are separated.

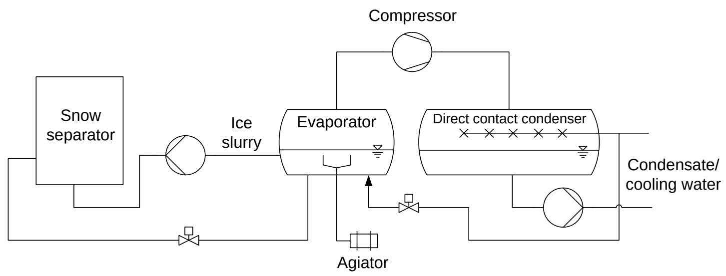

Fig. 1: Exemplary process scheme of an all-weather snow machine based on ice slurry generator with vacuum freezing method

Such a system has been realized by the company IDE Technologies Ltd. from Israel. The vacuum ice maker (VIM) of IDE uses an mechanical driven centrifugal compressor to create a vacuum in the evaporator where the ice slurry is generated. The suction vapor is fed to a direct contact condenser (mixing condenser). The direct contact condenser is cooled by a chilled water spray (coolant). The chilled water is produced by a conventional electric driven water chiller. A large surface area packing is installed in the condenser to increase the heat transfer between vapor and coolant. The liquid consisting of condensate and the sprayed coolant is fed via a coolant pump to the water chiller. The water vapor taken from the evaporator is replaced by chilled water. The ice slurry is pumped to the snow separator, where snow is separated from water. The water from the snow separator flows back to the evaporator. The IDE-system uses electricity as main driving energy to run the centrifugal compressor and the water chiller for condensation of the coolant. (IDE 2017), (Ophir et al. 2011)

1.3. All-weather snow machine based on steam jet ejector technology

An alternative concept for technical snow production based on an ice slurry generator with vacuum freezing method is presented in this paper. The system uses a thermally driven steam jet ejector to feed the water vapor from the evaporator to the condenser. The main components of an ejector are the motive nozzle, the mixing chamber and the diffuser. In the ejector an expansion of a motive steam takes place through a jet nozzle. Thereby the expansion of the motive steam is accompanied by a large increase in velocity up to Mach numbers of 3 to 4. The motive steam entrains the water vapor from the evaporator through momentum exchange. In a following diffuser, the kinetic energy of the mixed vapor streams consisting of motive steam and water vapor from the evaporator is gradually transformed into potential energy respectively in pressure. Both vapor streams are liquefied in the condenser. The condenser pressure represents the back pressure of the ejector and is related to the cooling water temperature. If the temperature of the cooling water decreases the condenser pressure decreases, too. Lower condenser pressure leads to a lower necessary motive steam pressure, which reduce the motive steam consumption. This improves the thermal coefficient of performance (COP) of the refrigeration process. The steam to run the steam jet ejector can be generated by e. g. waste heat, renewable sources or be provided from district heating systems. Electric power is only used for auxiliary drives such as pumps, fans etc. Therefore the power consumption is reduced to a minimum and makes an all-weather snow machine without using an electric driven compressor an ecologically friendly concept of snow supply. The development of the snow machine started in 2012 thanks to a regional grant which financed the development of the first test rig. The snow machine was commissioned in October 2014 in Val di Fiemme at the Nordic Ski stadium. Within an EU-project, which started in 2017, a first prototype has been developed with a snow production capacity of approx. 2 m³/h. Furthermore, it is planned to realize two demonstration plants with a snow production capacity of 4 m³/h, one with biomass and one with solar energy as heat source. The foreseen location to erect the solar driven one is a ski dome in the Netherlands and the biomass driven one in a ski resort in Italy. This paper presents first operational experience as well as performance figures of the prototype, which is electric driven. The prototype has been presented on the Interalpin trade fair in 2017.

2. Design and test operation of the prototype

The prototype of the snow machine consists of a single-stage steam jet ejector chiller (SJEC) with a snow production rate of approx. 2 m³/h. The thermal power for the steam supply is nowadays provided by an electric boiler for testing the technology. The mass flow of the steam is approx. 240 kg/h and the steam is provided at a pressure of 4.5 bar(abs). A plate heat exchanger is used as condenser, which is cooled via a secondary cooling water loop connected to a wet cooling tower. Two pictures of the prototype are shown in Fig. 2. The left picture shows the installation at the Interalpin trade fair in Innsbruck in April 2017 and the right picture shows the installation at the test site in Bolbeno, Italy, in July 2017. The system is integrated into a 20 ft container. The hydraulic scheme of the snow machine with the main process data is depicted in Fig. 3. The steam jet ejector is designed for a back pressure of approx. 40 mbar. According to the design of the cooling water circuit, this design point corresponds to a maximum temperature of 20 °C (e.g. wet bulb temperature t_{WB} of 15 °C with 60 % rel. humidity).

Fig. 2: Pictures of the prototype (left: trade fair installation at Interalpin 2017; right: installation at test side in Bolbeno, Italy)

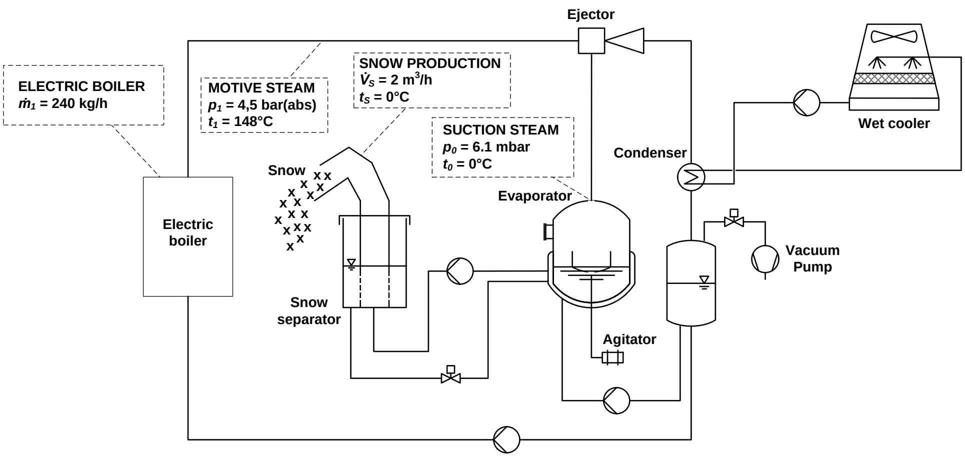

Fig. 3: Process scheme of the prototype

The prototype has been tested on 28. July 2017 in Bolbeno, Italy. At the test site well water with a constant supply temperature of approx. 10 °C was used for the heat rejection. Fig. 4 depicts the system temperatures and Fig. 5 the system pressures during the operation time from 16:10 – 17:40. The diagram in Fig. 6 depicts the heating and cooling capacity, the evaporator temperature and the thermal COP of the SJEC from 16:55 – 17:40. The SJEC was in operation from 16:22 – 16:40, from 16:43 – 17:08, from 17:15 – 17:31 and from 17:32 – 17:39. The evaporator temperature and pressure decreases during the SJEC operation. During the short operation breaks the evaporator was heated up with hot steam from the boiler (at 16:40 from 5 – 10 °C and at 17:07 from 0 – 15 °C).

The cooling water spreading was approx. 7.5 °C and larger than the designed 4 °C due to a lower flow rate of the cooling water pump. However, the SJEC was running close to the design point of the SJEC of 40 mbar back pressure. The condenser pressure was ranging between 32 – 41 mbar (see p_2 in Fig. 5). The cooling capacity plotted in Fig. 6 is calculated related to the evaporator water level and the temperature decrease of the evaporator during the operation of the SJEC. The cooling capacity is underestimated as no heat input from the ambience as well as from the ice slurry pump is considered. The heat consumption has been calculated to determine the thermal COP. The steam consumption was calculated according to (DIN 28430) and (GEA Wiegand 2017). The product of nozzle loss coefficient and outflow function is according to (Loschge 1914) constant for water vapor with a value of 0.459. The calculated value was crosschecked with the electricity consumption, which was measured by a current clamp.

$$\dot{m}_1 = \alpha \cdot \psi_{crit} \cdot A_1 \cdot \sqrt{2 \cdot p_1 \cdot \rho_1} \qquad \text{(eq. 1)}$$

with:

with:	α	–	Nozzle loss coefficient [-]
	ψ_{crit}	–	Outflow function [-]
	A_1	–	Nozzle cross-section [m²]
	p_1	–	Inlet pressure [Pa]
	ρ_1	–	Density of water vapor [kg/m³]

In Fig. 6 the thermal COP ranges between 0.1 and 0.9 and is decreasing with declining evaporator temperature. At a temperature close to 0 °C (time span after 17:30) the thermal COP is between 0.1 and 0.4. The calculation of the cooling capacity does not consider the freezing of the water in the evaporator, so that the thermal COP is underestimated in the temperature range close to 0 °C. Actually there is no possibility to directly measure the cooling capacity while water freezes in the evaporator and thus to calculate a more accurate thermal COP value.

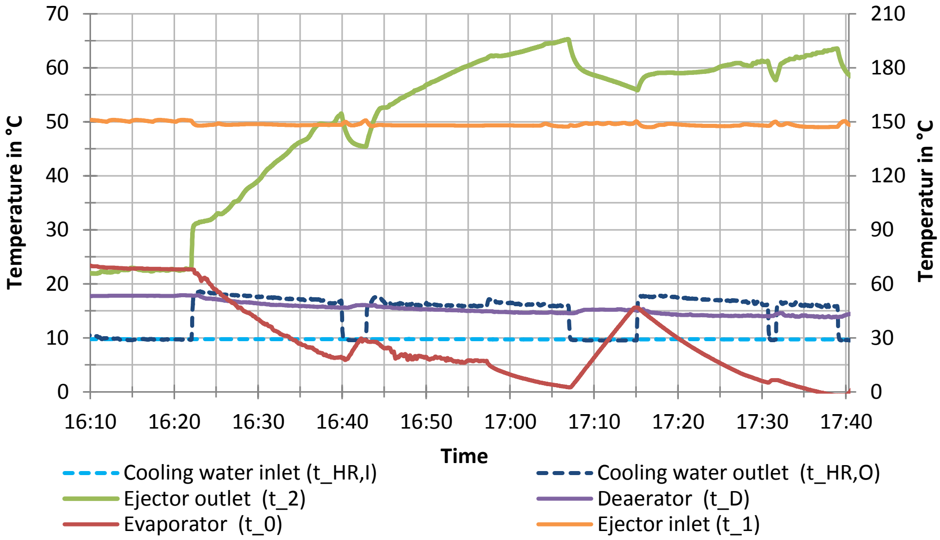

Fig. 4: Exemplary measurement data of the prototype: System temperatures (read ejector motive steam inlet temperature t_1 on right ordinate)

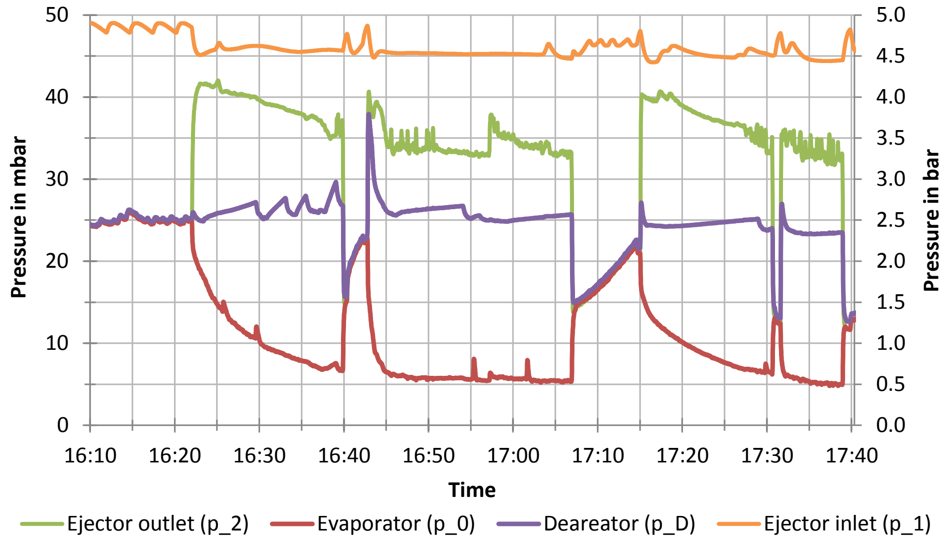

Fig. 5: Exemplary measurement data of the prototype: System pressures (read ejector motive steam inlet pressure p_1 on right ordinate)

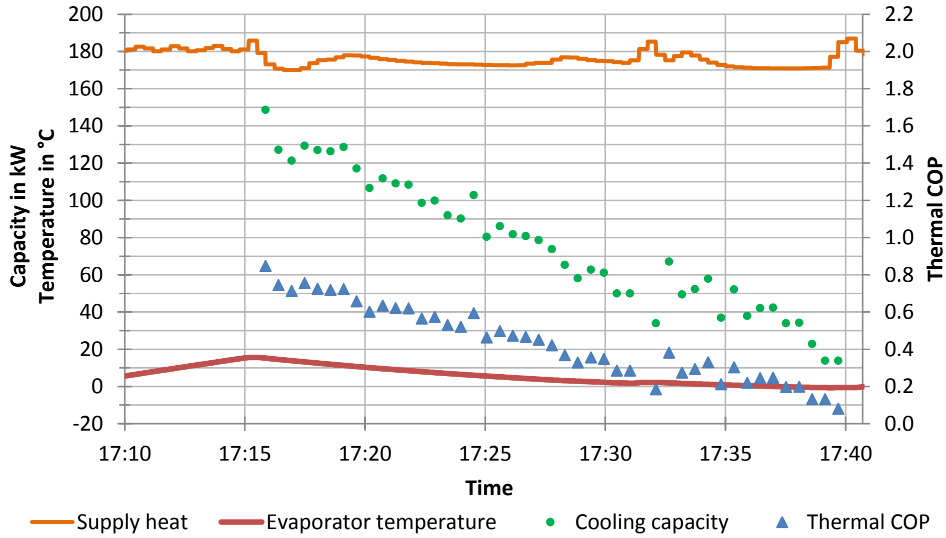

Fig. 6: Exemplary measurement data of the prototype: Heating and cooling capacity, evaporator temperature & thermal COP

3. Solar-assisted snow machine

3.1. Solar steam generation

There are parabolic troughs and Fresnel collectors, innovative evacuated tube and high-vacuum flat plate collectors available for the temperature range up to 250 °C. The collector types can be distinguished between concentrating and non-concentrating systems. The usable irradiation of concentrating collectors is often significantly lower compared to non-concentrating collectors. In case of 50 % lower irradiation the efficiency of the concentrating collector has to be twice as high to reach the same yield. Hence, important selection criteria for the choice of a suitable collector type are the solar profile at the system site and the required temperature.

Generally, there are two fundamental manners in which solar heat can be transferred to the ejector chiller system: direct steam generation (DSG) and indirect steam generation with pressurized water loop and steam drum. With DSG systems the motive steam is directly generated within the collector field and transported to the ejector. However, an intermediate steam drum, which can be bypassed, can be useful as an additional heat capacity to smooth the operation of the SJEC. In addition, it could also serve as droplet separator. The DSG technologies can be realized with parabolic trough collectors or linear Fresnel collectors. In case of a direct solar steam generation in the collector field, thermodynamic losses can be reduced. The control of the resulting two-phase flow has been extensively researched and has been already demonstrated in first commercial pilot plants. Alternatively to the line-focusing collector systems, evacuated tube collectors (ETC) can be used for DSG, too. This possibility has been investigated within the research project ProSolarDSKM (Joemann et al. 2016). However, until now there is no commercial ETC available, which is designed for DSG.

The second technical solution for the solar thermal steam generation is a pressurized water loop. This means, that boiling in the collector is inhibited by increasing the pressure in the solar collector. The water boils in a steam drum subsequent to the collector field at lower pressure. The advantages of this technical solution are an easier controllability of the solar thermal system as well as a more reliable operation condition of the collector field. The pressurized hot water loop can be realized in different manners, either by hydraulic separation (heat exchanger between solar loop and steam drum) or by applying a specially designed solar pump which maintains an overpressure in the solar circuit in combination with a throttle valve before expanding into the steam drum. If there is no hydraulic separation with a heat exchanger and thus no water-glycol mixture is used as heat transfer medium one has to consider antifreeze protection if the ambient temperature can drop below 0°C.

Fig. 7 provides an exemplary hydraulic scheme for a solar loop of the presented all-weather snow machine. However, there is a variety of possibilities. The following requirements have been defined for the system:

- Indirect steam generation with pressurized water loop and steam drum: ✓
- Solar storage: ✓
- Short start-up time of the SJEC (→ small volume/capacity of the steam drum): ✓
- Operation of SJEC only with heat from storage: ✓
- Backup heating system: ✓
- Reheating of solar fluid with backup system (prior to the steam drum): ✓
- Operation of SJEC only with heat from backup system ✓

To meet the various requirements of the solar loop, the system needs two pumps. One pump to feed the collector field and a second pump to enable the operation of the SJEC with the backup system without solar energy. The control throttle valve VRM-S1 is used to maintain the required overpressure in the system to inhibit boiling in the solar loop. In case that the solar loop is not in operation, the valves VAM-S1 and VRM-S1 are closed. The main operation modes are: 1) solar heating of the steam drum, 2) charging of the solar heat storage (no operation of steam drum), 3) heating of the steam drum with heat from the storage (no operation of the solar field) and 4) backup heating mode.

1) In case of solar heating of the steam drum both pumps are in operation. The proportion of the flow rates of the pumps defines, whether the storage is simultaneously charged (P1>P2) or discharge (P1<P2). If both flow rates are equal, the storage is neither charged nor discharged. To disconnect the storage the valve VAM-S3 can also be closed. In this operation mode, the valve VAM-S4 is open and the valve VAM-S2 is closed. If the solar irradiance is not sufficient to reach the desired temperature, the backup system is used to heat the water up.
2) If solar heat is available, but not required from the SJEC, the heat can be used to heat up the solar storage. In that case, only P2 is in operation, VAM-S3 and S4 are open and VRM-S1, VAM-S1 and S2 are closed.
3) P1 is in operation, when heating the steam drum with heat from the storage. VAM-S3 is open and VAM-S2 and S4 are closed. The backup system can be used to increase the temperature level.
4) To heat up the steam drum with the backup system only P1 is in operation. The valve VAM-S2 in the bypass is open and the valves VAM-S3 to the solar storage and VAM-S4 to the solar field are closed.

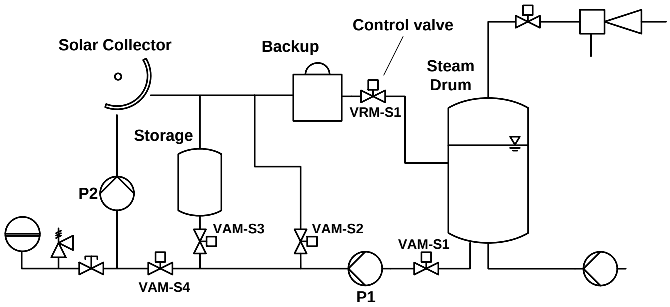

Fig. 7: Hydraulic scheme of the solar loop for a solar driven all-weather snow machine

3.2. Model of the snow machine

For the design of a solar-assisted snow machine, a calculation tool is required to evaluate different system designs and to investigate the operational behavior at varying climate conditions. Such a calculation tool has been developed based on a model of the system using mass and energy balances. The model describes the operational characteristic of the different components, e.g. solar collector, SJEC, pumps, fans, etc. It calculates the pumping power, pressure losses and flow rates of all hydraulic circuits for the determination of auxiliary power demand. Furthermore, the model enables to do year-round calculations on an hourly basis. Microsoft Excel with its spreadsheets in conjunction with VBA (Visual Basic for Applications) is used as software to compile parametric studies for different parameters such as solar field size, storage size or nominal motive steam temperature of the SJEC. Input data for the simulation are meteorological data, which are taken from Meteonorm database 4.0. Consumption and yield data for the components as well as various indicators, such as the solar fraction or the seasonal thermal and electrical efficiency of the system are calculated.

3.3. System design and system performance evaluation

In this chapter different aspects of the system design and system performance of a solar-assisted snow machine with SJEC will be discussed and evaluated. The application scenario is the snow production for an indoor ski dome. In order to assess the climatic influences on the system, three locations are considered (Doha in Qatar; Seville in Spain; Essen in Germany). Below, the reference scenario and design parameters are listed:

- Snow production rate: 4 m³/h (constant → additional heat provided by a backup system)
- Cooling capacity: 215.2 kW (snow density: 500 kg/m³)
- Operation time: 08:00 – 18:00; Season: all year round
- Condenser technology: mixing condenser; heat rejection: wet cooling tower
- Motive steam design temperature: optimized in dependence of system site (climatic conditions)
- Motive steam consumption at varying operating conditions:
    - Motive steam pressure p_1 increases/decreases by the ratio of the back pressure p_2 to nominal back pressure $p_{2,nom}$ (min: 80 % of $p_{1,nom}$; max: 115 % of $p_{1,nom}$)
- Nominal back pressure $p_{2,nom}$ of SJEC: selected in dependence of climatic conditions to be able to operate the SJEC ca. 95 % of the year (5 % no operation possible due to too high t_{WB})
    - Essen: $p_{2,nom}$ = 37.7 mbar (corresponds to t_{WB} of 15 °C + Δt 12 °C + $\Delta p_{Leakage\ Air}$ 2 mbar)
    - Seville: $p_{2,nom}$ = 48.3 mbar (corresponds to t_{WB} of 19.5 °C + Δt 12 °C + $\Delta p_{Leakage\ Air}$ 2 mbar)
    - Doha: $p_{2,nom}$ = 67.6 mbar (corresponds to t_{WB} of 25.8 °C + Δt 12 °C + $\Delta p_{Leakage\ Air}$ 2 mbar)
- Design of the internal and external cooling water loop and cooling tower
    - $\Delta t_{Cooling\ limit\ distance\ of\ wet\ cooler} = 3.0$ °C; $\Delta t_{Heat\ exchanger} = 5.0$ °C; $\Delta t_{Internal\ cooling\ water\ loop} = 4.0$ °C
- Collector field:
    - ETC (Paradigma AQUA PLASMA 15/27)
    - Fresnel (Industrial Solar LF-11)
    - Specific collector area: 7.5 m² per kW cooling capacity of SJEC (total collector area: ca. 1,600 m²)
    - Specific storage volume: 10 l per m² collector area (total storage volume: ca. 16 m³)
- Design data of auxiliaries
    - Nominal pump efficiency $\eta_P = 77.9$ % (efficiency varies with flow rate of the pump)
    - Nominal pump head H_P of the different pumps (pump head varies with flow rate of the pump):
        - Solar pump head $H_P = 6\ m$, storage pump head $H_P = 5\ m$, backup system pump $H_P = 5$ m, ice slurry pump $H_P = 15$ m, internal cooling water pump $H_P = 5$ m, external cooling water pump $H_P = 15$ m
          → to overcome pressure losses in the solar circuit a surcharge to avoid boiling in the collector has been considered for the solar pump, the storage pump and the backup system pump
- Nominal efficiency η_F and total pressure of the wet cooling tower fan p_F
    - Fan efficiency $\eta_F = 60$ %; total pressure p_F approx. 120 mbar

For the design of a solar-assisted snow machine based on SJEC technology, it is important to consider that the efficiency of the SJEC increases up to a certain level with increasing motive steam temperature, but contrary the solar collector efficiency decreases with increasing temperature, which governs the motive steam temperature. The design point is ought to be the optimum operation point taking both effects into account. Furthermore, each solar collector type has a specific decrease of the efficiency with temperature, thus the selection of the design point must be made in view of the selected solar collector type. Finally, it is an optimization problem, since both influence the system efficiency. In Fig. 8 (left part) the average annual thermal COP of the SJEC is plotted over the motive steam temperature for the three locations. For each location the respective dimensions of the SJEC (e. g. nozzle diameter) have been optimized for the motive steam temperature. The required motive steam temperature ranges between 140 and 160 °C to reach the highest possible efficiency. The comparison of the solar fraction for the different scenarios (see right diagram in Fig. 8) indicates that the highest solar fraction is reached at a lower temperature than the temperature level of the highest thermal COP. The reason for this is that the collector efficiency and hence the collector yield decreases with increasing temperature (see Fig. 9). Furthermore, the comparison points out that the ETC collector reaches the highest solar fraction (Essen ETC 100 °C; Seville ETC 120 °C; Doha ETC 140 °C).

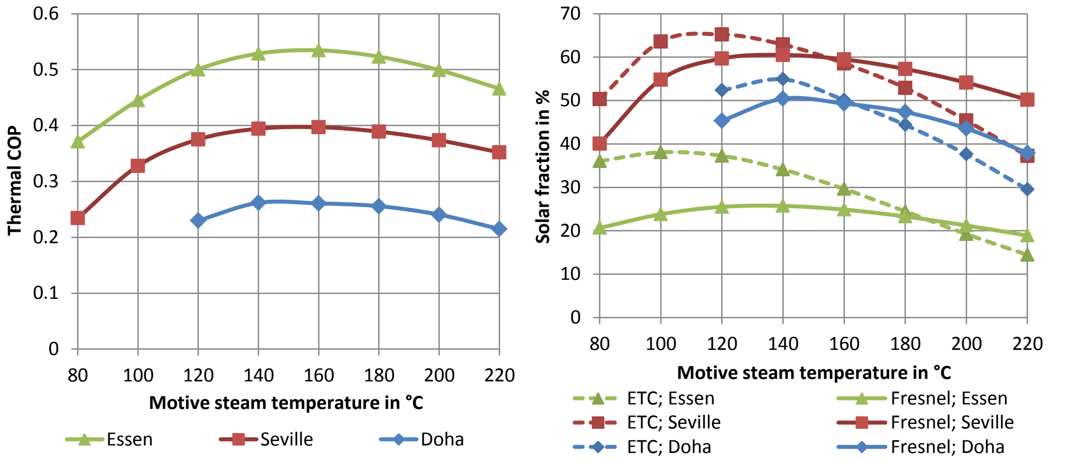

Fig. 8: Average annual thermal COP of the SJEC in dependence of motive steam temperature (left) and solar fraction (right) in dependence of motive steam temperature for ETC and Fresnel collector for the locations Essen, Seville and Doha

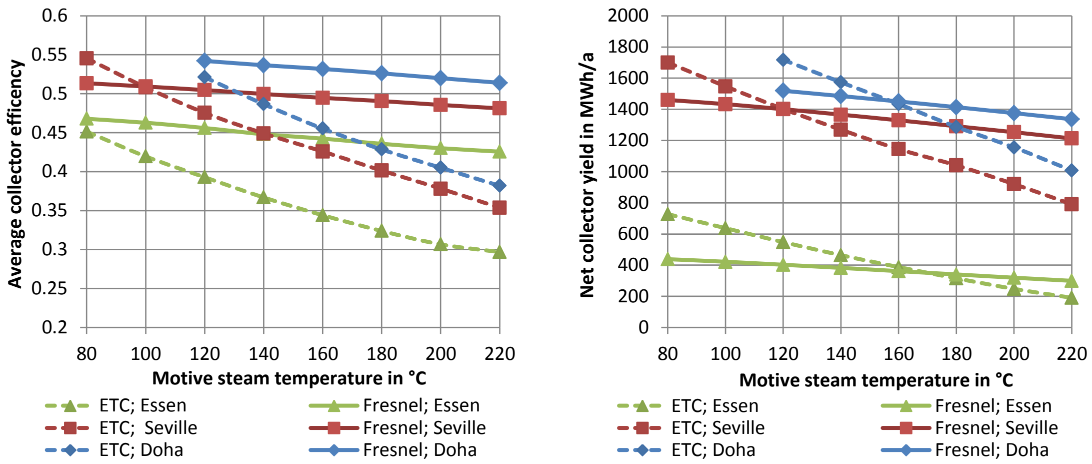

Fig. 9: Average collector efficiency (left) and net collector yield (right) in dependence of motive steam temperature for ETC and Fresnel collector for the locations Essen, Seville and Doha

The results of a parameter variation of the collector field and storage size for the location Seville are depicted in Fig. 10. For the reference scenario with a specific collector area of 7.5 m²/$kW_{Cooling}$ and a specific storage volume of 10 l/m² a solar fraction of approx. 67 % is reached. Small solar fields with a specific collector area of 2.5 – 5 m²/$kW_{Cooling}$ do not require heat storage. Larger solar fields require a storage size of 30 l/m², but not more. A further increase in storage volume would not result in a significant increase of the solar fraction. In comparison to solar driven cooling systems for comfort cooling, the required specific collector areas are significantly higher (cf. (Joemann 2015) page 118). This issue can be explained by different reasons. The cooling load profile of a building is often dominated by the solar irradiation profile, thus the Solar-Load-Match is significantly higher. Another reason is that the thermal COP of the SJEC is much lower due to a lower evaporator temperature of 0 °C. Also the average solar field temperature to run the SJEC is higher than for a single stage absorption chiller which decreases the solar heat yield.

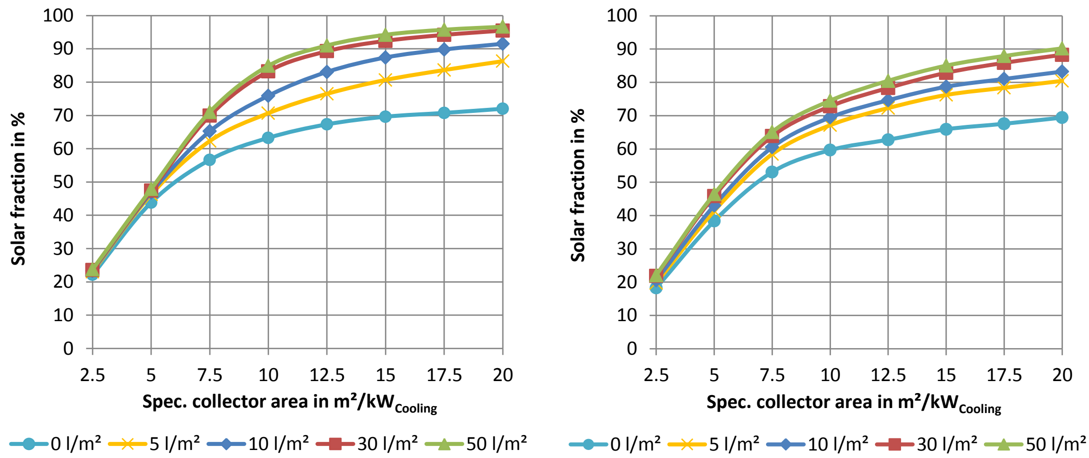

Fig. 10: Solar fraction in dependence of specific collector area and the specific heat storage volume for ETC (left) and Fresnel collector (right) for the location Seville; nominal system temperature: ETC = 120 °C, Fresnel = 140 °C

The annual sums of motive heat demand of the SJEC as well as gross and net heat supply are given by Fig. 11: monthly based (left diagram) and hourly based (right diagram) timeline. The data are given for ETC and Fresnel collector at the location Seville. The results prove that both collectors can provide the required motive heat to operate the SJEC in summer from May to August. The Fresnel collector even provides some surplus heat. From September to April, the backup heating system provides a significant amount of the heat to cover the demand. The data in the right diagram, which are the yield of heat energy and the demand of motive steam over the day, point out that the Fresnel collector provides the heat on a more constant level over daytime, whereby the solar heat provided by the ETC collector has a bell-shaped course so that some surplus heat is generated between 13:00 and 15:00.

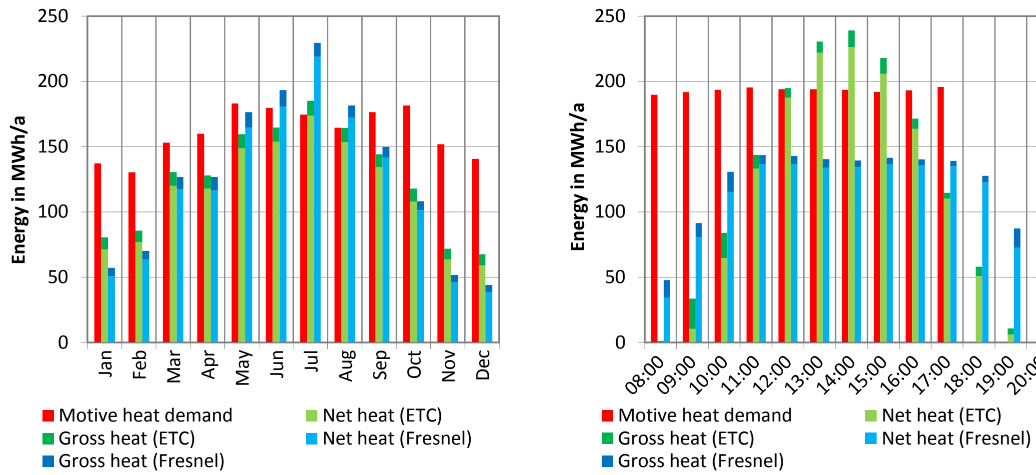

Fig. 11: Annual sums monthly for the year (left) and hourly for the day (right) of motive heat demand of the SJEC as well as gross and net heat supply of the ETC and Fresnel collector for the location Seville; nominal system temperature: ETC = 130 °C, Fresnel = 130 °C (same temperature to assure an identical motive heat demand of SJEC)

The electrical COP given in Fig. 12 is the ratio of the cooling capacity to the auxiliary power demand. The given data are average values for electrical COP monthly for the year (left) and hourly for the day (right) of systems with ETC collector. The nominal system temperature has been optimized to reach the highest possible solar fraction (see right diagram of Fig. 9). The values range between approx. 6 and 14. The annual average values are 10.5 for Essen, 8.8 for Seville and 7.1 for Doha. At the design point of the system, the nominal power consumption is 31.2 kW for Essen, 35.4 kW for Seville and 44.5 kW for Doha. At favorable ambient conditions, the power consumption is significantly lower due to the higher thermal COP of the SJEC. Thus the annual average values for power consumption are 20.6 kW for Essen, 24.3 kW for Seville and 30.2 kW for Doha. However, it has to be mentioned that the power consumption of the control cabinet, of the drives of valves and of the vacuum pump is not considered, thus the electrical COP is slightly overestimated.

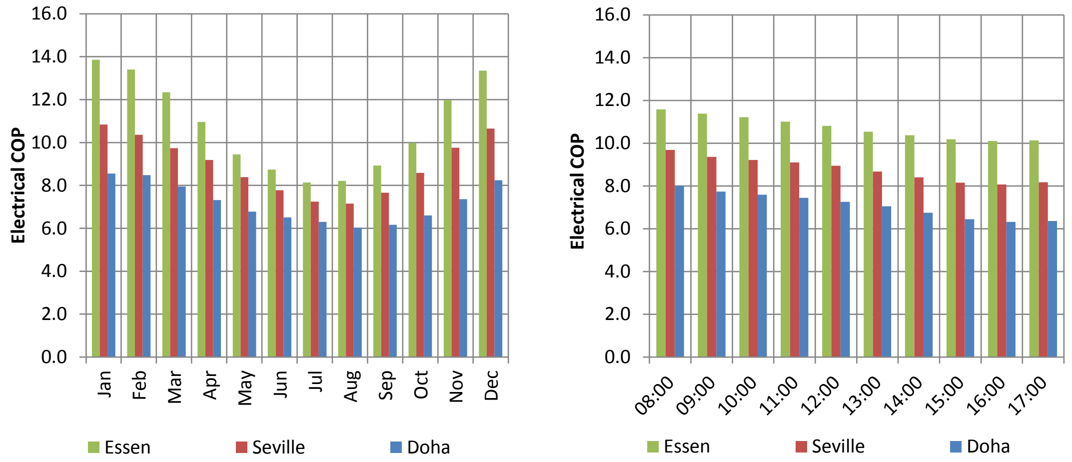

Fig. 12: Average electrical COP monthly for the year (left) and hourly for the day (right) for ETC and for the locations Essen, Seville and Doha; nominal system temperature: Essen = 100 °C, Seville = 120 °C, Doha = 140 °C

The average electrical energy consumption is depicted in Fig. 13 monthly for the year (left part) and hourly for the day (right part) of a system with ETC collector. The electrical energy consumption ranges between approx. 4 and 9 kWh/m³ depending on the climatic conditions. The annual average values are 5.1 kWh/m³ for Essen, 6.1 kWh/m³ for Seville and 7.6 kWh/m³ for Doha. In hot ambient climatic conditions the motive steam consumption of the SJEC increases and thus the power consumption increases, too. The reason is that higher amounts of heat have to be transported, mainly in the internal and external cooling water circuit. That also explains the higher values for electrical energy consumption in the summer time (compare for Essen) and for warmer regions like Doha. There is also a slight increase in power consumption throughout the day as the temperature increases over the day. Summarizing, a contrary behavior of the electrical COP (see Fig. 12) can be seen, which decreases while ambient temperature increases.

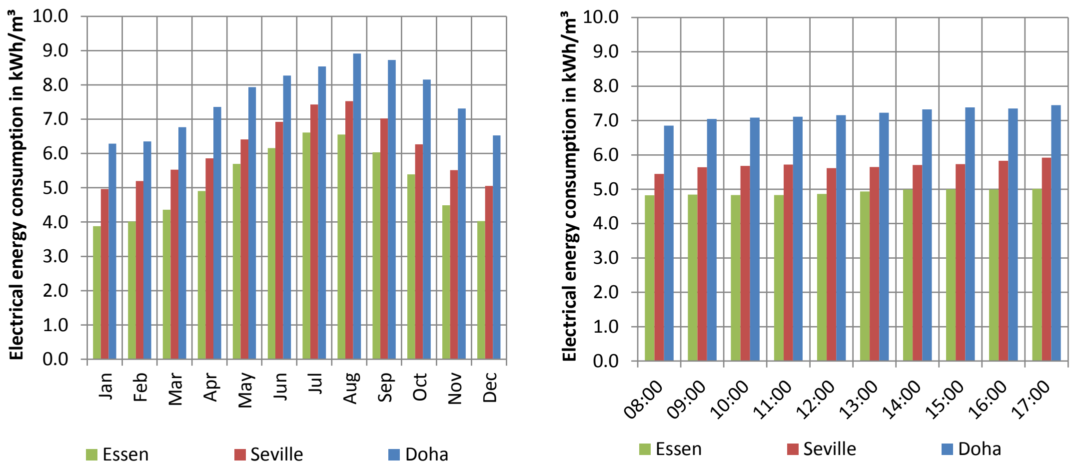

Fig. 13: Average electrical energy consumption monthly for the year (left) and hourly for the day (right) for ETC and for the locations Essen, Seville and Doha; nominal system temperature: Essen = 100 °C, Seville = 120 °C, Doha = 140 °C

The electrical energy consumption of the presented technology for snow production is about 2.7 to 6 times lower compared to other all-weather snow machines (cf. Tab. 1). Furthermore, it has to be highlighted that the presented values are seasonal values. The consumption data for all-weather snow machines from Tab. 1 are only given for the design point and are expected to be worse at a seasonal contemplation. Nevertheless, the snow machine based on SJEC technology needs thermal energy, too. This should come from renewable energy source to assure a sustainable system operation.

4. Conclusion

The developed pilot plant of the all-weather snow machine represents an innovative system for high quality snow generation at temperatures above 0 °C. The system is based on an ice slurry generator with vacuum freezing method and uses a steam jet ejector as thermal compressor for the refrigerant. Water is used as refrigerant in the process. The system can be operated with heat from renewable sources which enables a sustainable system operation. With the aid of the pilot plant it was possible to examine the performance system. The functionality of the system has been proven and the first measurement data of the pilot plant are giving a promising outlook.

Based on the experiences of the pilot plant, a concept for a solar-assisted snow machine has been developed. The development of the concept includes the determination of the design points and the evaluation of the system performance. Both have been realized by year-round calculation and simulating the operation of the system. The simulation proves that the electrical energy consumption of the presented technology for snow production is about 2.7 to 6 times lower compared to other all-weather snow machines, which are currently available.

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