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Industrial high temperature heat pump for simultaneous production of ice-water and process-heat

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ABSTRACT

Industrial processes like dairy production, brewing etc. often require simultaneous cooling and heating. High temperature heat pumps (HTHP) with natural working fluids offer a cost-, energy efficient and sustainable solution. However, availability and performance data and components for HTHP are rare, which is addressed in this work.

Here, the performance of a newly developed industrial scale propane-butane cascade HTHP is presented. It has a new developed semi-hermetic compressor, able to operate at high temperatures. The HTHP is installed in a Norwegian dairy and can lift from typical ice-water temperatures at 0.5°C to process hot-water at 112°C. Temperature glides are 4 K and 15 K at source and sink, respectively. Combined heating and cooling COP is 3.0 to 4.0 with a mean at 3.4 were measured. For this condition a temperature lift between heat sink and source was 88 K to 108 K. The highest temperature lift recorded between evaporation and condensation was 125 K. Condenser and evaporator capacities are within a range of 80 kW to 282 kW and 35 kW bis 134 kW, respectively.

The cascade HTHP enables to utilizes waste heat form e.g. chilled/ice-water or dry-coolers effectively to supply process heat at 112°C. The HTHP is suited for a both retrofit and new installations reducing energy consumption by up to 64% and CO_2-emissions by up to 94%.

Keywords: High Temperature Heat Pump, Cascade, COP, Waste heat, Ice water, Process heat.

1. INTRODUCTION

Today's thermal energy supply is based on fossil fuels. The European process thermal energy demand up to 200 °C is estimated to 720 TWh/a, being 37 % of the total thermal energy demand (de Boer et al., 2020). To limit the global warming the transition to sustainable CO_2-lean process heat supply is key. For industry sectors as food and beverages the temperature level is in the order of 100 °C to 150 °C and account to 123 TWh/a (de Boer et al., 2020). Here, process cooling is often required in addition to preserve raw materials and the products (Elmegaard et al. 2017), (Wolf et al. 2014), (IEAHeatPumpCentre 2014). This work presents an industrial high temperature heat pump combing process cold and heat supply, which can be applied in processes as: sterilisation, pasteurisation, distillation, and brewing. Compared to marked available heat pumps (>TRL8) with working at temperature ranges up to 90 °C, the presented industrial scale HTHP delivers up to 115 °C pressurised hot water, while simultaneously providing process cooling by means of 0.5 °C ice-water. For providing a temperature lift of 115 K, a cascade of R290 and R600 was found as best alternative (Bamigbetan et al. 2016), (Bamigbetan et al. 2018), and evaluations of laboratory scale measurements was presented earlier (Schlemminger et al. 2019). This work addresses the common R&D challenges, such as: i) extending the limits of heat supply temperature to higher values, ii) improving HTHP efficiency, iii) applying environmentally friendly natural refrigerants and iv) increase the TRL level from 3 to 7, lab scale to industrial pilot scale. The focus here is tuned towards the overall performance of the cascade-HTHP.

2. CASE DESCRIPTION

This section covers integration of the cascade-HTHP in the existing dairy process, the measurement equipment, DAQ and data post processing.

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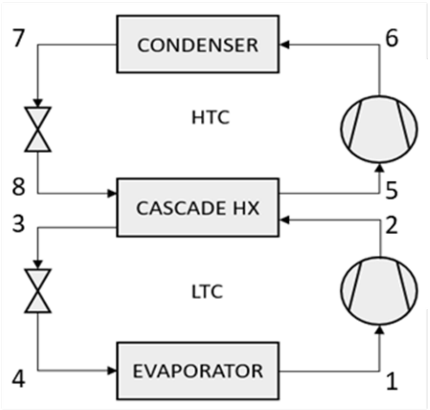
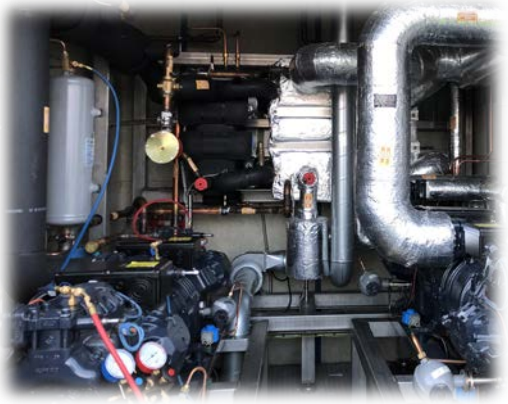
2.1. HTHP integration in existing thermal energy system

2.1.1. HTHP installation

The HTHP is constructed in a cascade configuration, shown simplified in Figure 1. Thus, a high temperature lift >100K is possible. For the low temperature cycle (LTC) R290 and for the high temperature cycle (HTC) R600 is applied as working fluid, both are class A3 working fluid (ISO 817:2014). In the cascade heat exchanger R290 condensates while R600 evaporates.



a)



b)

Figure 1: Industrial HTHP a) simplified cycle scheme, b) picture of installed prototype

The R290 evaporator has a design capacity of 150 kW$_{th}$ at temperature of -5 °C. The R600 condenser capacity is 300 kW$_{th}$, pressurised process water return from 95 °C to 115 °C, condensing temperature of 118 °C. An internal heat exchanger in the LTC and HTC provides sufficient suction superheat of the compressors. The compressors are frequency controlled to maintain condensing pressure and process water outlet temperature for LTC and HTC, respectively. The cascade HTHP is installed in a 10 feet shipping container equipped with ventilation, gas detection and separate ventilated electric cabinet, in accordance to NS-EN-378. Further, crankcase heaters are applied to avoid condensation in the compressor suction side.

2.1.2. Dairy thermal energy system

A simplified thermal energy supply system of the dairy is shown in Figure 2.

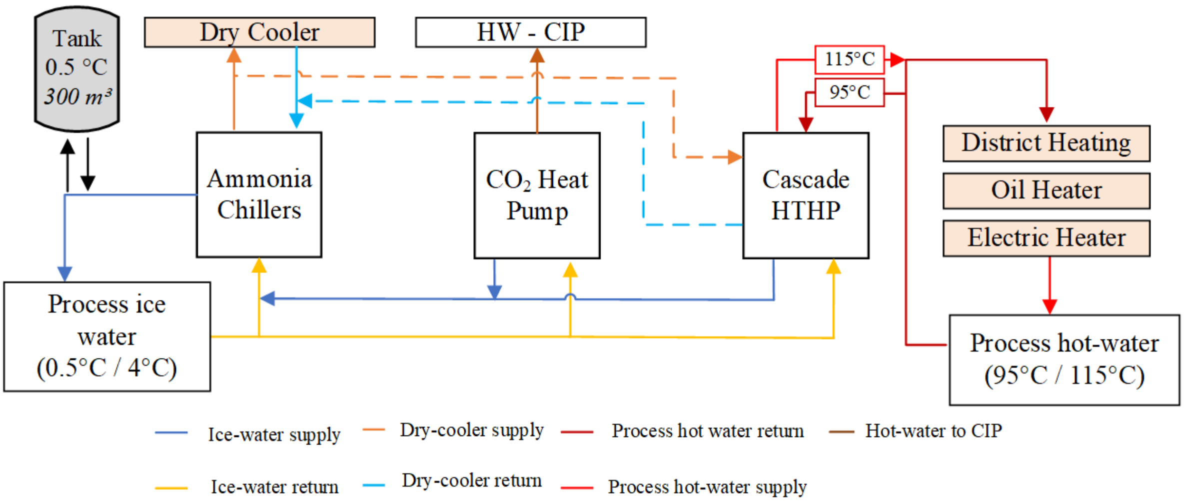


Figure 2: Simplified process diagram of dairy thermal energy system; dotted lines to HTHP are possible contention for utilisation of dry-cooler heat

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Here, the process cooling is realised by suppling 0.5 °C cold ice-water having a return temperature of 4 °C. In the cold supply a 300 m³ ice-water storage is implemented. The cascade HTHP and the transcritical CO_2 heat pump are supplied with the cooling return of the ice-water flow. As main cold supply four NH_3-chillers with a total cooling capacity of 2700 kW_{th} at -1.5 °C evaporation and 33 °C condensation temperature are installed. They reject the condenser heat to a secondary water/glycol circuit connected to dry coolers. The transcritical CO_2 heat pump heat has a capacity of 160 kW heating fresh water from 8 °C to 75 °C, used for cleaning in place (CIP). Here, the ice-water return is used as heat source and cooled from 4 °C to 0.5 °C.

The process heat supply is realised by means of a pressurised hot-water circuit having a supply and return temperature of 115 °C and 95 °C, respectively. Heat is supplied by electric boiler (3000 kW_{th} 95 °C/115 °C). As back-up both an oil-fired boiler and district heating are available.

The cascade HTHP is connected to the ice-water return with an additional secondary loop, to prevent refrigerant leakage into the ice-water supply system (not depicted in Firgure 2). Dimensioning conditions are cooling ice-water from 4 °C to 0.5 °C with secondary water/glycol having -1 °C to 3 °C, respectively. The R290 evaporating temperature is set floating. The R600 condenser is connected via a controlled bypass valve to the process heat return line, heating a part of the 95°C process return to 115 °C. Here, the 115 °C water outlet temperature is used as setpoint. An additional connection between the NH_3 dry-cooler circuit is foreseen to also test the ability for low grade waste heat utilisation at e.g. 10°C to 20°C.

The here presented results cover the cascade HTHP, excluding detailed analysis of dairy processes, building and storage heat and cold demands.

2.2. Data acquisition and post processing

The system analysis is covering a summer week operation. The DAQ of the HTHP is integrated in the dairy and stores data of every measurement signal independently when the deviation of the previous logged value exceeds a certain threshold. Thus, a function implemented in Python library Pandas is used to convert the moving time step dataset to constant time step dataset by means of interpolation/averaging between the readings. An averaging period of 60 minutes is applied in this study. Python in combination with the fluid database Cool-Prop is used to determine thermophysical properties. The energy analysis of the system was conducted applying Microsoft Excel 2016 and the thermo-physical property REFPROP 10.0.

The measurement equipment for temperatures in the range of -30 °C to 150 °C has an uncertainty range of ±1 K and ±0.3 K for absolute and relative measurements, respectively. Pressure measurements are within 1 bar to 40 bar and uncertainty of ±0,2% FS BSL. Heat flow at heat sink and source sider were measured with a commercial heat flow meter, having electromagnetic flow meter and calibrated temperature sensors with a combined uncertainty of ± 1% reading. Electric energy consumption is measured by the frequency drives with an uncertainty of ± 1%.

The combined coefficient of performance, tacking into account heat sink and source side of the heat pump, COP_{SI+SO}, is calculated in accordance to Eq. (1).

$$COP_{SI+SO} = \frac{\dot{Q}_{SI} + \dot{Q}_{SO}}{P_{comp\_LTC} + P_{comp\_HTC}} \qquad \text{Eq. (1)}$$

Here, Q_{SI} and Q_{SO} are the heat source and sink capacity in, determined by the thermal energy meters at the secondary side in W, and P_{comp_LTC} and P_{comp_HTC} are the compressor power in W, respectively. Eq. (1) can be reduced to a heating or cooling COP taking in account only evaporator/heat source or condenser/heat sink capacity.

In order to set the theoretical limit, the Carnot approach is used defining COP_{c_SI+SO} as described in Eq. (2).

$$COP_{c\_SI+SO} = \frac{\overline{T_{SI}} + \overline{T_{SO}}}{\overline{T_{SI}} - \overline{T_{SO}}} \qquad \text{Eq. (2)}$$

The temperatures for heat sink T_{SI} and source T_{SO} are applied in K and comprise of the average between inlet and outlet temperature, respectively. This approach is valid since temperature differences between in and outlet for heat sink are and source are small, less than 25K and less than 5 K, respectively.

The Carnot-efficiency η_{c_SI+SO} as describes in is Eq. (3) is used to compere HTHP performance against the theoretical limit.

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$$\eta_{c\_SI+SO} = \frac{COP_{SI+SO}}{COP_{C\_SI+SO}} \qquad \text{Eq. (3)}$$

3. RESULTS

3.1. Thermal supply analysis

The process cold, heat supply and temperature levels of the dairy are depicted in Figure 3, as analysed for a summer week in August 2021.

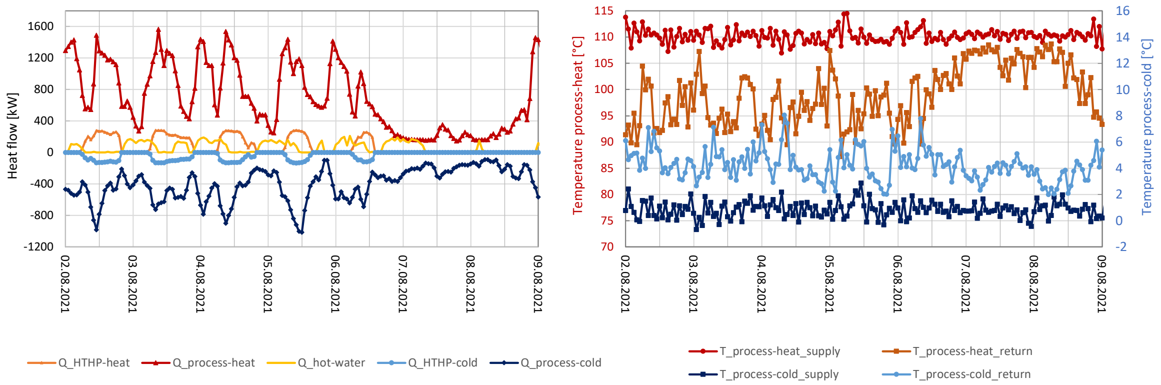

Figure 3: Summer week 2021: a) Process cold and heat supply, b) Temperature levels sink and source

The load/supply profiles show a cyclic process cold and heat demand on daily basis. During the weekend the production was closed. The process heat demand varies between 260 kW and 1565 kW, having a supply temperature in the range of 110 °C ±3 °C and return temperature of 96 °C ±10 °C. Similar behaviour is analysed for the process cold supply, varying daily between 100 kW and 1015 kW, having a supply and return temperature of 0,7 °C ±1 °C and 4.5 °C ±2.5 °C, respectively. It can further be noticed that peak load for heating and cooling result in variations of return temperature for both cooling and heating circuit, whereas supply temperature are rather constant. Process cold and heat demand seems to have a time shift of about 1 to 2 hours. The transcritical CO_2 heat pump operates about 13 hours per day with constant 160 kW gas cooler capacity to heat fresh water from 10 °C ±2 °C to 75 °C ±2 °C, supplying the cleaning in place system and charging the 6m^3 hot water storage. The freshwater heating usually starts after the process heat supply is reduced, since the equipment is cleaned after use.

3.2. HTHP operation

The analysis depicted in Figure 4 indicates the performance of the R290 evaporator and the R600 condenser.

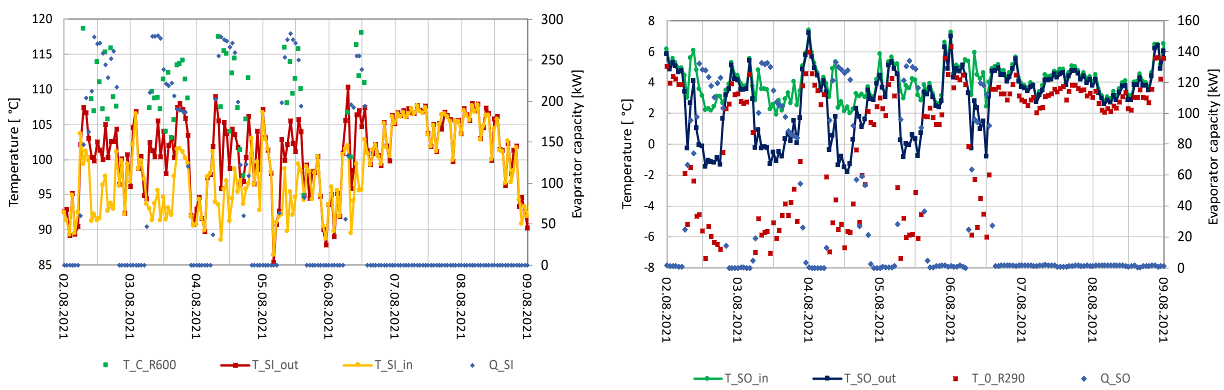

Figure 4: HTHP operation: a) R600 Condenser characteristics, b) R290 evaporator characteristics

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Despite the dimensioning values, both condenser and evaporator are following the heat source and sink temperature variations. Thus, a variation in R600 condensing temperature between 103 °C to 119 °C, with a span of condenser capacity from 80 kW to 282 kW were recorded. The R290 evaporator evaluation results in an evaporation temperature range of -7.5 °C to -2 °C, while having an evaporator capacity of 35 kW to 134 kW. The highest temperature lift between evaporation and condensation measured was 125 K. Values recorded outside the above describes range are linked to start/stop effects of the heat pump and are not considered in this analysis.

A graphical overview of heating and combined coefficient of performance together with the respective Carnot-efficiency over temperature lift is given in Figure 5.

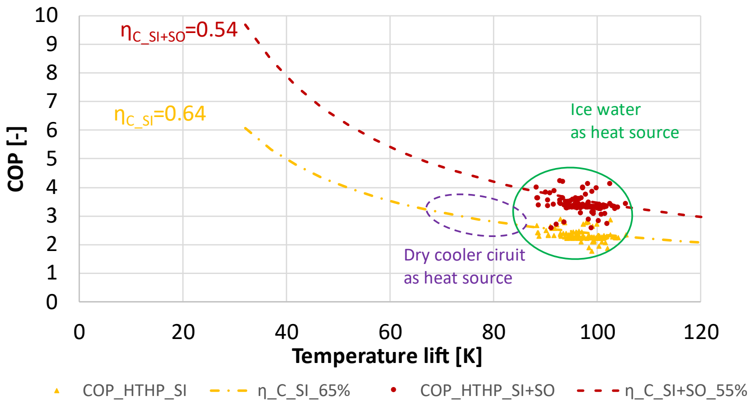

Figure 5: Coefficient of performance and Carnot-efficiency of cascade HTHP

The analysed COP_{SI+SO} is within 2.6 and 4.1 during the investigated period, having an average value of 3.4. The recorded temperature lift, is determined by averaging inlet and outlet temperature of source and sink, respectively. Thus, the temperature lift in the range of 88 K to 108 K. This leads to an average Carnot-efficiency for the combined utilisation of heat source and sink side η_{C_SI+SO} of the HTHP of 0.54. The heating coefficient of performance COP_{SI} spans for the same temperature lift in a range from 1.8 to 2.9 with an average of 2.5, giving a Carnot-efficiency of 0.64. The Carnot-efficiencies stated above are above the average values reported for similar temperature lifts (Arpagaus et al. 2018) and (Bless et al. 2021). As indicated in Figure 5, a extrapolation for the low temperature waste heat recovery from the dry cooler circuit can be conducted based on the heating Carnot-efficiency η_{C_SI} leading to a value in the range of 3 to 4 with a respective temperature lift of 60 K to 80 K.

3.3. Energy and CO_2 saving potential

In order to evaluate the energy and CO_2 saving potential, compared to an existing fossil fuel-based system, the HTHP performance is normalised to the process heat supply, as shown in Table 1. The reference system comprises of gas fired hot-water production at the same temperature level as the HTHP with an efficiency of 0.85, ice-water production with a chiller having a COP of 4.5. CO_2-emissions for gas and electricity are 180 gCO_2/kWh_{gas} and 22 gCO_2/kWh_{el}, respectively.

Table 1. Emission and energy consumption comparison

	Reference	HTHP
Process heat supply [kW]	1.00	1.00
Process cold supply [kW]	0.53	0.53
Electric power required [kW]	0.12	0.47
Gas boiler power [kW]	1.11	-
Total power [kW]	1.23	0.47
Energy saving [kWh]	0.76	
Relative energy saving	62%	
CO_2-reduction	94%	

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The comparison reveals an energy saving potential for the HTHP of 62 %. Whereas the implementation of CO_2-lean electricity in combination with the HTHP gives a CO_2-reduction potential of 94 %. The effect of the electricity based CO_2-emmissions is indicated by applying the European average from 2018 of 281 gCO_2/kWh_{el} (EEA 2022) reduces the CO_2-saving to 44%.

Assuming an integration of the cascade HTHP to cover the entire process hot water production of 117 MWh/week would lead to a 100 % reduction of fossil fuel or direct electric-based heat supply. Resulting in an ice-water production of 62 MWh/week from the HTHP. Thus, the remaining ice-water production form 15 MWh/week would be covered by the CO_2 heat pump with 8.9 MWh/week, and the remaining chillers. This indicated a reduction of primary energy demand for 126 MWh/week to 58 MWh/week. Here, the existing ice-water storage would help to overcome the mismatch between heat source availability and sink demands. Other, alternative heat sources, such as the cold storage refrigeration system can also be exploited.

4. CONCLUSIONS

This work gives an insight in the operation of a newly developed industrial HTHP, having 282 kW heating and 134 kW cooling capacity at 118 °C condensation and -5 °C evaporation respectively. The HTHP is integrated in a dairy. The cascade configuration, with environmentally friendly natural working fluids propane (R290) and n-butane (R600), allows high temperature lifts >100 K. Thus, the HTHP can produce process cold, i.e. ice-water at 0 °C and process heat, i.e. pressurized hot water at 112 °C, simultaneously. The utilisation of evaporator and condenser heat results in a COP_{SI+SO} of 3.4 having a temperature lift of 100 K. The heating COP_{SI} is with 2.5, respectively lower, at the same operation conditions. Here, Carnot-efficiency of 0.54 and 0.64 are achieved for COP_{SI+SO} and COP_{SI}, respectively. A potential evaluation indicated primary energy savings in the order of 62 % and CO_2-reduction potential of up to 94 %, compared to a dairy with natural gas-based process heat supply. Further work comprises of the analysis of the HTHP operation upgrading waste heat from dry-cooler circuit.

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NOMENCLATURE

Symbol		*Index*	
COP	coefficient of performance (-)	*C*	Carnot
P	power (W)	*Comp*	Compressor
p	pressure (bar)	*SI*	Heat sink
$\dot{Q}$	heat flow (W)	*SO*	Heat source
T	temperature (K)	*HTC*	high temperature cycle (R600)
η	Efficiency (-)	*LTC*	low temperature cycle (R290)
		in	inlet
		out	outlet

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