Manuscript ID: 1117
DOI: 10.18462/iir.icr.2019.1117

Performance of high temperature heat pump for simultaneous and efficient production of ice water and process heat

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ABSTRACT

Industrial processes like milk production often require simultaneous cooling and heating at temperature of 1 °C to 10 °C and 90 °C to 120 °C, respectively. High temperature heat pumps (HTHP) with natural working fluids are a cost-, energy efficient and sustainable solution.
In this work a 20 kW_{th} lab scale cascade heat pump is analysed to predict the performance of a 300 kW_{th} pilot plant. The test campaign covers an evaporation temperature range of -1 °C to +1 °C and a condenser temperature range of 113 °C to 118 °C enabling ice-water chilling from 10 °C to 4 °C and a hot-water heating from 85 °C to 116 °C
A Combined heating and cooling $COP_{combined}$ of 2.6 to 2.8 was achieved for a temperature lift of 110 K to 80 K, respectively. This performance gives a primary energy saving potential of 57% and a potential CO_2-emission reduction of 94%.
It can be concluded that HTHP can produce simultaneously chilled/ice-water and hot-process water efficiently for processes like drying, sterilization, pasteurization.
Keywords: Refrigeration, High Temperature Heat Pump, Natural Working Fluids, Efficiency, Combined Heating and Cooling,

1. INTRODUCTION

Demands for process heat between up to 150 °C are estimated to 172 TWh/year on European basis (Nellissen and Wolf, 2015), an equivalent energy consumption of about 8500000 Norwegian households. The presented work focuses on technology development for combined heating and cooling which can be applied in processes as: sterilisation, pasteurisation, distillation and brewing. In such processes product heating, e.g. with pressurized hot-water, is followed by a product cooling (Elmegaard et al., 2017), (Wolf et al., 2014), (IEAHeatPumpCentre, 2014).
A combination of ice- and pressurized hot-water production using a heat pump enables the efficient use of renewable electricity as CO_2-lean primary energy source, which enables: a) increased process efficiency, b) reduction in operation costs and c) reduction in CO_2-emissions.
Standard heat pump technologies for industrial applications (>TRL8) are restricted by simultaneous heat source and heat sink temperatures in the range 30 °C to 70 °C and 70 °C to 100 °C (Elmegaard et al., 2017). Heat pumps working at heat sink temperatures >100 °C are scare and commonly defined as high temperature heat pumps HTHP (Arpagaus et al., 2018). Research challenges related to HTHP development are: i) extending the limits of heat supply temperature to higher values, ii) improving HTHP efficiency, iii) applying new environmentally friendly refrigerants and iv) increase the TRL level from 3 to 7, lab scale to industrial pilot scale, respectively.
The presented work addresses this research challenges by presenting a lab scale HTHP lifting heat form an ice-water production at 4 °C to a pressurized water based heat sink temperature up to 118 °C. Environmental friendly natural working fluids are applied having low values of ODP, GWP and TEWI. The elevated condensation temperatures restrict the available natural working fluids for the compression

heat pump cycle. Suitable working fluids available for condensation temperatures >100°C are water (R718), ammonia (R717), pentane (R601) and n-butane (R600) (Lemmon et al., 2013). Comparisons of available working fluids for the operation temperature and capacity range, also including synthetic, concluded that butane (R600) is most suitable (Bamigbetan et al., 2016), (Bamigbetan et al., 2018). For the targeted temperature lift >100 K a cascade-HTHP system was selected with propane (R290) in the low temperature cycle (LTC) and butane (R600) in the high temperature cycle (HTC), as depict in Fig. 1.

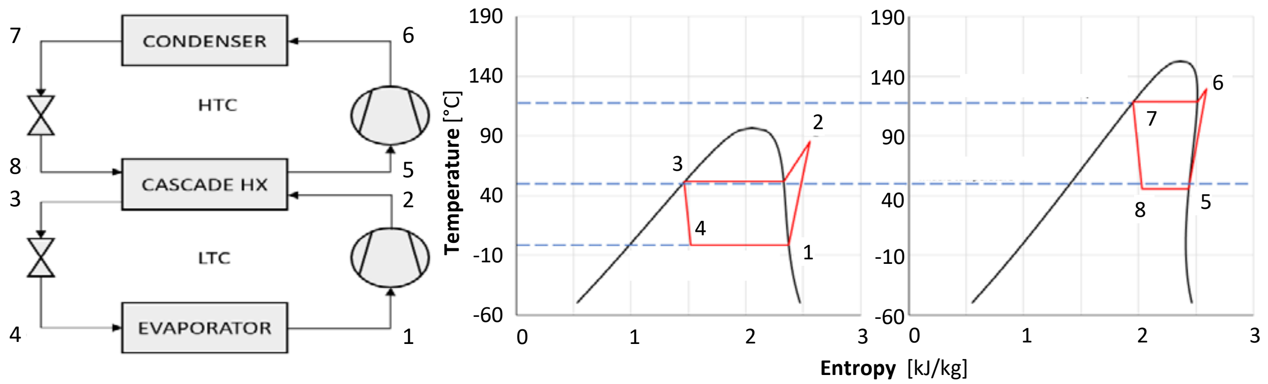

Figure 1: Simplified cycle and T-S diagram of R290/R600 cascade-HTHP

This work presents a detailed system analysis of a 20 kW_{th} R290/R600 cascade-HTHP lab scale system to show the technical feasibility, its energy efficiency an emission reduction potential.

2. MAIN SECTION

This section covered the design, measurement and analysis methods applied.

2.1. Test Setup Description

The R290/R600 cascade-HTHP (c-HTHP) is designed and build based on commercially available components as listed in Table 1.

Table 1 Main components of the c-HTHP

	Component	Manufacturer	Model
LTC - R290	LTC Compressor	DORIN	HEX551CC
	Evaporator	Kaori	K095 x 22
	Suction accumulator with internal HEX	CARLY	LCYE 69S
	High pressure receiver	KLIMAL	RCO139.40.4.80
	Frequency converter	Eaton	DA134046FB-B55C
	Electronic expansion valve	Carel	E2V14
HT – R600	HTC Compressor	DORIN	HEX1501CC
	Evaporator	Kaori	K070 x 60
	Suction accumulator with internal HEX	CARLY	LCYE 69S
	High pressure receiver	KLIMAL	RCO139.40.4.80
	Frequency converter	Eaton	DA134046FB-B55C
	Electronic expansion valve	Carel	E2V14

The working temperature range of the water-based heat sink and heat source was defined as 4 °C to 30 °C and 85 °C to 115 °C, respectively. The evaporator capacity is 10 kW at 10 °C water inlet and 4 °C water outlet. The condenser capacity is 20 kW at 100 °C water inlet and 120 °C water outlet temperature. The pinch point temperature of the heat exchangers is defined as 3 K.
A simplified P&ID schematic and the setup picture is depicted in Fig. 2. The temperature measurement range is -30 °C to 200 °C with an accuracy of ±1 K (absolute) and ±0,3 K (relative). Pressure measurement range is 1 bar to 30 bar with an accuracy of ±0.2 % FS BSL. The working fluids are measured separately by Coriolis mass flow meters with a measurement rang of 0,5 kg/min to 50 kg/min

and an accuracy of ±0.2 % FS. Heat source and heat sink water flow rates are measured using ultrasonic flow meters having an accuracy of ±2 % FS. Compressor speed and electric energy consumption can be manually adjusted using the frequency converters. The electric energy consumption reading has an uncertainty of ±1 %.
The c-HTHP setup was designed and manufactured in accordance to DIN EN 378. All working fluid containing components are placed within a separately ventilated cabinet, see Fig. 2. A gas detector and a ventilation control sensor are installed. The electricity supply into the cabinet is cut-off when the hydrocarbon concentration exceeds 800ppm and/or the ventilation is insufficient. The LTC and HTC are charged with 2.5 kg each.

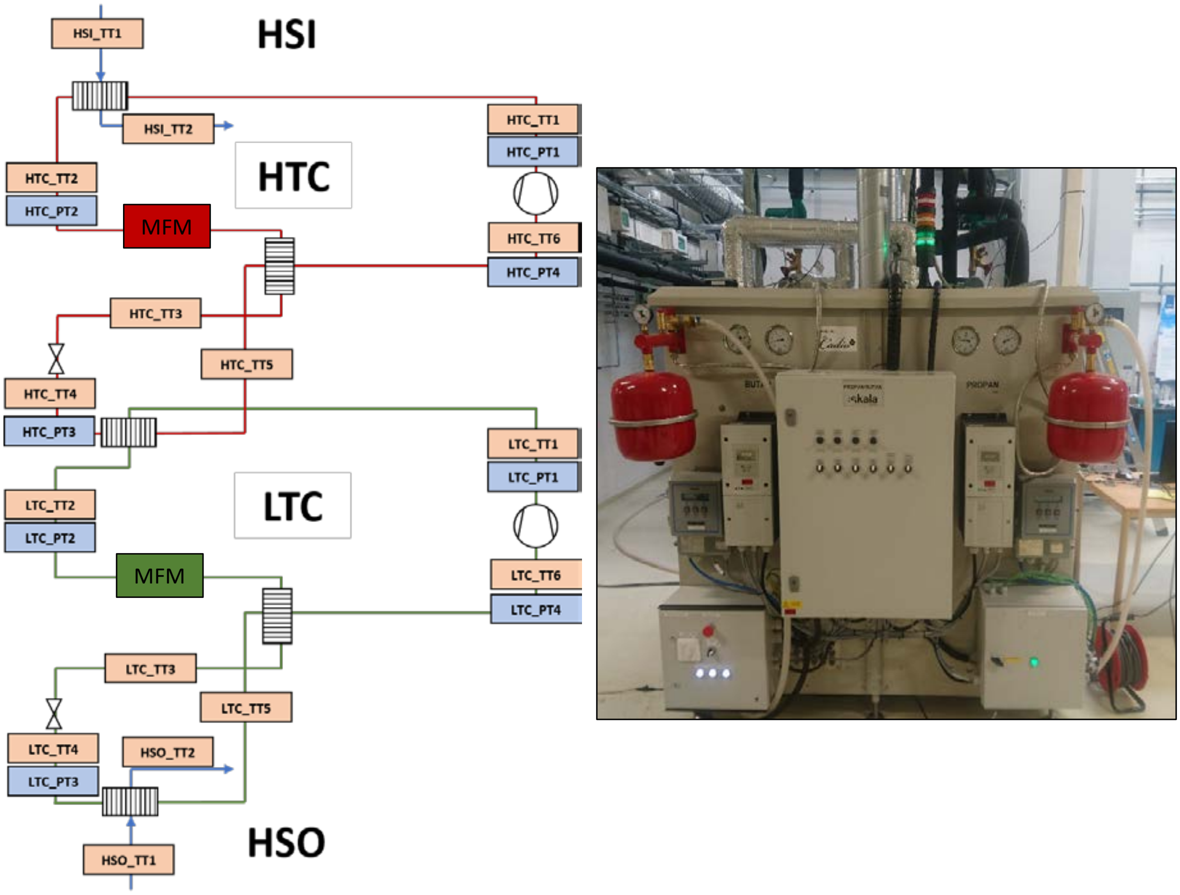

Figure 2: Simplified P&ID diagram (left) and picture of c-HTHP setup (right), (TT = Temperature Transmitter, PT = Pressure Transmitter, HIS = Heat Sink, HSO = Heat source)

2.2. Compressors

A standard R290-compressor was installed at the LTC. Evaporation temperatures exceed the standard working conditions of 10 °C of the compressor. The working temperature range was extended by Dorin for the specific test purpose. The standard condensation temperature working rang is limited to 65°C. The electric motor of the R290-compressor is suction gas cooled. The R600-compressor is a non-commercial prototype. It is modified to an operation range of 80 °C and 140 °C (160 °C) for suction and discharge temperature, respectively. This high temperature levels require a proper design and thermal management. The electric motor is suction gas cooled. It has an external gas manifold, the discharge temperature range is extended to 160 °C and the electric motor is about 25% overmentioned. Further, a suitable lubricant for high temperature applications and hydrocarbons was selected. Both compressors are semi hermetic, certified for use in ATEX 2014/34/EU zone 2. Operation frequency range is 35 Hz to 50Hz.

2.3. Experimental Campaign

The experimental campaign was designed to evaluate the coefficient of performance of the c-HTHP. As heat source water was cooled from 10 °C to 4 °C while pressurized hot-water was heated at the heat

sink side from 85 °C to 115 °C. The compressor frequency was increased in 5 Hz intervals from 35 Hz to 50 Hz. The seven measurements conducted are listed in Table 2.
The measurements were conducted at steady state condition, defined as stable heat source and heat sink in- and outlet temperature variations <±1 K. The measurement duration was minimum 10 minutes with a sampling rate of 1 second. Compressor electrical power consumptions and frequencies were recorded manually, whereas all other parameters were logged and post experimental averaged over the entire measuring period.

Table 2 Operating conditions of the experimental c-HTHP investigation

f [Hz]	Temperature [°C]						COP [-]		η [-]	Capacity [kW]		
	HSO			HSI								
HTC/ LTC	in	out	evap.	in	out	cond.	C_HSI +HSO	HSI+ HSO	C_HSI +HSO	cond.	evap.	el_ comp
35/35	9.8	4.8	0.8	83.9	112.5	113.4	6.2	2.84	0.46	19.0	8.45	9.7
40/35	9.1	3.7	-0.2	84.6	114.6	115.3	6.0	2.71	0.45	20.0	8.37	10.5
40/40	9.7	4.2	0.1	84.4	115.9	116.7	6.0	2.66	0.45	21.2	9.28	11.5
45/40	10.3	4.7	0.6	84.7	114.5	115.8	6.1	2.69	0.45	22.6	9.76	12.0
45/45	10.1	4.1	0.1	84.8	116.7	118.0	5.9	2.64	0.45	24.6	10.31	13.2
50/45	10.3	4.4	-0.1	84.8	116.1	117.6	6.0	2.63	0.45	25.7	10.62	13.8
50/50	9.9	3.9	-0.7	85.0	116.3	118.2	5.9	2.60	0.44	27.3	11.30	14.9

2.4. Data Analysis

Data analysis was carried out using Microsoft Excel 2016 in combination with the property database REFPORP 9.0 (Lemmon et al., 2013). All presented measurement values are minimum 10 minute averaged. The standard deviation was used as quality criteria and was below the measurement uncertainties.
Since heat source and heat sink is utilized in the application of simultaneous ice- and hot-water production a combined ideal COP based Carnot is defined as maximum thermal efficiency as given in Eq. (1).

$$COP_{C\_HSI+HSO} = \frac{T_{HSITT2} + T_{HSOTT2}}{T_{HSITT2} - T_{HSOTT2}} \qquad \text{Eq. (1)}$$

The condenser capacity is determined by the measured working fluid mass flow and the enthalpy difference between condenser inlet and outlet, as described in Eq. 2.

$$\dot{Q}_c = \dot{m}_{R\_HTC} \cdot (h_{HTC1} - h_{HTC2}) \qquad \text{Eq. (2)}$$

The evaporator capacity is determined in analogy, as given in Eq. 3.

$$\dot{Q}_E = \dot{m}_{R\_LTC} \cdot (h_{LTC5} - h_{LTC4}) \qquad \text{Eq. (3)}$$

As the ideal COP, takes the combined coefficient of performance heat source and heat sink into account, see Eq.4.

$$COP_{HSI+HSO} = \frac{\dot{Q}_E + \dot{Q}_c}{P_{comp\_LTC} + P_{comp\_HTC}} \qquad \text{Eq. (4)}$$

To benchmark the combined COP the Carnot-efficiency is defined as given in Eq. 5.

$$\eta_{c\_HSI+HSO} = \frac{COP_{HSI+HSO}}{COP_{C\_HSI+HSO}} \qquad \text{Eq. (5)}$$

In addition to the system, the compressor characteristics were evaluated. The total compressor efficiency is calculated as given in Eq. 6 and the volumetric efficiency as defined in Eq. 7.

$$\eta_{Comp\_total} = \frac{\dot{m}_R \cdot (h_{is\_discharge} - h_{suction})}{\dot{P}_{el\_Comp}} \quad \text{Eq. (6)}$$

$$\eta_{Comp\_vol} = \frac{\dot{m}_R}{\rho_{suction}\dot{V}_{Comp}} \quad \text{Eq. (7)}$$

The compressor displacement is calculated, as given in Eq. 8, with a standard displacement at 50 Hz of 23.13 m^3/h and 48.8 m^3/h for LTC- and HTC-compressor, respectively.

$$\dot{V}_{Comp} = \frac{\dot{V}_{Comp}(50Hz)f_V}{3600 \cdot 50} \quad \text{Eq. (8)}$$

3. REULTS AND DISCUSSION

The results are presented with focus on compressor and system performance and close with the prediction of emission and cost reduction potential.

3.1. Compressor Characteristics

The total compressor and the volumetric efficiencies are plotted over the pressure ratios in Figure 3 and were determined das given in Eq. 6 and Eq. 7, respectively.

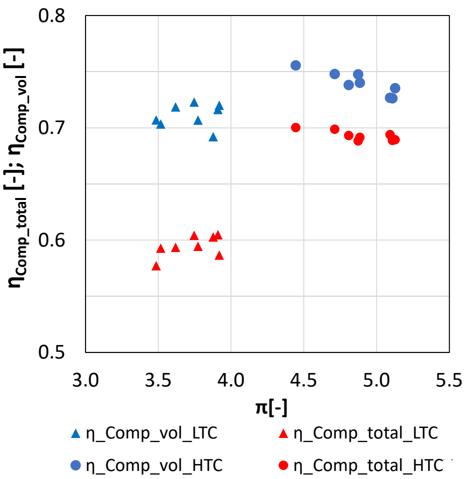

Figure 3: Compressor total compression efficiency and volumetric efficiency of experimental results

The total compressor and the volumetric efficiency are constant over the investigated pressure ration 3.5 to 4.0 and 4.4 to 5.2 for both the LTC- and the HTC-compressor, respectively. The total compressor efficiency is about 0.59 for the LTC-compressor and about 0.69 for the HTC-compressor. Even so the pips of the setup are insulated thermal losses occur between the compressor outlet and the temperature measurement position. The thermal losses are expected higher for the HTC-compressor, since the compressor head and the discharge manifold are external compared to the LTC-compressor. The volumetric efficiencies are in the range of 0.71 and 0.74 for the LTC- and the HTC-compressor respectively.

3.2. System Performance

The c-HTHP system performance is depict in Figure 4 as combined COP over the evaporation temperature. Further, the combined COP is plotted over the temperature lift which is defined as difference between evaporation and condensation temperature.

The measurement results show the expected increase of the combined COP by increasing evaporation temperature. Variations on HSI-inlet temperatures of ±5 K result in variations of ±5 % for the combined COP. For ice-water relevant evaporation temperatures range is 1 °C to +1 °C.
The variation of compressor frequency from 35 Hz to 50 Hz lead to an evaporator capacity increase from 8.37 kW to 11.3 kW. HSO-inlet-temperatures varied in the range of 9.1 °C to 10.3 °C resulting in a HSO-outlet temperature 3.7 °C – 4.8 °C.
The increase of compressor frequency resulted in increased temperature lift, as shown in Figure 4. HSI-inlet temperature was held constant with 84.5 °C ± 0.5 K. The condensation temperature increased from 113.4 °C to 118.2 °C effecting the HIS-outlet temperature 112.5 °C to 116.7 °C. The condenser capacity increased from 19.0 kW to 27.3 kW by increasing the compressor frequency. Consequently, decreased the combined COP. However, results the increased compressor frequency not significantly to a decreasing combined COP, indicated by the constant Carnot-efficiency as defined by Eq. 5 of 0.448 ±0.01, see Table 2.
The simultaneous production of ice- and pressurised hot-water with the developed c-HTHP system resulted in a combined COP range of 2.6 to 2.8.

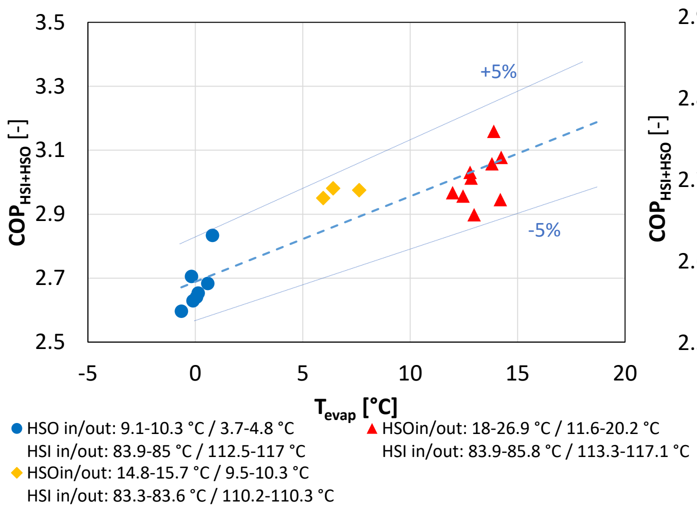

Figure 4: c-HTHP system performance: combined COP as function of evaporation temperature (left), combined COP as function of temperature lift (right)

A detailed investigation of the characteristic sate points of the cascade cycle at 50 Hz compressor frequency for both compressors, is given in Fig. 5.

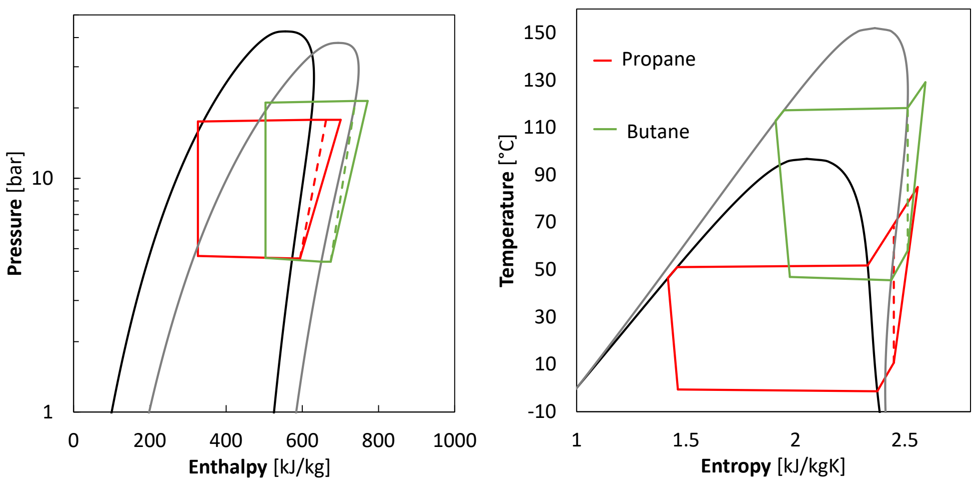

Figure 5: Characteristic sate points at 50 Hz operation for both compressors: log(p)-h diagram (left), T-S diagram (right)

It shows moderate working pressures of both cycles in a range of 4.5 bar to 21.5 bar. The super heat of the R290 cycle is about 15 K at the compressor suction, however the compressor discharge temperature reaches about 85 °C. The high efficiency of the R600-compressor indicates the necessity for sufficient superheat, otherwise compression into the two-phase region may occur. In the experiments a 13 K superheat was sufficient at the compressor suction side, resulting in discharge temperatures of about 130°C.
The T-S diagram reveals only 42 % R600 liquid are present after expansion. To enhance the cycle more R600-subcooling is suggested, further the implementation of ejectors for expansion work recovery would further improve the cycle.

3.3. Emission and Primary Energy Reduction Potential

The application of HTHP are the upgrade and utilisation of waste heat resulting in a replacement of fossil fuelled or direct electric heat supply. The investigated HTHP enables a high temperature lift enabling simultaneous production of ice- and pressurized hot-water. This additional cooling capacity available can either reduce the cooling load of existing chillers or increase the total cooling capacity of a production side.
A reference case is defined to evaluate the emission and cost reduction potential. The reference case assumes: 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 g_{CO2}/kWh_{th} and 22 g_{CO2}/kWh_{el}, respectively.
Standardised result for pressurised hot-water production are given in Table 3.

Table 3 Emissions and operating cost comparisons

	Reference case	HTHP
hot-water production [kW]	1	1
ice-water production [kW]	0.41	0.41
power required [kW]	0.09	0.54
gas boiler fuel [kW]	1.18	-
total energy required[kW]	1.27	0.54
total energy saving [kW]	0.72	
relative energy saving	57 %	
CO_2-reduction	94 %	

This simple comparison indicated significant primary energy saving by 57 %. For the production of 1,0 kWh pressurised hot-water and 0.41 kWh ice-water requires the reference case 151 g_{CO2}, whereas the HTHP only emits 8.5 g_{CO2}, a reduction by 94%.

4. CONCLUSIONS

A technology for efficient and CO_2-emission lean simultaneous production of ice- and pressurized hot-water was presented. This high temperature heat pump utilizes environmentally friendly natural working fluids with low ODP, GWP and TEWI values. Typical application of these type industrial size HTHP would be in processes like: sterilisation, pasteurisation, distillation and brewing etc..
The performance of the 30kW_{th} lab scale HTHP was analysed for an evaporation and condensation temperature operation range of 1 °C to +1 °C and 113.4 °C to 118.2 °C, respectively. A modified compressor was installed at the high temperature cycle of the cascade-HTHP system, enabling to increase standard equipment temperature limits. The total compressor efficiency was in the order of 0.74. The combined coefficient of performance was measured in a range of 2.6 to 2.8. This equals a Carnot-efficiency of 0.45 for a 119 K temperature lift.
The application of the developed cascade-HTHP system for simultaneous production of ice- and pressurised hot-water enables a significant reduction in primary energy consumption of 57 % and CO_2-emissions of 94% compared with a gas fired hot-water and chiller-based ice-water production.
The results show the great potential for industrial HTHP to increase industrial process efficiency and phase in of CO_2 lean electricity simultaneously.

ACKNOWLEDGEMENTS

This paper has been funded by HighEFF - Centre for an Energy Efficient and Competitive Industry for the Future, an 8-year Research Centre under the FME-scheme (Centre for Environment-friendly Energy Research, 257632/E20). The authors gratefully acknowledge the financial support from the Research Council of Norway and user partners of HighEFF.

The authors would also like to acknowledge the support of The Research Council of Norway and the industrial partners Equinor AS, Hydro Aluminium AS, Statkraft Varme AS, Vedde AS, member of TripleNine Group Vedde AS, Mars GmbH, TINE SA, Cadio AS, Hybrid Energy AS and EPCON Evaporation Technology AS through the grant NFR-243679 (HeatUp).

NOMENCLATURE

Symbol		*Index*	
COP	coefficient of performance (-)	*C*	Carnot
h	enthalpy ($kJ \times kg^{-1}$)	*c*	condenser
$\dot{m}$	mass flow rate ($kg \times s^{-1}$)	*Comp*	compressor
P	power (W)	*el*	electric
p	pressure (bar)	*HIS*	heat sink
$\dot{Q}$	heat flow (W)	*HSO*	heat source
T	temperature (K)	*HTC*	high temperature cycle (R600)
$\dot{V}$	volume flow ($m^3 \times s^{-1}$)	*is*	isentropic
η	efficiency (-) or ($W \times W^{-1}$)	*LTC*	low temperature cycle (R290)
Π	Pressure ratio (-) or (bar/bar)	*E*	evaporator
ρ	density ($m^3 \times kg^{-1}$)	*R*	refrigerant
		th	thermal

REFERENCES

Arpagaus, C., Bless, F., Uhlmann, M., Schiffmann, J., Bertsch, S.S., 2018. High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials. Energy 152, 985-1010.

Bamigbetan, O., Eikevik, T.M., Nekså, P., Bantle, M., 2016. Evaluation of natural working fluids for the development of high temperature heat pumps, 12th IIR Gustav Lorentzen Conference on Natural Refrigerants (GL2016), Edinburgh, United Kingdom.

Bamigbetan, O., Eikevik, T.M., Nekså, P., Bantle, M., Schlemminger, C., 2018. Theoretical analysis of suitable fluids for high temperature heat pumps up to 125° C heat delivery. Int. J. Refrigeration 92, 185-195.

Elmegaard, B., Zühlsdorf, B., Reinholdt, L., Bantle, M., 2017. Book of presentations of the International Workshop on High Temperature Heat Pumps, International Workshop on High Temperature Heat Pumps. Technical University of Denmark (DTU).

IEAHeatPumpCentre, 2014. Annex 35: Application of Industrial Heat Pumps - Final Report (No. Report HPP-AN35-1&2), Borås, Sweden.

Lemmon, E.W., Huber, M.L., McLinden, M.O., 2013. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.8 ed. National Institute of Science and Technology, Gaithersburg.

Nellissen, P., Wolf, S., 2015. Heat pumps in non-domestic applications in Europe: Potential for an energy revolution. Emerson Climate Technologies, Delta-ee 3rd Annual Heat Pumps & Utilities Roundtable.

Wolf, S., Fahl, U., Blesl, M., Voß, A., Jakobs, R., 2014. Analyse des Potenzials von Industriewärmepumpen in Deutschland. Institut für Energiewirtschaft und Rationelle Energieanwendung. Stuttgart. URL: http://www.ier.uni-stuttgart.de/publikationen/veroeffentlichungen/forschungsberichte/downloads/141216_Abschlussbericht_FKZ_0327514A.pdf (downloaded 01. 10. 2015).