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Theoretical Analysis of Transcritical HTHP Cycles with Low GWP HFO Refrigerants and Hydrocarbons for Process Heat up to 200 °C

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ABSTRACT

Transcritical heat pump cycles are suitable for processes that require large temperature glides on the heat sink, such as air or hot water heating. Refrigeration systems or hot water heat pumps often apply transcritical CO_2 cycles. For industrial waste heat recovery with high temperature heat sources, transcritical CO_2 heat pumps are not suitable. This study investigates the feasibility of low GWP HFO refrigerants and hydrocarbons for the delivery of process heat up to 200 °C using transcritical HTHP cycles. Steady-state models with IHX and parallel compression have been developed to analyse the cycle performance for heating air from 30 to 200 °C and water from 100 to 200 °C. The design aspects of the main cycle components are discussed, especially compressor and refrigerant-oil selection. The analysis shows that the gas cooler pressure is the most important optimization parameter to achieve highest efficiency up to 3.5. The transcritical cycle with IHX is easy to control and is technically feasible with one or two compression stages. R1233zd(E), R1224yd(Z) and R600 are considered the best refrigerant candidates.

Keywords: Transcritical Cycles, High Temperature Heat Pump, HFO Refrigerants, Hydrocarbons, Efficiency.

1. INTRODUCTION

1.1. Motivation

In industry, there is a large process heat demand in the temperature range of 100 to 200 °C. Today, this heat is primarely provided by burning fossil fuels (i.e. gas, coal, oil) or by direct electricity heating. With an increasing focus on reducing greenhouse gas emissions and higher energy efficiency, interest in the application of industrial high temperature heat pumps (HTHP) is growing (Arpagaus et al., 2018; Zuberi et al., 2018).

In the EU, Kosmadakis (2019) estimated a total waste heat recovery potential of 28.37 TWh/year by using industrial heat pumps. However, compared to the residential sector, heat pumps have been used to a lesser extent in industry due to limited availability of systems for high temperatures (i.e. lack of compressors, lubricants, and refrigerants).

Today, there are only a few heat pump suppliers on the market, which can provide industrial heat pumps with heat supply temperatures > 150 °C. Viking's heat booster reaches condensation temperatures of up to 165 °C using the hydrofluorolefin (HFO) R1336mzz(Z) (T_{crit} of 171.3 °C). The Steam Glow Heat Pump SGH 165/125 from Kobelco in Japan produces steam at 120 °C with R245fa and further compresses the steam to 165 °C. Another challenge is the lack of broad knowledge for an optimal process integration, e.g. by pinch analysis (Schlosser et al., 2019). Operation at the 150 to 200 °C temperature range of process heat would expand the possible applications of HTHP and address a larger range of industries.

Most of the available heat pumps are based on subcritical cycles where the maximal supply temperature is limited by the critical temperature of the refrigerant for heat rejection by condensation. However, the operating range can be extended beyond the critical temperature to the supercritical state. Transcritical cycles are a potential technical solution and are particularly suitable for supplying heat at a high temperature level (Angelino and Invernizzi, 1994). The condenser is replaced by a gas cooler where a sensible refrigerant cooling takes place at constant pressure with a large temperature glide, i.e. temperature difference between gas cooler inlet and outlet. In the evaporator, the fluid remains subcritical, absorbs heat from the heat source and undergoes a phase change.

Research on transcritical heat pump cycles began in the early 1990s to replace environmentally harmful chemicals like CFC and HCFC refrigerants by CO_2 (Lorentzen, 1994; Nekså, 1992). The critical conditions of

CO_2 (31.1 °C at 73.7 bar) makes a transcritical CO_2 cycle especially suitable for water heating applications, e.g. heating tap water from 5 to 60 °C (Nekså, 2002). Since then, the CO_2 heat pump technology has undergone remarkable development. Today, transcritical CO_2 cycles are well-established in supermarkets refrigeration (e.g. CO_2 boosters), car air conditioning, water heating (e.g. Eco Cute up to 90 °C or Thermeco$_2$ up to 110 °C), or air heating (e.g. Eco Sirocco up to 120 °C). However, for applications in industrial waste heat recovery with heat source temperatures higher than the critical temperature of CO_2 other fluids with higher possible evaporation and critical temperature are required.

1.2. Studies on transcritical heat pump cycles using other refrigerants than CO_2 (R744)

A thorough literature review has shown that studies on using refrigerants other than CO_2 in transcritical heat pump cycles for industrial applications are very limited. There are only a few examples available using R32, R1234ze(E), and R600. Table 1 gives a summary of those studies categorised by organization, heating capacity, refrigerant, heat source and sink temperature, cycle type, compressor type, and COP.

Table 1. Studies on transcritical heat pumps for industrial applications not using CO_2 as refrigerant.

Organization	Capacity [kW]	Refrigerant	Heat source (in/out) [°C]	Heat sink (in/out) [°C]	Transcritical cycle type	COP	Reference
MINES ParisTech, France	30	R32 (HFC)	50 – 60 (humid air)	60/120 (air)	Basic cycle	3.7 – 4.1	(Besbes et al., 2015, 2014)
MINES ParisTech, EDF, France	11	R1234ze(E) (HFO)	82 (humid air)	100/150 (air)	IHX cycle, parallel piston	3.3 – 3.7	(Chahla et al., 2019)
EDF, MINES ParisTech, France	30	R1234ze(E) (HFO)	82 (moist air)	90/150 (air) 100/150 (air)	IHX cycle, parallel piston	3.7 3.4	(De Carlan, 2019)
Mayekawa, Waseda Uni, Japan	300	R600 (butane)	80 (hot water)	80/180 (thermal oil)	IHX cycle, 1-, 2-, 3-stage turbo	> 3.5	(Kimura et al., 2018; Saito, 2017)
TU Graz, Austria	45	R600 (butane)	80/75 (hot water)	100/170 (water)	IHX cycle, piston	3.2 – 3.4	(Verdnik et al., 2019)

At MINES ParisTech, Besbes et al. (2015, 2014) developed a transcritical HTHP using R32. The technical feasibility was demonstrated by heating up air from 60 to 120 °C (dry air) using 50 °C moist air as heat source. The impact of the inlet air to the gas cooler and air humidity on the evaporator were investigated. A COP of close to 3.7 was observed with an exergy efficiency of 63%. Simulations predicted that the refrigerants R32, R134a and HFO R1234yf had a great potential to maximize the energy efficiency of a transcritical heat pump supplying 120 to 130 °C. Compared to a conventional subcritical cycle with R245fa, a transcritical R32 cycle reduced the energy consumption by a factor of 2. Based on that, a pilot R32 system with a heating capacity of 30 kW was built and tested. A COP of 3.7 was reached under the conditions of an industrial dryer (59 °C inlet air, 117 °C outlet air, 50 °C moist air as heat source) (De Carlan, 2019). Potential applications with similar drying conditions were identified in the starch industry, wood pellets, tiles and bricks, non-woven technical textiles, paper pulp, and pet food.

Later on, Chahla et al. (2019) investigated a 2nd generation of transcritical heat pump with HFO R1234ze(E) (T_{crit} of 109.4°C) to increase the hot air supply temperature to 150 °C for application as a preheater in a tunnel drying process using the effluents at 82 °C as heat source. Market available components (e.g. reciproacating compressor, electronic expansion valve), originally designed for R744 were used to build the prototype. Another adaptation was the application of synthetic polyolester (POE) oil. Technical options for improving the COP were discussed, such as adding an IHX in the cycle and using parallel compression economizing (Sarkar and Agrawal, 2010). The use of an IHX enabled 6% COP improvement (from 3.32 to 3.51), parallel compression 9% (COP of 3.61), and the combination of both options 12% of the COP (3.71) (Chahla et al., 2019). Indeed, both options allowed reducing the evaporator inlet quality, which reduced the expansion valve losses and increased the cooling capacity of the system. According to De Carlan (2019) the production of a 780 kW demonstrator in the paper industry is in progress where a COP close to 4 is envisioned when heating air from 90 to 150 °C using humid exhaust air at 80 °C as heat source.

In Japan, Kimura et al. (2018) developed a transcritical R600 (butane) HTHP prototype for heating thermal oil from 100 to 180 °C at a heat source of 80 °C. The gas cooler has a capacity of 300 kW. The system allows testing two compressor sets of oil-free radial compressors with magnetic bearings (1-stage, 2-stage, and 3-stage) (Saito, 2017). The target COP is more than 3.5 when heating from 80 to 180 °C (80 °C heat source).

Recently, Verdnik et al. (2019) from the TU Graz in Austria developed a 45 kW transcritical R600 heat pump using a suction gas cooled reciprocating compressor and an IHX for superheating. The simulation model was validated by experimental data of a R600 test rig operated at condensing temperatures up to 110 °C. The IHX increased the superheat and thus reduced the high side pressure at which an optimum COP was reached.

In summary, it is clear that transcritical heat pump cycles offer opportunities to improve the performance of air and water heating applications with high temperature glide compared to conventional subcritical cycles.

1.3. Goal of the study

The goal of this study is to theoretically evaluate the performance of different natural and F-gas-safe HFO and HCFO refrigerants with low global warming potential (GWP) for the generation of process heat of 200 °C using transcritical heat pump cycles. Refrigerants (pure fluids) with critical temperatures > 150 °C are investigated in a basic transcritical cycle, an IHX cycle, and a cycle with parallel compressors where a part of the refrigerant flow is decoupled by a flash tank after the first expansion stage.

In a parameter study, the performances are examined for two application examples: (1) heating of water from 100 to 200 °C (heat source: 80 °C hot water), and (2) heating of air (e.g. drying process) or water from 30 to 200 °C (heat source: 30 °C moist air). The heat sources represent industrial waste heat available at large amounts (e.g. humid exhaust air or waste water), so that the heat absorption is nearly at constant temperature (temperature glide of 5 K). The heat sinks are considered to be dry air or pressurized hot water.

2. SELECTED TRANSCRITICAL CYCLES AND REFRIGERANTS

Natural hydrocarbons and synthetic HFO/HCFO refrigerants with a low GWP were selected in this study. Table 2 shows the main properties including the critical temperature and pressure, GWP, ozone depletion potential (ODP), safety group (SG), and normal boiling point (NBP) (Arpagaus et al., 2018). R245fa, which is currently used in HTHP is examined for comparison, as it will be banned in the coming years according to the F-Gas regulation due to too high GWP.

Figure 1 shows the transcritical heat pump cycles investigated in this study with the corresponding T-s diagrams and heat exchanger approach temperatures.

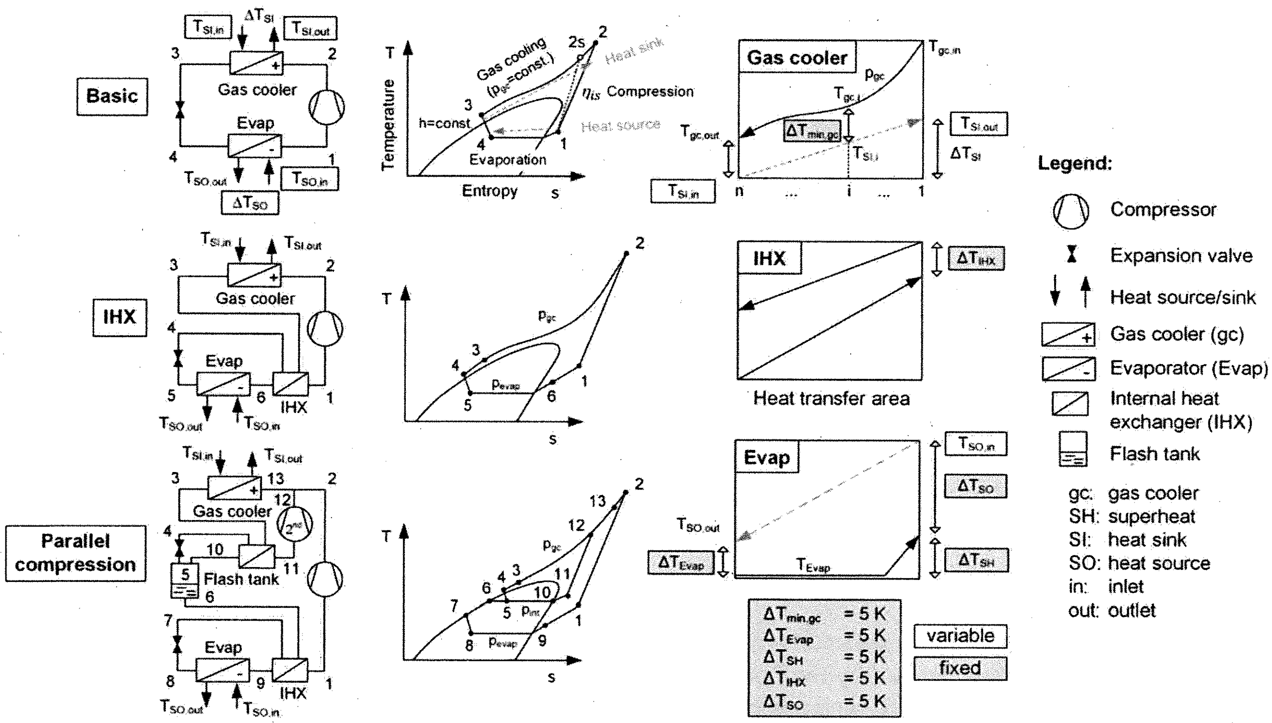

Figure 1: Schematics of the investigated transcritical heat pump cycles with T-s diagrams and fixed approach temperatures in the gas cooler, IHX, and evaporator.

A basic transcritical cycle was considered as a reference for keeping the system complexity low. As second cycle, an IHX was integrated with the aim to improve cycle efficiency. The IHX subcools the refrigerant from the gas cooler before it enters the expansion process, while at the same time, increasing the superheat at the inlet of the compressor. As a third option, the use of parallel compression is investigated (Chahla et al., 2019; Sarkar and Agrawal, 2010; Yu et al., 2019) where the refrigerant leaving the gas cooler expands first into a vapor-liquid separator (e.g. flash tank) from where the vapor portion enters a 2nd compressor and meets with the gas leaving the main compressor. The liquid portion is further expanded to the evaporator pressure. Such a cycle allows vapour compression on a smaller pressure ratio reducing the main compressor input power. The parallel compression has the additional advantage of reducing the refrigerant mass flow to be compressed by the main compressor, while compressing the rest of the mass flow at a lower pressure ratio.

Table 2. Properties of the selected refrigerants (Arpagaus et al., 2018; EES F-Chart software).

Refrigerant	**Type**	T_{crit} [°C]	P_{crit} [bar]	**GWP** (CO_2=1)	**ODP** (R11=1)	**SG**	**NBP** [°C]
R601	Natural	196.6	33.7	5	0	A3	36.1
R600	Natural	152.0	38.0	4	0	A3	-0.5
R514A	HFO	178.4	34.0	2	0	B1	29.3
R1233zd(E)	HCFO	165.6	35.7	1	0.00034	A1	18.0
R1224yd(Z)	HCFO	155.5	33.4	< 1	0.00012	A1	14.0
R1234ze(Z)	HFO	150.1	35.3	< 1	0	A2L	9.8
R245fa	HFC	154.0	36.5	858	0	B1	14.9

3. SIMULATION MODEL

The transcritical cycles were modelled in the Engineering Equation Solver (EES) F-Chart software (V10.6.43), and were simulated in steady-state operation with the following simplifying assumptions: no heat losses in the pipes and components, no pressure losses in the heat exchangers and pipes, isenthalpic expansion process, constant isentropic compressor efficiency ($\eta_{is} = 0.7$), and counter-current heat exchanger configurations. The superheat at the evaporator outlet (ΔT_{SH}) was set between 5 and 25 K, depending on the refrigerant and operating conditions, in order to prevent wet compression. Following the approach of Liang et al. (2019) and Sarkar et al. (2007), the gas cooler was divided into n elements (n = 50 in the simulations) to solve the pinch point problem. At the i-th element ($1 \leq i \leq n$) the refrigerant enthalpies were calculated as

$$h_{gc,i} = h_{gc,out} - \frac{i}{n}\left(h_{gc,in} - h_{gc,out}\right) \qquad \text{Eq. (1)}$$

At a given pressure in the gas cooler (p_{gc}), the temperature of the refrigerant is determined as $T_{gc,i}(h_{gc,i}, p_{gc})$. The pinch temperature difference of the i-th element can be approximated by $\Delta T_{pinch,i} = min\ (T_{gc,i} - T_{SI})$. Consequently, the minimal temperature difference of the whole gas cooler can be found by

$$\Delta T_{min,gc} = \min\left(\Delta T_{pinch,1}, \Delta T_{pinch,2}, \dots, \Delta T_{pinch,n}\right) \qquad \text{Eq. (2)}$$

A pinch point temperature difference of 5 K was set ($\Delta T_{min,gc,set}$) in the simulation program. The Min/Max-function of EES was used to find the minimum of the variable $y = \left|\Delta T_{min,gc} - \Delta T_{min,gc,set}\right|$, which enabled to evaluate the refrigerant temperature leaving the gas cooler ($T_{gc,out}$). By means of a parameter variation of the high pressure (p_{gc}) (and the intermediate pressure p_{int} in the flash tank) and calculation of the COP an optimal pressure ($p_{gc,opt}$ and $p_{int,opt}$) is determined at which the COP is maximized for a certain operating condition and refrigerant. The energy efficiency of the heat pump was evaluated by the coefficient of performance (COP) with the heat release to the heat sink divided by the compressor power(s).

$$COP = \frac{h_2 - h_3}{h_2 - h_1} \text{ (basic, IHX cylce) and } COP = \frac{h_2 - h_3}{x_5(h_{12} - h_{11}) + (1 - x_5)(h_2 - h_1)} \text{ (parallel compression)} \qquad \text{Eq. (4)}$$

In contrast to the Carnot COP in subcritical cycles, the theoretical maximum COP in transcritical cycles with large temperature glides is defined by the Lorenz COP, which uses the thermodynamic average temperatures during heat rejection (heat sink: SI) and absorption (heat source: SO), as follows:

$$COP_{Lorenz} = \frac{T_{SI}^{mean}}{T_{SI}^{mean} - T_{SO}^{mean}} \text{ with } T_{SI}^{mean} = \frac{T_{SI,out} - T_{SI,in}}{ln(T_{SI,out}/T_{SI,in})} \text{ and } T_{SO}^{mean} = \frac{T_{SO,in} - T_{SO,out}}{ln(T_{SO,in}/T_{SO,out})} \qquad \text{Eq. (5)}$$

The Lorenz efficiency is defined as $\eta = COP/COP_{Lorenz}$ and is 6.25 and 5.83 for the case studies 1 and 2.

4. SIMULATION RESULTS AND DISCUSSION

Table 3 summarizes the simulation results of the two case studies in the three investigated transcritical cycles. The values correspond to the optimized conditions obtained at 5 K gas cooler pinch temperature ($\Delta T_{min,gc,set}$). Listed are the optimized pressures in the gas cooler, COP, pressure ratios, VHC, efficiency, evaporation pressure, and inlet/outlet temperatures of the gas cooler.

The simulated results are reasonably consistent with the experimental results reported in the literature for R600 (Kimura et al., 2018; Verdnik et al., 2019) and R1234ze(E) (Chahla et al., 2019) (max. 19% overestimation).

Table 3. Heat pump performance factors of the selected refrigerants in the investigated transcritical cycles.

Refrigerant	Case study 1 (SI: 100 → 200 °C, SO: 80 → 75 °C)											Case study 2 (SI: 30 → 200 °C, SO: 30 → 25 °C)										
	p_{gc} [bar]	COP [-]	p_{ratio} [-]	VHC [kJ/m^3]	$T_{gc,in}$ [C]	$T_{gc,out}$ [C]	η [-]	ΔT_{SH} [C]	p_{evap} [bar]	p_{int} [bar]	$p_{ratio,2nd}$ [-]	p_{gc} [bar]	COP [-]	p_{ratio} [-]	VHC [kJ/m^3]	$T_{gc,in}$ [C]	$T_{gc,out}$ [C]	η [-]	ΔT_{SH} [C]	p_{evap} [bar]	p_{int} [bar]	$p_{ratio,2nd}$ [-]
Basic cycle																						
R601	36	3.3	12.6	2'922	204	105	0.53	25	2.8	-	-	35	2.9	62	917	203	35	0.50	30	0.6	-	-
R514A	44	3.3	12.0	3'710	205	105	0.54	10	3.7	-	-	43	2.9	60	1'138	204	35	0.50	13	0.7	-	-
R1234ze(Z)	56	3.3	8.3	6'071	208	105	0.52	5	6.7	-	-	47	2.9	31	2'175	209	35	0.50	5	1.5	-	-
R1233zd(E)	56	3.2	11.0	4'714	205	105	0.51	5	5.1	-	-	54	2.9	50	1'613	204	35	0.50	5	1.1	-	-
R1224yd(Z)	73	3.1	12.8	5'069	205	105	0.50	5	5.7	-	-	69	2.9	55	1'805	204	35	0.50	9	1.2	-	-
R245fa	89	3.1	14.5	5'598	205	105	0.49	5	6.1	-	-	83	2.9	68	1'867	204	35	0.50	7	1.2	-	-
R600	93	3.0	11.4	6'451	205	105	0.48	5	8.1	-	-	87	2.9	42	2'678	204	35	0.49	8	2.1	-	-
IHX cycle																						
R601	35	3.3	12.3	2'968	204	105	0.53	5	2.8	-	-	35	2.7	62	844	203	54	0.46	5	0.6	-	-
R514A	37	3.5	10.1	3'883	209	105	0.55	5	3.7	-	-	45	2.9	62	1'132	204	35	0.50	5	0.7	-	-
R1234ze(Z)	46	3.4	6.9	6'187	219	105	0.55	5	6.7	-	-	45	2.9	30	2'173	211	35	0.50	5	1.5	-	-
R1233zd(E)	44	3.4	8.6	4'936	207	106	0.54	5	5.1	-	-	52	2.9	48	1'622	205	35	0.50	5	1.1	-	-
R1224yd(Z)	52	3.4	9.0	5'395	205	105	0.54	5	5.7	-	-	67	2.9	54	1'805	204	35	0.50	5	1.2	-	-
R245fa	60	3.4	9.8	5'939	205	105	0.54	5	6.1	-	-	79	2.9	65	1'872	205	35	0.50	5	1.2	-	-
R600	62	3.3	7.7	6'796	205	106	0.53	5	8.1	-	-	83	2.9	40	2'681	204	35	0.49	5	2.1	-	-
Parallel compression cycle with IHX																						
R601	34	3.2	12.0	2'799	205	115	0.51	5	2.8	7.3	4.7	36	2.6	63	811	203	60	0.45	10	0.6	1.5	24.0
R514A	38	3.5	10.3	3'809	206	105	0.56	5	3.7	7.7	4.9	45	2.7	62	1'066	204	49	0.47	10	0.7	1.3	34.2
R1234ze(Z)	47	3.4	7.0	6'080	216	106	0.55	5	6.7	13.8	3.4	39	2.8	26	1'991	217	54	0.47	5	1.5	3.7	10.5
R1233zd(E)	41	3.2	8.0	4'519	210	119	0.52	5	5.1	13.9	3.0	41	2.7	38	1'453	207	61	0.45	5	1.1	3.5	11.6
R1224yd(Z)	50	3.2	8.7	5'014	207	115	0.52	5	5.7	14.2	3.5	55	2.7	44	1'655	205	56	0.46	5	1.2	3.4	16.2
R245fa	66	3.3	10.8	5'832	206	105	0.53	5	6.1	12.5	5.3	65	2.7	53	1'702	205	56	0.46	5	1.2	3.2	20.3
R600	68	3.3	8.4	6'703	206	105	0.52	5	8.1	15.2	4.5	75	2.7	36	2'529	205	48	0.47	5	2.1	4.0	18.6

Figure 2 visualizes the optimal COP, volumetric heating capacity ($VHC = \rho_1(h_2 - h_3)$) and p_{gc} results for the different refrigerants derived in the basic transcritical cycle (light area) and with IHX (darker area) for case study 1. The optimal COP was found by varying the gas cooler pressure. As an example, the optimal COP-line for R1233zde(E) is indicated as a dotted line in Figure 2(A). The COP increases fast with higher pressure until it reaches a maximum and then decreases continuously. This is due to the change of the pinch location in the gas cooler at different pressures. The COP values increase for all refrigerants in the IHX cycle, e.g. for R600 from 3.0 to 3.3. The shift to lower gas cooler pressure by IHX integration is obvious in Figure 2 (A) and (C). For R600 this extends from about 93 to 62 bar, i.e. a noticeable reduction of 33%.

As shown in Figure 2 (B), a compromise between COP and VHC needs to be found when selecting the refrigerant. The VHC is a good indicator for the required size of the compressor and for the investment costs. The VHC values of the refrigerants exhibit a slight increase of 2% to 6% when adding an IHX in the cycle, which is due to a slightly higher enthalpy difference during heat rejection in the gas cooler.

The pressure ratios, listed in Table 3 (case study 1), vary depending on the refrigerant between 8.3 to 14.5 in the basic cycle and 6.9 and 12.3 with IHX. The required pressure ratios can be achieved with two-stage compression assuming a pressure ratio of at least 3.8 per stage. R1234ze(Z) requires the lowest pressure ratio of 6.9 in the IHX cycle. R1234(Z) together with R600 show the highest VHC values, which is advantageous when dimensioning the compressor size. The COP of the investigated refrigerants are closer together in the IHX cycle and range from 3.3 to 3.5. Therefore, the selection of refrigerants can focus on VHC and the safety properties. In comparison, case study 2 (heating from 30 to 200 °C) requires much higher pressure ratios in the range of 30 to 65. At least three compressor stages are required, which is typically not economical.

R1234ze(Z) with the highest VHC, R1224yd(Z), R1233zd(E), and R514A are particularly suitable as potential synthetic refrigerants. As R1224yd(Z), R1233zd(E) and R514A are already commercially available as Amolea™ 1224yd, FORANE® HTS 1233zd or Solstice® zd, and Opteon™ XP30 for low pressure centrifugal chiller applications, these refrigerants can also be suggested for transcritical HTHP. R1233zd(E) and R1224yd(Z) have in particular a low pressure ratio of 8.6 and 9.0, respectively. Regarding the hydrocarbons, R601 requires a relatively low gas cooler pressure, but has low VHC, which indicates a large dimensioned system. In contrast, R600 has a higher VHC but requires a high pressure in the gas cooler. As several studies in the introduction show, R600 is typically preferred.

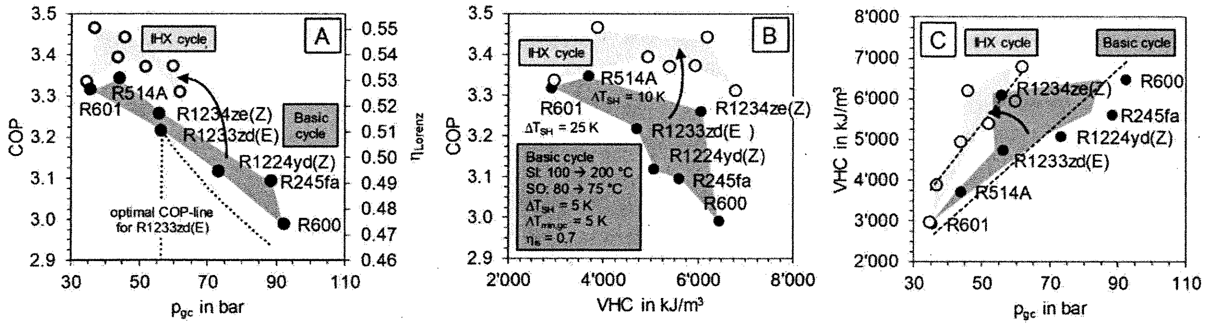

Figure 2: Optimal COP and VHC derived by variation of the gas cooler pressure (p_{gc}) for the different refrigerants in a basic transcritical cycle and with IHX.

Figure 3 illustrates the influence of the variation of the external parameters ($\Delta T_{min,gc}$, η_{is}, ΔT_{SH}, and $T_{SO,in}$) on the optimal COP in a basic R600 transcritical cycle. The gas cooler pressure is the varible of interst for optimization as there exits an optimum leading to a maximum COP for a given condition. The location of the pinch point in the gas cooler depends on the gas cooler inlet state and thus on the discharge pressure (state point 2). This is due to the gliding temperature at heat rejection in a transcritical operation and the curvature of supercritical isobars around the critical point. The arrows in Figure 3 (A) to (D) indicate the extent of the factor influencing the optimal COP. As expected, the COP increases with lower temperature difference in the gas cooler, higher compressor efficiency, higher superheat, and higher heat source temperature.

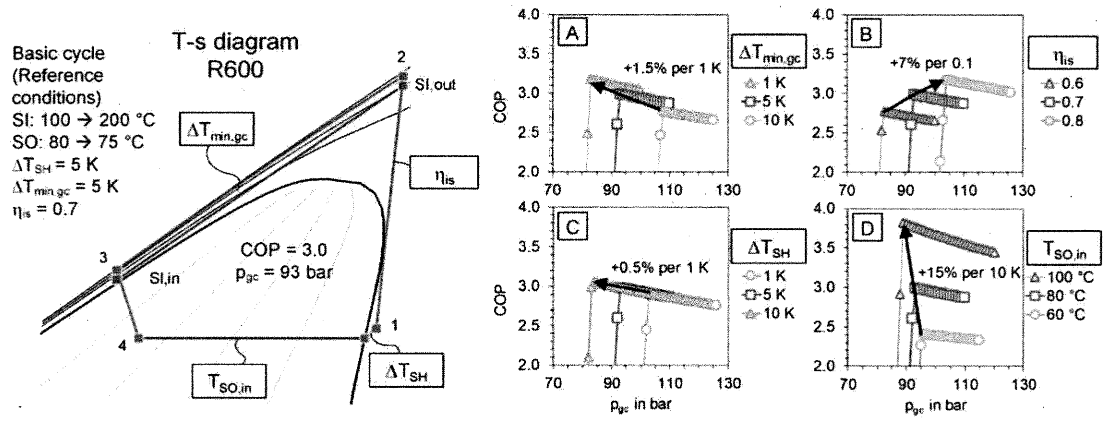

Figure 3: Influences of (A) gas cooler pinch temperature, (B) isentropic compressor efficiency, (C) superheating, and (D) heat source inlet temperature on the COP in a basic R600 transcritical cycle.

A lower evaporation temperature leads to a larger temperature lift that results in a lower cycle performance. The optimum pressure varies almost linearly with the external parameter changes. Therefore, a simple control device of the gas cooler pressure is sufficient, which can be achieved by adjusting the refrigerant mass in the cycle by a tank and a pressure control valve (Verdnik et al., 2019).

Figure 4 compares the three cycles for the refrigerant R1233zd(E). The IHX cycle achieves the highest COP, followed by parallel compression and the basic cycle. The IHX cycle is easy to control and offers the advantage of significantly reducing the discharge pressure, and thus the pressure ratio. A pressure ratio of 8.6 is technically feasible with one or two compression stages.

In the parallel compression system, the intermediate pressure is an additional optimization variable. The simulation results are shown in Table 3. In contrast to refrigeration applications, the concept of parallel compression in general does not lead to improvement in efficiency compared to an IHX cycle. Major reason is that economizing reduces the discharge temperature. In order to reach the desired heat sink outlet temperatures, higher discharge pressures are needed in this cycle. Under optimized COP conditions, the quality in the flash tank (x_5) remains in the range of only 4% to 5%. The parallel compression cycle is more complex to control and an additional compressor is necessary, which results in additional costs. There is also the risk of wet compression, especially with R601, R514A, R245fa, and R600, which requires sufficient superheat at the inlet of the 2nd compressor. Overall, parallel compression has more effect in refrigeration applications when the system operates at lower evaporating temperatures.

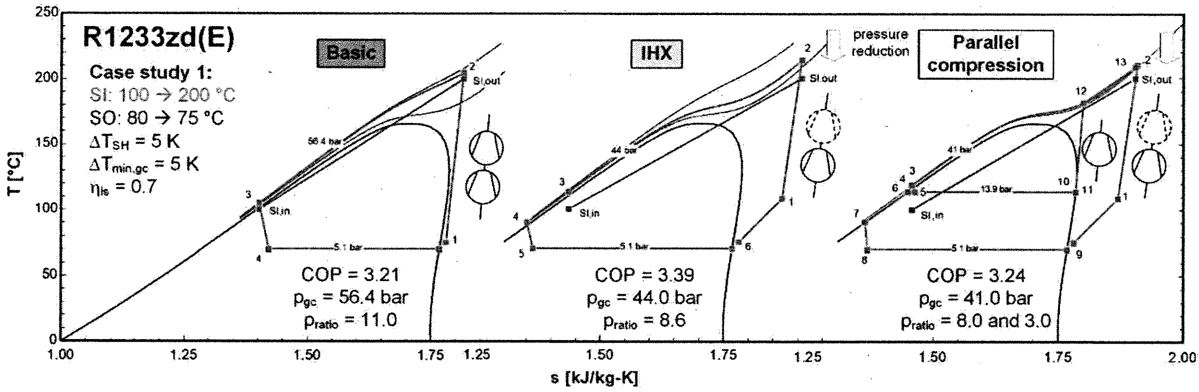

Figure 4: Comparision of transcritical cycles with R1233zd(E).

Considering the technical feasibility, the high discharge pressure of the transcritical cycles, the discharge temperature of about 203 to 219 °C, as well as the stability of lubricant oils at high temperature can be a technological brake. As can be seen in Table 3, the highest gas cooler pressure in the IHX-cycle is in the order of 63 bar for R600. The currently available CO_2 compressors can be operated up to a discharge pressure of about 150 bar (Sarkar et al., 2007). The pressure limits of available hydrocarbon compressors are about 40 bar. Therefore, it seems to be possible to adapt CO_2 compressor designs to other refrigerants for the development of high-pressure compressors in transcritical heat pump cycles.

A suitable refrigerant-oil combination is required to ensure sufficient lubrication of the compressor at temperatures above 200 °C. The lubricant must maintain its stability at high temperatures, which requires sufficient solubility of the refrigerant, sufficient viscosity and other factors. Special synthetic lubricants based on ester oils such as polyolester (POE) or polyalphaolefin (PAO) oils are recommended (Verdnik et al., 2019). According to oil manufacturers, the miscibility of R1233zd(E) and R1224yd(Z) with high viscosity POE is satisfactory. In addition, there are possibilities to apply stabilisator technology.

Furthermore, oil-free compressors (e.g. turbo compressors) are expected to increase in popularity, mitigate the challenges of oil migration and improve compressor efficiency. This is also shown by the study of Kimura et al. (2018) testing oil-free 1- to 3-stage turbo compressors with a speed of 45 to 70 krpm in a R600 transcritical cycle up to 180 °C. Another advantage is the lower maintenance required of turbo compressors. Nevertheless, compressor sizing and oil selection needs a case-by-case assessment.

5. CONCLUSIONS

In this study a simplified simulation model of transcritical heat pump cycles was developed to determine the efficiency of natural hydrocarbons (R601, R600) and HFO/HCFO refrigerants (R1233zd(E), R1224yd(Z), R1234ze(E), R514A) for two heating applications up to 200 °C with large temperature glides on the heat sink. The pressure of the gas cooler during heat rejection is the variable of interest for optimization. The COP drops quite rapidly at pressures below the optimum pressure. The evaluation shows a heat pump cycle with IHX as the best cycle option. For heating applications from 100 to 200 °C (80 °C heat source), the COP of the investigated refrigerants are between 3.3 and 3.5. The IHX cycle is easy to control and significantly reduces the discharge pressure, and thus the pressure ratio. R1233zd(E), R1224yd(Z) and R600 are the best candidates in terms of high VHC and reasonable pressure ratio and can be selected for further evaluation. The required pressure ratios of 6.9 to 12.3 with IHX is technically feasible with one or two compression stages. Technical challenges arise in the selection of the refrigerant/oil combination and the application of temperature and pressure resistant compressors. Overall, it is worth investigating transcritical heat pump cycles in order to extend the operating range of HTHP beyond the critical refrigerant temperature towards higher supply temperatures and thus to achieve a higher application potential in industry. For future research other transcritical cycles e.g. with multi-stage compression, ejectors, expanders etc. and other temperature conditions could be analysed.

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NOMENCLATURE

COP	coefficient of performance (-)	*HCFO*	hydrochlorfluorolefin	*pinch*	minimal temperature difference (°C)
crit	critical	*HTHP*	high temperature heat pump	*POE*	polyol ester
CFC	chlorofluorocarbon	*i*	iteration step	*s*	spec. entropy (kJ/kg K)
evap	evaporator	*in, out*	inlet, outlet	*SG*	safety group
gc	gas cooler	*is*	isentropic	*SH*	superheat (°C)
GWP	global warming potential	*IHX*	internal heat exchanger	*SI, SO*	heat sink, heat source
h	spec. enthalpy (kJ/kg)	*NBP*	normal boiling point	*T*	temperature (°C)
HFC	hydrofluorcarbon	*ODP*	ozone depletion potential	*VHC*	vol. heating capacity (kJ/m^3)
HFO	hydrofluorolefin	*p*	pressure (bar)	*x*	quality (-)

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