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Theoretical investigation of high-temperature heat pump cycles for steam generation

Frédéric Bless*, Cordin Arpagaus, Stefan Bertsch

NTB University of Applied Sciences of Technology Buchs, Werdenbergstrasse 4, 9471 Buchs, Switzerland

Abstract

This paper presents a thermodynamic analysis of different heat pump cycles that are capable of delivering 150 °C (4.8 bar) saturated steam from 80 °C recovered waste heat. A detailed comparison of the advantages and disadvantages of the following cycles is presented:

1. Transcritical cycle using butane (R600),
2. Classical closed-cycle using low GWP hydrofluoroolefin (HFO) refrigerant (e.g. R1233zd),
3. Reversed Brayton cycle using carbon dioxide (R744) or argon (R740),
4. Closed-cycle using R600 or R1234ze(Z) with flash tank and additional steam compressors for mechanical vapor recompression, and
5. Open-cycle using water (R718) as working fluid and multiple steam compression stages.

Performance parameters such as COP, compressor efficiency, pressure ratio, steam generation rate, cycle controllability, safety, running and investment costs, and component availability are evaluated and discussed. Under the assumed operating conditions (e.g. temperature lift of approx. 70 K), the theoretically calculated COPs of these cycles range between 1.3 and 3.1. In the transcritical R600 cycle, only a small part of the heat can be used to generate steam, limiting the COP to about 1.3. However, the efficiency can be increased by parallel hot water generation in a second gas cooler. Decisive for a system with a classical closed-cycle heat pump (COP of 1.8) are the use of an environmentally friendly refrigerant (e.g. HFO) and a temperature-resistant compressor with cooling function and stable lubrication. The Reversed Brayton cycle (COP of 2.3) requires a compressor with very high isentropic efficiency (> 0.9) in order to achieve reasonable efficiencies. Depending on the application, a temperature glide on the heat source can be advantageous to transfer more heat. The availability of oil-free compressors is crucial for open steam compression cycles (COP of 3.1). Their achievable pressure ratio determines the number of compression stages. Overall, the study contributes to a better understanding of steam generating heat pump cycles for industrial heating processes and it highlights the current research and technological gaps.

Keywords: high-temperature heat pump; heat pump; steam generation; cycle analysis, transcritical heat pump

1. Introduction

1.1. Steam generation heat pumps (SGHP)

Electrification of heat generation in the industrial sector is a trend, which is continuously growing. Heating with fossil fuel has a higher environmental impact. Furthermore, due to the decreasing cost of renewable electricity and the introduction of a carbon tax in some countries, heating by fossil fuel is not always the cheapest solution anymore [1]. Heat pump technology using energy from waste heat, ground heat or air, offers

* Corresponding author. Tel.: +41 81 755 31 48
E-mail address: frederic.bless@ntb.ch

an efficiency that is much higher than a heating system based on combustion [2]. However, heat pump systems are limited in the maximal achievable supply temperature. Thus, high-pressure steam (e.g. 15 bar and higher) cannot easily be generated efficiently by heat pump systems, but low-pressure steam (e.g. < 6 bar) can be produced by heat pump processes, especially if a heat source with a high temperature (e.g. waste heat of 60 °C to 80 °C) is available. The advantages of steam compared to hot water are especially the much better heat transfer due to the latent heat release during condensation and the easier temperature control [3].

Mechanically driven vapor compression heat pump systems for steam generation is an evolving technology that is becoming more widespread in the coming years to reduce primary energy consumption and greenhouse effect gas emission [4]. A characteristic constructive difference to a conventional heat pump is the condensing part. The generated steam is either used directly or further recompressed by mechanical vapor recompression (MVR) [5]. A flash tank is normally used as a reservoir for the two-phase water.

The advantages of a steam generation heat pump (SGHP) are in particular the recovery of industrial waste heat in the range of approx. 60 °C to 100 °C and the conversion into low-pressure steam (e.g. < 6 bar) for many potential application fields, e.g. distillation of alcohol, condensation, disinfection, sterilization of food and beverages, concentration of liquids and juices, humidification, or drying processes [4, 6, 7, 8]. Steam generation takes place at a relatively low temperature when using waste heat, thus avoiding exergy losses compared to the combustion of fossil fuels at high temperatures [6, 7]. Moreover, a SGHP can generate more heat from the same primary energy source than a fossil-fueled boiler [9].

1.2. Literature review on SGHP

Several authors investigated the suitability of SGHP by thermodynamic simulations [2, 10, 11, 12]. Experimental investigations are rather rare. Table 1 summarizes a selection of experimental research studies on SGHP categorised by organization, heating capacity, heat source and steam temperature, heat pump cycle, compressor, refrigerant, and COP.

Table 1. Selection of experimental research studies of steam generation heat pumps (SGHP)

Organization, country	Heating capacity (kW)	Heat source temperature (°C)	Steam temperature (°C) (flow rate kg/h)	Heat pump cycle, compressor	Refrigerant	COP (source/sink temperature °C)	Reference (year)
Korea Institute of Energy Research (KIER)	300	60	128 (422)	HTHP + flash tank, piston	R245fa	n.a.	Chang et al. (2016) [13]
	100	70	120	HTHP + flash tank, open screw	R245fa	3.05 (70/120)	Wang et al. (2018) [14]
	25	60	104 - 123	HTHP + IHX + flash tank + valve	R245fa	~ 3.5 (60/105)	Lee et al. (2017) [15]
Seoul National University, Korea	6 - 8	60 - 70	115 - 125 (10.8)	HTHP, piston	R245fa	2.95 (60/125) 3.59 (60/115)	Kang et al. (2019) [9]
	6 - 12	60 - 80	115 - 125	HTHP + steam reservoir + MVR	R245fa	3.39 (80/125) 2.72 (60/115)	Yoo et al. (2017) [3]
Tokyo Electric Power, Mayekawa, Japan	400	80 - 90	130	HTHP, screw	R601 (pentane)	4.5 (80/130)	Yamazaki and Kubo (1986) [16]
Kobe Steel, Ltd., CRIEPI, and electric companies, Japan	660	35 - 70	165 (890)	HTHP + MVR, screw	R134a/R245fa (SGH165)	2.5 (70/165)	Kaida et al. (2015) [4]
	380	25 - 65	120 (20)		R245fa (SGH120)	3.2 (65/120)	Kuromaki (2012) [8]
Mayekawa, Waseda University, Japan	300	80	100 - 180 (thermal oil)	Transcritical HTHP, centrifugal	R600 (butane)	3.5 (80/180) calculated	Kimura et al. (2018) [17]
Shanghai Jiao Tong University, China	285	75 - 85 (evaporation)	111 - 150 (condensation)	VHTHP + flash tank, twin-screw	R718 (water)	6.10 (85/117) 1.96 (85/150)	Wu et al. (2020) [18]
ECN, IBK, Bronswerk, Smurfit-Kappa, Netherlands	160	60	125 (2.4)	HTHP + IHX + subcooler, piston	R600 (butane)	1.9 (60/125)	Wemmers et al. (2017) [19]
Olvondo Technology, TINE dairy, Norway	449	80 - 90	184 (10)	HTHP (reversed Stirling cycle), piston	R704 (helium)	2.1 (85/183)	Tveit (2017) [20]
NTNU, SINTEF, Norway	20	25 - 35	115	HTHP cascade + IHX	R290/R600 (propane/butane)	2.1 (25/115)	Bamigbetan et al. (2019) [21]
AlterECO project, EDF, France	200	35 - 60	80 -140 (condensation)	HTHP + IHX + subcooler, two scroll	ECO3 containing R245fa	2 - 3 (50-60/125) (evap/cond)	Bobelin et al. (2012) [22]
PACO project, Uni Lyon, EDF, France	380	85 - 95	130 -140 (condensation)	HTHP + flash tank, twin-screw	R718 (water)	~5.5 (94/121)	Chamoun et al. (2012) [23, 24]
National Research Council Canada	45	55 - 80	103.5 - 135.5	HTHP + IHX, piston	R113 & R123 (ozone depleting)	2.7 (75/135, R113) 3.6 (60/120, R123)	Linton (1990, 1993) [25, 26]

The Korea Institute of Energy Research (KIER) has researched on SGHP since about 2013. Lee et al. [15] investigated a 25 kW HTHP for the production of low-pressure steam in a suitable cycle configuration. An internal heat exchanger (IHX) was implemented in the HTHP cycle to ensure superheating of the refrigerant R245fa. The generated heat from the HTHP was transferred in the condenser to a circulating pressurized water circuit. By reducing the pressure, part of the water was converted into steam (flashing method). A steam reservoir separated the steam and the water. In an open circuit configuration, additional water is supplied to the flash tank. With this configuration, steam temperatures of 104 to 123 °C (1.2 to 2.2 bar) were reached. At 60 °C heat source and 105 °C steam temperature a COP of about 3.5 was achieved.

Wang et al. [14] developed a larger SGHP that recovered waste process heat and generated steam at 120 °C. A flash tank served as phase separator. By optimizing the experimental setup, a heating capacity of 101.4 kW and a COP of 3.05 was achieved at 70 °C heat source temperature.

Chang et al. [13] studied the design of a high-temperature condenser for a SGHP with R245fa and 300 kW capacity. For saturated steam of 128 °C (2.3 bar) a steam production rate of 0.422 t/h was evaluated.

Yoo et al. [3] from the Seoul National University designed a laboratory-scale SGHP system and observed a 25% and 80% increase in COP and heating capacity, respectively, with increasing heat source temperature from 60 to 80 °C. The SGHP showed a much better performance compared to a steam boiler.

From the same research group, Kang et al. [6] reviewed recent research studies on SGHP systems. More recently, Kang et al. [9] conducted an experimental study on the performance of a laboratory-scale SGHP with IHX for performance increase. The system consists of an HTHP with a second stage using a flash tank and MVR. The COP was 2.95 at 125 °C, 3.24 at 120 °C and 3.59 at 115 °C steam temperature. Up to 10.8 kg/h steam was generated. The integration of an IHX could increase the amount of steam by 22%.

In Japan, already in 1985, the researchers Yamazaki and Kubo [16] presented a 400 kW heat pump operated with pentane (R601) providing condensation temperatures of up to 135 °C. The heat pump generated low-pressure steam of 130 °C using waste heat of 90 °C as heat source. A variable-capacity screw compressor was used with a motor power of 75 kW. A COP of 4.5 was reached at 80 °C evaporation and 135 °C condensation.

In 2011, Kobe Steel Ltd., CRIEPI (Central Research Institute of Electric Power Industry), and Japanese electric companies have jointly developed and commercialized the KOBELCO Steam Glow Heat Pump (SGH) SGH120 and SGH165, which enable steam temperatures of 120 °C and 165 °C (6 bar). The rated performance of SGH120 is 520 kg/h of steam with a COP of 3.5 at 65 °C heat source [8]. The SGH165 uses an additional water flash tank and MVR to achieve a higher temperature and steam pressure. About 890 kg/h of steam at 165 °C is generated with a COP of 2.5 when using waste heat of 70 °C. Kaida et al. [4] from CRIEPI investigated experimentally the steam generation rate, energy efficiency and controlled performance of the SGH165 under various operating conditions. More recently, Kaida et al. [27] also presented first experimental results of R1224yd(Z) as an environmentally friendly alternative to R245fa, which is to be phased out in the foreseeable future due to high Global Warming Potential (GWP) of 858. At operating point W50/W95, R1224yd(Z) exhibited a 3% higher heating capacity and 12% higher COP compared to R245fa.

Latest developments in heat pump technology in Japan include a transcritical butane (R600) heat pump with a target maximum temperature of 200 °C and a calculated COP above 3.5 at 80 °C heat source (i.e. waste heat). In a joint project with Mayekawa and Waseda University, Kimura et al. [17] developed such a prototype for heating thermal oil from 100 ˚C to 180 ˚C. The experimental setup consists of two oil-free centrifugal compressors with active magnetic bearings. The gas cooler has a capacity of 300 kW. Due to the flammability of the refrigerant, the test apparatus is explosion-proof and enclosed in a housing with active ventilation. Experimental performance evaluation of the system is ongoing.

For condensation temperatures beyond 140 °C to 150 °C, water (R718) is becoming increasingly efficient and very attractive compared to other refrigerants. There are a few studies on HTHPs using water as refrigerant, as summarized by Bertsch et al. [28]. Bless et al. [2] theoretically analyzed different steam generation methods with open cycles. Results showed that water vapor compression methods using water injection require about three times less energy than direct heating systems (i.e. gas-fired boilers or electrical heating).

Shibata et al. [7] from Hitachi proposed an SGHP with water as refrigerant and a four-stage centrifugal compressor with water spray intercooling. The proposed system generates 5'000 kg/h of steam at 130 °C by recovering the waste heat from 60 °C hot water. A test apparatus was developed to validate the first stage of the steam compressor at sub-atmospheric pressure, i.e. compression from 0.2 bar (65 °C) to 0.4 bar (148 °C).

In China, Wu et al. [18] from Shanghai Jiao Tong University investigated the performance of a very high temperature heat pump (VHTHP) prototype using water as refrigerant. An oil-free twin-screw water vapor compressor with water injection was developed. With condensing temperature rising from 117 °C to 150 °C and constant evaporating temperature of 85 °C, the COP decreased from 6.10 to 1.96. The heating capacity decreased from 285 kW to 162 kW, respectively.

In France, Bobelin et al. [22] from EDF (Électricité de France) were among the first to build a 200 kW VHTHP pilot plant in 2012, which can supply hot water of up to 140°C, or low-pressure steam. The heat pump operates with IHX, subcooler, and two parallel scroll compressors. ECO3, a refrigerant mixture containing R245fa, was used. COPs of 2 to 3 were achieved at 125 °C condensation and 50 °C to 60 °C evaporation temperature. Chamoun et al. [23, 24] presented experimental investigations of a water (R718) vapor HTHP pilot plant in 2012 for industrial heat recovery. After condensation, the water was collected in an accumulator and was expanded to low pressure in a flash tank. The main technical feasibility was demonstrated attaining a condensation temperautre of up to 145 °C. Waste heat of 85 °C to 95 °C was used as source providing a heat output of more than 380 kW. At 94 °C source and 121 °C sink temperature, a COP of about 5.5 was measured.

In the Netherlands, Wemmers et al. [19] designed a pilot R600 heat pump able to deliver low-pressure steam of up to 2.4 bar (125 °C) from 60 °C waste heat at a COP of 1.9. The HTHP was developed from commercially available components to ensure quick market introduction. The heat pump configuration also includes a subcooler in order to heat process water at 70 °C.

In Norway, in a cooperation project between NTNU and SINTEF, Bamigbetan et al. [21] reported on the experimental performance of a 20 kW cascade HTHP with propane (R290) in the low-stage and butane (R600) in the high-stage for high temperature heating up to 115 °C. The heat pump was found to have an average COP of 2.1 for a temperature lift of 98 to 101 K and 3.1 for 58 to 72 K, which makes it a better alternative to gas boilers at current carbon emissions and electricity cost rates.

Tveit [20] from Olvondo Technology AS presented an industrial HTHP capable of producing steam up to 10 bar (184 °C) using 80 °C to 90 °C from district heating as the heat source. The so-called SPP HighLift heat pump is based on a reversed Stirling cycle with helium (R704) as working medium, which is continuously in the gas phase. A closed hot water circuit heats the feed water to the saturated steam temperature. In a case study in a dairy plant, the heat pump generates steam for sterilizing UHT milk (for long-life products) at 135 °C to 150 °C for a few seconds. Simulated data showed a heat capacity of 449 kW and a COP of 2.1 at 183 °C heat sink and 85 °C heat source temperature.

Finally, it is worth mentioning the works of Linton [25, 26] from the National Research Council Canada from 1990, who presented a 45 kW steam generating HTHP test facility running with R113 (ozone depletion potential ODP of 0.85) and R123 (ODP of 0.03). This was before the Montreal Protocol entered into force, which banned ozone-depleting substances. The heat pump contained an IHX and a piston compressor with a water-cooled oil cooler. The test stand operated with 55 °C to 80 °C hot water as heat source and produced saturated steam from 103.5 °C to 135.5 °C. The COP with R113 was 2.7 at 75 °C/135 °C and 3.2 with R123 at 65 °C/135 °C.

In summary, the above literature review shows that experimental research on SGHP has intensified worldwide in recent years. Most studies are based on laboratory and prototype tests with heating capacities in the range of a few 100 kW. The majority of pilot plants generate steam temperatures of 130 ± 20 °C. Often a multi-stage configuration with one HTHP and one condenser with pressurized water circuit, flash tank and MVR is used. The average COP is around 3.2 at a temperature lift of 60 K using 70 °C waste heat.

As can be seen, R245fa is predominantly used in SGHP as refrigerant as it offers a suitable temperature range. For subcritical processes, a high critical point for two-phase heat exchange at the steam generation temperature is crucial. However, because of its high GWP of 858 researchers turned to the use of natural refrigerants like hydrocarbons (R600) or water (R718). New synthetic hydrofluoroolefins (HFO) and hydrochlorofluoroolefins (HCFO) with very low GWP such as R1233zd(E), R1224yd(Z) and R1336mzz(Z) are also preferred and investigated. More details on HTHP can be found in Arpagaus et al. [29, 30].

Beside mechanically driven vapor compression heat pumps, thermally driven absorption cycles are also an option for waste recovery and steam generation. As these cycles are not the focus of this study, reference is made to relevant review literature for further details [31, 32].

1.3. Goals of this study

The goal of this study is to analyze different heat pump cycles in order to produce saturated steam at 150 °C (4.76 bar). The influence of various refrigerants, heat source temperatures, and isentropic efficiency of the compressor(s) on their efficiency, cost, and complexity are investigated theoretically. This study compares the following five different heat pump cycles:

(1) ***Transcritical cycle:*** A transcritical cycle is when the heat sink temperature is above the critical temperature of the refrigerant. The heat is suppled via a gas cooler (not a condenser) with a temperature glide. The glide can be an asset in terms of efficiency for certain applications. However, when it comes

to evaporating a fluid, a temperature glide is a disadvantage. In this study, the natural refrigerant butane (R600) is used to generate heat above 155 °C, making the cycle transcritical.

(2) ***Classical closed-cycle HTHP:*** A HTHP is simulated using the common subcritical closed-loop cycle. The condensation temperature was fixed at 155 °C. At this temperature there are only a few refrigerants available, which can be used below their critical temperature. R1233zd(E) was used in this simulation as is it one of the few low GWP HCFO refrigerants with a critical temperature higher than 150 °C.

(3) ***Reversed Brayton cycle:*** A reversed Brayton cycle, also known as gas refrigeration cycle or Bell Coleman cycle, was simulated as well. In a reversed Brayton cycle, the refrigerant stays in gaseous state. In order to use refrigerants that are in the gas state at the low temperature of the heat source, the refrigerants used in the simulation were R744 (CO_2) and R740 (argon).

(4) ***Closed-cycle heat pump with mechanical vapor recompression:*** As only a few refrigerants are subcritical up to 155 °C a combined cycle was simulated, consisting of a low-stage HTHP creating steam at temperatures between 90 °C to 130 °C and a second-stage steam recompression system. This temperature was then fixed to 110 °C in order to optimize the COP while keeping the number of compression stages to two. Depending on the pressure ratio of the steam compressor, multiple compression steps are necessary to reach the desired high pressure. The refrigerants used in the HTHP simulations were R600, R1234ze(Z), and R1233zd(E). The cycle is called high temperature heat pump with mechanical vapor recompression (HTHP with MVR).

(5) ***Open-loop water heat pump:*** This cycle first expands the water to the necessary low pressure in order to evaporate it directly by the heat source. The steam is then re-compressed to reach the desired temperature at the saturation state. Depending on the evaporating pressure and the pressure ratio of the steam compressor, multiple compression stages and inter-cooling sections are needed.

1.4. Refrigerants

The refrigerants studied need to be compatible with the future standards otherwise this study will contain information which will be quickly irrelevant. The European directive on mobile air conditioning systems (MACs) (Directive 2006/40/EC) will prohibit the use of F-gases with a global warming potential (GWP) more than 150 times greater than carbon dioxide [33]. The refrigerants used in the simulations are R1233zd(E), R1234ze(Z), R600 (butane), R718 (water), and R744 (CO_2) with a GWP of less than 10. Some refrigerants like R600a (isobutane) were not simulated because they have very similar properties to the ones mentioned above. Table 2 shows some properties of these refrigerants.

Table 2. Properties of the refrigerants used in this simulation study. ODP = Ozone Depletion Potential (R11 is the ODP reference of 1), GWP = Global Warming Potential (CO_2 is the GWP reference of 1), ASHRAE Standard 34 ranks the toxicity (A, B) and the flammability (1, 2L, 2, 3) of a refrigerant.

Refrigerant	ODP	GWP	Safety group
R1233zd(E)	0.00034	1	A1
R1234ze(Z)	0	6	A2L (slightly flammable)
R600 (butane)	0	4	A3 (flammable)
R718 (water)	0	0.2	A1
R744 (CO_2)	0	1	A1

2. Methodology

To compare the cycles, saturated steam is generated at 150 °C (4.762 bar), and the initial water state is assumed to be 10 °C at 1 bar. The generation of steam is split into two heat exchangers. First, the water is compressed to its evaporating pressure, then the first heat exchanger preheats the water to the evaporating temperature, and finally the second heat exchanger evaporates the water.
Some cycles use steam compression, which compresses the steam to the final state. For example, the open-loop water cycle is different, it first expands the water to a lower pressure, evaporates it, and then compresses it. An IHX is used in some cycles to recover some energy from the subcooling to superheat the refrigerant. The superheating at the compressor inlet is fixed at 10 K for the HTHP and the HTHP with MVR cycles and to 14 K for the transcritical cycle using R600, to avoid wet compression.

The COP of a cycle corresponds to the ratio of the enthalpy difference between the final water state (saturated steam at 150 °C, 4.76 bar) and the initial water state (10 °C, 1 bar) and the work needed by the compressor(s). Each compressor work is calculated by the enthalpy difference between the enthalpy h_1 at the outlet of the compressor and the enthalpy h_0 at the inlet (compressor work = ($h_1 - h_0$)). The outlet enthalpy depends on the isentropic efficiency of the compressor (η_{is}).

EES (Engineering Equation Solver, from F-Chart) is used to simulate the different cycles. However, code for the Reversed Brayton could not converge, thus a self-made Python code using the Coolprop library for the refrigerant data was used instead. The following assumptions are made:

- Pinch point of 5 K in every heat exchanger
- Isentropic efficiency of 0.75 (if not otherwise specified)
- No heat losses in the system (e.g. pipes, heat exchanger, etc.) and no pressure losses
- No energy consumption of water pump(s) (negligible compared to the compressor power)
- Constant temperature glide of 20 K on the heat source
- In the case where heat is delivered at higher and lower temperature than 155 °C, only heat to create the steam is considered (5 K pinch point) and the rest of the energy is used to warm the water from the initial temperature to the final temperature before the evaporation.
- The fresh water is first pumped to its final pressure for all cycles except for the HTHP with MVR cycles and for the open loop R718 cycle (see Figure 1 for more details on the cycles).
- Fixed pressure ratios for steam compressors with a maximum of 2.5. In case of steam compression, the number of compressor stages is thus depending on the pressure ratio of each compressor.
- Between each compression stage, intercooling is added by injecting liquid water at the same pressure to cool the overheated steam to its saturated state.
- Controllability: The heat pump cycles are assumed to be regulated to keep the generated steam constant while the heat source varies.
- Safety and emissions depend on the refrigerant used. The safety concerns may origin from the high pressure needed in the heat pump cycle or from the toxicity or flammability of the refrigerant. The environmental impacts only occur if there is a leakage of the refrigerant. Each refrigerant has a certain GWP summarized in Table 2.

Operating costs are directly dependent on the COP and capital costs vary by cycle. A cost estimation of each component is made to compare the investments costs of the cycles. Using the COP of the cycle, the total cost of the systems is estimated over a period of 15 years. For the reversed Brayton cycle, the expander price is estimated to be the same as a compressor. Equation (1) is used to estimate the price (see Chapter 2 in [34]):

$$C_e = C_b \left({}^{Q}/_{Q_b} \right)^M \cdot f_m \cdot f_p \cdot f_t \tag{1}$$

where C_e is the equipment cost, C_b the known base cost for a capacity of Q_b, M the constant depending of the equipment, and the f's are correction factors for the material of construction, pressure, and temperature. The correction factors where fixed for all components at 1, 1, and 1.6 respectively. C_b, Q_b, and M are given in Table 2.1 of reference [34]. The capacity Q is calculated as function of the cycle for a particular condition.

For the cost analysis, the heating capacity of the heat pump is fixed at 200 kW, which corresponds to about 266 kg/h of saturated steam at 150 °C (4.762 bar). General investment costs are hard to determine and the results have to be taken with care. The yearly electricity consumption cost is calculated using the following equation:

$$\mathit{Yearly\ electricity\ cost} = 7'000\,[\mathrm{h}] \cdot cost_{elec} \left[{}^{\$}/_{\mathrm{kWh}}\right] \cdot \frac{200}{COP}\,[\mathrm{kW}] \tag{2}$$

The cost of electricity ($cost_{elec}$) is 0.13 \$/kWh with taxes, which is approximatively the EU-28 weighted average retail electricity price for industry for the last couple of years (Figure 4 in reference [35]). The yearly maintenance cost was fixed to 6% of the capital cost according to Table 5 from Wang and Zhang [36]. The capital cost of the cycle is an estimation of all the component costs according to equation (3), and the two formulae for the cost of the expansion valve and the piping and instrumentation coming from Table 3 of reference [36].

$$C_{\text{expansion valve}} = 114.5 \cdot \dot{m} \tag{3}$$

where $\dot{m}$ is the mass flow of the refrigerant. The piping and instrumentation costs are assumed to be 28% of the sum of all the components cost (except the steam compressors), as described in [36].

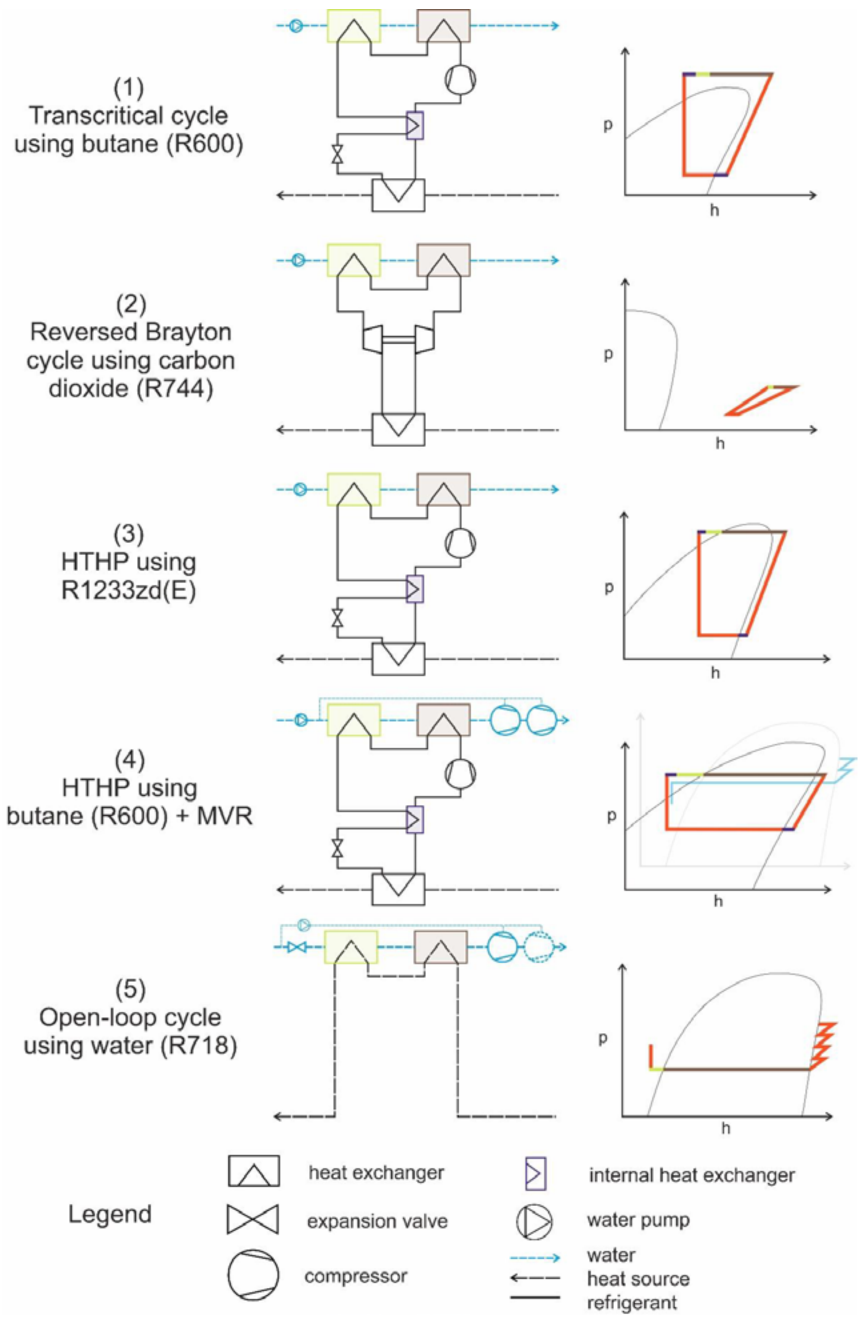

Fig. 1. Schematics and p-h diagrams of the five investigated heat pump cycles. (1) Transcritical R600 cycle, (2) Reversed Brayton cycle with R744 (CO_2) or R740 (Argon). (3) Classical closed-cycle HTHP using R1233zd(E), (4) Closed-cycle HTHP with mechanical vapor recompression (MVR), and (5) Open-loop cycle with R718 (water).

3. Results

3.1. COP simulations

The COP is simulated with different heat source temperatures and different isentropic efficiencies of the compressor(s). Figure 2 shows the COP for the different heat pump cycles generating steam of 150 °C (4.76 bar) as a function of the heat source outlet temperature (with constant temperature glide of 20 K on heat source) with an isentropic efficiency of 75%. As expected, the COP of each cycle increases with higher heat source temperature. The reversed Brayton cycle is the cycle least affected by the increase in source temperature. For the open-loop R718 heat pump cycle, the number of compressions stages varies as follows: six stages for temperatures below 40 °C, five for temperatures between 40 and 50 °C, four for temperatures between 50 and 70 °C, and three for temperatures above 70 °C. Since the steam compressor is not perfect, each compression step leads to a loss in process performance. This explains the lower COP of the open-loop cycle with water at lower source temperature compared to the HTHP with MVR cycles. For heat source temperatures of 75 °C and above the cycle's efficiencies are comparable. The transcritical cycle and the reversed Brayton cycle are not as efficient due to their temperature glides in their heat sink whereas the other cycles have a constant temperature.

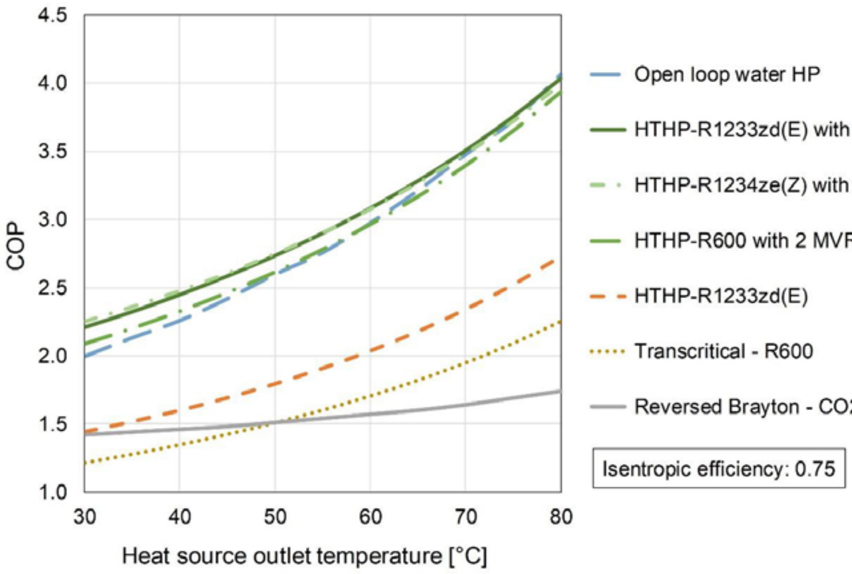

Fig. 2. COP for the different heat pump cycles generating steam of 150 °C (4.76 bar) as a function of the heat source outlet temperature (heat source temperature glide: 20 K, heat sink water inlet: 10 °C, isentropic compressor efficiency: 0.75)

Figure 3 shows the COPs as a function of the isentropic efficiency of the compressor(s) using a heat source inlet/outlet temperature of 80/60 °C. The increase of the COPs is linear except for the reversed Brayton cycle where the relation is much stronger than linear. The reversed Brayton cycle becomes most efficient for steam generation if it is operated with a compressor with a very high efficiency. The HTHP+MVR cycles and the open-loop water HP are still more efficient up to η_{is} of 95%, only with 98.5% and above the reversed Brayton cycle is above all the other cycles.

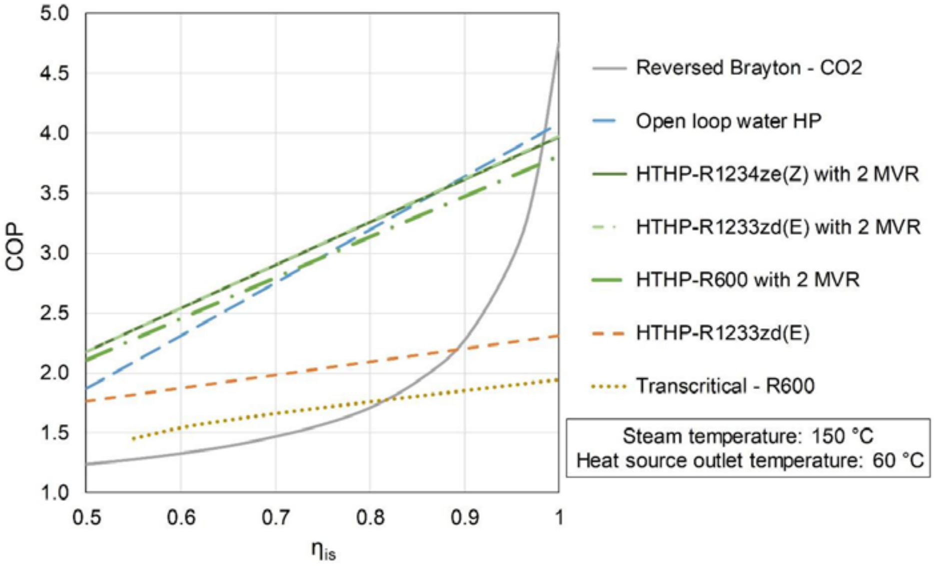

Fig. 3. COP for different heat pump cycles generating steam of 150 °C (4.76 bar) at 60 °C heat source outlet temperature (20 K temperature glide) as a function of the isentropic efficiency of their compressor(s) (η_{is}).

3.2. Cost analysis

Table 3 shows some estimated cost for the representative cycles at 60 °C heat source outlet temperature and η_{is} 75 %. The HTHP-R600 with 2MVR is simulated with a HTHP having a condensation temperature of 110 °C. The HTHP and the transcritical cycles have roughly the same capital cost due to their similarity. The open loop cycle with R718, as seen in Table 3, has a price similar to the previous cycles even though it needs 4 compressors; the reason is this cycle needs less heat exchangers. The reversed Brayton cycle shows the highest cost of all cycles. The main reason is that its compressor needs around twice the power compared to the other cycles. This can be schematically seen in the p-h diagram of the cycle in Figure 1 (2), where the ratio of the enthalpy difference of the compressing step over the heating step is small. Furthermore, the expander also costs much more than the expansion valve used in the other cycles.

Table 3. Estimation of the capital costs, the maintenance, and the electricity costs for a particular condition (60 °C heat source outlet temperature, η_{is} = 75 %) for each cycle. The HTHP-R600 with 2MVR cycle is simulated with a condensation temperature of 110 °C.

Cycle	Capital cost (CC) [$]	Yearly maintenance cost (6% of CC) [$]	Yearly electricity consumption cost [$]
(1) Transcritical R600 cycle	174’054	10’443	106’433
(2) Reversed Brayton with R744	516’697	31’002	115’924
(2) Reversed Brayton with R744 with η_{is} = 95%	640’451	38’427	59’477
(3) HTHP with R1233zd(E)	182’258	10’935	89’216
(4) HTHP with R600 with 2 MVR	217’388	13’043	61’279
(5) Open loop cycle with R718 (4 MVR)	181’868	10’912	61’279

Figure 4 shows an estimation of the total costs (i.e. investment and operating costs) as a function of the year according to the costs displayed in Table 3. The graph shows, which cycles are the most interesting in terms of cost depending on its usage. The transcritical R600 has the lowest capital cost, however, after only one year of operation (by the conditions assumed), the standard HTHP-R1233zd(E) and the open loop R718 HP are cheaper. One year later, the HTHP-R600 with 2 MVR also becomes a cheaper option.

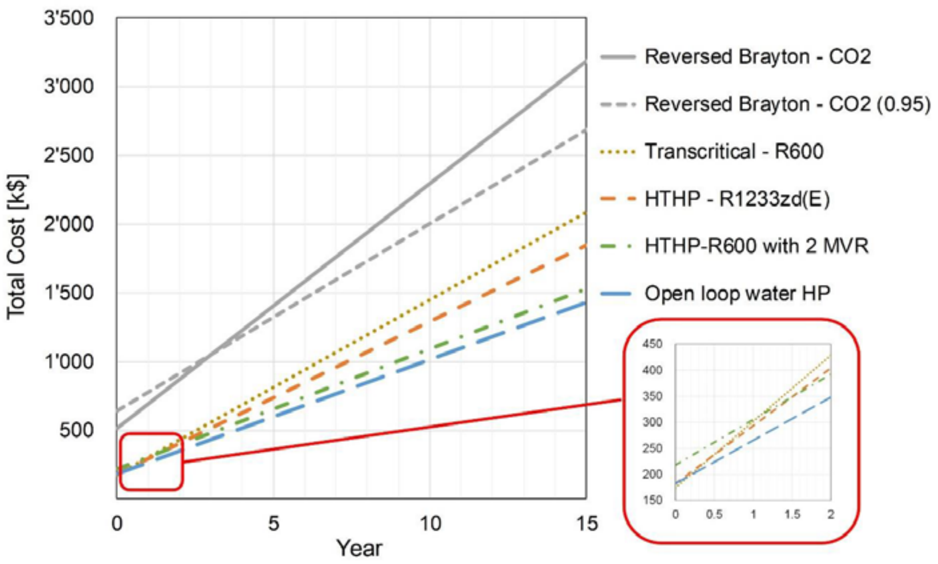

Fig. 4. Total cost (capital cost, maintenance cost, and operating cost) for the different heat pump cycles for 15 years with the following conditions: heat source inlet temperature: 80 °C, isentropic efficiency: 0.75 (except for the second reversed Brayton where it is 0.95), k$ corresponds to 1'000 $.

The maintenance costs were estimated simply by estimating a percentage of the capital cost, but do not consider the simplicity of the maintenance. To get an overview of the complexity of the different cycles, Table 4 summaries the number of components included in each cycle.

Table 4. Components of the five heat pump cycles.

Cycle	Heat exchanger (evaporator, condenser, gas HEX, or IHX)	Compressor	Steam compressor	Expander
(1) Transcritical	4	1		
(2) Reversed Brayton	4	1		1
(3) HTHP	4	1		
(4) HTHP with MVR	4	1	2	
(5) Open loop	2		4	

Table 4 shows that the first three cycles are very similar in terms of components. The reversed Brayton cycle has an additional expander, which is linked to the compressor. Such a system can be achieved with a turbo compressor and an expander on the same shaft [37]. The cycle with the additional steam compressor(s) has to deal with more compressors and more control. The open loop R718 cycle has only half the number of heat exchangers, which reduces the need for cleaning depending on the fluid used, but the high number of steam compressors makes this cycle particularly difficult to control and the complexity makes it more difficult to optimize and operate at its best efficiency.

4. Discussions

No combination of cycles was analyzed in this study, which could increase the COP by using different refrigerants with lower temperature lifts, instead of having one refrigerant with a high temperature lift. The cycle combining the transcritical heat pump as low stage with the reversed Brayton cycle as high temperature stage was nevertheless analyzed, as it seemed like a good combination due to the temperature glide in the heat sink of the former cycle and in the heat source of the latter. Although this system shows a 28% higher COP compared to the transcritical R600 cycle and has similar costs (the system has more components but with lower unit prices), this combination is much more complex to control and offers no increased performance over the HTHP cycle in terms of efficiency.

The reversed Brayton cycle is interesting when the isentropic efficiency of the compressor is very high. When η_{is} of the compressor(s) are above 90%, the cycle is a good alternative for HTHP especially when the temperature lift is higher than 100 °C. However, the only heat pump compressor with isentropic efficiency higher than 90% found nowadays is the Rotational Heat Pump from ECOP [38]. This near market-ready machine is using an original idea: a rotational heat pump where the entire system (i.e. heat exchangers, expansion) is located inside a rotating cylinder that acts as a compressor [38].

The availability of heat pump components for temperatures higher than 120 °C is limited. Although components exist, they are far more expensive than standard low-temperature components for conventional heat pumps. This issue should dissipate in the future as soon as HTHP become more common. For the two cycles, which recompress the steam, an oil-free compressor is mandatory depending on the usage of the steam in order to avoid contamination. Oil-free steam compressors are not common in the market for low- to mid-range power (e.g. tens of kW to hundreds of kW). Steam compressors are more often used in the high-power range for MVR in order to recover the steam, which hasn't condensed [39].

The number of case studies of HTHPs in industry producing steam is very scarce [12, 15, 20, 40]. There are some articles that propose closed-cycle R718 heat pumps [23, 24, 41], and many articles on HTHPs [30, 42, 43]. As described in the literature review at the beginning of this paper, there are also some publications on experimental studies producing steam by heat pumps from Korea [3, 9, 13, 14, 15], Japan [4, 8, 16, 17, 27], China [18], Europe [19, 20, 21, 22, 23, 24], and Canada [25, 26], as well as theoretical works [2, 10, 11, 12].

Validation of this theoretical work is important to gain the trust of industries in order to build pilot projects.

In terms of technical readiness level, the five cycles analyzed can be put into three categories. The transcritical HP and HTHP are at the production level, the reversed Brayton cycle with high isentropic efficiency is according to [38] at the validation level, and the low- to mid-range HTHP with MVR and the open-loop R718 HP are considered to be at the development level.

5. Conclusions

Different heat pump cycles were simulated in order to generate saturated steam at 150 °C (4.76 bar) using only low GWP refrigerants that are future-proof according to the F-gas regulations. They show different COP trends depending on the heat source temperature and the isentropic compressor efficiency. A transcritical heat pump or a reversed Brayton cycle can be very efficient to generate heat at different temperatures thanks to the temperature glide in their heat sink. However, these cycles are not as efficient when used to generate steam at constant heat sink temperature. The heat pump cycles using steam recompression offer higher efficiency and are cheaper after a few years, taking into account maintenance and operating costs.

The authors hope that this study will promote further research into heat pump cycles for steam generation and, above all, motivate the realization of prototypes and pilot projects. The validation of such a study is crucial to rapidly increase the technical readiness level of heat pump cycle for low-pressure steam generation.

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