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Off-design of high temperature hybrid heat pump

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Abstract

Most heat pumps are designed for optimal performance at one specific operation point. It is however quite usual that the operating load and conditions change. Because of this, the heat pump should not only be considered at its optimum, but also, how and to which extend the off-design conditions affect its overall performance. A holistic approach for the most suitable choice of heat pump thus should include calculations and/or simulations of the heat pump at varying load and conditions, to evaluate its off-design performance. In this work, a hybrid high temperature heat pump, using water and ammonia, was studied. The load was varied from 10 to 130% of the designed point, giving COPs between 1.81 and 2.19, and the operating conditions, such as evaporation, condenser, absorption and desorption temperature and heat exchanger contamination, were changed separately and in combinations, giving COPs between 1.34 to 2.82. Compressor efficiency was not kept constant, to fully exploit the effects. The results are mapped. The most important factor for influence was temperature.

Keywords: compression-absorption cycle; Hybrid heat pump; Ammonia-water; Binary working fluid; Hot water production

1. Introduction

Many industries have heat demands in temperature ranges 80-120°C [1] and current R&D in heat pump technology is investigating how to reach this temperature levels. . So far only a limited number of industrial heat pumps operate at these levels, mainly due to the properties of the working fluids and available equipment. The TRL of heat pumps working above 80°C is generally low, and their potential is only commercialized to a small degree. The difference in energy prices in European countries of fossil and electric energy is surely another inhibitor that the potential of high temperature heat pumps was not exploited in times with "cheap" fossil energy. Thus, high amounts of low temperature surplus heat in industrial processes are not exploited, but wasted to the ambient [2]. Utilization of waste heat would enhance the energy efficiency [3], but the temperature is low for direct utilization. The development of high temperature heat pumps, or the so-called next generation heat pumps, can hence reduce the wasting of heat and the need for primary energy [4]. A high temperature heat pump is able to operate with a high temperature lift and deliver heat above 80°C. Boilers or electricity normally supplies heat at these temperatures. An important factor in the development of the next generation heat pump is to make sustainable solutions and environmentally friendly heat pump systems [4]. A big part of that is to use natural working fluids [4], which should not have an impact on the ozone layer or contribute to global warming.

An example of a high temperature heat pump with natural working fluids is the compression-absorption heat

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pump cycle using ammonia and water, This was first patented in 1895 by Osenbrück, and was known as the Osenbrück cycle [5]. Yet, this was still in the experimental phase in 1998 [6]. The cycle differs from a normal vapour-compression cycle by the use of a binary working fluid. Its emergence the last few decades has probably occurred due to the energy savings and environmental benefits [6] as it combines the advantages of compression cycles (high coefficient of performance - COP) with those of absorption heat pumps (high temperature lifts and gliding temperature). It also has advantages of both fluids involved: the high volumetric heat capacity of ammonia [7], the good heat transfer properties of water [7] and whereas the pressures of water are generally lower than desired (sub-atmospheric) and those of ammonia higher, those of the mixture are appropriate [2]. It can supply heat above 110°C with high performance below 25 bars at the high-pressure side [8]. Different studies show that the performance of the these hybrid heat pumps can result in a higher COP than conventional heat pumps for processes which have a corresponding temperature glide in the heat source and demand [2, 6, 9, 10]. This is due to temperature glide. The ideal compression-absorption cycles would achieve the COP of a Lorenz cycle, not a Carnot cycle as conventional heat pumps [6]. This can be used as a first evaluation tool of the most appropriate process. Itard [6] claimed to have shown that the compression-absorption cycle was better if the ratio of the temperature lift to the temperature glide in the absorber was less than two, but only for temperature glides in the absorber below 30°C. Hultén and Berntsson [9] found that the performance was similar to conventional alternatives (effective heat pumps) for temperature glides of 10°C in both absorber and absorber and 12% better at 20°C. However, higher temperatures and higher lifts changes this [9]. The performance (COP) is also significantly enhanced if longer tubes in heat exchangers and higher absorber pressures are applied [10]. This improvement was determining for whether the absorption compression system performed better than a conventional, realistic (effective) cycle [10]. They stated that *"It could be concluded that only heating cases where both the sink and the source temperature changes are high (>20 K) give superior performance for the CAHP."* An important feature of the study was that it was based on equal economic investments. A main challenge at high temperatures is the high discharge temperatures and the ability of components and oil to manage this, creating a temperature constraint for the process. Oil-free compressors exist, but this is cost-intensive technology compared to conventional cycles [10]. Jensen [11] argued that another choice of oil (synthetic rather than mineral oil) increases the achievable temperature, and found that supply temperatures up to 125°C are achievable with the compression-absorption cycle using ammonia-water if the temperature constraint is 170°C. This number was used because other researchers had used stated that the limit was 160°C [12] and 180°C [13], and 170°C was used as a compromise [11]. Supply temperatures up to 175°C were feasible in this study if the limit was instead 250°C. This limit was assumed because this is the current limit of components of for transcritical CO_2 [11]. Optimizing heat exchangers and intermediate pressure in another study gave a maximum supply temperature of 171°C with the 250°C constraint [2]. It was also found that optimum intermediate pressure should be 1.16-1.35 times the normally applied geometric mean value, and that use of a desuperheater had a low effect of the highest supply value. Oil-cooling is a possible way to reduce the problems with high discharge temperatures [10]. The relationship between the water and the ammonia, and the optimal circulation ratio (mass flow in the pump compared to mass flow in absorber) were studied by [14]. These were highly dependent on one another and the operational conditions, hence a set of graphs for several cases were made. A good fit between the temperature profiles of the heat sink/source and the working fluid will yield less irreversibility in the system [2].

The off-design behaviour of heat pumps will influence the overall performance for industrial cases with significant part-load demands. It is therefore desirable to have heat pumps that achieve high performance also at part loads, and that manage a higher capacity than the designed one. Capacity control can be performed in several ways, and is mainly related to the compressor. On-off control is one option, but at low load this means many start-ups, a phase in which the performance was found to be poor [15]. Apart from a significant impact of the start-up phase, Waddicor [15] found that this method generally showed high COPs most of the time (3-6) for a water-water system with 5-10 and 42-55°C on the low and high pressure sides. How the process was controlled was highly important for performance: Controlling supply temperature showed a strong drop in performance, whereas control of return temperature yielded a much smaller fall in COP, which was then almost constant [15]. Unloading of pistons gives a quite good result for lower capacities, in the sense that the reduction in power input and capacity are nearly identical [16], making the COP nearly constant when no other effects are considered. This is in some way an on-off control, where some of many series coupled pistons are inactivated. However, even though pistons are no longer used, the pistons keep moving, causing deviations (increase) from the ideal power consumption (identical reduction), especially at low loads. About 35% power was required at 25% capacity in [16], and the power and capacity are very close to linear [16, 17]. Another capacity control measure is to regulate the compressor speed, which has improved the COP at lower loads significantly [18, 19]. It has been verified experimentally that this technique works for a wide range of conditions, and the COP at 20 Hz was about 80-130% better than at 90 Hz, depending on the source and sink temperatures [19]. Variable speed made the COP increase at lower loads between 50 and 100% instead of dropping, and optimal flowrates kept the performance high at all capacities, particularly below 50% load [18]. At 100% capacity, the difference was minor, but at 10%, the COP was about 160% better with optimal flow rates. For binary fluids, like water-ammonia mixtures, another capacity control is change in concentration, as this changes the properties of the mixture [6].

To the best of the authors knowledge, no study of off-design performance for compression-absorption heat pumps, are performed. The objective of this article is to investigate the off-design performance of the compression-absorption heat pump. In industrial processes where the operating load and/or conditions change it is vital that the performance of the heat pump does not decline substantially in off-design operation. Different off-design scenarios were considered, including change in either heat load or temperature level.

Nomenclature

β	Pattern angle	[radians]	$\dot{m}_{comp}$	Mass flow in the compressor	[kg/s]
D_h	Hydraulic diameter	[m]	$\dot{m}_{pump}$	Mass flow in the pump	[kg/s]
η_v	Clearance volume efficiency	[-]	n	Frequency	[Hz]
η_{is}	Isentropic efficiency	[-]	Π	Pressure ratio	[-]
η_d	Loss factor	[-]	p_m	Intermediate pressure	[bar]
φ_0	Clearance volume ratio	[-]	p_{abs}	Absorber pressure	[bar]
f	Circulation ratio	[-]	p_{des}	Desorber pressure	[bar]
κ	Ratio of specific heats	[-]	Pr	Prandtl number	[-]
k	Thermal conductivity	[W/(mK)]	ρ	Density	[kg/m^3]
h_l	Heat transfer coefficient of working fluid	[W/(m^{2}K)]	Re	Reynolds number	[-]
			v_{comp}	Specific volume	[m^3/kg]
h_w	Heat transfer coefficient of water	[W/(m^{2}K)]	x	Ammonia concentration	[-]
ʎ	Volumetric efficiency	[-]	x_{abs}	Mass fraction of water entering the absorber	[-]
μ	Dynamic viscosity	[Ns/m^2]			
$\dot{m}_{abs}$	Mass flow in the absorber	[kg/s]	x_{pump}	Mass fraction of water entering the pump	[-]

2. Method

A compression-absorption heat pump was designed to provide a heat load of 1.1 MW for heating process water from 95°C to 115°C. Surplus heat of 30°C was used as heat source. At design point it exits at 21°C, yielding a temperature lift of 94°C for the heat pump between the lowest and highest temperature of the heat source and sink. The logarithmic mean temperature difference for the process was 79.4°C. For standard vapour-compression heat pumps, these conditions would be challenging with respect to e.g. high pressure ratio and superheating, which would lead to low performance of the system. By using a mixture of ammonia and water as working fluid, the boiling point will be reduced, which leads to lower pressure level compared to pure ammonia.

2.1. The Principle of a Compression-Absorption Heat Pump

A principle sketch of the cycle is show in Figure 1. A lean ammonia-water solution (point 8) and ammonia vapour (point 2) enters the absorber. Through the absorber the vapour is condensed and absorbed in the ammonia-water solution while heat is rejected to the heat sink. The absorber corresponds to the condenser in a vapour-compression cycle. The ammonia concentration of the solution will increase in the absorber and the mixture exits as a rich ammonia-water solution (point 3). Next, the rich solution is cooled in an internal heat exchanger and then throttled to low pressure. In the desorber, ammonia vapour is formed by heat supply from a heat source. Thus, the concentration of ammonia in the solution will decrease. The two-phase mixture exiting the desorber consists of saturated vapour and liquid in thermal equilibrium. The saturated vapour, consisting of nearly pure ammonia, is compressed to the high pressure (point 1-2). The liquid, which is a lean ammonia-water solution, is first pumped to the high pressure (point 6-7) and then heated in the internal heat exchanger (point 7-8). The internal heat exchanger will enhance the cycle performance as it provides internal heat recovery [20].

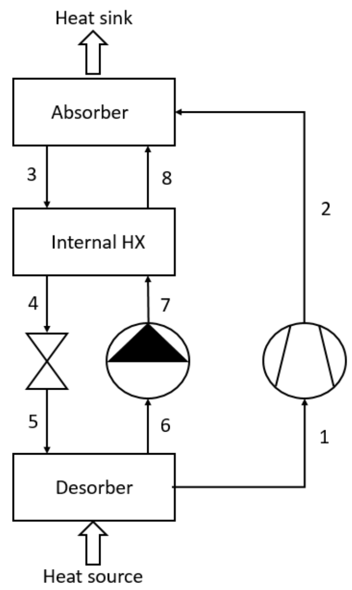

Figure 1: A principle sketch of the compression absorption heat pump, based on the sketch in [2]

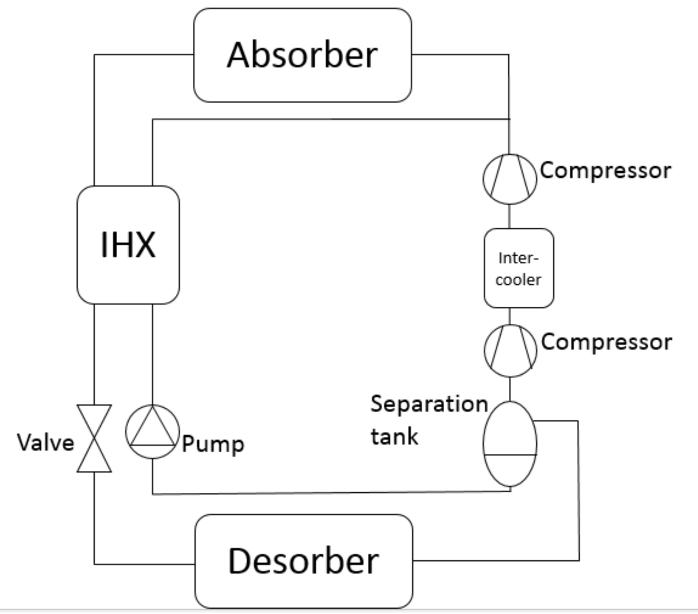

Figure 2: Model applied: Three PI-controllers are steering the process towards the desired operation point, giving inputs to two compressors (right) and an expansion valve (left). A pipe connected to a thermal boundary of constant temperature (red) functions as an intercooler between the compressors. Vapour is sucked from a separation tank of 90 l, whereas the liquid phase in the bottom of this is pumped to the high pressures side by the pump in the middle. The mass flow in the pump is controlled to give a fraction of ammonia in the absorber and desorber of 0.6. Two heat exchangers (top and bottom) exchange heat with water (blue lines). The amount of hot water to be heated is varied by a ramp function (upper left corner). The internal heat exchanger is seen above the valve to the left.

A simulation model of a compression-absorption heat pump has been constructed in Dymola (Dynamic Modelling Library, version 2015, Dassault systems) This is a modelling and simulation environment that uses the language Modelica (3.2.1). Components and functions (objects) from the TIL library (TIL 3.4, TLK-Thermo GmbH, Braunschweig, Germany) were used. Library design and the principles for the objects based on equations can be found in [21]. How the components can be combined into a heat pump is demonstrated by [22].

Figure 2 shows the model made in Dymola, where two-stage compression with intercooling was used. This was due to the high pressure ratio, but also reduces the overall compression power, see section 2.4. Regulation of the system is explained in section 2.6. The model was simulated using the algorithm "Dassl" and 500 intervals.

2.2. Concentration

To achieve high enough temperature in the desorber at an acceptable pressure the concentration of the ammonia in the ammonia-water mixture entering the desorber must be set at an appropriate value. Figure 3 was used to estimate an optimal concentration of the ammonia. This figure shows the relation between the pressure and saturation temperature at different concentration of ammonia. The line to the far left is pure ammonia, while the line to the far right is pure water. A mass fraction of 0.6 of ammonia was chosen and used as this seemed appropriate from the diagram, but this is a degree of freedom to be optimized for enhancing the COP. Optimal concentration will also depend on operational conditions [11, 14].

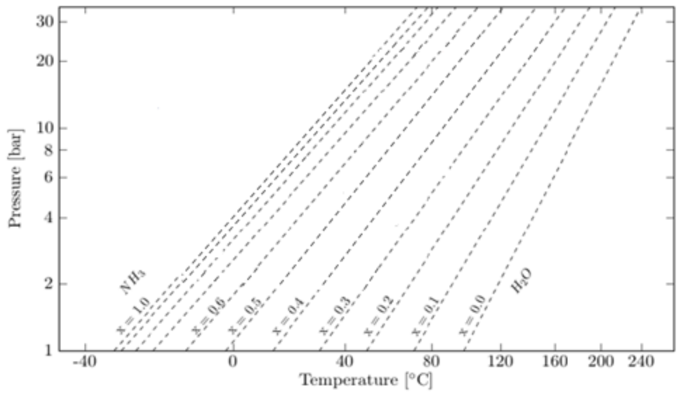

Figure 3: Ammonia-water mixture depicted in a log P-(1/T) diagram with x, the saturated/solvable concentration of ammonia at the given conditions. The diagram is taken from [2].

2.3. Plate heat exchangers model for absorber and desorber

The desorber, absorber and internal heat exchanger have been designed as plate heat exchangers, which are normally applied according to suppliers of the technology. Both the geometry and the heat transfer model are inputs to the component in Dymola.

The heat transfer model used assumes constant heat transfer coefficients. No correlation for desorbing/absorbing fluids was found. The formula used for the ammonia-water side described in equation (1), which is a formula for phase-changing fluids, which should be a useful approximation. For the heat source and sink sides, the correlation in equation (2) has been used, using the Reynolds and Prandtl numbers. The Reynolds number for a plate heat exchanger can be found from equation (3), [23].

$$h_l = 0.0077\left(\frac{k_l \rho_l (\rho_l - \rho_g) g}{\mu_l^2}\right)^{\frac{1}{3}} \mathrm{Re}^{0.4} \tag{1}$$

$$h_w = 0.259\left(\frac{k_w}{D_h n}\right) \mathrm{Re}^{0.64} \mathrm{Pr}^{0.32} \left(\frac{\Pi}{2} - \beta\right)^{0.09} \tag{2}$$

$$\mathrm{Re} = \frac{4\dot{m}_w}{\mu_l D_h} \tag{3}$$

The hydraulic diameter was obtained from Dymola. The rest of the thermodynamic properties for the ammonia-water mixture have been calculated from the relation of [24]. The geometry parameters of the heat exchangers are given in Table 1. The number of plates for each heat exchanger has been chosen to achieve the needed heat load. The rest of the parameters have been chosen so that the heat exchanger shall have a realistic size and shape.

Table 1. Parameters of plate heat exchangers (HX)

	Desorber	Absorber	Internal HX
Number of plates	30	50	20
Plate length [m]	0.8	0.8	0.8
Plate width [m]	0.5	0.5	0.5
Pattern angle	40	40	40
Wall thickness [m]	0.001	0.001	0.001
Pattern amplitude [m]	0.005	0.005	0.005
Pattern wave length [m]	0.005	0.005	0.005

2.4. Compressor

The built-in component "EffCompressor" in the TIL library in Dymola was used to model the piston compressor. This component has constant volumetric and isentropic efficiency. To make the model more realistic, the "EffCompressor" was modified to have volumetric and isentropic efficiency as defined in equations (4) through (6). Equations (4) and (6) are taken from [16] and equation (5) from [23]. These equations apply to piston compressors, which are normally applied in real compression-absorption installations with water-ammonia, according to the manufacturer. Screw compressors have higher deviations from ideal performance [16], whereas piston compressors are closer to the ideal [16], and their efficiency mainly depends on the pressure ratio like in equation (5). The compressor speeds were controlled to give the desired water temperature from the heat sink and the desired intermediate pressure. The clearance volume efficiency is defined as equation (6), [23]. In the simulation model, the clearance volume ratio was 4% and the loss factor is set to 1.

$$\lambda = \eta_v \eta_d \tag{4}$$

$$\eta_{is} = 0.3862 + 0.0016\,\Pi^3 - 0.0333\,\Pi^2 + 0.1892\,\Pi \tag{5}$$

$$\eta_v = 1 - \varphi_0 \left(\Pi^{\frac{1}{\kappa}} - 1 \right) \tag{6}$$

A two-stage compression was used in the simulation model to enhance the efficiencies of the compressors, which is optimal at a pressure ratio of 4, but quite poor above 8. More stage compression reduces the energy consumption if there is intercooling between, and will thus result in a higher COP for the system. It will also reduce the discharge temperatures after compression, and hence, this system was chosen. The model of the intercooler was simplified to a tube connected to a heat boundary with a temperature of 30°C and the heat transfer in the tube was set to a very high value, 10^{10} W/K, to ensure sufficient intercooling at all times and thereby stabilize the model.

The stroke volumes of the compressors are calculated by equation (7), [23]. Assumptions about temperature, pressure, compressor frequency and mass flow have been made to estimate the stroke volume. This was 0.022 m^3 for the first compressor and 0.00185 m^3 for the second compressor at their respective design points.

$$V_S = \frac{\dot{m}_{comp} v_{comp}}{\lambda n} \tag{7}$$

2.5. Pump

The built-in component "simple pump" in the TIL library was used to model the solution pump. The pump efficiency was held constant at 40%, which is the default setting in Dymola. The input parameter of the solution pump was the mass flow, calculated by equation (8).

$$\dot{m}_{pump} = \frac{\dot{m}_{comp} x_{abs}}{x_{pump} - x_{abs}} \tag{8}$$

2.6. Regulation

To achieve the wanted temperature lift of the water at varying conditions, three PI-controllers were used to regulate the system. One controller adjusted the frequency of the second compressor so that the temperature of the water exist at the design temperature of 115°C. Another controller adjusted the frequency of the first compressor so that the intermediate pressure, p_m, followed equation (9). The last controller adjusted the opening of the expansion valve so that the pressure in the desorber was kept at 2 bars.

$$p_m = \sqrt{p_{high} p_{low}} \tag{9}$$

2.7. Off-design conditions considered

To investigate the off-design performance of the heat pump, several cases have been simulated. Firstly, the heat load to the heat sink has been changed from 10% to 130% of design load. The temperature levels of the heat sink and source were then held constant.

Secondly, the performance of the heat pump has been tested for seven changes of the temperature levels of the heat sink and source while the design mass flow was held constant. In cases 1 and 2, the heat sink inlet and outlet temperature have been respectively reduced and increased with 10 K. For cases 3 and 4, the inlet temperature of the heat source have respectively been reduced and increased with 5 K. Furthermore in case 5, the heat sink inlet and outlet temperature have been reduced with 10 K at the same time the heat source inlet temperature have been increased with 5 K, thus reducing the temperature lift to 81°C. For case 6, the temperature levels have been moved the opposite way compared with case 5, yielding a temperature lift of 105°C. The results were compared to the Carnot and Lorenz efficiencies as described in [14], which represent the COPs of ideal compression and ideal absorption heat pump cycles.

Table 2. Temperature cases considered, regarding the temperatures of the sink and the source of the heat pump: the outlet sink temperature and the resulting lift are results of the other three and the system efficiency, but are shown here for the sake of order.

Case no.	Inlet sink temperature [°C]	Outlet source temperature[°C]	Outlet sink temperature [°C]	Inlet source temperature [°C]	Temperature lift [°C]
1	85	20	105	30	85
2	105	22	125	30	103
3	95	18	115	25	97
4	95	24	115	35	91
5	85	24	105	35	81
6	105	20	125	25	105

3. Results

Discharge temperatures, COPs including heat in the intercooler due to the high discharge temperatures after compression, compressor frequencies and absorber pressures and mass flows are reported for all simulations.

3.1. Variation of heat load

The mass flow of the heat sink has been varied from 10% to 130% of design point, and the results are tabulated below. The discharge temperatures 1 and 2 are respectively the temperatures out of the first, low-pressure compressor and the second, high-pressure compressor. The sink temperature was always 95°C in and 115°C out, and the inlet source temperature was always 30°C. The source' mass flow rate was constant, and its outlet temperature was 21°C at design point (100%), 29°C at 10% load and 19°C at 130% load.

Table 3. Simulation results for 10% to 130% heat load, showing pressure and mass flow in the absorber, COP and the discharge temperatures and frequencies of the low pressure compressor (denoted by 1) and high pressure compressor (denoted by 2)

Capacity [%]	Absorber pressure [bar]	Mass flow [kg/s]	COP [-]	Discharge temp. 1 [°C]	Discharge temp. 2 [°C]	Frequency 1 [Hz]	Frequency 2 [Hz]
10	26.53	0.35	2.19	169.06	143.50	4.32	2.94
20	26.58	0.69	2.18	169.10	146.10	8.52	5.83
30	26.56	1.04	2.18	169.10	148.60	12.73	8.77
40	26.57	1.36	2.17	169.10	150.90	16.85	11.68
50	26.54	1.70	2.16	169.00	153.20	21.04	14.67
60	26.52	2.04	2.15	168.90	155.50	25.24	17.72
70	26.47	2.40	2.13	168.60	157.80	29.57	20.91
80	26.42	2.77	2.10	168.30	160.10	33.98	24.21
90	26.29	3.19	2.06	167.70	162.60	39.01	28.07
100	26.06	3.73	1.98	166.70	165.60	45.20	32.96
110	26.02	4.21	1.94	166.10	168.40	50.49	37.13
120	25.99	4.30	1.88	165.40	171.60	56.22	41.68
130	25.93	5.52	1.81	164.60	175.00	62.72	46.89

3.2. Variation of temperature level

The results of changing the sink and source temperatures are given in Table 4. The temperatures of the sink and source were presented in Table 2. In case 3 and 6 the heat source temperature was lowered, and to retrieve enough heat and keep the temperature difference of the inlet of the desorber to 5 K the desorber pressure was reduced to 1.8 bar.

Table 4. Simulation results for various temperature levels

Case no.	Absorber pressure [bar]	Mass flow [kg/s]	Discharge temp. 1 [°C]	Discharge temp. 2 [°C]	Frequency 1 [Hz]	Frequency 2 [Hz]	COP [-]	Lorenz efficiency [-]	Carnot efficiency [-]
Design	26.06	3.73	166.70	165.60	45.20	32.96	1.98	4.76	4.13
Case 1	21.84	3.17	157.60	152.90	38.26	30.34	2.39	5.26	4.45
Case 2	30.93	4.39	176.20	179.50	53.4	35.91	1.67	4.37	3.87
Case 3	25.55	4.60	164.8	179.30	58.85	42.40	1.61	4.53	3.99
Case 4	23.77	2.55	177.50	154.40	34.62	23.50	2.52	5.02	4.28
Case 5	22.75	2.38	167.10	143.40	32.06	24.07	2.82	5.63	4.67
Case 6	29.75	5.82	173.20	194.60	75.00	50.02	1.34	4.19	3.78

4. Discussion

In all the results, it is a challenge that the absorber pressure became a few bars higher than the designed one. Commercial equipment for ammonia is available for up to 40 bars [25], so operation with these pressures is possible, but this will be more expensive and energy demanding than operation at 25 bars. Thus, it is desirable to reduce the absorber pressures. This could be accomplished either by a larger absorber or a change in concentration. Achieving more than 110°C with pressures less than 25 bars should be possible [8] and 140°C was achieved at 20 bars in other studies [11]. Some studies claim that high-pressure equipment is required to reach the higher temperatures with an also high COP [2, 10]. The highest supply temperature of 171°C required 47.5 bars [2]. The difference in this study might be the higher temperature lift than in [9, 11]. Cooling of the compressors, three-stage compression or a change in the intermediate pressure are possible solutions.

4.1. Variation of heat load

When the heat load is changed, referring to Table 3, the COP generally increases for lower heat load, but decreases for higher heat load. The increase at low load slowly stagnates around 60%. At design load the COP is 1.98, for the lowest load, the highest COP of 2.19 is reached, and for 130% load, it is reduced to 1.81. This trend should be expected for two reasons. One is the more favourable temperature conditions due to the higher outlet temperature of the source, as mentioned in section 3.1. The Carnot efficiencies at 10, 100 and 130% load are respectively 4.51, 4.13 and 4.06. The Lorenz efficiencies are, in the same order, 5.01, 4.76 and 4.71. Thus, the potential of the absorption-compression cycle is higher than that of the compression cycle, but the obtained COPs are far from the ideal ones. At 10 and 100% load, they are about 43% of the ideal, at 130% load this has dropped to 38%. This might be related to the second reason to expect higher COP at lower load, namely that the heat exchangers are over-dimensioned at lower heat loads, and the absorber pressure should therefore be reduced with lower loads, reducing the COP. However, as can be observed in Table 3, the pressure in the absorber increases. This increase is a result of the regulation of the desorber pressure. In the beginning of the simulation, the valve starts to close, reducing the mass flow through the valve, but also increasing the absorber pressure. As the process water at this point already has reached its design temperature, and the increased pressure would increase the temperature, the compressor frequency will be reduced. The absorber pressure should then be reduced, and these two opposite effects bring the process to a stable balance point with a lower mass flow and a slight change in absorber pressure, in this case a very small increase in absorber pressure, 0.65 bar. Since the desorber pressure is held constant, the absorber pressure is also nearly constant once the balance is reached.

Another way to explain the increase in COP at lower loads is the fact that the temperatures before the valve and the desorber are lower leading to more heat per mass retrieved in the desorber. The over-dimensioned heat exchangers gives higher heat transfer per mass, until a limit when the temperature differences are so small at the exit that the system stagnates. At 20% load for example, there is 245 kJ/kg heat transferred compared to 203 kJ/kg heat transferred for design load. The mass flow is then reduced more than proportionally to the load. At 20% heat load, the mass flow is 18% of design mass flow. This makes the work input, which is proportional to the mass flow, less than proportional to the heat load, because the mass flow is less than proportional.

Isentropic efficiency of piston compressors is little influenced by the mass flow, and this small influence was therefore not accounted for in this work. At the lowest capacities, this might be unrealistic, thus the COP would increase somewhat less in the lowest range of capacities, and future work should also include this effect. However, there are compressors with about 15 % higher efficiency. As no reference to this was found, it was not applied here, but could potentially improve performance quite significantly. For very high pressure lifts, piston compressors might not be an option, and screw compressors would have to be applied. These show even more degradation in efficiency at off-design loads [16].

An important question for the cycle is whether the compressors have acceptable working conditions. Discharge temperatures of the compressors is one such important topic. Most of the discharge temperatures in Table 3 are in the range of 160-170°C. This is in general very high, and is a typical problem with ammonia compressors. Some are even above 170°C. The discharge temperature of the first compressor and at higher loads are the worst in this sense. Again, larger heat exchangers, compressor cooling or more stages of compression could be necessary to achieve low enough temperatures, unless components that can withstand these temperatures are available as assumed in other studies [2, 11]. However, this might be more costly, and costly solutions should be avoided to be a fair alternative to existing cycles [9]. Costs are a main challenge for implementation of this technology

For the first compressor the discharge temperature increases for lower loads, which can be explained by the small increase in intermediate pressure. The opposite happens for the second compressor, where the discharge temperature decreases for lower loads. The intercooler before the second compressor is able to cool the ammonia more at lower load, thus the ammonia will have a lower temperature in to the compressor and the discharge temperature will therefore be lower. Another solution to the problem could therefore be to reduce the intermediate pressure, thus reducing the discharge temperature of the first compressor and increasing the intercooling, but most studies indicate that better COPs are achieved with higher, not lower intermediate pressures [2]. If the desorber pressure was allowed to float, this would allow the COP to increase at lower loads due to lower pressure ratios instead of lower mass flow. This would also potentially reduce the discharge temperatures because the pressure lifts would also be lower, and the temperature differences and enthalpies before the compressors lower. Attempts with this control strategy (steering superheat before compression) gave unstable simulations that could not run, as the pressure, heat transfer, mass flow and concentration all alter it and simulations cannot start properly. Managing such a control measure is still highly desirable. For higher loads than the design, the situation cannot be improved for neither temperatures nor pressures by floating lower pressures, thus, one of the other means to reduce this would have to be used.

4.2. Variation of temperature level

The simulation results for variation of temperature level, referring to Table 4, are as expected. When the temperature level of the heat sink is lowered, the absorber pressure is reduced and the COP is then increased. The COP increased to a value of 2.39 when the heat sink temperature level was reduced with 10 K, in comparison, the COP at the design temperature was 1.98. The opposite occurred when the heat sink temperature level was increased with 10 K, the COP is then reduced to the value 1.67. This shows a quite strong temperature dependency. The results for 10 K increase results in a very high absorber pressure and discharge temperatures above 170°C which has used as an upper limit in other studies [11]. A change in pressures and/or concentration could perhaps improve this, as the intermediate pressure should increase more than the geometric mean of the high and low pressures [2], but the correlation for this is not yet implemented. This also means that the process needs either very stable operational conditions or advanced steering equipment to optimize the concentration and pressures with the temperatures.

A reduction of heat source inlet temperature with 5 K yielded a reduction in COP to the value of 1.61. In this case the desorber pressure was reduced to 1.8 bar to ensure that heat could be transferred in the desorber. This also demonstrates the need for floating pressure in the desorber if the process shall handle varying conditions. The absorber pressure was as expected not changed much. An increase of heat source temperature of 5 K yielded an increase in COP to the value of 2.52. The absorber pressure however, did also increase. This might be because a higher temperature of the mixture out of the desorber means a lower density in the vapour transferred to the absorber, which leads to a higher absorber pressure.

In cases 5 and 6 the temperature lift is respectively reduced and increased with 15 K. As expected the absorber pressure and mass flow decreases, yielding an increase in COP for the reduced temperature lift. The opposite occurs for an increase in temperature lift. For comparison, the COP for the increase in temperature lift is 1.34 and for the reduction in temperature lift, the COP is 2.82. The results for supplying heat at 105°C (case 5) are quite good and within acceptable conditions for both temperatures and pressures. Case 6 on the other hand, has conditions that are far from acceptable. The COP is far too low to compensate for the costs of a heat pump compared to alternative heat sources. To be interesting for the industry, the COPs should at least approach 2.00. Regarding temperatures, much intercooling was necessary if 170°C is a limit, and the pressure is also much higher than the design pressure. Comparing case 2 and 6, both supplying heat at 125°C, but case 6 also retrieving heat from 25°C source, the highest discharge temperature, after the first compressor is clearly mainly related to the low side, and three-stage compression would probably have been more appropriate. It was also crucial that the heat in the intercooler could be utilized; otherwise, the COP would be about 0.5 lower. This was the case for all scenarios. Bad performance for the conditions in case 6 was expected, and would not be acceptable for industrial use. The process would require redesign to be used for such temperature conditions.

It seems very likely that one could make the system run with a COP of about 2.00 or better at design load. The obtained COPs were around 40% of the achievable, while 50-60% is usual, suggesting room for improvement. Small changes in temperature improved the system very much, and 50% of the Lorenz efficiency was reached for case 4 and 5, whereas it was 32% for case 6. As the Lorenz efficiency represents only the effect of the temperatures, this means that there was more than the temperature conditions that made the COP much lower for case 6 than the other cases. Design and regulation also matters. Changing the ammonia concentration x, for example, is a way to regulate the capacity, as an alternative to increase or decrease the pressure levels [14], and was not exploited here. Jensen [14] studied the influence of circulation ratio, defined in equation (10), and x, finding that their values changed the COP. Comparisons showed that f and x at design point in this study were already very close to the optimum values in the graphs by [14], and no room for improvement was found here. In this model, the concentration determined the circulation ratio. Suggested further work on the model would therefore be to also make the circulation ratio, defined in equation (10), an independent input to the system.

$$f = \frac{\dot{m}_{pump}}{\dot{m}_{abs}} \tag{10}$$

With the circulation ratio also as an input the system can be more easily controlled. This gives the opportunity to find the optimal values of concentration and circulation ratio to maximize the COP as has been done by [26]. The optimal values should not only be found for the design conditions, but also for off-design conditions. In this way, the concentration and circulation ratio can be optimized for the operating conditions, design or off-design, and the COP can always stay at its maximal obtainable value. Higher efficiency compressors, optimization of intermediate pressure and more compression stages are other options.

5. Conclusions

The performance of off-design operation of a compression-absorption heat pump using ammonia-water as working fluid has been investigated. The heat pump was designed for an industrial process where process water was heated from 95°C to 115°C. The heat source was surplus heat at 30°C. Changes in demand mainly altered the mass flow in the heat pump as the pressure was controlled. The COP was only weakly influenced by changes in load and improved with lower heat loads. It was 1.98 at the design point, at its highest value, 2.19, at 10% load and declined to 1.81 at 130% load. Changes in temperature lift had a much stronger influence on performance, and otherwise the heat pump behaved as expected: for reduced lift the COP increased and for increased lift the COP decreased. The worst case scenario was an increase in temperature lift to 105°C, which yielded a reduction in COP to 1.34, and the COP, temperature and pressure conditions became unacceptable. The highest COP achieved was 2.82 for the a reduced temperature lift to 81°C. Thus, very high COPs can be obtained, but system improvement is necessary for some of the off-design conditions. Using the intercooler heat was crucial, and enhanced the COPs from about 1.5 to about 2.0. The maximum compressor efficiency was 0.71 in the performed investigation. With recent compressor development it should be possible to achieve higher efficiencies, which will increase the COP accordingly.

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