OFF-DESIGN OF HIGH TEMPERATURE COMPRESSION-ABSORPTION HEAT PUMP

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ABSTRACT

Most heat pumps are designed for optimal performance at one specific operation point. In real industrial applications though, the operating load and conditions change, thus, 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. In this work, a compression-absorption high temperature heat pump using water and ammonia was studied. The load was varied from 17% to 159% of the design point based on a measured demand in the industry, yielding average COPs of 2.02-2.08 for the different cases considered, with minimum and maximum values of 1.72 and 2.26. The absorption and desorption temperatures were also changed, showing strong impact on performance. The COP then varied between 1.67 and 2.35 Compressor efficiency was not kept constant, to fully exploit the effects. Several suggestions for improvement were identified.

1. INTRODUCTION

The potential for heat pump technology to supply heat between 80 and 150°C in industrial context is estimated to be 3.126 PJ in Europe below 150°C in 2012 [1]. However, no commercial heat pump technology today is capable of reaching these levels, as this requires oil with low viscosity and high boiling point [2, 3], components and working fluids that are applicable in a much higher temperature range than manufacturers usually offer. It is also a challenge that in several European countries, fossil fuels are up to four times cheaper than electricity, demanding a COP higher than this ratio around four for a heat pump to be the more economic choice. Most industries have several heating and cooling needs at different temperature levels, and would enhance their energy efficiency save upon utilizing surplus heat [4] through for example heat integration. However, this is a difficult and sophisticated task, and is not commonly applied. Instead, large amounts of waste heat are rejected to the ambient rather than utilized [5]. The reason is often that the heat has a too low temperature for direct utilization. New concepts are emerging in this field though. The next generation of heat pumps will enable higher degree of surplus heat utilization and reduce the total need for energy [6]. Developing high temperature heat pumps for the future and making them sustainable and environmentally friendly through the use of natural refrigerants is important [6]. Natural refrigerants, unlike the synthetic ones, have low or no impact on global warming and the ozone layer, and synthetic fluids also have no natural mechanisms for decay in the nature.

Among the emerging high temperature heat pump technologies is the hybrid or compression-absorption heat pump. It was invented and patented as the Osenbrück cycle in 1895 [7], but its application has only awoken interest quite recently, as it in 1998, still had not advanced from experimental level [8]. The reason is probably that some of the key benefits are energy savings and being environmentally friendly [8], which has not been a matter of high importance until the last decade. Unlike other heat pumps, this heat pump applies two working fluids, a binary mixture, enabling heat transfer at gliding temperatures. Because of this feature, the cycle is able to perform better than compression cycles when the heat source and sink also have gliding temperatures [2, 3, 5, 8]. It can also perform better even if equal investment costs are required [2]. However, this varies with operational conditions, and for small temperature lifts (10 K or less) the performance was not found to be better than for compression cycles [3]. For a larger glide of 20 K, an improvement of 12 % was obtained [3]. Another study showed that if

$$\frac{\Delta T_{lift}}{\Delta T_{glide,sink}} > 2 \text{ and } \Delta T_{glide,sink} > 30\ K, \quad (1)$$

the hybrid cycle would outperform the compression cycle [8]. This is however not valid at high temperature levels and for high lifts [3]. Due to the gliding temperatures, an ideal compression-absorption cycle is not the Carnot cycle, but rather the Lorenz cycle [8]. The difference between these cycles gives an indication of whether conventional or hybrid heat pumps will be better for a given application.

The two refrigerants in hybrid heat pumps can be the natural refrigerants ammonia and water, which are used in some first installations which operate above 110°C, yet below 25 bars, and achieves high, competitive COPs [9]. Compression-absorption heat pumps with ammonia/water have benefits from both the compression and the absorption cycles. Examples are high performance, gliding temperature when exchanging heat and high temperature lifts (at least up to about 80°C) [9]. It also has good heat transfer properties like water, appropriate pressure levels (between 1.0 and 25 bar in the relevant temperature ranges, e.g. from below 60°C to above 100°C, those of water would be below 1.0 bar and those of ammonia above 62.6 bar) [5], and the volumetric heat capacity is high like for ammonia [10]. However, the high discharge temperatures often associated with ammonia is a challenge for the components and the oil, which today are not meant to be operated at such high temperatures. Oil free compressors are expensive and probably will prevent the cycle from being competitive [2], but oil cooling is a possible measure to reduce the discharge temperatures from the compressors [2]. The highest temperature today's equipment can be used at is about 160°C [11] to 180°C [12] according to Jensen [13], who used 170°C as a limit and managed to reach sink temperatures of 125°C.

Varying operational conditions appear in industrial processes, and makes the off-design performance of industrial heat pumps important for overall performance. To control capacity, one can apply techniques like

- unloading compressor pistons: This lowers compressor efficiency slightly at lower loads (35% power input at 25% thermal capacity in [14]). Power and load are as good as linear [14, 15].
- on-off operation: This is also likely to reduce performance, as the start-up phase has low performance [16]. On-off control has given COPs in the range 3-6 most of the time when heating and cooling water and having temperature levels of 42-55°C and 5-10°C [16]. Proper regulation of the process was important for its performance. The COP depended on temperature to a high extent and decreased at lower load.
- speed/frequency control: In some studies [17, 18], this measure enhanced the COP at reduced capacities. Experiments have shown its applicability at various conditions, and also showed higher COP at lower loads (around twice as high at 20 Hz compared to 90 Hz, depending on temperature) [18]. Another study, [17], showed that optimal control of the flowrates was important for the speed variable compressor to achieve higher COPs at lower loads. Above 50% load, this measure improved performance, but the COP also increased without it. Below 50%, optimal flowrates were crucial to keep performance high. For a 10% capacity, the improvement of the COP was in the range 140-175% compared to cases without optimized flowrates.
- altering the concentration and thereby the mixture properties (only for fluid mixtures) [8].

The third option is the most promising of the three first with respect to keeping the COP high at off-design conditions, there is too little data on the fourth to compare.

In a previous study [19], a model of a hybrid heat pump was established in Dymola. Its performance was evaluated at different loads, ranging from 10-130% of the design point, 1.1 MW. At the design point, the COP was 1.98, and improved to 2.19 at lower loads and decreased to 1.81 at the highest load. The heat sink was surplus heat at 30°C, and the sink was process water to be heated from 95°C to 115°C. The performance at varied temperature levels was also considered. However, all these simulations were steady state simulations, not taking into account the transient effects of varying loads. This study investigates the heat pump model at varying conditions, using fluctuating and measured conditions.

2. METHOD

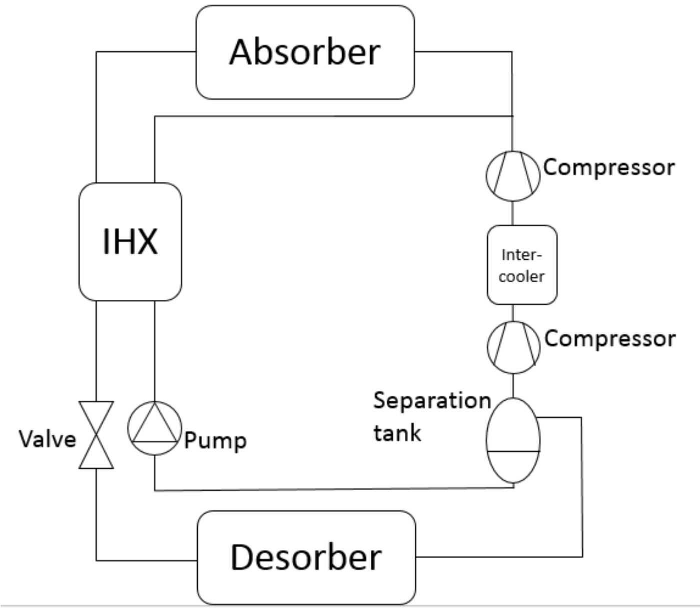

Figure 1. The modelled hybrid heat pump included an internal heat exchanger (IHX) for heat recovery; a separation tank removing gaseous ammonia from the liquid mixture; two ammonia piston compressors with an intercooler in between and a pump for bringing the liquid mixture to the high-pressure side. The figure is taken from [19].

A hybrid compression absorption heat pump was modelled and simulated in dynamic modelling programme Dymola (Dynamic Modelling Library, version 2015, Dassault systems), using the Modelica (3.2.1) as programming language. It also used objects from the TIL library (TIL 3.4, TLK-Thermo GmbH, Braunschweig, Germany) as components in the system, as these are readily built-in and adapted to fit real components. How it is designed and built up is described by Richter [20]. The very same model was used in [19] at steady state conditions, and full details can be found there.

2.1. Regulation

In order to deliver process water at the desired temperature of 115°C, regardless of load and inlet temperature, the compressor speed and expansion valve opening had to be adjusted. Built-in PI-regulators in Dymola were used to steer the process. The valve kept the desorber pressure at 2.0 bars by adjusting the flow from the high-pressure side. The high-stage compressor adjusted its speed to achieve the desired process water outlet temperature. The low-stage compressor had to follow the adjustments of the first, but not proportionally, as the inlet densities to the two compressors differed. Thus, this compressor was controlled to achieve an appropriate pressure lift for both compressors. Equation (9) defines the desired intermediate pressure, which gives equal pressure lifts for both compressors. The liquid pump adjusted its mass flow to achieve an ammonia concentration of 0.40 in the absorber, which could be calculated exactly from mass balance equations for ammonia and water entering the compressor and pump. A mass fraction of 0.40 of ammonia was chosen and used as this seemed appropriate considering temperature glides and pressure conditions, but this is a degree of freedom which could be optimized for enhancing the COP. Optimal concentration will also depend on operational conditions [13, 21].

$$p_m = \sqrt{p_{high}\, p_{low}} \qquad (2)$$

2.2. Simulated transient cases

The modelled cases was based on an existing real-life industrial process, for which the heat demand is not at static conditions. Both the demand and the inlet temperature of the process water to the heat pump changes. The effect of changing these values were studied in [19], using the same model as in this study. However, the effect of dynamic, simultaneous change in both demand and temperature was not considered. The present study considers both the effects and the performance of the heat pump when the inlet temperature of the heat sink fluctuated between 85.0°C and 105.0°C while the temperature of the heat source fluctuated between 27.5°C and 32.5°C. The fluctuation was simulated as sine waves with details as in Table 1. The sine wave of the heat source started 10 seconds later than that of the heat sink to not let the waves harmonize, and the different frequencies also ensured that they did not change in parallel.

Table 1. Parameters for sine wave used to obtain fluctuating sink and source temperatures

	Heat sink	Heat source
Amplitude [K]	10	2.5
Frequency [Hz]	0.01	0.05
Offset [K]	368.15	303.15

Two additional cases involved simulating the dynamic, typical heat demand for an entire day, as shown in Figure 1. The heat loads in Figure 1 are measured demands from an industrial production process, provided by an industry partner. Case 1 is a day with a high total heat demand and Case 2 is a day with a lower total heat demand. The data was measured every hour at the industrial plant, and a 2nd degree polynomial interpolation method was used to make a smooth curve of the demand, which would be more realistic compared to discrete hourly values with abrupt changes. The simulation ran a 24 hours scenario where the mass flow of the process water changed according to the smooth curve. In these two simulations, all temperatures for sink and source were held constant at design conditions, as no data on temperature changes were available.

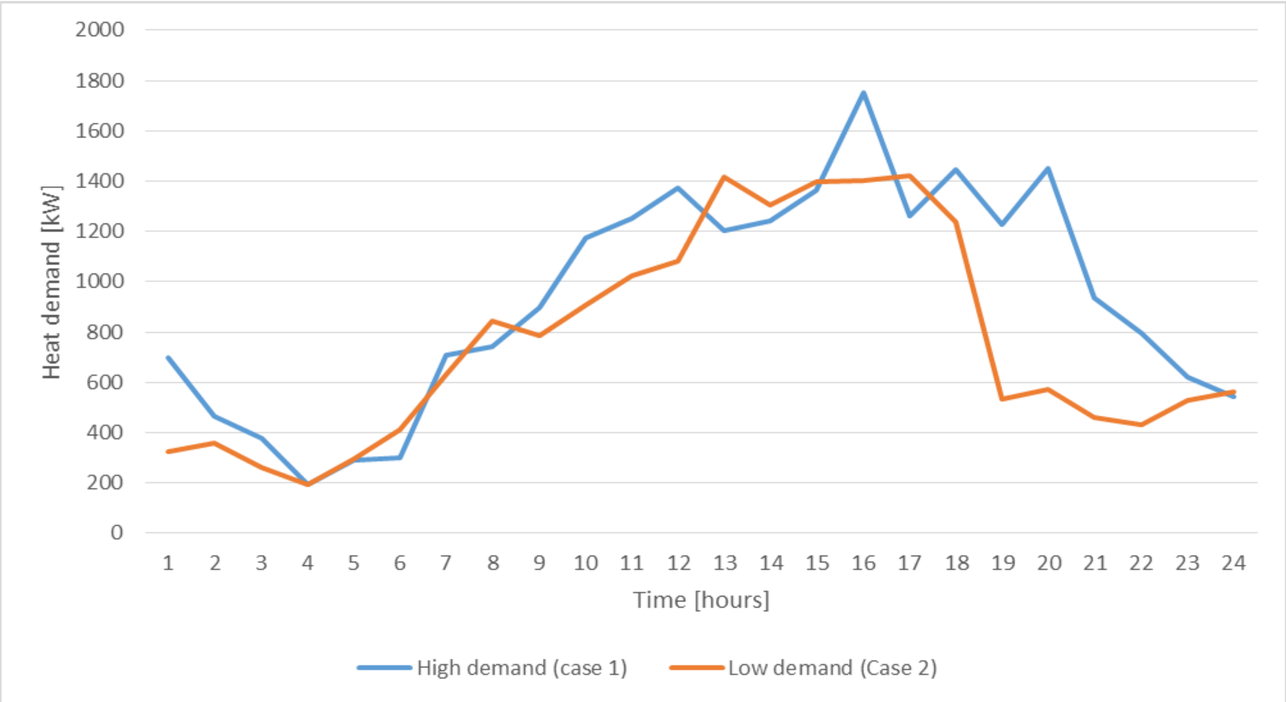

Figure 2. The daily heat demand of two days measured in an industry process: one day with high and one day with low demand

The results were compared to the results of Wersland et al. [19], using the same model and assumptions, which considered steady state loads ranging from 10 to 130% of the design point, which was 1.1 MW, and also steady state changes in temperature levels. The intercooler heat was also used in the process and included in the COP.

3. RESULTS

For the case with oscillating changes in both demand and temperature, a constant start-up period was necessary for the numerical calculations to stabilize. The start time for the fluctuations was therefore delayed by approximately 45 minutes. The average, minimum and maximum values of the simulation results are tabulated in Table 2. The results for the two cases using daily measured heat demand as input are presented in Table 3. These are also presented as the average, minimum and maximum values obtained.

Table 2. Simulation results for fluctuation of heat sink and source temperature

	Average value	Minimum value	Maximum value
Absorber pressure [bar]	26.13	21.93	30.19
Desorber pressure [bar]	2.01	1.93	2.11
Mass flow of H_2O/NH_3 [kg s^{-1}]	3.70	3.09	4.04
Discharge temp. 1 [°C]	166.12	138.31	190.29
Discharge temp. 2 [°C]	166.48	143.04	194.79
Frequency 1 [Hz]	44.56	36.40	49.28
Frequency 2 [Hz]	33.32	22.82	43.84
COP [-]	**1.99**	**1.67**	**2.35**

Table 3. Simulation results for a daily heat demand: Case 1 is for a high demand day and Case 2 for a low demand day.

	Case 1			Case 2		
	Average	Minimum	Maximum	Average	Minimum	Maximum
Absorber pressure [bar]	26.24	25.79	26.82	26.36	25.92	26.81
Desorber pressure [bar]	2.00	1.86	2.15	2.00	1.91	2.21
Intermediate pressure [bar]	7.25	6.90	7.51	7.26	7.04	7.65
Mass flow of H_2O/NH_3 [kg s^{-1}]	3.14	0.63	6.27	2.50	0.60	5.28
Discharge temp. 1 [°C]	167.08	162.20	171.33	167.88	163.09	171.02
Discharge temp. 2 [°C]	161.65	145.27	184.49	157.92	145.02	174.84
Frequency 1 [Hz]	37.61	7.77	78.28	30.29	7.48	62.38
Frequency 2 [Hz]	27.49	5.32	57.77	21.87	5.11	46.61
COP [-]	**2.02**	**1.72**	**2.25**	**2.08**	**1.83**	**2.26**

4. DISCUSSION

Despite changes in mass flow and inlet temperatures, the heat pump, continuously controlled by the regulators, managed to keep the outlet temperature at the desired target point, 115°C. The discharge temperature of the second compressor is quite high, up to 194.79°C. It could be reduced by:

- using equipment for very high temperatures, available for CO_2 system (up to 250°C) [13]
- cooling the fluid during compression, for example through oil cooling [2]
- increase the middle pressure and thus allow more intercooling and less superheat gain in the second compression
- increasing the number of compression stages to three, and add another intercooler
- reduce the high pressure by applying more cooling (larger heat exchanger)

Comparing the simulation results for the simulation with temperature conditions fluctuating about the design point, shown in Table 2, to the existing data with steady state changes in mass and temperature in [19], the results are as expected: The average values are similar to the values for 100% heat load. The COP was 1.99, almost identical to 1.98 for the static simulation in [19]. The variations in COP are larger for the simulation with oscillating temperatures than for the two cases with dynamic, daily heat demand, where only the mass flow was changed. The extrema for the COPs are 2.9-8.7% lower and 4.0-4.4% higher than for the simulations with changing demand at constant temperature levels. This shows that temperature affects the result more than the load, which was also concluded in [19]. The simulation results for the daily heat demand of case 1 and case 2 are shown in Table 3. The average values do not coincide exactly with the values obtained for 100% load in static simulation, yet they are in the same range. The average COPs are 2.0 and 5.1% higher than for the design load in [19]. This is because the daily average load is lower than the design load. Yet, the design load was set higher than the average because otherwise, far too poor performance would be expected at the highest loads. There is only a small difference between the two cases with changing demand: The one with a lower load generally gave a higher COP just like in [19], on average it performed 3.0% better. The difference might be small because the improvement with lower load in [19] stagnated below 60% load. As the two cases considered here both operated at loads around 60% or less during at least nine hours of the day, the COP would be similar for both cases during large parts of the simulations. The simulated demands also have a similar trend in increase and decrease, making the COP similar all day. The main difference between them seems to be the peak of the high demand day. Yet, one peak does not affect the average value very much, as it is only one of 24 data points. The highest load for Case 2 is 130% of design load, and the COP of 1.83 was then very close to the value for 130% heat load in Wersland et al. [19], which was 1.81. The highest load for Case 1 on the other hand, was 159% of the design load, yet it was still possible to achieve a COP of 1.72. A reduction in COP of only 9-11% points from 130% to 159% capacity, compared to about 16-17 % points from 100 to 130% load, can be explained by that the 159% demand is only a short-lasting peak, and its effect is reduced by the thermal energy of the heat exchanger, which probably works as a peak shaving device. At short high or low demands, the performance will not be strongly affected because there is already heat stored in the mass of the exchanger. If the load is higher for a longer period, it is expected that the performance will drop more. Most of the difference in average COP is probably due to the difference in the demands between 18 and 22 o'clock in Figure 2.

The pressures for both Case 1 and 2 did not change much, which is in accordance with the previous work [19]. The highest deviation was only 2.2% from the average value. This is as expected due to the control of the low side pressure, and because all the external temperatures were kept constant. The simulations were performed with a variable compressor efficiency, which was depending on the compression ratio. The maximum compressor efficiency achievable was 0.71. Recent development of compressors will enable better compressor performance and higher efficiencies. The determined absolute COPs of the system in the present work can therefore be improved accordingly with the improved compressor efficiency.

5. CONCLUSION

The presented work investigated the performance of compression adsorption heat pump for two industrial cases, which were characterized by a dynamic heat load demand. Despite large variations, the system did manage to control the outlet water temperature very well. Its ability to work at varying conditions with controllers and variable speed compressors thus seems promising. The performance was more temperature than load sensitive. The highest COP in the study was 2.35, and the lowest was 1.67, both obtained when the temperature fluctuated. The discharge temperature and pressure conditions were higher than what could be permitted in some cases, especially when the sink temperature was raised. The system would require changes in heat exchanger area and eventually number of compression stages to manage operation at such conditions. Results are very close to the ones at design point in [19]. It can be expected that the determined performance indicators can be improved with recent, more efficient compressor developments.

For dynamic simulations with change in heat load, the average heat pump performance was very similar to that in the static simulations, and the results for both cases were also quite similar to each other. Peaks in the demand had a smaller effect on the instant COP than one might expect, probably due to the thermal capacity of the system. The low pressure was kept constant and neither was any optimization of the system performed. The discharge temperatures were very high, both for design conditions and for most off-design operations. Further work should allow floating pressure, higher compressor efficiency, decreasing with load, reduce discharge temperatures and optimize the system concentration and intermediate pressure. This could potentially result in an improved COP.

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