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Two-stage high temperature hybrid heat pump with parallel heat sinks

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ABSTRACT

Hybrid heat pumps have operated well at temperatures above 100ºC for a decade and several improvements have been made during this period. Heat delivery at two temperature levels with a single, integrated heat pump with two absorbers in parallel is yet not exploited. Thus, current solutions deliver to one heat demand, even though industries often have several heat demands at various temperature levels; a more efficient and advanced solution should be able to deliver heat at several temperature levels. An ammonia/water hybrid heat pump was simulated at various loads, achieving ratios of delivered heat to work input in the range 2.1-2.8. Both static and dynamic simulations were performed, and values for storage capacities to even the loads were predicted. The system manages to operate and deliver the requested capacities from both 20°C and 27°C bottom cycle refrigeration, and achieves required setpoint at all loads, steady and dynamic ones. The COP at 20°C bottom cycle might be not acceptable for commercialization, whereas at 27°C the COP values are improved by 15% (total COP), 18 % (HT COP) and -18% (LT COP).

Keywords: Hybrid heat pump, high temperature heat pump, integrated system, energy efficiency

1. INTRODUCTION

Energy efficient processes leads to lower use of valuable resources, less greenhouse gas (GHG) emissions and also lower running costs. Integrated heat pump systems with combined utilization of heat source and sink have been proven to be very efficient and reduced GHG emissions for retail stores (Hanne Kauko, Kvalsvik et al. 2016). Here, improvement is mainly in the refrigeration system. However, heat pumps enable industrial heat demands to be covered more efficiently than e.g. boilers and are a promising technology with respect to reduction of climate gas emissions for thermal process industry. Combined gas and heat pump heating, in which a gas-fired burner generates electricity for a heat pump and both these systems deliver heat (Mondot, Caillet et al. 2017, Nienhuis 2017), can reduce energy demand and reach high temperatures. Other developments involve pure heat pump solutions, reaching up to 180°C. These involve the rotatory heat pump (Bernhard Adler and Mauthner 2017), the hybrid or compression-absorption heat pumps (Hybrid Energi AS 2016), MVR with new developed, highly efficient turbo-compressors for steam (Bantle 2017) and the butane heat pumps in various configurations (Bamigbetan 2017, Popovac, Lauermann et al. 2017). Heat pumps delivering above 80°C are in this paper classified as high temperature heat pumps (HTHP).

Industries with both heating and cooling demand will benefit, in terms of energy demand, from integrating high temperature heat pumps. Using new efficient refrigeration systems and HTHPs, this is possible for a wide range of industries. Industries requiring heat between 100°C and 160°C include the paper, food, drying, chemical distillation, rubber and synthetical material production, machine building, wood and textile industries (Wolf, Fahll et al. 2014), and the amount of surplus heat is large. Germany alone used 14.4 TWh of electricity for cooling in 2012 (Wolf, Fahll et al. 2014). With COPs of 3-4, this gives 57.4-71.8 TWh of surplus heat.

Leakage of refrigerants contribute to reduced HSE, ozone depletion and global warming. Yearly leakage is estimated to around 15-20% (Hafner 2014). Natural refrigerants have no or low impact on ozone depletion and global warming, whereas synthetic fluids have surprised with detrimental effects twice before. New synthetic fluids involve the risk of introducing another such problem. Natural refrigerants cannot cause an effect that was not already there, and are therefore desirable to use.

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A green field dairy plant investigation is characterized by cooling demands down to -1.5°C and heat demands up to 102°C. A comparative study to determine the most environmentally friendly solution was carried out to evaluate different relevant alternatives (Kvalsvik and Bantle 2018). As a part of this, one alternative was the implementation of a hybrid ammonia/water HTHP. Hybrid heat pumps were introduced in 1895 by Osenbrück (Osenbrück 1895), but not fully exploited until recently (Itard 1998). Today, several installations with water and ammonia have been running successfully for the last years, delivering above 120°C at less than 25 bars (Hybrid Energi AS 2016). Further development and improvement is still ongoing. The working principle and potential has been studied in detail (Itard 1998, Nordtvedt 2005, Jensen, Markussen et al. 2015a, Wersland, Kvalsvik et al. 2017) and show very promising results. The investigated dairy requires a heat pump to deliver heat at two temperature levels and the present paper investigates if a single hybrid heat pump could be operated with one heat source and two heat sinks. The HTHP should use the surplus heat from a bottom cycle refrigeration system operated by R744 (CO_2), condensing at 20°C or 27°C, depending on which gives best total results. As direct systems are more energy efficient than indirect ones, it was desirable to utilize heat from the R744 refrigeration system directly. The challenge is that the hybrid heat pump has gliding temperature, whereas CO_2 has most of its surplus heat at a constant temperature. A model was made to investigate whether the HTHP can take heat from CO_2 at these two conditions, and whether investment cost could be reduced.

2. METHODOLOGY

2.1 Feasibility

Operating conditions for the heat pump and the proposed heat pump configuration are given in Table 1. The working principle of an absorption-compression heat pump is explained in (Wersland, Kvalsvik et al. 2017). Feasible operation conditions for the binary working fluid had to be identified as a first step. The temperature glide of a mixture upon heat transfer changes with its composition. Thus, a mixture of water and ammonia, at an enthalpy corresponding to expected absorber outlet conditions, should give a temperature well below the CO_2 condensing temperature (20 or 27°C) upon throttling to a pressure above 1 bar. A log P-(1/T) diagram can be used to estimate this, but calculations were required to find the more exact number.

Higher ammonia concentration gives flatter temperature slope and higher possibility to retrieve heat. However, pure ammonia cannot deliver the HT demand, and the system efficiency increases with higher share of water, as this increases the amount of working fluid that can be pumped rather than compressed. The temperature of different mixtures at 1.2, 1.45, 1.95, 2.05, 2.44 and 2.51 bar was found after throttling from assumed absorber outlet conditions of 22 bar and either 90°C or 70°C. Then, the best mixture was used in modelling of a HTHP system with two parallel absorbers, see Figure 1, with rated loads of 950 kW for HT and 350 kW for LT.

Table 1: Required delivery conditions for the heat pump

Heat demand	High temperature (HT)	Low temperature (LT)	Comment
Inlet	85°C	40°C	Assumed constant in this work.
Outlet	102°C	67°C	Target values

2.2 Modelling tool

To simulate and dimension the hybrid heat pump, the dynamic, object-oriented modelling software Dymola (Dassault Systemes 2002) was used with the TIL-libraries (TLK-Thermo GmbH 2010) were used, which include fluid properties and component models, verified with experiments to be as realistic as possible.

2.3 Demand data

The industrial heat demand will vary, and hence, running the heat pump model with the best mixture for the two different source temperature conditions at different load combinations was performed to evaluate the performance in various situations. The HTHP with a source at 27°C is expected to perform better, but the total system solution for the dairy was found to be better in terms of energy and climate impact with 20°C in the refrigeration system, provided one can achieve a COP above 2.0 (Kvalsvik and Bantle 2018). Thus, the least energy efficient solution obtained here could still be the preferred solution. After the initial steady-state evaluation, a dynamic simulation was performed. Dynamic load profiles were taken from (Kvalsvik and Bantle 2018), and to make simulations faster and simpler, equations that modelled these demands very closely (deviation of 1.2% and 5.4%) were made and used, seen in Figure 2. The base load for processes was 7% load.

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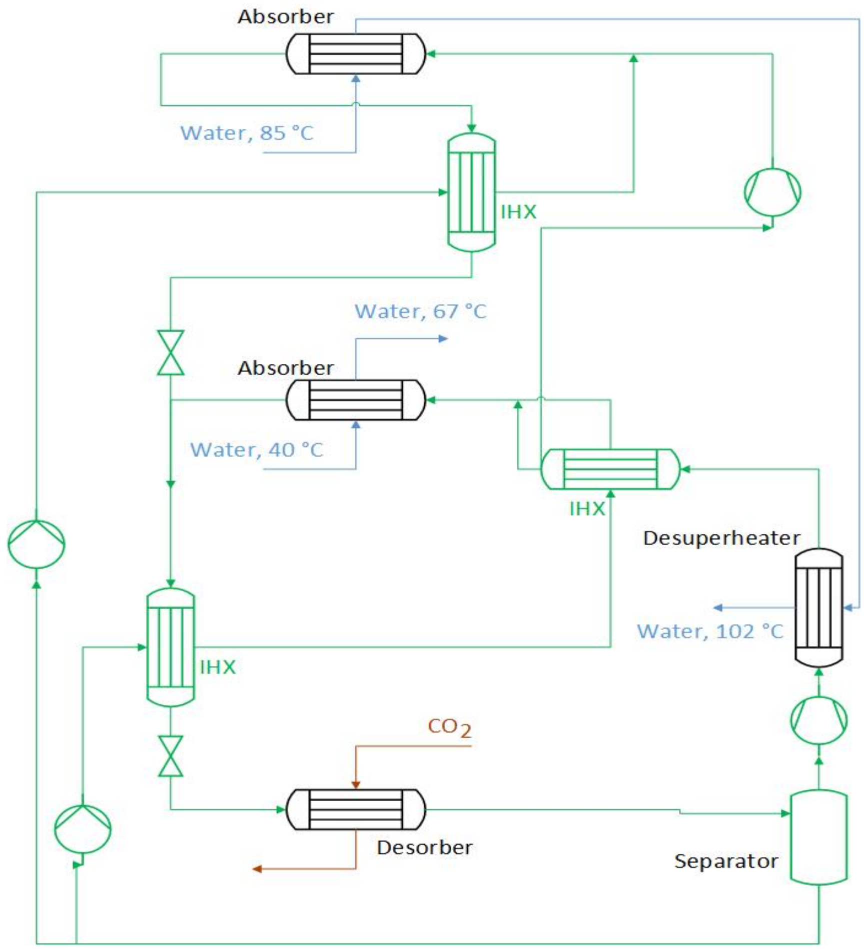

Figure 1 Setup of the parallel absorber hybrid system: CO_2 is condensed in the desorber. The separator separates ammonia gases for compression from the liquid. Water is heated in the two absorbers and the desuperheater. Internal heat exchangers (IHX) improve the efficiency.

2.4 Peak shaving

Peak shaving involves moving heat demands in time by storing heat/cold, in tanks or wells, to periods with higher demand, enabling reduced capacity and investments for heat pumps, as well as higher average loads for/better utilization of the equipment. This was also investigated with the aim to reduce the capital costs of the system. The HT demand is for processing, and can be planned. Reducing the HT peak demand by 30% (this was 875 kW in the demand data, lower than rated capacity), the demand exceeding this was moved to the hours before. To obtain a smooth start and stop, a cut-off sine-curve was used rather than a step-function. The LT demand, covering heat demand of buildings, was peak shaved by assuming that 70% of the original capacity is delivered when the storage tank is not full and the required demand (but no more than 70% of maximum) otherwise. The tank size for LT was assumed to be 50 m^3 and initially 60% filled, the HT tank was 90 m^3 and initially 1/6 filled. Reported exit temperatures are taken from the entrance to the tanks. Heat and pressure losses are neglected in the model for simplification.

The COPs of the high temperature and low temperature demands were used to evaluate system performance. To get separate COPs of the combined system, component power usage, mass fractions and heat transferred in each absorber and desuperheater were used in equation 1. Q_i is the sum of heat transferred to either the high or low temperature water line and the summation is the mass fraction of ammonia mixture going to either high or low temperature heating through component j multiplied by power consumption of component j.

$$\mathrm{COP_i} = Q_i \left(\sum_{j=1}^{N} w_{i,j} \cdot P_j \right)^{-1} \qquad 1$$

Internal heat exchange was applied to improve the COPs and a part of the superheat after the first compression stage was utilized to deliver heat to the HT demand (Wersland, Kvalsvik et al. 2017). The compressors had variable isentropic efficiencies η_{is} following

$$\eta_{is} = 0.3862 + 0.0016\pi^3 - 0.0333\pi^2 + 0.1892\pi \qquad 2$$

where π is the pressure ratio across the compressor (Eikevik, Hafner et al. 2016).

3. RESULTS AND DISCUSSION

3.1 Steady state solution

The system according to Figure 1 was simulated for different capacities for HT and LT in steady state for two ammonia-water concentrations. The results are shown in Table 2. A "Total COP", the ratio of total heat delivered to the total work input was used an indication of the overall system performance.

Table 2: Steady state results for assorted load and concentration cases

40% water concentration					30% water concentration				
LT load [%]	HT load [%]	COP LT [-]	COP HT [-]	COP tot [-]	LT load [%]	HT load [%]	COP LT [-]	COP HT [-]	COP tot [-]
2.5	100.0	3.55	2.10	2.11	0.01	100	4.96	1.54	1.54
10.0	100.0	3.66	2.08	2.11	25	25	4.95	1.59	1.94
25.0	25.0	3.61	1.97	2.24	25	50	4.95	1.58	1.76
25.0	50.0	3.40	2.04	2.17	25	75	4.95	1.56	1.69
25.0	75.0	3.61	2.05	2.16	25	100	4.95	1.54	1.63
25.0	100.0	3.83	2.05	2.13	50	25	4.94	1.56	2.20
50.0	25.0	4.07	1.84	2.40	50	50	4.94	1.57	1.92
50.0	50.0	3.84	1.96	2.25	50	75	4.94	1.56	2.20
50.0	75.0	3.87	1.99	2.20	50	100	4.94	1.54	1.72
50.0	100.0	3.98	2.00	2.16	75	25	4.94	1.54	2.41
75.0	25.0	4.31	1.74	2.53	75	50	4.93	1.56	2.06
75.0	50.0	4.04	1.89	2.33	75	75	4.93	1.55	1.90
75.0	75.0	4.03	1.94	2.25	75	100	4.93	1.53	1.80
75.0	100.0	4.09	1.95	2.20	100	0.5	4.94	0.18	3.62
100.0	1.0	5.31	0.02	2.83	100	1.5	4.94	0.50	3.68
100.0	25.0	4.45	1.66	2.63	100	25	4.93	1.51	2.58
100.0	50.0	4.19	1.83	2.39	100	50	4.92	1.55	2.18
100.0	75.0	4.14	1.89	2.23	100	75	4.92	1.54	1.99
100.0	100.0	4.17	1.92	2.24	100	100	4.92	1.53	1.88
Carnot COP (all loads)		8.50	5.00	-	Carnot COP (all loads)		7.24	4.58	-

The steady state solutions used concentrations of (0.3, 0.7) for the 20°C case and (0.4, 0.6) for the 27°C case. The modelled system efficiency retrieving heat at different temperature levels and using different ammonia concentrations are mostly identical. Apart from slightly different setpoints resulting from the different temperatures, the main difference is that the displacement volume of the second stage compressor is twice as large when using a source temperature of 20°C compared to 27°C. The loads in the two absorbers were different combinations of lowest achieved (≤2.5%), 25%, 50% and 100%.

Results show that the suggested system can provide heat to both high and low temperature water lines in parallel, and can still obtain COPs in a sufficiently high range for a water/ammonia mixture of (0.4, 0.6) and a source temperature of 27°C. The COP is above 1.8 at the highest temperature level for all HT loads ≥50%, and

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always above 3.4 for LT. On average, it is around 1.9 for the HT load and 4.0 for the LT load, and at the best values are 2.1 and 5.3. The solution with a source temperature of 20°C was possible, but required a water concentration of 0.30. Based on results in Table 2, increasing the water concentration to 0.40 improves the COPs for most load conditions. For all load conditions, using a concentration of water/ammonia of (0.3, 0.7), gives high temperature COP values below 1.6, low temperature COP close to 5.0 and a total COP between 1.7 and 3.7 depending on load.

Using a concentration of (0.4, 0.6) gives high temperature COP values in the area 1.9-2.1, a more erratic low temperature COP (in the area 3.4-5.3) and an improved and more consistent total COP (in the area 2.1-2.8). Figure 2 summarizes the total COP for different load combinations between 25% and 100%. When the high temperature load was reduced below 25%, there was a clear decrease in COP. Since the demands are coupled, heat transferred to the high temperature water increase as load decrease in the low temperature one improving high temperature COP. The concentration of 0.4/0.6 at 27°C generally performed more effectively and was therefore used in the further dynamic simulations.

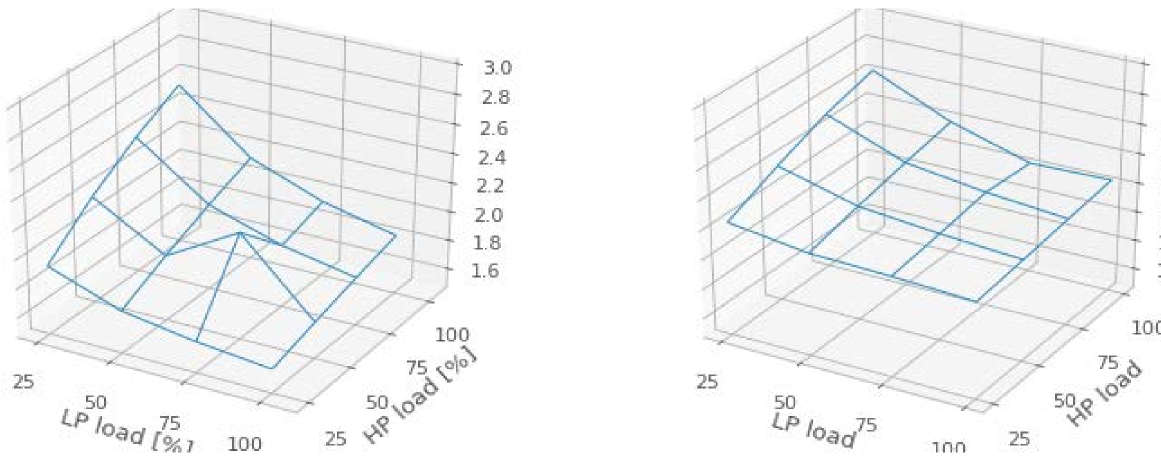

Figure 2 Graphical comparison of total COP for two concentrations, (0.3, 0.7) to the left and (0.4, 0.6) to the right, for loads between 25% and 100%

3.2 Dynamic solution

Using the dynamic demand profiles, the system is modelled for a 9-day period, following demand profile in Figure 3. The simulated period includes some of the highest and lowest low temperature loads and several on/off-situations (weekend and nights). The purpose of these simulations is testing the applicability of dynamic behaviour and peak shaving. Both HT and LT dynamic demands were successfully simulated simultaneously, with the best of the two concentrations (40% water concentration). All setpoints were reached and maintained closely. Delivered temperatures never deviated more than 0.02% from their setpoints during dynamic simulation. The largest deviation for absorber water concentration was 0.45%.

Average COPs for the period are shown in Figure 4. The LT COP is high on average, higher than in steady-state and better when peak shaving is applied. For HT the trend is opposite; average COP is 1.52 without peak shaving and 1.43 with. Considering only periods of demand >7%, the COP is 1.87 or 1.82 or average. Thus, the COP increases with higher load. This is even more apparent in Figure 5, showing instant COPs and loads as a function of time. The systems are closely coupled, and the COPs affected by both loads, even when trying to separate their power consumption. The opposite trends might be a result of this coupling. The LT COP in Figure 5 is quite stable around an average value of 6.32, but generally peaks when the HT demand falls (sending more superheat to the LT absorber) and opposite when the HT demand peaks. The HT COP peaks at 2.17-2.28, and generally increase with load.

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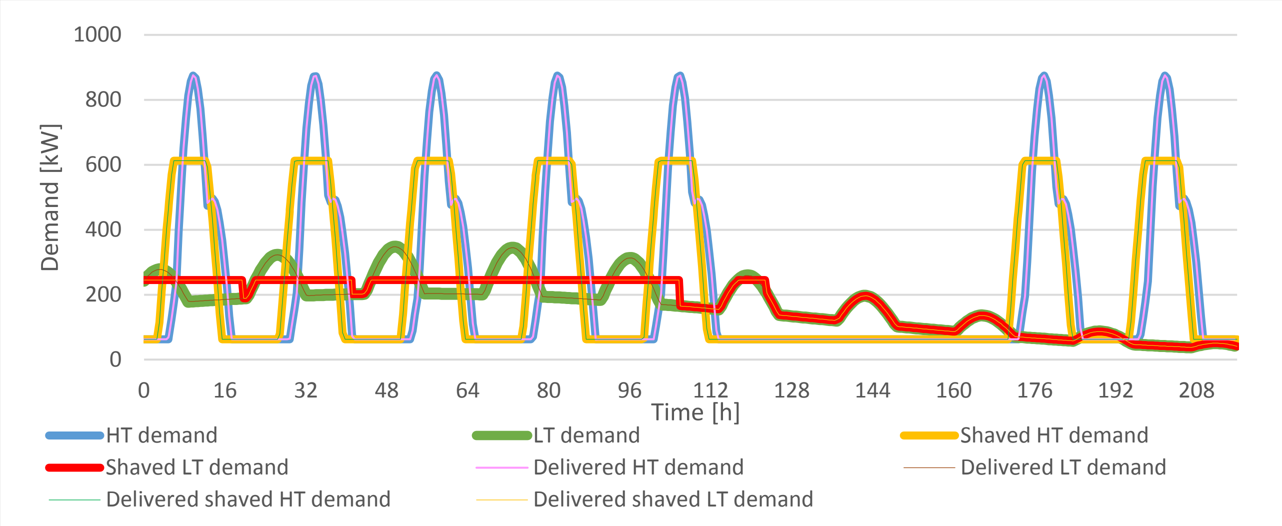

Figure 3: Modelled dynamic demands (equation based), deviating 1.2 and 5.4% from the demand data based on ambient temperature and process demands, and delivered heat for all cases

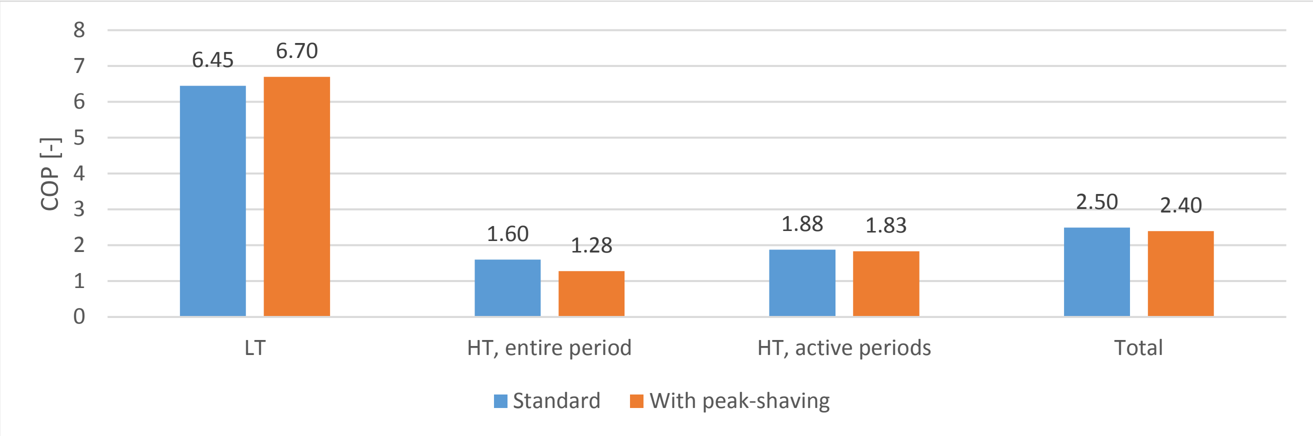

Figure 4: Average COPs for Low Temperature (LT) demand, High Temperature (HT) demand and Total COP with and without peak shaving (Standard): For HT, both the average COP for the entire period (including the 7% loads) and only for the active periods of HT heat supply are shown.

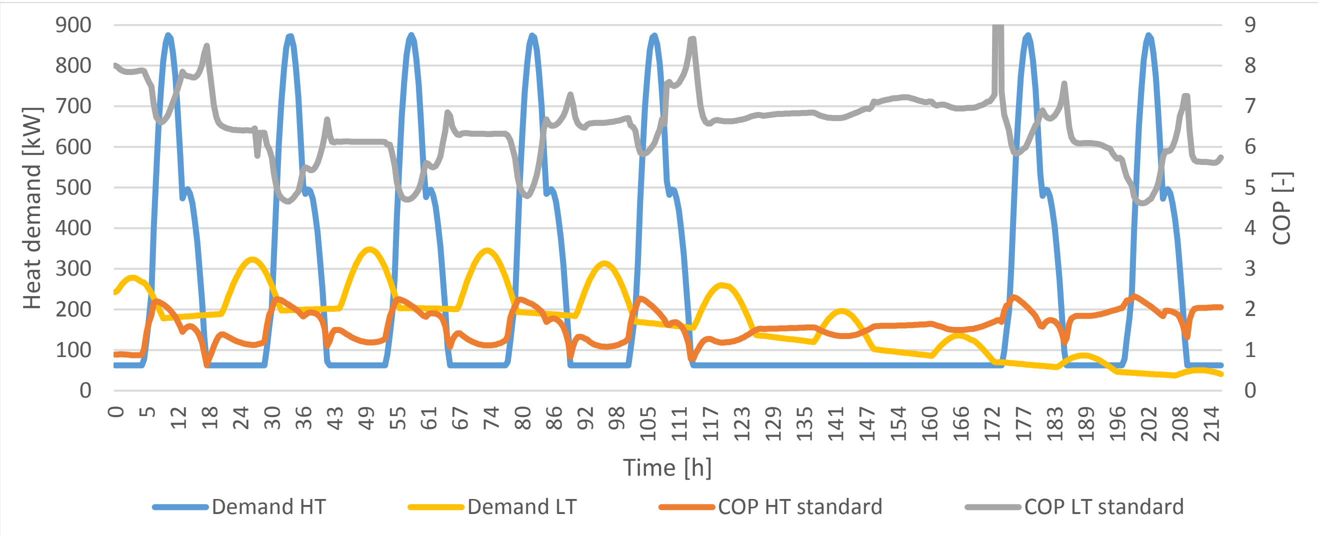

Figure 5: Correlation between the COPs and loads in dynamic simulation

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Peak shaving was successfully applied, and demonstrated that the heat pump size could be reduced by 30% by means of a tank of about 90 m^3 for HT (largest difference in filling volume was 76.1 m^3 seen in Figure 6) and a 50 m^3 tank for LT (largest difference of 24.7m^3). Even though a 50 m^3 tank was assumed for LT, the filling level was never below 25.6 m^3, hence, a smaller tank could be used. The LT COP increased 3.9% and the HT COP decreased by 2.1% when peak shaving was applied, which was considered insignificant.

Currently, the modelled pumps and compressors react to change of water flow through absorbers. Implementing predictive flow rate through pump and compressor might improve performance while reducing investment costs. This is partially done when peak shaving the HT demand, as this demand can be planned to a large extent. However, combining a smart control system with production planning and weather forecasts can help predicting heat demands and minimise both CAPEX and OPEX. To keep the COP high, it is suggested to keep the heat pump load constant during weekdays and off in the weekends, as it keeps HT load above 25%. An economic consideration is whether it is more cost efficient to have a large HT storage tank to be filled during night or other periods of the day with low electricity price.

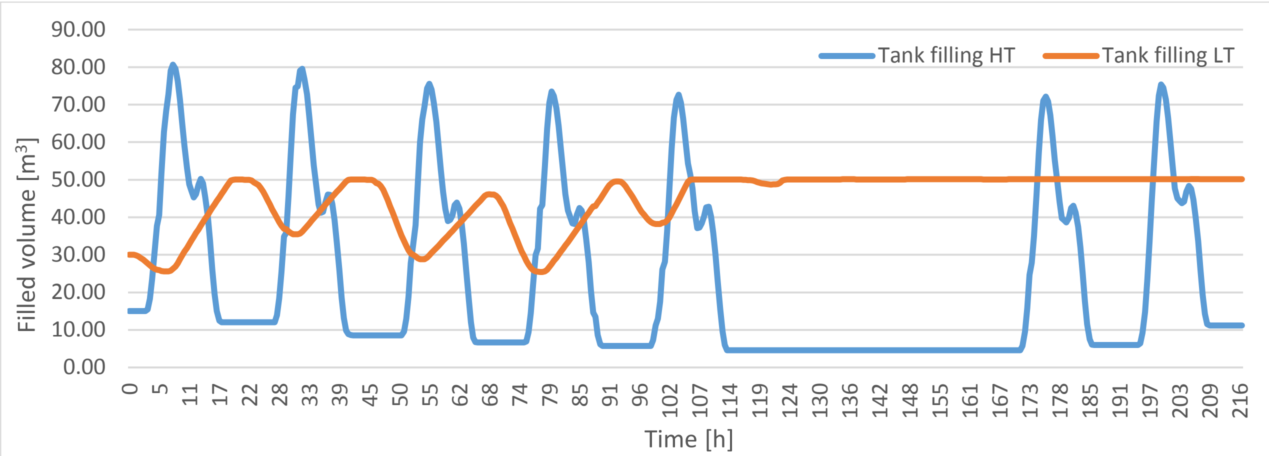

Figure 6: Filling level of the two tanks used for peak shaving of the high temperature (HT) and Low Temperature (LT) demands

Steady state and dynamic simulation gave similar results for HT, which generally achieved around 40% of the theoretical maximum, whereas the results for LT varied strongly, and achieved 40-60 % of maximum in steady state and around 77% in dynamic conditions. This could suggest that Eq. 1 is not sufficient to decouple the demands completely, as also seen in Figure 5. The chosen approach might be not precise enough for a combined system. However, as the loads in industry are often variable, this demonstrates the necessity of dynamic simulating systems in order to evaluate a realistic operational behaviour. The steady state results are useful for first evaluations.

Naturally, two separate hybrid heat pumps would give better flexibility, and hence be easier to control and optimize. From an operational point of view, this might be the better solution. However, the cost of two separate system is higher. Decoupling the demands would also remove the benefit of being able to utilize the surplus heat after the first compression stage to heat the high temperature sink, and this option was not compared to the combined solution, thus, better efficiency is not guaranteed with separate systems. The constant temperature heat source is far from ideal for a hybrid heat pump with gliding temperature. One could run the R744 refrigeration system transcritical to improve the HTHP COP, but results for the overall system suggests that this would be detrimental for the overall performance the dairy plant.

4. CONCLUSIONS

The system manages to operate and deliver the requested capacities from both 20°C and 27°C bottom cycle refrigeration, and achieves required setpoint at all loads, steady and dynamic ones. The COP at 20°C bottom cycle might be not acceptable for commercialization, whereas at 27°C the COP values are improved by 15% (total COP), 18 % (HT COP) and -18% (LT COP). Using a 40% water concentration (instead of 30%) gives an improved COP of 2.24 at 100% load. The HT and LT COPs show opposite trends: LT COP is higher when

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using 30% water mixture or peak shaving, whereas the HT COP is lower at 30% water mixture or when peak shaving is applied. As the HT load is larger, increasing the HT COP should have priority.

The dynamic simulation was stable when modelling several days including the lowest and highest heat demands and steepest gradients. Results show that 30% peak shaving could be obtained by tanks of 30-50 m^3 for LT and 80-90 m^3 for HT. Recommended future work within this project include optimization of pressure levels and ammonia/water concentration to improve COP. Inclusion of the rest of the system could alter performance. To properly asses the system for industrial use, an investigation regarding economic benefits of different storage tank sizes and filling regimes should be undertaken. Cost efficiency is a strong motivator for the industry for phasing in renewable energy solutions.

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NOMENCLATURE

COP	Coefficient Of Performance (-)	P	power (W)
HT	High Temperature	Q	heat (W)
HTHP	High Temperature Heat Pump	η	Efficiency (-)
LT	Low Temperature		

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