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Design considerations for dynamically operated large-scale ammonia heat pumps

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ABSTRACT

Large-scale heat pumps (HPs) are being implemented to provide heat for district heating (DH) systems from renewable heat sources. They are further expected to play a key role in integrated energy systems as flexible loads that can be used to balance the electricity grid. However, most large-scale HPs installed are designed for base load operation and are thus not optimized with regard to dynamic behaviour. Within this work, the dynamic behaviour of different HP configurations was assessed with regard to performance during load change. For this purpose, a two-stage and a one-stage ammonia HP were modelled using Modelica. Further, different adaptions of the two-stage HP to avoid condensation in the suction line were assessed. Fast ramping resulted in oscillation of the liquid level in the receivers. It would thus be necessary to design receivers carefully and include suction traps to avoid entrainment of droplets into the suction lines. The results indicated that condensation in the low-pressure suction line was a limiting factor for how fast the HPs may ramp down. This could be avoided by preheating of the suction line. In that case, the HPs under assessment were able to regulate between 100 % and 40 % load in less than five minutes, and they would thus be capable of providing secondary regulation power.

Keywords: Large-scale heat pump, dynamic operation, dynamic model, ammonia, regulation power

1. INTRODUCTION

Denmark has decided to reduce CO2 emissions drastically in order to comply with international climate change mitigation targets and national energy targets. The target is to supply all heating and electricity from renewable energy sources in 2035 (The Danish government - Energi-Forsynings- og Klimaministeriet, 2013).Renewable energy sources such as wind or solar power are fluctuating and difficult to predict. In order to secure grid stability even with high shares of fluctuating power production, flexible units on the production and demand side can be used to balance out the frequency in the grid.

Heat pumps (HPs) enable the use of ambient or low-grade heat sources for heating purposes by lifting the temperature from the source temperature to values high enough to supply the required heating service. The fuel for the cycle is most cases electricity. Thus, HPs provide a link between the electricity and the heating sector, and thereby they may act as flexible loads while supplying renewables-based heat. The flexibility is provided by the possibility to store heat in storages, the buildings or in the district heating (DH) grid.

Flexible operation of heat pumps may results in additional losses on the heating side (Meesenburg et al., 2018). However, the flexibility to adapt the electric capacity of HPs can be used to provide frequency regulation and thus act on the reserve markets. This may offer additional income for plant operators. In order to do so the regulation rate needs to be fast enough in order to comply with the standards of the transmission system operator (TSO).

While small-scale HPs can switch on and off rather quickly, large-scale HPs for DH supply are usually designed for base load operation and thus, not optimized for fast regulation (David et al., 2017). Accordingly, ramping rates in many applications are limited (Averfalk et al., 2014). In order to improve the ability of large-scale HP systems to deliver regulation services, it has to be identified what causes the limitations with regard to ramping rates and thereby, how the HP design and control can be improved (Fischer and Madani, 2017).

Danish law forbids the use of HFC refrigerants in large-scale applications because of their high global warming potential (Miljøstyrelsen, 2016). Natural refrigerants such as ammonia are mostly used for large-scale applications in Denmark today and accordingly, we chose to focus on ammonia as a refrigerant for this study.
This study focused on the description of dynamic behaviour during load change of large-scale ammonia HPs in a one-stage and a two-stage configuration. The aim was to identify what limits fast ramping up and down of the HP. Liquid formation in the suction line during ramp-down has been identified as a limitation to fast ramping. Accordingly, different strategies to avoid this were assessed. The study was based on a dynamic simulation of the HP configurations, which is further described in section 2. The results are presented in section 3 and discussed in section 4, before conclusions are presented in section 5.

2. METHOD

2.1. Heat pump application case

The HPs simulated in this study were designed to fit a real HP application, supplying DH to a small grid in the development area of Nordhavn in Copenhagen, Denmark. The design heating capacity of the system was 800 kW. As heat source groundwater at 10.5 °C was used. It was cooled down in the evaporator and ejected into the sea. This was possible as the groundwater reservoir is infiltrated by seawater anyway. The HP was able to deliver forward temperatures of 65 °C to 84 °C. For this study, we assumed a forward temperature set point of 70 °C and a constant return temperature of 35 °C. The assessed system was a two-stage ammonia HP, which comprises a flooded evaporator, separator, low-stage compressor, open flash intercooler, high-stage compressor, desuperheater, condenser, subcooler and two expansion valves as main components. Pipe models were included for both suction lines of the compressors. For better understanding of the nature of limitations during ramping of the HP, the results for the two-stage HP were compared to a one-stage ammonia HP, supplying the same services. This was comprised of flooded evaporator, separator, compressor, desuperheater, condenser, subcooler and one expansion valve. A sketch of the two configurations is shown in Figure 1.

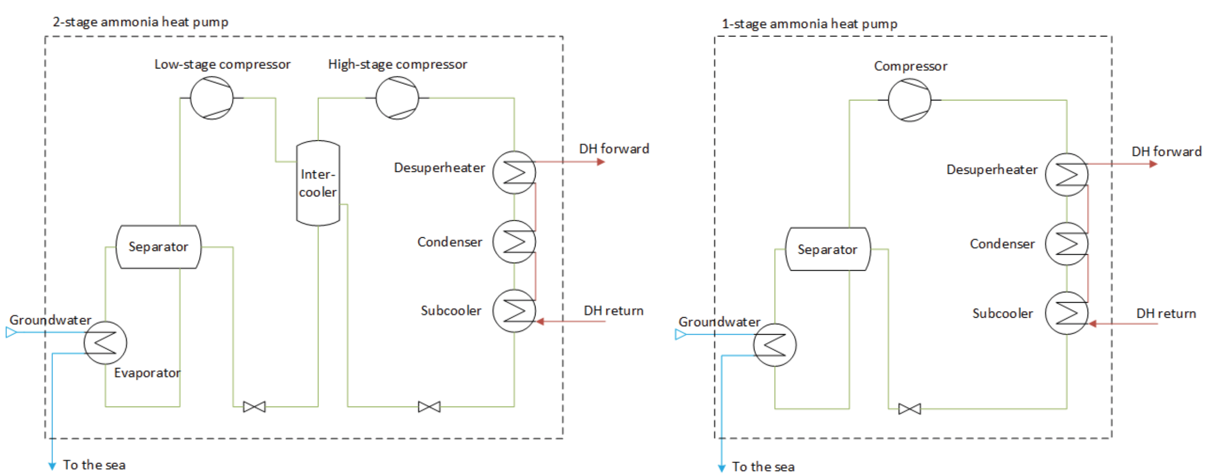

Figure 1 Sketch of 2-stage and 1-stage ammonia HP

2.2. Dynamic model

In order to simulate the behaviour of the system during up- and down ramping a dynamic model of the system was built in the object-oriented programming language Modelica (Modelica Association, 2017) using Dymola software (Dassault Systèmes, 2017) and the TIL library (TLK Thermo GmbH & IfT, 2017). The latter includes validated component models for the simulation of refrigeration systems. All components were modelled using dynamic energy and mass balances. Momentum balances were modelled using steady state equations, as the time constants for momentum balances are relative small compared to energy and mass balances.
The heat exchangers were modelled as corrugated plate heat exchangers using heat transfer correlations of Steiner (VDI, 2010) for evaporation, of Shah (VDI, 2010) for condensation and of

Gnielinski, Dittus & Boelter (Baehr and Stephan, 2011) for one-phase flow and a quadratic mass flow dependent pressure drop for ammonia. The pressure drop on the water side was neglected and the heat transfer coefficient was calculated according to (VDI, 2010). The pipe model included Dittus & Boelter heat transfer model and Konakov's pressure drop correlation for smooth pipes (VDI, 2010). Heat conduction from the ambient through steel pipe and insulation was considered.

The compressors were piston compressors equipped with variable speed drive for part-load operation. The oil-cooling was not modelled in detail, but was considered as a heat loss from the compressor. The compressor efficiencies were calculated from compressor polynomials, which were functions of condensation temperature, temperature lift, pressure ratio and compressor frequency. The displacement volumes were assumed to be 0.009425 m^3 and 0.0044 m^3 for the low-stage and high-stage compressor of the two-stage HP. The high-stage compressor polynomial was also used for the compressor in the one-stage HP, due to the high temperature tolerance (Johnson Controls, 2019). The displacement volume was 0.0088 m^3.

The separators and open-flash intercooler were modelled as ideal tanks, i.e. equilibrium between the gaseous phase and the liquid phase and uniform pressure and temperature in the tank were assumed. As the liquid levels were assumed to be low, the pressure difference between gas- and liquid outlet was neglected.

2.3. Control of the heat pumps

The aim of controlling the HPs was to ensure a certain DH forward temperature and heat output at high efficiency and safe operation of the HPs.

In the case of the two-stage HP there were 6 manipulating variables, namely the rotational speed of both compressors, the source water flow, the DH water flow and the cross sectional area of both expansion valves. The pairing of manipulating and controlled variables was deducted from a relative gain array (RGA) analysis for the two-stage and one-stage HP (Seborg et al., 2010). Resulting in the following control structures:

- *Two-stage HP*: The source water flow was fixed and the water temperature at the evaporator outlet was controlled with the low-stage compressor rotational speed. Thereby, the capacity of the HP was controlled indirectly. The rotational speed of the high-stage compressor controlled intermediate pressure level, ensuring similar pressure ratios for both compressors. The DH flow was altered to control the condensation temperature and thereby indirectly the DH forward temperature. The high-stage expansion valve controlled the fluid quality at the condenser outlet, and the low-stage expansion valve was used to control the liquid level in the separator.
- *One-stage HP*: the forward temperature was controlled by the DH water flow directly and the heat output was controlled by the compressor rotational speed. The separator level was not controlled, but the valve was used to control the condenser outlet quality of the fluid.

All controls were implemented as PI-controllers into the model. The parametrization was conducted according to the method described by Kuhn (1995).The integration time constants for the 1-stage system were reduced slightly to match those found for the two-stage cycle, in order to ensure that the results are comparable. This resulted in slightly faster regulation of the one-stage HP.

2.4. Operation strategy

In order to deliver frequency regulation, the HPs need to adapt their electricity consumption quickly. It was assumed that only upward regulation defined from an electricity supply perspective was provided. Accordingly, the HP can supply upward regulation by reducing electricity consumption. This could be achieved either by

- ramping down into part-load and ramping up again when the regulation period is over
- shutting down the HP completely and starting up again afterwards.

To avoid the necessary dead times before restart of the compressors and increased wear through recurrent starts and stops the first option was considered. Tests with the existing two-stage HP have shown that the HP can operate at part loads of down to 40 % of full load. We thus assumed that the HP capacity can be reduced by 60 % during regulation periods. The response time for frequency normal reserve (primary reserve), secondary reserve and tertiary reserve in Eastern Denmark is 150 s and 5 min and 15 min, respectively (Energinet.dk, 2017).. The frequency (normal) reserve market is the economically most attractive option and thus the aim of this study was to find out whether it could be possible to deliver the regulation power within 150 s. The electric capacity of the assessed HP is too low to bid into the markets. That means an exceptional allowance would be needed or the HP could be clustered with other flexible assets.

2.5. Evaluation criteria

The ramping time was defined as the time during which the set point of the heat load was changed, thus forcing the system to a slower or faster change of load. This value was varied to be 150 s, 100 s, 50 s and 10 s. The time between start of the ramping and reaching steady state (ss) was defined as the response time Δt_{r}

$$\Delta t_{\mathrm{r}} = t_{\mathrm{start}} - t_{\mathrm{ss}} \qquad \text{Eq. (1)}$$

Steady-state was assumed to be reached when the heat output was within a tolerance of 0.1 % of the overall heat output. This corresponded to a tolerance of < 0.2 % of the steady-state value for the power uptake.

The steady-state COP was defined as the ratio of the steady state heat output $\dot{Q}_{\mathrm{ss}}$and power consumption $\dot{W}_{\mathrm{ss}}$.

$$\mathrm{COP}_{\mathrm{ss}} = \dot{Q}_{\mathrm{ss}} \big/ \dot{W}_{\mathrm{ss}} \qquad \text{Eq. (3)}$$

2.6. Strategies to avoid condensation in the suction line

The observed condensation in the suction line was a limiting factor to how fast the HP may be ramped down. There are a couple of possible technical solutions that may avoid condensation in the suction line during fast ramp-downs. These could help to allow a faster ramping of the HP. Here, we assessed four different strategies for the two-stage HP:

- Controlling the source outlet temperature from the evaporator to a constant temperature

The idea is to reduce the increase of evaporation pressure during fast ramp down by keeping the mean source temperature constant. This may be obtained by controlling the source outlet temperature by changing the source mass flow through the evaporator.

- Replacement of the flooded evaporator with a direct expansion evaporator (DEX)

In this case, the superheat after the evaporator could be controlled, to a value high enough to avoid liquid entering the compressor during ramp-down. The heat exchanger size of the evaporator was increased for this simulation by a factor 2, which was large enough to ensure superheating at the outlet, but it was not optimized. The superheat was controlled by the low-stage expansion valve.

- Implementation of an internal heat exchanger (IHX) to superheat the fluid in the suction line

This design would keep the flooded evaporator by installing an IHX to preheat the suction line using the liquid exiting the open-flash intercooler as heat source. The IHX was designed using manufacturer software to provide a superheating of 5 K. No further control was considered.

- Implementation of an electric preheating (el. PH) of the suction line

Ensuring superheated gas in the suction line by external electric heating is a flexible option, which may easily be added to existing systems and may allow adapting the superheating to the requirements of different operation modes. Due to simplicity for this study, we assumed a constant heating power of 2 kW throughout the simulation time.

3. RESULT

This section presents the simulation results regulation of the two-stage HP and compares them to the one-stage HP. Different possibilities of avoiding condensation in the suction line are compared.

3.1. Two-stage heat pump

The dynamic behaviour of the two-stage HP was simulated for four different ramping times. The power uptake and heat output for an exemplary case is depicted in Figure 2 and the steady state COPs were calculated as 3.45 during full-load and 3.43 at 40 % part-load. The response times of the system until steady-state are listed in Table 1.

It can be seen that the given system was not able to reach steady state in less than 150 s, which corresponds to the necessary reaction time for primary reserve in Denmark. The full regulation capacity might be delivered within 182 s to 306 s. That means that under the implemented control strategy, the system reacted too slowly to provide primary reserve, but it might be possible to provide secondary regulation power. The integration time constants of the controllers had a large influence on the system response time, i.e. optimizing the controller would allow faster ramping.

Table 1 Response time for different ramping times (change of set value) for the two- and one-stage ammonia HP cycle

Ramping time [s]	Two-stage HP		One-stage HP	
	Δt_{r} ramp-down [s]	Δt_{r} ramp up [s]	Δt_{r} ramp-down [s]	Δt_{r} ramp up [s]
150	278	306	285	303
100	238	270	240	265
50	205	221	203	232
10	182	190	179	209

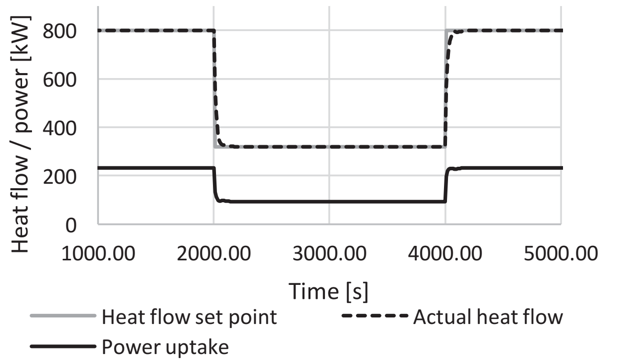

Figure 2 Operation scheme for the provision of regulation power. Depicted is the set value for the heat flow, the actual heat flow and the power uptake

The limiting factor was found to be a thermodynamic effect. As can be seen in Figure 4 the fast ramp-down from full-load into part-load led to formation of liquid in the suction line. The effect was stronger the faster the HP ramped down. This was due to a sudden increase in pressure on the low-pressure side, which caused the saturation temperature of the fluid to increase. The walls of the suction line warm up more slowly than the fluid due to the thermal inertia of the piping material. Accordingly, the wall temperature was below the saturation temperature of the fluid, which would lead to condensation along the pipe walls. The suction line in the model is only discretized in one dimension, i.e. the quality values given here are averaged over the control volumes and it is uncertain how much fluid would actually condense on the cold surface. Potentially, any droplet may however harm the reciprocating compressor and condensation in the suction line should be avoided. This limited how fast the HP capacity can be ramped down. Thus, an ammonia HP that should adapt its load quickly needs to be designed in a way that condensation in the suction line is avoided. During ramp-up, the suction line is warmer than the saturation temperature and thus there is no problem with regard to condensation.

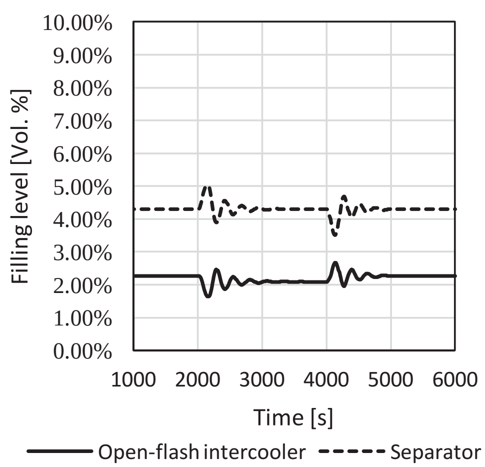

Figure 3 Filling level of intercooler and separator. Ramp-down start at 2000 s. Ramp-up starts at 4000 s.

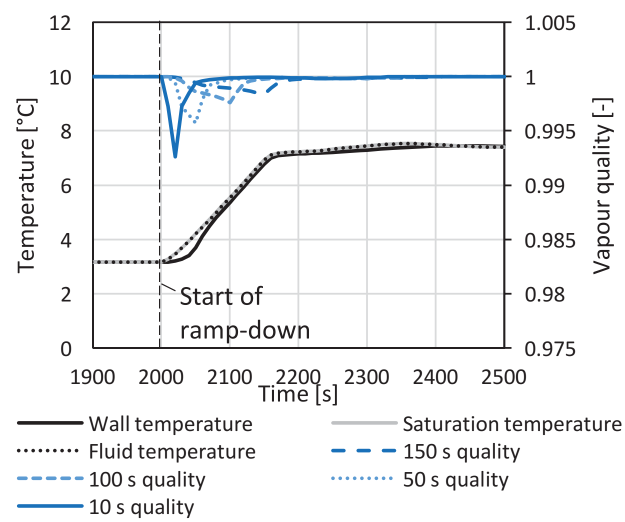

Figure 4 Fluid temperature, wall temperature, saturation temperature of the fluid and vapour quality at the end of the low-pressure suction line in a two-stage ammonia heat pump. Ramp-down into part load starts at 2000 s. Ramp-up to full load starts at 4000 s.

Figure 3 shows the liquid level of the open-flash intercooler and the separator. The fast load change induced large peaks in the filling level compared to steady-state level. This was especially critical for relatively high filling levels in steady-state, as the absolute peaks in filling level were high, too. This means that under the assumed control strategy the fast ramp down led to large agitation, which could influence the separation of liquid and vapour in the vessel negatively, and which also influences the low and high stage pressure level - and thus the operation conditions of the heat pump. The dynamic response of the filling level depended mainly on the controller parameters chosen for the low-pressure expansion valve controller. This indicated that the applied controller design method was not sufficient for this control pair and that a more elaborate controller design is necessary.

3.2. One-stage heat pump

In order to assess whether the more simple cycle design of a one-stage HP can lead to faster response time of the HP system during ramping, a one-stage cycle with the same nominal heating capacity as the two-stage cycle was simulated.

The response time for different ramping times of the one-stage HP are presented in Table 1. The response times obtained did not differ significantly from the response times of the two-stage system. This was due to the similar time constants of the controllers.

The filling level in the separator was not controlled in the one-stage cycle. Nonetheless, the induced oscillations were as low as 3 % of the steady state filling level in full load and thus significantly lower than for the two-stage cycle. Even though no advantage in terms of response time was observed, the system reaction was more robust and fewer oscillations were induced. Thus, a more simple cycle might be well suited for flexibly operated heat pumps. However, the forward temperatures that may be delivered with a one-stage ammonia HP are limited.

As with the two-stage HP the fluid state in the suction line was in the two-phase region under fast ramp-down of the HP capacity, i.e. condensation might occur. The effect was stronger for faster ramping times.

Table 2 Results for different cycle adaptions to avoid condensation in the suction line

Design	Risk of fluid in suction line [yes/no]	Δt_r down [s]	Δt_r up [s]	COP_{ss} full-load [-]	COP_{ss} part-load [-]
Base case	yes	182	190	3.45	3.43
Source control	yes	176	159	3.41	3.24
DEX	no	205	182	3.42	3.31
IHX	no	193	206	3.40	3.39
El. PH	no	254	176	3.41	3.35

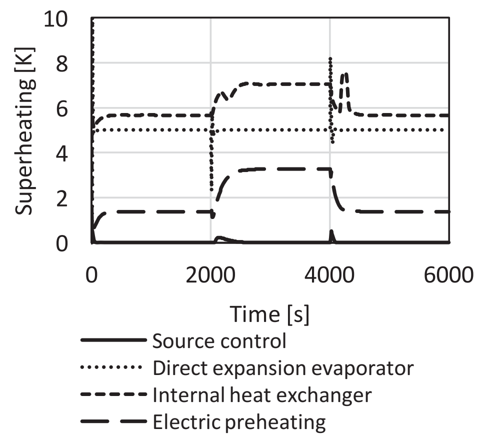

Figure 5 Superheating for four different strategies to avoid condensation in the suction line

3.3. Strategies to avoid condensation in the suction line

The results for these four adaptions to avoid condensation in the suction line are summarized in Table 2. The response times were in a similar range as for the two-stage cycle without adaption. The first option resulted in condensation in the suction line, too. Even though the increase in the evaporation pressure was less than with the original control strategy, it still increased and even peaked during ramp down. This meant that the same phenomenon as in the original cycle design occurred.

The COP in full load was highest for DEX and el. PH. In both configurations the part-load COP was comparably low. Further, Figure 5 shows that the superheating for DEX and el.PH decreased shortly during ramp-down, this means that it is crucial to design the control in a way that the superheating is large enough to accommodate this negative peak. The part-load COP of the el. PH configuration might be increased to the value of the two-stage HP base case by shutting the preheating off during part-load operation. The COP in part-load was highest for the internal heat exchanger configuration. This was advantageous in the case of the proposed operation strategy, as the heat pump would be operated in part-load during regulation periods. Thus, the loss due to regulation would be smaller and the cost of regulation would be lower. The full-load COP in the IHX case was slightly lower than for the alternative adaptions. As shown in Figure 5, this was due to the larger superheating compared to the other technologies. The superheating increased in part load and did not show any downward peaks during ramp-down of the heat pump. It might be possible to reduce the IHX size, and thereby the set point for the superheating, and thus reduce the decrease in COP compared to the base case. In the IHX case, the reduction in COP due to superheating was partly outbalanced by the effect of additional subcooling of the liquid before entering the low-pressure expansion valve. This explains the better COP in part-load compared to the other cases.

4. DISCUSSION

The results indicated that the thermal inertia of the components as well as inertia of the system due to refrigerant migration actually had a minor impact on the system dynamics, as the system was not ramped down completely and started up again, but only the load was changed. It was observed that the dynamic behaviour is strongly influenced by the controller design. In the present study, only PI controllers were considered. It is expected that the controller design may be improved by advanced controller design methods, which may possibly reduce the reaction time of the two-stage system far enough to be able to provide frequency normal reserve (150 s response time). It should however be noted that the presented simulation results did not consider dead-times of actuators or waiting times during compressor ramping. The latter was especially relevant for the two-stage cycle, as the compressor load needs to be changed in an orchestrated way during ramping. This may possibly lead to longer actual regulation times for the two-stage HP. Further, the heat pump was controlled according to the desired heating capacity, as this is the main purpose of the heat pump. In future applications, it might be considered to control the heat pump according to the power uptake and thereby force the heat pump to follow the desired power consumption profile for frequency regulation.

The results for the one-stage cycle did not show faster regulation performance than the two-stage cycle model. However, during load change fewer oscillations occurred and the overall system performance seemed to be more robust. The one-stage ammonia cycle is usually not chosen for DH applications due to the limited supply temperatures. For 70 °C forward temperature the discharge temperature exiting the compressor in the one-stage case was calculated to be ca. 172 °C. This might still be allowable with regard to thermal stability of the lubrication oil (Ommen et al., 2015), but it is at the upper end of discharge temperatures tolerated in state-of-the-art compressor technology. However, it seems promising to investigate whether one-stage cycles with other working fluids may be more suitable for flexible operation.

Additionally, fast regulation led to sudden changes in the hold up of vessels in the system, such as receivers, open flash intercooler and separators. This can lead to increased agitation of the vapour and liquid flow, which may in extreme cases lead to two-phase flow in the lines towards compressor or valve. Both should be avoided. The effect may be decreased by improved control of the valves during the ramping routine. Further, the layout of vessels as suction traps (ASHRAE, 2006) can help to secure that no liquid enters the compressors during fast load changes.

The results showed that the crucial factor when ramping the HP down quickly was the danger of condensation in the suction line due to sudden increase of evaporation pressure. It should however be noted, that from this system level analysis it cannot be determined how much fluid actually condensates in a real system, as the flow characteristics and heat conductions along the pipe might have an influence, as well. Thus, the results only can give an indication on whether there is a risk of condensation or not. Different possible strategies to avoid this risk were compared in the results of this study. Among those, the implementation of an IHX and electric preheating of the suction line seemed most promising. The first is preferable, if the system is supposed to ramp up and down frequently, as condensation in the suction line can safely be avoided under all conditions. The part load COP is comparably high for this system. The benefit of el. PH is that the superheating control is very flexible and may be switched off during part-load or when the HP is not delivering regulation power. Further, el.PH can be added relatively easy to existing HP systems. The necessary heating power needs to be chosen carefully in order to deal with the downward peaks in superheating that were observed. Another option would be to combine the flexible approach and the IHX, by bypassing the IHX during part load operation or when no flexible operation is scheduled. This would need further implementation of three-way-valves and controllers. The influence of these on the system should be assessed further.

5. CONCLUSION

This paper assessed the dynamic behaviour of a two-stage ammonia heat pump during ramping from full-load into part-load and up to full-load again. The results were compared to a one-stage cycle. The response times for both systems was comparable, but the one-stage cycle showed less oscillation during fast ramping. By ramping the heat pump down quickly, large oscillations were observed in the filling levels of intercooler and separator in the two-stage cycle. This should be considered in controller design. Further, suction traps could avoid liquid entering the suction line.
Condensation in the low-pressure suction line was identified as a major limiting factor with regard to possible ramping times. Condensation was more likely to happen the faster the heat pump was ramped down. In order to avoid this, several possible adaptions were assessed. Out of these, an internal heat exchanger between the liquid outlet from the intercooler and the suction line and electric preheating seemed to be most feasible. If condensation could be avoided, the heat pump set up would be able to deliver secondary regulation power, i.e. response time below five minutes.

6. ACKNOWLEDGEMENT

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