13th IEA Heat Pump Conference
April 26-29, 2021 Jeju, Korea

Medium capacity low charge ammonia chiller and heat pump.

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Abstract

Large-scale ammonia refrigeration and heating systems have been widely used in industry due to their high efficiency, broad temperature ranges of operation and innocuous environmental effects. However, implementation of this technology in small to medium systems has not yet been very successful. Factors like the toxicity of the refrigerant and prices of ammonia equipment are some of the obstacles that this technology must overcome before being able to enter a market dominated by systems running on HFC refrigerants. This paper presents a concise review of the research and development of ammonia systems designed to supply small to medium capacity applications. Based on this review, a low charge ammonia chiller and heat pump unit intended to supply both needs providing 180 kW of cooling and 200 kW of heating was proposed. A market overview was performed to find the available equipment that can guarantee proper operation with the lowest possible refrigerant charge. Furthermore, a numerical study of three concepts for this system was performed to study the performance, as well as refrigerant charge estimation and distribution within the equipment. The analysis laid the foundation to propose a suitable configuration to embrace the intended market segment not only considering the high coefficient of performance but also specific refrigerant charge.

Keywords: Low charge, Ammonia, Chiller, Heat Pump;

1. Introduction

The latest regulations promoting the phase-out of HFC refrigerants e.g. The Kigali Amendment [1], are forcing the refrigeration and heating industry to develop new systems that make use of refrigerants with low or preferably zero GWP. Thus, systems with natural refrigerants like ammonia are a suitable option to keep supplying cooling and heating demands. Ammonia has proven to be an efficient and environmentally harmless refrigerant. That is why industrial-sized refrigeration systems have used ammonia since the beginning of the refrigeration industry. Besides, district energy applications have increased the use of ammonia systems in the last decades. Thus, in large-scale systems, shifting from HFC towards ammonia can prove to be a viable solution for manufacturers and industrial consumers. However, toxicity is an issue that has excluded this refrigerant from smaller scale refrigeration and heating applications for many years. Consequently, the re-introduction of small (≈10 kW) to medium-sized (≈200 kW) ammonia systems as a replacement for HFC systems, would require the development of new methods to prevent hazards from leakages. Hence, as presented by Ciconkov [2], researchers and companies have recognised that one main improvement to keep embracing the use of ammonia systems is the reduction of refrigerant charge.

Regardless of the system's application, most of the studies and developments to minimize charge or improve performance and security during operation are for large ammonia systems and, to a low extent, studies for small-scale ammonia systems. However, the optimization of medium capacity ammonia systems has not received much attention. Moreover, there are few reference cases about the construction or performance of this kind of equipment on the market. The present paper will present a concise review of the current research to minimize charge in ammonia systems. This acquired knowledge allows the introduction of a medium-capacity chiller-heat pump unit concept. A numerical model of the system will show not only the performance but also the specific charge of the system when using the equipment available in the market.

1.1. Review of Ammonia Chiller and Heat Pump Research

The application of ammonia cooling and heat pump systems has been a topic of interest in the last decades. Stoecker [3] introduced expanded applications for ammonia within the refrigeration industry. The author presented water chiller and ice thermal storage applications that, at the time, were considered as prototypes of what is now recognised as conventional systems. Furthermore, the effects of ammonia leakages from the systems was also a topic of concern in the study. Thus, the author presented methods and numerical models to analyse the consequences of releasing ammonia into the atmosphere. Similarly, Korfitsen [4] showed the suitability of applying ammonia heat pumps in several food refrigeration applications. Not only they presented case studies to assess the advantages of ammonia over other refrigerant systems but also, they are the ones who introduced the high-pressure/temperature (40 bar) ammonia heat pump concept, which nowadays is a standard operation mode in high-temperature heat pump applications. The market and technology development have increased the acceptance of large scale ammonia systems in the last decades, as a matter of fact, Skačanová et al. [5] showed that by 2017, more than 90% of the large refrigeration installations in Europe use ammonia as working fluid. Besides, Skačanová et al. discussed the substantial increment of low charge ammonia systems in industrial applications and regions. Low charge systems have also received attention from researchers mainly related to industrial applications. Chapp [6] and Poggi [7] describe the possibilities of charge reduction with emphasis on industrial refrigeration systems and the effect of charge minimization in each component. Likewise, Pearson [8] explained the advantages and disadvantages of charge reduction in industrial ammonia systems. These three studies [6]–[8] concluded that industrial systems tend to accumulate substantial amounts of refrigerant within the liquid lines. Furthermore, inappropriate charge reduction could have a counterproductive effect on the operation and performance of the system. From a component-oriented perspective, Hrnjak [9] demonstrated that ammonia has the best potential for being used in low charge systems if a rational approach to minimize charge within the components is applied. Similarly, Ayub [10] showed the reduction of charge and price of an industrial heat pump through optimizing the selection of the evaporator.

On the other hand, low charge small ammonia systems have also received attention from researchers. Palm [11] proposed a 9 kW ammonia heat pump with a specific charge of 11 g/kW. The author also performed a market analysis to spot different components that can be embedded in this kind of systems concluding that there is a lack of proper components for small ammonia systems. Similarly, Hrnjak [12] proposed an air to water ammonia chiller concept with a 13 kW capacity and specific charge of 18 g/kW. Hrnjak also compared the specific refrigerant charge of the prototype with different chillers available in the market to show how commercial systems are still not optimized in terms of refrigerant charge. These works have proven that there is plenty of opportunities to reduce refrigerant charge in ammonia systems, thus improving the competitiveness of this refrigerant in the market. Nevertheless, approximately a decade has passed since these last two studies were published and it is still not possible to embrace small ammonia systems and their entrance in the market. For instance, Zajacs [13] introduced a small-scale ammonia heat pump concept for space and water heating. They also recognized that the lack of components for small-scale ammonia systems still is a drawback on introducing them to the market. In the same way, Wu [14] presented a study of the progress in the development of ground source heat pumps with natural refrigerants. In this comprehensive survey, Wu showed that large-scale systems are still embracing the market. However, they were not able to recognise further improvements in small ammonia systems. Among the literature related to medium capacity ammonia chillers or heat pumps, Nielsen [15] modelled the performance of a 150 kW heat pump with different refrigerants. Aside from proving that the heat pump performance is higher when using ammonia as a refrigerant, the author also performed a charge estimation concluding that the ammonia system with a specific charge of 23 g/kW was the one with the lowest refrigerant charge needed to operate. Furthermore, Nelson [16] examined the long-term suitability of medium-sized ammonia chillers. Using a 280 kW unit as a case study, Nelson showed the economic benefits of replacing HFC units with ammonia ones. The two afore-mentioned papers are recent and do not refer to prior studies of suitability or optimization of systems working with these capacities.

Few companies produce medium-capacity ammonia systems. Most of these companies do not have a complete description of the operating conditions or sizes. On top of that, few companies can provide systems with the so-called "very low" refrigerant charge, ranging 100 g/kW to 200 g/kW. However, the approach used to assure low charge seems to be the same, a compact designed system with plate heat exchangers as evaporator and condenser. Moreover, manufacturers agree that the proper implementation of the receiver helps to minimize the amount of ammonia within the system. Based on the performed review related to ammonia chiller and heat pumps, the current work presents a concept that could be a suitable option to supply medium-capacity cooling and heating applications. The inspiration for the current concept comes from a real case study in which there is a need to provide cooling for different industrial processes of a mineral wool manufacturer plant. On the other hand, the heat pump mode will serve a district heating network of a municipality located in the vicinities of the plant. The concept proposes a combined ammonia chiller and heat pump with a nominal cooling capacity of 180 kW and a heating capacity of 200 kW.

2. Methods

2.1. Medium Capacity chiller-heat pump concept

The rating test conditions for process chillers of the European standard EN 14511:2018 [17] was used as an inspiration to define the operating conditions of the evaporator and condenser when the system is working in chiller mode. However, the current standards developed for heat pumps are focused on space heating applications, which are not the same as district heating applications. Thus, the specifications of heat pumps available in the market that have been used in district heating applications were used as an inspiration to define the operating conditions of the condenser when the system is working in heat pump mode. Moreover, in the design of the evaporator, the temperature difference is maintained in both chiller and heat pump modes. Thus, when the system is working in heat pump mode, the temperature difference in the evaporator is maintained as the one from the chiller mode operation. Finally, the chosen evaporator and condensation temperatures entail the same design pressure ratio in both chiller and heat pump mode. Table 1 compiles the design operating conditions of the proposed concept.

Table 1. Operating conditions of the Chiller-heat pump concept

	Chiller		Heat Pump	
Data	Value	Units	Value	Units
Capacity	180	kW	200	kW
Evaporation Temperature	0	°C	30	°C
Temperature of inlet/outlet water in evaporator	12/7	°C	42/37	°C
Condensing Temperature	40	°C	80	°C
Temperature of inlet/outlet water in condenser	30/35	°C	65/75	°C

With the operating conditions defined, the three configurations presented in figure 1 were analysed to quantify the difference in performance and refrigerant charge. First, a direct expansion (DX) configuration (figure 1a) was studied to define the minimum charge needed by the system to operate. However, there will be a difference in the amount of refrigerant needed by the cooling and heating modes. Consequently, a receiver is required to store the excess refrigerant [18]. That is why the second configuration analysed was the DX system with the addition of a high-pressure receiver (figure 1b). Finally, Pearson [8] proposed a suitable configuration to minimize refrigerant charge in industrial ammonia systems. Thus, using this concept as inspiration, a flooded evaporation system with a low-pressure receiver was analysed (figure 1c). The analysis of these three systems required additional operating conditions to be defined. For the DX systems, superheating of 10K at the outlet of the evaporator was assumed to guarantee stable operation whereas, subcooling is not considered.

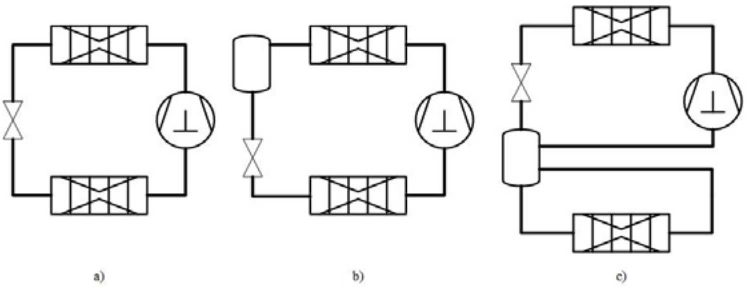

Fig. 1. Proposed configurations to study charge distribution. a) Direct Expansion, b) Direct expansion with high-pressure receiver, c) Flooded evaporator with low-pressure receiver.

The main components in the system were selected based on the market availability of equipment for ammonia systems. In the case of the compressor, it is necessary to find a compressor that fulfils the operating conditions of both chiller and heat pump modes. The considered reciprocating compressors available in the market and developed for ammonia chiller applications were not able to operate with the current cooling capacity proposed for the system. Thus, an open screw compressor seemed to be the only option for the current application. However, according to the manufacturers, these compressors can only operate under the chiller operating conditions, not the heat pump conditions.

On the other hand, suppliers of ammonia heat pump compressors also offer open screw compressors that can be used in the current heat pump application. Nevertheless, they do not assure that the compressors could operate within the chiller mode operating conditions. In the current concept, the pressure ratio of both chiller and heat pump operation modes are similar. Thus, it is assumed that if a screw compressor can operate in chiller mode, it would also be possible for it to operate in heat pump mode. Finally, at the current condensation temperatures, the discharge temperature is high. Consequently, manufacturers restrict the discharge temperatures to a lower value with the addition of oil cooling systems to the compressor. In this case, the suggested temperatures by the manufacturers of the compressors were used as a reference. The important dimensioning specifications of the compressor are summarized in Table 3.

Table 2. Compressor specifications

Type	Displacement [m^3/h]	RPM	Isentropic Efficiency	Discharge Temperature Chiller mode [°C]	Discharge Temperature Heat pump mode [°C]
Screw Compressor	250	2900 (50Hz)	0.6	80	110

Regarding the evaporator and condenser, plate heat exchangers were analysed. The supplier used as a reference for the plate heat exchangers was SWEP, which recommends the use of stainless steel plate heat exchanger for the proposed application. The specifications of the chosen heat exchangers are summarized in Table 2. In the case of the evaporator, the same heat exchanger was used in all the configurations, with the difference that the pressure drop will be lower for the flooded evaporator with low-pressure receiver configuration. Moreover, for the flooded evaporator, SWEP also stated that the circulation number for the brazed plate heat exchanger should be around 1.1 (outlet quality of 0.9) which is the value used for the current concept. The condenser is smaller in dimensions when compared with the evaporator. Since ammonia has a high heat transfer coefficient during condensation, the condenser needs smaller heat transfer areas than the evaporator.

Table 3. Plate heat exchangers specifications

Heat Exchanger	Heat Load [kW]	Heat Transfer Area [m^2]	# Plates	# Channels	Volume [dm^3]
Evaporator	180	11.9	94	93	7.9
Condenser	200	3.9	100	99	4

2.2. Numerical Model

The analysis was based on a numerical model of the chiller-heat pump unit to study the thermodynamic cycle for the operating conditions presented in Table 1. This calculation yielded an estimation of the expected coefficient of performance for both chiller and heat pump modes. The model was implemented in Matlab R2018b making use of ammonia properties from REFPROP 10. The compressor used in this concept is a generic proposal based on different specifications from manufacturers. That is why it was not possible to obtain performance curves given as polynomials. A simple model was used for the compressor in which the assumed isentropic efficiency is used to estimate the difference between an ideal compression and a real compression process. Moreover, the heat loss from the compressor was also calculated based on the assumed discharge temperatures and the theoretical discharge temperature. Finally, the amount of refrigerant in the compressor was calculated based on the method proposed by Poggi [7] making use of the volumetric displacement.

For the evaporator and condenser, detailed models were implemented for analysing component performance and refrigerant inventory. The detailed models accounted for the specifications provided by the manufacturers of each component. The models utilized a 1D-discretization approach with a Newton-Raphson solver in which the heat exchanger was divided into 100 control volumes of equidistant enthalpy steps during the phase change of ammonia through the plate heat exchangers. Mass and energy balances as well as heat transfer equations were formulated for each control volume. Calculation of the heat transfer coefficient for two-phase flow was based on the correlations from Amalfi [19] for the evaporator and Yan [20] for the condenser. In case of single phase heat transfer coefficient, the correlation of Longo [21] was used for ammonia and Kim [22] for water. The models also computed the pressure drop along the plates based on the correlations for the fanning friction factor of Amalfi [19] in case of the evaporator and Arman [23] for the condenser. For the single phase friction factor, Martin [24] was used for ammonia and Kim [22] for the water. The calculation of refrigerant mass inside the evaporator and condenser needs the implementation of a void fraction model. However, there is no model developed for ammonia within plate heat exchangers. In this case, aside from the homogeneous model, the void fraction models of Zivi, Smith, Rouhani-Axelsson, Premoli, Hughmark, Yashar and Boher, presented by Godbole [25] and Poggi [7], were selected as suitable models for the current application. The component models for compressor, condenser and evaporator were coupled to simulate the operating conditions of the units when working in the chiller and heat pump mode as well as to simulate the dependency of the refrigerant charge between these components.

The dimensioning of the receiver is based on its capacity to store the excess of refrigerant during a change of operation mode. However, in the current study, we were mainly interested in the amount of refrigerant needed to act as a seal between the incoming and outgoing refrigerant streams at design conditions. Following a conservative approach, Rajapaksha [18] recommends that the amount of refrigerant in the receiver has to be one-sixth of the charge needed by the system in design conditions. Finally, the piping between the different components was also considered as part of the charge estimation. In this case, a fixed pipe length of 1 m was assumed for suction and discharge pipes of the compressor and 0.5 m for the liquid lines. However, the pipes that connect the low-pressure receiver with the flooded evaporator were dimensioned to have a suitable driving force to run the evaporator with thermosiphon effect. Moreover, standard pipe diameters for ammonia systems were used considering that the flow velocities of the refrigerant would not exceed the values recommended for ammonia systems by Granryd [26].

3. Results

The coefficient of performance (COP) of the vapour compression cycle with ideal compression for chiller mode using the temperatures specified in table 1 is 5.6 and likewise 5.6 for the heat pump mode using the temperatures specified for the heat pump mode. However, the use of real components specifications and the different configurations will affect this ideal COP. For instance, as can be seen in figure 2, the operating cycle for the DX systems (dashed red line) for both chiller and heat pump, varies considerably from the ideal compression cycle (blue line).

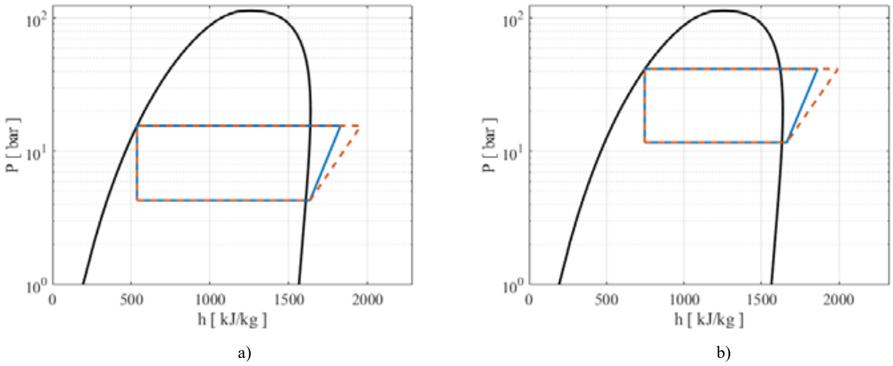

Fig. 2. Log-ph diagram of the Ideal compression cycle and the DX cycle configuration for: a) Chiller operating mode, b) Heat pump operating mode.

For each system configuration presented in figure 1, the mass calculation with the eight void fraction models was applied. Table 4 shows the COP and the average total charge for chiller and heat pump modes and, for the different configurations.

Table 4: COP and refrigerant charge of the systems

	Chiller			Heat Pump		
	DX	DX-HPR	Flooded-LPR	DX	DX-HPR	Flooded-LPR
COP	3.4	3.4	3.5	3.0	3.0	3.2
Average Total Charge [kg]	1.0	1.2	2.2	1.2	1.5	2.7

A quantitative comparison of refrigerant charge presented in Figure 3 for the chiller and Figure 4 for the heat pump, shows the estimation in each component as well as the total refrigerant charge for the three proposed configurations. As can be seen, the refrigerant charge in the compressor is negligible when compared to the total charge of the systems. On the other hand, there is a significant amount of refrigerant that the system accumulates inside the pipes that connect the condenser, expansion valve and evaporator, where the refrigerant is mainly in the liquid phase. Thus, the distance, as well as diameters of these pipes, should be optimized to accumulate the least possible amount of liquid. In the case of evaporator and condenser, the mass distribution shows that most of the refrigerant charge is located inside the evaporator. On the contrary, the charge inside the condenser remains steady despite the type of system. The results for the DX system with high-pressure receiver showed that the addition of the receiver imposes and increment of refrigerant charge due to the liquid level needed at the bottom of it. Besides, more pipe is needed to connect the receiver with the condenser and evaporator. Hence, the share of liquid refrigerant accumulated inside the pipes increases. The flooded configuration presents a different tendency in the charge distribution compared with the previous configurations. In this case, the largest share of refrigerant is still inside the evaporator. However, the share of the pipes slightly increased mainly due to the addition of more pipe circuits to connect the low-pressure receiver.

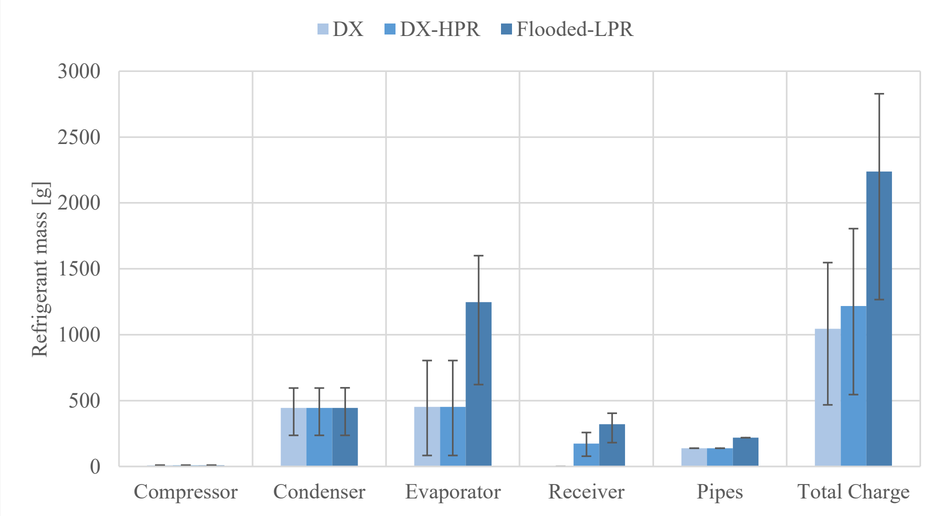

Fig. 3. Refrigerant inventory of the proposed configurations working as a Chiller

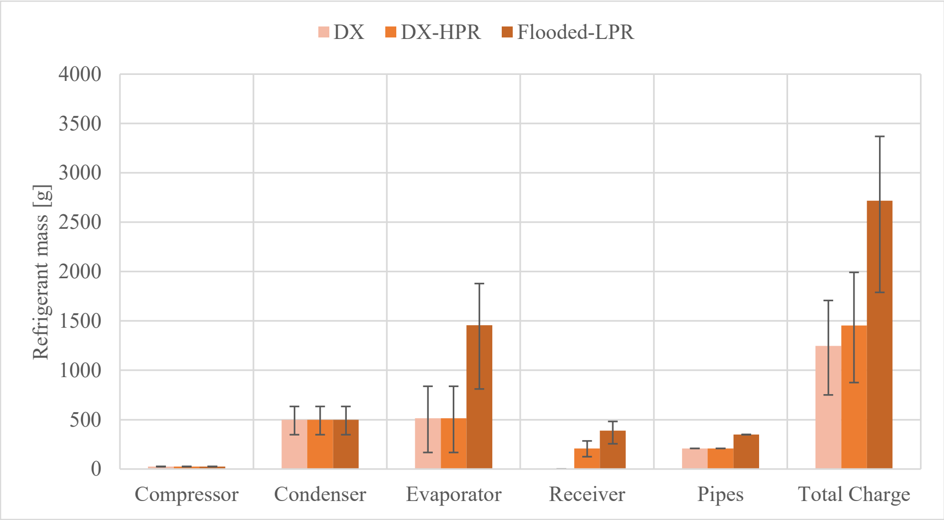

Fig. 4. Refrigerant inventory of the proposed configurations working as a Heat Pump

The error bars shown in Figure 3 and Figure 4, represent the variation that can be expected in the refrigerant charge estimation based on the different void fraction models. To properly depict this dependency on the calculations, Figure 5 shows the eight void fraction models used for a saturation temperature of 0 °C and a saturation temperature of 80 °C. When increasing operating saturation temperature of ammonia, the void fraction models tend to estimate a lower fraction of the heat exchanger being occupied by the vapour phase. Thus, a higher share of the liquid phase is present inside the heat exchangers. Even though between the saturation temperature of 0 °C and saturation temperature of 80 °C, there is a reduction of 20% in liquid density, there is more than 800% increment in vapour density. Therefore, a larger refrigerant charge estimation would be obtained when the concept is operating as a heat pump.

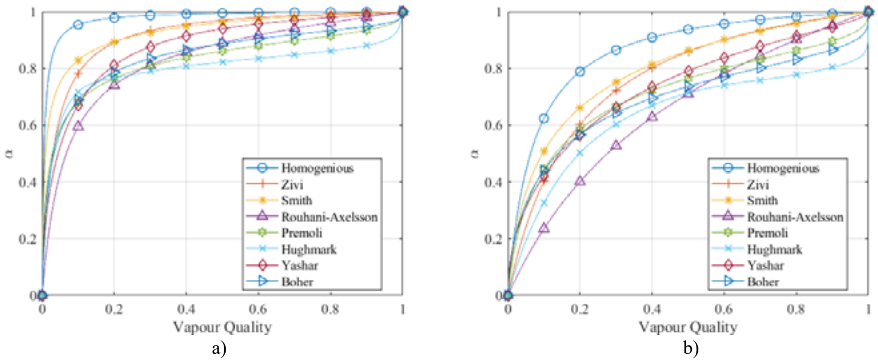

Fig. 5. Void fraction models for Ammonia at: a) Evaporation temperature of 0 °C, b) Condensation temperature of 80 °C.

For each system configuration presented in figure 1, the minimum, average and maximum specific refrigerant charge is presented in table 5. Moreover, the table shows the values of specific charge presented in the introduction to verify the current results.

Table 5: Specific refrigerant charge comparison

	Chiller			Heat Pump			Literature			
	DX	DX-HPR	Flooded-LPR	DX	DX-HPR	Flooded-LPR	Palm [11]	Hrnjak [12]	Nielsen [15]	Industry
Minimum Specific Charge [g/kW]	2.6	3.0	7.0	4.2	5.0	10.0				
Average Specific Charge [g/kW]	5.8	6.8	12.4	7.0	8.0	15.0	11.0	18.0	23.0	100-200
Maximum Specific Charge [g/kW]	8.6	10.0	15.7	9.5	11.0	19.0				

4. Discussion

The COP of the three systems is considerably lower when compared with the Carnot COP. The explanation for this is because of the current selection of equipment for the three configurations. The isentropic efficiency of the compressor plays an important role for the performance of the system. Regarding the calculated refrigerant charge of the three configurations, the average values of specific charge presented in Table 5 are remarkably low when compared with the refrigerant charge of the systems designed by Palm [11], Hrnjak [12], Nielsen [15] or the available systems in the market. The amount of refrigerant charge calculated in this study has a direct dependency on the void fraction model used to estimate the refrigerant charge inside the plate heat exchangers. The results of the simulations showed that the homogeneous model yields the lower estimation for both evaporator and condenser in either chiller or heat pump mode. The higher estimation for evaporator and condenser of the DX systems, in either chiller or heat pump mode, was obtained with the Hughmark model. However, for the flooded evaporator, the highest charge estimation was obtained with Rouhani-Axelsson. Moreover, among the three proposed configurations, the major variation in the refrigerant charge estimation is on the evaporator. Nonetheless, if the values of maximum specific charge are considered as valid, the specific charge will be within the same range as the results presented by Palm [11] or Hrnjak [12].

The selection of larger evaporator and condensers will substantially increase the charge accumulated inside of them. Besides, the pressure drop through these components will decrease and the COP of the system will increase. Hence, a further study to analyse the trade off between the increment in COP and the increment in refrigerant mass will define the optimum size of the system. The amount of refrigerant accumulated in the receiver is also an assumption that affects the charge estimation of the proposed systems. If a higher value for the excess refrigerant in the receiver was assumed, a higher refrigerant inventory would have been estimated. However, the receiver needs further analysis to account for its influence on the performance of the system. In a more detailed study, simulations of the system at different operating conditions are essential to estimate the difference in refrigerant mass needed for the components. Thus, a more accurate value of refrigerant mass within the receiver can be determined. Finally, the consideration of more components that are normally part of these type of systems like liquid or oil separators will improve the estimation of the refrigerant charge. Furthermore, the analysis of more configurations would help to improve the knowledge of the charge distribution of medium capacity ammonia chiller and heat pumps.

5. Conclusion

Medium capacity ammonia systems still need to be developed and optimized to keep improving components as well as the portfolio of ammonia systems within the market. Furthermore, it will facilitate the inclusion and development of small ammonia systems. Three concepts of a medium-capacity low charge ammonia chiller and heat pump able to provide 180 kW of cooling capacity and 200 kW of heating capacity were analysed. The calculation showed that the average amount of refrigerant needed for the system to operate either in cooling or heating mode is 1.2 kg for a DX system, 1.5 kg for a DX with a high-pressure receiver and 2.7 kg for a flooded system with a low-pressure receiver. Even though the flooded system will tend to accumulate a higher amount of liquid refrigerant than the DX system, it is still possible to conceive a flooded system with a low refrigerant charge. The main conclusion from the refrigerant inventory calculation is that industrial or commercial ammonia systems can still be optimized to reduce the amount of refrigerant. Nonetheless, the study of more complex configurations will extend the knowledge and development of low charge ammonia systems.

Acknowledgements

This paper presents part of the work developed for the CLAUS project (64018-0100) partially funded by the EUDP Energy development and demonstration program of the Danish Energy Agency.

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