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ELIMINATION OF WET RETURN LINES IN AMMONIA SYSTEMS

High-efficiency systems for energy reduction in industrial refrigeration plants with ammonia

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Preface

The project "High-efficiency systems for energy reduction in industrial refrigeration plants with ammonia" was carried out from May 2008 till December 2010.

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I would like to thank the participants and their companies for the support to the project and for bringing in new ideas to the project.

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1. Abstract

The objective of this project is to reduce the energy consumption in industrial refrigeration and freezing systems that use ammonia as refrigerant by removing the pressure loss in the wet return from flooded evaporators.

The objective is to reduce the energy consumption by 10% compared to existing systems corresponding to energy savings of 30 GWh/year in Denmark alone. The project aims at industrial refrigeration and freezing systems as the project ensures low energy consumption and the technology can be used in new as well as existing systems.

Project objectives:

- Determine future ammonia low-temperature plant design
- Reduce energy consumption
- Increase evaporator efficiency
- Reduce pressure loss when eliminating the wet return line

Ammonia properties in low-temperature applications:

- 0.1 bar pressure loss in the suction line corresponds to 2°C at $t_e = -35^\circ\text{C}$
- 2°C improvement on the suction temperature gives a COP improvement of app. 8% for a one-stage ammonia compressor.

2. Introduction

In large industrial refrigeration systems, safety and practical considerations usually dictate that refrigerant piping should be located on the roof of the building. However, as most equipment runs with a liquid overfeed system, the return flow is a two-phase flow. Lifting a two-phase flow requires careful attention to the piping design to avoid static or dynamic pressure penalties that result in a reduced refrigeration capacity and/or increased power consumption. Whatever the situation is, the performance of the system is less than optimal which always gives a financial penalty, either in the system output or in increased electrical compression costs. The height difference between the flooded evaporator and the pump separator/liquid separator can be up to 6 m which means that the pressure in the pump separator will be lower than in the evaporator. The pressure difference results in ammonia evaporating at a higher temperature than the corresponding temperature in the separator. A return line from the evaporator to the pump separator (dimensioned according to the existing standards) will have a pressure loss of 0.3 bar corresponding to a temperature difference of 6.5°C at an evaporation temperature of -40°C.

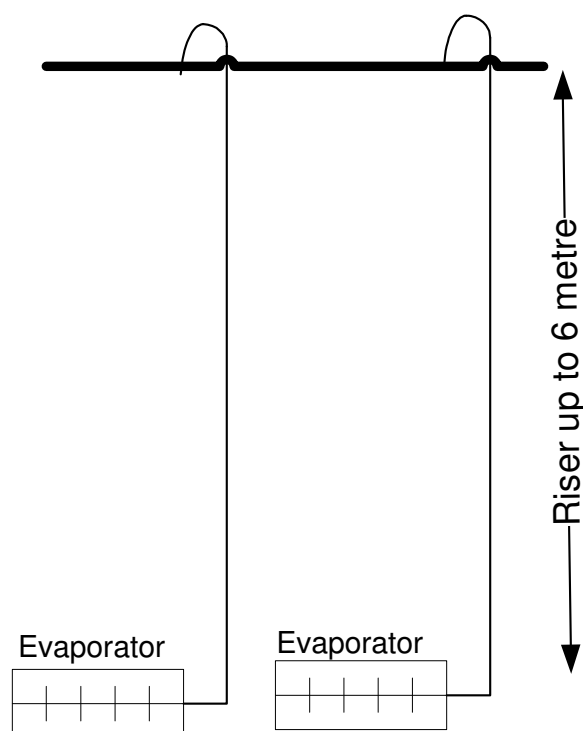


Figure 1: Riser in an industrial plant.

2.1 Project content and brief description

Phase 1: Theoretical evaluation of the selected evaporators

In the project definition it was defined that three different evaporators should be evaluated and tested in the course of this project. It was a plate heat exchanger, a plate freezer and an air cooler. The evaporators tested in this project are the newest type on the market and the calculations give an optimum performance at full load conditions. In the project, the challenge was to evaluate the evaporators at different conditions varying from the standard conditions.

Phase 2: Dimensioning of the evaporator and auxiliary equipment

The evaporators were dimensioned by the suppliers to fulfil the specification of the laboratory test system. However, the requirements to each individual evaporator differed and to achieve optimum performance of the auxiliary equipment the ejector and the refrigerant pump were designed especially for the plate heat exchanger operation.

Phase 3: Building the test rig

The three evaporators were connected to the existing refrigeration system at Danish Technological Institute. The laboratory unit was built so all 3 evaporators could be operated from the same system and could be isolated individually with full flow ball valves. That gave some limitations in the setup and operation but the controlled circulation test could be achieved for all evaporators and that was the essential part. A consequence of this was that the pressure loss in the wet return was only measured for the plate heat exchanger.

Phase 4: Operating the evaporators

The three evaporators required very different operation setups and therefore the measuring equipment setup was specially designed for each evaporator test. The setup gave a high degree of flexibility and each evaporator test could be improved as knowledge was gained on the operation. The measurements were logged with an interval of 10 seconds over a period of stable operation of 10 to 15 minutes.

Phase 5: Evaluation of the evaporator performance

The data was evaluated and compared to the dimensioning data sheet for each evaporator.

Phase 6: Optimisation of the system

In this phase, focus was on optimisation of the operation and circulation rate control. The setup was adapted as knowledge was gained on the operation.

Phase 7: Selection of the optimum system for each evaporator

The objective of the test was the development of a guideline of how to design each individual evaporator circuit so the wet return can be eliminated. That is described in the chapter for each individual evaporator.

2.2 Challenges with wet return in industrial refrigeration systems

The solution to the vertical two-phase flow problem is referred to as a "riser" and it is quite well-known, but often executed incorrectly. To ensure liquid transport upwards, the velocity must be high enough as the liquid is transported through shear forces on the gas/liquid surface. When the velocity is sufficient, the liquid will flow on the inside surface of the pipe and the gas will flow in the centre of the pipe.

Failing to obtain the correct flow pattern can result in a number of different situations, but all are a result of too large vertical piping. All situations involve penalties. Usually, the vertical pipe is almost completely filled with liquid. The gas will bubble through it, either as small bubbles or as larger plugs. In any case, there is a static pressure difference between the top and bottom of the vertical pipe that imparts a pressure/temperature penalty on the evaporator, reducing its capacity, or on the compressor, lowering its suction pressure - or a combination of both. Another scenario is liquid collecting in a low part of the piping until its volume has increased the gas velocity to a point where the liquid is shot to the top of the vertical piping. Naturally, that involves a very unsteady operation of the plant and should also be avoided.



Elforsk Non riser project

A satisfactory design of a riser involves checking all running conditions of the system. Especially part load needs to be considered as gas flow is proportional to the load and therefore the gas flow is lowest at part load. At the same time, the pressure loss at full load needs to be within acceptable limits to avoid the dynamic penalty.

In some applications (such as a batch freezing process) the load approaches zero at the end of the freezing cycle. A riser designed for this very small load will have an unacceptably high pressure loss at the beginning of the freezing cycle and will most likely make a satisfactory freezing performance impossible. In these cases, the riser can be dimensioned so it will fail at an appropriate time in the cycle where the freezer can perform even with the static penalty. Another solution for a batch process is to have a double riser which has a large riser for the start-up process with high evaporating temperatures and a smaller riser for operation at low temperatures. However, that is a costly as well as a complicated installation.

3 The laboratory test unit

3.1 System description

In order to validate that a refrigeration system built in a more optimised way can increase the energy efficiency to the degree stipulated, a laboratory test unit has been built. The system is designed to operate at an evaporation temperature of -35°C and at a capacity of 50 kW. Three different evaporator types will be installed and tested:

1. A liquid-based cooling system with a plate heat exchanger

The plate heat exchanger will be installed so it can run at self-circulating mode with a traditional wet return into a liquid separator. In self-circulating mode the circulation depends on the pressure drop in the evaporator and wet return and the inlet pressure which is given by the liquid column at the inlet which again depends on the height of the liquid separator. Self-circulating operation is only feasible for the plate heat exchanger and not for the other two evaporators.

2. A batch freezing process with a plate freezer

As the plate freezer freezes the product in batches, the thermal load on the plate is not constant but declines during the freezing period. Therefore, the traditional definition of circulation number as the ratio between the circulated refrigerant mass flow to the gas mass flow out of the freezer falls somewhat short of sufficient. With a constant liquid flow to the plate freezer, the circulation rate will change throughout the entire cycle. Furthermore, the flow fed to the plate freezer cannot be assumed to be constant as the pressure loss through the freezer is highly dependent on the thermal load. The flow rate can be defined either as an average value or as a function that changes over time.



Figure 2: Plate freezer.

3. Air-cooled system with a forced convection air cooler

The air cooler is normally placed in a high position and the regulation system is a simple on/off type where the liquid flow is controlled by a solenoid.

To optimize the system performance for all three evaporators, two different system designs were tested:

- 1) An ejector-driven system, as described in chapter 3.3
- 2) A pump-driven system, as described in chapter 3.4

3.2 Pressure loss in wet return

In low-temperature ammonia systems the pressure loss in the return line to the compressor is significant for the capacity of the compressor and the efficiency of the system. The design of the return line whether wet or dry will have to be carried out to give the minimum pressure loss.

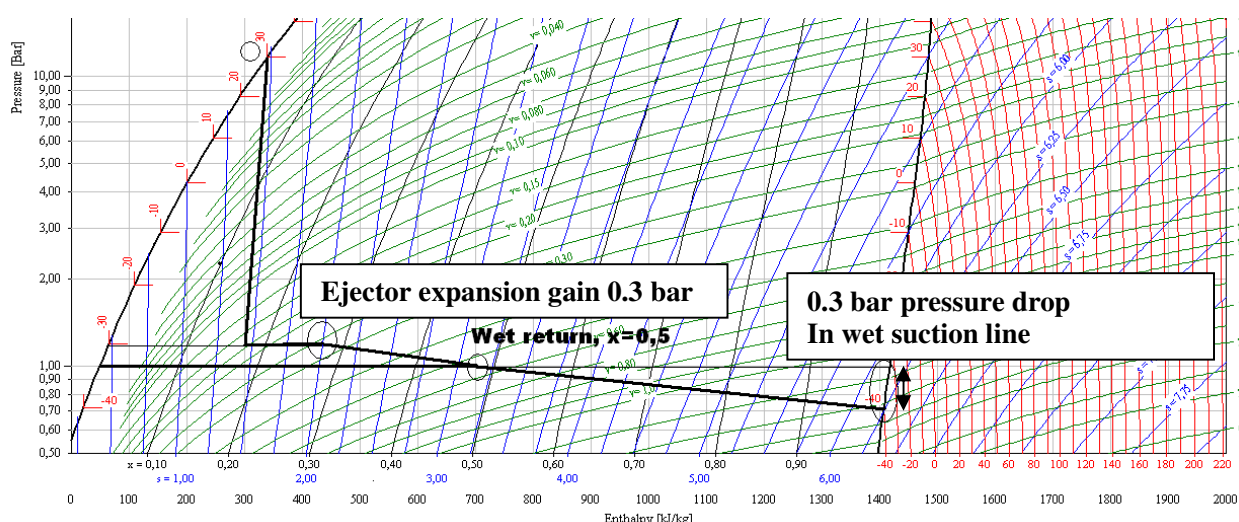


Figure 3: R717 process diagram for ejector and traditional wet return.

Figure 4 shows the influence of the pressure loss on the saturation temperature. In connection with ammonia, a pressure loss of 0.3 bar at an evaporation temperature of -40°C will result in a temperature change of 6.7°C . In connection with CO_2 , the same pressure loss at -40°C will result in a temperature change of 0.8°C . The reason for the large difference between ammonia and CO_2 is the difference in the incline on the saturation curves [Figure 4].

By using a dry return line where there is no demand for liquid carry over it will be possible to make the pressure loss much lower and in that way the compressor can operate at a higher evaporation temperature resulting in smaller energy consumption. It will be possible to reduce the energy consumption by up to 10% by removing a pressure loss of 0.3 bar. At lower evaporation temperatures, the savings potential will be even higher.

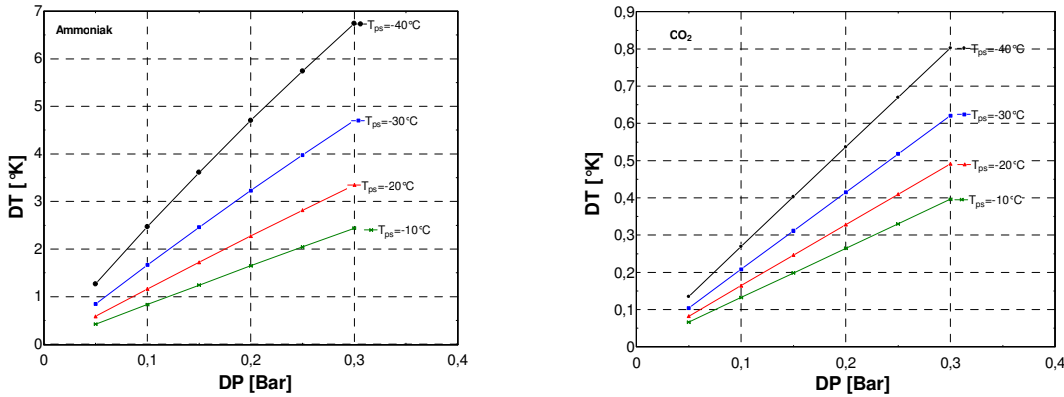


Figure 4: How the pressure loss influences the temperature change of ammonia and CO₂.

3.3 Ejector-driven system

In the return line from the evaporator an ejector will be used. It will be operated by the high-pressure liquid (4) directly from the condenser/high pressure receiver and will be transmitted through a nozzle. The liquid leaves the nozzle with high velocity and little static pressure that sucks refrigerant from the evaporator (2) and mixes it with the high-pressure liquid. The liquid is transmitted through a diffuser and ends at a higher pressure than at the low pressure side of the ejector. By installing an ejector in the wet outlet of the evaporator it will be possible to have a lower pressure in the evaporator than in the separator (3). The ejector shall be mounted on each individual evaporator if several parallel evaporators are connected to the same separator.

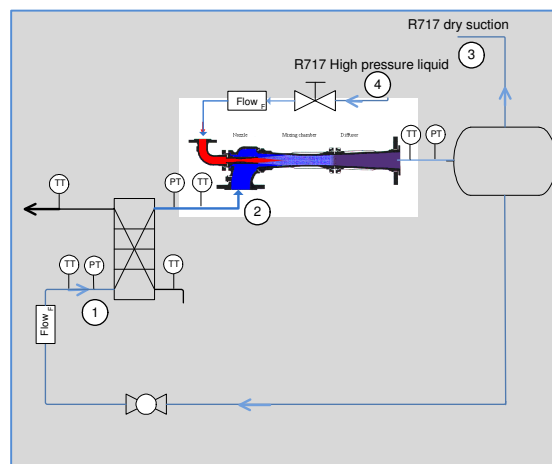


Figure 5: Ejector and plate heat exchanger.

Especially for that purpose, an ejector will be developed in the course of the project, but it is not expected that it will be able to operate with optimum efficiency. The ejector will form part of the systems as a passive component that replaces the expansion valve. It is a newly-developed type especially designed for two-phase expansion where the energy from the high-pressure liquid is used to lower the suction pressure in the evaporator. The ejector is calculated to give a pressure lift of 0.3 bar.

3.4 Pump-driven system

In the pump circulation system a small refrigerant pump is used and it is placed so the liquid flow to the evaporator can be controlled independently of the load. The refrigerant pump shall ensure an optimum circulation number in the evaporator. The pump is a functional model or very early prototype developed by Grundfos for the range of very small circulator pumps that can be placed in each individual evaporator circuit. In principle, that is similar to a modern heating system where small automatic control circulators constantly are controlled so they give the needed flow in the system. In the refrigeration system it means that the circulation rate always will be adjusted to give the optimum heat transfer coefficient for the individual evaporator system.

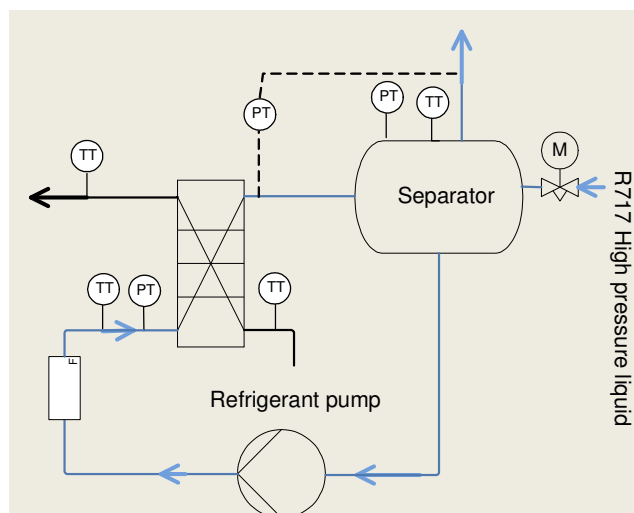


Figure 6: Process diagram for the test unit with plate heat exchanger and CO₂ brine.

The liquid pump has variable speed control and was designed for generation of a flow of 0.24 m³/h at a head of 2.5 m. For ammonia that corresponds to a mass flow of 164 kg/h and a pressure difference of 0.2 bar. The power consumption of the prototype pump is app. 200 W. In the design phase focus was on flow and head performance and not on energy efficiency. The power consumption of the pump has less importance in relation to the overall power consumption of the refrigeration system.

3.5 Construction of test rig and measuring equipment

The three evaporators will be connected to the existing ammonia and CO₂ refrigeration system in the laboratory. Due to limited capacity of the ammonia refrigeration system at low temperature the ammonia rig was connected to an ammonia / CO₂ cascade heat exchanger that is connected to a CO₂ compressor with a rated capacity of 100 kW at -45°C. Measuring equipment will be mounted so it will be possible to measure pressure loss in the wet return from the evaporator. In addition, the refrigerant flow, pressure and temperatures will be measured to determine the capacity at different circulation numbers.

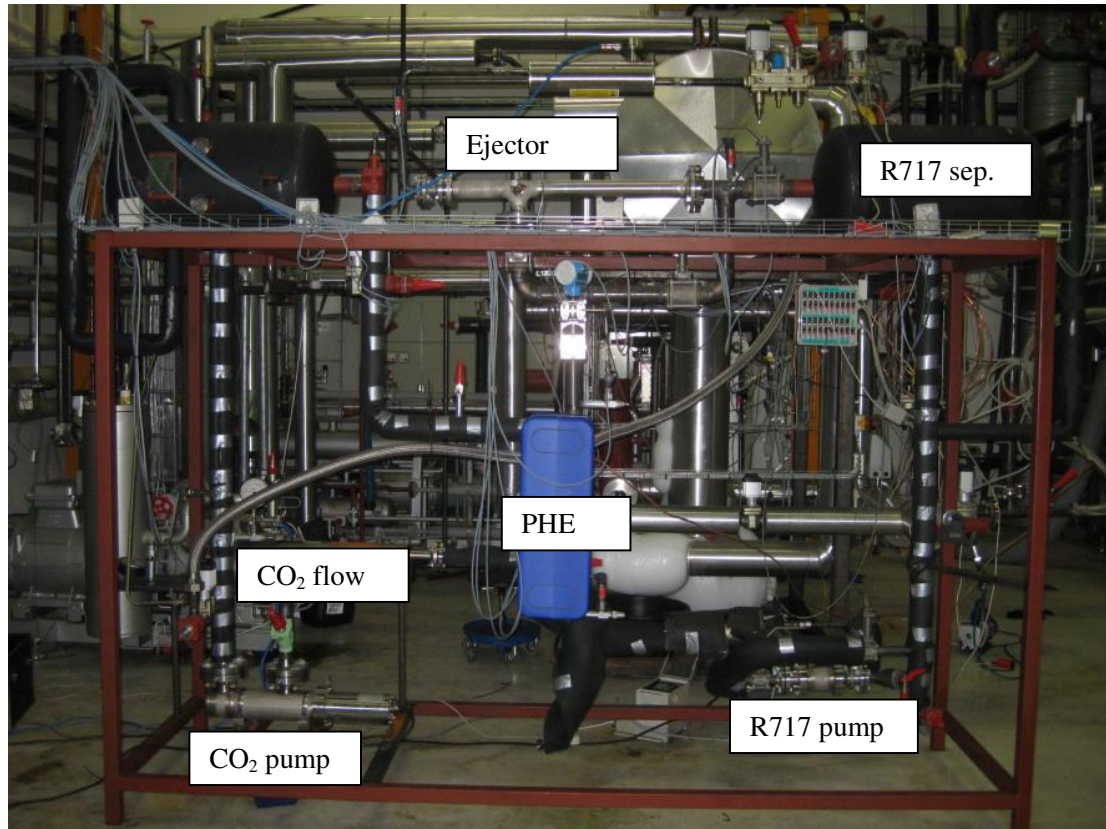


Figure 7: Test unit with plate heat exchanger [PHE].

4. Theory

4.2 Circulation rate control

In an evaporator and condenser the heat transfer coefficient of the boiling/condensing refrigerant depends on the wall temperature and therefore of the heat transfer coefficient and temperature on the other side of the heat exchanger. Boiling is the process where the phase shift from liquid to vapour takes place as bubble formation on a solid surface while the phase shift at evaporation takes place on the liquid-vapour surface.

In most evaporators, the refrigerant is forced through the evaporator by external means and the heat transfer is a combination of boiling and convection. The heat transfer coefficient is highly dependent on the flow pattern, which is dependent on the mass flux and the quality.

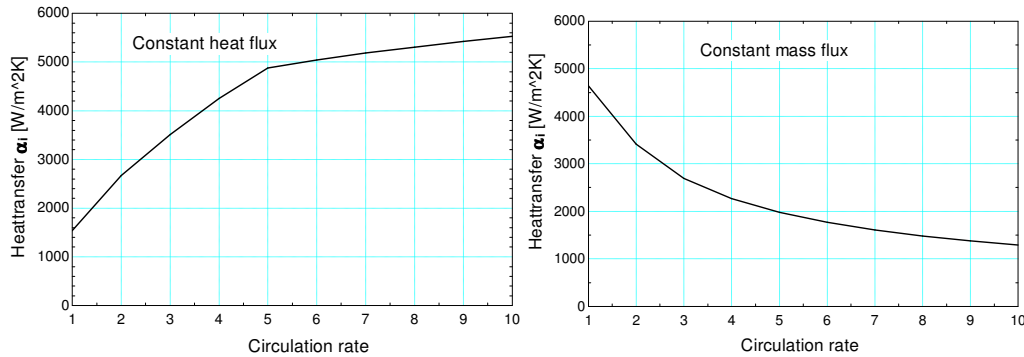


Figure 8: Calculated average heat transfer in a horizontal tube with evaporating ammonia at -35°C.

The figure for the constant heat flux illustrates that a high evaporator output or a low temperature difference can be achieved by increasing the circulation rate. The limit is given when the pressure loss in the evaporator is higher than the gain from the increased mass flow.

The constant mass flux chart illustrates an evaporator where the flow is not controlled during part load meaning the heat transfer will be low and in worst case there is a risk that the compressor suction temperature will be lower at part load than at full load.

By using a pump, the circulation rate can be controlled so optimum heat transfer is achieved for the individual system. In addition, it will be possible to optimise the evaporator design independent of the available pressure from a liquid column.

4.2 How the evaporation temperature influences the ammonia COP

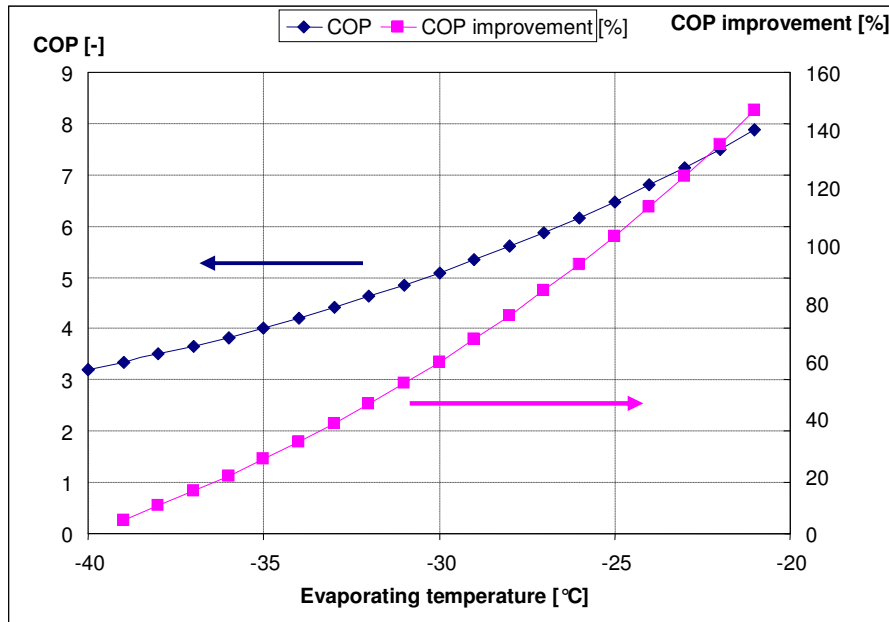


Figure 9: COP improvement in % in relation to -40°C for the screw compressor with ammonia.

Figure 9 shows which COP values can be obtained for a screw compressor at different evaporation temperatures. The calculation was carried out with a maintained condensation temperature of -5°C.

If it is assumed that the pressure in the separator is -40°C and the pressure loss in the wet return between the evaporator and the separator is 0.3 bar (corresponding to a temperature difference of 6.5°C), then COP at -40°C will be 3.2. If the pressure loss in the wet return can be overcome so there is no pressure loss, then COP will be 4.3. That corresponds to a COP improvement of 34%.

5. Test result for the plate heat exchanger (PHE)

The plate heat exchanger that was used in this test was the newest type of brazed plate heat exchanger from Alfa Laval, Alfa Nova HP 76. The heat exchanger is designed for CO₂ brine on the secondary side which is a very efficient heat transfer media. The capacity is measured on the brine side where the flow and temperature are measured. No phase change takes place on the brine side and the CO₂ is circulated as a liquid. The design data are as stated below.

Capacity	50	kW
Evaporation temperature	-35	°C
Brine temperature in / out	-22,5 / -34	°C
Pressure drop ammonia	8880	Pa
Pressure drop brine	7500	Pa
Heat transfer value, fouling	2198	W/m ² K
Heat transfer value, clean	2700	W/m ² K




Figure 10: PHE design data and installation

5.1 PHE test matrix and calculations

For the plate heat exchanger 35 different test situations were tested in 5 different system configurations:

- Full load capacity with pump circulation. The only parameter that varied was the ammonia flow.
- Part load capacity with pump circulation. Same conditions as a) but at a constant part load.
- Load variation with constant mass flux. The brine inlet temperature varied while the ammonia flow was kept constant.

- d. Load variation with self-circulation. The brine temperature varied while all other parameters were kept constant.
- e. Load variation with constant circulation rate (two test setups). The brine temperature varied while the ammonia flow was decreased to maintain a constant circulation rate.

The evaporating temperature was regulated so that the suction temperature in the liquid separator was kept constant. In this case, the evaporating temperature varies with the pressure drop in the system. The system capacity was regulated so the only variable in the system was the brine inlet temperature. The brine flow was kept constant and the capacities varied as a function of the temperature difference on the brine side.

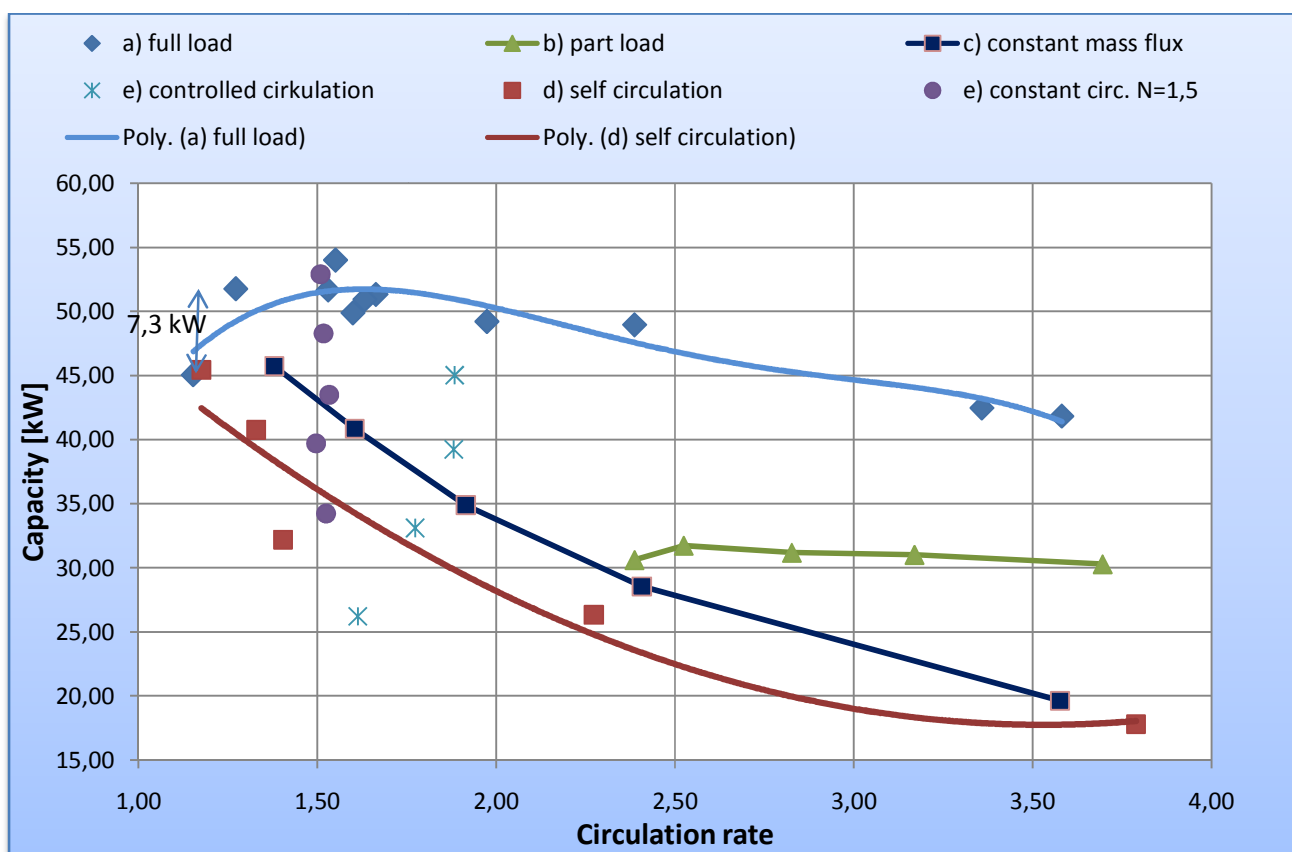


Chart 1: PHE test data for all the load situations.

In the chart above, the set of result are shown for the 5 different system configurations. The chart is only shown to give an overview of the test setup that was done for the PHE.

A quick glance at the chart shows that the capacity of the PHE in full load is 7.3 kW higher at pump circulation a) than with self circulation d). Secondly it is seen that in self- circulation mode it appears that the circulation rate increases at part load (red line) which causes the system pressure drop to be

higher than necessary. When the circulation rate is changed at a constant part load it does not influence the capacity.

The results for load situation a), d), e) will be evaluated further in the following chapters.

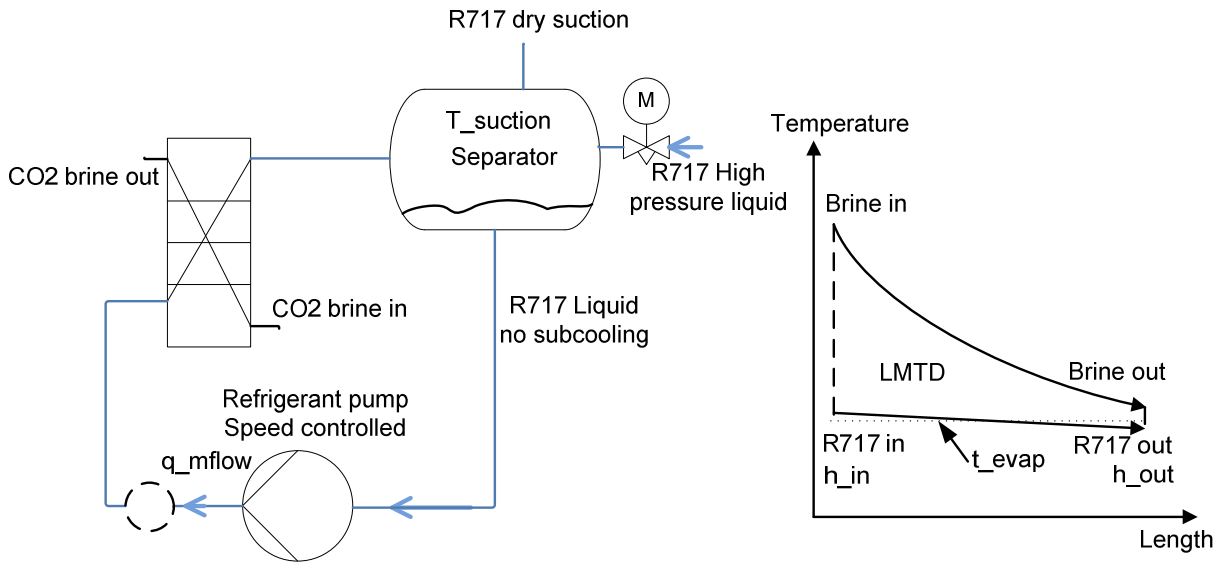


Figure 11: Test setup for PHE with pump circulation

Equations used to calculate circulation rate and heat transfer values

The brine flow and temperatures give the capacity $Q = q_{mbrine} * c_p * (t_{in} - t_{out})$

The capacity is used to calculate the theoretical ammonia flow $qm_{flow} = \frac{Q}{(h_{in} - h_{out})}$

The circulation rate is calculated as ratio of theoretical and measured flow $N_c = \frac{qm_{flow}}{qm_{evap}}$

LMTD is calculated from the brine temperatures and evaporating temperature (p)

$$LMTD = \frac{(t_{ind} - t_{evap}) + (t_{out} - t_{evap})}{\ln\left(\frac{t_{in} - t_{evap}}{t_{out} - t_{evap}}\right)}$$

The heat transfer value is calculated from the capacity and LMTD: $U = \frac{Q}{LMTD * A}$

5.2 Result of full load capacity test, load situation a)

The purpose of this test was to evaluate how the efficiency of the evaporator changes with the circulation rate. In this test the only variable was the ammonia flow and all other parameters were kept constant.

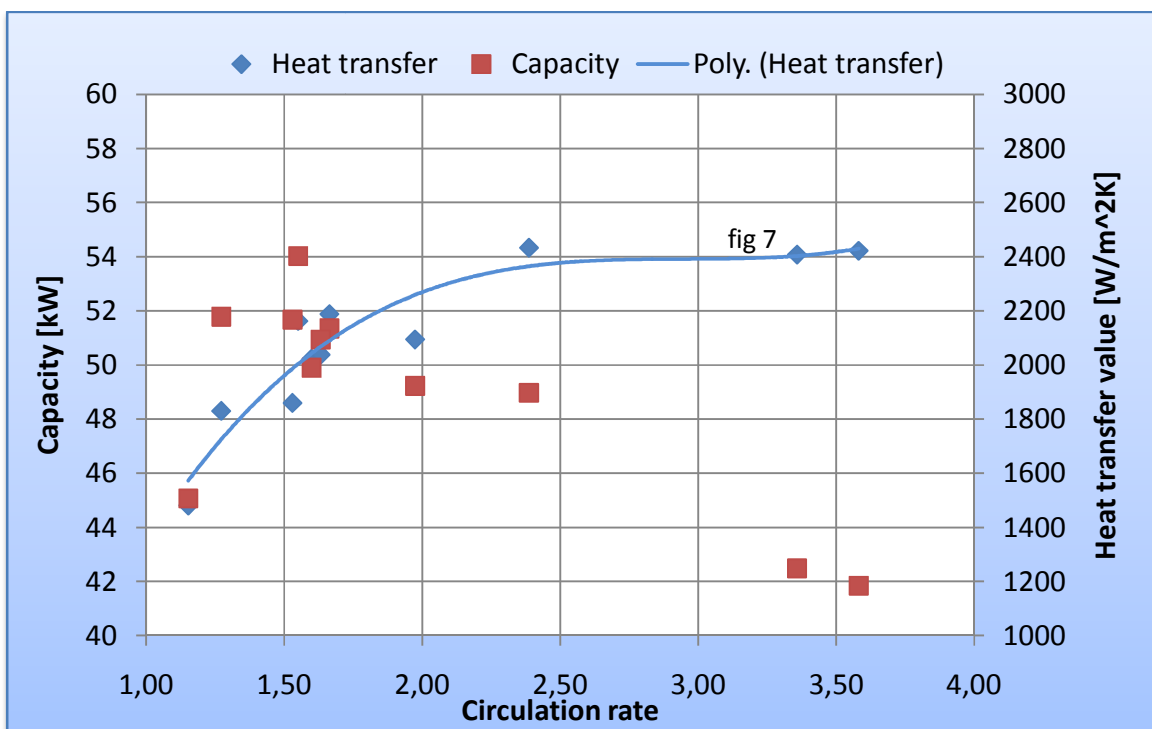


Chart 2: Cooling capacity and heat transfer values for the full load test data.

The results show that there is an optimum circulation rate between 1.2 and 1.6 at full load. The heat exchanger is designed for a circulation rate of 1.25 at full load which is 50 kW. In the test, the heat exchanger delivered 54 kW at a circulation rate of 1.5 so in the test series the constant circulation rate was kept at 1.5. During the test it was discovered that it is difficult to control the evaporator operation at a circulation rate of 1.25. At this low circulation rate, even small changes in the brine flow or temperatures will result in some of the plates running dry or not have efficient boiling to operate optimal and this will cause fluctuations in the system

The heat transfer value reached a maximum at a circulation rate of 2.5. The result can be compared to the theoretical result shown in figure 7 for the constant heat flux. At the lowest circulation rate at n=1.15 the capacity decreases from 52 kW to 47 kW corresponding to 13 %. That is most likely due to

the optimised distribution of refrigerant to various plates and indicates that the PHE must be redesigned if it is to operate with a circulation rate below the design value of 1.25.

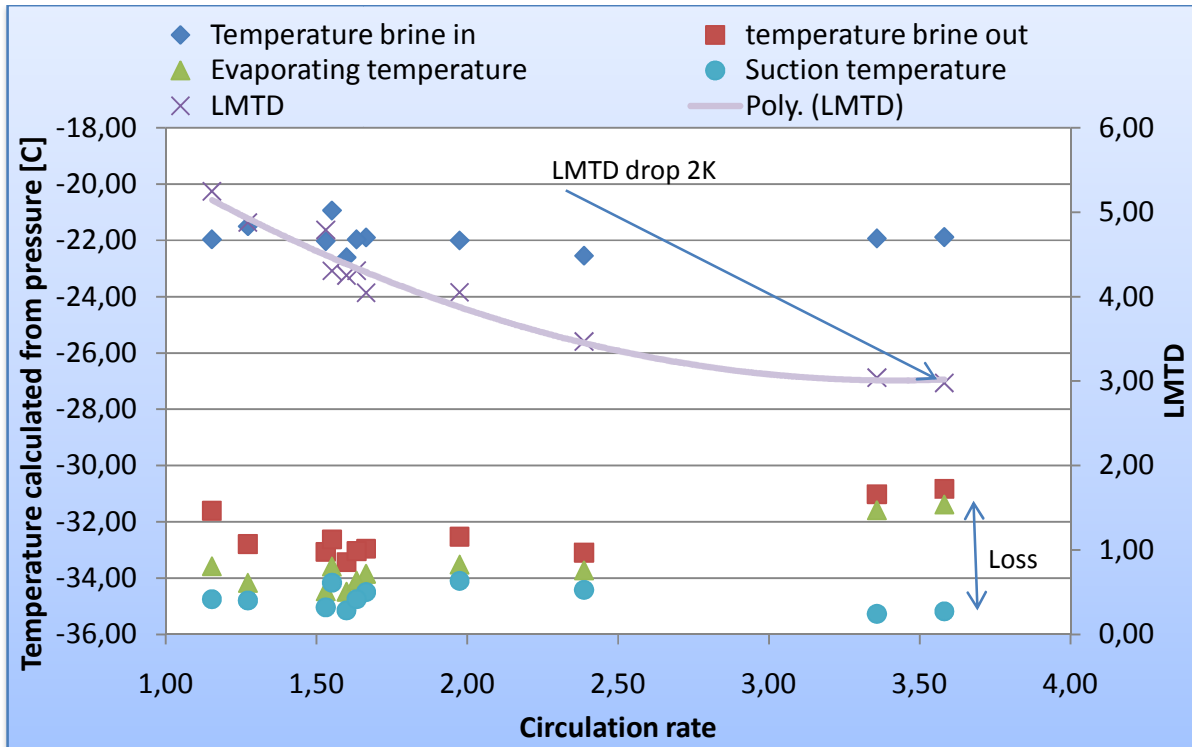


Chart 3: Evaluation of temperatures with varied circulation rate at full load.

The temperatures in the chart above are calculated from the measured pressures. It appears from the temperature chart above that the temperature difference between evaporation and suction temperature increases due to the pressure drop in the wet return. That is the main reason why the LMTD and the capacity decrease with a high circulation rate. The connection between the LMTD and heat transfer values can be seen in the chart below.

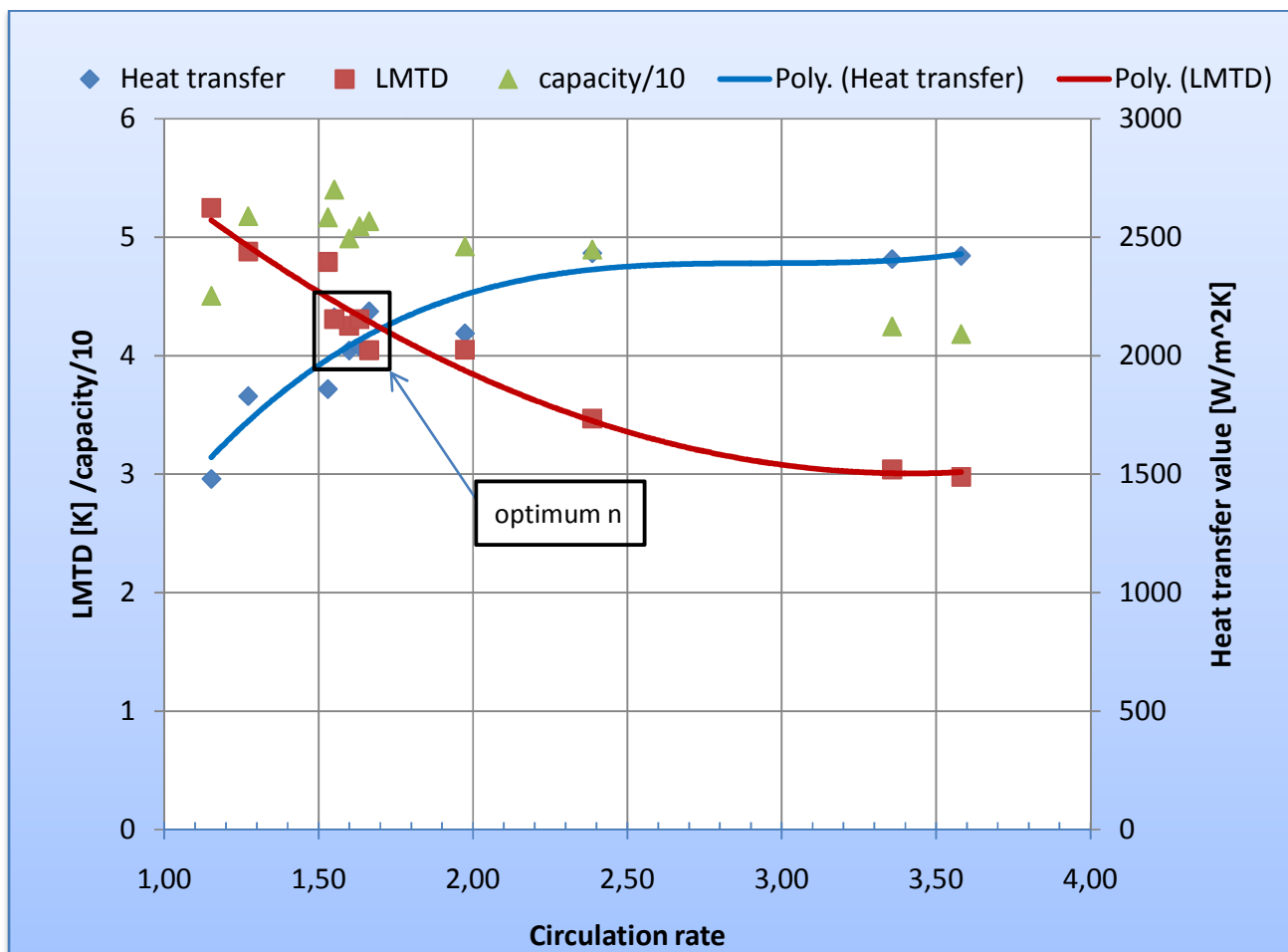


Chart 4: Capacity is a product of LMTD and Heat transfer value.

In chart 4 above, the two factors that influence the capacity are compared in the same chart. The relative capacity (capacity divided by 10) is indicated with green triangles. The LMTD decreases due to the increasing pressure drop in the plate heat exchangers while the heat transfer increases due to the increased flow. The optimum performance of the exchanger is in the area where the two lines cross each other.

5.3 PHE with constant circulation rate versus self-circulation load situation d) and e)

To validate the importance of the circulation rate at part load the values from self-circulation (load situation d) were compared to the values from the pump regulated capacities (load situation e).

When the PHE was run with a constant circulation rate (load situation e) the circulation rate was constantly calculated based on the capacity and the measured flow on the ammonia side.

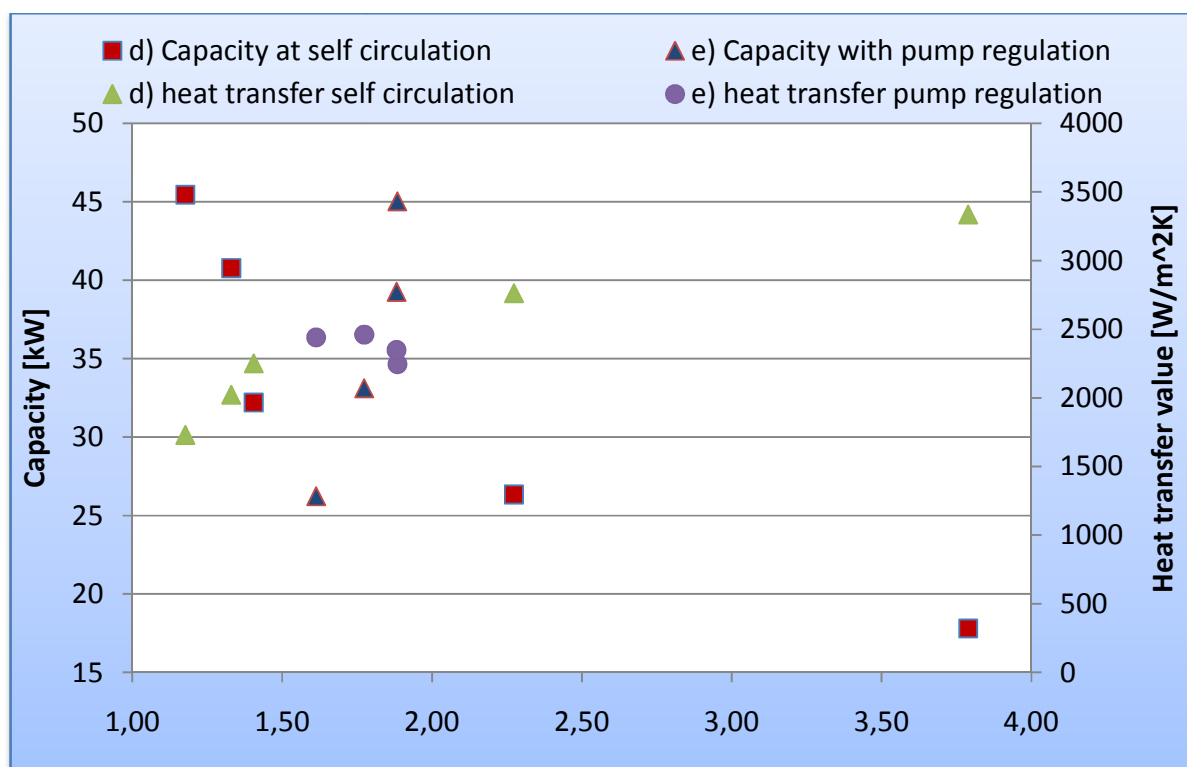


Chart 5: Comparison of the capacity and the heat transfer values for load situation d) and e).

When the test was carried out it was not possible with the existing regulation system to regulate the circulation rate so precisely that it was kept constant. The system regulation must be very accurate because many parameters affect the circulation rate. When system regulation takes place to maintain the various set points the circulation rate is influenced. It appears from the data that the capacity can be reached even at part load with a low circulation rate. If the system was optimised in terms of the regulation system it could be possible to operate the system with even lower circulation rates at part load.

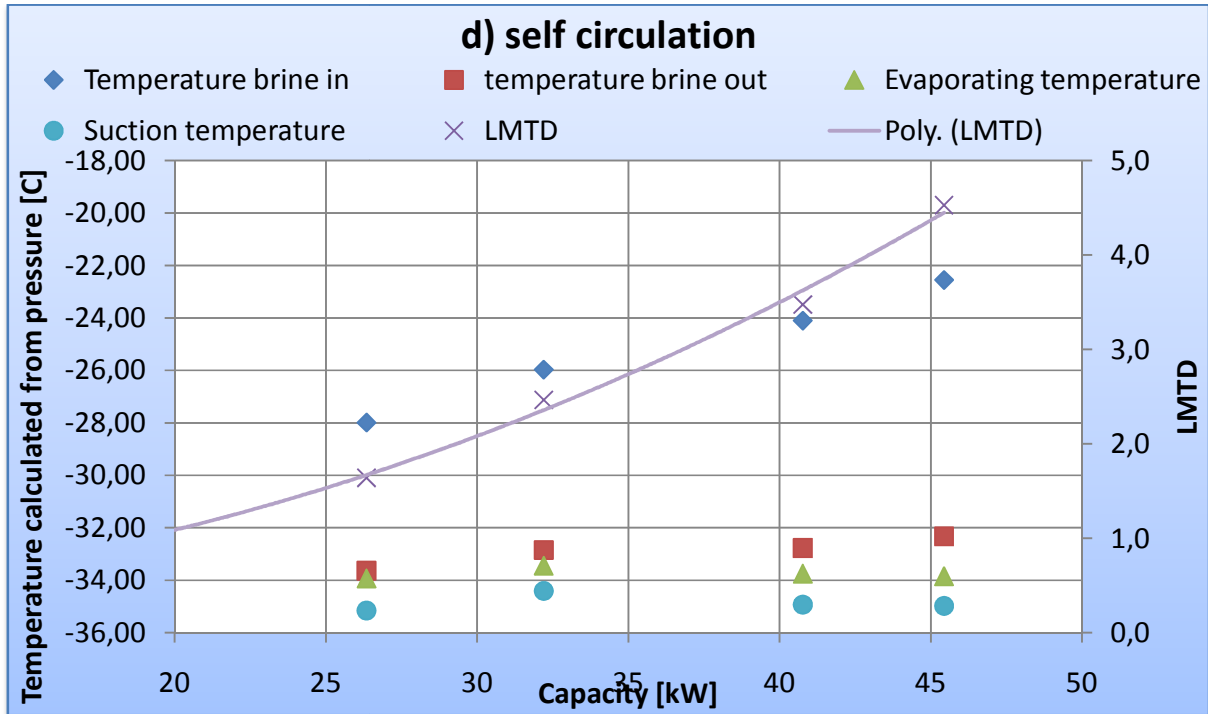


Chart 6: Brine and ammonia temperatures at self-circulation.

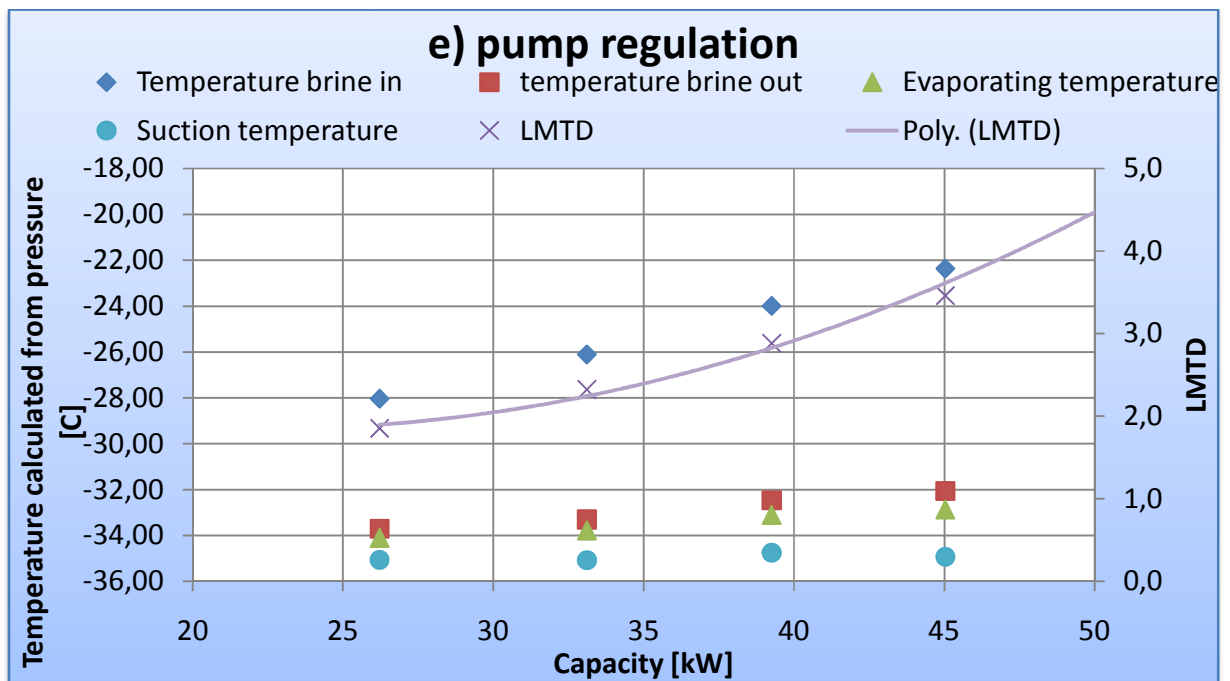


Chart 7: Brine and ammonia temperatures at pump regulation.

In the two above charts, the suction temperature equals the temperature calculated from the pressure in the liquid separator and the evaporator temperature is calculated from the pressure in the outlet of the evaporator.

The LMTD is lower at part load when the flow is regulated with the pump due to a smaller difference between the evaporating temperature and the brine outlet temperature. The PHE clearly performs better when the circulating rate is controlled. At 45 kW capacity the LMTD is 4.5 in self circulation (chart 6) and 3.6 at pump regulation (chart 7).

5.4 Pressure drop in the wet return line and comparison to calculation

When the system was constructed it was decided that the liquid separator should be located so the PHE could run in self-circulating mode. The height of 0.87 m in the wet return made it possible to evaluate the pressure drop in the wet return. The pressure drop in this line was measured with a differential pressure transmitter so that the importance of the circulation rate could be evaluated. The return line is DN50 with 4 x 90° elbows, 0.8 m horizontal pipe and 0.87 m horizontal pipe.

The calculation program used for the calculation of pressure loss was the Danfoss DirCalc program that is widely used in the industrial sector to calculate pressure drop in the system.

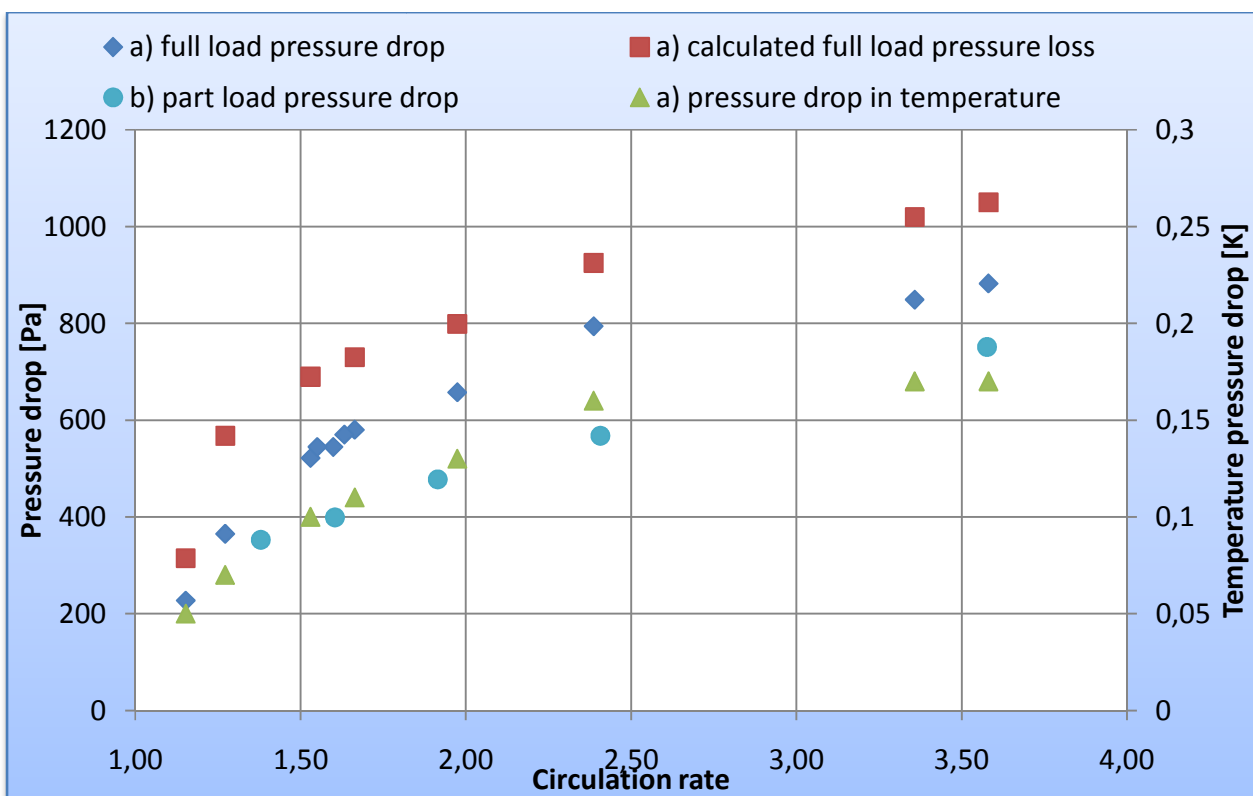


Chart 8: Comparison of calculated and measured pressure loss in the wet return line from the PHE.

It appears that if the circulation rate is kept constant at part load the pressure loss in the system can be minimized. In the test system, the pressure drop is only reduced by 500 Pa. If that value is timed by 7.5 which corresponds to a system tem with a riser of 6 metres the value would be 3750 Pa or 0.038 bar. That corresponds to 0.7°C or app. 3% on the COP.

From the chart it can be seen that there is a good coherence between the measured and calculated values. The calculated values are higher which indicates that the figures given from this program are conservative and gives the designer an extra margin when designing the system. The result verifies that in relation to the two other evaporators this program can be used to calculate the pressure drop in the wet return. Due to this a pressure differential manometer was not installed in the return line from these evaporators.

5.5 Conclusion of the plate heat exchanger

The test of the PHE shows that it has been carefully designed to operate with low circulation rates and therefore the pressure loss is minimized due to excessive liquid in the wet return. The systems with heat exchangers are often designed so that there is a local separation of liquid in a liquid separator which also gives the necessary liquid column to start the self-circulation in the system. However, while designing the plate heat exchanger the pressure drop on the ammonia side was often limited to 7.5 kPa or 1.5 m liquid column due to limitations in the total height of the system.

First of all, when an ammonia pump is used, the ammonia pressure drop is no longer a limitation and there is one less fixed factor when the PHE is designed. In that way, it is possible to design the PHE so it gives a minimum pressure drop on the secondary side and minimizes the required pump power. Secondly, the height of the system can be limited and the unit can be made more compact. The data from this test only gave a 3% saving on the COP by constructing a system without a wet return.

Finally, it appears that the PHE has 7.3 kW more capacity when the flow is regulated with a pump. Besides that there is good potential for minimising the pressure drop in the system. The additional capacity when using a pump cannot be directly transferred to a saving as that requires that the compressor removes the additional capacity. However, it is estimated that if the evaporator was redesigned, a running a dry suction could improve the system performance by 10%.

6 Plate freezer test results

6.1 Plate freezer test method

The plate freezer is a batch process and the test data are handled differently for the other evaporators. Several tests were carried out on the plate freezer but several challenges have to be met to obtain stable conditions during the batch period of app. 1.5 hours. The plate freezer has a considerable higher circulation rate and pressure drop than the plate heat exchanger, and the pump was only able to feed 3 of the 5 plates on the freezer resulting in water in 2 of the 4 cells.



Figure 12: Plate freezer test.

The water temperature was measured in the middle of the media and the process value did not completely correspond to the average process temperature. The measuring method was modified in the last test so with a Pt1000 sensor attached to a copper tube the sensor was placed in the centre of each of the two cells. That method made it possible to easily detect the three different phases. The measurement method is also the reason why the ice temperature declined in the freezing period. The water freezing process was visually controlled during the entire process and the build-up of ice on the plate was manually detected.

The cooling capacity was not measured as there was no flow meter in the dry gas return line. The cooling capacity was calculated based on the mass of water in the system.

The calculation was carried out for the 3 following process periods:

- 1st period: Cooling down water to start freezing
- 2nd period: Freezing of water
- 3rd period: Cooling down the ice

6.2 Plate freezer temperature profiles

Several tests were carried out for the plate freezer but only one test result appears in the report. In the other tests the temperature of the water approached 0°C during the freezing period.

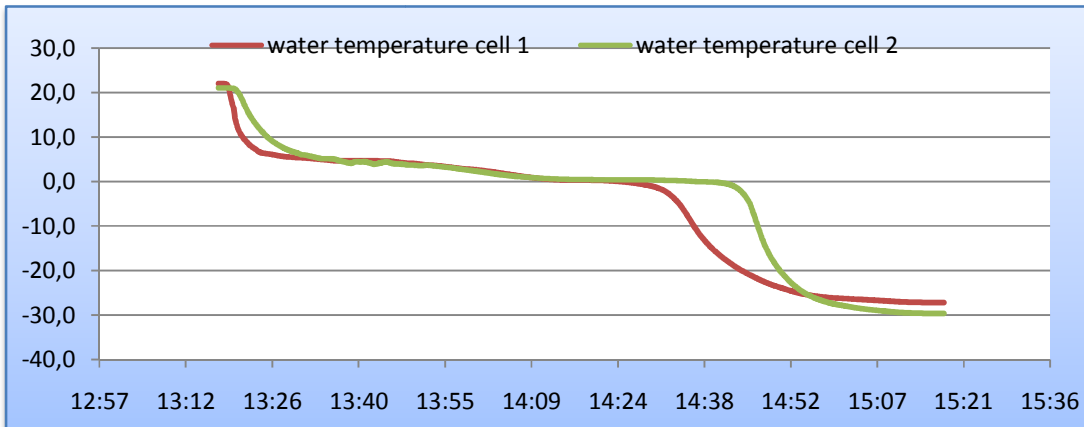


Chart 9: Water temperature with pt 1000 sensor placed in the middle of the cells.

The chart above clearly illustrates the 3 phases in the process. The result is only included to show the difference of the temperature measuring method.

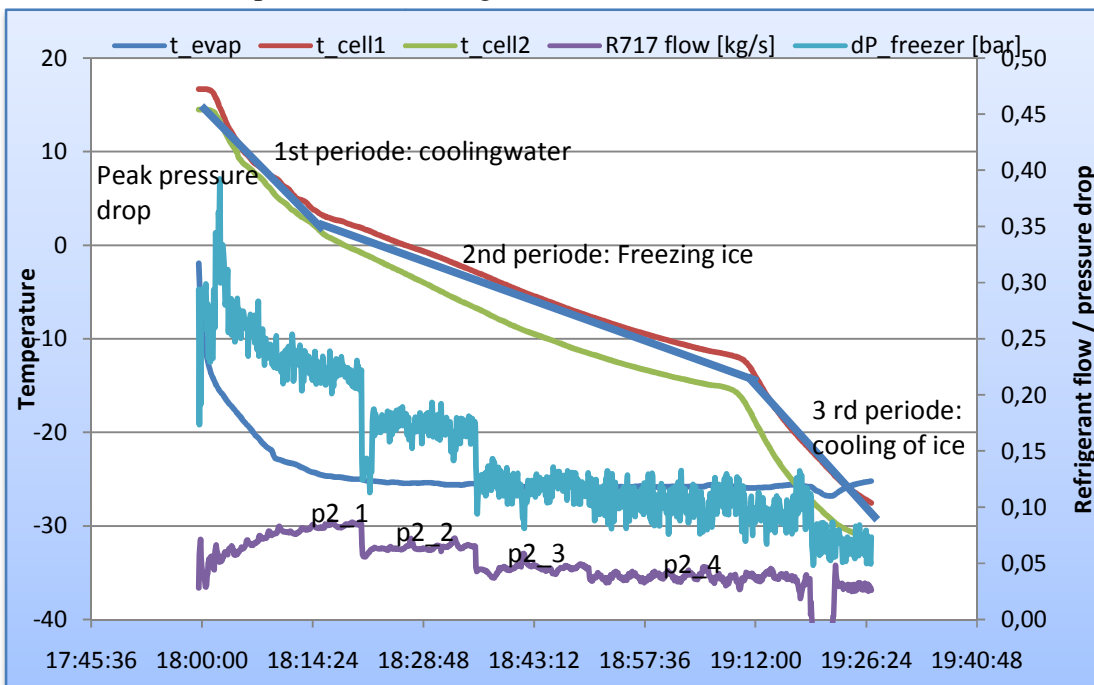


Chart 10: Test result of the water freezing process. Pressure drop and flow are shown on the axis to the right

In chart 10 the freezing process is shown. During period 2, the ammonia flow was varied (purple curve) in four values and the flow was lowered as the freezing of the water took place. It appears that the pressure drop in the freezer (dp_{freezer}) drops as a function of the ammonia flow. During the start-up of the process when the plate freezer was warm, the ammonia pump had difficulties in pumping in the liquid due to the high pressure that builds up in the freezer. First of all, that result means that the pump has to be able to deliver a higher pump pressure than the current pump does. Secondly, the pump must run at maximum speed during the start-up phase.

6.3 Plate freezer test result

The total process time for the three periods was 1 hour and 27 minutes.

1st period: Cooling down water to start freezing (15°C to 0°C)

Process time: 15 minutes
 Cooling capacity: 7.1 kW
 Ammonia flow: 0.068 kg/s (average)
 Circulation rate: 13.0 (average)

The cooling capacity was lower in this period due to the start-up process with high superheating in the return gas.

2nd period: Freezing of water

Process time: 57 minutes
 Cooling capacity: 9.4 kW (average)

The 2nd period was divided into 4 sub periods where the flow varied during the freezing period. In that way, the process time of the freezing period could be evaluated and as it is the longest period the gain from decreasing the ammonia flow and thereby the pressure loss in the system would be considerable. To complete the test data, a period from another test with even lower circulation rates was included in the test series.

	Pump flow [kg/s]	Rated capacity	Capacity n=1 [kW]	Circulation rate	Pressure drop in freezer [bar]	Calculated wet return [bar]
Period 2_1	0,084	9,6	114	12,2	0,220	0,105
Period 2_2	0,064	9,5	87	9,3	0,169	0,095
Period 2_3	0,045	9,5	62	6,6	0,119	0,085
Period 2_4	0,038	9,3	52	5,5	0,105	0,08
Other test	0,014	9,3	19	2,8	0,09	0,065

Figure 13: Test result from the freezing period. One additional result from another test was included.



Figure 14: Block of ice in the freezer at the end of the test.

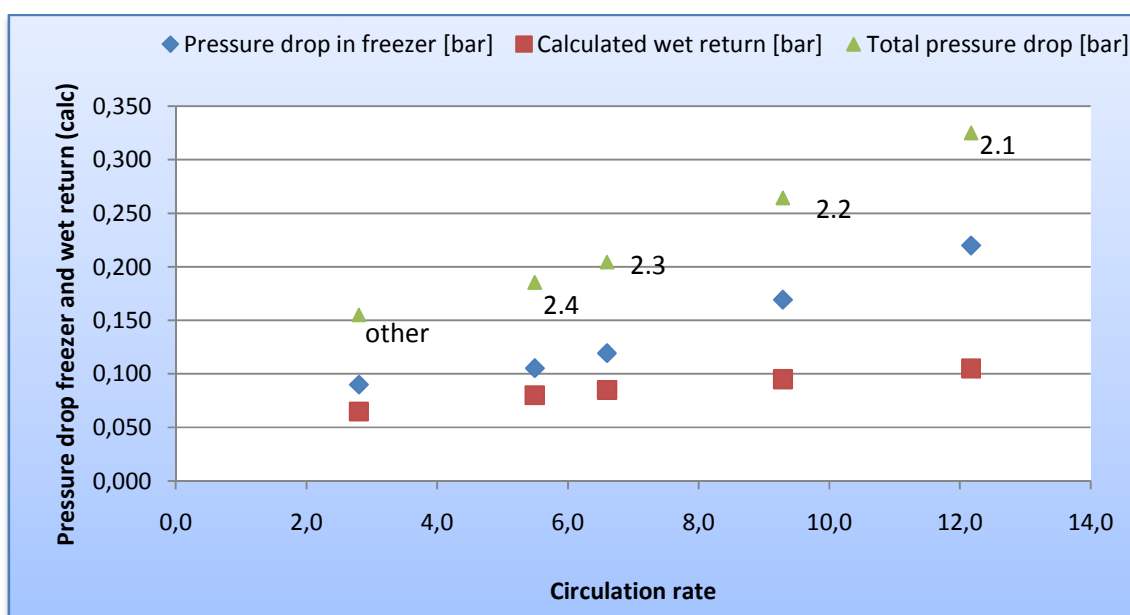


Chart 11: Measured pressure drop in freezer and calculated pressure drop during the freezing period.

The results show that if the plate freezer operates in a system with a 6 meter high riser, the pressure drop will increase with the circulation rate. However, the pressure drop in the plate freezer is up to twice as high and has the greatest impact on the total loss when the circulation rate is increased. The total pressure drop is increased from 0.170 bar at $n=2.8$ to 0.325 bar at $n=12.2$ which is an increase of 110 %; or a possible loss on the COP of 13%.

3rd period: Cooling down the ice

Process time:	15 minutes
Cooling capacity:	8.1 kW
Ammonia flow:	0.033 kg/s (average)
Circulation rate:	5.5 (average)

6.4 Plate freezer conclusion

The result of the plate freezer test shows that there is a much greater potential for regulating the flow so the circulation rate is kept at a lower rate due to the high pressure drop in the freezer and wet return.

In this test, the pressure loss in the wet return was not measured but as the calculation method was verified in the plate heat exchanger test the pressure was calculated for a riser with a height of 6 metres.

The result from the freezing period indicated that the freezing capacity of the freezer only changed slightly when the circulation rate was changed. In this test, the circulation rate varied from 2.8 to 12.2 without influencing the performance. However, the loss in the system increases with the circulation rate. The reason why plate freezers often operate with high circulation rates is to give an optimum performance in all the various phases. As the flow is not regulated during the process, the freezer operates with the highest possible circulation rate.

Operating the plate freezer with a circulation rate of 2.8 compared to 12 gives a saving on the pressure drop in the freezer of 0.13 bar and a pressure drop in the wet return of 0.040 bar.

The overall saving on the COP in relation to the plate freezer and wet return pressure drops is 13%.

A process industry where plate freezers are used was visited during the project and the temperatures in the system were controlled. The freezer was operated at an evaporating temperature of -30°C and the compressor was operated at -40°C. That corresponds to a pressure loss of 0.46 bar in the system. If the circulation rate of each freezer was optimized, the pressure loss could be reduced by 50% making it possible for the compressor to operate at -34.4° giving an improved COP of 27%.

The result from this test is confirmed by the factory observations and it is obvious that there is a saving potential on the COP in relation to plate freezer operation of at least 13%.

7 Air cooler test result

7.1 Air cooler test method

The Alfa Laval Airmax II 562 air cooler has a rated capacity of 60kw at an evaporation temperature of -35°C and a room air temperature of -15°C . The test method of this evaporator is again different from the two previous evaporators. The evaporation temperature varied while the airflow and room temperature were kept constant and that caused a variation in the outlet air temperature. The lower the outlet air temperature, the higher the capacity and therefore there was temperature difference between the evaporation temperature and air temperature out of the air cooler.



Figure 15: Air cooler mounted in climate chamber.

The reason for the low evaporation temperature is the low heat transfer values on the air side, which requires a high temperature difference on the evaporator. The overall heat transfer values are not affected very much by the ammonia heat transfer values. To test the importance of the ammonia flow and thereby the circulation rate on an air cooler, test run 62 was carried out at 3 different ammonia flows. However, the pump characteristics only make it possible to vary the flow within a small range. The test matrix for the air cooler appears from the 6 points below.

Run	Evaporating temperature	pump flow	Air temp. In	Air temp. out	Calculated evaporator flow	Circulation number	Evaporator capacity
	°C	kg/s	°C	°C	kg/s		kW
60	-35	0,0837	-15,3	-21,9	0,047	1,8	63,9
61	-32	0,0690	-15,3	-20,7	0,038	1,8	52,7
62	-29	0,0557	-15,3	-19,6	0,031	1,8	42,5
63	-26	0,0417	-15,3	-18,5	0,023	1,8	31,8
64	-23	0,0258	-15,3	-17,2	0,014	1,8	19,7
65	-20	0,0117	-15,3	-16,1	0,006	1,8	8,9

Figure 16: Test program for the 5 different evaporating temperatures and related capacities.

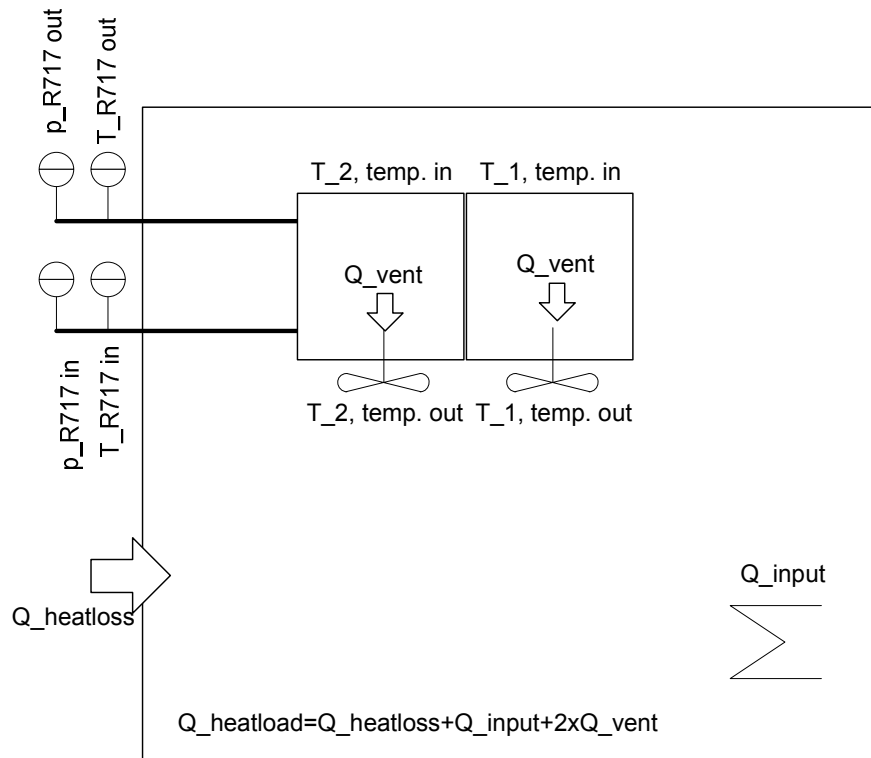


Figure 17: Climate chamber with air cooler

Formulas for the calculations

Air flow $q_{mair} = airflow * rho_{air,-15}$

Heat load $Q_{heatload} = Q_{heatloss} + Q_{input} + 2 * Q_{vent}$

Cooling capacity calculation: $Q_{capacity} = q_{m,air} * c_{p,air} * (t_{air,in} - t_{air,out})$

7.2 Test data

It was not possible to test the evaporator at maximum capacity due to a limited electrical power supply to the heaters. The heat load of the ventilator fan is added to the total electrical heat load. The heat loss from the climate chamber was measured and calculated to 107 W/K which gives at total average heat loss with ambient temperature of 22°C and a room temperature of -15°C at 3.86 kW. That heat loss was added to the measured electrical heat load and resulted in the total heat load of the experiment.

Run	Evaporator temperature	Pump flow	Right side air in	left side air in	Right side air out	left side air out
	T_p_e [C]	f_ford - [kg/s]	t_air_in_1 - [°C]	t_air_in_2 - [°C]	t_air_o_1 - [°C]	t_air_o_2 - [°C]
Run 60	-33,7	0,0710	-14,4	-14,2	-17,9	-18,7
Run 61	-32,0	0,0631	-14,7	-14,5	-17,8	-18,5
Run 62.2	-29,1	0,0378	-14,4	-14,4	-16,9	-17,6
Run 62	-29,0	0,0441	-13,4	-13,1	-16,1	-17,8
Run62.1	-29,0	0,0487	-13,7	-13,7	-16,6	-17,3
Run 63	-26,2	0,0335	-14,4	-14,7	-16,6	-17,0
Run 64	-23,1	0,0331	-14,7	-15,0	-16,1	-16,4

Figure 18: test data from air cooler test

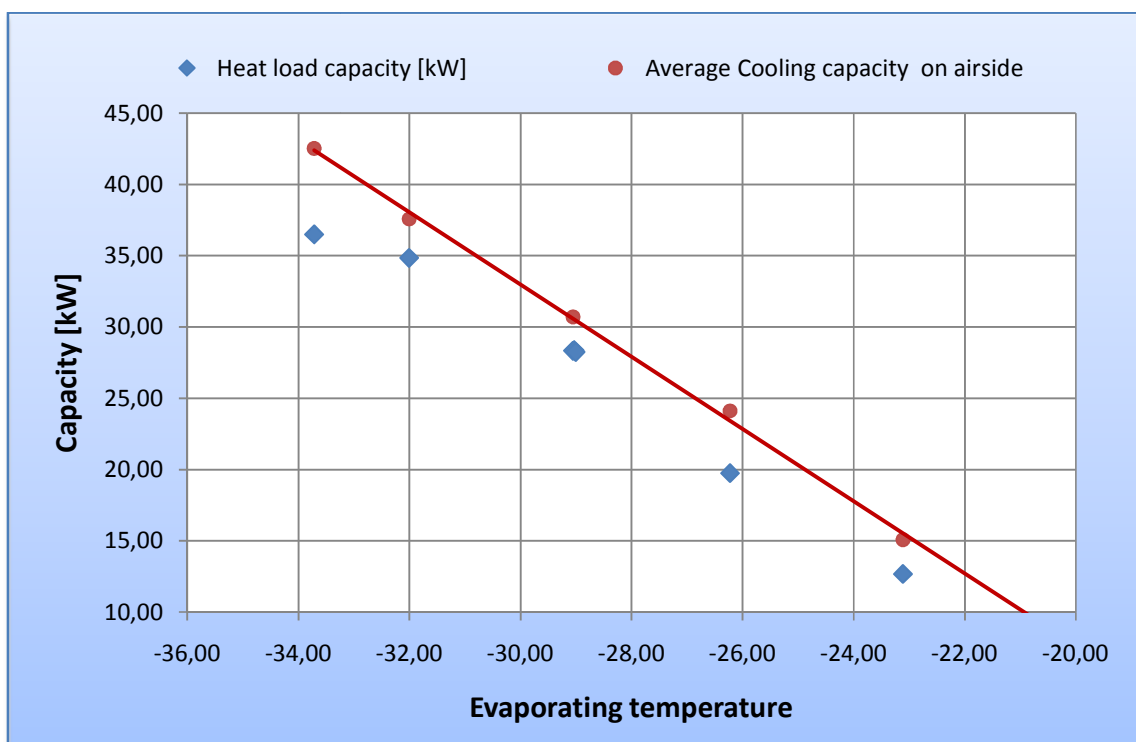


Chart 12: Cooling capacity calculated on air side compared to heat load on climate chamber.

From the chart above it appears that the heat load capacity is lower than the calculated capacity based on the temperature difference and airflow. On the calculated capacity there is an inaccuracy on the temperature and due to the low temperature difference that could be as high as 5%. The airflow measurement of the air cooler was carried out at ambient temperature and it was not taken into account that the performance of the ventilator was lower at the operating temperature of -15°C.

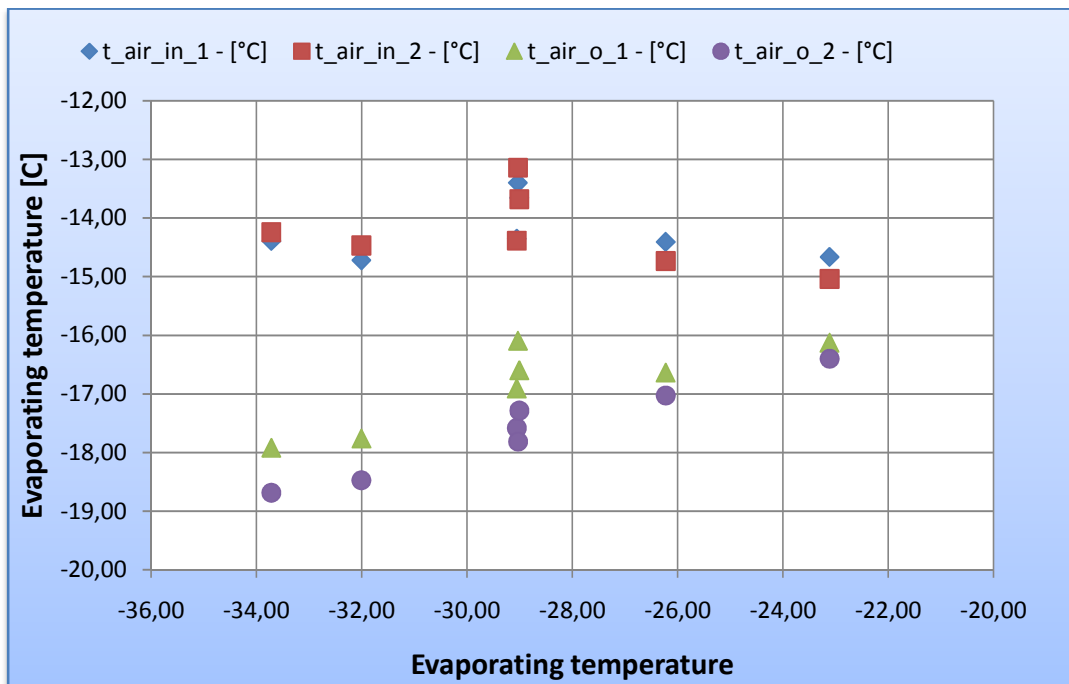


Chart 13: Air temperatures measured for each of the two sections.

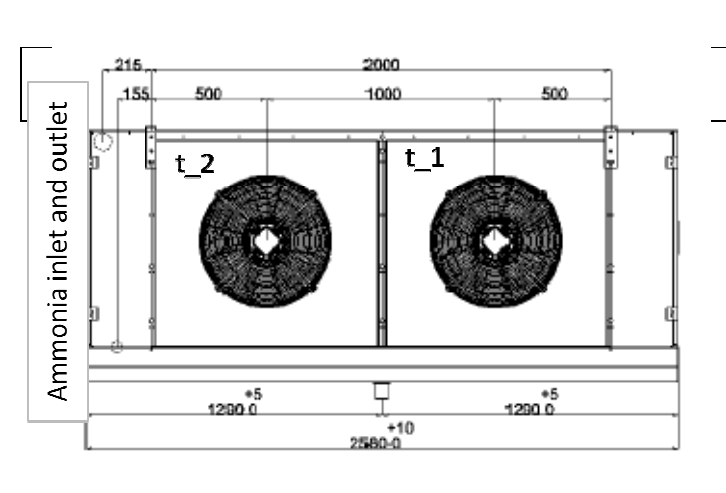


Figure 19: air cooler design.

The chart 13 above shows that there is a difference between the section of the evaporator. The section t₂ that produces the coldest air is the one closest to the ammonia inlet and outlet. Air measurements showed that section t₁ has an average airflow of 7.5%, which is higher than in section t₂. The flow variance explains some of the difference but there is up to 22% difference on the calculated capacity

between the two sections for the full load data. The difference is most likely due to biased distribution of refrigerant in the coil but it was not possible to further investigate the matter in the course of this project.

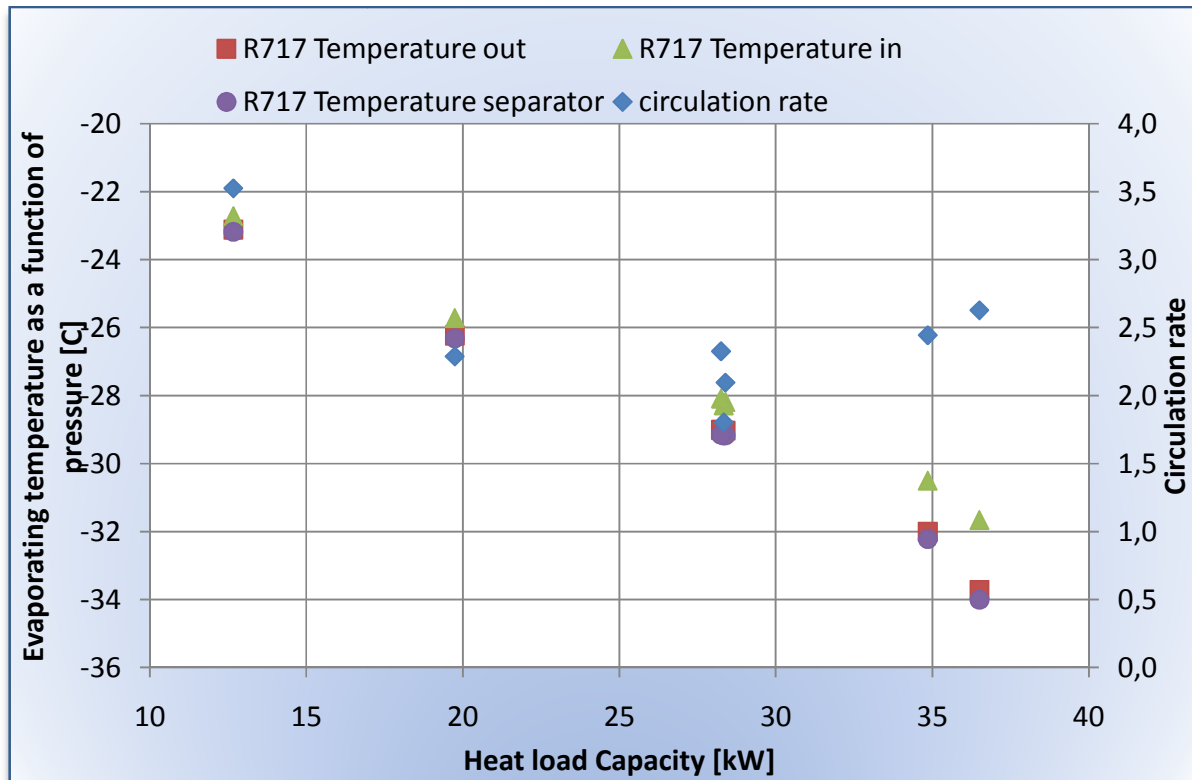


Chart 14: Ammonia temperatures as a function of the electric heat load capacity

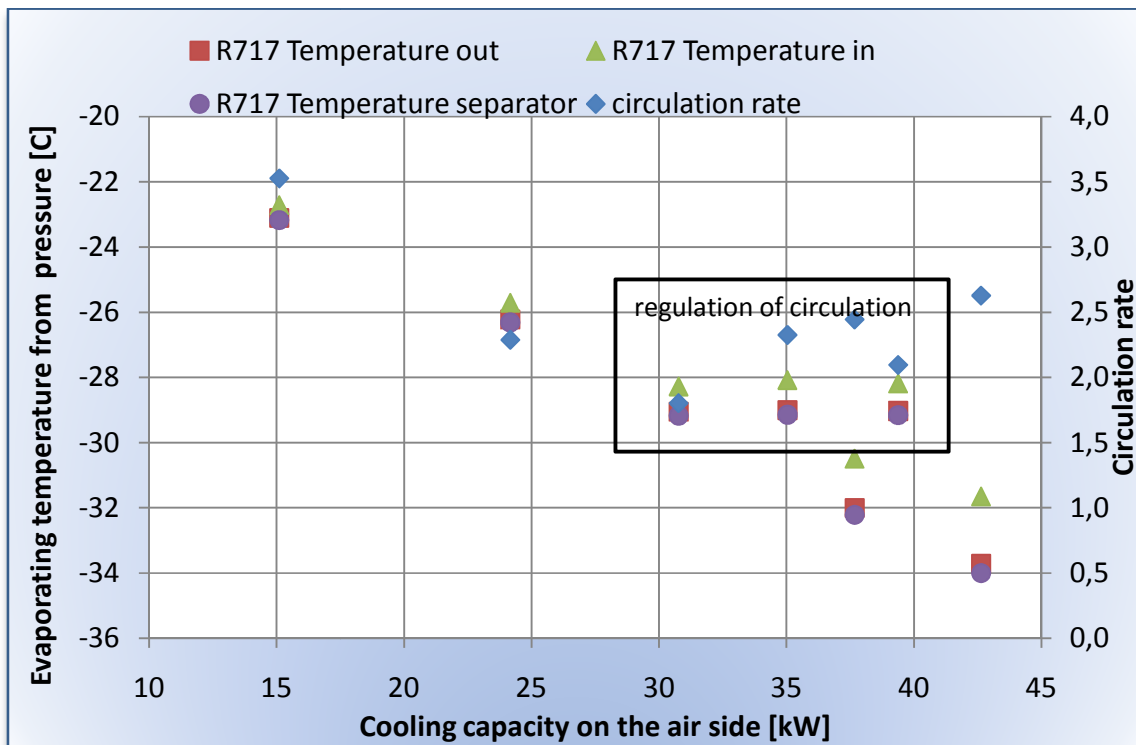


Chart 15: Ammonia temperatures as a function of the calculated cooling capacity of the air side.

The chart above shows that the air cooler gain is greater in relation to capacity regulation as the heat transfer value on the air side is the dimensioning factor. At full load an ejector that could give 0.2 bar lift on the suction side equal to 4 K in this area of operation could increase the capacity of the evaporator by between 3 and 12 kW.

7.3 Variation of the circulation rate in relation to the efficiency

Due to pump limitations and high pressure loss in the complete system circulation rate regulation was limited. To be able to regulate the flow of an air cooler, the pump characteristic will have to have a higher degree of flexibility and the inlet header of the air cooler will have to be redesigned to give a better distribution of the liquid at the various capacities.

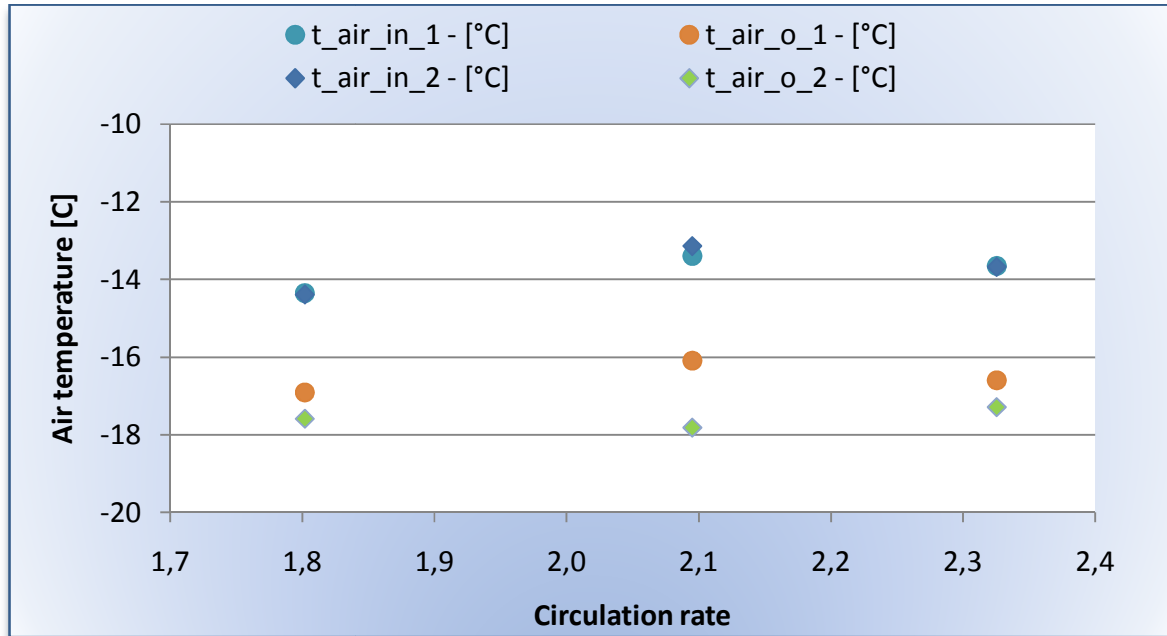


Chart 16: Air temperatures in and out of the air cooler.

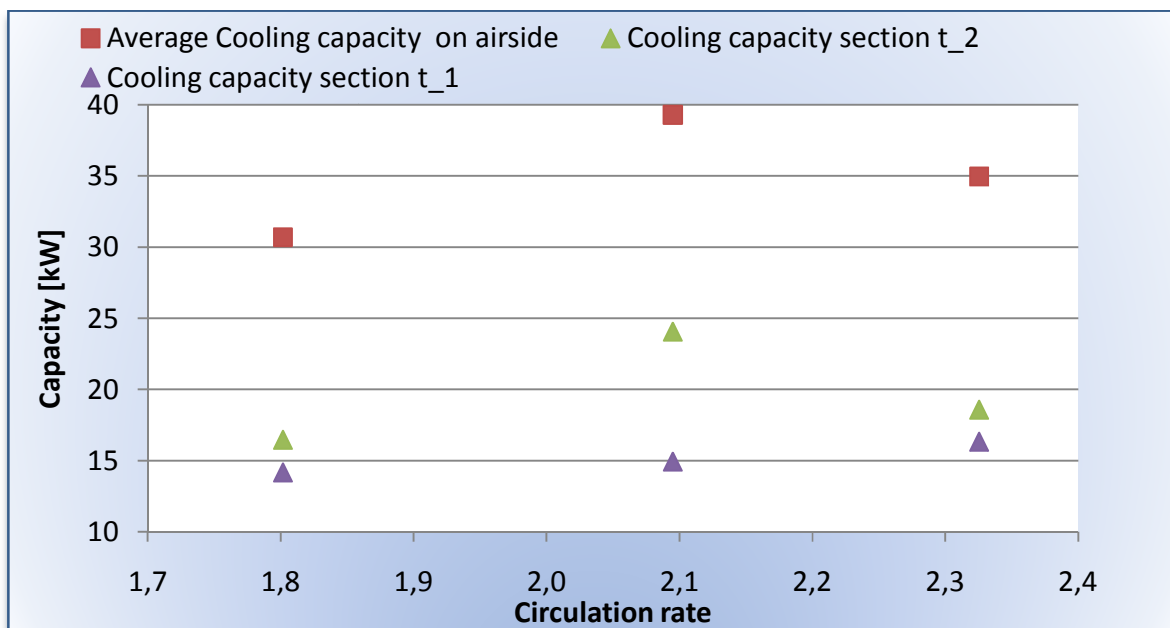


Chart 17: Calculated cooling capacities in the two sectors of the cooler.

The different circulation rates (between the value at $n=1.8$ and $n=2.2$) of the 3 tests, gave a capacity difference of 9 kW. In the test with a circulation rate of 2.1 there was a large difference between the calculated capacities of the two sections. The difference cannot be explained on the basis of the circulation rate alone.

7.4 Air cooler test conclusion

The optimum circulation rate could not be defined merely in the light of this test, but the test indicated that there was some optimisation in running the cooler at a specific circulation rate. However, more tests are required with a pump that has a larger span of operation.

The test clearly indicated that there is some potential for improving the air cooler capacity by controlling the circulation rate. An increased air cooler capacity of 9 kW corresponds to 30% was achieved by varying the circulation rate. However, the test could not be carried out at full load and the result would probably be less significant at full load as the limiting factor would be the heat transfer on the air side.

In relation to evaluating the wet return, the potential of the air cooler is limited as the air cooler often is placed at the highest point under the roof. There is a limited loss when lifting the wet return back to the central pump separator and that has not been evaluated in this test.

Another special factor of the air cooler is the build-up of ice on the fins that will limit the heat transfer on the air side. In the test mode, the test was run for 5 hours in a closed environment so the ice build-up was limited. In cold store operations the ice build-up would be the greatest limiting factor on the air-flow and the heat transfer values.

The test does not give a clear answer to the savings potential in relation to pump circulation for air cooler operation. In the test, there was an indication that the air cooler performs better at certain circulation rates. However, the biggest potential is related to lifting the evaporation temperature at full load. If the difference between the room temperature and evaporation temperature could be increased by a few degrees, the savings potential would be considerable. In the test, a cooling capacity of 42kW and climate chamber temperature of -15°C gave an evaporation temperature of -33.7°C . If that could be increased to -30°C by improving the heat transfer values, the COP would be increased from 4.2 to 5 (figure 9) corresponding to 19% .

8 Ejector in the wet return line

8.1 Ejector description

An ejector is a thermal compressor without moveable parts. It is operated by vapour/liquid with high pressure that is transmitted through a nozzle. The vapour/liquid leaves the nozzle with high velocity and little static pressure that sucks refrigerant from the evaporator and mixes it with the high pressure liquid/vapour. The mixture is transmitted through a diffuser and ends at a higher pressure than at the low pressure side of the ejector.

By installing an ejector in the wet return the pressure loss is overcome, and in that way the same pressure is obtained in the evaporator and separator. Furthermore, a lower pressure in the evaporator than in the separator can be achieved. The objective is to use the ejector in new systems as well as a substitute for existing flooded systems. The ejector shall be mounted on each individual evaporator if several parallel evaporators are connected to the same separator. Especially for that purpose an ejector was developed in the course of the project, but it was not expected that it would be able to operate with optimum efficiency.

That means that this is not an ejector development project, but to a certain point the knowledge that already exists on ejector design can be used. The ejector will form part of the system as a passive component that replaces the expansion valve. No additional external energy source has to be supplied to operate the ejector but the energy that is available in the process will be utilised. The ejector is operated by means of high pressure liquid and will remove the pressure loss that will be in the wet return from the evaporator to the liquid separator and in that way ensure that the compressor will suck from a higher temperature. In that way, the compressor will have lower energy consumption.

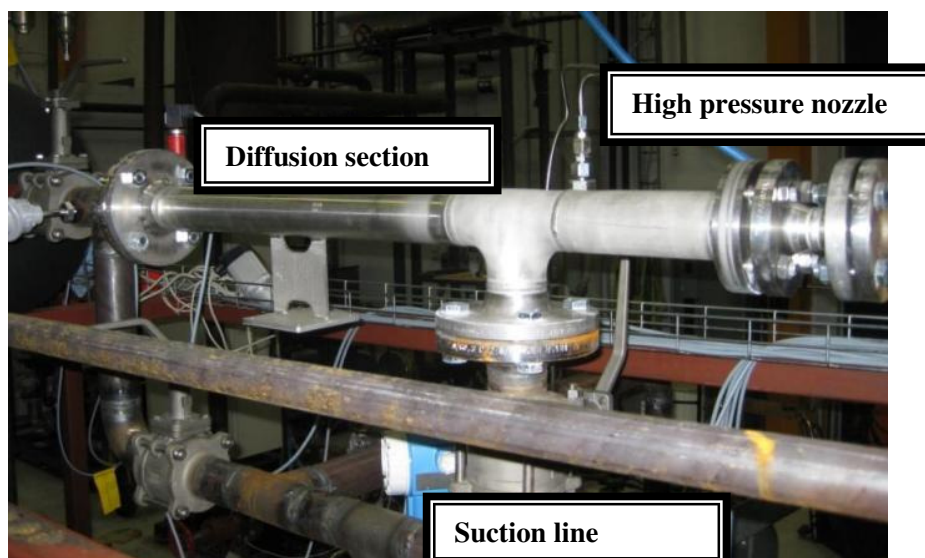


Figure 20: Ejector installation and design

The capacity regulation of the system was designed so that there was constant flow to the ejector. The capacity regulation is handled by an expansion valve that was controlled by the liquid level transmitter in the separator to maintain the ammonia level in the liquid separator as seen below.

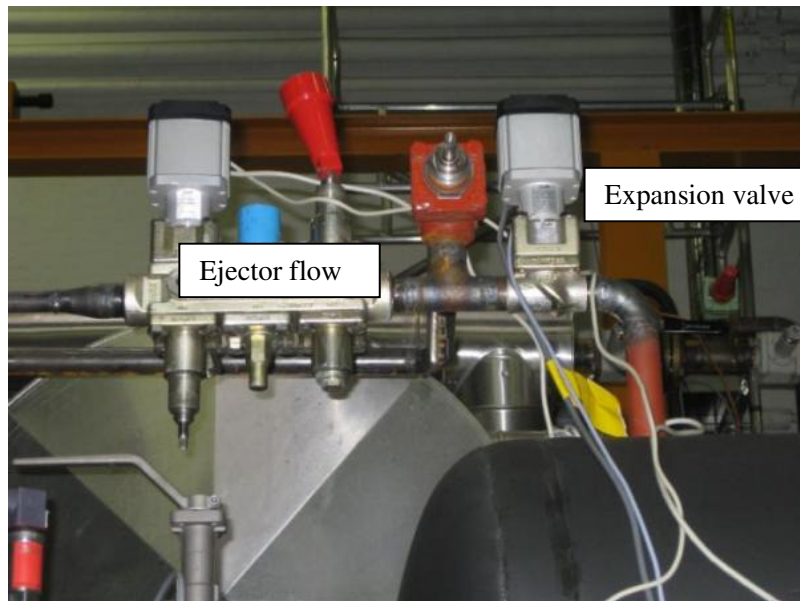


Figure 21: Ejector flow regulation system and system expansion valve.

8.2 Ejector test results

The ejector was tested only with the plate heat exchanger but it did not perform optimally in the system because the operation point of the ejector did not fit the system design. The ejector was run in a mode with zero load on the evaporator and the test results can be seen below.

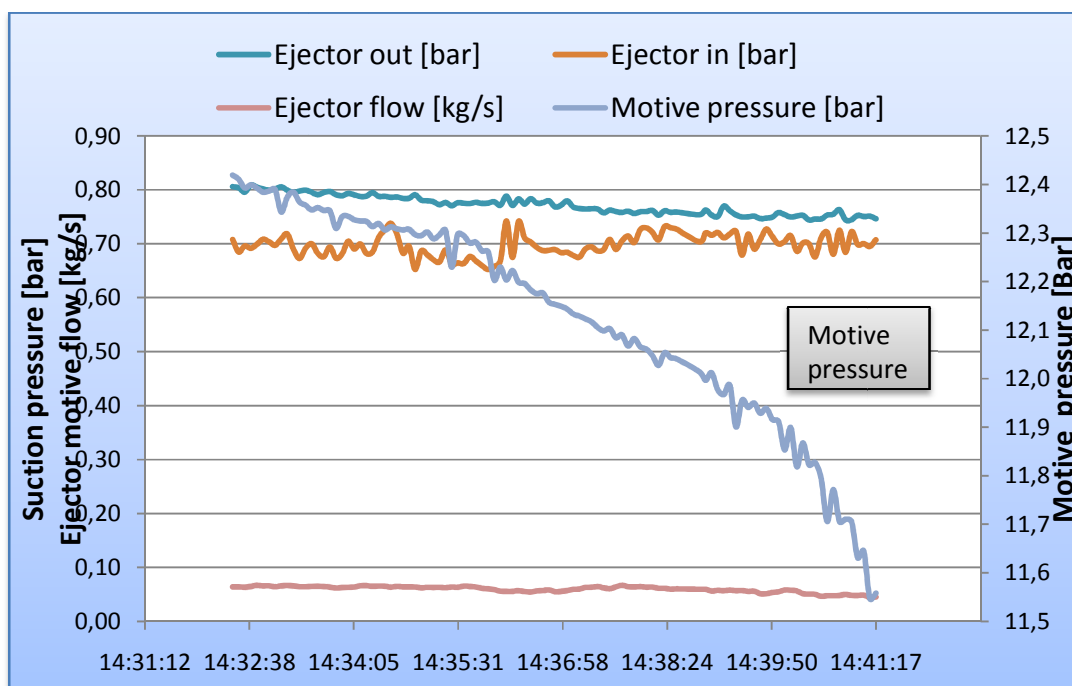


Chart 18: ejector test data

It appears from the result that the ejector could deliver a pressure lift of 0.1 bar, equal to 2K on the evaporation temperature. The ejector performance was only varied in relation to the motive pressure and it has a large impact on the ejector performance. The result confirms the challenges that are related to ejector operation were the operation point of the ejector is very narrow and must fit the operation point of the system to perform optimally.

8.3 Ejector test conclusion

The ejector that was delivered for the test setup had too large a capacity for the test rig and was not able to perform optimally. That was the main reason why we did not get any useable results in relation to operation at low-temperature ammonia plants. To be able to have an ejector in an ammonia system it must be possible to regulate the capacity of the ejector so the performance of the ejector automatically is adapted to the operation point of the system. One way to do so was to use a nozzle that could regulate in the expansion chamber so it is constantly adapted to the motive flow and pressure. It will require extensive development work for the ejector, before the ejector is commercially available for low temperature ammonia systems.

9 Pump circulation result and pump design

9.1 Small refrigerant circulators for ammonia

In the current industrial refrigerant market, pumps for controlling each ammonia circuit individually do not exist. Grundfos' contribution to the project was a proof of concept prototype of a small refrigerant pump. The pump had been used to regulate the refrigerant flow of all three evaporator types. The refrigerant pump and the control system of the pump provided opportunities for investigating an optimum circulation number in the evaporator. During the course of the project, it has been verified that it is feasible to develop a pump for this application and to prove the concept of the pump.

The separator was placed locally near the evaporator so the return from the small separator to the compressor/central separator functions with dry gas. By using small decentralised liquid separators located by each evaporator the charge of ammonia will be minimised. Therefore, inconsistency does not exist when going from large separators to small separators.

The pump was a centrifugal type with speed regulation as seen below. The pump was designed to deliver a flow for a capacity of 50 kW and a pressure drop in the system fitted to the plate heat exchanger.

The characteristics of the pump will have to be adjusted to the characteristics of each individual evaporator system and will have to be flexible enough to fit many evaporators. In most evaporator systems it is important that the pump has a steep curve.

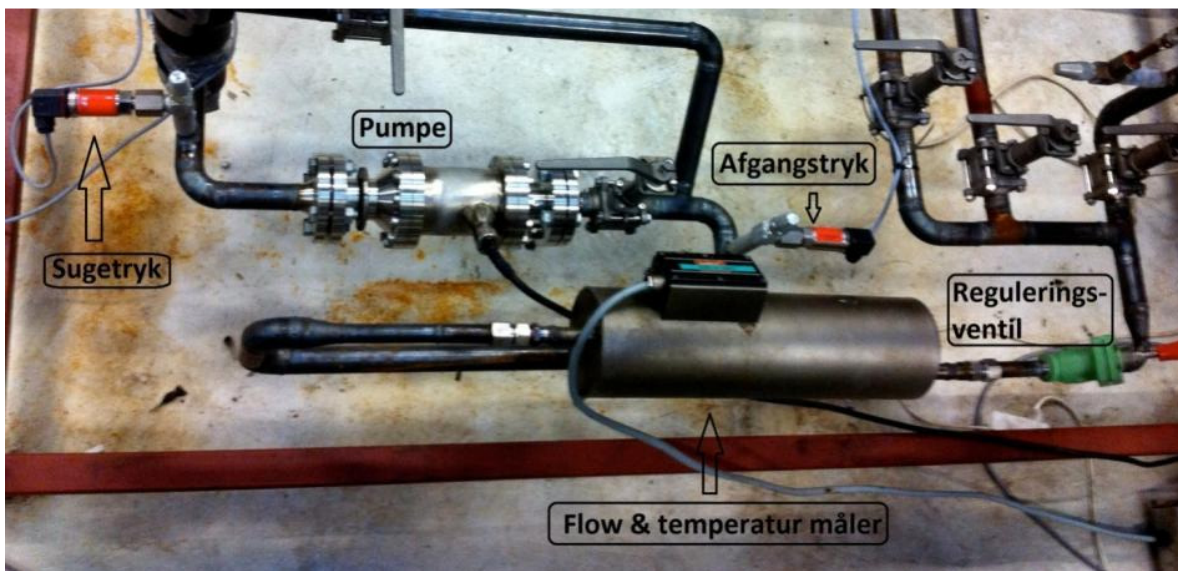


Figure 22: Pump test setup in existing non riser system.

9.2 Pump test data

In order to determine the pump characteristics, the test rig was adapted so the pump data could be obtained.

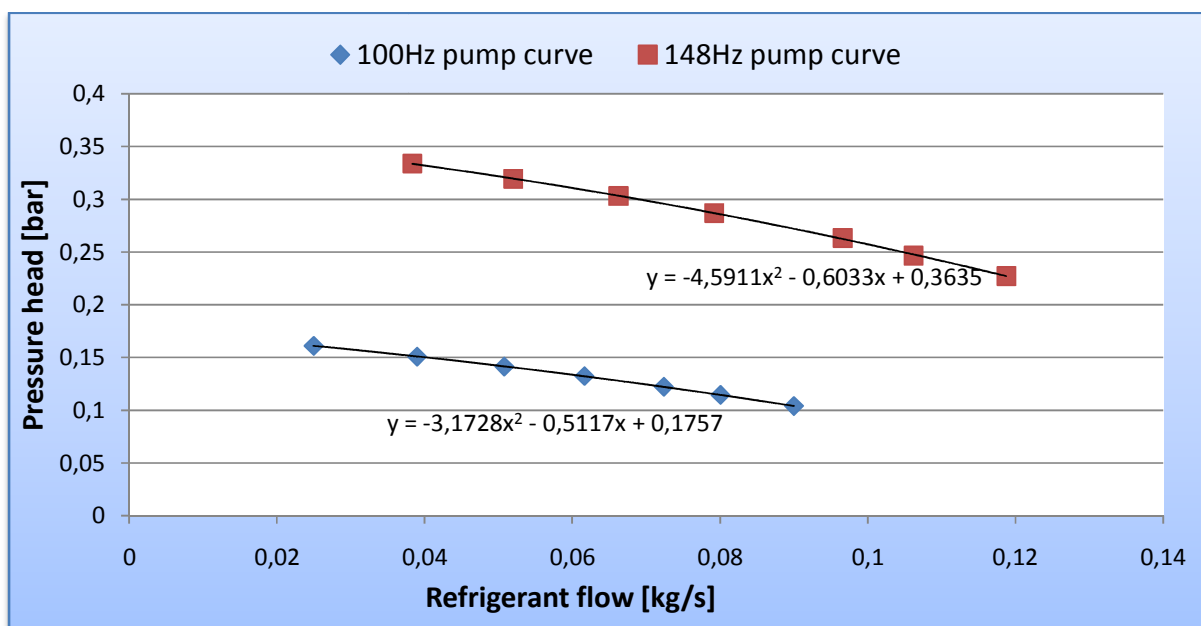


Chart 19: Pump curves at two different pump speeds.

It can be seen from the characteristics above that the pump has a limited area of performance in relation to pressure regulation. That is the main reason why the pump showed some limitations in the plate freezer and air cooler test.

9.3 Cavitation in the pump and regulation of the system

The pump that had been used in the laboratory test was a small centrifugal single stage pump (only one impeller). The pump has a limited operation range as the flow ratio is limited. The best suited pump for a refrigeration system is able to have a large span in the flow within in a small pressure range. The regenerative turbine pump is more suited for this purpose as it has a steeper pump curve. That makes the regenerative pump more stable. Secondly, the pump is less sensitive to cavitation as the velocity in the wheel is smaller than in the centrifugal pump.

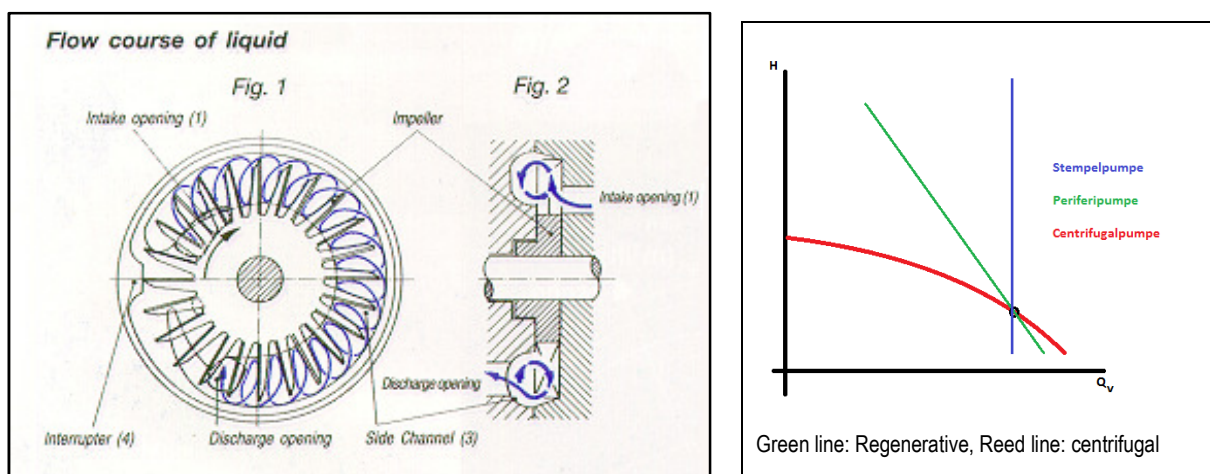


Figure 23: Regenerative pump and pump curves.

The regenerative pump is also good to couple in parallel connection which allows for easy variation of the pump capacity. The efficiency of the pump is of little importance when selecting pump type as the consumption of the pump is insignificant compared to the refrigerant compressor.

When it comes to the design of the regulation system in relation to controlling the circulation rate in a system so an optimum rate is maintained at all capacities no simple method has been developed during the course of this project. When it comes to regulating the ammonia circulator in a plate heat exchanger system it is recommended to measure the capacity on the secondary side as it is done in the test system and then use the pump curve to reach the flow that gives the optimum circulation rate. For the plate freezer, air cooler and other evaporator types it could be recommended to find a simple measuring method that could define the ratio of gas and liquid in the outlet of the evaporator. This simple method needs to be invented.

9.4 Conclusion of pump circulation

In the light of the test results of the three different evaporators, the performance and efficiency of the refrigeration process were improved when a circulating pump was used. In low-temperature systems even small improvements in the efficiency of the system will have a huge impact on the COP of the total system. First of all, the result from the test shows that optimising the circulation rate to minimizing the pressure drop in the evaporator and wet return has an impact. Secondly, there is an improvement in the evaporator performance and design that could have an even greater impact on the overall efficiency. It is therefore a clear conclusion from the project that using small pumps for local ammonia circulation is beneficial to both the performance and the energy efficiency of the system.

10. Conclusion

The industrial refrigeration system design for ammonia plants has changed very little in the past years and the regulation system for each individual evaporator station is kept at a minimum with a solenoid for the liquid line and a regulator for the hot gas defrost. However, in the machine room there has been extensive development with the introduction of high-performance screw compressors with speed regulators and more efficient regulation systems.

With this project, Danish Technological Institute has evaluated that possibilities exist for building more advanced systems at each evaporator system as the savings potential lies in the low-temperature control. There are some technical challenges involved in the development of cost-effective pumps and ejectors for this purpose and the work has commenced. It is possible that the components will be available in the market within few years. The measurement gave some basic data for Koerting to develop an ejector for the refrigeration market. Similarly, the project has provided Grundfos with basic knowledge about the performance requirements for small refrigerant circulation pumps used for establishing local circulation.

Another challenge that has to be met when implementing the proposed system is the way industrial plants are constructed. Black pipes are used and they are not cleaned before installation. Grinding and welding splatter does not improve the cleanness and the dirt is often not properly evacuated or it is simply impossible to evacuate the dirt. When refrigerant is circulated, the more sensitive components such as pumps, ejectors or modern regulation valves will be destroyed or damaged. To make it possible to implement more advanced systems within refrigeration it must be considered how cleaner industrial refrigeration systems can be built. When the ammonia pump was disassembled after the test it appeared that there was a lot of dirt in the pump, but that did not prevent the pump from operating. The magnet and filter protected the pump successfully.

The plate heat exchanger is the most optimised evaporator of the 3 selected for the test. By using an ammonia pump it is possible to design the PHE to give a minimum pressure drop on the secondary side and in that way minimize the required pump power. The test result gave a 3% saving on the COP by constructing a system without a wet return. However, the capacity of the evaporator was increased by 16% when the pump was used to circulate the refrigerant compared to self-circulation.

Operating the plate freezer with a circulation rate of 2.8 compared to 12 gives a saving on the pressure drop in the freezer of 0.13 bar and a pressure drop in the wet return of 0.040 bar. The result from the test was confirmed by the factory observations and it is clear that there is a saving potential on the COP in relation to plate freezer operation of at least 13%.

The air-cooler test did not give a clear answer to the savings potential in relation to pump circulation for air-cooler operation. The greatest potential is linked to the low evaporation temperature at full load. If the difference between the room temperature and evaporation temperature could be increased by a few degrees, then the savings potential would be considerable. In the test, a cooling capacity of 42kW and climate chamber temperature of -15°C gave an evaporation temperature of -33.7°C. If that could be



Elforsk Non riser project

increased to -30°C by improving the heat transfer values, the COP would increase from 4.2 to 5 (figure 9) corresponding to 19% .

Basic system data and design parameters have been forwarded to Grundfos for further development. When the new components are available for the market it will be possible to redesign the existing cooling system and build a new system with a higher efficiency than today.

It is a clear conclusion of the project that using small pumps for local ammonia circulation is beneficial to the performance as well as the energy efficiency of the system. The overall savings potential of the three evaporators amounts to between 13 and 19% .

REFERENCES

1. Hansen, Svann; Hansen, 2003, Analysis of the Danish energy consumption for refrigeration, divided into areas of use and unit types, ESO case 4, *Danish Technological Institute*.
2. Knudsen, Hans Jørgen Høgård; 2002, Refrigeration and heat transfer. Part 1: Evaporator and condensers, *DTU*.
3. Reinholdt, Lars; May 2006, freezing equipment with CO₂ as refrigerant, *Danish Technological Institute*.
4. Pressure loss in the wet return from the evaporator to the liquid separator described in the project application has been informed by Johnson Control Inc., Supplier of industrial refrigeration systems with ammonia.
5. Jesper Weinkauff Kristoffersen and Søren Bøgelund Madsen, Refrigerant pump for flooded evaporation, Bachelor project, Engineering College of Aarhus, December 2010.