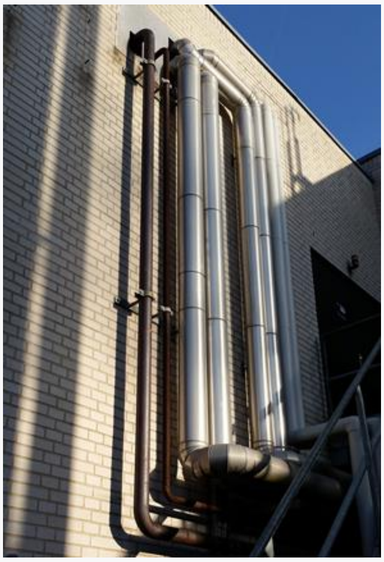
DANISH TECHNOLOGICAL INSTITUTE

Flexible Energy Optimized Split Condenser Ammonia Heat Pump (FOSCAP)

Final report



Title:
Flexible Energy Optimized Split Condenser Ammonia Heat Pump (FOSCAP)

Prepared for:
Project consortium

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1. Preface

This report is the final report of the study: "**F**lexible Energy **O**ptimized **S**plit **C**ondenser **A**mmonia Heat **P**ump" (the FOSCAP-concept). The objective of the project is to demonstrate an improvement in the energy efficiency of heat pumps with up to 30 % by using a novel heat exchanger technology where the condenser is split and may, therefore, be able to provide heated water at two temperature levels. This will result in an improved overall performance and a possibility to supply some of the heated water at a higher temperature level, which will be advantageous for many applications.

This research project is financially supported by the Danish Energy Agency's EUDP programme (Energy Technology Development and Demonstration).

Project number: J.nr. 64013-0543.

The project is carried out in cooperation with the Technical University of Denmark and the following industrial cooperating partners: Svedan Industri Køleanlæg, Innotek (Lund & Sørensen), Alfa Laval, Arla Foods, Hofor, MYCOM, Egå Smede og Maskinværksted, and Bjerringbro Varmeværk (now Gudenådalens Energiselskab).

The following persons have participated in the project:

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- Kim Holm, INNOTEK and Lund & Sørensen
- Rolf Christensen, Alfa Laval
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The project team would like to thank Povl Frich from the EUPD programme (Danish Energy Agency), who has supported the project with valuable inspiration.

1.1. Project Details

Project title	Flexible Energy Optimized Split Condenser Ammonia Heat Pump (FOSCAP)
Project identification (program abbrev. and file)	Område: Energieffektivitet, EUDP 13-II, Proj. Nr: 64013-0543
Name of the programme which has funded the project	Energiteknologisk Udviklings- og Demonstrationsprogram (EUDP),
Project managing company/institution (name and address)	Danish Technological Institute, Gregersensvej 1, DK-2630 Taastrup
Project partners	Svedan Industri Køleanlæg Innotek (Lund & Sørensen) Alfa Laval Arla Foods Hofor Mycom Egå Smede- og Maskinværksted Bjerringbro Varmeværk (Gudenådalens Energiselskab) Technical University of Denmark
CVR (central business register)	56976116
Date for submission	27. November 2018

1.2. Short Description of Objectives and Results

English version

The objective of the project is to improve the energy efficiency of ammonia heat pumps with up to 30 % by means of a novel heat exchanger technology, i.e. the condenser is split whereby it provides heated water at two temperature levels.

The split condenser concept has been installed in a heat pump at the food company Co-Ro, and it has been shown that this solution can reduce the condenser cost with up to 80%. A test heat pump installed at Danish Technological Institute (DTI) has been built with two partial flows of which one has a high outlet temperature of 120°C and 26% of the capacity. This has not been done previously.

The expected efficiency improvement was 10 to 30%. The work has shown that it is possible to obtain a performance improvement of 15% under normal conditions.

The project has provided input for a Ph.D., a master, and two bachelor projects. Additional to the theses, four papers have been produced for international conferences.

Danish version

Formålet med projektet er at forbedre ammoniakvarmepumpers energieffektivitet med op til 30 % ved brug af en ny varmevekslerteknologi, hvor kondensatoren er delt og derved producerer opvarmet vand på to temperaturniveauer.

Split kondensatorkonceptet er installeret i en varmepumpe på fødevarevirksomheden Co-Ro, og det har vist sig, at denne løsning kan reducere kondensator omkostningerne med op til 80%. En testvarmepumpe installeret på Teknologisk Institut er bygget med to delstrømme, hvoraf den ene har en høj udløbstemperatur på 120°C og 26% af kapaciteten. Dette er ikke sket tidligere.

Den forventede effektivitetsforbedring var 10 til 30%. Arbejdet har vist, at det er muligt at opnå en effektivitetsforbedring på 15% under typiske forhold.

Projektet har givet input til en ph.d., et master og to bachelorprojekter. Ud over de tilknyttede afhandlinger er der blevet produceret fire papers til internationale konferencer.

1.3. Executive Summary

English

Today, ammonia heat pumps are constructed with an oversized condenser unit, compared with the design capacity, to achieve low temperature approaches. By splitting up the condenser into two separate heat exchangers, the low temperature approaches can be maintained, and the total heat transfer area is reduced, or the design capacity is raised. The concept is proven in a laboratory test unit in the PSO project; Energy Efficient Ammonia Heat Pump. In that project, only the fundamental advantages of the concept have been proven. In this current project, further investigations and scientific research work have been performed to understand the depth of the concept by obtaining detailed knowledge about the heat transfer mechanisms involved. This has made it possible to design and optimize demonstration plants. The concept and other improvements found during the project show a potential efficiency improvement of up to 30%. The complexity of the complete heat pump system makes the designing of an optimum application for specific units without calculation tools very challenging and not economically feasible. In this project, the required calculation tools and control algorithms have been analysed, which renders it possible to commercialize the concept.

The split condenser concept makes it possible to increase the ammonia heat pump efficiency without increasing the cost of components, compared with present solutions. The concept also makes heat pump solutions suitable for a broader range of temperature requirements without the necessity of more expensive components. The flexible design can contribute to cheaper custom solutions and hereby increase the possible applications for the ammonia heat pump. Hence, the concept contributes to the electrification of heat production in all sectors in an energy efficient and cost-effective manner.

The industrial partners in the project will be able to differentiate themselves from their competitors because of the optimized heat pump technology, enabling them to increase their sales and exports.

Danish

Kondensatoren på ammoniakvarmepumperne, som findes på markedet i dag, er typisk overdimensioneret med hensyn til varmeydelsen for at sikre små temperaturforskelle. Hvis kondensatoren deles i to varmevekslere, opnås de lave temperaturforskelle stadig, og det samlede varmeoverførselsareal reduceres, eller varmeydelsen hæves. I PSO-projektet, Energieffektive ammoniakvarmepumper, blev konceptet eftervist på laboratorieniveau. I dette projekt blev kun de grundlæggende fordele ved konceptet eftervist. Der er derfor i det nye projekt foretaget en dybdegående videnskabelig undersøgelse af konceptet for at opnå en detaljeret viden om, hvilke termodynamiske mekanismer der ligger bag. Med denne viden har det været muligt at projektere og optimere demonstrationsanlæg. Ved split kondensorkonceptet og et antal andre forbedringsmuligheder er det vist, at der samlet kan opnås en effektivitetsforbedring af ammoniakvarmepumpen med op til 30 %. Den samlede kompleksitet af ammoniakvarmepumper gør det svært og tidkrævende at projektere optimale enheder, som er økonomisk gangbare, uden beregningsværktøjer. I FOSCAP projektet er de nødvendige beregningsværktøjer og styringsalgoritmer, som gør det muligt at kommercialisere konceptet, blevet analyseret.

I PSO-projektet havde de to varmevekslere samme pladegeometri, som tydeligt efterviste konceptet. Konceptet har muliggjort en effektivisering af ammoniakvarmepumpen uden at forøge komponentomkostningerne i forhold til eksisterende løsninger. Samtidigt kan ammoniakvarmepumpens anvendelse bredes ud over et større temperaturinterval uden et behov for dyrere komponenter. Med det fleksible koncept er det nemmere og billigere at projektere brugerdefinerede løsninger og at forøge ammoniakvarmepumpens anvendelsesområde. Hermed bidrager konceptet til elektrificeringen af varmeproduktionen i alle sektorer på en energieffektiv og økonomisk fordelagtig måde.

De industrielle partnere i projektet vil blive i stand til at differentiere sig fra deres konkurrenter på grund af den optimerede varmepumpeteknologi, hvilket giver dem mulighed for at øge deres omsætning og eksport.

1.4. Project Objectives

The objective of the project is to improve the energy efficiency of ammonia heat pumps with up to 30 % by means of a novel heat exchanger technology, i.e. the condenser is split whereby it provides heated water at two temperature levels. The system can be generalized to other types of refrigerants. This will result in an improved overall performance and a possibility to supply some of the heated water at a higher temperature level, which will be advantageous for many applications.

The objective is achieved by carrying out research on how the different elements are combined most optimally through gained experience from the testing of the concept and the selection of control concept as well as by calculating the expected performance.

1.5. Project Results, Dissemination and Utilization of Results

In the project, the FOSCAP concept is elaborated by carrying out research encompassing both experimental work and calculation of the performance. Moreover, the control scheme of the system has been established and analyzed. Also, a calculation tool has been developed.

The commercial status is that the project partners have begun to market and sell the concept as a part of the quotations, which they give. The participation in the project have been fruitful for the partners due to the developed knowledge and design tools.

The objectives stated in the project proposal have been obtained by the conducted research, which shows that the technology is feasible for the intended purposes. The completed work showed more challenges than what was foreseen in the proposal. Several issues related to the technology should be considered in order to obtain an optimal solution. The outcome of the project will also be a valuable foundation for further research in other technologies, e.g. heat pumps, which apply higher temperatures than usual.

The project results are disseminated in terms of reports and four papers presented at conferences as well as an extended know-how information for the participants involved in the project. The project has also provided input for one Ph.D. project, one master thesis and two bachelor theses at the Technical University of Denmark.

1.6. Utilization of Project Results

The results of the project are expected to be incorporated in future products and will thereby increase the turnovers, exports, and employments of the companies involved. The companies have gained experience by participating in the project, and they will use this experience commercially - both in relation to the utilization of the FOSCAP-concept and in other projects.

The project results will contribute to the energy policy objectives by improving the performance of heat pumps and thereby increasing the possibilities of achieving flexibility in the electric grid as well as increasing the share of electricity used in the Danish energy system.

1.7. Project Conclusion and Perspective

The primary purpose of the project has been to provide an understanding of the advantages of using of the split condensation concept. In addition, focus has been on the optimization of the heat pump. Several heat pump rebuilds have been made in order to provide the right conditions to demonstrate this objective.

The thesis saying that it is advantageous to split a condenser in two sections, when the temperature lift is large enough, has been demonstrated in the project in which Alfa Laval has played a central role. One final result is that it is possible to achieve a significant gain in the form of a significant reduction in the cost of heat exchangers and vice versa - it is possible to achieve a lower condensation temperature if the area is maintained.

Through two specific cases, it has been documented that the price of a solution, where the heat exchanger is split into two, can in some cases be reduced by up to 80%. However, the size of the savings is highly dependent on the specific temperature requirements, the capacity, the design pressure, and the range of available heat exchangers as well as a good portion of knowledge on heat exchanger design including a design tool that can reduce the time spend on the design work considerably.

Svedan Industri Køleanlæg, who has contributed to the design and the construction of the test heat pump installed at Danish Technological Institute (DTI) in Aarhus and used for all the tests, has also supplied a heat pump with built-in split condensers to the food company Co-Ro on commercial terms.

The project has demonstrated the potential of splitting the condenser into two with the aim of maximizing the capacity of the warm partial supply flow and delivering this partial supply flow at a very high temperature far beyond the condensation temperature. A test run has shown that it is possible to deliver a capacity equivalent to approx. 26% of the total capacity at a supply flow temperature of not less than approx. 120°C. The second partial flow was delivered at about 70°C with a supply temperature of 35°C and a condensation temperature of approx. 71°C.

Another project participant, Innotek, currently acquired by Lund & Sørensen, has developed and installed a control system at the test system at DTI in Aarhus, which allows for the control of a high and fixed outlet temperature.

The company Egå Smede- og Maskinværksted has assisted in the rebuilding of the test plant.

Mycom has been involved in the discussion, the planning, and the design of the compressor types most suitable for the FOSCAP concept.

Following project partners have participated in the discussions on possible applications for the FOSCAP concept and on the possibilities of hosting a demonstration plant: Hofor, Arla Foods, and Bjerringbro Varmeværk (Gudenådalens Energiselskab).

The concept has been tested and found operational. Several possible improvements of the concept have been identified, e.g. how to split the two sections of the condenser most optimally.

The improvement in efficiency by using the FOSCAP concept was expected to be in the area between 10 and 30 % depending of the actual design, the reference setup, and the operating conditions. An increase of 15 % was found in a case based on an assumption of an often-used system configuration with heating from 40°C to 80°C. In this case, the condensation temperature is assumed to be 82°C. By applying the FOSCAP concept with a split condenser, it will be possible to reduce the condensation temperature to 74°C and thereby achieve the 15% performance increase. The largest improvement can be expected when there is a large temperature difference (lift) of the media to be heated and a large operating time. The improvements in the lower end of this interval are expected when the temperature lift is small, and the number of operating hours limited.

Some of the challenges of using the FOSCAP technology are the additional investments compared to the obtained savings. The need for design of each single application will also be a barrier for an extended implementation of the concept, but the developed design tool will be advantageous in this respect.

However, the project shows that the FOSCAP concept is feasible in practice and that it can provide a significant improvement of heat pump efficiency in comparison with traditional single condenser heat pump systems. The project is also a valuable basis for further research and development of the technology.

2. Introduction

The primary focus of this project is on developing, designing, and demonstrating the split condenser technology as devised in the PSO project, Energy Efficient Ammonia Heat Pump. The results of the PSO project indicate that this technology results in increased heat transfer coefficients compared with a conventional setup with a single condenser with or without a separate de-super-heater.

The possibilities of using the technology for heating two different water flows can have large potentials, also within areas where industrial ammonia heat pumps are not commonly used, i.e. for combined heating of domestic hot water and room heating. This application requires a combined heating of two different water flows, typically to be covered by one heat pump.

The PSO project proved the main advantage of the new concept in terms of laboratory tests on a heat pump. The project also proved that the concept is complex, and that it requires more research and development regarding heat transfer mechanisms, system design, and control strategy to understand the concept in depth and to take full advantage of its potential. Two main features of the concept are:

- A substantial share of the total water flow (considerably more than a traditional de-super-heater) can be heated to a very high outlet temperature compared to the condensing temperature. This is possible in that the first condenser (following the refrigerant flow) acts as both the de-super-heater and the condenser. The optimization will be done in relation to the area of the heat exchangers, temperature conditions, and load distribution.
- A higher heat pump efficiency can be obtained with the same total heat exchanger area compared to the principles used on industrial ammonia heat pumps of today. The split of the heat exchangers enables higher heat transfer coefficients and smaller temperature differences, which reduce the necessary condensing temperature and power consumption. The optimization concerns especially the areas of the heat exchangers and the temperature levels involved in the split.

2.1. Discussion on coupling of split condensers

Different methods of coupling split condensers are possible. The following principles are discussed below:

- Combined condenser, where there is only one single condenser
- Split condensers, where the condenser is split into two sections, which can be combined with these two types of coupling on the water side:
    - Serial coupling
    - Parallel coupling

2.2. Discussion of Parallel and Serial Couplings

Experience shows that a larger amount of the energy is dissipated in the first section of the condenser (Q1) with a parallel coupling compared to a serial coupling on the water side.

This consideration applies only under the assumptions of:

- The same inlet temperature.
- The same mixing temperature.
- The number of plates in a parallel or serial coupled solution has the same number of plates as in a single condenser.
- The refrigerant side is always connected serially.
- The same channel plates.

When designing a condenser for a given task, it is relevant to ask the question whether one could always split a heat exchanger into two sections and then end up with a solution with a lower condensation temperature.

A relatively important parameter in this analysis is the maximum permissible pressure loss on the water side. To make this analysis relatively simple, only the pressure loss is dealt with. If a heat exchanger fully utilizes the permissible pressure loss, then a serial coupled solution will not be an option as this solution will generate a larger pressure loss than the permissible pressure loss, when the total number of plates is to be maintained. In this case, a parallel coupled solution may possibly result in a better solution, but only if a relatively large portion of the pressure loss in the single condenser is located somewhere else than in the heat transferring channels.

In cases, where the temperature lift on the water side is relatively small, the solution with a single condenser will often end up being the most optimal solution. The picture looks somewhat different if the permissible pressure drop is not fully utilized, and the further from full utilization, the more interesting other solutions will appear to be.

Thus, the recommendation is to use as much of the permissible pressure loss as possible to achieve a higher heat transfer rate to minimize the plate area.

However, when the temperature lift on the water side is relatively large, a design for a single condenser will often provide a relatively small pressure loss as the plates in general are not configured for such a lift. Therefore, in this case, one should aim at applying either a serial or a parallel coupled condenser.

When the pressure drop ends up being quite low, it is often because heat exchangers typically are developed so that they can be used for more than just one type of application. Thus, they should be able to work sensibly as evaporators as well as condensers and typically they should also be suitable for other types of tasks. At the same time, there must be a relatively large volume of sales, which can justify the many costs involved in developing a heat exchanger. However, this is a simplified image, because there are many other parameters, which play a key role in connection with this kind of development work.

The advantage of splitting the condenser into two parts is that uneven formed channels can also be selected for the two sections to achieve a better performance match. Thus, the channels can be better tuned to the respective capacity streams, while the total pressure loss can be distributed in a much more favourable way relative to the two heat exchangers.

With serial coupled condensers, the pressure drop increases considerably on the water side as the entire mass flow of water is to flow through both sections. The lowest pressure drop is obtained when both sections have the same type of channels and the same size. However, this does not imply that the capacity is evenly distributed across the two sections. The fewer number of channels. which are used in either the first or the second section, will cause an exponentially increasing pressure difference on the water side. The same is not the case on the refrigerant side. For the first section, where the refrigerant has its inlet, the pressure drop will increase exponentially as well by reducing the number of channels and consequently with a reduced condensation pressure. Ultimately, this results in a final design with a plate area increase relative to the single condenser. Similarly, the pump work will take a larger and larger part in the energy accounts. Regarding the condensation temperature, it will approximately be lower than that of a single condenser.

Two serial coupled heat exchangers of the same type and with the same type of channels are comparable to a double-length heat exchanger, which has a long-thermal length, with the exception of the outlet manifold in the first section, the inlet manifold in the second section, and the influence of an uneven distribution of gas and liquid in channels in the second section.

With two serials coupled heat exchangers, additional degrees of freedom are obtained when comparing with a similar design with a long heat exchanger. In the case of a small temperature approach, it does not matter whether there are two sections or not, whereas with an increasing temperature approach, it becomes appropriate to choose the different types of uneven channel types, where the pressure loss and, thus, the heat transfer rates can be adjusted.

In addition, it is possible to apply several channels - especially in the first section, which provide far better options for customizing the heat exchangers for the current task.

How will the split offset the quality (i.e. ratio of gas and liquid, x-value) in the longitudinal direction? A small first section with a small thermal length will result in superheated gas out of the first section. Two sections with equal thermal lengths will result in a relatively high quality (x-value) while a long thermal length in the first section and a small thermal length in the second section will generate a much lower quality.

With a given configuration of the two sections, the quality will be relatively unchanged at both full and partial load. The two sections in the split condenser will behave in the same way as a single condenser during partial load, which in both cases results in a lower condensation temperature.

In conjunction with serial coupled design, the first section (Q1), seen from the water side, will end up being relatively oversized if the split is chosen to be in close proximity to a quality of one. When commissioned, the additional area will be converted to a lower temperature approach. If this is not desired, the condenser section must be made smaller so that a part of the condensing surface is moved to the last part of the "desuperheater".

It is often expected that a solution with two heat exchangers, which are serially connected, versus a single condenser will lead to a very expensive total solution due to the need for additional piping and welding. However, vendors with large product programs will often be able to find more optimal solutions and cheaper alternatives with two condensers, serial or parallel coupled, than the traditional solution with a single condenser. It shall, however, be noted that the serial coupled condensers do not reduce the fluid velocity, which is a key parameter in terms of decreasing the size of the heat exchangers.

A coupling in parallel may have two advantages. One advantage is to obtain a smaller plate area at an unchanged condensation temperature or vice versa, i.e. to apply the same plate area and thus generate a lower condensation temperature. The other advantage is to simultaneously achieve heat recovery in section one (Q1) and minimize the total plate area.

By integrating heat recovery into a plant, it is typically the energy in the desuperheater, which is recovered. In practice, it will not only be the superheating, which is collected, due to the design of the heat exchanger. What typically happens in the design process of heat exchangers is that they often end up being relatively oversized, which is the result of limitations on the maximum pressure loss on the refrigerant side in connection with the design phase. An excessive pressure loss results in a reduced condensation temperature and thus a lower COP. The fact that a heat exchanger is oversized for a given task means that it provides a better performance with the selected design flow and inlet temperature. Simultaneously in the outlet on both sides, there will be a change of the conditions until a balance occurs, which corresponds to the case where the capacity currents match what the heat exchanger will provide when there is no oversizing. In practice, this means that when the two exchangers are connected in parallel and section one (Q1) is oversized for the current task, then there will be an adjustment of the operating point for both heat exchangers (Q1 and Q2). As section one (Q1) has been designed and built to deliver a higher performance, it will automatically affect the performance of section two (Q2). Thus, it will become oversized, relative to the design conditions. The result is that the two condensers in combination with the condensation temperature and other parameters find an equilibrium, which provides a final performance of the system. This relationship leads to a position where it is possible to design with the purpose of maximizing the yields relative to the heat recovery. Ultimately, the large oversizing will be converted into a smaller temperature approach during operation, which improves the operating economy. However, at the same time, it would have been possible to reduce the total plate area if the two sections were optimally designed from the beginning.

When a maximum of heat recovery is requested, it is important to do the right calculations so that the largest possible proportion of the condensation heat can be extracted based on the inlet temperature, the outlet temperature, and the temperature approach.

When one of the condensers are oversized, it will be possible to estimate the current temperatures at which the system will run during operation. This applies especially to the condensation temperature as it is a part of the calculation of the COP of the plant, and it will be lower than indicated by the design. The calculation is very rarely done as it is not easy to perform, and it would not be noticed if the full benefits of the process were omitted.

However, more energy is dissipated from a serial coupled condenser than from a parallel coupled condenser, when the speed (RPM) is the same, and the subcooling is just about unchanged. When the condensation temperature (Tc) increases at the same evaporation temperature, the work of the compressor will increase causing the condenser load to increase as well. If the flow is slightly lower, the outlet temperature will increase naturally. The flow may be slightly less in a parallel coupled condenser.

The first section of the condenser (Q1) is designed to generate a given supply temperature and to provide a certain capacity. If there is a requirement for both the temperature and the capacity or the temperature has to be maintained during part load, this can be done by controlling the mass flows.

When the flow through section one and two (Q1 and Q2) is reduced equally (expressed in percentages), the temperature will remain relatively unchanged and the load of section one and two (Q1 and Q2) will correspondingly be reduced evenly (also expressed in percentages). If the load of section one (Q1) must be maintained, it is necessary to further reduce the mass flow through section two (Q2) while the mass flow through section one (Q1) is increased. The consequence of this is an increasing condensation pressure, which must be sufficiently buffered in the design. Thus, depending on the design pressure of the system, there is a limit to how low the operating range can be before the limit of the high-pressure alarm is reached. If it should be possible to only operate section two of the condenser (Q2), the system must be designed to run at a system pressure which corresponds to approximately the water outlet temperature.

There are also limits to how far down it is possible to operate the load in section one (Q1), when there is also a load on section two (Q2), if the temperature cannot be higher than the desired supply temperature. However, this limit can be increased if a bypass with a pump is being built over section one (Q1).

The pump will pump water back to the inlet on section one (Q1), when the outlet temperature begins to rise more than the desired supply temperature. This increases the temperature of the inlet of the heat exchanger, and the performance decreases as the temperature difference over the heat exchanger is reduced. There is, however, still a minimum capacity of section one (Q1), but at a far lower level. It should be noted that at high loads, the flow through the bypass might be relatively large, and the pressure loss will, therefore, increase.

3. Final Scientific Dissemination of the Research in the FOSCAP Project

3.1. Introduction

The research part of the project is focused on supporting the development of the concept by the industrial partners and on conducting scientific research at an international level as well as on publishing the results internationally in journals and at conferences. The main aims were:

- Documenting and generalizing the idea of FOSCAP by generic, thermodynamic analyses.
- Suggesting improvements for process design and control.
- Developing tools for use by the industrial partners in further development of the technology.

The work resulted in the following publications:

Christensen, S. W., Elmegaard, B., Markussen, W. B., & Madsen, C. (2017). Modelling of Split Condenser Heat Pump: Optimization and Exergy Analysis. In Proceedings of the 7th conference on Ammonia and CO2 Refrigeration Technologies International Institute of Refrigeration.

Christensen, S. W., Elmegaard, B., Markussen, W. B., & Madsen, C. (2017). Modelling of Split Condenser Heat Pump with Limited Set of Plate Heat Exchanger Dimensions. In Proceedings of ECOS 2017: 30th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems.

Christensen, S. W., Elmegaard, B., Markussen, W. B., Rothuizen, E. D., & Madsen, C. (2015). Modelling of Ammonia Heat Pump Desuperheaters. In Proceedings of the 24th International Congress of Refrigeration.

Nielsen, S., Christensen, S. W., S. Thorsen, R., & Elmegaard, B. (2018). Comparison of Heat Pump Design and Performance for Modern Refrigerants. In Proceedings of the 13th IIR-Gustav Lorentzen Conference on Natural Refrigerants International Institute of Refrigeration. DOI: 10.18462/iir.gl.2018.1149.

The publications were part of a PhD project that was completed by Stefan Wuust Christensen. This is reported also independently in the thesis. The research also included development of a model of dynamic operation of a heat pump unit. Furthermore, one master thesis and two bachelor of engineering thesis were developed in relation to the project under supervision of involved researchers:

Jørgensen, P. H. (2016). Energy optimization by utilizing heat pumps in the dairy industry, Energioptimering ved anvendelse af varmepumper i mejerier.

Christiansen, R. (2015). Eksperimental undersøgelse af nyt split kondensator koncept til varmepumper. Afgangsprojekt. Danmarks Tekniske Universitet.

Bork, A. R. L., & Gunnarsson, H. (2015). Optimization and simulation of Energy-efficient Ammonia Heat Pump, Optimering og simulering af energieffektiv ammoniakvarmepumpe.

3.2. Results

The main results of the work were:

3.2.1. Performance Model for General Use in FOSCAP System Design

The basis of the work was the development of numerical models for analysis and optimization of the split condenser system as well as models for comparing the FOSCAP system to traditional systems. These models were implemented in the software Engineering Equation Solver, EES, which is commonly used in the industry and allows for flexible modeling of complex systems, including fluid properties. The models were delivered as EES models and as executables. The models included graphical user interfaces which allow for easy access to inputs and results of the models. Figure 1 and 2 show the user interfaces of the models of the traditional heat pump and the split condenser heat pump, respectively.

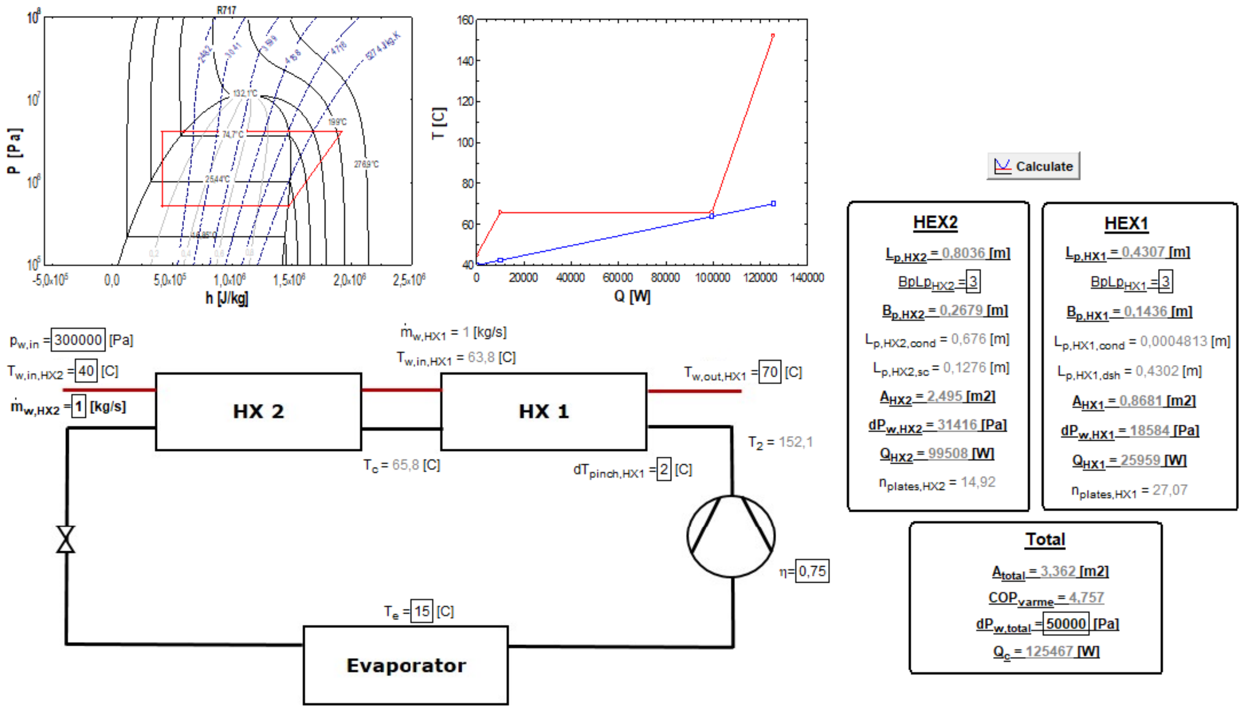

Figure 1: Diagram window of THP (traditional heat pump) model built in EES.

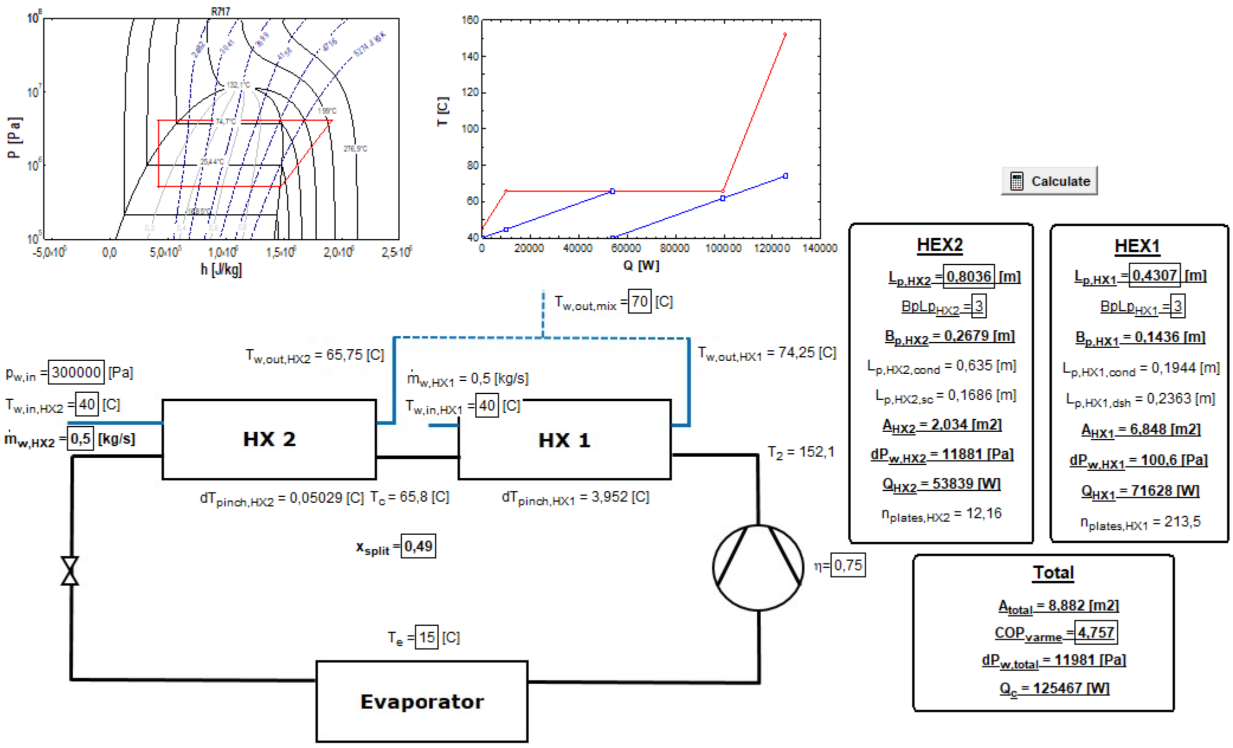

Figure 2: Diagram window of SCHP (split condenser heat pump) model built in EES.

3.2.2. Modelling of Split Condenser Heat Pump with Limited Set of Plate Heat Exchanger Dimensions

This work included a numerical study of optimal plate dimensions in a split condenser heat pump (SCHP) using ammonia as refrigerant. As illustrated in Figure 3, the SCHP setup differs from a traditional heat pump (THP) setup in the way that two separate water streams on the secondary side of the condenser are heated in parallel to different temperature levels, whereas only one stream is heated in a THP.

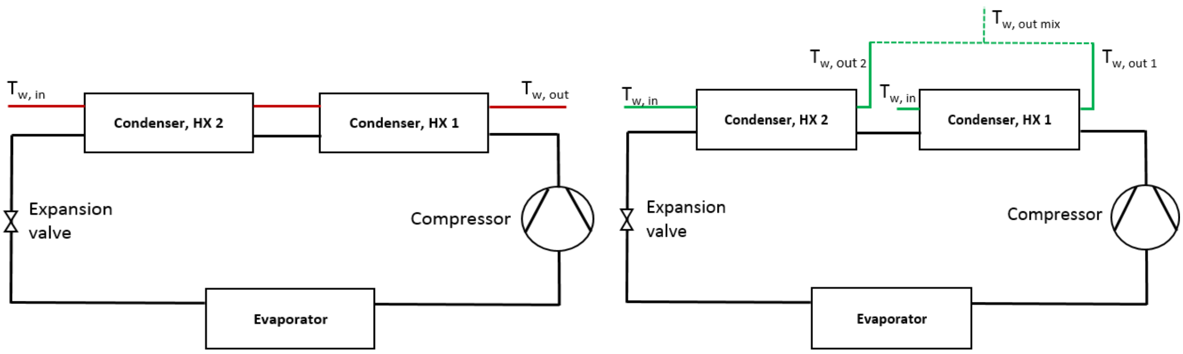

Figure 3: Traditional heat pump configuration (left) and split condenser configuration (right).

The length-to-width ratio of the plate heat exchangers on the high-pressure side of a SCHP was investigated to find the optimal plate dimensions with respect to minimum area of the heat exchangers. Figure 4 indicates that the total heat exchanger area would decrease in line with an increasing length-to-width ratio of the plates. The marginal change in the heat exchanger area was proven to be less significant for heat exchangers with a high length-to-width ratio. In practice, only a limited number of plate dimensions are available and feasible in the production. This was investigated to find the practical potential of a SCHP compared to a THP. Using plates optimized for a SCHP in a THP, the total required heat exchanger area was increased by approximately 100% for the conditions investigated in this study. This indicates that the available plate dimensions influence whether a THP or SCHP is beneficial.

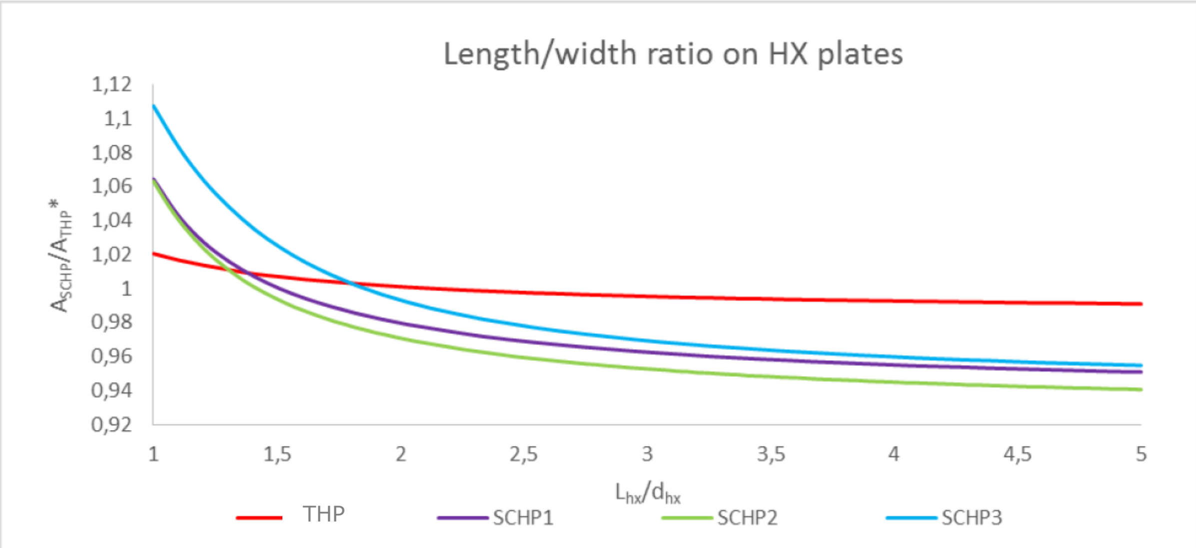

Figure 4: Relative area of plate heat exchangers on the high-pressure side of the heat pump as a function of the length-to-width ratio of the plates for the traditional configuration (THP) and three different split unit configurations (SCHP1, SCHP2, SCHP3).

3.2.3. Optimization and Exergy Analysis

The SCHP setup differs from a traditional heat pump setup in the way that two separate water streams on the secondary side of the condenser are heated in parallel to different temperature levels, whereas only one stream is heated in a traditional heat pump. The comparison between the SCHP and a THP was made for equal heat load and equal total pressure drop on the secondary side. It was found that the SCHP setup offered solutions that resulted in smaller and more compact plate heat exchangers, while assuming the same COP. For a water temperature of 40°C/85°C and an evaporating temperature of 5°C, the total area of the two plate heat exchangers was reduced by 3%. When using the SCHP setup, the exergy destruction was slightly smaller compared to the THP. The SCHP seems to have no significant theoretical benefits in terms of COP and area, but the SCHP can produce two water streams with different temperatures, which might be beneficial for some industrial processes. The SCHP might have practical benefits in terms of COP and area as only a finite number of plate dimensions are available in the production. Figure 5 illustrates the difference in exergy destruction between the THP and the SCHP.

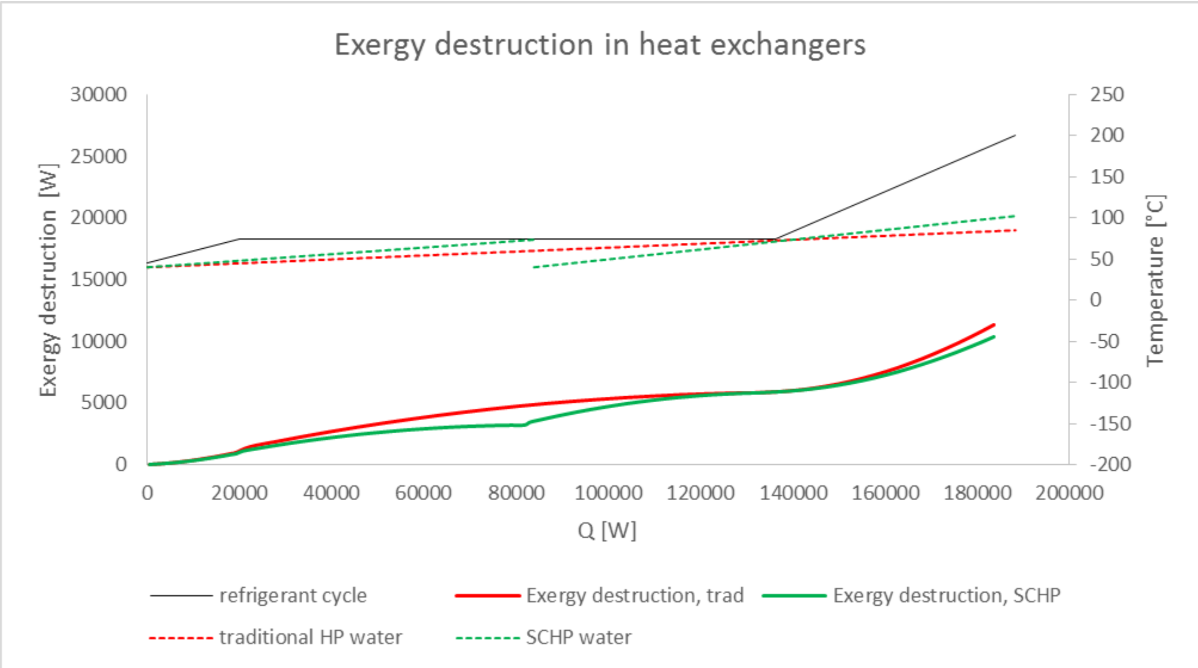

Figure 5: Exergy destruction in the heat exchangers in the heat pump.

3.2.4. Modelling of Ammonia Heat Pump Desuperheaters

This study was concerned with modelling desuperheating in ammonia heat pumps as illustrated in Figure 6.

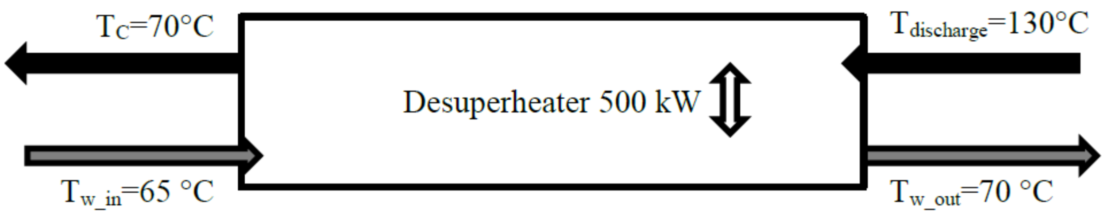

Figure 6: Conceptual sketch of desuperheater used for discretizing.

The focus was on the temperature profile of the superheated refrigerant. Typically, the surface area of a heat exchanger is estimated by using the Log Mean Temperature Difference (LMTD) method. The assumption of this method is that the specific heat is constant throughout the temperature glide of the refrigerant in the heat exchanger. However, when considering ammonia as refrigerant, the LMTD method does not give accurate results due to significant variations of the specific heat. By comparing the actual temperature profiles from a one-dimensional discretized model with the LMTD, it is found that the LMTD method provides a higher temperature difference than the discretized model and would, therefore, lead to an underestimation of the needed condenser area as illustrated in Figure 7. This figure shows the deviation of UA-values (heat transfer coefficient times the area) in dependence of the discharge temperature. There are two ways to compensate for the lower temperature difference in the discretized model. The area of the heat exchanger can be increased, or the condensation temperature can be raised to achieve the same temperature difference for the discretized model as for the

LMTD. This would affect the compressor work - hence, it would affect the COP of the system. Furthermore, for a higher condenser pressure and, thus, a higher pressure in the desuperheater, a large deviation between the two temperature difference models is observed. When using the discretization model, the number of discretizations in order to get accurate estimates is found to be 20.

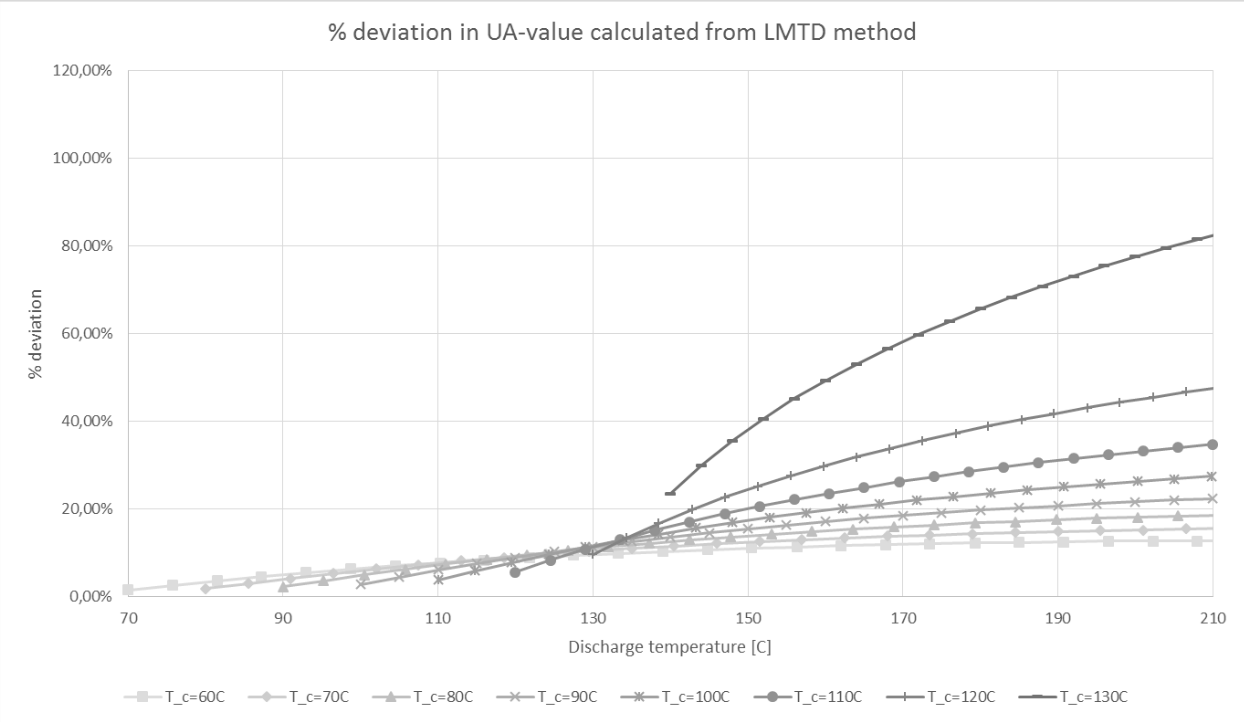

Figure 7. Deviation of UA-values compared to the LMTD method for various conditions of the desuperheater.

3.2.5. Dynamic Simulation of Heat Pumps

The work was focused on the development of a generic model for dynamic operation of heat pump units intended for studying performance during startup, shutdown, and transients. The development was done in collaboration between three different EUDP-projects FOSCAP, SVAF Phase 2, and Energylab Nordhavn. All three projects include elements related to analysis and optimization of dynamic operation for fast response and optimal control. The model was formulated in the Modelica language, and it was implemented in the software Dymola.

3.2.6. Comparison of Heat Pump Design and Performance of Modern Refrigerants

Due to an increasing awareness of global warming, the types of refrigerants used in heat pumps are changing globally. Regulations concerning HFC refrigerants are being introduced due to their high global warming potential (GWP). This can create a shift in the demand for different refrigerants since the HFCs are still commonly used in many countries. As a result, the refrigerant charge will play a significant role when determining the most feasible refrigerant. The analysis was based on a numerical study of the performance of natural refrigerants, HFCs, and HFO refrigerants for a one-stage cycle, and it focused on the influence of the refrigerant charge. The study showed that R717 is the most optimal refrigerant, exhibiting a 51-87% smaller charge and 12-27% lower cost of heat compared to other refrigerants. In addition, the results show that the refrigerant price should be included when conducting economic evaluations.

4. Control Loops

Apart from the basic controls of the heat pump, two new control strategies were investigated. One control strategy, where two temperature levels were obtained, and another control strategy, where a new control strategy for the compressors were developed in order to increase its efficiency. The two temperature levels were implemented in the actual heat pump installation and tested, but the compressor control strategy could not be implemented due to the lack of resources in the project.

4.1. Two Temperature Levels

The purpose of this control method was to utilize heat recovery from the condensing side of a chiller on two temperature levels. The lower temperature level for e.g. floor heating of buildings, and the higher temperature level for e.g. tap water production. Many other applications are available, where two temperature levels are needed. The lower temperature level comes from the condenser (Q2) in Figure 8, and the higher temperature level comes from the desuperheater (Q1) in Figure 8.

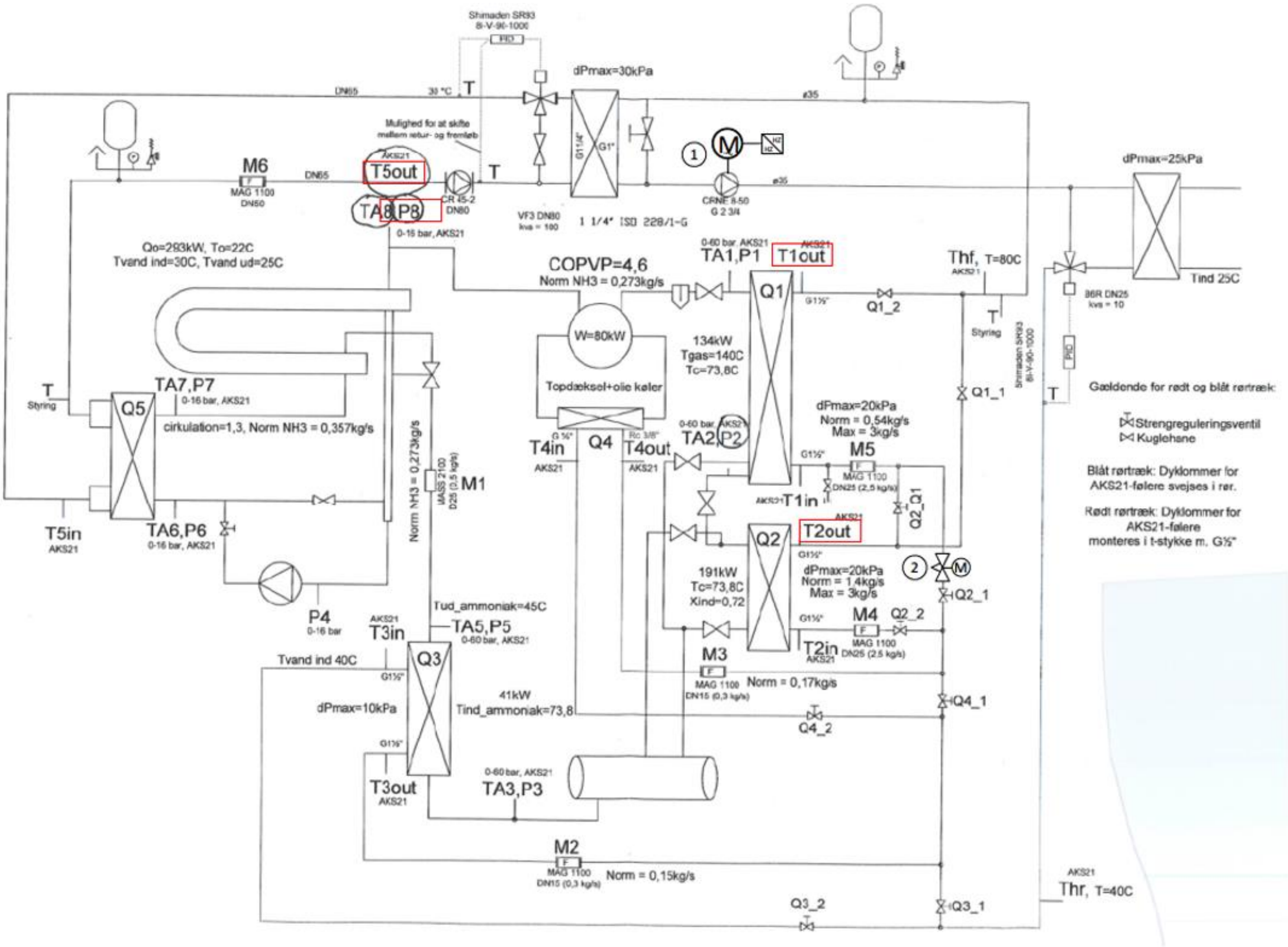

Figure 8: PI diagram of the test heat pump installation.

The lower temperature level coming from the condenser is controlled by a PI regulator according to the water temperature out of the condenser (T2out) by changing the speed (RPM) of the water pumps (1) by means of a frequency converter. When the water temperature is rising above the set point, the pumps are controlled upwards to increase the water flow and vice versa.

The higher temperature level is taken from the desuperheater, and it is controlled by opening and closing valves (2) and by measuring the temperature by means of the sensor T1_out according to the wanted set point. When the temperature rises above the set point, the valve is opened more to increase the water flow and vice versa.

The concern about the control loops was that they would influence each other, and it would be impossible to get a stable control. The control loop was implemented in the test setup, and the gain and the integral time for the PI controllers were adjusted.

A measurement of the temperatures out of the heat exchangers (T1_out and T2_out) along with the temperature into the heat exchanger (T1_In and T2_In) and the set point (in dotted lines) (T1_Out_SP and T2_Out_SP) can be seen in the graph in Figure 9.

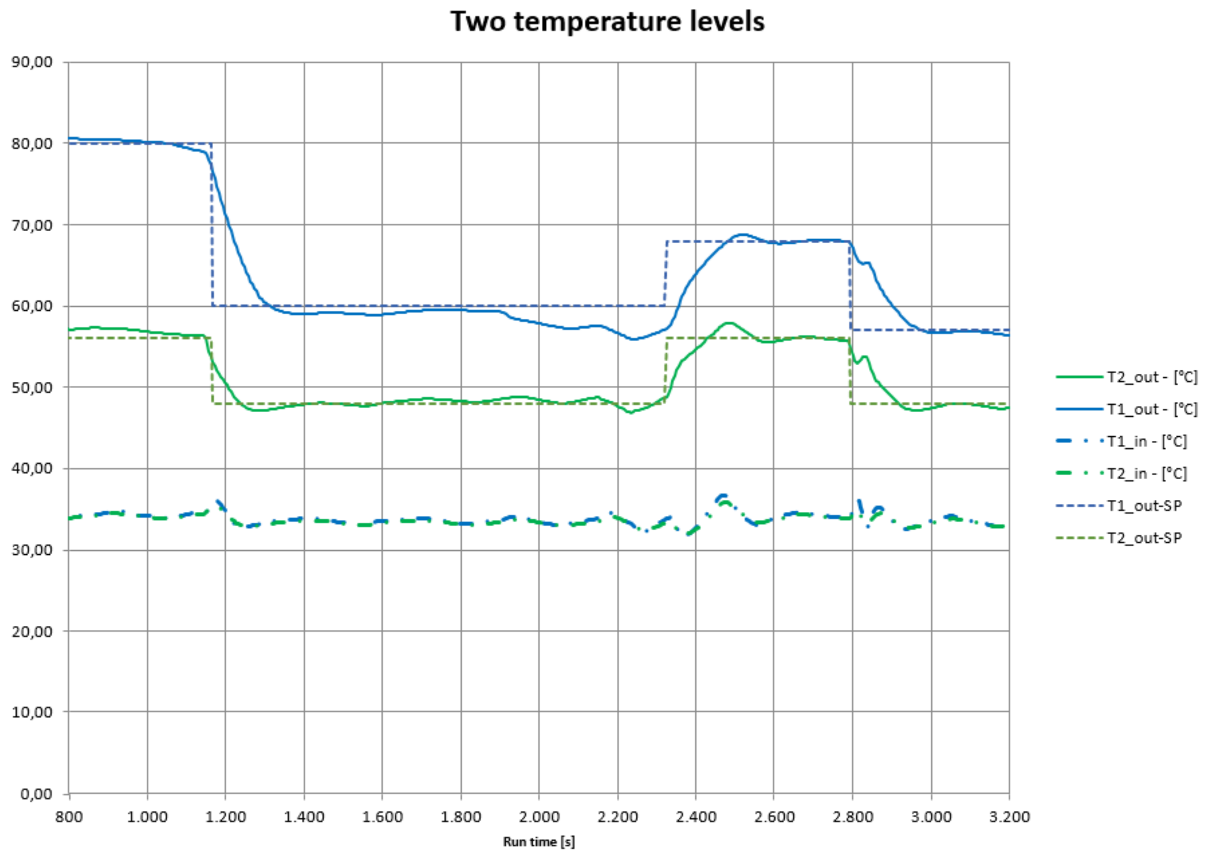

Figure 9: Temperature measurements for control of two temperature levels.

After adjusting the gain and the integral time of the two PI controllers, the system was running with stable conditions until the temperature set point for both control loops was changed around 1170 seconds. The set points were changed from 80°C to 60°C of the water from the desuperheater and from 56°C to 48°C from the condenser. This was a large step change in the set points, but as can be seen from the graph, the controllers managed to adjust the temperatures to the new values within 10 minutes for the desuperheater and 4 minutes for the condenser.

After the first attempt, the control parameters where adjusted again, and another step was performed by changing the set point from 60°C to 68°C for the desuperheater and from 48°C to 56°C for the condenser. Both control loops had a small overshot and then the temperature was adjusted to the required set point in about 3 minutes for both the desuperheater and the condenser.

The graph also shows that the control loops worked satisfactorily even though the loops are interconnected in the test setup through the water pump. The change in water flow from the water pump changes both the water flow to the condenser and the desuperheater. In real situations, the water flow to the desuperheater and the condenser will be uncoupled and coming from two sources. This will make the control easier.

The drifting of the temperature from the desuperheater away from the set point at around 1900 seconds is caused by a change in the control parameters.

4.2. Compressor

The capacity change of the compressor is made by both changing the frequency of the electrical motor and by coupling pistons in and out. The frequency of the motor can be changed from 25 to 60 Hz, which changes the speed of the compressor from 750 RPM to 1800 RPM. The compressor has six pistons. Four of these can be coupled in and out in a set of two. When increasing the capacity, two pistons can be activated, i.e. it is possible to go from two to four and later from four to six.

The original control of the heat pump unit was to start by coupling in the pistons and subsequently to activate the frequency control, when all the pistons were active. A piston compressor has its highest efficiency at the lowest possible speed for the required capacity. This is because the mechanical losses increase with higher speeds, which decrease the isentropic efficiency of the compressor. Another challenge with the original control method is when the wanted capacity is lower than the lowest achievable capacity on the frequency controller. In this case, the heat pump is solely controlled by shifting pistons in and out, which gives a large deviation in the controlled temperature since a large capacity is shifted in and out.

To increase the efficiency of the compressor, another control strategy was suggested. The idea is to run the compressor on the lowest possible speed for the required capacity. This is done by utilizing both the piston in and out coupling and the frequency drive. The capacity control starts with two active pistons on the lowest frequency state 1 shown in Figure 10. Then, as the load requirements rise, the frequency control increases the RPM of the compressor until the maximum frequency is obtained at state 2. If a higher capacity is needed, the control couples two extra pistons in so that the compressor is now running with four pistons. At the same time, the frequency is reduced to around 30 Hz to capture the same capacity as before at state 2. This puts the capacity point shown in Figure 10 at state 3.

In connection with increased capacity requirements, the frequency drive increases the RPM up to 45 Hz, i.e. state 4 in the graph. From here, the last two pistons are coupled in and the frequency is reduced to 30 Hz at state 5. From this point and to the point where the maximum capacity is reached, the frequency is increased to 60 Hz (state 6).

When reducing the capacity requirements from maximum, the reverse sequence is activated. The compressor goes from state 6 to state 7 or 25 Hz. Subsequently, two pistons are deactivated, and the frequency is increased to 38 Hz or state 8 in the graph. On further reduction, the compressor is reduced to state 9 before changing to state 10, and finally from there down to state 1.

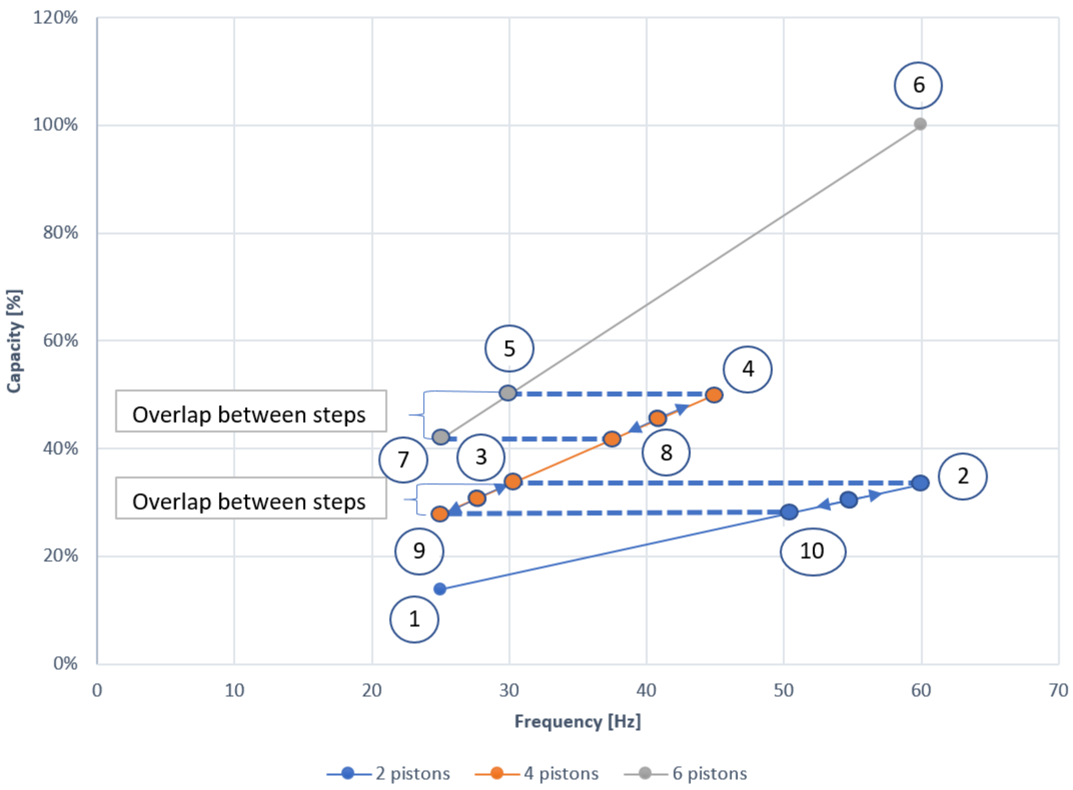

Figure 10: Compressor capacity control.

The overlaps shown in the graph from state 2 to 10, 3 to 9, 4 to 8, and 5 to 7 are used to ensure that the control of the compressor is stable. In connection with the capacity requirements that lies in-between two states, e.g. state 2 and 3, the compressor would oscillate with pistons coupled in and out constantly. To overcome this, the overlaps are used. When the compressor changes state, e.g. from state 2 to 3, the capacity requirements must drop down to state 9 before the pistons can be coupled out again.

This control method gives smooth control in all the control areas from minimum capacity to maximum capacity.

The plan was to implement this control logic in the compressor controller, but due to the lack of resources this was not done.

4.3. Split Condenser, Serial and Parallel Test

In this section, the results from the heat pump test with a serial (A) and parallel (B) coupling of the condenser are discussed. The comparison is based on two test runs.

In test run B, the purpose was to document how the condensation temperature and the performance of the two condensers vary as a function of the distribution of the liquid through the first section (PHE1 (Q1)) and the second (PHE2 (Q2)) condenser. See Figure 8.

In test run A, the aims were to document the changes in the distribution of the performance of PHE1 and PHE2 and the changes in the condensation temperature by reducing the mass flow through the heat exchangers PHE1 and PHE2.

Table 1 and 2 show two situations, one situation with a serial coupling and one with a parallel coupling on the water side. As it can be seen, the COP is larger with a serial coupling than when using a parallel coupling.

However, the test conditions in test run B are not optimal in order to obtain the lowest possible COP, e.g. regarding the distribution of flows between the Q1 and Q2. Therefore, this test is not sufficient in order to conclude that serial coupling is better than parallel coupling.

Table 3 shows the results from test B, where the distribution of the mass flows through Q1 and Q2 is varied. As shown in the Table, the COP is dependent on the ratio between the mass flows. Even if the sum of the mass flows M4 and M5 is not constant in all the test runs, the COP will have a maximum value around 4.8.

What to keep in mind when interpreting the results of COP is that each of the two heat exchangers (Q1 and Q2) is designed to manage the entire capacity alone.

Legend for Table 1, 2, 3 and 4:

M_i: Mass flow at point i (l/s)

M_6: Volume flow (m^3/h)

P_sup: Power (kW)

phi_Q1w: Heat transfer rate (kW)

COP: Coefficient of Performance of heat pump ( )

P_i: Pressure at point i (bara)

Ti: Temperature at point i (°C)

TAi: Temperature of ammonia at point i (°C)

Test scheme A, serial coupling,
(run time from Excel spreadsheet:
Average in the interval 720 - 740)

	Start	End	Parameter	Min.	Max.	Std. dev.	Average
Run time [s]:	3602	3750	M_1 - [kg/s]	0,24	0,31	0,02	0,28
Row #:	720	740	M_2 - [l/s]	0,16	0,16	0,00	0,16
			M_3 - [l/s]	0,056	0,056	0,000	0,056
Samples:	20		M_4 - [l/s]	2,22	2,23	0,00	2,23
			M_5 - [l/s]	2,26	2,27	0,00	2,26
			M_6 - [m³/h]	27,0	27,2	0,1	27,1
			P_sup - [kW]	75,7	77,4	0,5	76,7
			phi_Q1w - []	86,4	89,7	1,3	88,0
			phi_Q2w - []	251,8	256,1	1,4	253,8
			phi_Q3w - []	21,5	21,8	0,1	21,7
			phi_Q4w - []	4,48	4,59	0,04	4,54
			phi_Q5w - []	307,1	313,2	1,8	309,6
			COP - []	4,72	4,90	0,06	4,80
			P_1 - [bara]	31,5	32,1	0,2	31,8
			P_2 - [bara]	31,4	32,0	0,2	31,7
			P_3 - [bara]	31,3	31,9	0,2	31,6
			P_5 - [bara]	31,3	31,9	0,2	31,6
			P_8 - [bara]	9,14	9,38	0,09	9,26
			T2_out - [°C]	65,0	65,5	0,2	65,3
			T_hf - [°C]	74,6	75,3	0,2	74,9
			T_hr - [°C]	34,5	35,1	0,2	34,9
			T1_out - [°C]	74,9	75,3	0,1	75,1
			T1_in - [°C]	65,5	66,1	0,2	65,9
			T2_in - [°C]	37,9	38,3	0,2	38,1
			T3_out - [°C]	66,5	66,8	0,1	66,7
			T3_in - [°C]	34,9	35,3	0,2	35,1
			T4_out - [°C]	57,3	57,4	0,0	57,4
			T4_in - [°C]	37,8	38,2	0,1	38,0
			T5_out - [°C]	25,0	25,3	0,1	25,2
			T5_in - [°C]	34,9	35,0	0,0	35,0
			T_disch - [°C]	127,5	127,9	0,1	127,7
			TA_1 - [°C]	125,8	126,1	0,1	125,9
			TA_2 - [°C]	67,9	68,4	0,2	68,2
			TA_3 - [°C]	66,9	67,3	0,1	67,1
			TA_5 - [°C]	49,3	52,2	1,1	51,1
			TA_liq - [°C]	57,7	58,2	0,2	57,9
			TA_8 - [°C]	22,6	23,1	0,2	22,9
			TA_gas - [°C]	23,3	23,8	0,2	23,6
			T_sat-suct - []	22,0	22,8	0,3	22,4
			T_sat-disc - []	67,9	68,6	0,3	68,2

Table 1. Results from test scheme A with serial coupling of condensers on the warm water side. Legend for this table is shown above the table.

Test scheme B, parallel coupling,
(run time from Excel spreadsheet:
Average in the interval 1500 - 3500)

	Start	End	Parameter	Min.	Max.	Std. dev.	Average
Run time [s]:	1998	3000	M_1 - [kg/s]	0,23	0,31	0,03	0,28
Row #:	381	581	M_2 - [l/s]	0,17	0,17	0,00	0,17
			M_3 - [l/s]	0,06	0,06	0	0,057
Samples:	200		M_4 - [l/s]	1,79	1,80	0,00	1,80
			M_5 - [l/s]	0,47	0,47	0,00	0,47
			M_6 - [m³/h]	27,0	27,3	0,0	27,2
			P_sup - [kW]	77,4	79,5	0,5	78,5
			phi_Q1w - []	108,6	111,8	0,8	110,3
			phi_Q2w - []	218,5	223,4	1,3	220,7
			phi_Q3w - []	23,1	23,4	0,1	23,3
			phi_Q4w - []	4,57	4,70	0,03	4,63
			phi_Q5w - []	303,6	313,6	2,5	308,4
			COP - []	4,50	4,66	0,04	4,57
			P_1 - [bara]	32,4	33,3	0,2	32,9
			P_2 - [bara]	32,4	33,2	0,2	32,8
			P_3 - [bara]	32,3	33,1	0,2	32,7
			P_5 - [bara]	32,3	33,1	0,2	32,7
			P_8 - [bara]	9,18	9,46	0,08	9,33
			T2_out - [°C]	67,1	67,9	0,2	67,5
			T_hf - [°C]	73,2	74,2	0,3	73,8
			T_hr - [°C]	34,6	35,1	0,1	34,9
			T1_out - [°C]	93,8	94,7	0,2	94,4
			T1_in - [°C]	37,9	38,5	0,1	38,2
			T2_in - [°C]	38,0	38,4	0,1	38,2
			T3_out - [°C]	67,8	68,2	0,1	67,9
			T3_in - [°C]	34,9	35,2	0,1	35,1
			T4_out - [°C]	57,4	57,6	0,0	57,5
			T4_in - [°C]	38,0	38,3	0,1	38,1
			T5_out - [°C]	25,2	25,5	0,1	25,4
			T5_in - [°C]	35,0	35,2	0,0	35,1
			T_disch - [°C]	129,8	130,7	0,2	130,2
			TA_1 - [°C]	128,0	128,3	0,1	128,1
			TA_2 - [°C]	69,2	70,0	0,2	69,6
			TA_3 - [°C]	68,2	68,9	0,2	68,5
			TA_5 - [°C]	49,3	52,9	1,1	51,6
			TA_liq - [°C]	58,0	58,6	0,2	58,3
			TA_8 - [°C]	22,7	23,4	0,2	23,0
			TA_gas - [°C]	23,4	24,0	0,2	23,7
			T_sat-suct - []	22,1	23,1	0,3	22,6
			T_sat-disc - []	69,1	70,2	0,3	69,7

Table 2. Results from test scheme B with parallel coupling of condensers on the warm water side. Legend for this table is shown above the table.

In addition, the heat exchangers are identical regarding both the height and the width and they both use the same types of channels. However, they use different numbers of plates. In the test setup, the number of plates of the second heat exchangers, Q2, is 56, and the number of plates in the first heat exchanger, Q1, is 130.

Thus, the heat exchanges are not optimized for an optimal split. They are composed so that it was possible to switch from full capacity in Q1 and gradually change the run of the water and, thus, the capacity in Q2 in order to end up by having a full capacity in Q2. Therefore, it has been possible to compare the conditions, which only deal with the number of plates, and thereby neglect the influence of the plate geometry.

Figure 11 shows the change in the energy transferred to Q1 and Q2 as a function of the distribution of the water flow with a parallel coupling. The results are taken from Table 3.

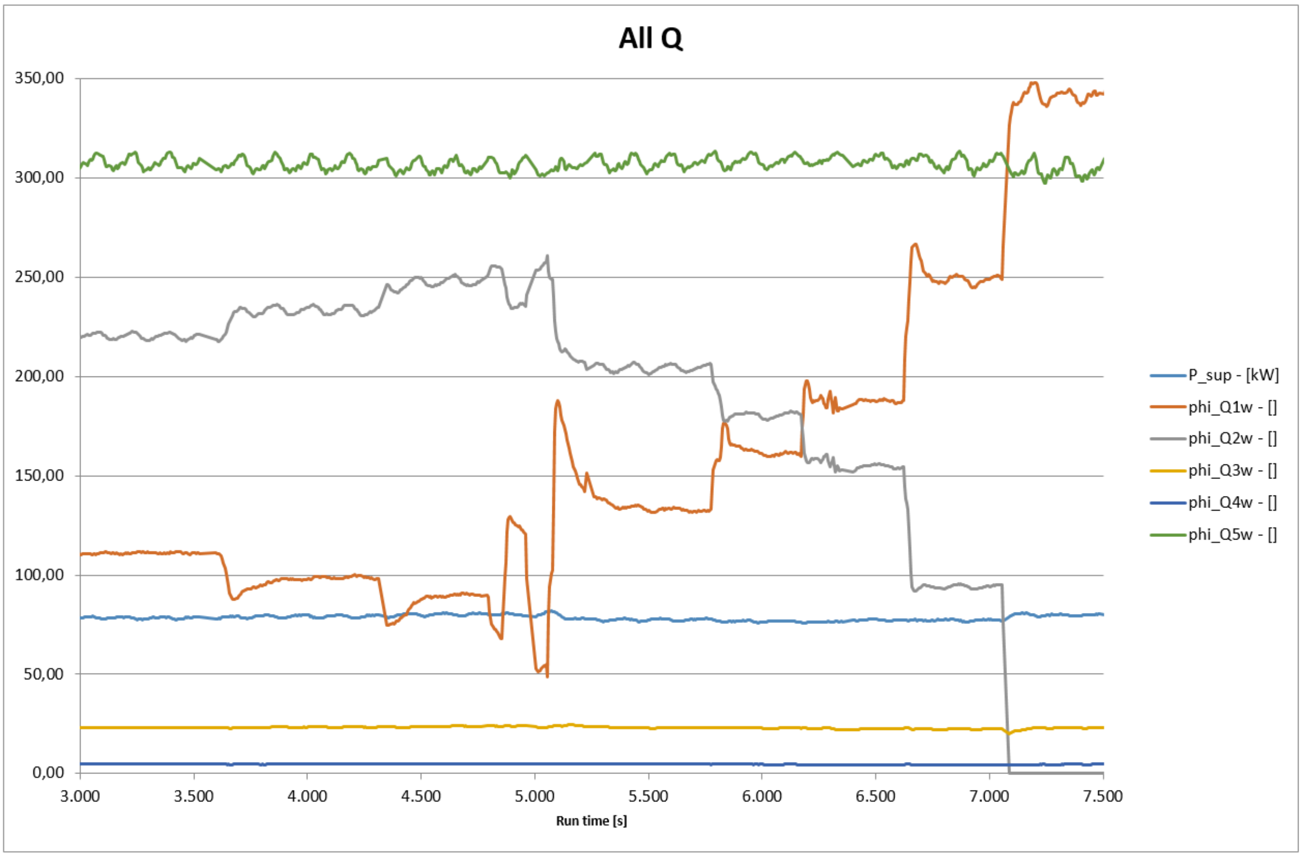

Figure 11. Measurement of the energy transferred in the different heat exchangers in the heat pump. The configuration can be seen from Figure 8. Test Scheme B

Run time	3000 til 3500	3951 til 4200	4614 til 4780	5551 til 5700	6001 til 6150	6451 til 6600	6848 til 7000	7125 til 7200
Parameter	Average	Average	Average	Average	Average	Average	Average	Average
M_1 - [kg/s]	0,2847	0,2755	0,2805	0,2787	0,2860	0,2797	0,2838	0,2581
M_2 - [l/s]	0,1691	0,1671	0,1657	0,1716	0,1738	0,1684	0,1669	0,1581
M_3 - [l/s]	0,0571	0,0564	0,0559	0,0577	0,0583	0,0565	0,0559	0,0530
M_4 - [l/s]	1,8012	1,8801	1,9552	1,6979	1,5146	1,2563	0,7422	-0,0002
M_5 - [l/s]	0,4698	0,3656	0,2657	0,5954	0,8050	1,0028	1,4942	2,1266
M_5 + M_4	2,2710	2,2457	2,2209	2,2933	2,3196	2,2592	2,2364	2,1264
M_6 - [m3/h]	27,1674	27,1287	27,0822	27,1966	27,1973	27,1588	27,1044	26,9453
P_sup - [kW]	78,3987	79,2705	80,1970	77,3769	76,6954	77,3628	77,1225	80,4783
phi_Q1w - []	111,0557	98,4937	90,1052	133,0256	160,8753	187,8039	248,2582	343,3838
phi_Q2w - []	220,3522	233,8855	248,1308	204,4447	180,4009	154,9646	94,1253	-0,0235
phi_Q2W + Q1W	331,4080	332,3792	338,2360	337,4703	341,2761	342,7685	342,3835	343,3603
phi_Q3w - []	23,2038	23,4160	23,8162	23,0322	22,8215	22,4773	22,3805	22,3149
phi_Q4w - []	4,6342	4,6632	4,7420	4,6446	4,5595	4,4922	4,4705	4,4254
phi_Q5w - []	307,8733	307,4897	305,8126	308,1670	308,3863	308,9983	307,3497	306,2754
COP - []	4,5825	4,5475	4,5739	4,7192	4,8068	4,7794	4,7877	4,5991
Q5w+P_sup - []	386,2719	386,7602	386,0096	385,5439	385,0818	386,3611	384,4722	386,7537
Q1w+Q2w+Q3w+Q4w -	359,2459	360,4583	366,7942	365,1470	368,6571	369,7381	369,2345	370,1006
P_1 - [bara]	32,8004	33,3330	33,9520	32,2275	31,7566	32,1855	32,0960	34,0889
P_2 - [bara]	32,7402	33,2700	33,8846	32,1649	31,6982	32,1296	32,0462	34,0381
P_3 - [bara]	32,6202	33,1475	33,7618	32,0514	31,5871	32,0249	31,9520	33,9837
P_5 - [bara]	32,6395	33,1667	33,7804	32,0715	31,6072	32,0435	31,9691	34,0041
P_8 - [bara]	9,3133	9,3350	9,3552	9,2471	9,2597	9,2684	9,2463	9,3936
T2_out - [°C]	67,4145	67,9567	68,6424	66,8495	66,5157	67,6134	68,4180	68,3547
T_hf - [°C]	73,7130	74,1425	74,6643	73,1107	72,7305	73,5284	73,9517	76,6884
T_hr - [°C]	34,8868	34,8501	34,8738	34,8342	34,8694	34,9522	34,8994	34,9423
T1_out - [°C]	94,6316	102,5450	119,2467	91,4347	85,8005	82,9277	77,8575	76,9968
T1_in - [°C]	38,2121	38,2513	38,3121	38,1113	38,1044	38,2319	38,2042	38,4588
T2_in - [°C]	38,2180	38,2662	38,3537	38,1123	38,0897	38,1749	38,1510	38,1413
T3_out - [°C]	67,8368	68,5063	69,3831	67,0656	66,4232	66,9575	67,0885	68,9164
T3_in - [°C]	35,0779	35,0717	35,0706	35,0395	35,0829	35,1062	35,0794	35,2379
T4_out - [°C]	57,4957	57,9311	58,4885	57,2598	56,7292	57,0461	57,1911	58,2611
T4_in - [°C]	38,1322	38,1918	38,2497	38,0644	38,0579	38,0629	38,0895	38,3149
T5_out - [°C]	25,3714	25,4109	25,4999	25,3319	25,3078	25,3974	25,4313	25,5527
T5_in - [°C]	35,1081	35,1493	35,2018	35,0674	35,0500	35,1728	35,1739	35,3185
T_disch - [°C]	130,1294	131,3075	132,8886	129,1370	127,8032	128,4506	128,6254	131,7446
TA_1 - [°C]	128,1679	129,1774	130,6203	127,5606	126,3271	126,4353	126,6292	128,8340
TA_2 - [°C]	69,4987	70,2009	71,0040	68,7685	68,0814	68,6578	68,5627	69,7852
TA_3 - [°C]	68,4316	69,1238	69,9641	67,7048	67,0138	67,6082	67,6453	70,1689
TA_5 - [°C]	51,5256	51,9457	52,6500	50,6405	50,6039	50,9581	51,1103	53,3083
TA_liq - [°C]	58,2341	59,0824	59,9338	57,1452	56,4668	56,9425	57,5927	64,8049
TA_8 - [°C]	22,9969	23,1053	23,1433	22,8091	22,7946	22,8802	22,7983	23,3108
TA_gas - [°C]	23,6331	23,7346	23,7788	23,4585	23,4176	23,5149	23,4539	23,8831
T_sat-suct - []	22,5881	22,6625	22,7322	22,3596	22,4035	22,4337	22,3567	22,8637
T_sat-disc - []	69,5930	70,2939	71,0975	68,8291	68,1936	68,7730	68,6526	71,2741
(T_sat-disk) - (T2_out)	2,1786	2,3372	2,4551	1,9796	1,6779	1,1596	0,2346	2,9194
(T_sat-disk) - (T_hf)	4,1200	3,8486	3,5668	4,2816	4,5369	4,7554	5,2991	5,4144

Table 3. Results from test scheme B with parallel coupling of condensers on the warm water side. (run time from Excel spreadsheet: Average in the 8 different intervals). Legend for this table is shown above Table 1.

As previously mentioned, test A has been performed by having changing flows in serial coupling of the condensers with a constant compressor capacity. The changes are shown in Figures 12 and 13, and the associated test results are shown in Table 4.

Only the number of revolutions on the pump are changed. However, as shown in Figure 13, the mass volume flow M6 has decreased slightly during the period. This is due to the fact that the warm and the cold side of the heat pump are connected in such a way that only the power of the compressor is removed from the system.

As seen from Figure 13, the water flow through the evaporator Q5 is only slightly reduced. The reason for this is the three-way valve, which function is to supply a constant return temperature to the evaporator (T5_in), which is influenced by changes in the water flow on the warm side.

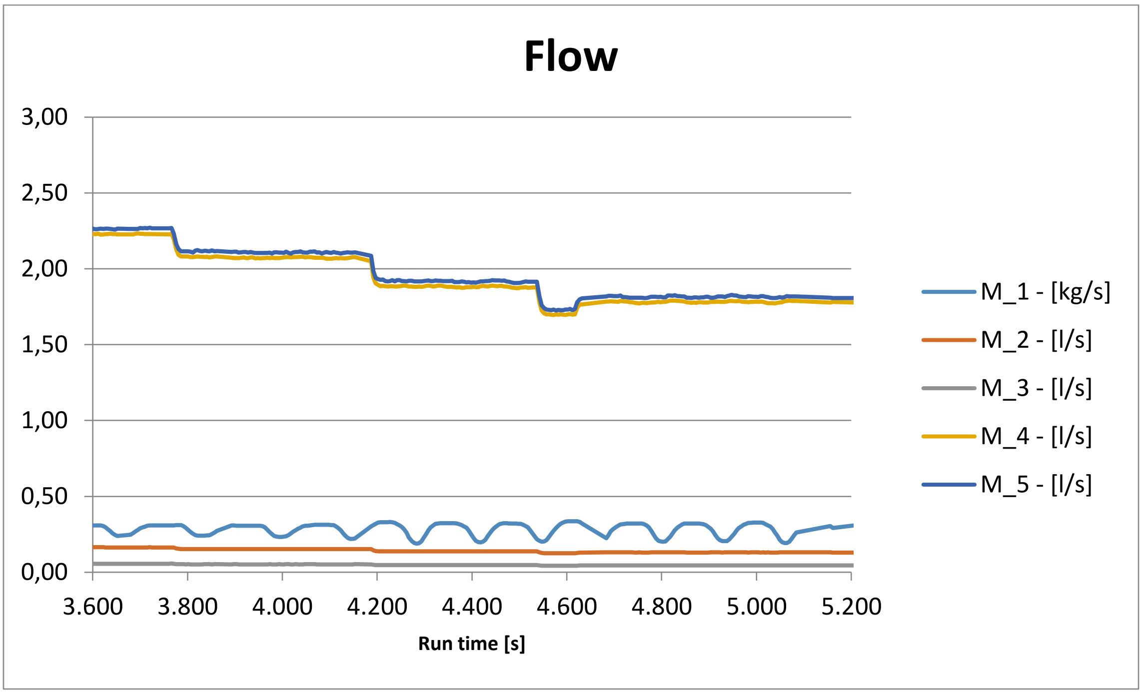

Figure 12: Measurement of mass flows through the different sections during test scheme A. M_5 is the mass flow through Q1, and M_4 is the mass flow through Q2.

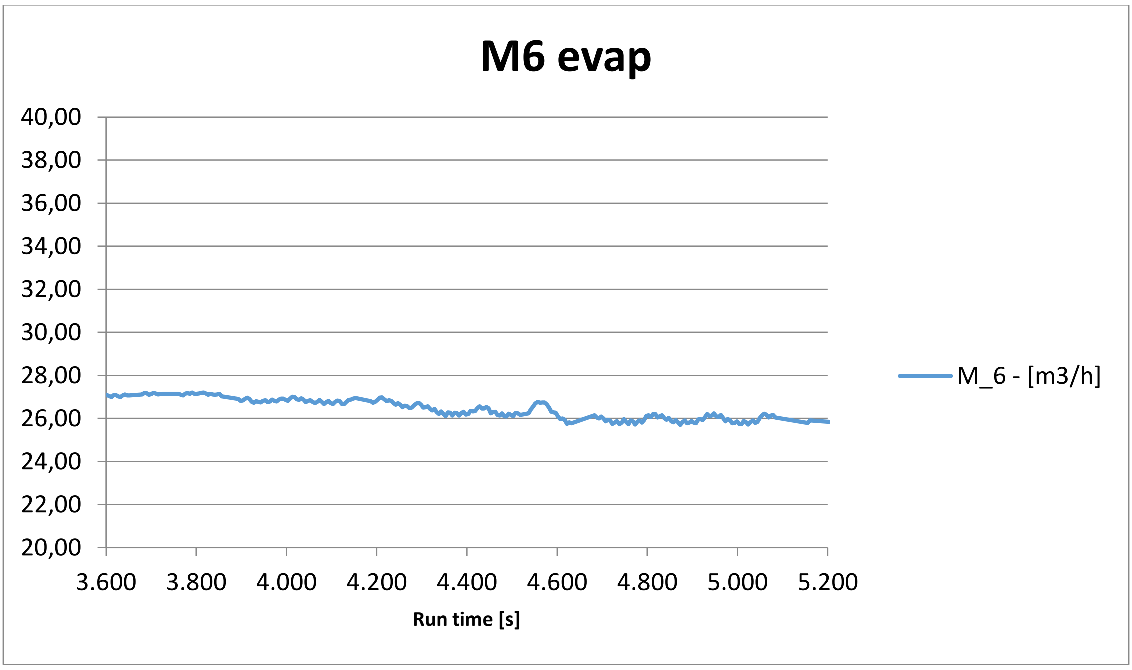

Figure 13: Measurement of volume flow M_6 through the evaporator Q5 during test scheme A.

In this case, the evaporation temperature rises and does not fall as it would be expected by a decreasing water flow. This is because the cooling load is reduced at the same time. As the condensation load is decreasing due to the increasing condensation temperature, the consequence is an increase in the evaporation temperature. It is also apparent from Table 4 that the mass flow of ammonia (M_1) is reduced due to the high condensation temperature, which also results in a high pressure-gas temperature (TA_1). The high-pressure gas temperature is favourable in this situation as it helps to limit the increase of the condensation temperature.

Figure 14 (PHE, subcooler incl. water) shows the temperatures around the subcooler in the same periods as in the previous figures. It is should be noted that TA_liq, which is the liquid temperature at the exit of section two, Q2, is lower than the saturation temperature, T_sat disc, which means the liquid is subcooled.

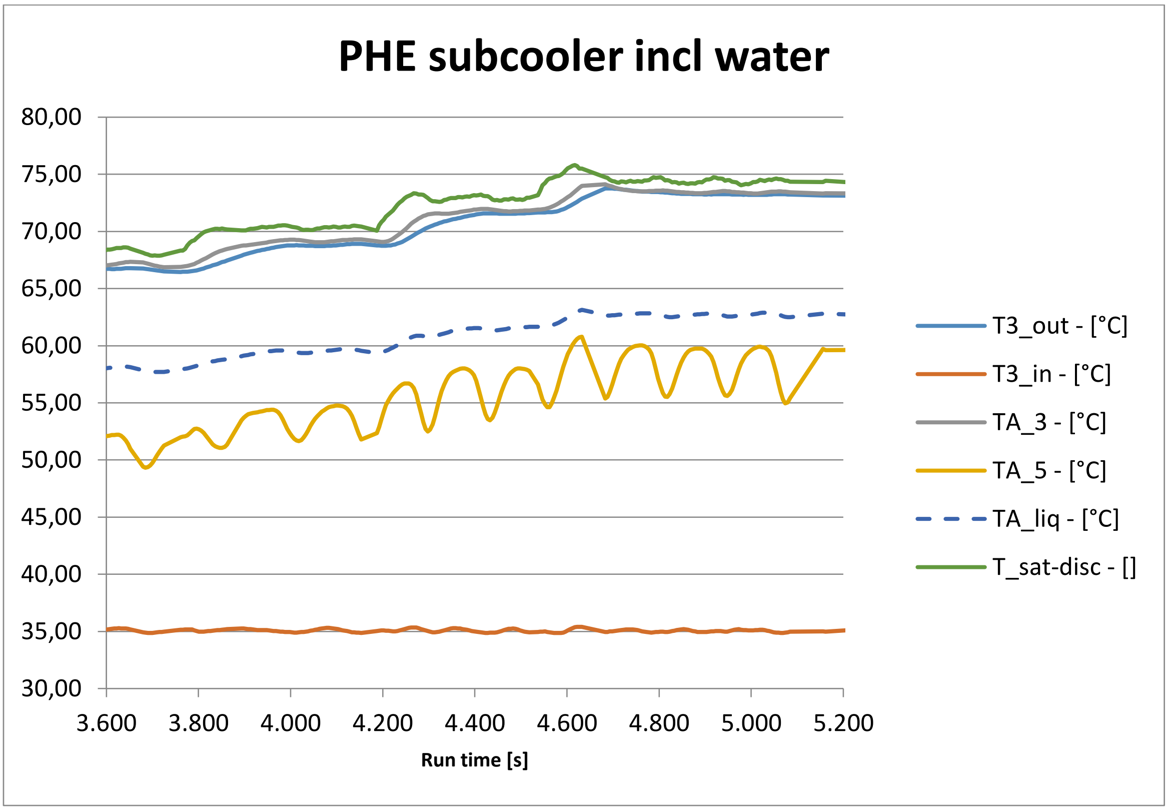

Figure 14: Measurement of temperatures during test scheme A.

Run time	3602 - 3750	4048 - 4150	4398 -4500	4898 - 5000
Parameter	Average	Average	Average	Average
M_1 - [kg/s]	0,29	0,29	0,27	0,27
M_2 - [l/s]	0,16	0,15	0,14	0,13
M_3 - [l/s]	0,06	0,05	0,05	0,05
M_4 - [l/s]	2,23	2,07	1,88	1,78
M_5 - [l/s]	2,26	2,11	1,92	1,82
M_6 - [m³/h]	27,1	26,8	26,3	26,0
P_sup - [kW]	76,7	79,4	82,3	84,0
phi_Q1w - []	88,0	84,3	81,3	81,5
phi_Q2w - []	253,8	254,2	253,5	251,2
phi_Q3w - []	21,7	21,5	21,1	20,9
phi_Q4w - []	4,5	4,5	4,6	4,8
phi_Q5w - []	309,6	305,7	301,0	296,6
COP - []	4,80	4,59	4,38	4,27
EER - []	4,04	3,85	3,66	3,53
P_1 - [bara]	31,8	33,4	35,4	36,6
P_2 - [bara]	31,7	33,3	35,3	36,5
P_3 - [bara]	31,6	33,2	35,2	36,4
P_5 - [bara]	31,6	33,2	35,2	36,4
P_8 - [bara]	9,3	9,4	9,4	9,4
T2_out - [°C]	65,3	67,6	70,7	72,4
T_hf - [°C]	74,9	77,6	81,2	83,5
T_hr - [°C]	34,9	34,9	34,8	34,9
T1_out - [°C]	75,1	77,7	81,3	83,6
T1_in - [°C]	65,9	68,2	71,2	72,9
T2_in - [°C]	38,1	38,3	38,5	38,7
T3_out - [°C]	66,7	68,8	71,6	73,2
T3_in - [°C]	35,1	35,1	35,0	35,1
T4_out - [°C]	57,4	58,9	61,5	64,0
T4_in - [°C]	38,0	38,2	38,3	38,4
T5_out - [°C]	25,2	25,5	25,5	25,4
T5_in - [°C]	35,0	35,3	35,3	35,2
T_DX - [°C]	24,4	24,5	24,5	24,5
T_riser - [°C]	23,4	23,8	23,8	23,9
T_disch - [°C]	127,7	130,8	135,9	139,3
TA_1 - [°C]	125,9	128,4	132,6	136,5
TA_2 - [°C]	68,2	70,2	72,8	74,3
TA_3 - [°C]	67,1	69,2	71,9	73,4
TA_5 - [°C]	51,1	54,1	56,0	57,8
TA_liq - [°C]	57,9	59,6	61,5	62,7
TA_8 - [°C]	22,9	23,3	23,3	23,4
TA_gas - [°C]	23,6	23,9	24,0	24,0
T_sat-suct - []	22,4	22,9	22,9	22,9
T_sat-disc - []	68,2	70,4	72,9	74,4

Table 4. Results from test scheme A with serial coupling of condensers. (run time from Excel spreadsheet: Average in the 4 different intervals). Legend for this table is shown above Table 1.

Throughout the whole test period, the liquid is subcooled in the range of 10 to 12 K. The sinusoidal shaped variation of the temperature TA_5 is caused by the fluctuation of the expansion valve control device. There is a relatively good coincidence between the variation of TA_5 in Figure 14 and the variation of the ammonia flow, M_1, in Figure 12.

The temperature of the water from the subcooler T3_out (Figure 14) is very close to the condensation temperature T_sat_disc, which indicates that either the heat exchanger Q3 is unnecessarily large for the task or the mass flow of water, M_2, is set too low.

If the water flow M_2 had been larger, it would have generated a larger LMTD (logarithmic mean temperature difference), and thereby a lower TA_5. As shown in Figure 14 (PHE subcooler incl. water), the test is run with a temperature difference between T3_in and TA_5 of approximately 20 to 25 K. A larger flow on M_2 would have reduced this temperature difference and resulted in a larger COP.

Another interesting observation is the temperature of ammonia TA_3, which is the temperature measured at the output of the high-pressure receiver. This measured temperature is in line with the measured saturation temperature T_sat_disc. This indicates that an amount of liquid of ammonia is damming up in Q2, which is probably because the valve in the bypass over Q2 has been open.

This means that saturated gas flows through the bypass to the receiver in an amount corresponding to the condensation energy and the energy required to bring the subcooled liquid back to the saturated temperature at the corresponding pressure in the receiver. Unfortunately, the result is a reduced condensation area, which otherwise could have been used to reduce the condensation temperature further down.

5. Split Condenser and Design Considerations

This report gives a brief description of the split condenser concept and its design considerations. It also shows the potential benefits using this concept as well as new products developed by Alfa Laval in order to meet market demands of high temperature ammonia heat pumps. Finally, the report provides an example of a commercial installation in an industry in Denmark, where the benefits of the concept is demonstrated.

5.1. Split Condenser Concept

The concept of having two condensers instead of a single condenser is often used in situations, where there is a need for de-superheating and/or subcooling. Particularly the benefits of splitting the condenser duty in more than one unit can be seen in high temperature ammonia heat pump applications, where the temperature rise on the water side is high. In such applications, the superheat of the ammonia is normally high as well, which requires a quite large de-superheating zone in the condenser or a separate desuperheater. A standard basic layout of a heat pump can be seen in Figure 15.

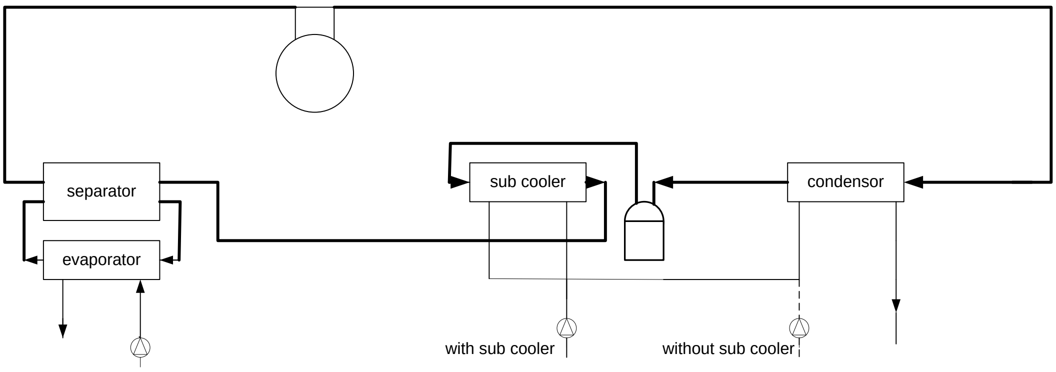

Figure 15. Basic layout of standard heat pump with subcooler and condenser.

Separating the condenser and the desuperheater into two units, where the condensing takes place in the condenser and the de-superheating part takes place in the desuperheater. Here, the thermal and the hydraulic characteristics of the two heat exchangers can be adopted to the different thermal duties, and various heat exchangers can be selected to have the best fit. Such a layout can be seen in Figure 16.

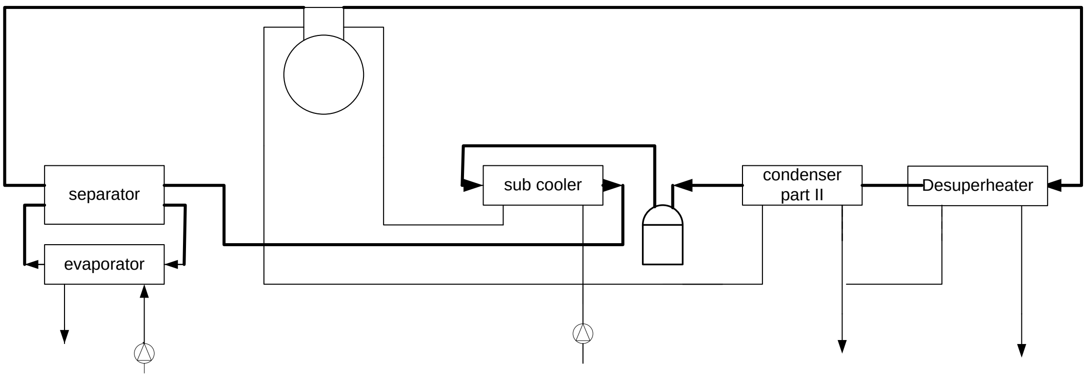

Figure 16: Basic layout of heat pump with subcooler, condenser, and desuperheater.

When making a split condenser design, the difference between the layout above and the split condenser layout is merely the fact that the cold inlet water is distributed to both the condenser and the desuperheater, see Figure 17.

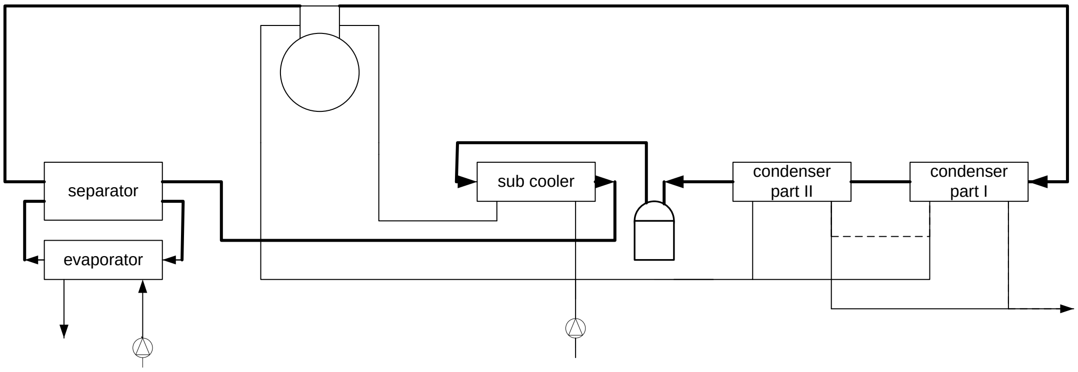

Figure 17: Basic layout of heat pump with subcooler and split condenser.

In this case, the larger temperature difference in the desuperheater can now be used to enable substantial condensing in the desuperheater as well. The benefits of choosing this kind of design could be several.

1. It is possible to obtain lower condensing temperatures
2. It is possible to have a large quantity of outlet water with high temperatures
3. In many cases, it is possible to use less heat transfer areas and/or smaller heat exchangers than is the case for a single unit.

5.2. Lower Condensing Temperature

For large temperature glides on the water side, it is possible to reduce the condensing temperature to such an extent that it is possible to have a lower condensing temperature than the outlet water temperature. This is shown in the example below. The inlet water temperature is 40°C, the outlet water temperature is 80°C and the inlet refrigerant temperature is 140°C. A standard design point for a single condenser would require a condensing temperature about 82°C. However, it is possible to achieve lower condensation temperatures as well. In the example below, a theoretical comparison of a single unit design with that of a split condenser design is shown.

For the FOSCAP heat pump at DTI, a condensing power of approximately 319 kW is achieved.

Figure 18 shows the design of a single unit. The design requires 186 plates in order to do the duty at a minimum condensing temperature of 75.1°C. It should be noted that this design is purely theoretical as the maximum number of plates, which the AlfaNova76 unit can have, is 150 plates. In practical terms, this means that one needs two units in parallel or a bigger and more expensive unit. One such realistic alternative would be a high pressure semi-welded heat exchanger. A comparative design is shown in Figure 19.

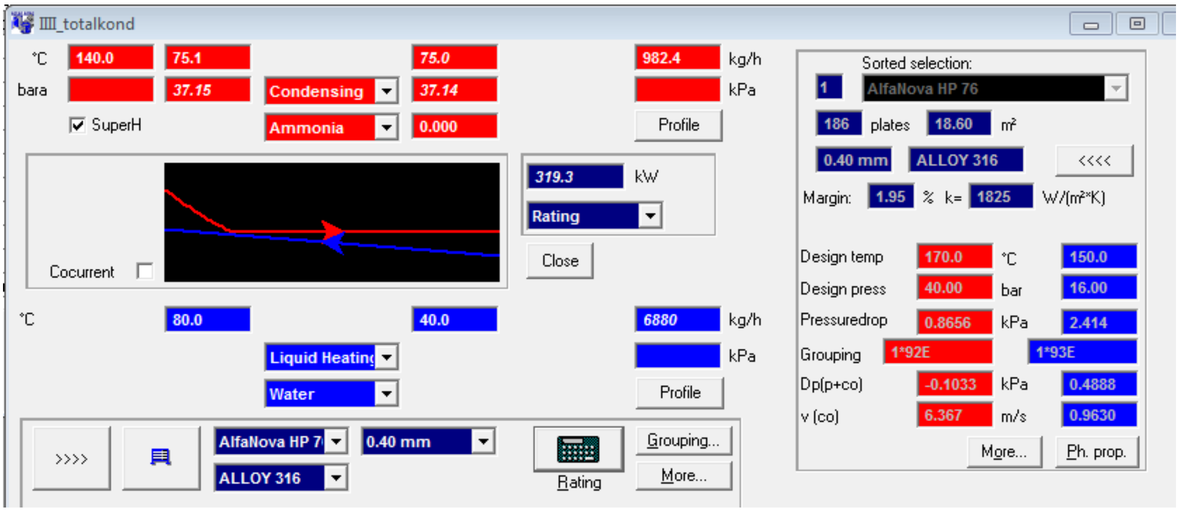

Figure 18. Example of calculation of a single unit with AlfaNova76 plates.

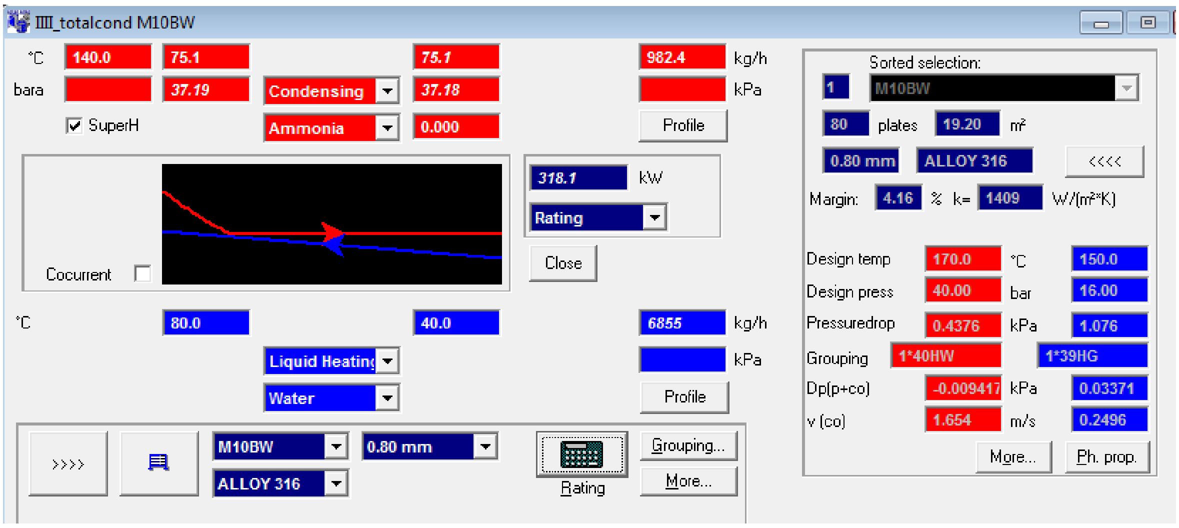

Figure 19. Example of calculation of a single unit with M10BW plates.

There are several things that differ between the two alternatives. The semi-welded unit requires 0.8 mm plates compared to 0.4 mm for the AlfaNova76. The M10BW also requires more heat transfer area and comprises heavy frames in order to stand the pressure. Furthermore, the footprint and the size of the unit are much larger, and hence this is not a very attractive alternative compared to a split condenser selection.

An alternative is to use the same amount of plates as the AlfaNova76 single condenser design above, but divide it into two units in series on the refrigerant side and parallel on the water side. In the pictures below, the unit is divided into three sections; desuperheater, condenser part 1, and the second unit condenser part 2. The desuperheater is shown in Figure 20 followed by the condenser part 1 in Figure 21.

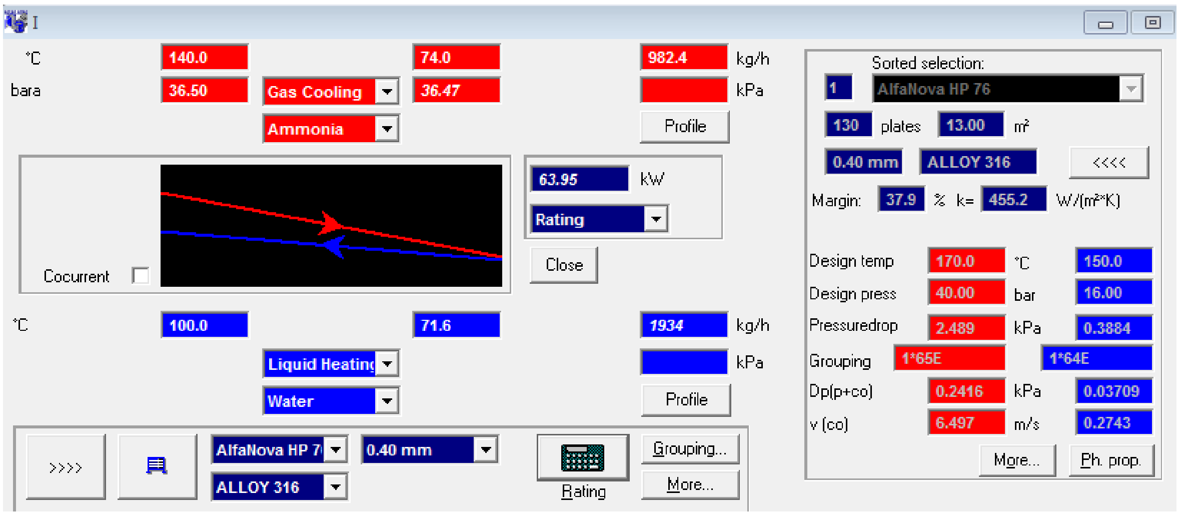

Figure 20. Example of calculation of the de-super-heater part of a unit divided into two sections coupled in series with AlfaNova76 plates.

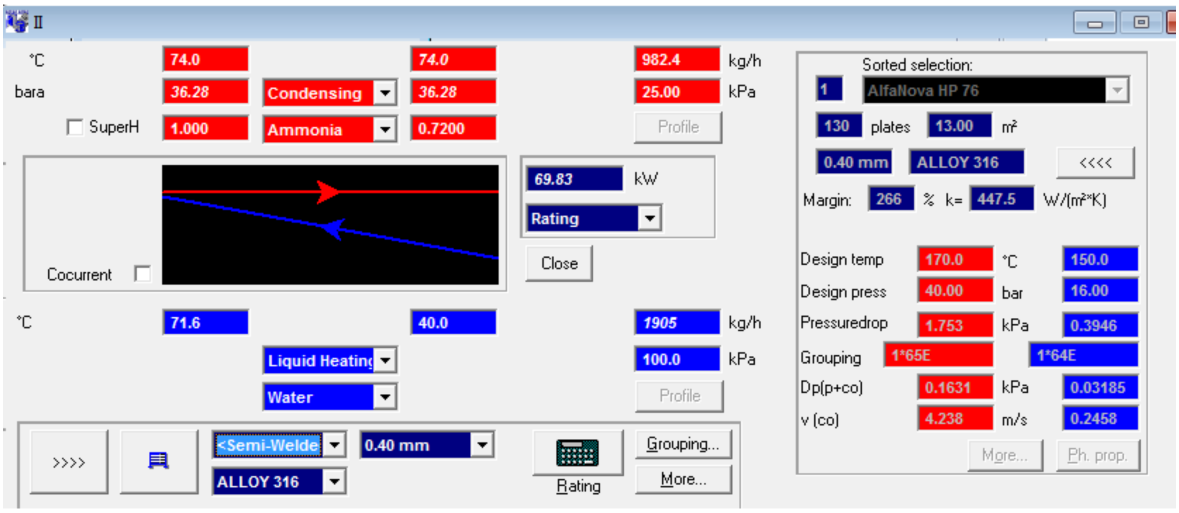

Figure 21. Example of calculation of condenser, part 1 of a unit divided into two sections coupled in series with AlfaNova76 plates.

It should be noted that in real life this is one and the same unit. Hence, the surface margin for the desuperheater part must be inverse of the condenser part as the total area is the same.

It should also be noticed that the balancing point for the condensing temperature in this design is 74°C compared to 75,1°C for the single unit design. This boosts the efficiency of the heat pump.

The design of the second condenser is shown in Figure 22.

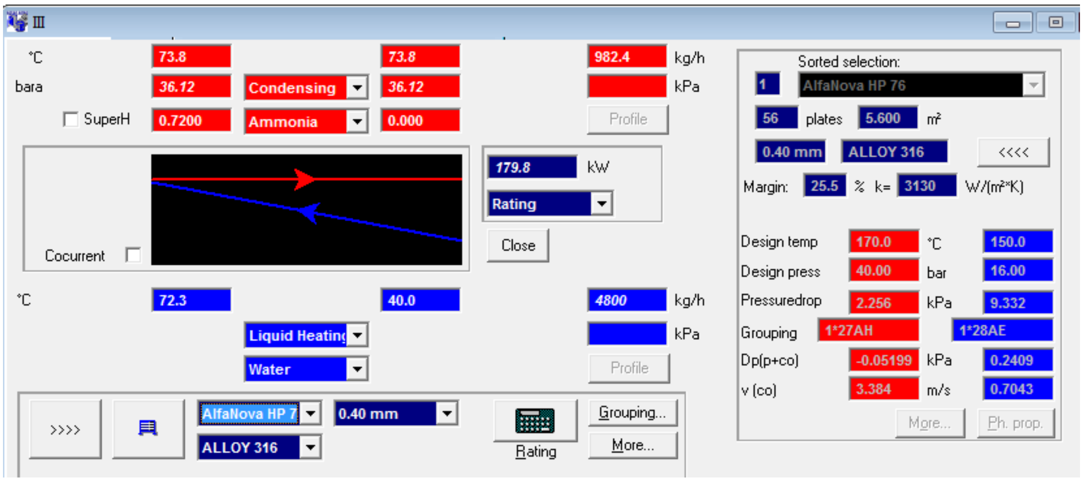

Figure 22. Example of calculation of condenser, part 2 of a unit divided into two sections coupled in series with AlfaNova76 plates.

Furthermore, it is worth noticing that the unit can perform better than required, which is seen by the lower condensing temperature and the surface margin. This means that it is possible to do the thermal duty of a single condenser design with a smaller heat transfer area (in total) and at a lower condensing temperature at the same time. One reason for this is that it is possible to select different plate types for the two units, each optimized for the thermal duties, which they do.

5.3. Higher Outlet Water Temperature

One potential advantage with a split condenser is that it is possible to have high outlet water temperatures. In a single condenser design, the outlet water temperature is limited by the pinch point in the heat exchanger, see Figure 23.

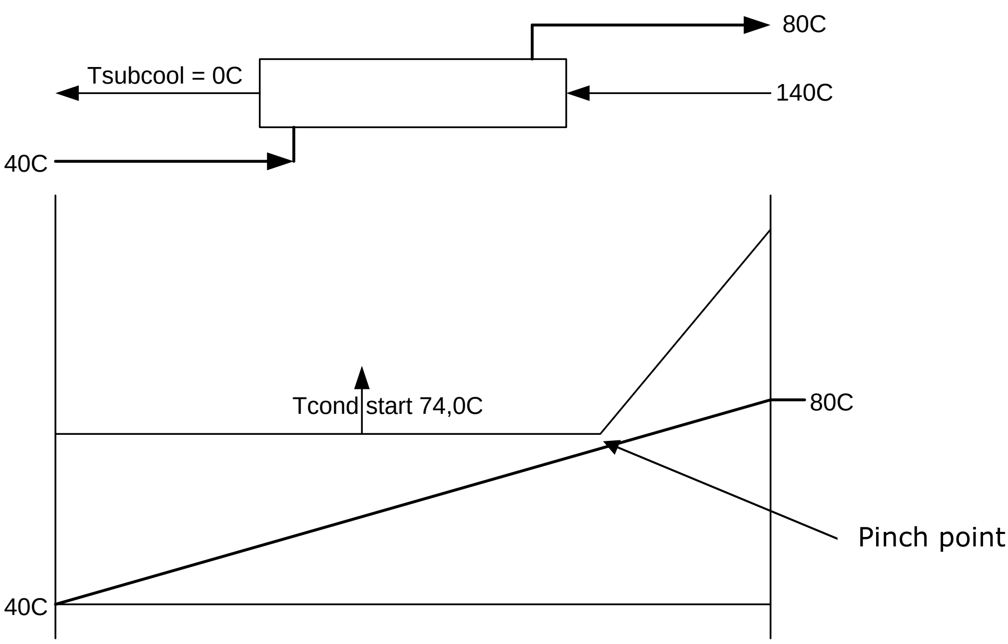

Figure 23. Principle of single condenser design.

This also means that the amount of heat, which can be transferred at a high temperature, is limited to the superheat of the refrigerant.

For a split condenser design, there are not only one pinch point, but two to consider. One pinch point in the second condenser and one in the first condenser, which is the combined desuperheater – condenser.

Simply by selecting the condensing temperature, the outlet temperature from the de-superheater and by varying the water flows in the different heat exchangers, one can achieve a large amount of heat at high temperature levels, see Figures 24 and 25.

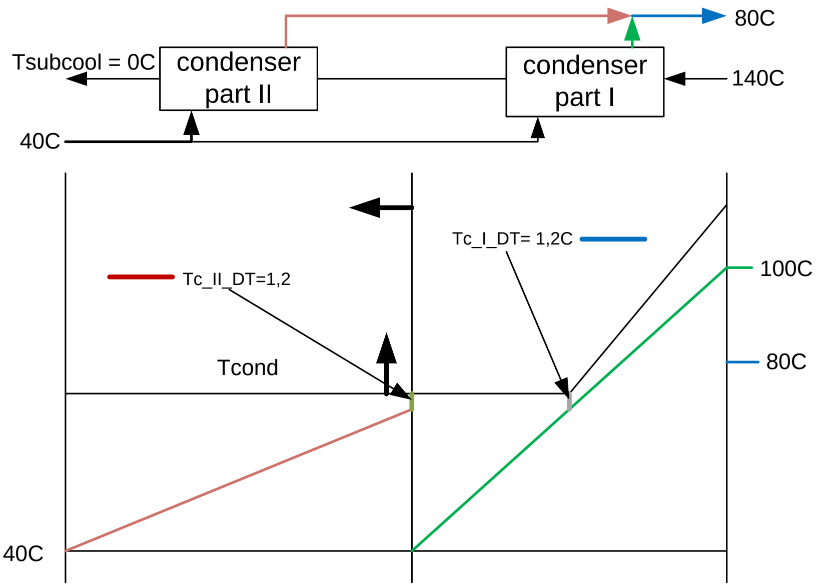

Figure 24. Example 1 of a split condenser.

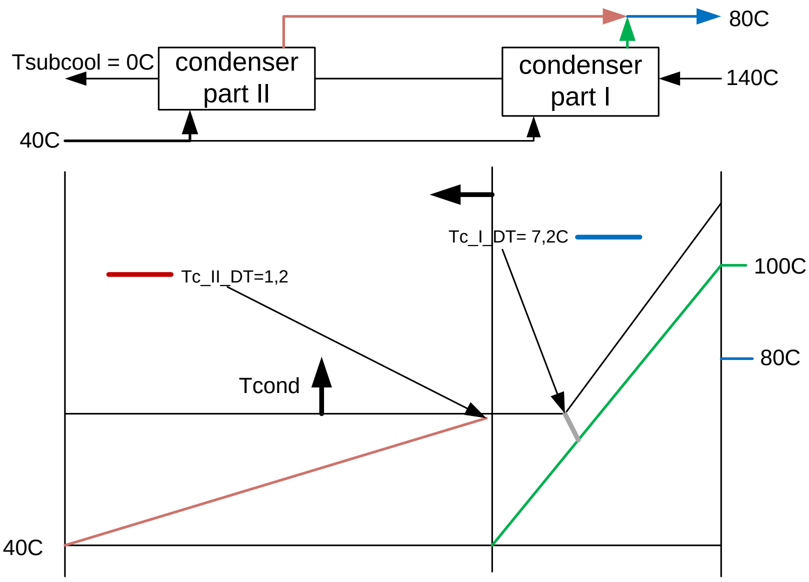

Figure 25. Example 2 of a split condenser.

5.4. Selecting Smaller Units

In most of the design cases, it is preferable to select small heat exchangers as they are less expensive, have lower weight, and are more compact. In the case of a single condenser solution, the entire water flow must pass through the same heat exchanger, which can give rise to pressure drop restrictions. Normally, this leads to the need to select larger heat exchanger sizes. In the example below of a heat pump delivering 820 kW at 75 °C outlet water temperature, a semi-welded high-pressure unit must be selected to meet the required design pressure of minimum 50 bar. See Figure 26.

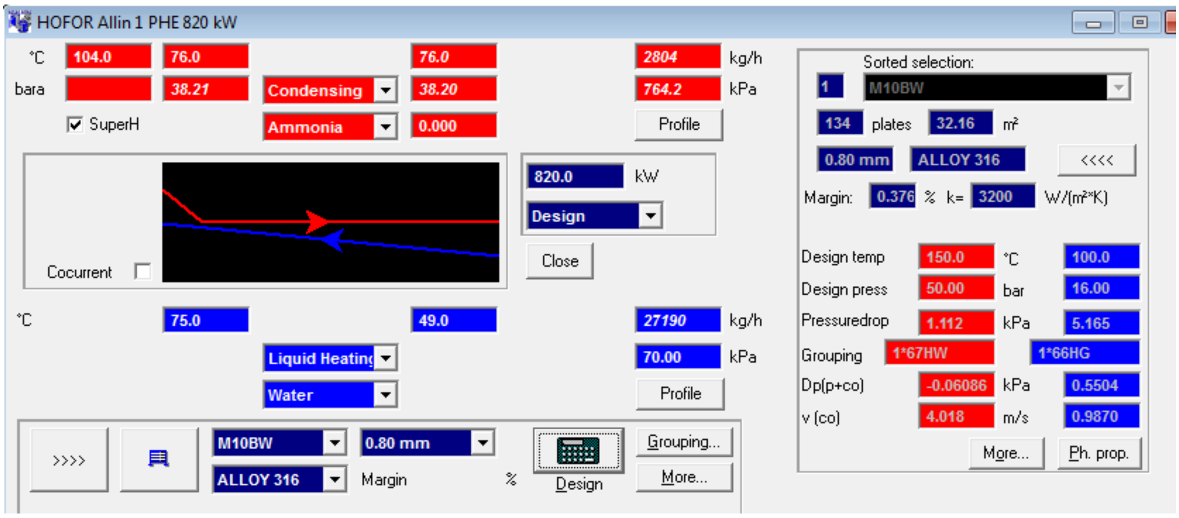

Figure 26. Example of calculation of a single condenser unit with M10BW plates.

This heat exchanger weighs about 800 kg, and the cost is roughly 15.000 €. The dimensions are:

Overall length x width x height mm 1285 x 470 x 1133

When comparing this to an optimized split condenser design using AlfaNovaXP52 heat exchangers, it is possible to do the duty in two small units and still respect the pressure drop limitations:

1 pcs of AlfaNovaXP52-150H as a combined desuperheater/condenser and
1 pcs of AlfaNovaXP52-130H as the second condenser.

In this case, the condensing temperature can be substantially lower, 73°C instead of 76°C as see above. See the design in Figures 27, 28, and 29.

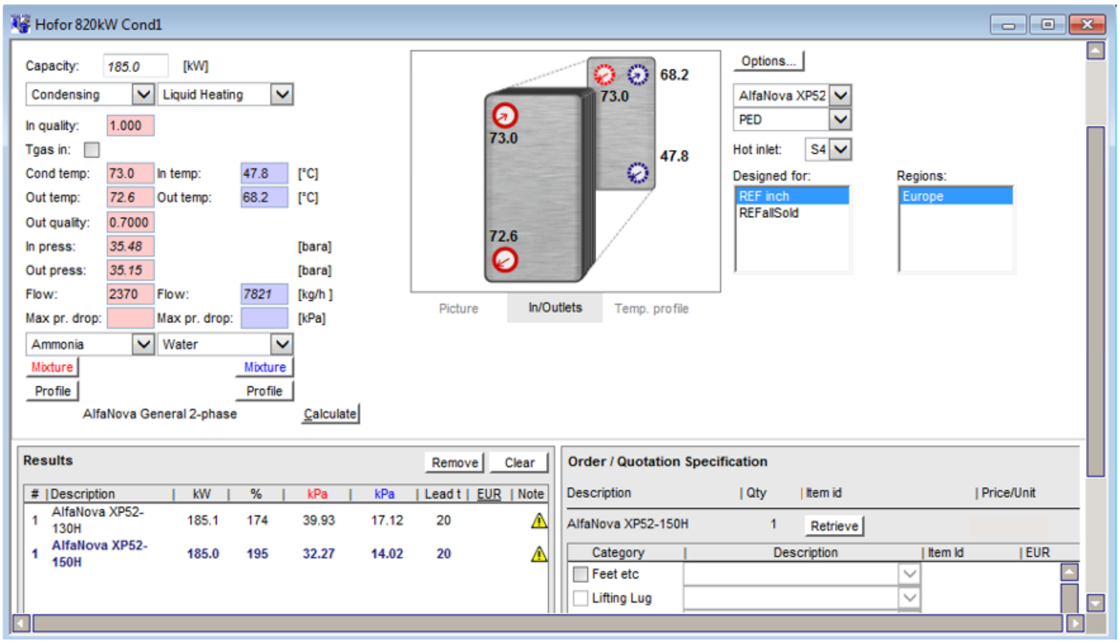

Figure 27. Example of calculation of condenser, part 1 of two units in series with AlfaNovaXP52-150H plates.

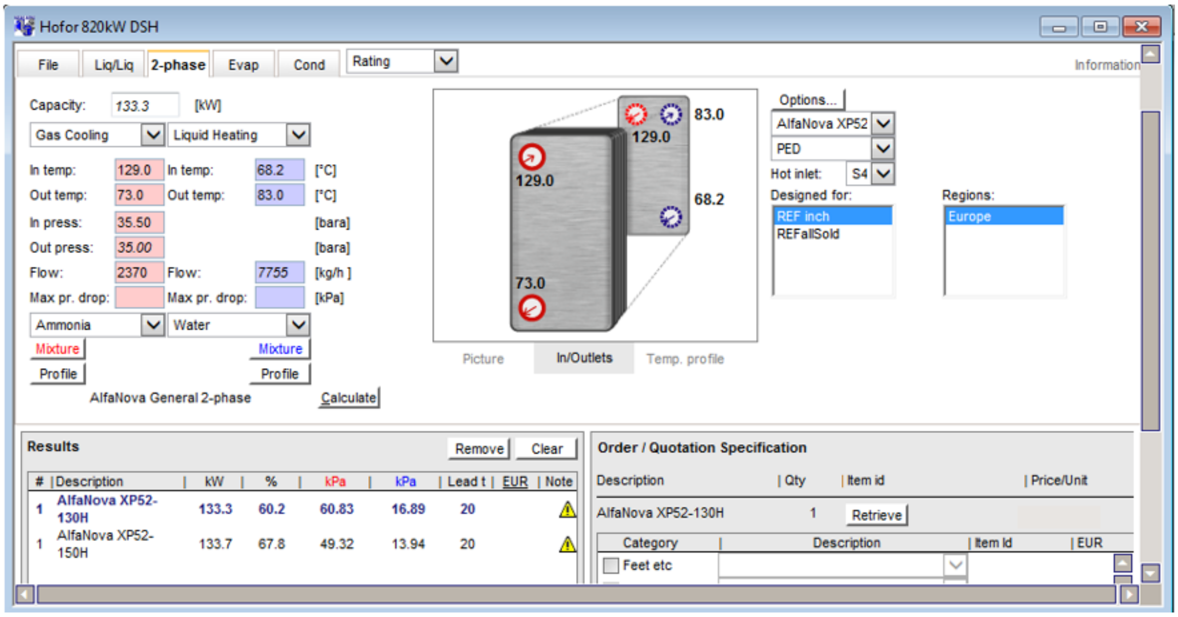

Figure 28. Example of calculation of de-super-heater part of a unit divided into two sections coupled in series with AlfaNovaXP52-150H plates.

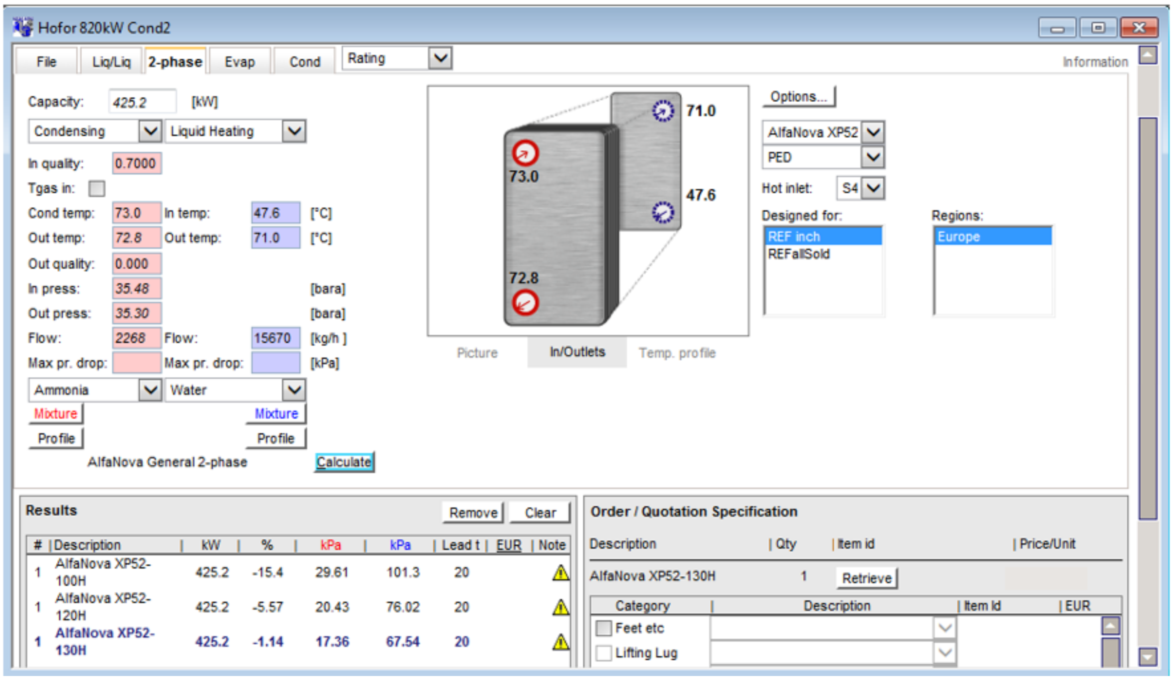

Figure 29. Example of calculation of condenser, part 2 of two units in series with AlfaNovaXP52-130H plates.

The weight and the dimensions are:

AlfaNovaXP52-150H:

Overall length x width x height	mm	471 x 111 x 526
Net weight, empty / operating	kg	35.8

Cost is about 1.500 €

AlfaNovaXP52-130H:

Overall length x width x height	mm	421 x 111 x 526
Net weight, empty / operating	kg	31.5

Cost is about 1.300 €

Apart from a much smaller weight, about 70 kg compared to 800 kg, the split condenser design is much more compact than the single design unit, and it costs less than 20%!

This example clearly shows the potential benefits of a split condenser design.

5.5. Pinch Point Optimisation Software

In all the calculations shown above, there is a need for a large number of iterations to be made in order to find the best selection of heat exchangers, and at the same time, to find the lowest condensation temperature. To do so in a practical way, a software has been developed to provide help in the iterative design process. This software takes care of the heat balances between the refrigerant side and the water side, and it determines the pinch points and the appropriate flow rates for the different sections, see Figure 30.

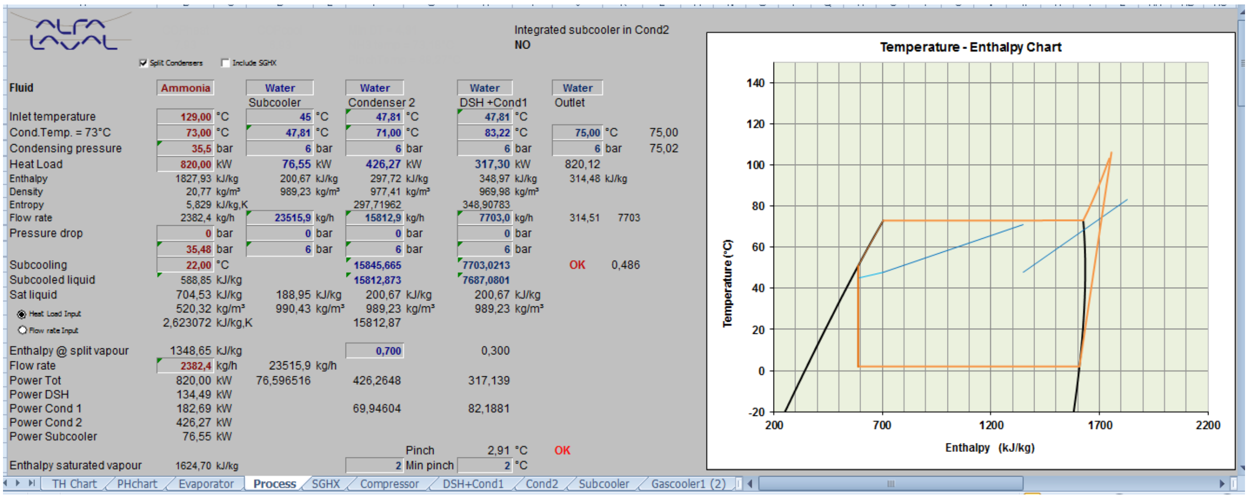

Figure 30. Example of a calculation with the split condenser software. (HOFOR 820 kW).

With this software, it is relatively easy to find a suitable outlet temperature from the de-superheater/Condenser1, and to balance the two pinch points together with the CAS design software. This tool has been essential for the design process in order to find the optimum layout to achieve smaller heat exchangers and the correct plate types. Figure 30 represents an iteration stage for the split condenser design for the case described in chapter 5.4 - see Figures 27, 28 and 29.

Another relevant case for the FOSCAP project is a design made for a commercial project at an industrial site located in Frederikssund, Denmark. The layout of the system is designed to use small heat exchangers with split condenser design, see Figure 31.

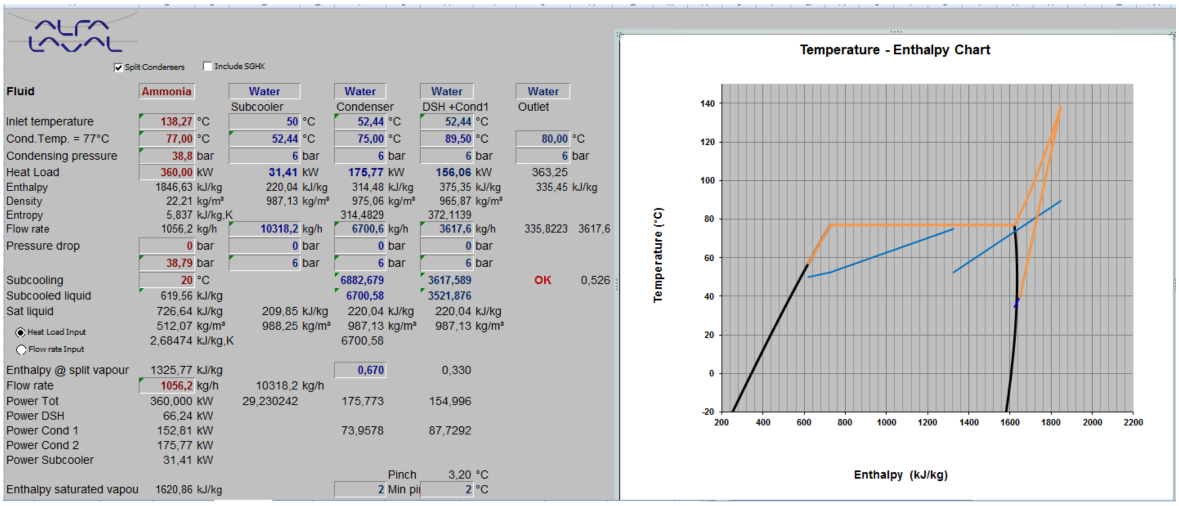

Figure 31. Example of a calculation with the split condenser software (Frederikssund).

More details on this commercial installation is shown in chapter 5.7.

5.6. Alfa Laval Product Portfolio Development

During the FOSCAP project and thereafter, Alfa Laval have developed several new products aimed at meeting market demands for efficient high temperature ammonia heat pumps. The start of this development was to develop new frames for semi-welded heat exchangers followed by a new range of high pressure AlfaNova heat exchangers, see Figure 32.

	AlfaNovaXP 27	AlfaNovaXP 52	AXP 27 AN	AXP 52 AN
Design pressure (bar @ 150°C)	70	70	110	110

Figure 32. Examples of AlfaNova heat exchangers.

The products, which have been used in industrial ammonia heat pump projects, are mainly AlfaNovaXP 52 and AlfaNova 27.

The AlfaNova portfolio was also extended with a new evaporator, which has been used in the FOSCAP heat pump. The new heat exchanger is named AlfaNova 200, and it has a capacity up to approximately 350 kW as an evaporator. The evaporator has been used together with a U-turn separator, see Figure 33.

Figure 33. U-turn separator.

Furthermore, it was decided to develop a large semi-welded unit, where the scope included a high-pressure version with 63 bar design pressure, directly aiming at the industrial size ammonia heat pump market, i.e. the TK20-BW. Complementing the TK20-BW on the market, the latest development is a new 10 size high pressure heat exchanger, the T10-EW. This heat exchanger is focusing on the new high-temperature ammonia heat pumps, where the plate has been adopted in order to fit the large temperature differences on the water side.

The new semi-welded portfolio is shown in Figure 34. New products are indicated in red writing.

	M6-MW	M10-BW	T10-EW	MK15-BW	TK20-BW	T20-BW	MA30-W
16 bar	Yes	Yes	Yes	Yes	Yes	Yes	Yes
25 bar	Yes	Yes	Yes	-	-	-	Yes
30 bar	-	Yes	-	Yes	Yes	Yes	Yes
40 bar	-	Yes	Yes	Yes	-	-	-
55 bar	-	Yes	-	-	-	-	-
63 bar	-	-	Yes	-	Yes	-	-

Figure 34. Semi-welded heat exchangers.

It should be noted that the FOSCAP project has been an inspiration, and the information gained in the project has been useful for the development of the high-pressure product portfolio - mainly in terms of reducing uncertainties and verifying performances in split condenser designs.

5.7. Industrial Installation Example

As mentioned in chapter 5.5, a split condenser design has been made for a commercial project at an industrial site located in Frederikssund, Denmark. The layout of the system is designed in order to use small heat exchangers with split condenser design. The capacity of the heat pump is 360 kW delivering hot water at an outlet temperature up to 80°C. The condensation temperature at design condition is 3 K lower than the outlet water temperature.

Here, AlfaNovaXP 52-90H is used as de-superheater/condenser and AlfaNovaXP 52-50H is used as condenser 2. The system also includes a subcooler to enhance system performance. This is an AlfaNova 27-30L. The heat pump can be seen in the following picture from the site, see Figure 35.

Figure 35. Heat pump installation in Frederikssund, Denmark, with split condenser.

A closer view of the split condenser layout is shown in the Figure 36.

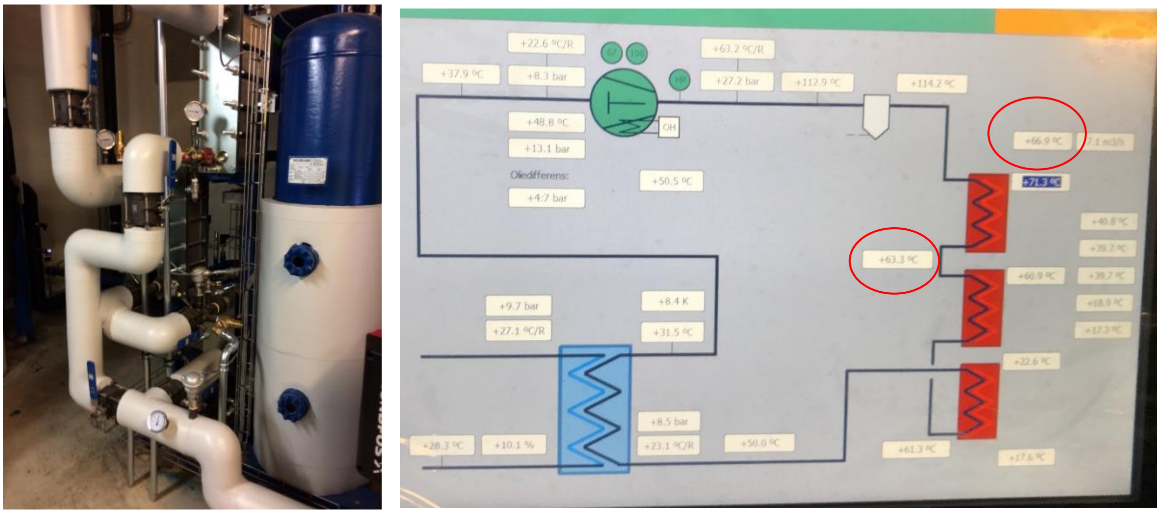

Figure 36. Details of heat pump installation in Frederikssund, Denmark, with split condenser.

It is also very interesting to view the performance of the split condenser system. One example is shown in the above picture to the right.

It is noticeable that an outlet water temperature of 66.9°C is achieved at a condensing temperature of 63.3°C, i.e. the approach is -3.6°C, which is in line with the predictions.

Furthermore, it is estimated that the benefits in terms of weight, compactness, and cost are in the same magnitude as described in chapter 5.4.