NTNU – Trondheim
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Optimizing a Small Ammonia Heat Pump with Accumulator Tank for Space and Hot Tap Water Heating

Arturs Lalovs

Sustainable Energy
Submission date: June 2015
Supervisor: Trygve Magne Eikevik, EPT

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NTNU
Det skapende universitet

MASTERKONTRAKT

- uttak av masteroppgave

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2. Studieopplysninger

Fakultet **Fakultet for ingeniørvitenskap og teknologi**
Institutt **Institutt for energi- og prosessteknikk**
Studieprogram **Sustainable Energy**

3. Masteroppgave

Oppstartsdato **12. jan 2015**	Innleveringsfrist **08. jun 2015**
Oppgavens (foreløpige) tittel **Optimizing a Small Ammonia Heat Pump with Accumulator Tank for Space and Hot Tap Water Heating**	
Oppgavetekst/Problembeskrivelse It will be built a smaller plant outdoors for use in detached to heat domestic water and space heating via underfloor heating. In this task it evaluated various solutions for ammonia refrigerant in residential heat pumps, and instrumenting and monitoring system of measurements of the heating season to look at energy use, operating conditions, management strategy and component selection. The following tasks are to be considered: 1. The literature review focusing on the scope of work 2. Develop a calculation tool (EES) for evaluation of energy over the year 3. Evaluate the size and needed for accumulation tank in the floor heating system at different tariffs for electricity throughout the day/week 4. Develop operational strategy for the heat pump system with regard energy prices throughout the day, as well as alternative management of compressor - on / off or frequency control 5. Verification of calculation model by measuring on new NH3 heat pump systems with ground source heat as a heat source. 6. Writing a scientific "paper" with the main results from the task 7. Make proposal for further work	
Hovedveileder ved institutt **Professor Trygve Magne Eikevik**	Medveileder(e) ved institutt
Merknader **1 uke ekstra p.g.a påske.**	

Side **1** av **2**

4. Underskrift

Student: Jeg erklærer herved at jeg har satt meg inn i gjeldende bestemmelser for mastergradsstudiet og at jeg oppfyller kravene for adgang til å påbegynne oppgaven, herunder eventuelle praksiskrav.

Partene er gjort kjent med avtalens vilkår, samt kapitlene i studiehåndboken om generelle regler og aktuell studieplan for masterstudiet.

20/01/2015 **Sted og dato** 20/1 - 2015

Student **Hovedveileder**

Originalen lagres i NTNUs elektroniske arkiv. Kopi av avtalen sendes til instituttet og studenten.

Changes in the tasks

Before work started plan of the project was to find missing components for the ammonia heat pump and to install it in one family house for further experiments and measurements. Due to technical reasons this task was not completed what leads to substitute practical part with design drawings for the heat pump with specification of components.

22/5 - 2015

Agreed with supervisor

TRYGVE M. EIKEVIK

SUPERVISOR

ABSTRACT

The heat pump market offers a wide variety of different residential heat pumps where most of them utilize refrigerant R-410A which has high global warming potential. Considering the fact that global policy starts to focus on issues related to energy efficiency and harmful impact to the environment, it is necessary to investigate over new refrigerants. As an alternative solution is to utilize natural refrigerants, such as ammonia, which has almost zero global warming potential and zero ozone depletion potential. Since the beginning of refrigeration era, ammonia has been used in large capacity systems, whereas no attention has been paid to residential sector.

In this study, small ammonia heat pump with capacity of 8,4 kW for one family house has been designed to provide heating of domestic water and space heating via floor system.

Theoretical part of this study involves analysis of the present heat pump market and comparison of different heat pump applications. Previous experience in researches of small capacity ammonia systems have been summarized to highlight related problems and design futures.

Practical part of this study involves calculations in the Engineering Equation Solver with estimation of the heat pump performance and evaluation of necessary storage tank volume for the heating system to provide efficient operation during variable electricity prices.

Engineering part consists of principal drawing of the small ammonia heat pump with list of components. Additional drawing of a house with floor heating system is included as an example which shows possible application for the ammonia heat pump.

The thesis consists of 80 pages, 73 equations, 7 tables, 44 figures, 3 drawings and 3 appendices. 24 literature sources used for this thesis.

Keywords: small capacity ammonia heat pump; variable electricity prices; storage tank; heat pump operational regimes.

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INTRODUCTION

Nowadays, the problem with climate change is gaining importance. Due to big CO_2 emissions and greenhouse effect, global policy starts to focus on actions which could contribute reduction of harmful impact on the environment. As an example is "20 - 20 -20" climate and energy package released by European Union (EU) where are placed 3 main objectives with deadline of 2020:

- 20% reduction in EU greenhouse gas emissions from 1990 levels;
- raising the share of EU energy consumption produced from renewable resources to 20%;
- 20% improvement in the EU's energy efficiency. (EU Cimate and Energy package, 2008)

To achieve those targets there is need to implement environmental friendly technologies and one of them is a heat pump technology.

A heat pump is a device that provides heat energy from a source of heat to a destination called a "heat sink". Heat pumps are designed to move thermal energy opposite to the direction of a heat flow by absorbing heat from a cold space and releasing it to a warmer one. A heat pump uses some amount of external power to accomplish the work of transferring energy from the heat source to the heat sink.

According to data about household energy consumption in Norway (2012), almost 80% of energy is used for space and hot tap water heating.(Birger Bergesen and Ingrid H. Magnussen 2013) Considering the fact that the biggest part of this demand is covered by electrical energy, wide use of residential heat pumps could make significant energy savings. Although residential heat pump market offers wide range of heat pumps, the most of them use refrigerant R410A which has negative influence on the environment due to big global warming potential. It leads to search for natural and environmental friendly refrigerants and one of them is ammonia.

Ammonia has been used in large capacity systems since the beginning of refrigerating era. However, there are still no available small capacity systems for residential buildings on the market. It is related to lack of components and specific design due to physical and chemical properties of the ammonia. The main focus of this thesis is to find designing features and to estimate performance of the small ammonia heat pump for one family house.

The following tasks are considered:

- the literature review focusing on the related topic;
- development of a calculation tool (EES) for evaluation of energy consumption over the year;
- evaluate the size and need for accumulation tank for the floor heating system at different tariffs for electricity throughout the day/week;
- development of operational strategy for the heat pump system with regard to energy prices throughout the day, as well as alternative management of compressor - on / off or frequency control.

1. RESIDENTIAL HEAT PUMPS

In this chapter will be described different types of residential heat pumps, working principles and applications. After evaluation of present heat pump market trends, one of the modern heat pumps will be chosen for detailed analysis. This analysis will lead to better understanding of components interaction and will help to design optimal ammonia heat pump.

Heat pumps could be classified by utilized heat source. The most widely used are air source, water source and ground source heat pumps.

1.1 Air source heat pumps

Air source heat pump (ASHP) utilize ambient air as a heat source. It can provide efficient heating and cooling during different periods of year.

Simple **air-to-air** heat pump consists of a compressor and 2 coils made of copper tubing, which are surrounded by aluminium fins for better heat transfer. Air movement is forced by fans inside of each unit. In heating mode the outdoor coil is used as a heat source (evaporator) and indoor unit as heat sink (condenser). Cooling mode is achieved by reverse heat pump cycle changing refrigerant flow direction by 4-way valve. In that case the outdoor unit works as the condenser and indoor unit as the evaporator. Capacity range is from 2 to 15 kW. Figure 1 shows working principle of air-to-air heat pump in both cooling and heating mode.

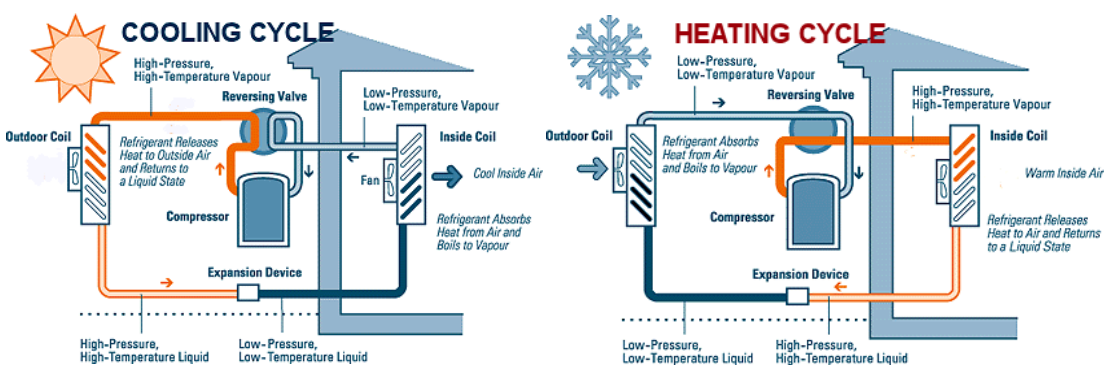


Figure 1 Air-to-air heat pump operation modes (U.S. Energy)

Air-to-air heat pumps product range varies from simple split systems (one indoor and one outdoor unit) to multi-split and variable refrigerant flow (VRF) systems. Multi-split system is very similar to the split system with the same capacity range. The only benefit of the multi-split system is possibility to connect more than one indoor unit and cover bigger heating area. (U.S. Energy)

The VRF system has also possibility to connect multiple indoor units. Capacity range is from 9-130 kW. Usually those systems are equipped with variable speed compressor to achieve bigger efficiency at different loads. It has also an option of cooling and heating mode. 3 pipes VRF system could provide simultaneous heating and cooling utilizing internal building energy. This solution is very efficient for buildings with large number of windows where heat load varies in each part of the building. Such systems are designed only for space heating and cooling without possibility to preheat hot tap water.

Another type of air source heat pump is **air-to-water**. This type of system utilizes ambient air as a heat source and transfers heat to water heating system. Typically, for heat transfer in building is used coaxial copper tube-in-tube heat exchanger. Heat pump works much more efficient with low temperature heating applications. It is preferable to use under-floor heating or large area radiators. It can also provide hot tap water preheating. Figure 2 shows possible applications for the air-to-water heat pump.

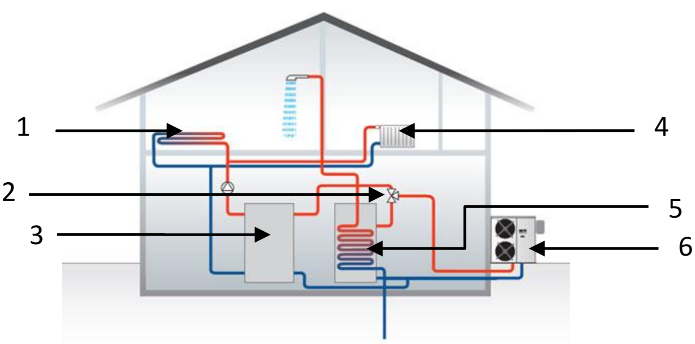


Figure 2 Air-to-water heat pump scheme (1 - floor heating,2 - 3-way valve,3 - water tank for heating system, 4 - radiator, 5 - domestic water tank, 6 - air source outdoor unit) (Sprint Energy Solutions)

ASHPs are definitely the most widely used because of low price and easy installation. However, they are not supposed to supply high efficiency heat during low ambient air temperatures. It this case it is necessary to install additional peak load heating source (electrical heaters, central heating or gas boiler) to cover heating demand during low ambient air temperatures in the winter. Air-to-water heat pumps could supply hot water temperature at 60^{o}C. Some manufacturers offer high-temperature models which could heat up water up to 80^{o}C. Biggest part of those heat pumps use refrigerant R-410A and scroll compressors with frequency controller. Manufacturers offer air source heat pumps with coefficient of performance (COP) higher than 4 and seasonal coefficient of performance (SCOP) higher than 5. Operation limit for those heat pumps usually is -20 C^{o} of ambient air temperature. However, it should be taken into account that COP values are given at certain conditions. Typically, at outdoor temperature 7^{o}C and indoor temperature 21^{o}C. Due to decreasing of ambient air temperature COP drops significantly and close to -20^{o}C it is almost the same as electrical heater (COP = 1). Those conditions makes system non-effective which lead to conclusion that application of air source heat pump is limited by seasonal temperature variation. In regions like Scandinavia this kind of heat pump is not efficient due to low ambient air temperature during the winter time.

1.2 Geothermal heat pumps

Water and ground source heat pumps very often are called as geothermal heat pumps (GHPs). GHPs working principles are similarly to ASHPs. The only significant difference is that they use the ground or the water sources as the exchange medium instead of ambient air. It allows to reach high COP (from 3-6) even during the coldest winter days due to relatively insignificant temperature variations between different seasons. Those heat pumps could provide heating, cooling and also supply the house with hot tap water. Relative to ASHPs, they are quieter, last longer, need little maintenance and almost do not depend on ambient air temperature.

GHPs has several types of different heat exchangers which provide heat transfer from the ground to the heat pump. The most utilized are horizontal (a), vertical (b) and pond/lake (c), shown in Figure 3.

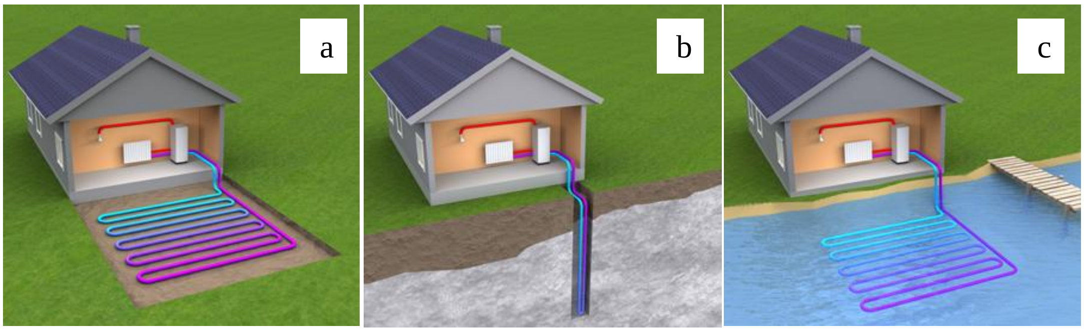


Figure 3 Ground source heat exchanger types (a - horizontal, b - vertical, c - pond/lake) (Siltumsūkņu Cents)

Horizontal loop is generally most effective for residential buildings because of low installation costs. Plastic tube (normally high density polyethylene) is placed into the ground under the freezing level which depends on region. Disadvantage of this system is big land area which has to be trenched. From one meter of pipe it is possible to gain approximately 20-35 W of heat.

However, in houses without sufficient amount of free land the only option is to use vertical boreholes. They have stable, deep soil temperatures with greater potential for heat exchange with ground water. Depth of vertical borehole could vary from 30 to 150 meters. In each borehole is placed "U" shaped tube. It is possible to connect several boreholes if one is not enough. Heat gains from one meter of tube are 50-60 W.

Other solution is to place heat exchanger into a lake or a pond. Minimal depth is 2,5 meters to prevent freezing. Due to stratification, lower possible water temperature at the bottom is 4^{o}C which is enough to run heat pump at high efficiency. Tube at the bottom should be fixed by weight. Average heat gains from one meter of tube placed in water are 30 W. The pipe is filled with mixture of water and ethylene/propylene glycol or ethanol alcohol to avoid freezing. (Energy 2011)

GHPs are more efficient and more reliable than ASHPs. The more stable operating temperature/pressure of the compressor in a GHP also promises extended compressor life compared to air-source heat pumps. For example, ASHRAE (2003a) estimates the life expectancy of a commercial water-source heat pump to be over 25% greater (19 years) than its commercial air-source counterpart (15 years). (Phetteplace 2007)

Table 1 Comparison of different heat pumps

Type	**Capacity range, kW**	**Design solutions**	**Limitations**
Air-to-air	Split 2-15 Multi-split 2-15 VRF 9-130	Easy to install Only space heating/cooling Mostly R-410A	Ambient air temperature (-20^{o}C)
Air-to-water	3,5 - 60	Easy to install Space heating/cooling Hot tap water preheating Mostly R-410A	Ambient air temperature (-20^{o}C)
Ground source (horizontal)	3,5 - 80	Constant efficiency during the year Space heating/cooling Hot tap water heating Mostly R-410A	Big land area for installation
(vertical)			High installation costs
(pond/lake)			Availability of heat source

1.3 Components of GHP

Theoretical study of geothermal heat pump has been made to evaluate modern trends of heat pump manufacturers. Tight competition on heat pump market forces all manufacturers to follow technological progress and use, basically, the same technologies and components. For detailed review has been taken geothermal heat pump from one of the leading manufacturers with capacity of 11 kW with hot water generation. This is complete package which is ready to be connected to water loops. Both vertical and horizontal ground source loops can be used.

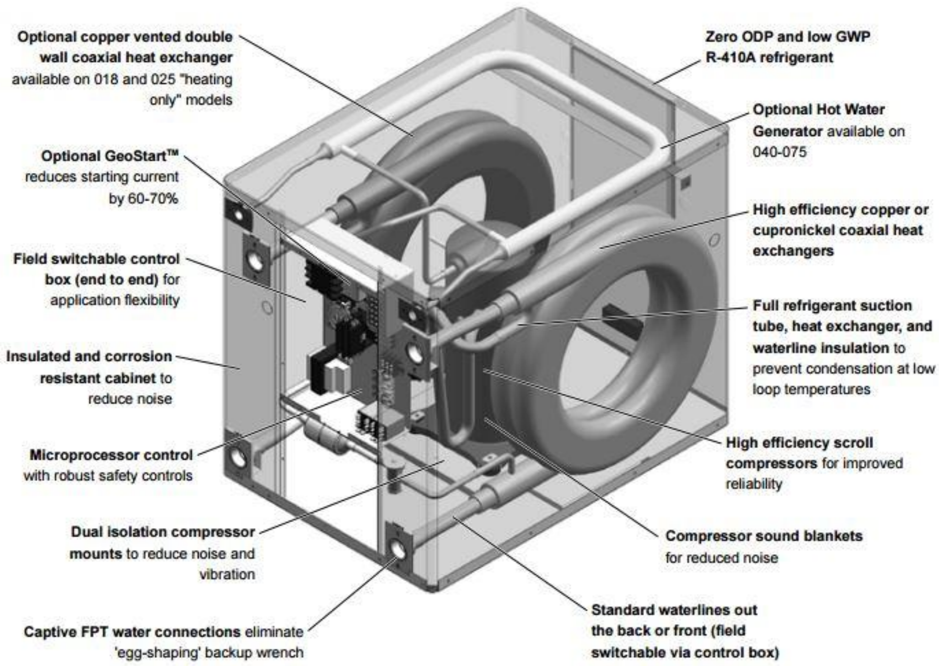


Figure 4 Geothermal heat pump (Johnson Controls, 2010)

Cabinet is constructed of resistant galvanized sheet metal which provide long-term operation and easy access to basic components. There is a high efficiency scroll compressor which works on R-410A refrigerant. Scroll compressors provide both the highest efficiency available and great reliability. For control of refrigerant is used thermostatic expansion valve (TXV). TXV allows precise refrigerant flow in a wide range of entering water variations (-7^{o}C to 49^{o}C) found in geothermal systems. The TXV is located in the compressor compartment for easy access. However, nowadays is mostly used electronic expansion valve (EXV) which provide even better accuracy and performance. Unit has 2 large oversized coaxial refrigerant-to-water copper heat exchangers. They are designed for low pressure drops and low flow rates. Both heat exchangers have insulation (coating) to prevent condensation in low loop temperatures. Suction and discharge lines are equipped with service connections for field charging or service access. 4-way reversing valve provides possibility to operate unit in both cooling and heating modes. Unit is supplied with soft-start for reduction of compressor starting current. This unit can achieve maximum 54^{o}C of hot water temperature. Depending on temperature difference between ground and water loops, COP varies from 2,5 (ΔT = 45^{o}C) to 8 (ΔT=10^{o}C). (Johnson Controls, 2010)

1.4 Typical GHP application

This system application is designed for hydraulic heating. There are many options how to distribute the heat through the building. The most acceptable for this type of system is low temperature heating, such as radiant floor heating or fancoil units because of low necessary hot water temperature.

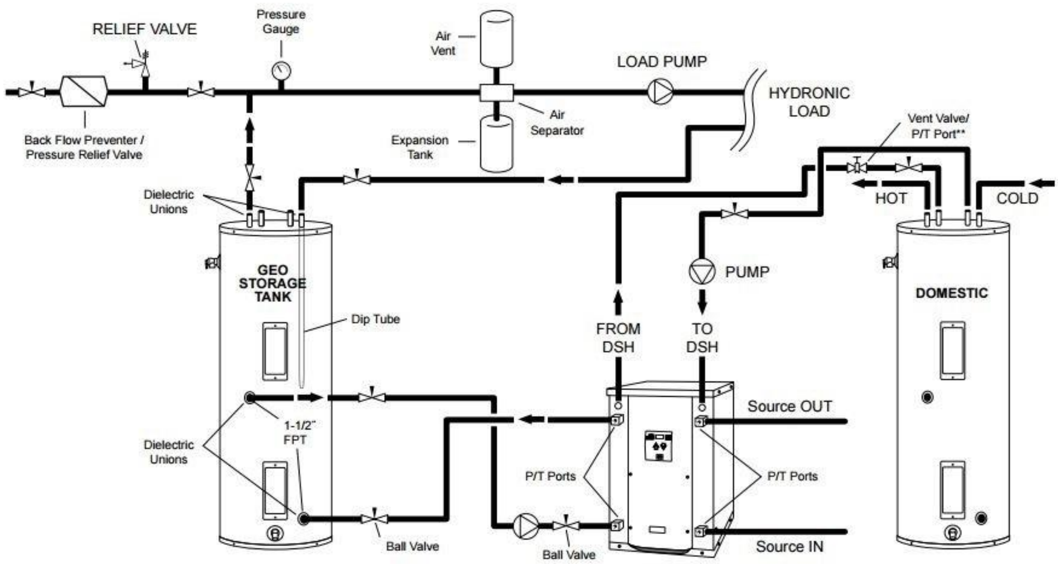


Figure 5 Typical application of ground source heat pump (Johnson Controls, 2010)

This system (Figure 5) consists of heat pump unit and two water tanks. One water tank is for domestic hot tap water and the other is for heating system.

Domestic water tank is equipped with electrical heater to heat the water to temperature from 55 to 70^{o}C to avoid presence of bacteria. Usually, this type of tank has internal coaxial heat exchanger to prevent refrigerant leakage to tap water system. This heat exchanger does not allow to achieve highest possible water temperature because of temperature differences in each heat exchanger.

Water tank of heating system has no strict safety limitations. It allows to use water of heating system for circulation through heat pump's heat exchanger. This type of heat exchange is much more efficient because of lower temperature difference between substances. It is very important to design right volume of those tanks (mostly heating storage tank) because they affect on/off periods of the compressor. To small water tank leads to very frequent on/off operation of the compressor which makes its lifetime shorter. Too large water tank leads to bigger installation costs. Important influence on the tank size has electricity price which could vary during the day. Manufacturers offer tank sizes from 200 - 500l. (Johnson Controls, 2010)

2. REFRIGERANTS

Refrigeration and heat pump industry has experienced many changes in the last 25 years in case of refrigerants. The beginning of changes was Montreal protocol in 1989 with main idea to protect ozone layer. Many refrigerant substitutes were made. For example, R22 which has ozone depletion potential (ODP) of 0,055 was substituted with R-407A (ODP=0). It is important to find refrigerant blend with similar or very close pressure and performance to allow smooth transition from one refrigerant to another. After some time R-407A was substituted with R-410A (ODP=0), which nowadays, is the most used refrigerant in residential sector. However, R-410A is not long term refrigerant because of high global warming potential (GWP) of 1725. High GWP of R-410A leads to explore new refrigerants and one of them are natural refrigerants, such as ammonia (NH_3). (Emerson Network Power, 2010)

Ammonia has been used in large capacity cooling and heating systems since the beginning of refrigeration era. Nowadays, came to focus the use of ammonia in small capacity systems. During the last 10 years many attempts were made to make prototypes of small capacity ammonia system. Unfortunately, most of them have failed, basically due to lack of components.

Ammonia is thermodynamically preferable. It has higher heating capacity and heat transfer coefficients compared to R-410A. GWP and ODP are very close to 0. Main disadvantages of ammonia are toxicity, flammability and its destructiveness towards materials, such as copper or brass. In applications with big pressure ratio, high temperature of the ammonia discharge gas could be an obstacle.

Table 2 Comparison of ammonia and R-410A refrigerants (EES Library)

Parameters	Ammonia	R-410A
Boiling point (^{o}C)	-33	-48,5
Critical point (^{o}C)	133	72,8
Gas heat capacity (kJ/(kg·°C)) *	2,68	1,13
Liquid heat capacity (kJ/(kg·°C))*	4,16	1,52
GWP	<1	1725
ODP	0	0

Values given at T = 0 ^{o}C

Table 2 shows that ammonia has more than two times higher heating capacity compared with R-410A. It allows to use less refrigerant charge and design smaller system. However, it is impossible to use ammonia as substitute for R-410A because of big difference in pressure and performance. Another issue is that majority components for residential systems are made of copper and ammonia is very destructive towards them. Specific components should be used to provide long term operation of ammonia heat pump.

Another substance, which could be used instead of ammonia is R723 which is mixture of ammonia (60%) and dymethilether (40%). Benefit of using R723 is lower discharge gas temperatures and slightly lower pressure ratio than for R717. (Danish Technological Institute, 2005)

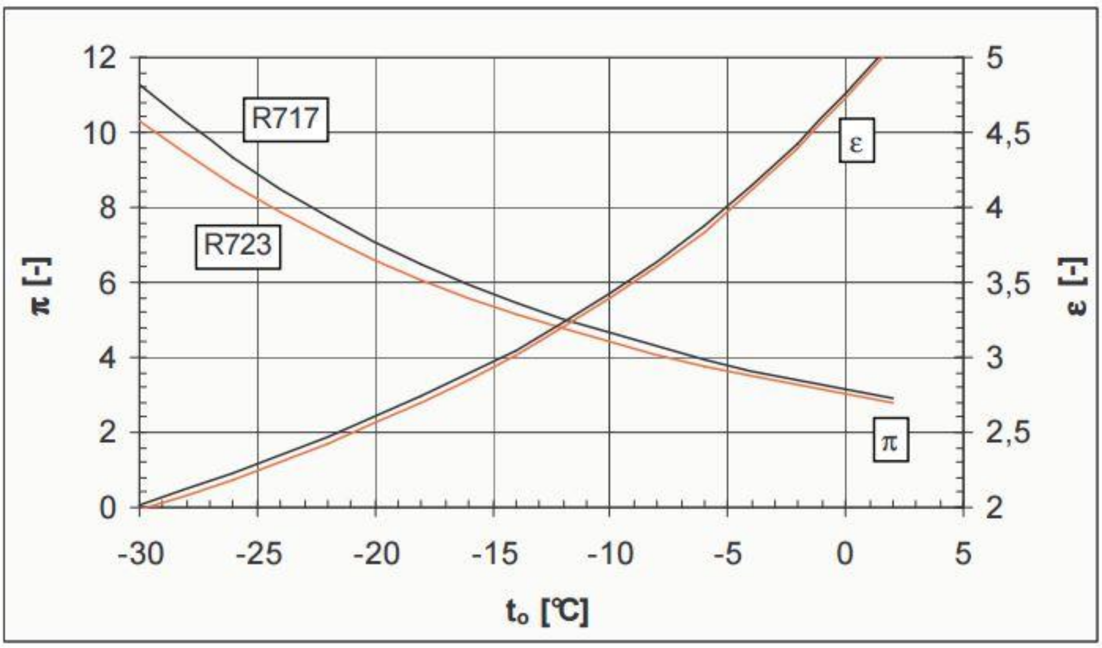


Figure 6 Comparison of R723 and R717 (Danish Technological Institute, 2005)

3. ELECTRICITY MARKET

The Norwegian electricity market supplies power with variable prices during the day. This method of price difference started earlier in big cities, where power demand changes are significant during the day. Variation in the price is an effective method how to equalize power demand and protect the grid from overload. The power price is determined by the balance between supply and demand. Factors such as the weather or power plants not producing to their full capacity can impact power prices as well. Today, there is general agreement among politicians and other stakeholders in the Nordic and Baltic power markets that this power model serves society well. Usually variable electricity price had been applied only for industrial sector but in the very close future it will be available also for residential sector. Figure 7 shows maximal price frequency during different day hours in Trondheim, Norway.

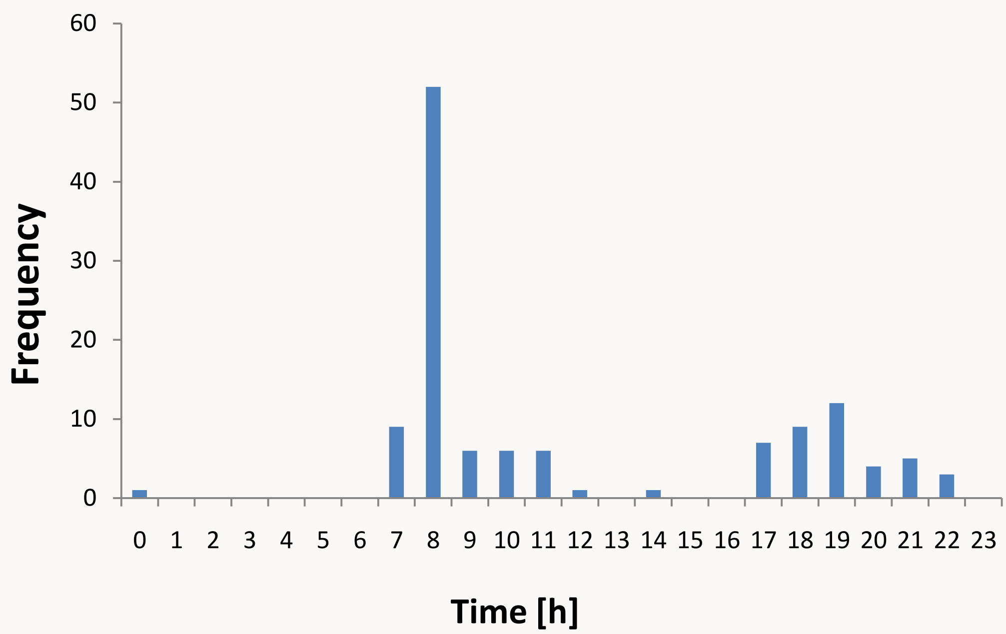


Figure 7 Maximal price frequency for period from 04.02.2013 to- 04.06.2013
(Bjørneklett 2013)

Maximal price of the power appears in the morning hours (7-8 AM) and during the evening hours (17-19 PM). Cheapest energy is available during the night when total load on the grid is very low.

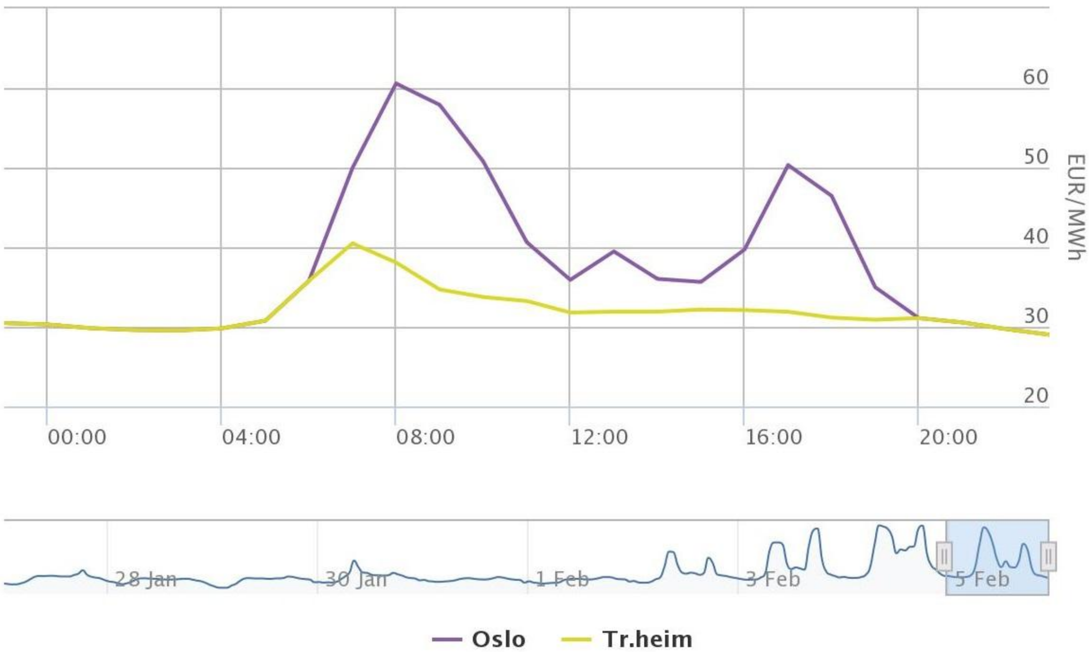


Figure 8 Price variation during the day in different regions (5th of February 2015) (NordPoolSpot, 2015)

Price difference between the minimum and the peak is much more higher in regions with high population density which is explained with bigger load on the grid. In smaller regions, such as Trondheim, variation of the price is insignificant, however the peak and the low price difference may rise when the electricity market will apply hourly counting system to each household.

Designing proper systems gives a possibility to decrease or even avoid electricity consumption during the peak hours what can make significant savings and decrease peak loads on the grid. For further calculations 3 models of price difference have been applied to evaluate savings depending of the peak price (Figure 9).

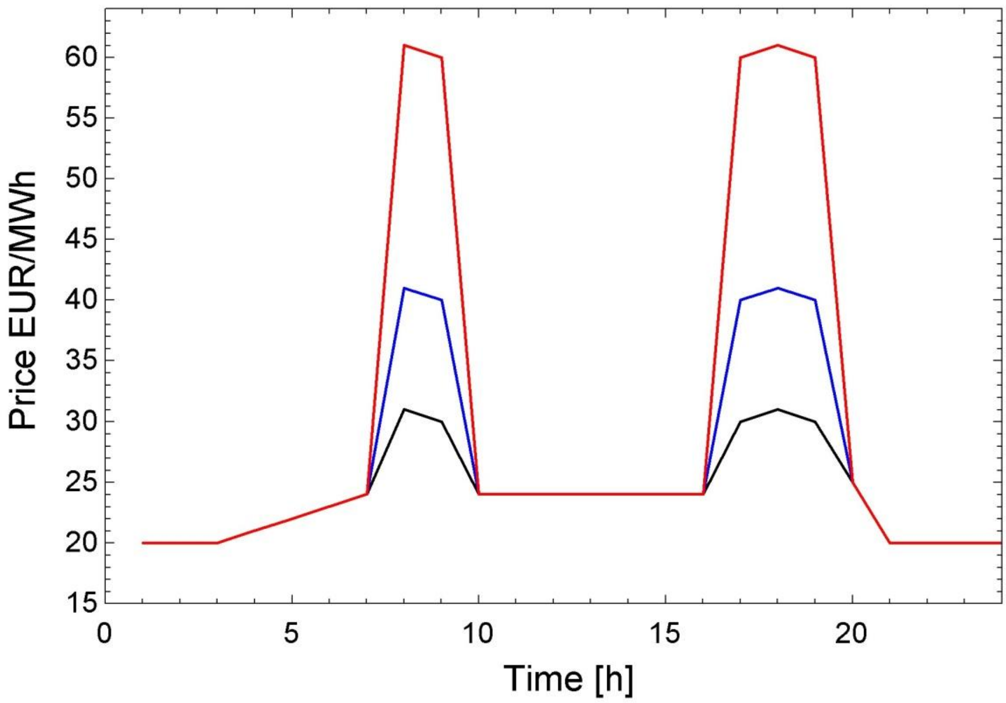


Figure 9 Different price tariffs during the day (EES)

4. SMALL AMMONIA HEAT PUMPS

Nowadays, heat pumps market does not offer small capacity ammonia heat pumps, however few pilot plants were made for scientific purposes. Analysis of those plants was made to evaluate specific components and designing methods regarding small capacity ammonia heat pump.

4.1 Plant No 1

One of the first projects in this field was done at Royal institute of Technology (KTH), Stockholm, Sweden by Björn Palm. Goal was to develop prototype of residential ammonia heat pump for small single-family house to provide space and hot tap water heating.

The heat pump was designed according to the regular ground source heat pump technology for borehole as a heat source. Necessary capacity for small single-family house was assumed as 9 kw for both space heating and hot tap water heating. Condensation temperature was selected as 40 ^oC to provide optimal temperature level for floor heating system. Evaporation temperature was taken according to the ground temperature - around 0 ^oC. Hot tap water heating was provided basically by desuperheating of discharge gases.

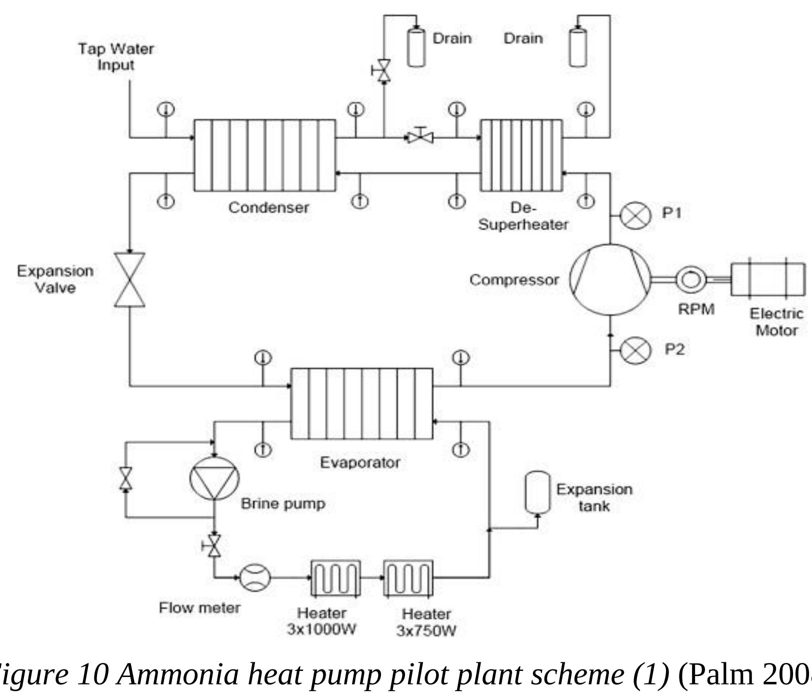


Figure 10 Ammonia heat pump pilot plant scheme (1) (Palm 2008)

Inlet water temperature to the condenser was kept constant at 40 ^{o}C and at the outlet of the desuperheater was fixed at temperature of 60^{o}C. Brine loop was heated with electrical heaters to provide necessary amount of heat to the evaporator. Because of low evaporation temperature, brine loop was filled with ethanol alcohol to avoid freezing.

Open ammonia piston compressor F2 from Bock was used with displacement of 10,5 m^3/h at 1450 rpm. The compressor was directly connected to 4,5 kW electrical engine. During the test period, 3 of expansion devices were changed. First TXV was TEA 20 from Danfoss. It was designed for large capacities therefore it was not be able to provide stable run of the system. Second was manually regulated valve SVA/REG 6-10 also from Danfoss. The third one was EXV Carel E2V from Carel. Despite EXV was designed for large capacities, it showed very good performance. Small fraction filter was installed to avoid blocking of the expansion device. First tests were made with AlfaNova plate heat exchangers as the evaporator (AlfaNova 52 with 20 plates), the condenser (AlfaNova 52 with 10 plates) and the desuperheater (AlfaNova 27 with 10 plates). Oil separator was not installed and it caused problems with oil returning to the compressor. The oil collected at the bottom of the evaporator and affected performance of the heat pump. The condenser and the desuperheater did not cause any problems with oil return. To solve this problem without installation of oil separator, redesigning was made to achieve refrigerant flow downwards from the inlet of the desuperheater to the inlet of the compressor. After redesigning there was no problem with oil returning to the compressor. However, evaporator performance was far from expected. Probably, because of bad distribution of the refrigerant and oil trapping in the passages of the plate heat exchanger. To improve distribution of the refrigerant, spiral channel type heat exchanger from Spirec was reviewed as a possible solution. However, this solution increased necessary refrigerant charge to several hundred grams and it made this heat exchanger as unacceptable due to safety requirements. At the final test were used self made 2 mini-channel (flat aluminium multiport tubes) heat exchangers as the evaporator and the condenser. They were specially designed for propane systems to reduce refrigerant charge in the system. Another type of oil (miscible with ammonia) and oil separator were used as well. This design improved performance and oil return. Total charge of the refrigerant was a little bit more than 100g.

The water flow (tap water input) ran through condenser and divided into two flows. Main part of the flow ran to simulated heating system and the other proceed way to the desuperheater. Output water of the desuperheater flowed through the heat exchanger inside of the hot tap water tank and returned back to the heating system tank. The flow to the desuperheater was regulated by the automatic 2-way valve to achieve necessary output temperature of at least 60 ^{o}C.

COP was measured by inlet and outlet temperatures of the compressor excluding efficiency of the electrical motor. The COP was in range of 3,8 - 4,8 at evaporation temperature from -13 ^{o}C to +2 ^{o}C and at condensation temperature of +40^{o}C. Hot water was supplied at temperature of 64 ^{o}C. The main problem to design this type of plant was lack of components, especially heat exchanger for evaporator. (Palm, 2008)

4.2 Plant No 2

Plant No 2 was designed few years later at the same university as the previous one. Heat pump application was the same - space and hot tap water heating for small single-family house. This plant was based on experience gained in previous study. Expected working conditions were almost the same - evaporation temperature of -5 to 0 ^{o}C and condensation temperature of 40 ^{o}C.

An open-type reciprocating HKT-GOELDNER O 12 3 DK100 compressor for ammonia was selected for this application. The compressor was coupled with TGT6-2200-20-560/T1P electrical motor with frequency controller. At given temperatures and at 1450 rpm of the compressor heating capacity provided by the manufacturer was 6 kW. For the desuperheater was chosen heat exchanger AlfaNova 27-10H and for the condenser AlfaNova 52-20H. As the evaporator was selected the most successful mini-channel aluminium heat exchanger from the first plant. As expansion device was selected EXV from Carel.

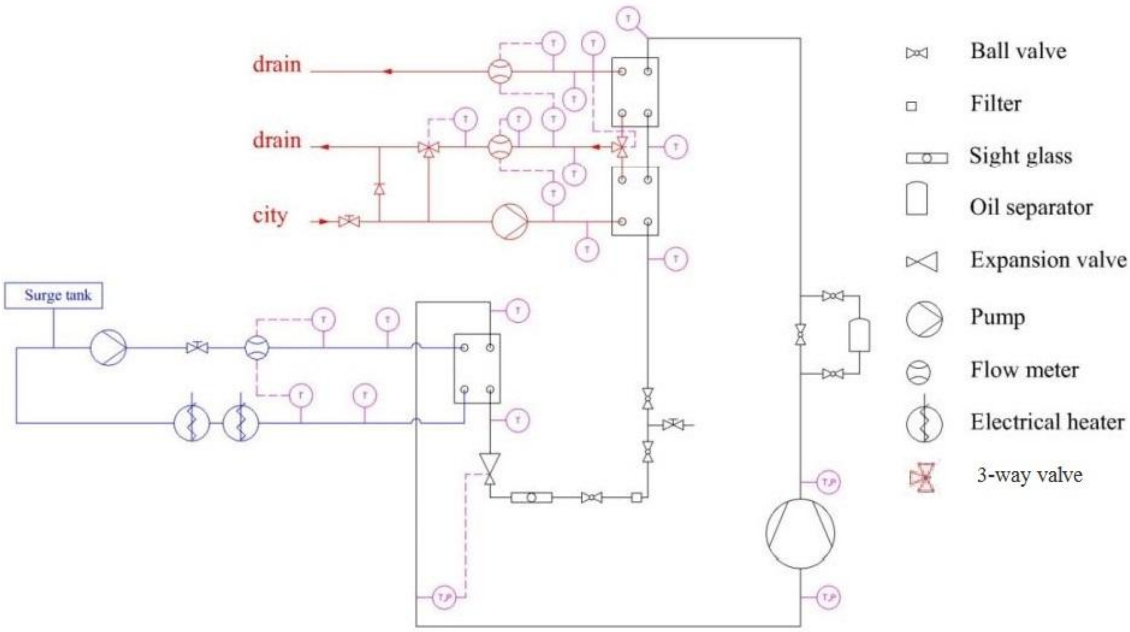


Figure 11 Ammonia heat pump pilot plant scheme (2) (Monfared 2010)

The water flowed inside of the condenser as in the first plant and at the outlet of the condenser was divided in two parts. Through automatic operated 3-way valve the one part flowed to the simulated heating system and the other to the desuperheater. At the outlet of the desuperheater the water temperature was fixed at 60^{o}C. The water flowed from the desuperheater to heat exchanger in the hot tap water tank. Then it returned to the tank of the heating system. It means that the heat pump was running constantly while control from water side was achieved.

All the components were placed in aluminium frame to adjust it to regular size of GHP. COP was 3,2 at evaporation temperature of -5 ^{o}C and condensation temperature of 40 ^{o}C. It was lower than COP of the previous plant because in the COP calculation was included electricity consumption both of the water pumps and the compressor.

This plant had no significant problems. However, evaporator efficiency was very poor and it was suggested to change it to another type of heat exchanger what would improve performance and make lower pressure drops. (Monfared, 2010)

4.3 Other applications

Another studies were performed in 2005 by Danish Technological institute. They developed and designed 6 plants with R723 for different applications in capacity range from 3 to 20 kW.

Table 3 Parameters of the plants

Plant	I	II	III
Refrigerating capacity	17-18 kW	10 kW	3 kW
Evaporating temperature	.-10℃ (t_{b_out} - 5℃)	-21℃	-24℃
Condensing temperature	.+30℃ (t_{w_out} +45℃)	28℃	+40℃
Plant	**IV**	**V**	**VI**
Refrigerating capacity	10 kW_o	10 kW_o	20 kW_o
Evaporating temperature	-10 ℃	-28℃	-14 ℃ (brine) $t_{Brine\ Outlet}$
Condensing temperature	42℃	42℃	34℃

(Plant I: Heat pump (Chiller) system for milk cooling, Plant II: Chiller system for marine applications, Plant III: Ice flake machine, Plant IV: refrigerating system for cooling cells (normal cooling - NC), Plant V: refrigerating system for cooling cells (deep freezing - DF), Plant VI: Brine chiller)

Basically, they used standard components available on market, such as copper tubes and hermetic compressors. Special oil return mechanisms were not developed. It was estimated that partial oil solubility in ammonia should provide guaranteed oil return, therefore installation of oil separator was renounced.

The heat pump system (Plant I) at the beginning was equipped with Maneurop SM 110-3 scroll hermetic compressor from Danfoss. Total charge of refrigerant was 2,5 kg (R723). The hermetic compressor operated only 5 hours. Afterwards there was a short circuit.

Figure 12 SM 110-3 compressor after breakdown (Danish Technological Institute, 2005)

The insulation of the inlet pipes was dissolved by the refrigerant. Dissolved varnish collected at the bottom of the stator windings. The plastic around the stator windings was dissolved. After this compressor there was installed semi hermetic reciprocating compressor Bock AM 3/233-4 which was suitable for ammonia. This compressor operated well until the end of the project.

Another unsuccessful try was at the ice flake machine (Plant III) with reciprocating compressor LT- 28-4VI from Danfoss. The compressor stopped after 4 days of being charged with R723.

Figure 13 LT- 28-4VI compressor after breakdown (Danish Technological Institute, 2005)

Conclusion for this compressor breakdown was damage of the varnish on the stator windings because of R723 use. This compressor was replaced with similar one and it operated only for 2 days before breakdown. Before the breakdown colour of the oil changed to white-brown and had some brown substance inside (dissolved varnish).

Materials, such as copper and brass, did not show any corrosion in short term operation. However, after 6 month some signs of corrosion were visible. In some plants problem with oil return occurred. As a solution was evaporator feeding from the top or use of flooded evaporator and it provided good oil return. However, this solution affected performance of the evaporator. The most suitable compressors for ammonia are reciprocating compressors from Bock (open type) and Frigopol (separated-hood). They were used in the other plants and no problems were detected. (Danish Technological Institute, 2005)

4.4 Design features

According to the information collected about standard and ammonia heat pumps it is possible to highlight some design features which should be taken into account in new residential ammonia heat pump construction.

Due to high ammonia discharge gas temperature, it is efficient to use it for both space heating and hot tap water heating. Usually, for this purpose is installed one more heat exchanger (desuperheater) which cools down discharge gases thereby providing high water temperature. Due to limit of discharge temperature, the most suitable condensation temperature is around $40^{o}C$ what is totally enough for underfloor heating. Both space heating system and hot tap water heating system should be equipped with storage tanks. Volume of the tanks affects heat pumps efficiency and lifetime of the compressor. There are two possible solutions how to connect tanks to the heat pump. One of them is to use separate loops for the heating system and the hot tap water. In this case capacity control is achieved by regulating of refrigerant flow. The other solution is to make only one water loop, which at the beginning goes in to the condenser and afterwards divides in two flows (regulated by 3-way valve). One goes to the heating system tank and the other goes to the desuperheater. After the desuperheater it goes through heat exchanger in the hot tap water tank and returns to the tank of the heating system.

Components suitable for ammonia should be used. Experimental plants showed that compressors, which are not specially designed for ammonia lasts not longer than 4 days. The best compressors for small ammonia systems are open type from Bock or separated hood from Frigopol. For heat sink the best heat exchangers are stainless steel plate heat exchangers, such as Alfa Nova. Very often plants had problems with oil return because of oil accumulation in the evaporator. As a solution is to feed evaporator from the top and make whole flow of refrigerant downwards. However, no good enough evaporator has been found during the experiments. For better performance it was suggested to use single-channel heat exchanger which makes better refrigerant distribution and lowers possibility of oil accumulation. As an expansion device EXV is more preferable because of accurate operation. Stainless steel piping and valves should be used as well.

5. COMPONENTS

According to the collected information small ammonia heat pump for single-family house has been designed. Estimated heat load during the winter coldest day for single family house located in Trondheim, Norway was 7,5 kW for space heating and 2 kW for hot tap water heating. Design solution for water connections is similar to regular GHP application (Figure 5) with 2 separate loops for space heating and hot tap water heating. As a heat source will be used borehole with depth of 150m. The main components have been chosen according to simple heat pump cycle. Stainless steel valves and piping should be used to avoid corrosion possibility. Data for the pumps has been taken from previous work done for this topic. (Bjørneklett, 2013)

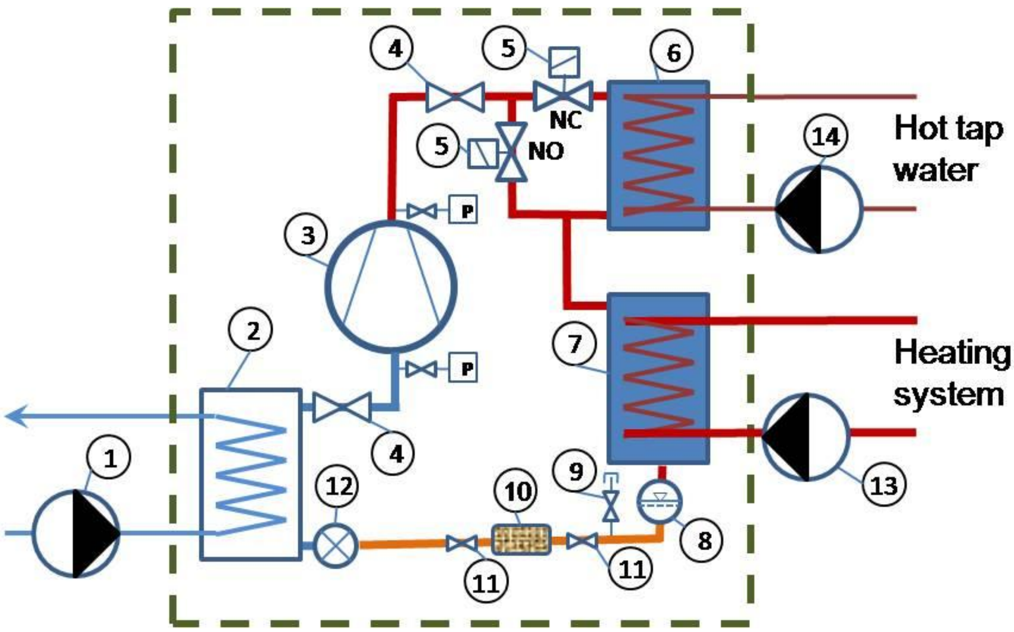


Figure 14 Principal scheme of the ammonia heat pump (Bjørneklett, 2013)

(1 - pump for ground loop MAGNA 25-100 (m=0,3kg/s); 2 - evaporator; 3 - compressor; 4 - shut off valves for the compressor; 5 - solenoid valves; 6 - desuperheater; 7 - condenser; 8 - receiver; 9 - pressure release valve;10 - mechanical filter;11 - shut off valves for the mechanical filter;12 - expansion valve; 13 - pump for the heating system ALPHA2 25-40 (m=0,3kg/s); 14 - pump for the hot tap water heating system ALPHA2 25-40 (m=0,1 kg/s))

5.1 Compressor

Considering previous experience of the small ammonia heat pump plants regular compressors should not be used in applications with ammonia. For that reason 10-DLZC-2.2 reciprocating separate hood compressor from Frigopol has been chosen. This compressor is specially designed for applications with R717 or R723. Refrigerating capacity of the compressor is 7kW at condensation temperature of 40^{o}C and evaporation temperature of -3^{o}C. 40^{o}C is the highest possible condensation temperature the compressor can achieve. This limitation is related to the discharge gas temperature which is normally very high for the ammonia. Usually, this type of compressor is used for refrigeration applications. Because of the separating hood between rotor and stator, any chemical reaction between ammonia and the copper winding of the motor is absolutely impossible (Figure 15). Compared to the open type compressors, separate hood compressors do not have any drive element from the motor to the compressor and it makes them very reliable and long lasting. It is suggested to use MOBIL SHC 226 oil compatible with ammonia. (FRIGOPOL)

The compressor is supplied with frequency controller from Danfoss which allows to work at 40% of the full capacity.

Figure 15 Separate hood compressor from Frigopol (FRIGOPOL)

5.2 Evaporator

As previous studies shown, choice of evaporator is very important, because it is related to problems with refrigerant distribution and oil return. It was suggested to use one channel heat exchanger to be sure that refrigerant distribution will be good enough. Another useful suggestion was to feed evaporator from the top. This type of flow provides good oil return and does not allow to trap oil in the evaporator.

For this application special stainless steel coaxial tube-in-tube heat exchanger from SEAB GmbH Kleinostheim has been chosen (Figure 16).

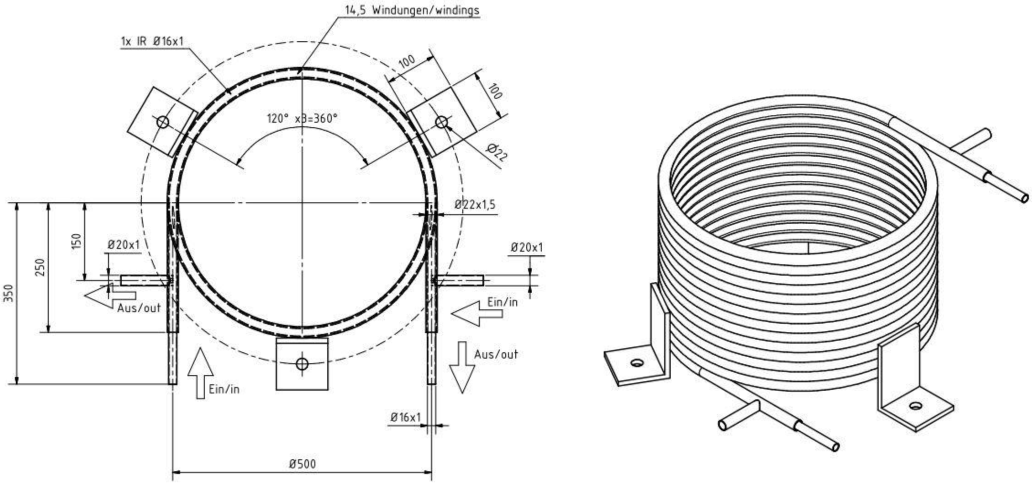


Figure 16 Coaxial tube-in-tube heat exchanger

For the evaporator parallel flow pattern has been chosen because of better temperature suitability. Brine HX-35 (25 % ethanol alcohol and 75 % water mixture) will be circulating in the inner pipe and the ammonia in the outer pipe of the heat exchanger. The evaporator capacity given by manufacturer is 6.5 kW at the evaporation temperature of $0^{o}C$ and the water inlet temperature of $8^{o}C$ with the mass flow rate of 0.3 kg/s. Feeding of the refrigerant to the evaporator should be achieved from the top to avoid problems with the oil return. Internal heat exchanger will not be used therefore suction gas superheating should be achieved in the evaporator to avoid droplets of liquid ammonia in the compressor. Heat transfer area is 1 m^2 with total length of 22,76 m.

5.3 Desuperheater and condenser

Two similar Alfa Nova 27-20H stainless steel plate heat exchangers (PHX) from Alfa Laval have been chosen as the desupearheater and the condenser. Those heat exchangers are specially designed for applications with aggressive medias, such as ammonia. At the previous plants this type of heat exchangers showed very good performance without causing any oil return problems in hot gas and condensation side. To achieve highest possible outlet water temperatures counter-flow should be used.

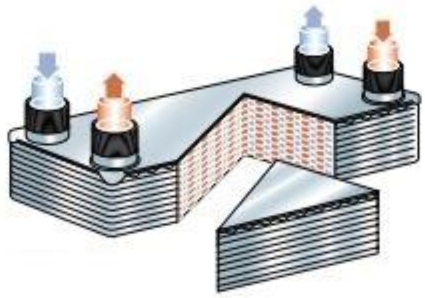


Figure 17 Flow principle of an Alfa Nova plate heat exchanger (Alfa Laval)

Each heat exchanger consists of 20 plates with specific flow pattern which depends on corrugation angle. Corrugation angle affects heat transfer coefficient and pressure drop. Inlet and outlet connections on the water side have threads but on the ammonia side they are prepared for welding. Maximum working pressure at the temperature of 225^{o}C is 23 bar. Heat transfer area is 0,45 m^{2}. (Alfa Laval)

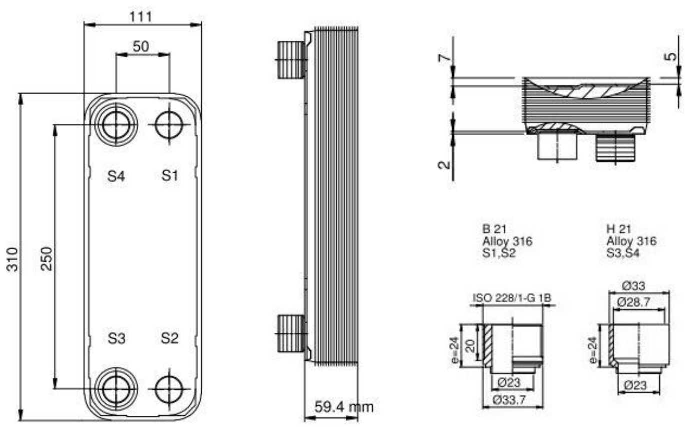


Figure 18 Dimensions of Alfa Nova 27-20H plate heat exchanger

5.4 Expansion device

Expansion device AKVA-10-2 from Danfoss has been selected for this application. AKVA is electric expansion valve specially designed for ammonia refrigerating plants. Individual capacity of the valve can be adjusted by changing orifice. With orifice No. 2 rated capacity of the valve is 6.3 kW. Design of the valve is shown in the Figure 19.

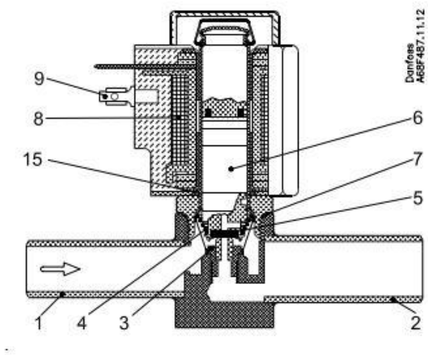


Figure 19 AKVA 10-2 expansion device
(1- inlet, 2 - outlet, 3 - orifice, 4 - filter, 5 - valve seat, 6 - armature, 7 - aluminium gasket, 8 - coil, 9 - DIN plug, 15 - o-ring) (Danfoss)

The valve capacity is regulated by means of pulse-width modulation. Within a period of six seconds a voltage signal from the controller will be transmitted to and removed from the valve coil. This makes the valve open and close for the flow of refrigerant. It can be used both as expansion and solenoid valve. Mechanical filter with fraction of 100μm should be installed before EXV to avoid blocking possibility. (Danfoss)

6. CALCULATIONS

6.1 Engineering Equation Solver

Calculations with engineering equation solver (EES) have been done to estimate heat pump performance and evaluate necessary size of water tank for the heating system. Obtained heat transfer coefficients are available in Appendix 2. Flow diagram of the calculations described in Appendix 3.

EES is equation solving program that can numerically solve algebraic and differential equations. A major feature of EES is high accuracy thermodynamic and transport property database that is provided for hundreds of substances. It makes it very suitable for hydraulic and heat transfer calculations. (Klein)

6.2 Compressor efficiency

At the beginning simple heat pump cycle has been modelled. According to the compressor technical data have been calculated mass flow rate, pressure ratio, volumetric and isentropic efficiencies at different condensation and evaporation temperatures. Compressor refrigerating capacity is given with 10K of superheating.

Table 4 Frigopol 10-DLZ-2.2 compressor refrigerating capacity (kW)

T_{evap} °C / T_{cond} °C	-20	-15	-10	-5	0	5	10	15
20	3,11	4,2	5,53	7,13	9,01	11,2		
30	2,8	3,77	4,97	6,42	8,13	10,15	12,47	15,14
40			4,43	5,73	7,29	9,12	11,25	13,7

Entered all this data to EES it was possible to calculate mass flow rate from Equation 1 at different evaporation and condensation temperatures. Pressure ratio was calculated from Equation 2 as a relation between condensation and evaporation pressures. Necessary displacement volume is obtained from Equation 3. Volumetric efficiency (Equation 4) is relation between necessary and real swept volume of the compressor (given by the manufacturer). Isentropic efficiency is obtained from the compressor power data given by manufacturer relation with necessary power for ideal compressor cycle.

$$Q_{ref} = m_a(h_1 - h_5) \text{ [kW]} \tag{1}$$

$$\pi = \frac{P_{cond}}{P_{evap}} \tag{2}$$

$$V_{necessary} = m * v_{comp.in} \text{ [m}^3\text{/s]} \tag{3}$$

$$\lambda = \frac{V_{necessary}}{V_{swept}} \tag{4}$$

$$W_{ideal} = m_a(h_2 - h_1) \text{ [kW]} \tag{5}$$

$$\eta_{is} = \frac{W_{real}}{W_{ideal}} \tag{6}$$

m_a - *mass flow of the refrigerant [kg/s]*

$h_2;h_1;h_5$ - *enthalpies in the related points [kJ/kg]*

$\upsilon_{comp.in}$ - *specific volume of the refrigerant at the compressor inlet [m^3/kg]*

For further calculations all necessary parameters of the compressor were expressed as a functions at condensation temperature of 40°C because this temperature is suitable for floor heating application. EES feature "Curve Fit" has been used to obtain functions from following graphs (Figures 20, 21, 22, 23).

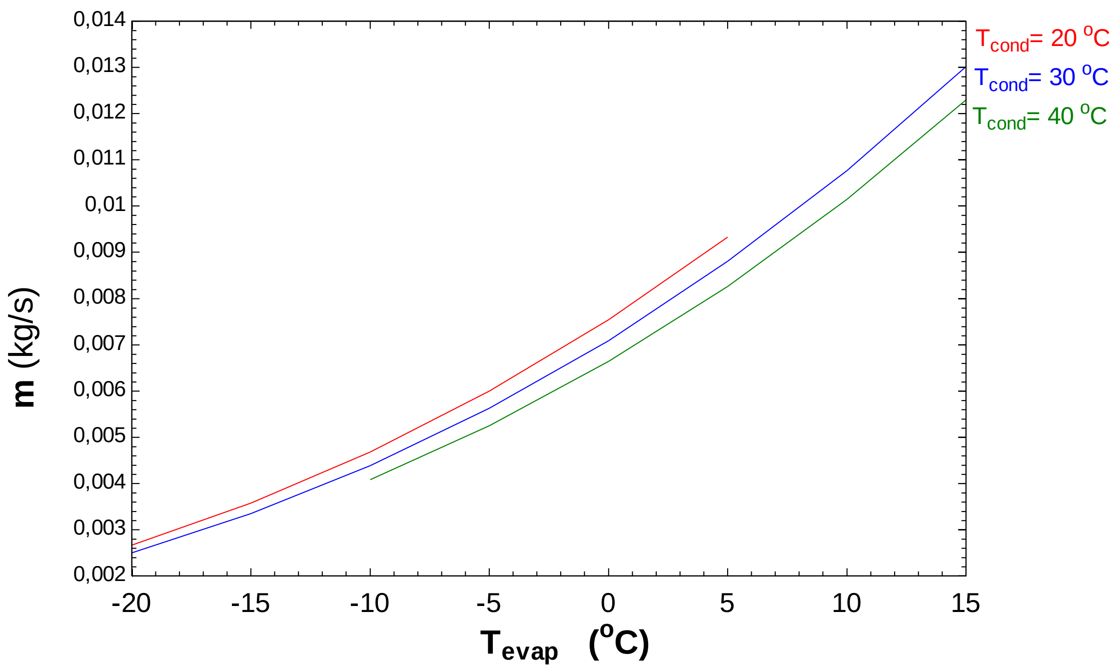


Figure 20 Mass flow rate as a function of T_{evap} at different T_{cond} (EES)

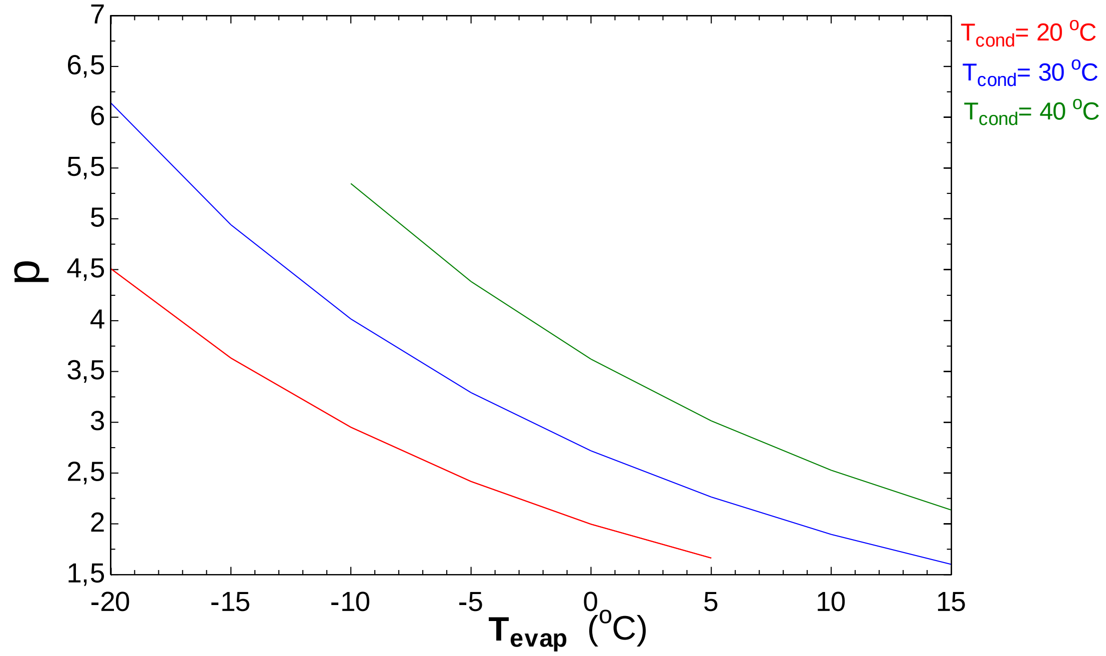


Figure 21 Pressure ratio as a function of T_{evap} at different T_{cond} (EES)

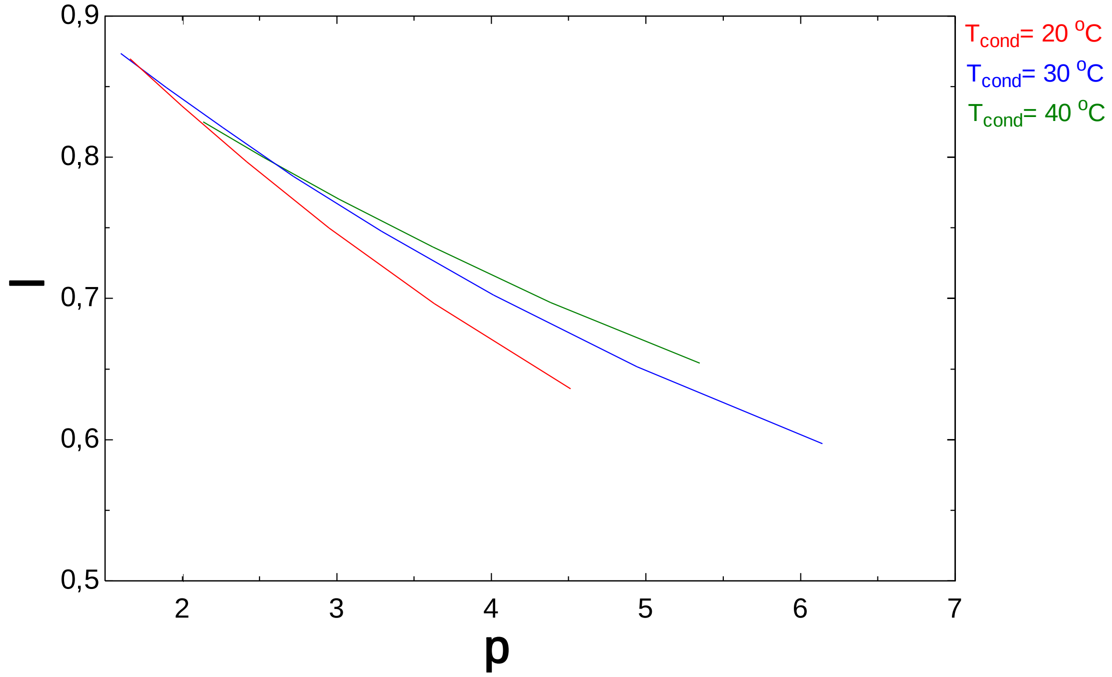


Figure 22 Volumetric efficiency as a function of pressure ratio at different T_{cond} (EES)

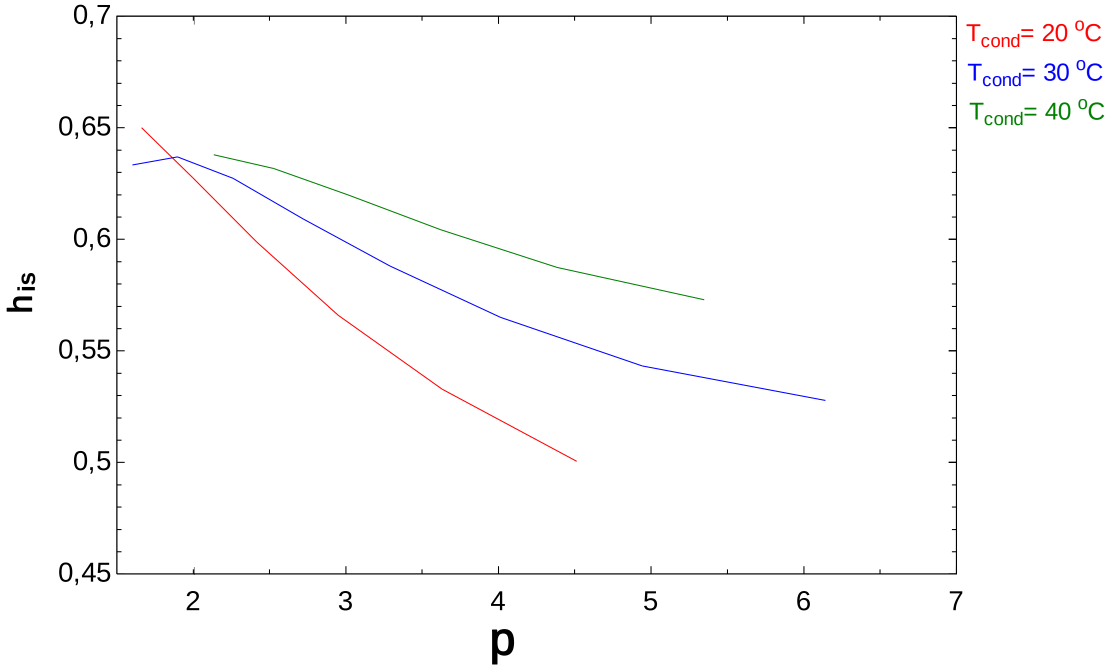


Figure 23 Isentropic efficiency as a function of pressure ratio at different T_{cond} (EES)

Pressure ratio and mass flow rate are expressed as a function of the evaporation temperature T_{evap} (Equations 7, 8). Volumetric and isentropic efficiencies are expressed as functions of pressure ratio (Equations 9, 10). All those functions are related to condensation temperature of 40°C and evaporation temperature range from -10°C to +15 °C.

$$\pi = 3{,}65764exp(-0{,}0367333T_{evap}) \quad (7)$$

$$m_a = 0{,}0065054exp(0{,}0440417\ T_{evap})[kg/s] \quad (8)$$

$$\lambda = 0{,}95981exp(-0{,}0723387\,\pi) \quad (9)$$

$$\eta_{is} = 0{,}687633exp(-0{,}0348549\,\pi) \quad (10)$$

All system process has been visualized at logarithmical pressure/enthalpy (Figure 24) and temperature/entropy (Figure 25) diagrams to make obvious understanding of the heat pump cycle. All values for entropy, enthalpy, vapour quality and pressure were obtained at different points of the cycle.

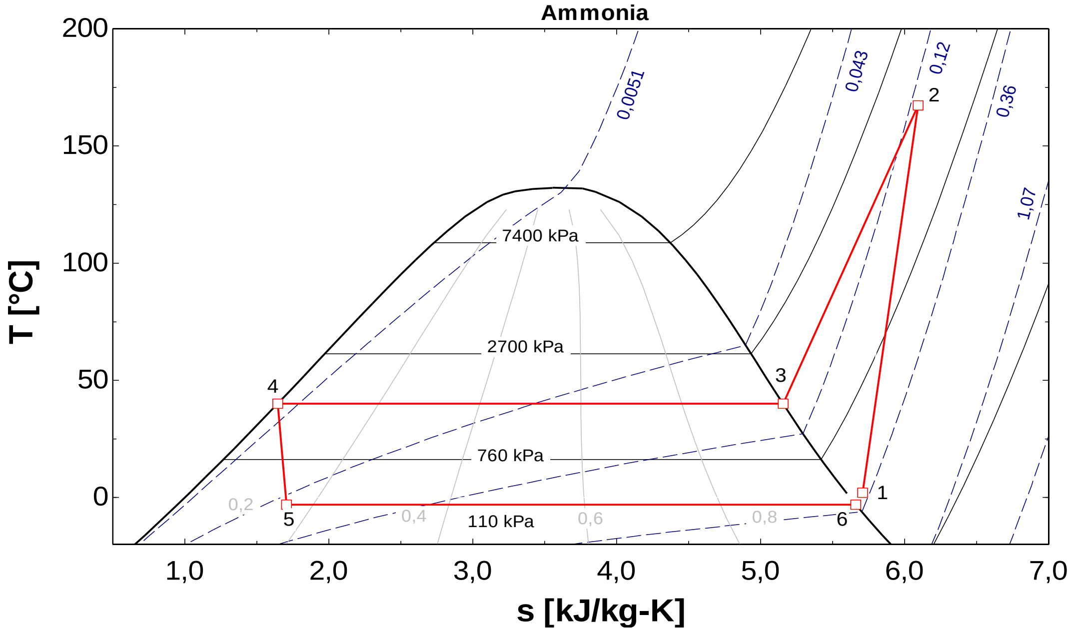


Figure 24 Ammonia temperature/entropy diagram T_{cond}=40^oC, T_{evap}=-3^oC (EES)

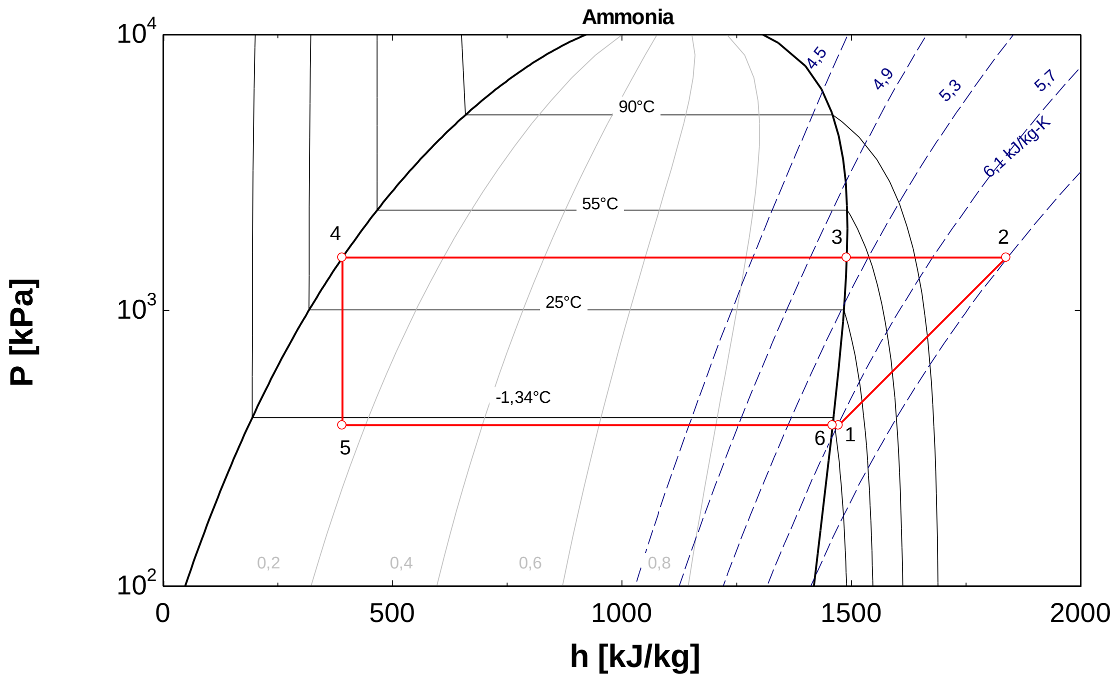


Figure 25 Ammonia logarithmical pressure/enthalpy diagram T_{cond}=40^oC, T_{evap}=-3^oC (EES)

At this part also estimation of the heat pump performance has been made, according to the efficiency of the compressor. Condensation capacity is calculated from the mass flow rate and the enthalpies between points 3 and 4 (Equation 11). Capacity of the desuperheater has been estimated by the same principle between points 2 and 3 (Equation 12). COP is calculated as a ratio between rejected heat (desuperheating and condensation) relation with real work added to the compressor (Equation 13).

$$Q_{cond} = m_a(h_3 - h_4) \tag{11}$$

$$Q_{des} = m_a(h_2 - h_3) \tag{12}$$

$$COP = \frac{Q_{cond} + Q_{des}}{W_{real}} \tag{13}$$

Flexible diagram window has been made to show all points and parameters of the heat pump at different evaporation temperatures in range of -10°C < T_{evap} < 15 °C (Figure 26). Condensation temperature is assumed to be 40°C because it is maximal condensation temperature of the compressor and it corresponds to the temperature regime for the floor heating system.

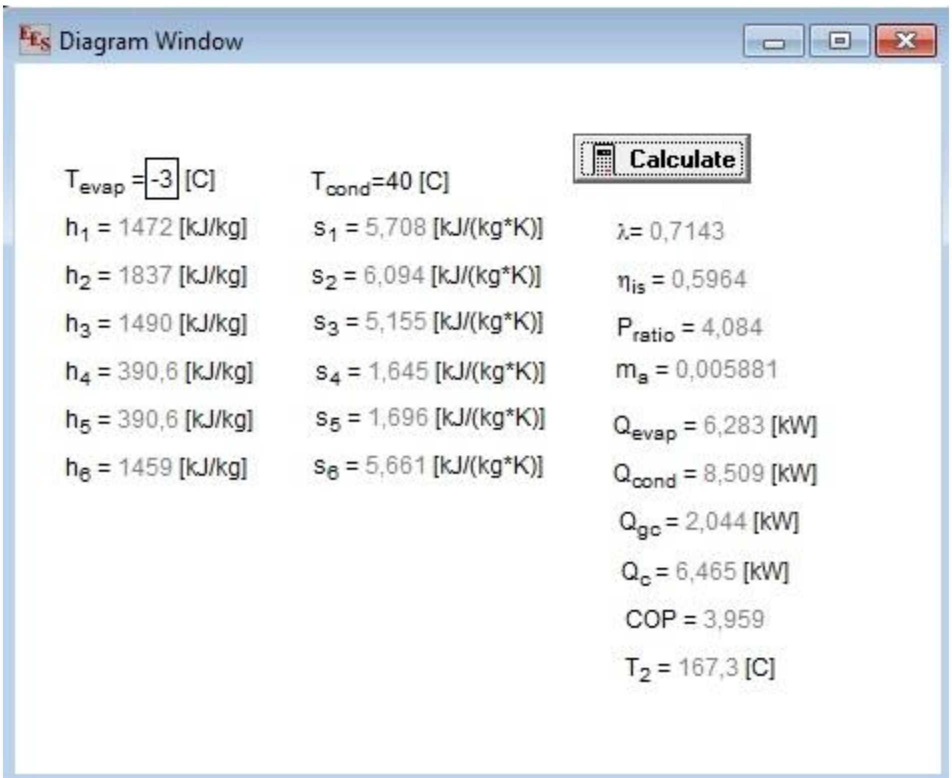


Figure 26 Diagram window with parameters of the heat pump (EES)

6.3 Evaporator performance

As the evaporator has been chosen coaxial tube in tube heat exchanger with parallel flow of refrigerant and HX-35 (water and 25% ethanol alcohol mixture) further in text - water. Heat transfer model has been made to evaluate evaporation capacity and performance at different flows. At the beginning all necessary geometric calculations have been made. From Equation 14 and 17 was calculated cross-sectional area of the pipe. From Equations 15 and 18 wetted perimeter and from Equations 16 and 19 hydraulic diameter of the pipe.

Water side (inner tube):

$$A_w = \pi r^2 \ [m^2] \tag{14}$$

$$P_w = 2\pi r \ [m] \tag{15}$$

$$d_{hw} = \frac{4A_w}{P_w} \ [m] \tag{16}$$

r- radius of the inner pipe [m]

Ammonia side (outer tube):

$$A_a = \pi(R^2 - r^2) [m^2] \tag{17}$$

$$P_a = 2\pi(R + r) \ [m] \tag{18}$$

$$d_{ha} = \frac{4A_a}{P_a} \ [m] \tag{19}$$

R - radius of the outer pipe [m]

Heat transfer coefficient from the water side is calculated by correlation for helically coiled tubes offered by VDI Heat Atlas. (Gnielinski, et al., 2010)

To determine character of the flow critical (Equation 20) and actual (Equation 21) Reynolds numbers are calculated. Prandtl number and other fluid properties are obtained from EES library.

$$Re_{crit} = 2300 \left[1 + 8{,}6 \left(\frac{d_{hw}}{D}\right)^{0{,}45}\right] [\ -\ ] \tag{20}$$

$$Re_w = \frac{m_w d_w}{\mu} [\ -\ ] \tag{21}$$

D - diameter of the coil [m]

μ - dynamic viscosity [kg/m s]

For laminar flow when $Re_w < Re_{crit}$ is suggested Equation 22. For turbulent flow when $Re_w > Re_{crit}$ should be used Equations 24 and 25. Heat transfer coefficient is obtained from Equation 27.

$$Nu_l = 3{,}66 + 0{,}08 \left[1 + 0{,}8 \left(\frac{d_{hw}}{D}\right)^{0{,}9}\right] {Re_w}^i Pr^{1/3} \left(\frac{Pr}{Pr_w}\right)^{0{,}14} [\ -\ ] \tag{22}$$

$$i = 0{,}5 + 0{,}2903\left(\frac{d_w}{D}\right)^{0{,}194} \; [\text{ - }] \tag{23}$$

$$Nu_t = \frac{(\xi/8)Re_w Pr}{1+12{,}7\sqrt{\xi/8}(Pr^{2/3}-1)}\left(\frac{Pr}{Pr_w}\right)^{0{,}14} \quad [\text{ - }] \tag{24}$$

$$\xi = \left[\frac{0{,}3164}{Re_w{}^{0{,}25}} + 0{,}03\left(\frac{d_w}{D}\right)^{0{,}5}\right]\left(\frac{\mu_w}{\mu}\right)^{0{,}27} \; [\text{ - }] \tag{25}$$

$$Pr = \frac{c_p \mu}{k} \; [\text{ - }] \tag{26}$$

$$\alpha_w = \frac{kNu}{d_w} \; [W/\, m^2\, K] \tag{27}$$

Pr ,Pr_w - Prandtl's number at the saturation and at the wall temperature respectively [-]

μ, μ_w - dynamic viscosity at the saturation and at the wall temperature respectively [kg/m s]

cp - specific heat capacity [kJ/kg K]

k - thermal conductivity [W/m K]

Calculation of heat transfer coefficient from the ammonia side is very complicated due to flow geometry and two phase flow. Therefore has been made assumption, that all heat from the water side will be transferred to the ammonia side. This assumption is based on ammonia heat transfer coefficients which are normally very high. Figure 27 shows principle of the heat transfer which is used for the calculations. Equilibrium of the heat transfer is shown in Equation 28.

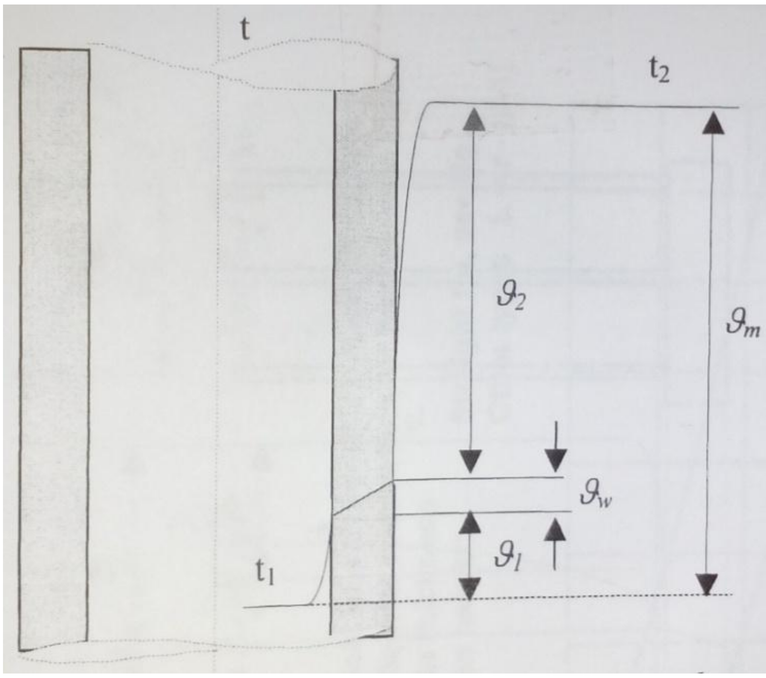


Figure 27 Heat transfer through the wall (Granryd, et al., 2005)

$$Q = UA\vartheta_m = \alpha_1 A_1 \vartheta_1 = k_w A_w \vartheta_w / \delta = \alpha_2 A_2 \vartheta_2 \ [kW] \qquad (28)$$

U - overall heat transfer coefficient [kW/ m^2 K]

A - overall heat transfer area [m^2]

ϑ_m - logarithmical mean temperature difference [K]

α_1- heat transfer coefficient from the refrigerant side [kW/m^2 K]

A_1 - heat transfer area from the refrigerant side [m^2]

ϑ_1 - temperature difference between the refrigerant and the wall[K]

α_2 - heat transfer coefficient from the water side [kW/m^2 K]

A_2 - heat transfer area from the water side [m^2]

ϑ_2 - temperature difference between the wall and the water [K]

A_w - heat transfer area of the wall [m^2]

ϑ_w - temperature difference between both sides of the wall [K]

k_w - thermal conductivity of the wall [W/m K]

δ - thickness of the wall [m]

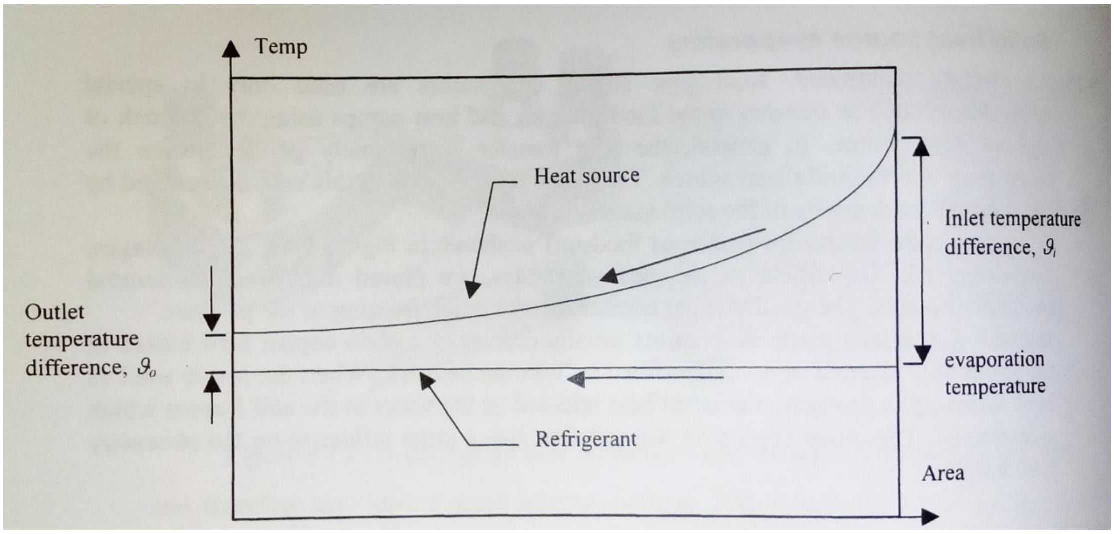


Figure 28 Temperature profile in evaporator (Granryd, et al., 2005)

$$\vartheta_m = \frac{\vartheta_i - \vartheta_o}{\ln\left(\frac{\vartheta_i}{\vartheta_o}\right)} \ [K] \qquad (29)$$

In real process evaporation curve is not straight as in Figure 28 due to pressure drop. The pressure drop usually is expressed in change of temperature, which affects logarithmical mean temperature difference shown in Equation 29. Due to lack of correlations for evaporation process in given geometry flow there has been used correlation for two phase flow pressure drop in straight tube using hydraulic diameter of the existing tube. Momentum pressure drop is insignificant therefore it is not taken into account. (Granryd, et al., 2005)

Following equations have been used:

$$\Delta p_f = f_m G^2 v_m \frac{L}{d_{ha}} \ [kPa] \qquad (30)$$

$$f_m = 0{,}0185 K_f^{1/4} Re_a^{-1/4} \ [\text{ - }] \qquad (31)$$

$$v_m = x_m v_m'' \ [m^3/kg] \qquad (32)$$

$$x_m = 4{,}4\, d_{ah}{}^{1/4} L^{-1/2} \ [\text{ - }] \qquad (33)$$

$$K_f = \frac{\Delta h}{Lg} \ [\text{ - }] \qquad (34)$$

f_m - friction factor [-]

K_f - Pierre's boiling number [-]

Δh *- change in enthalpy [J/kg]*

Re_a *- Reynolds number calculated with liquid ammonia parameters [-]*

v_m $(1/\rho_m)$ *- mean density [m^3/kg]*

v'' *- specific volume of the vapour at average* T_{evap} *[m^3/kg]*

x_m *- mean vapour fraction [-]*

L *- length of the pipe [m]*

Pressure drop was expressed as temperature and adjusted linearly to evaporation curve. Wall temperature was assumed as 1°C higher than evaporation temperature. To modulate evaporation process in the heat exchanger it was divided in 100 sections. Heat transfer in each section has been calculated. Water temperature from each section was obtained from Equation 36 considering the heat transfer equilibrium. Enthalpy for ammonia has been calculated from Equation 37. Through EES features vapour quality has been calculated in each section as well. When vapour quality reached 100% additional Equation 38 has been introduced to calculate temperature rising of superheated ammonia gas. Model sample shown in Figure 29.

$$Q_{section} = \alpha_1 A_1 \vartheta_1 \ [kW] \tag{35}$$

$$Q_{section} = m_w c_{pw} (T_{w,in} - T_{w,out}) \ [kW] \tag{36}$$

$$Q_{section} = m_a (h_{a,in} - h_{a,out}) \ [kW] \tag{37}$$

$$Q_{section} = m_a c_{pa} (T_{a,in} - T_{a,out}) \ [kW] \tag{38}$$

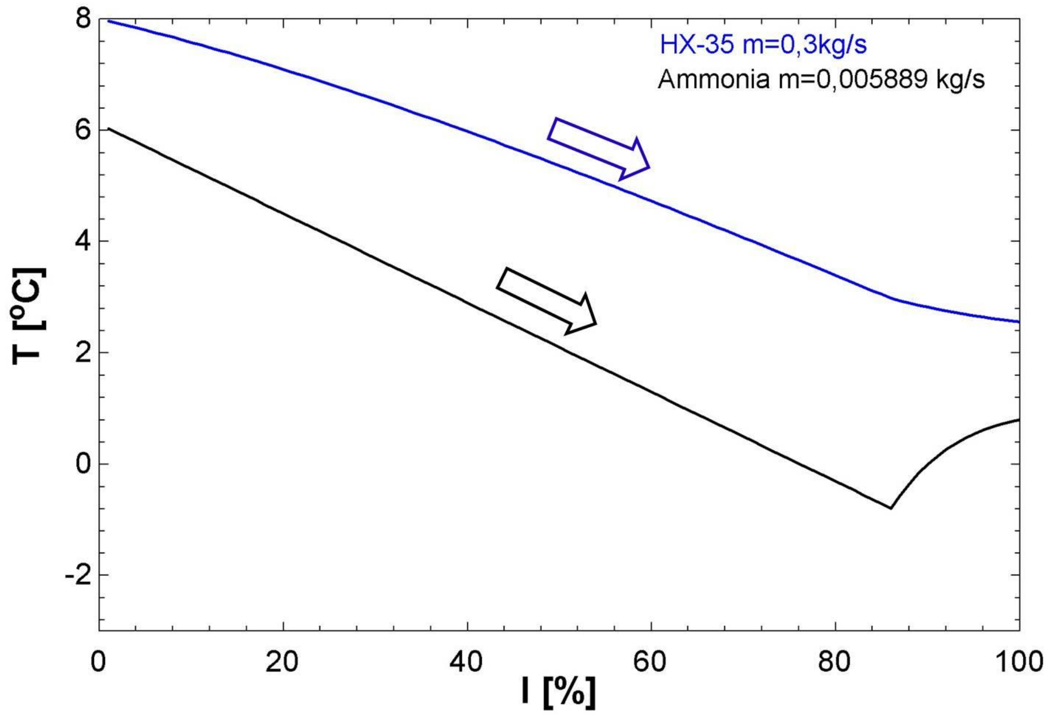


Figure 29 Evaporator temperature curves at T_{evap}=-3°C (EES)

Inlet brine temperature was assumed as 8°C without changes during the year, because borehole is deep enough (150m) to not be influenced by the variation of ambient air temperature. Ground loop goes through ground water layers therefore possibility of extracting all useful heat from the ground is neglected.

Calculated ammonia pressure drop is 1,33 bar which is 9°C expressed in the temperature. It means that higher possible evaporation temperature is -3°C. At T_{evap}=-3°C and brine mass flow rate of 0,3 kg/s it is sufficient to achieve total evaporation of the ammonia at 85% of total length of the heat exchanger. With lower evaporation temperature total evaporation appears faster. For further calculation data related to T_{evap}=-3°C has been taken.

6.4 Desuperheater and condenser performance

As the desuperheater and the condenser have been chosen 2 similar Alfa Nova 27-20H brazed plate heat exchangers. Provided capacity from the manufacturer was 1,5 kW for the desuperheater ($T_{ammonia\_in}$=130^{o}C; $T_{ammonia\_out}$=40^{o}C, $m_{ammonia}$=0,001096 kg/s; $\Delta P_{ammonia}$=0,005 kPa and T_{water_in}=35^{o}C; T_{water_out}=40^{o}C; m_{water}=0,07181kg/s; $\Delta P_{ammonia}$=0,426 kP) and 8,7 kW for the condenser ($T_{ammonia\_in}$=130^{o}C; $T_{ammonia\_out}$=40^{o}C, $m_{ammonia}$=0,0063 kg/s; $\Delta P_{ammonia}$=0,15 kPa and T_{water_in}=30^{o}C; T_{water_out}=37^{o}C; m_{water}=0,29 kg/s; $\Delta P_{ammonia}$=0,426 kP).

Due to changing conditions during different heat pump working regimes additional calculation for those heat exchangers has been made. Obtained mass flow for ammonia and inlet gas temperature also differs from data given by the manufacturer. In this calculation heat exchangers have been divided in 20 sections each. Heat transfer in each section has been calculated and plotted.

The heat transfer and the pressure drop characteristics in PHEs are related to the hydraulic diameter, the increased heat transfer area, the number of the flow channels, and the profile of the corrugation waviness, such as the inclination angle, the corrugation amplitude, and the corrugation wavelength.

At the beginning all geometric calculations have been made. The hydraulic diameter of the channel is calculated from Equation 39.

$$D_h = \frac{4A_f}{Pw} = \frac{4bL_w}{2L_w\varphi} = \frac{2b}{\varphi}\ [m] \quad (39)$$

A_f - cross flow area [m^2]

P_w - wetted perimeter [m]

L_w - horizontal length of the plates [m]

φ is the enlargement factor which is given by manufacturer (usually 1,17). The mean channel spacing, b, is defined as

$$b = p - t\ [m] \quad (40)$$

t - plate thickness [m]

Where p is the plate pitch (Equation 41).

$$p = \frac{L_c}{N_t - 1}\ [m] \quad (41)$$

L_c - distance between the end plates [m]

N_t - total number of plates

Heat transfer coefficient from the water side is calculated from following equation. (Lee, et al., 2003)

$$\alpha_w = 0{,}295\left(\frac{k_w}{nD_h}\right)Re_w^{0,64}Pr^{0,32}\left(\frac{\pi}{2}-\beta\right)^{0,09}\ [W/\ m^2\ K] \qquad (42)$$

k_w - water thermal conductivity [W/m K]

β- chevron angle [rad]

n - number of channels on the plate

Reynolds number for water is obtained from Equation 21.

Condensation heat transfer is obtained from correlation for condensation on vertical surfaces. Depending on flow pattern (laminar or turbulent) 2 equations have been used. (Granryd, et al., 2005)

For laminar flow (Re < 1800) has been used Equation 43 and for turbulent (Re > 1800) Equation 44.

$$\alpha_l = 0.943\left(\frac{\rho_l g(\rho_l-\rho_v)k_l}{\mu_l L^3(t_{sat}-t_{wall})}\right)^{1/4}\ [W/\ m^2\ K] \qquad (43)$$

$$\alpha_t = 0.0077\left(\frac{k_l^3\rho_l(\rho_l-\rho_v)g}{\mu_l^2}\right)^{1/3}Re^{0.4}\ [W/\ m^2\ K] \qquad (44)$$

ρ_l - density of ammonia liquid [kg/m^3]

ρ_v - density of ammonia vapour [kg/m^3]

k_l - conductivity of liquid ammonia [W/m K]

t_{sat} - temperature of saturated refrigerant [K]

t_{wall} - temperature of the refrigerant at the wall temperature [K]

L- length of the surface [m]

g - gravitational acceleration [m/s]

Heat transfer coefficient for superheated gas has been obtained from following equation,

$$\frac{1}{\alpha_g} = \frac{1}{U} - \frac{1}{\alpha_w} - R_{wall}\ [W/\ m^2\ K] \qquad (45)$$

U- overall heat transfer coefficient given by the manufacturer [W/m^2 K]

where R_{wall} is resistance of the wall (Equation 46).

$R_{wall} = \frac{\delta}{k_w}$ [W/ m^{2}K] (46)

Additional calculation of pressure drop for the refrigerant at high pressure side has been skipped due to insignificant impact on the heat pump cycle.

For condensation and desuperheating counter-flow pattern is most effective because of good temperature glide. In case of the counter-flow (Figure 30) it is possible to reach higher possible outlet temperatures.

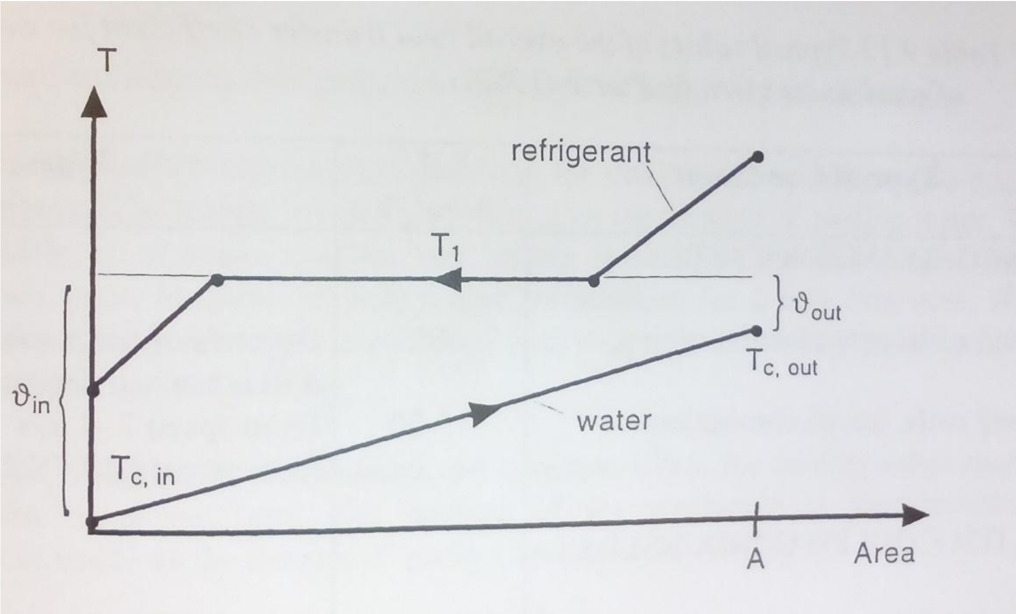


Figure 30 Temperature glide in counter-flow heat exchangers ($T_{C,in}$, $T_{C,out}$ - temperatures of the cold fluid; $T_{H,in}$, $T_{H,out}$ - temperatures of the hot fluid) (Granryd, et al., 2005)

To modulate heat transfer process in the heat exchangers with counter flow pattern LMTD method has been used. Briefly, this method is about outlet temperature prediction using simple heat exchanger energy balance. Afterwards those temperatures are included in the heat transfer calculation from the one side of the heat exchanger. At the end temperatures are adjusted to correspond with the inlet fluid temperatures. (Nellis, et.all., 2009)

Similar principle with the heat transfer equilibrium like in the evaporator has been applied for the heat transfer calculations in each section of the heat exchanger (Equation 28). Equation 47 describes heat transfer from the refrigerant side, 48 through the wall and 49 from the water side.

$Q_{section} = \alpha_a A_a (T_a - T_{wall,a})$ [kW] (47)

$$Q_{section} = k_{wall} A_{wall} (T_{wall,a} - T_{wall,w})/\delta \ [kW] \quad (48)$$

$$Q_{section} = \alpha_w A_w (T_{wall,w} - T_w) [kW] \quad (49)$$

$$Q_{section} = m_a c_{pa} (T_{a,in} - T_{a,out}) \ [kW] \quad (50)$$

$$Q_{section} = m_w c_{pw} (T_{w,in} - T_{w,out}) [kW] \quad (51)$$

$$Q_{section} = m_a (h_{a,in} - h_{a,out}) [kW] \quad (52)$$

α_a - *heat transfer coefficient from the refrigerant side [kW/m^2 K]*

A_a - *heat transfer area from the refrigerant side [m^2]*

α_w - *heat transfer coefficient from the water side [kW/m^2 K]*

A_w - *heat transfer area from the water side [m2]*

k_{wall} - *thermal conductivity of the wall [kW/m K]*

A_{wall} - *heat transfer area of the wall [m^2]*

T_a - *mean refrigerant temperature in the section [K]*

$T_{wall,r}$ - *mean wall temperature from the refrigerant side [K]*

$T_{wall,w}$ - *mean wall temperature from the water side [K]*

T_w - *mean water temperature in the section[K]*

Outlet temperatures and enthalpies have been obtained from Equations 50, 51, 52. In case of condensation, ammonia temperature is assumed as constant and Equation 50 is not taken into account. Vapour quality has been obtained from values of the enthalpy at given condensation pressure. Sample of the model is shown at Figure 31 (desuperheater) and Figure 32 (condenser) when the heat pump runs at the full capacity providing hot tap water heating and covering heating demand. Due to division of the heat exchanger only in 20 sections there are some rapid drops of the temperature which in real process are more smooth.

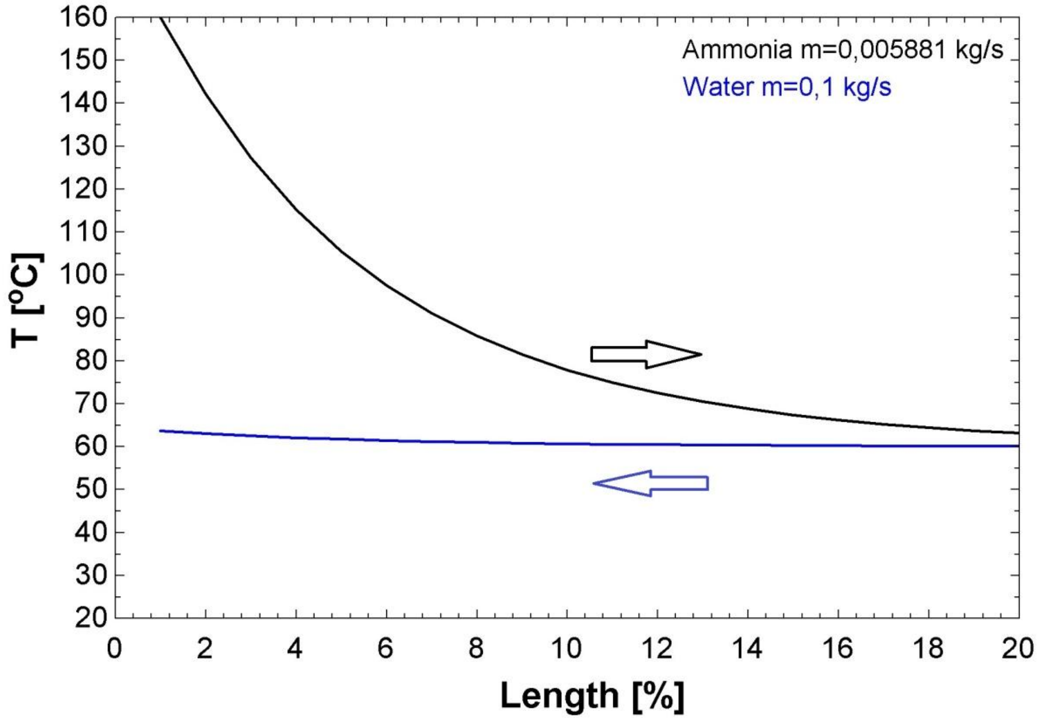


Figure 31 Change of temperatures in the desuperheater (EES)

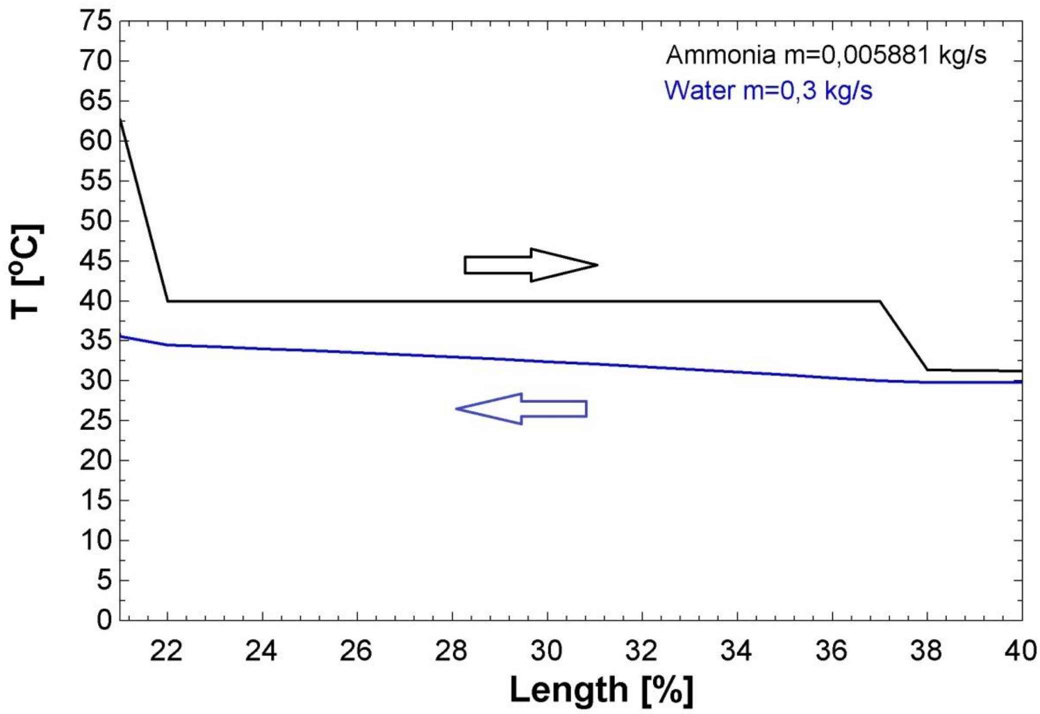


Figure 32 Change of temperatures in the condenser (EES)

Several calculations with different working regimes and different inlet water temperatures have been made. Obtained data of the water temperature change in each of the heat exchangers has been plotted and converted to functions. The functions show average temperature change in the heat exchanger in the unit of time for certain mass flow rate (volume of the heat exchanger is 0,9 l).

1st regime. Desuperheater (hot tap water heating) and condenser (heating system) working in the row simultaneously (Figures 31, 32). Assumption has been made that the tap water temperature is higher than the condensation temperature because of complicated calculations of the superheated gas condensation on cold surfaces.

$$T_{w\_out\_dsh}=T_{w\_in\_dsh}+0,453143t\ [^oC] \tag{53}$$

(m_w=0,1 kg/s; 40 $^oC < T_{w\_in\_dsh} <$ 75 ^oC)

$$T_{w\_out\_c}=T_{w\_in\_c}+1,79333t\ [^oC] \tag{54}$$

(m_w=0,3 kg/s; 15$^oC < T_{w_in_c} <$36 ^oC)

2nd regime. When hot tap water tank reaches 70^{o}C desuperheating and condensation are provided in one heat exchanger, which works only for the floor heating system.

$$T_{w\_out\_c}=T_{w\_in\_c}+2,26t\ [^oC] \tag{55}$$

(m_w=0,3 kg/s; 15$^oC < T_{w_in_c} <$36 ^oC)

3rd regime. The heat pump works only for hot tap water pre heating with 40% of the total capacity (decreased by frequency controller).

$$T_{w\_out\_c}=T_{w\_in\_c}+1,66667t\ [^oC] \tag{56}$$

(m_w=0,1 kg/s; 15 $^oC < T_{w\_in\_c} <$ 30 ^oC)

6.5 Hot tap water tank

The hot tap water tank is insulated vessel with volume of 200 l. Due to safety reasons there is a built-in coaxial heat exchanger to avoid refrigerant presence in the water in case of leakage. The tank is supplied with electrical heater as well. Water is circulating between 2 heat exchangers with mass flow rate of 0,1 kg/s.

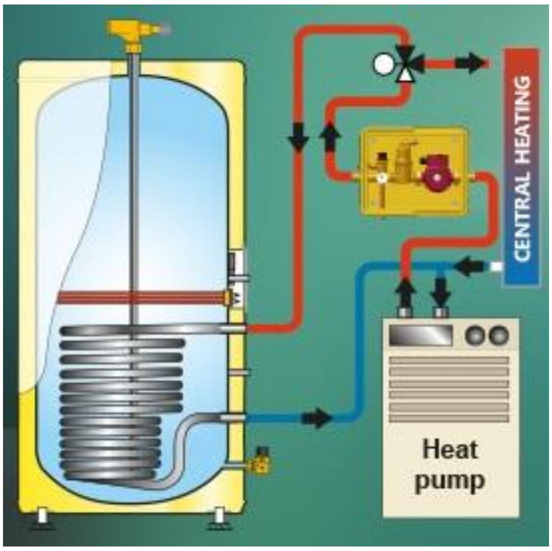


Figure 33 OSO Ecoline GEO 200l tank (OSO 2015)

The heat transfer coefficient from the circulating water side in the coaxial heat exchanger is calculated using the same equations as in the evaporator (Equations 14-15 for the pipe geometry and Equations 22-27 for the heat transfer coefficient) .

The coaxial heat exchanger has been assumed as a cylinder inside of the tank to calculate convective heat transfer coefficient inside of the vessel suggested by VDI Heat Atlas. (Kast, et al., 2010)

$$Nu = \left(0{,}60 + 0{,}387(Raf(\mathrm{Pr}))^{1/6}\right)^2 \; [\text{ - }] \tag{57}$$

$$f(Pr) = \left[1 + \left(\frac{0{,}559}{\mathrm{Pr}}\right)^{9/16}\right]^{-16/9} \; [\text{ - }] \tag{58}$$

$$\beta = \frac{1}{T_*} \; [K^{-1}] \tag{59}$$

$$Ra = \frac{\beta g \,\Delta T L^3}{v k_d} [\text{ - }] \tag{60}$$

$$\alpha_w = \frac{kNu}{L} \; [W/m^2 \, K] \tag{61}$$

β - thermal expansion coefficient [K^{-1}]

T^* - media temperature [K]

g - gravitational acceleration [m/s^2]

ΔT - temperature difference between the media and the wall [K]

L - effective length of the cylinder [m]

υ - kinematic viscosity [m^2/s]

k_d - thermal diffusivity [m^2/s]

k - thermal conductivity [W/m K]

The same principle of the heat transfer equilibrium as in previous calculation has been applied for further analysis (Equations 47-51). Equations 53 and 56 (depending on the heat pump working regime) have been introduced in the equilibrium to determine inlet water temperature to the coaxial heat exchanger depending on the outlet temperature (shows received heat from the desuperheater/condenser).

At the 1st heat pump working regime, when it works at full load for both hot tap water heating and space heating, water temperature in the tank rises from 40^{o}C to 70^{o}C in 50 minutes (Figure 34). If water temperature is lower than 40^{o}C condensation will appear in the desuperheater and it will decrease capacity of the condenser. Due to high condensation heat transfer coefficients of the ammonia, hot tap water will reach temperature of 40^{o}C very fast and it will not make significant affect on the whole system.

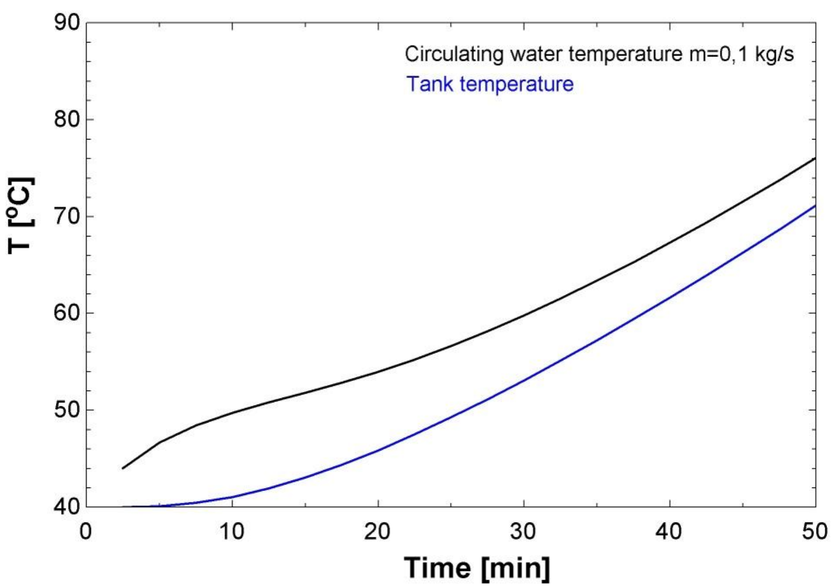


Figure 34 Temperature curves in the water heating system (EES)

3rd working regime, when the heat pump works only for the hot tap water preheating has been rejected due to limits of inlet water temperature to the condenser ($T_{w\_in\_c}$=30^{o}C) to provide condensation of the refrigerant. With temperature difference of 5 ^{o}C to provide good heat transfer hot tap water can be preheated only up to 25 ^{o}C. Even at low operating frequency of the heat pump (40% of the total capacity) heat transfer rate from the heat pump side is much more higher than the heat transfer rate in the tank. It leads to very rapid temperature rise in the loop between 2 heat exchangers which would provoke rising of condensation pressure and very frequent on/off operation of the compressor what is not acceptable. As a solution for this operation regime could be oversized pump which could provide higher mass flow of the water during this operating regime thereby decreasing the rapid temperature rise of the water inside the loop. Bigger pump means higher electricity consumption which leads to make a conclusion that only electrical heater during the periods without any space heating demand should be used. Hot tap water heating could be provided only as a by-product to the heat pump operation mode when it covers heating demand of the house.

6.6 Heating system storage tank

Usually the storage tank for heat pumps is designed to prevent very frequent on/off operation of the compressor due to variations in the heating load. In this case the main aspect for storage tank design is possibility to cover heating demand during the peak hours of high electricity costs. As shows Figure 7 in Chapter 3 the peak hours appears twice per day and peak period lasts for at least 2 hours. According to this period optimal volume of the tank has been found. Required temperature in the room is 23^{o}C therefore temperature of the circulating fluid in the system should be at least for 5 ^{o}C higher to provide good heat transfer. The temperature difference in the tank is assumed to be 8^{o}C because the heat pump is able to provide water up to 36^{o}C with condensation temperature of 40^{o}C. The volume of the tank has been found from Equations 62-64 which describe necessary tank volume at different heat loads to cover heating demand for 2 hours. Data for the heating demand was used from previous study done for this topic. (Bjørneklett, 2013)

$$E_{d2h} = Q_d t_{2h} \text{ [kWh]} \tag{62}$$

$$m_t = \frac{E_{d2h}}{c_{pw}\,\Delta T} \text{ [kg]} \tag{63}$$

$$V = \frac{m_t}{\rho_w} \text{ [m}^3\text{]} \tag{64}$$

E_{d2h} - necessary tank capacity to cover 2 h of heating demand (Q_d) [kWh]

m_t - water mass in the tank [kg]

V - volume of the tank [m^3]

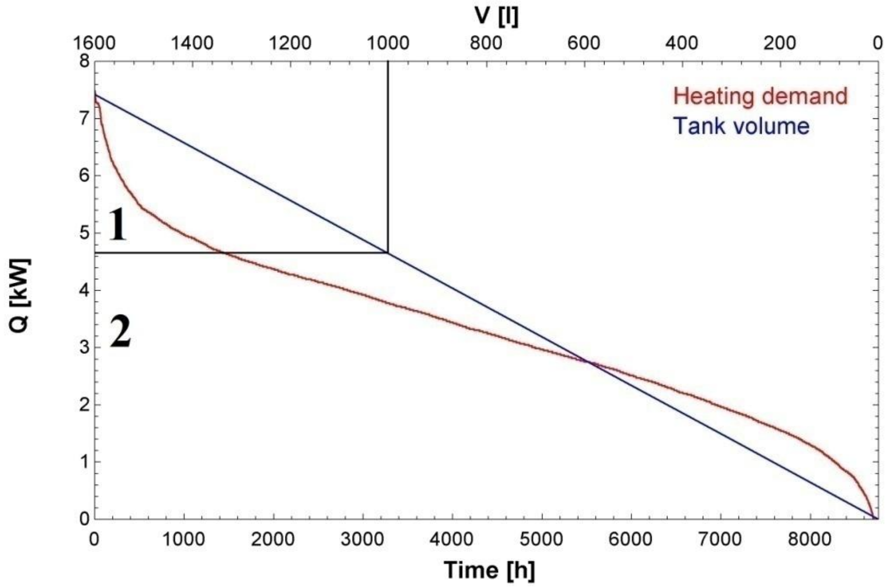


Figure 35 Optimal tank volume to cover 2 hours of heating demand ($\Delta T=8^{o}C$)(EES)

Figure 35 shows that volume of 1000l (with $\Delta T=8^{o}C$) is enough to cover 4,63 kW heating demand for 2 hours. It means that heat pump will not be working during the period with high electricity price and all heating demand will be covered by the storage tank. In the section 1 it will cover heating demand less than for 2 hours and in the section 2 it will be enough to cover heating demand even for longer time. Usually heat pump systems are designed to cover 60% of the maximal heating load which is approximately 90% percents of the total energy amount during the year. The same principle with the tank volume is applied here.

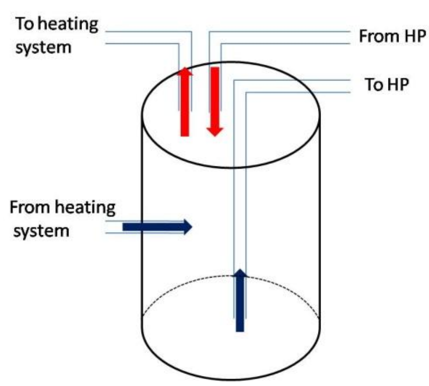


Figure 36 Principal scheme of the tank

In this solution there is no additional heat exchanger inside of the tank and water is circulating directly to the heat exchanger of the heat pump. Water intake should be provided from the bottom of the tank because there is the lowest temperature due to stratification. When water temperature reaches $36^{o}C$ at the bottom it means the tank reached its full capacity. Additional temperature sensor should be installed at the top of the tank to estimate when the tank is empty in terms of energy. Total capacity of the energy in the tank and the floor heating system is 11,5 kWh with $\Delta T=8^oC$.

6.7 Operational strategy

Heat transfer from the heat pump to the tank has been calculated taking into account different working regimes during variable heating demand. 7 patterns has been made for heating demand from 1 to 7 kW (described in the results). Price of electricity affected when the heat pump is running or when it is switched off. Assumption has been made that no mixing in the tank between hot and cold water occurs. Heat losses of the tanks has been neglected as well. Equation 62 describes transferred amount of energy to the tank.

$$E_t = E_{HP} - E_D \ [kJ] \tag{65}$$

E_{HP} - *energy from the heat pump in the unit of time [kJ]*

E_D - *energy for the heating demand in the unit of time [kJ]*

Heat transferred from the heat pump varies depending on the working regime. At the 1st regime, when the heat pump covers heating demand and heats up hot tap water tank is introduced Equation 54 which describes change of the water temperature in the heat exchanger. As the heat pump has full load capacity of 8,4 kW but maximal heating demand is 7,43 kW, surplus energy accumulates in the storage tank. Heating demand affects the time of heating up the tank.

$$E_{HP} = m_w * c_{pw} \left(T_{w_{out_c}} - T_{w_{in_c}}\right) t \ [kJ] \tag{66}$$

$$E_D = Q_D t \ [kJ] \tag{67}$$

Q_d - *heating demand [kJ/s]*

t - time [s]

When the temperature in the hot tap water tank reaches 70^oC, Equation 54 is substituted with Equation 55 which describes temperature change of the water in the heat exchanger when the heat pump works only for the space heating system.

When the tank reaches its full capacity (temperature at the bottom is 36^oC) heat pump either is switched off (Equation 68) or starts to work at lower frequency to cover heating demand only (Equation 69), depends on electricity price in that moment. When the heat pump works at lower frequency it is assumed that the heat transfer rate decreases proportional to decreased frequency but isentropic efficiency of the compressor does not change.

$$E_{HP} = 0 \text{ [kJ]} \quad (68)$$

$$E_{HP} = E_D \text{ [kJ]} \quad (69)$$

When the temperature at the top is < 28^{o}C (E_t <=0) , the heat pump starts to work at full load or at decreased frequency, depends on heating demand and electricity price. During the working days, when nobody is at home it is possible to allow drop of the house temperature until 16^{o}C (T_{drop}). Temperature drop is calculated from Equation 70 and 71 according to the heat loses. Equation 71 shows energy capacity of the air with temperature difference between room and allowed drop temperature. Divided with heating demand has been calculated time which house can stay without any heating supply. In calculations has been used average value of the heating demand because heat transfer through the walls decreases with decreased temperature difference between the room and the ambient air. Figure 37 shows the time which takes to cool down house at certain ambient air temperature from 23 ^{o}C - 16 ^{o}C.

$$Q_d = U_h A_h (T_r - T_{amb}) \text{ [kW]} \quad (70)$$

$$E_d = m_{air}\, c_{pa} (T_r - T_{drop}) \text{ [kWh]} \quad (71)$$

$$t = \frac{E_d}{Q_d} \text{ [h]} \quad (72)$$

U_h- overall heat transfer coefficient for the house [kW/m^2 K]

A_h - heat transfer area of the house [m^2]

T_r - room temperature [K]

T_{amb} - ambient air temperature [K]

m_{air} - air mass in the house [kg]

c_{pa} - air specific heat capacity [kJ/kg K]

T_{drop} - room temperature drop [K]

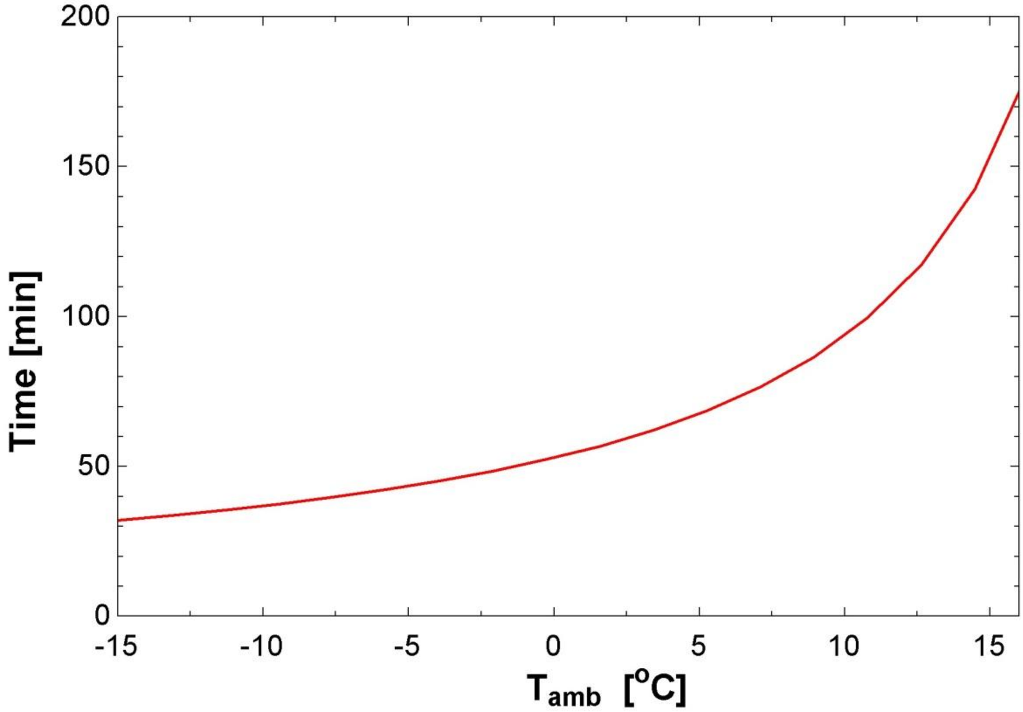


Figure 37 Time which takes to cool down house from 23 ^oC - 16 ^oC at different ambient air temperatures (EES)

Regime, when the heat pump works for both hot tap water heating and space heating is applied two times per day, before the morning and the evening peaks.

7. RESULTS

According to the main task, EES tool with optimal working regimes for energy consumption during the year has been developed. This model is adjusted to different tariffs of the electricity price and different heating loads (from 1 to 7 kW). In the Figures 28 and 39 is described pattern for 6kW heating load. The other patterns are available in Appendix 1.

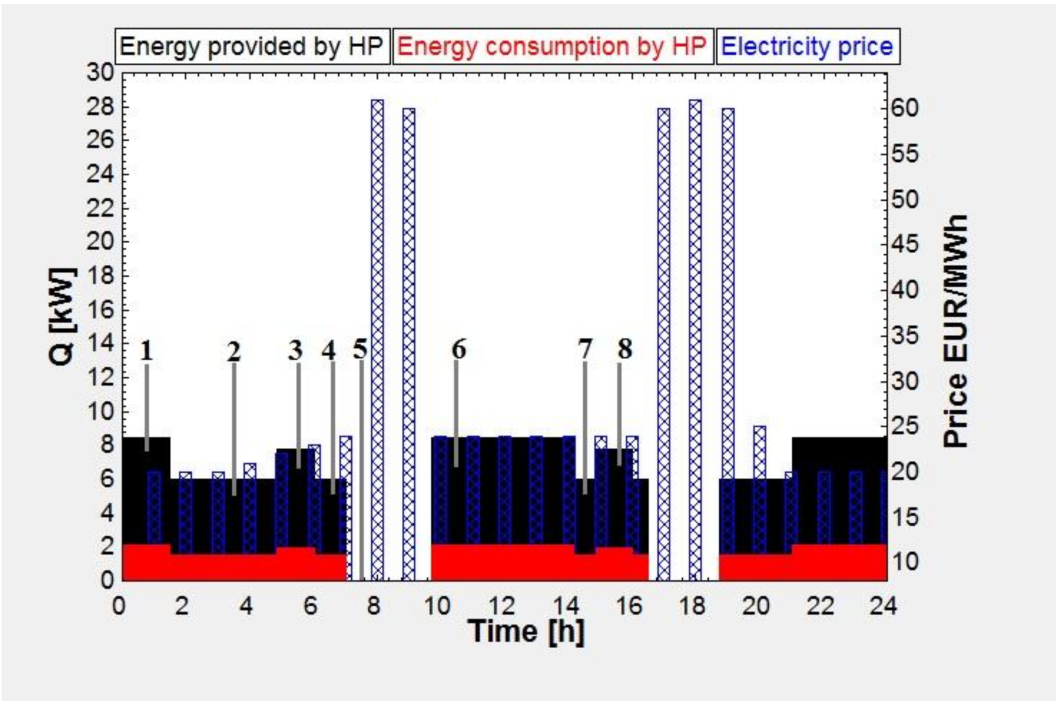


Figure 38 Energy consumption during the day with heating demand of 6kW (EES)

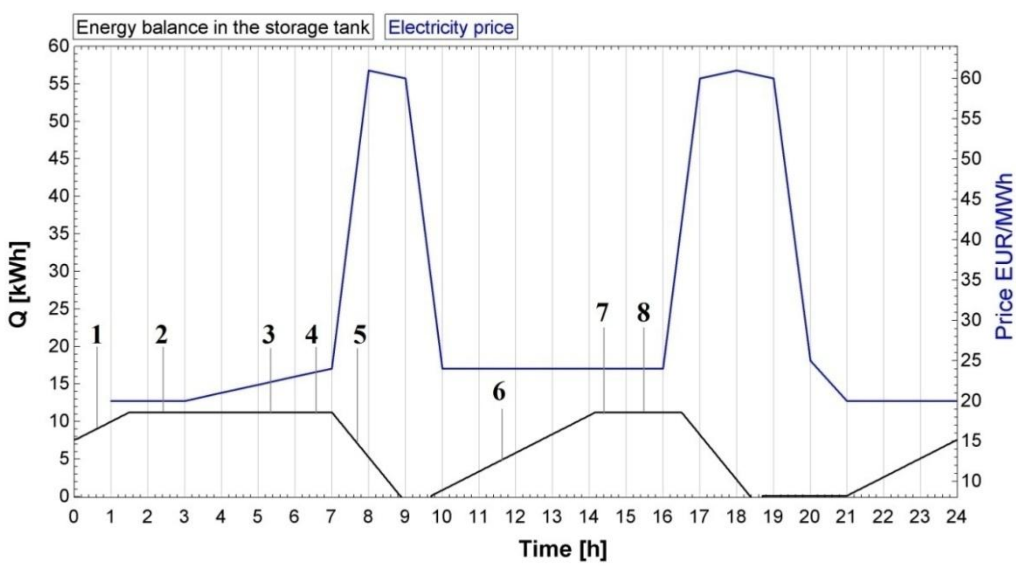


Figure 39 Balance of the storage tank during the day with heating demand of 6kW (EES)

In **Section 1** The heat pump works at the full load of 8,4 kW. It covers heating demand and surplus energy is accumulating inside of the tank. When the tank reaches its full capacity (temperature at the bottom is 36^oC) the heat pump starts to work at decreased frequency to cover heating demand only which is described in **Section 2**. In the **Section 3** the heat pump works to cover heating demand and to heat up hot tap water simultaneously. When the hot tap water tank reaches 70^oC, the heat pump proceeds to work at decreased frequency to cover heating demand only (**Section 4**). Before the morning peaks of high electricity price (7AM) the heat pump is turned off and heating demand is covered from the storage tank, described in the **Section 5**. When the storage tank exceeds its capacity (temperature at the top is <28^oC) the heat pump is not turned on until the room temperature drops to 16^oC. This gap without any heating supply can be achieved only during the working days when no occupants are at home. In the **Section 6** the heat pump starts to run at the full load to cover heating demand and heat up the storage tank before the evening peaks. When the tank reaches its full capacity, the heat pump works at decreased frequency to cover heating demand only (**Section 7**). Before the evening peaks it is assumed that there is one more hot tap water heating demand therefore in the **Section 8** the heat pump works at the same regime as in the Section 3. The heat pump is turned off before the evening peaks until the storage tank exceeds its capacity. Afterwards it starts to run at decreased frequency due to high electricity price. When the electricity price is low, the heat pump starts to work at full load to heat up the storage tank. This regulation model can be achieved by linking time and temperature controls in one.

The price of consumed energy has been calculated for 7 patterns. Results of each pattern with different price tariffs are described in the Table 5.

Table 5 Price of the consumed energy during the different tariffs and heating demands (EES)

Peak price 30 EUR/MWh Regular price 20 EUR/MWh				Savings	
Heating demand (kW)	Only electrical heating (EUR/day)	Only heat pump (EUR/day)	Heat pump with storage tank (EUR/day)	Heat pump compared with electrical heating (COP=4)	Heat pump with/without storage tank
7	4,22	1,06	0,96	75%	8,83%
6	3,51	0,88	0,81		7,24%
5	3,03	0,76	0,69		9,03%
4	2,47	0,62	0,54		11,92%
3	1,92	0,48	0,40		16,76%
2	1,34	0,33	0,27		18,77%
1	0,75	0,15	0,11		22,76%
Peak price 40 EUR/MWh Regular price 20 EUR/MWh					
7	4,57	1,14	0,98	75%	13,85%
6	3,79	0,95	0,82		13,62%
5	3,27	0,82	0,68		16,26%
4	2,66	0,66	0,54		18,83%
3	2,06	0,52	0,40		22,34%
2	1,43	0,36	0,27		23,73%
1	0,80	0,16	0,11		27,17%
Peak price 60 EUR/MWh Regular price 20 EUR/MWh					
7	5,31	1,33	1,05	75%	21,19%
6	4,39	1,10	0,84		23,25%
5	3,77	0,94	0,69		27,29%
4	3,06	0,76	0,54		29,37%
3	2,36	0,59	0,40		32,19%
2	1,63	0,41	0,27		32,93%
1	0,90	0,17	0,11		34,46%

Compared to the electrical heating, the heat pump consumes 4 times less energy with assumption that COP changes are insignificant during a year. Installation of the storage tank does not reduce amount of consumed energy but it can significantly reduce the total price of consumed energy in case of variable tariffs.

Table 6 Annual savings with the storage tank (EES)

	Peak price EUR/MWh		
	30	40	60
	Costs EUR/year		
Only electrical heating	751,61	808,64	928,94
Only heat pump	187,90	202,16	232,23
Heat pump with storage tank	161,57	161,45	162,36
Annual savings for heat pump with/without storage tank	14,01%	20,14%	30,09%

The Table 6 shows annual costs for different heating applications. The equal energy price for the heat pump with the storage tank shows that the tank with volume of 1000l is optimal because it is not affected by 2-3 hour peaks of the electricity price. Savings are proportional to the price of the electricity at the peak hours.

Another benefit of the storage tank is possibility to heat up the hot tap water tank during low heating demands, therefore avoiding pure electrical heating. Figures 40 and 41 shows that the heat pump provides surplus energy even at lowest possible frequency (40% of the maximum capacity) and the hot tap water tank is not able to accumulate it (described in Chapter 6.5), therefore the storage tank is used as a heat sink. Without storage tank it would be very frequent on/off operation of the compressor which is not acceptable.

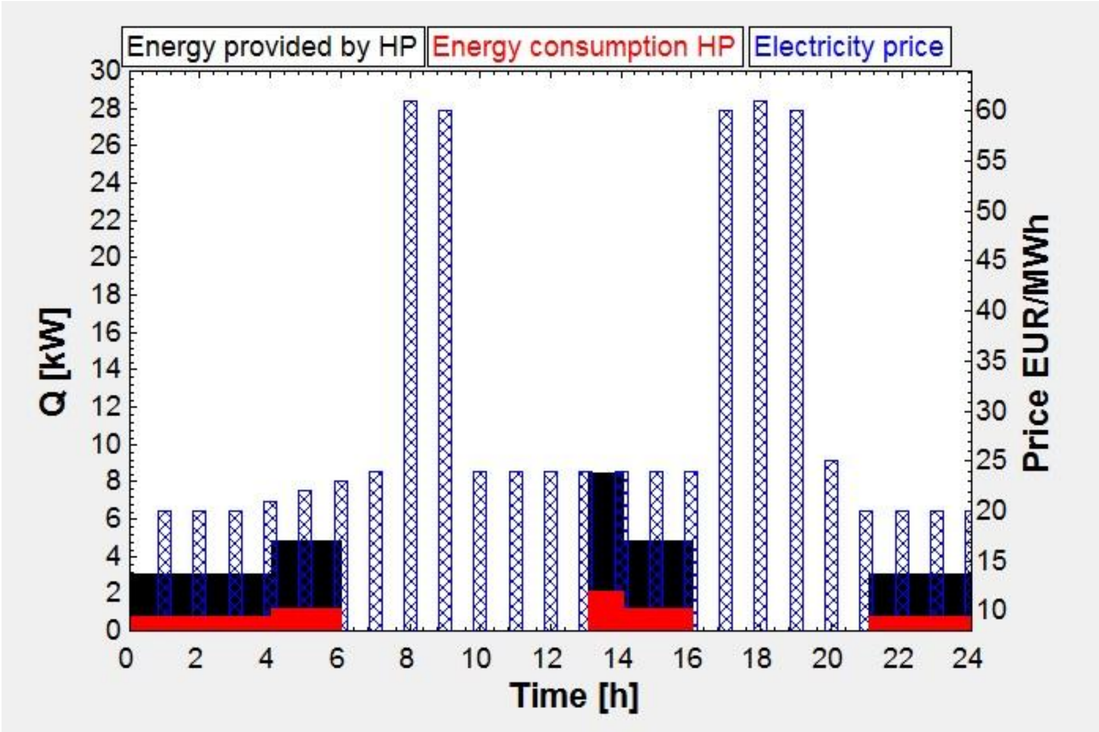


Figure 40 Energy consumption during the day with heating demand of 2kW (EES)

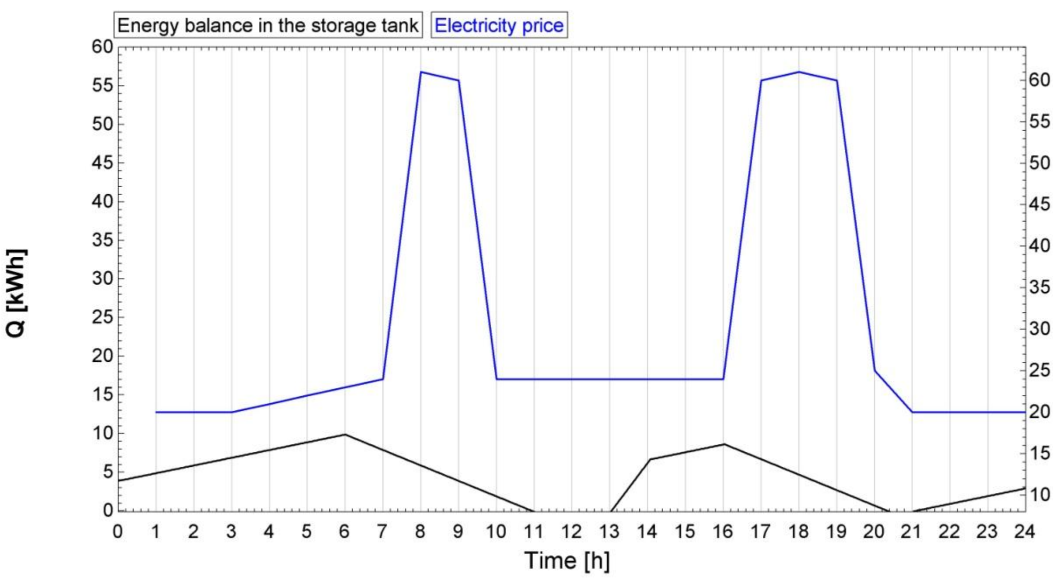


Figure 41 Balance of the storage tank during the day with heating demand of 2kW (EES)

8. ENGINEERING PROJECT

Engineering project consists of 3 main parts:

- specification list of the components;
- principal drawing for the ammonia heat pump;
- example of a house with ammonia heat pump application.

The aim of the engineering project is to design small capacity ammonia heat pump according to the collected information. Analysis of previous experiments in this field allows to highlight important design features which would provide efficient and long lasting operation of the heat pump.

Components

Due to corrosivity of the ammonia, only stainless steel components have to be used. The heat pump market still does not have enough components for small capacities suitable with ammonia. It leads to use some parts from industrial sector, which makes the system oversized, thereby necessary charge of the refrigerant is bigger.

Refrigerant flow

Many plants built before had a problem with oil return and refrigerant distribution in the evaporator. As an effective solution for this problem is to achieve refrigerant flow from the top of the desuperheater downwards. It means that the desuperheater has to be at the highest point of the system and the compressor at the lowest point. Special type of the evaporator has to be used as well to provide good distribution of the refrigerant and to avoid oil trapping in the evaporator. Oil, miscible with ammonia has to be used, for example, MOBIL SHC 226.

Pipe dimensions

In refrigerating systems there are 3 types of pipes: suction line, discharge line and liquid line. The main aspect for design is velocity of the refrigerant which is related to oil return to the compressor. Typical vapour velocities used in suction and discharge lines are in the order from 5-10 m/s. Typical velocities in liquid lines are from 0,7 - 1,5 m/s. (Granryd, et al., 2005)

Calculation of velocities in the lines has been done to evaluate necessary diameter of pipes. Following equation has been used:

$$v = \frac{m_a}{A\rho} \text{ [m/s]} \tag{73}$$

m_a- mass flow rate of the refrigerant (kg/s)

A - cross sectional area (m^2)

ρ - density of the refrigerant in the section (kg/m^3)

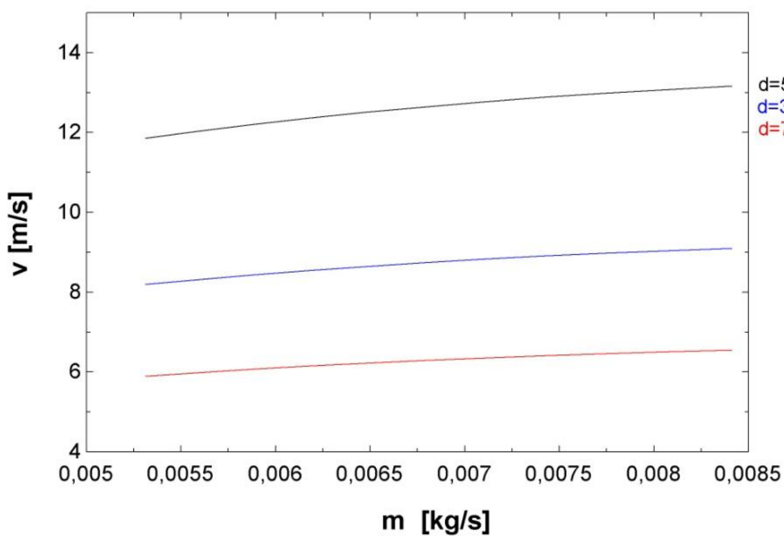


Figure 42 Velocities at different mass flow rates in suction line (EES)

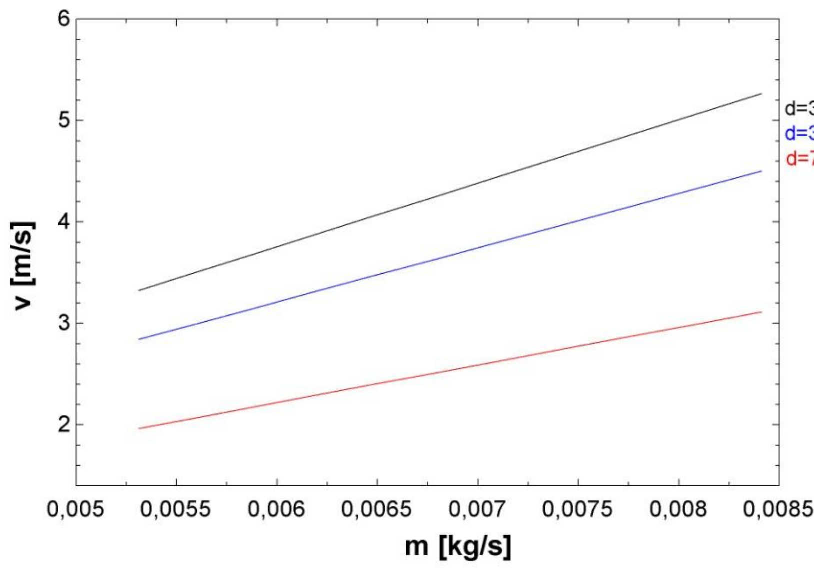


Figure 43 Velocities at different mass flow rates in discharge line (EES)

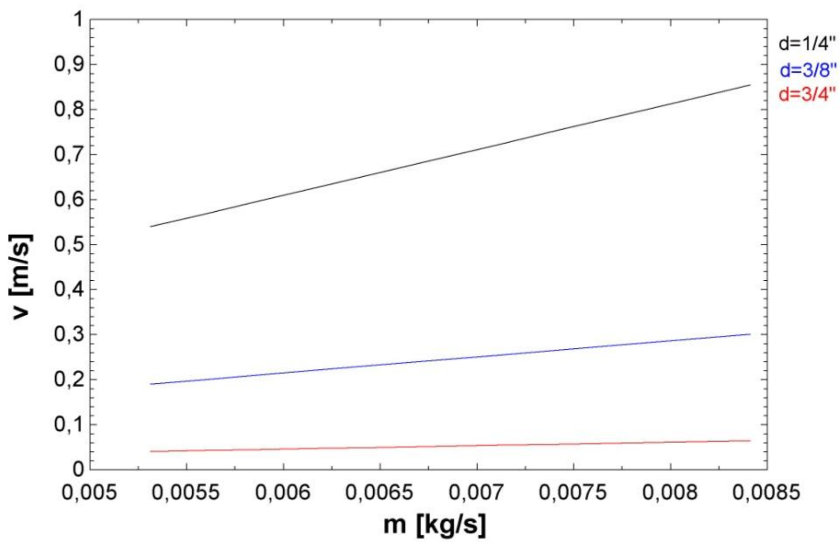
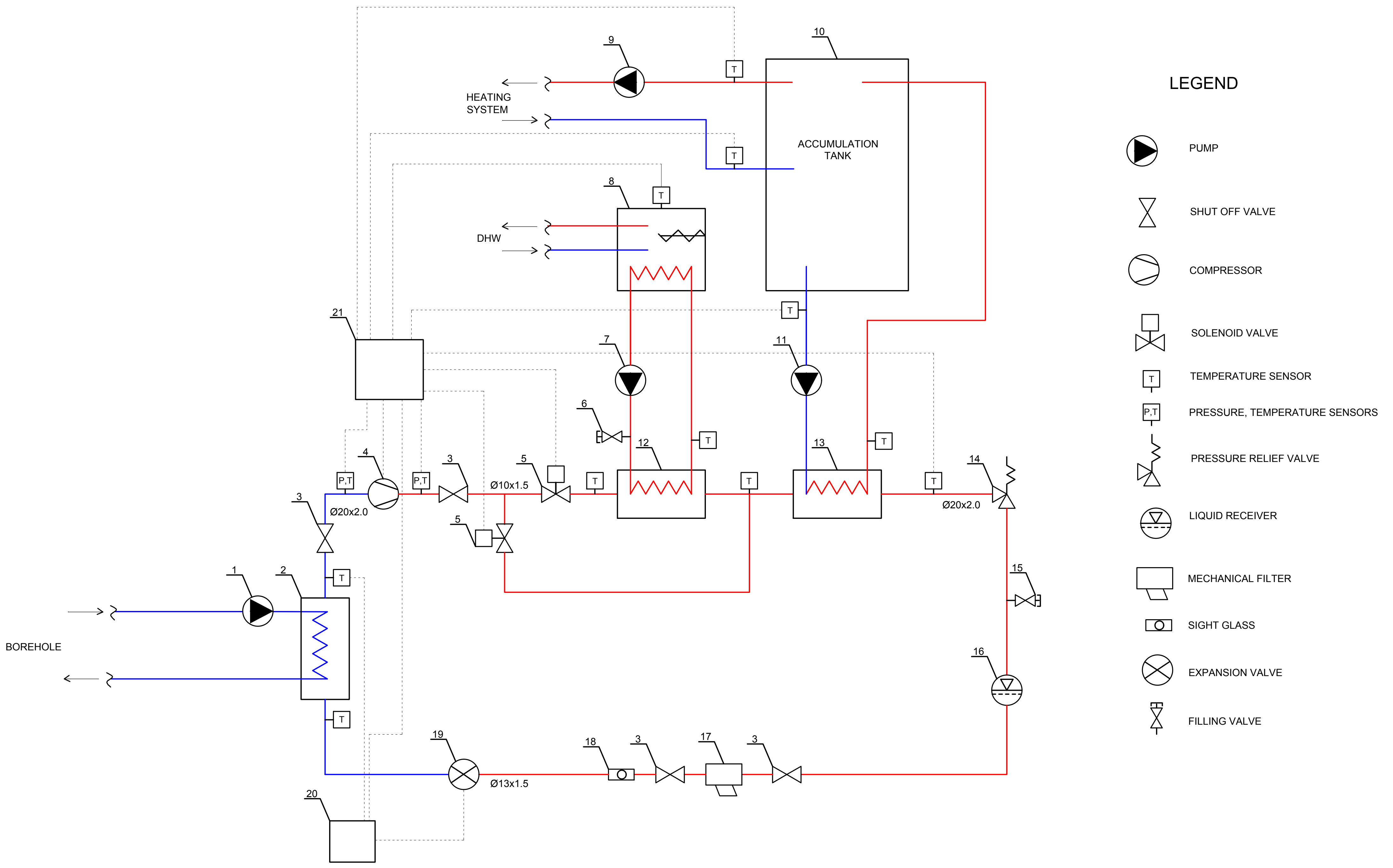


Figure 44 Velocities at different mass flow rates in liquid line

Figures 42 shows that minimal velocity requirement in the suction line is achieved with pipe diameter of 3/4". To achieve required velocity in discharge line (Figure 43) pipe with diameter of 3/8" has to be used. For liquid line (Figure 44) required pipe diameter is 1/4". Due to relatively large components on the liquid line (basically 3/4"), it is not useful to install pipe with smaller diameter because it requires big amount of transitions which can cause unnecessary pressure drops. As a solution is to install liquid receiver which would provide smooth operation even with small velocities. Moreover, the most important parameter is velocity in the suction line, because it is related to problems with oil return and it is fulfilled. According to a volume of the system, estimated refrigerant charge is approximately 5 kg. To obtain more precise value of the refrigerant charge it is necessary to make practical measurements of the heat pump performance. Due to toxicity and flammability of ammonia, the heat pump has to be installed outside of the house or in special facility.

Calculations of pressure drop in the pipes has been renounced since the system is very compact and total length of the pipes is much more smaller than the total length of the evaporator.



				8,4 kW ammonia heat pump for space and hot tap water heating	Stage	Page	Pages
					Scale	N/A	
	Name,surname	Signature	Date	Principal scheme of ammonia heat pump application	RBCS01		
Drawn	A. Lalovs						
Checked	A. Borodiņecs						

Table 7 List of components

List of components					
Number	**Name**	**Model**	**Units**	**Amount**	**Description**
1	Pump for the ground loop	Grundfos MAGNA 25-100	pc.	1	m=0,3 l/s, HX-35
2	Evaporator	SEAB GmbH Kleinostheim	pc.	1	Custom design
3	Shut off valves (3/4")	Danfoss 148B5380	pc.	4	
4	Compressor (3/4")	Frigopol 10-DLZC-2.2	pc.	1	
5	Solenoid valves with coils (3/8")	Danfoss 032F3080/018F6881	set	2	
6	Filling valve (1/4")	Continental CNA1275	pc.	1	
7	Pump for the desuperheater	Grundfos ALPHA2 25-40	pc.	1	m=0,1 l/s
8	Domestic water tank	OSO Ecoline GEO 200l		1	
9	Load pump	-		1	
10	Storage tank for the heating system	-		1	Insulated, $V=1m^3$
11	Pump for the condenser	Grundfos ALPHA2 25-40	pc.	1	m=0,3 l/s
12	Desuperheater	Alfa Nova 27-20H	pc.	1	
13	Pressure switch	Danfoss 017-500766	pc.	2	
13	Condenser	Alfa Nova 27-20H	pc.	1	
14	Safety relief valve	Danfoss SFV 20 T 225	pc.	1	
15	Filling valve	CNA1275	pc.	1	
16	Liquid receiver	Custom	pc.	1	V = 5 l
17	Mechanical filter 0,1mm (3/4")	Danfoss FIA20D/148H3122	set	1	
18	Sight glass (3/4")	Danfoss DN 20	pc.	1	
19	Expansion valve	Danfoss AKVA-10-2/BG220DS	set	1	
20	Controller for the expansion valve	Danfoss EKC 315 A	pc.	1	
21	Main controller	-			

ROOM LEGEND						Legend	
No	NAME	m²	No	NAME	m²		
1.01	LOBBY	22.4					Monolith reinf. concrete wall
1.02	HALL	23.6					Brick wall
1.03	BATHROOM	24.7					Wood decoration
1.04	TECH. ROOM	1.6					Hydro insulation
1.05	REST ROOM	27.6					Thermal insulation
1.06	LIVING ROOM	71.1					Terrace boards
1.07	STORAGE	9.3					
1.08	CABINET	25.6		TOTAL	213.2		

LEGEND

5. - NUMBER OF CONTOUR
9,7m² - AREA
400W - HEATING DEMAND
72l/h - FLOW
72m - LENGTH OF PIPE
200mm - DISTANCE BETWEEN PIPES

- FLOOR HEATING, SUPPLY
- FLOOR HEATING, RETURN
- HEATING, SUPPLY
- HEATING, RETURN
- FLOOR SYSTEM CONTOUR

				8,4 kW ammonia heat pump for space and hot tap water heating	Stage	Page	Pages
					Scale	1:100	
	Name,surname	Signature	Date				
Drawn	A. Lalovs			Ammonia heat pump in one family house	RBCS01		
Checked	A. Borodiņecs						

9. CONCLUSION AND SUGGESTIONS

Calculations and collected information in this study proves that it is possible to use ammonia as a refrigerant not only in large capacity systems but also for household applications. Nowadays, the heat pump market offers sufficient variety of components to build small capacity ammonia heat pump, however some of the components are oversized, what leads to relatively big refrigerant charge and non-standard design. Due to specific design of the ammonia systems (all components have to be from stainless steel only) it will be more expensive compared to the R-410A counterpart, therefore an effort from government should be applied to introduce those systems in the market.

The main benefits of the ammonia heat pump system are:

- ecology (GWP < 0,1; ODP = 0);
- possibility to heat up domestic water to high temperatures;
- better thermo-physical properties in comparison with R-410A.

This system has following disadvantages:

- due to toxicity of the ammonia, the heat pump has to be installed outside of the house or in special facility;
- high discharge gas temperature makes limitations for condensation temperature, which does not allow to use this system in high-temperature heating applications.

Calculated COP = 4 (T_{evap} = -3 ^oC , T_{cond} = 40 ^oC) provides energy savings up to 75% compared to the electrical heaters. In case of variable price of the electricity it is possible to achieve more savings with installation of the storage tank. Optimal volume of the tank for a single-family house is 1000l to cover 2-3 hour peaks of the high electricity price. EES tool with optimal working regimes shows, that even with large storage tank it is necessary to use frequency controller for the compressor to achieve precise operation of the heat pump.

For further work the designed small ammonia heat pump has to be built with following evaluation of the performance. Additional studies of control system have to be done in order to combine control possibility by both time and temperature parameters.

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APPENDICES

APPENDIX 1 Operational strategies with different heating demands

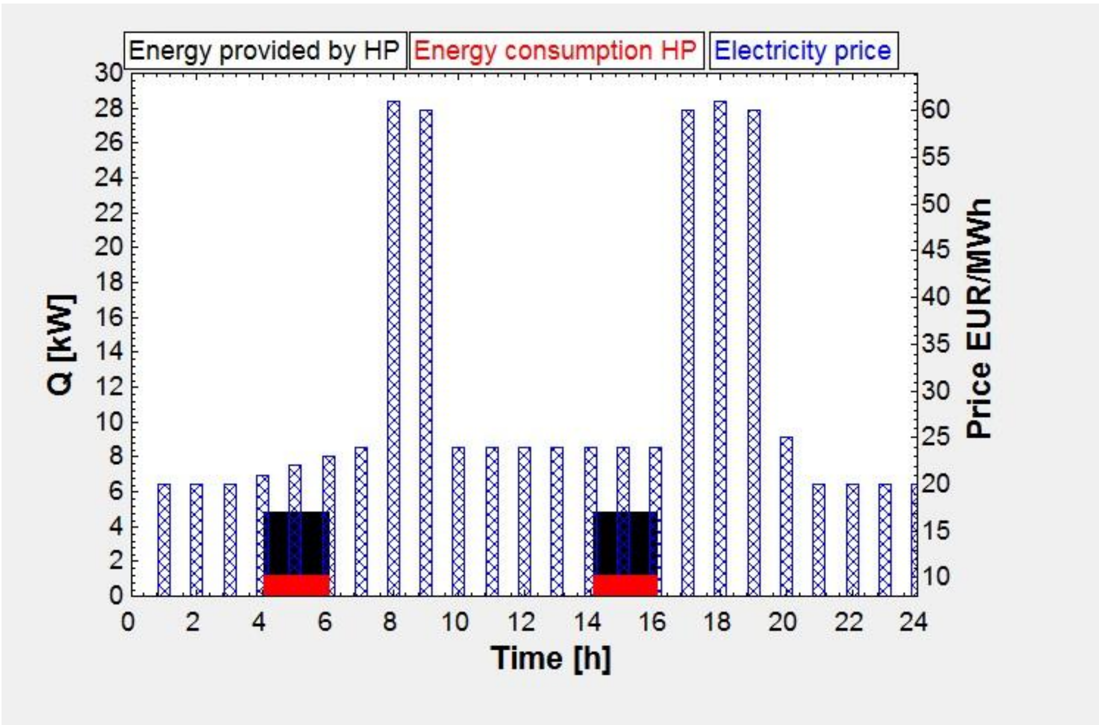
APPENDIX 2 Heat transfer coefficients

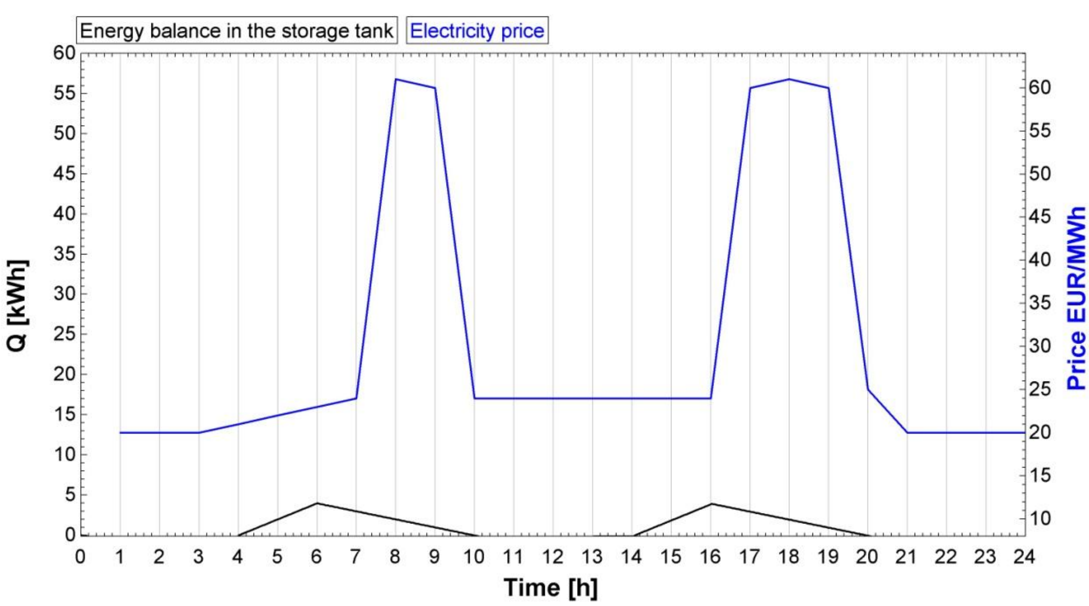
APPENDIX 3 Flow diagram of calculations

APPENDIX 1

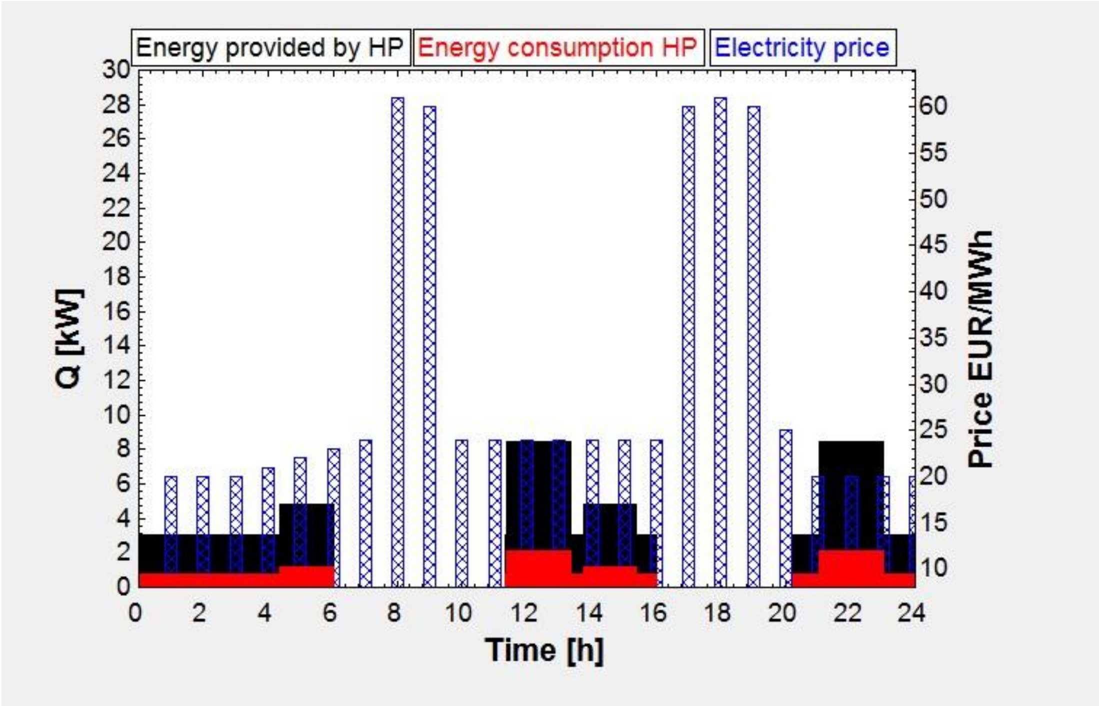
Operational strategies with different heating demands

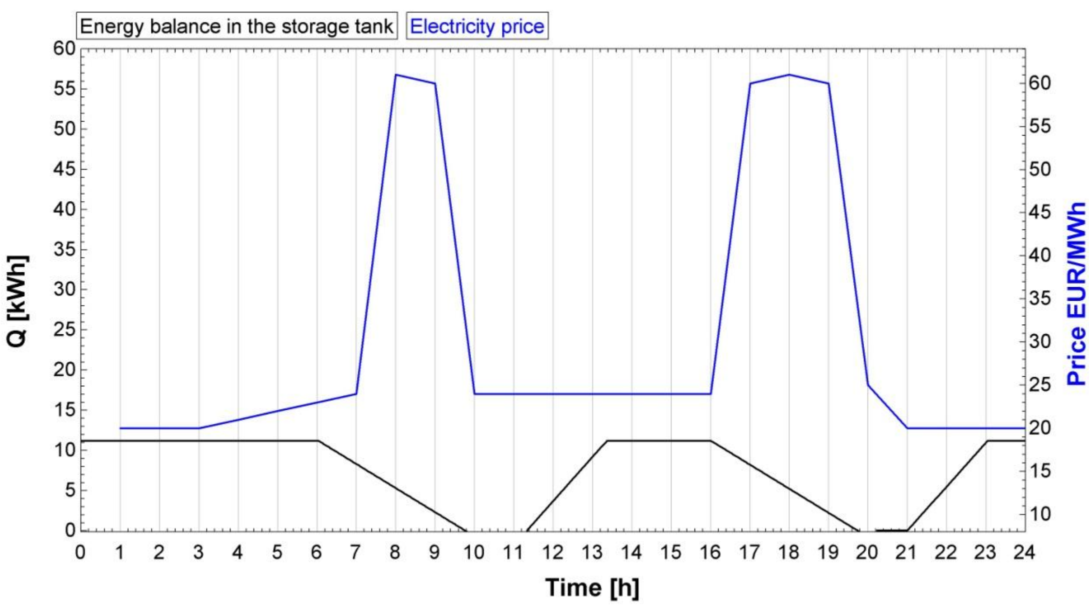
Heating demand 1kW



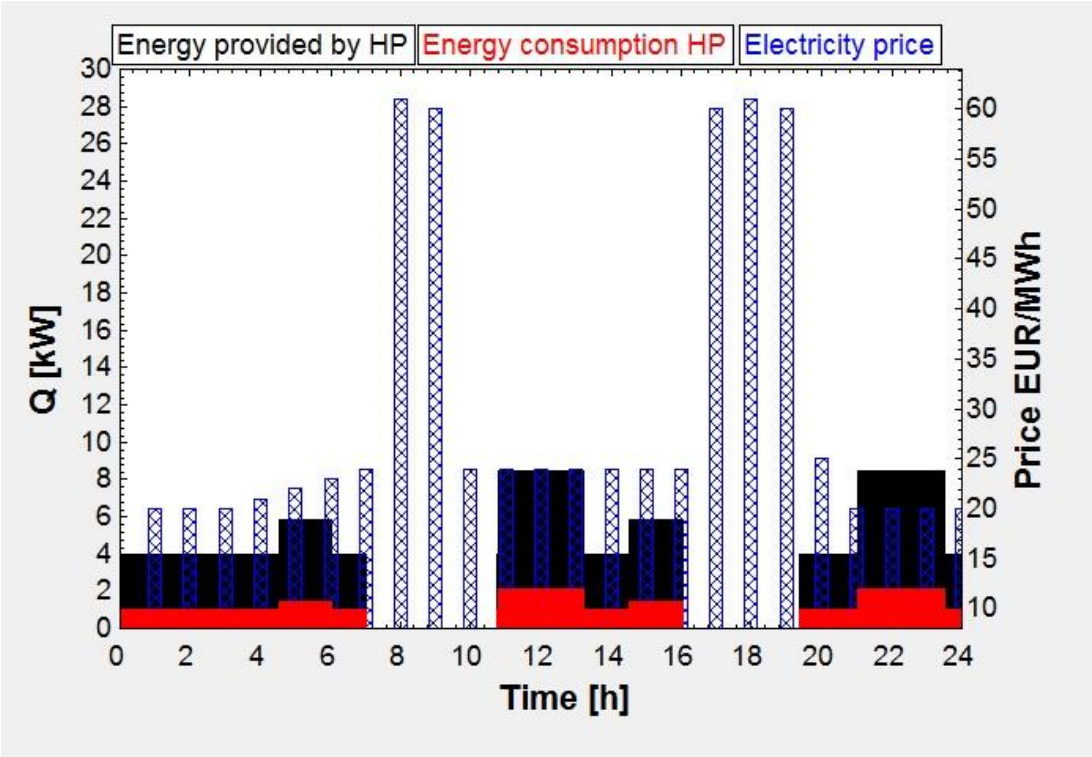


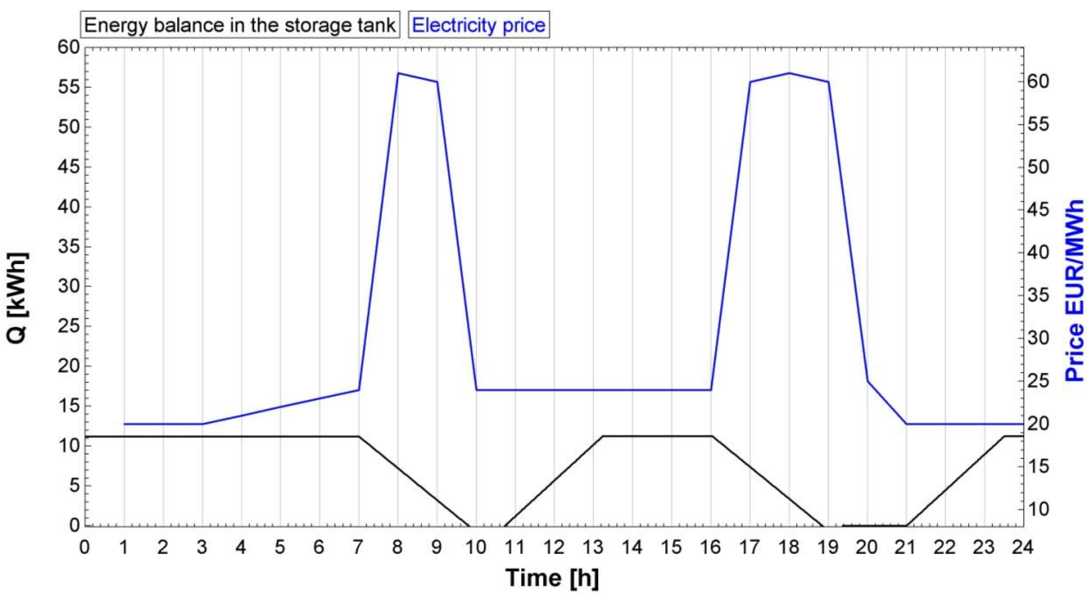
Heating demand 3kW



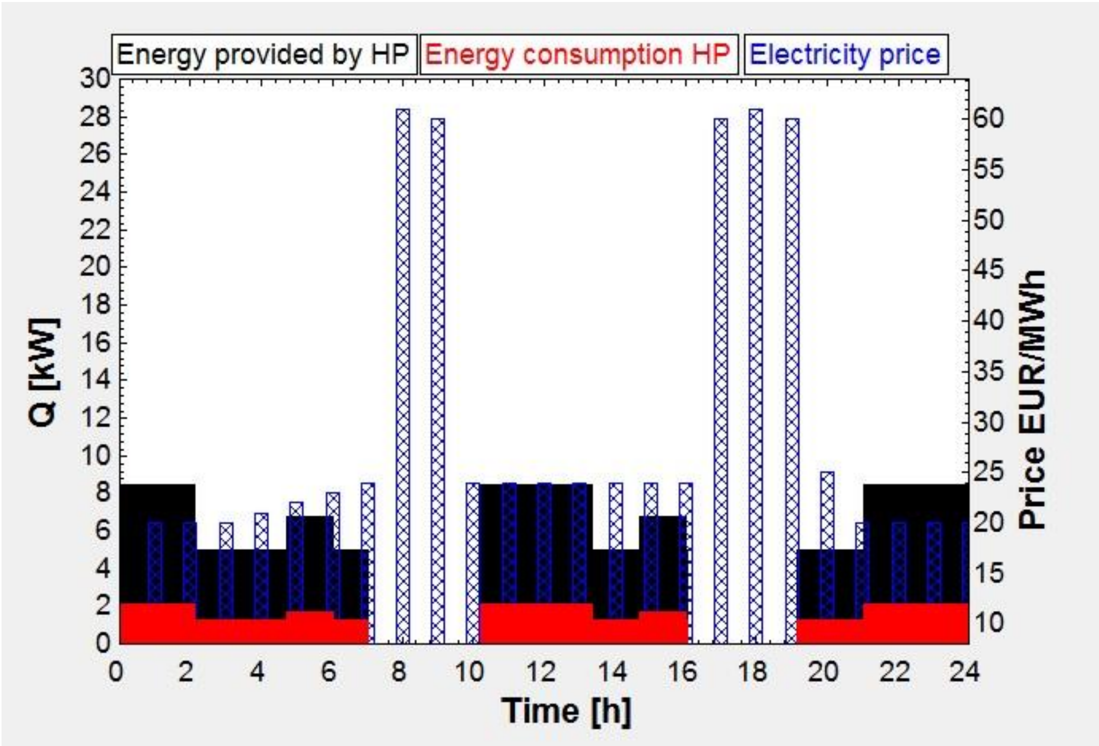


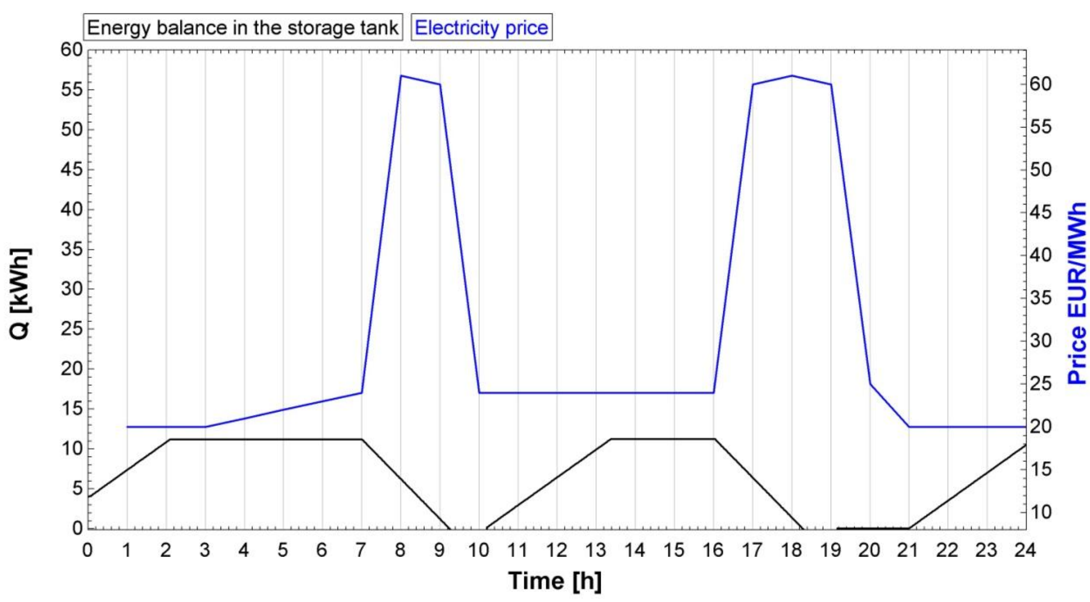
Heating demand 4 kW



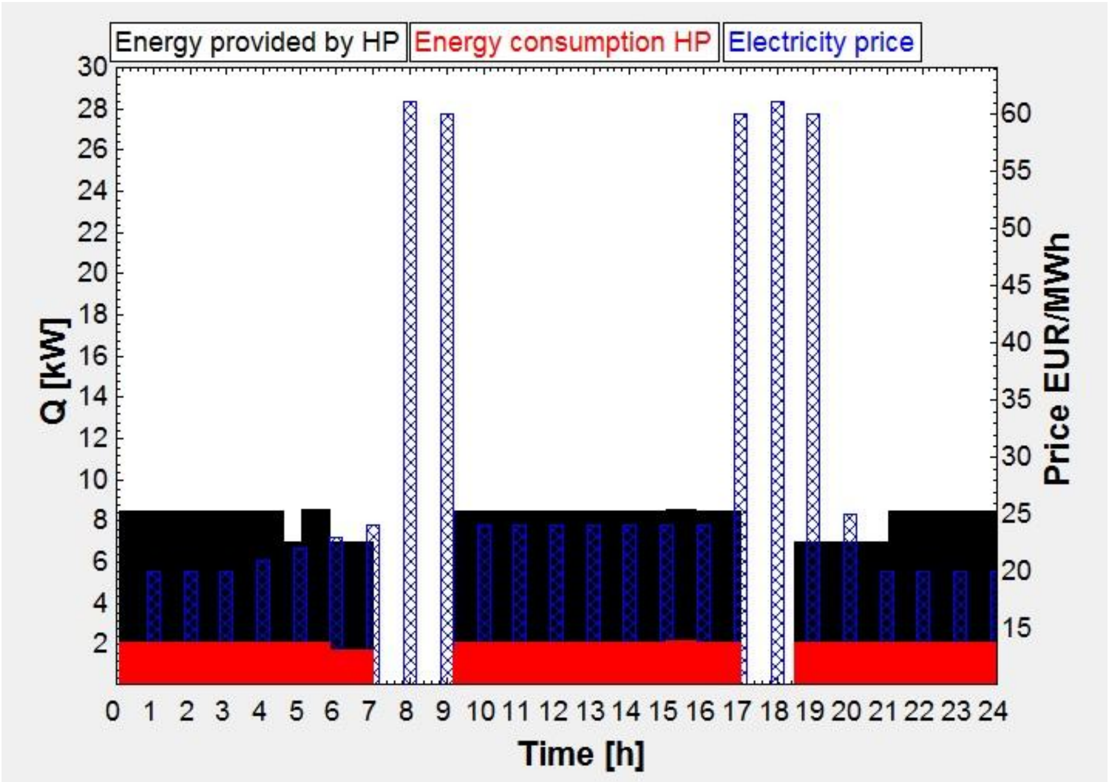


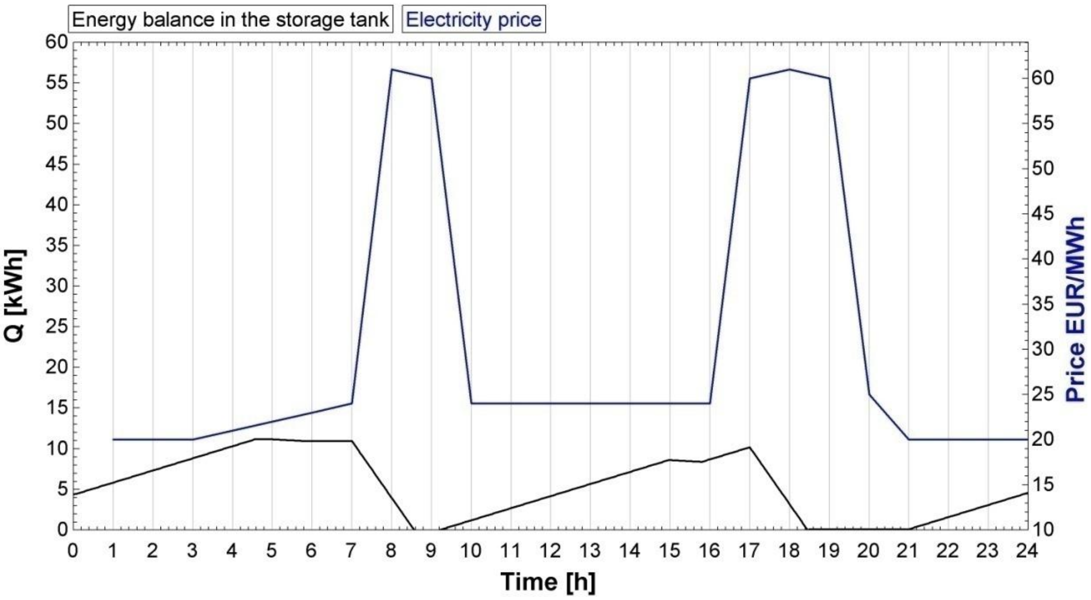
Heating demand 5 kW





Heating demand 7kW





APPENDIX 2

Heat transfer coefficients (W/m^2 K)

Evaporator	Heat transfer coefficient (W/m^2 K)
HX-35 (m=0,3 kg/s)	1098
Ammonia evaporation (m=0,05889 kg/s)	>> 1098 *(heat transfer calculated from HX-35 side)*
Ammonia superheated gas (m=0,05889 kg/s)	90

Desuperheater	Heat transfer coefficient (W/m^2 K)
Water (m=0,1 kg/s)	3614
Ammonia superheated gas (m=0,05889 kg/s)	130

Condenser	Heat transfer coefficient (W/m^2 K)
Water (m=0,3 kg/s)	5367
Ammonia condensation (m=0,05889 kg/s)	4337
Ammonia liquid (m=0,05889 kg/s)	1091

Hot tap water tank	Heat transfer coefficient (W/m^2 K)
Water (m=0,1 kg/s)	2875
Water (tank side)	105,5

APPENDIX 3

Flow diagram of calculations