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Thermodynamic modeling of absorption heat pumps

Master's Thesis within the Sustainable Energy Systems programme

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CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden, 2010

MASTER’S THESIS

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Cover:
[One of the Sanyo absorption heat pumps at Renova waste incineration plant [5]]

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Abstract

In order to recover as much as possible of the energy released in the waste incineration process at Renova, six lithium bromide absorption heat pumps are utilized to raise the temperature of a recovered low temperature stream.

The recovered heat utilized in the absorption heat pumps comes from the flue gas cleaning process, and is after the temperature has been raised in the heat pumps, delivered to the district heating network.

Four of the six heat pumps is of an older brand called Sanyo and has nominal effect of 4MW each. The two more modern heat pumps have a nominal effect of 6MW and are of brand called Entropie.

As a part of an energy optimization project initiated at Renova, together with a consultant company called “Energy project E&S”, the entire plant, including the absorption heat pumps, have been represented in an Excel based model (here referred to as Model-Renova). This model started with simple equations describing a small part of the energy recovery process and the choice to use Excel as a base for the modeling, was explained by the simplicity of Excel and the fact that Excel is a common and well known program,

The aim of this model is to find the optimal process solutions that would result in a more optimal use of the energy released in the combustion.

The parts of the Model-Renova that represents the absorption heat pumps are based on empirical correlations formulated from collected process data and represent the six heat pumps as two units. The Model-Renova has an unknown range of validity, accuracy and a limited possibility to model the individual heat pumps.

The accuracy of this heat pump model was evaluated by comparing the result of this model with real process data. The old model has a fairly god accuracy in predicting the district heating temperature, but give slightly over or under predicted estimations of the heat recovered by the heat pumps and the coefficient of performance. However a possibility to improve the model by changes in the governing equations could improve the accuracy.

This master’s thesis work focus on creating an improved model (New-model) of these heat pumps with a better defined range of validity and the possibility to individualize the model to each separate heat pump.

Some theoretical background concerning the heat transfer and the absorption heat pumps is briefly presented in this report as a base for new models of the absorption heat pump.

Energy and mass balances along with common calculation methods used in calculation with heat exchangers (the NTU method and the logarithmic mean temperature method) were used in the representation of the absorption heat pumps as an excel base models (New model).

The two brands of heat pumps (Entropie and Sanyo) were modeled in two separate theoretical models based on their design. These models should then be adapted to the six real heat pumps by altering a factor on the heat transfer coefficient.

Due to some problems and difficulty in the modeling and the use of Excel as a modeling tool descried in the report, only one of the two types of heat pumps was completely modeled.

It could also be concluded that the Model-Renova could, after some smaller alterations, be a fairly god representation of the real heat pumps from a process perspective.

Key words: Absorption heat pump, Thermodynamic model

Table of content

Nomenclature

$Q = Heat\ flow\ [W]$
$U = Overall\ heat\ transfer\ coefficient\ \ [W/m^2K]$
$A = Heat\ transfer\ area\ [m^2]$
$T = Temperature\ of\ the\ hot\ and\ the\ cold\ streams\ [K]$
$h = Enthalpy\ \ [kJ/kg]$
$u = Velocity\ [m/s]$
$h_{conv} = Convection\ \ coefficient[W/m^2K]$
$hf = Fouling\ coefficient\ \ [W/m^2K]$
$k = Heat\ conduction\ coefficient\ \ [\ W/m^2K]$
$D, d =\ Tube\ outer\ and\ inner\ diameter\ [m]$
$COP = Coefficient\ of\ performance\ heat\ pump$
$x = Mass\ fraction\ of\ LiBr\ [kg_{LiBr}/kg_{solution}]$
$DH = District\ heating\ water$
$MKS = Water\ from\ the\ flue\ gas\ cleaning\ (Cooling\ water)$
$MKS = (Swedish)\ Mellankylsystem$
$\rho = density\ [kg/m^3]$
$u = velocity\ [m/s]$
$cp = specific\ heat\ [kJ/kg\ K\ ]$
$Q = heat\ transfered\ \ [W]$
$\theta = Saturation\ temperture\ \ for\ pure\ water$ [K]
LMTD= Logarithmic mean temperature difference
NTU = number of heat transfer units
$\varepsilon = Heat\ transfer\ efficicnecy$
Model-Renova = Existing model of the heat recovery process at Renova
New- model= Model created during this master thesis
DH-District heating
Strong solution – High mass concentration of lithium bromide
Weak solution – Low mass concentration of lithium bromide

Superscript

$''' = per\ unit\ volume\ [1/m^3\ ]$
$\dot{} = per\ unit\ time\ [1/s]$

Subscript

h=hot
c=cold

i=inside
o=outside
$\sigma = saturation$
1, 2, …. .20 = stream number

1 Introduction

The continued growth of the world's population and the increased standard of living will bring out an increased demand for varies types of energy. The increase in the world's energy consumption is estimated by the American energy information administration, to be 44% from 2006 to 2030 *([1] EIA Energy outlook report, 2009)*.

All energy production will result in some harmful influence on the environment including global warming, emission of pollutions, land use and resource depletion.

As a way to manage the increasing amount of waste produced in the world and to contribute to meeting the increasing energy demand, the waste is incinerated and the released energy is recovered to produce electricity and district heating.

The energy produced in the waste incineration process can replace energy production by use of combustion of fossil fuels. The amount of district heating produced in waste incineration plants in Sweden each year could provide heat for 700000 villas and at the same time produce enough electricity to supply 200000 villas. *([2] Avfall Sverige, Avfallsförbränning)*

Using industrial heat pumps to make use of waste heat at low temperature can optimize the heat recovered from the incinerated waste.

1.1 Background

As always in industrial processes there is a need to optimize the operation in order to get a high overall efficiency of the plant. There is a project at Renova with the aim to find methods to optimize the production. This project is conducted together with a consultant company called "Energi projekt E&S (EPRO)". A thermodynamic model of the whole process and all important components has been made for this purpose (Model-Renova).

The Existing-model is built up in the software Excel and represents all parts of the energy recovery process. The equations representing the different parts of the process have been gradually improved and developed as the need for simulation of the different parts emerged. These have sometimes been found from simple thermodynamic equations and correlations between different parameters given as process data. All six heat pumps are modeled in the same way from the same identical correlation.

The accuracy and the range of validity in the Model-Renova of the heat pumps are somewhat uncertain and a need for improvement was therefore identified.

The fact that the model does not consider the characteristics of each individual heat pump will be a deficit in modeling operation changes or when the characteristics of the heat pumps change as a result of rebuilding or aging. A need to get a more detailed model that can be adapted to each individual absorption heat pump and their dependence on each other and the surrounding process was recognized.

The equations and relations that constitute the Model-Renova are partly made up from adaption of the model to measure data and might not give sufficient transparency and understanding of the process.

1.2 Objective

The aim is to create a model with an improved defined range of validity that can be converted into the Excel format of the Model-Renova.

The model should be a good thermodynamic representation of the individual heat pumps and be representative for the real heat pumps at different operation conditions. As the model will be a detailed representation of the reality there will also be an option to alter the model as the operation of the heat pump with aging and increased fouling on the equipment. This will also give an understanding which parameters that are important for the performance of the equipment.

The models should correspond to real measured operational data found from the real process in operation.

1.3 Scope

Since the main purpose of the models will be to simulate a stationary system and not to simulate failures or rapid changes, there will be no need to include transients into the model. The limits for the model of the absorption heat pump cycles will be at the heat pumps input and output of low pressure steam, condensate from the flue gas cleaning and the district heating water respectively. (See Appendix I, figure 26)

An absorption heat pump is a complex machine containing several different components and simultaneous mass and heat transfer. The amount of details in the model equations, should therefore always subjected too careful considerations.

Some assumptions have been made concerning the state of steam or condensate at certain points in the machine.

The properties of the working medium are assumed to be uninfluenced by properties of the additives included in the working medium.

1.4 Method

Existing literature was evaluated to find thermodynamic models that are used for simulating heat pumps. The examined correlations and models will, together with knowledge obtained in courses within the master program, be used when constructing the model for this application.

An evaluation of the accuracy of the model is made by statistics analysis of process data. The resulting model will then be adjusted to represent the individual heat pumps. This will characterize how the heat pumps differ in their operation.

2 Description of the Process

2.1 Renova AB

Renova AB is a waste management and recycling company owned by 11 municipalities in the western part of Sweden. The company takes care of various types of waste coming from both households and industry.

After the waste has been discarded and sorted into hazardous waste, recyclable, compostable and combustible waste it is transported by garbage trucks to Renova´s different waste management facilities.

2.2 The incineration plant

The part of the waste that is combustible is incinerated in Renova's incineration plant.

The incineration plant is located in Sävenäs close to the small river Säveån outside Gothenburg. The plant produced 228 GWh of electricity and 1197GWh heat in 2009 which corresponds to about 5% and 26% of Gothenburg's electricity and heat need respectively *([3] Renova AB, Miljörapport, 2009).*

There are four furnaces installed to combust the allowed amount of waste, around 550000 tons per year. (*[4] Renova AB, Homepage)*

The waste is combusted from a sloping grate in the bottom of the furnace. The grate is penetrated by a part the combustion air coming from beneath the grate.

As a way to reduce the formation of nitrogen oxides, the combustion is first preformed in a fuel rich conditions and the rest of the air is introduced higher up in the combustion chamber. Nitrogen oxides are also further reduced by injection of ammonia, a so called selective non-catalytic reduction (SNCR) *([5] Renova AB, Waste to energy plant in Göteborg).*

The walls in the combustion chamber are covered with tubes containing water. The water within the tube wall will be evaporated into saturated steam.

The steam is superheated to 400°C and 40 Bar in the superheater, located after the combustion chamber. The temperature of the flue gases leaving the superheater will be rather high and is therefore used to preheat the water in a feed water economizer.

As the flue gases leaves the feed water economizer they enter an electrostatic precipitator that is used to clean the flue gases from large particles and soot.

The flue gas will then be cooled down from about 230 °C to 140 °C in a flue gas economizer by heating the district heating.

The next step is the wet flue gas cleaning consisting of a wash reactor and a condensing reactor. The heat released in these processes is delivered to the district heating via six absorption heat pumps. The flue gas will then be reheated by use of steam and as a last step the flue gases are cleaned in a fabric filter before it leaves the process in the chimney. (Figure 1)

The superheated steam leaving the superheater is expanded in a turbine that is connected to a generator producing electricity. The electricity produced is delivered to the electricity grid via a transformer.

Not all of the steam will pass all the way through the turbine; some steam is extracted at intermediated levels 3, 5 and 7 bars. The wet steam that passed all the way through the turbine will be condensed in a heat exchanger that will deliver heat to the district heating network.

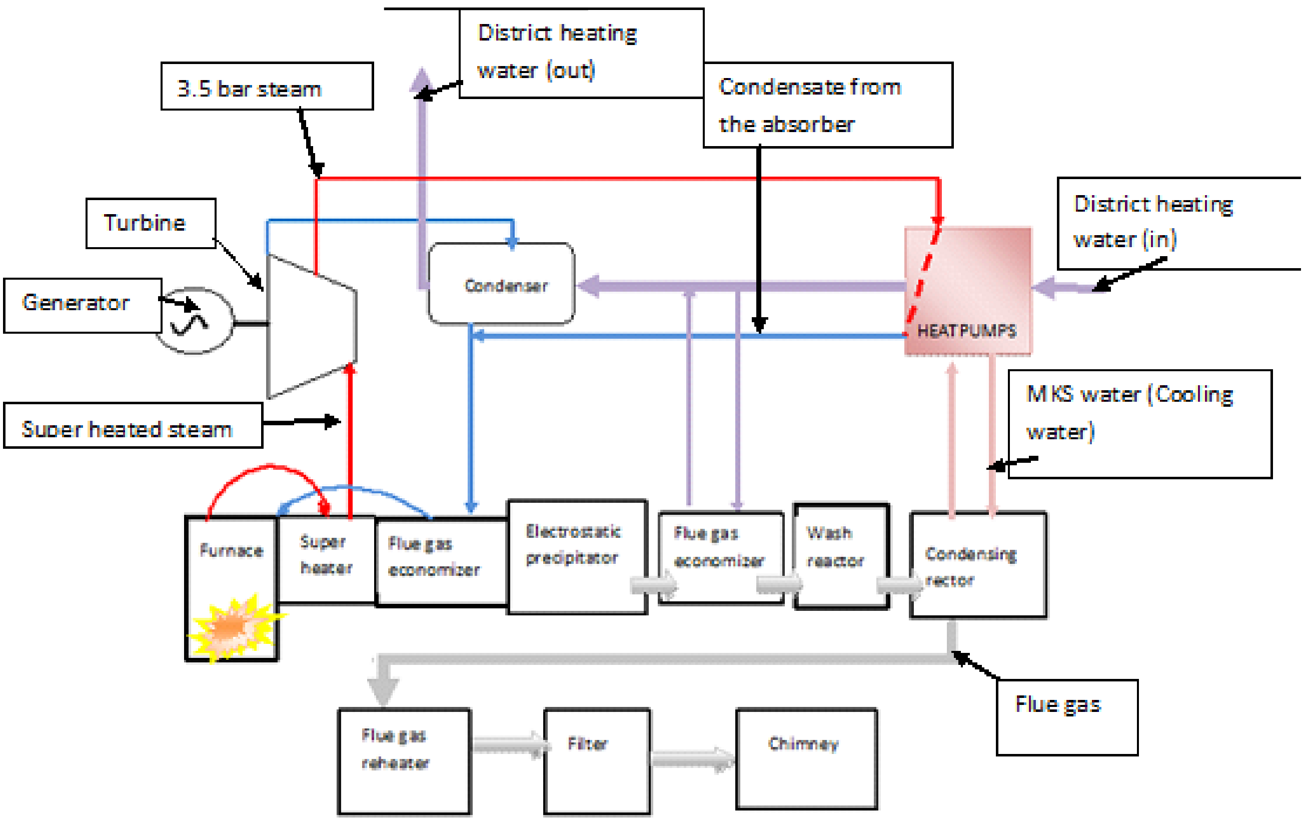


Figure 1. The entire energy recovery process at Renova waste incineration plant

2.3 The absorption heat pumps

The second law of thermodynamic states that:

Energy has both a quantity and a quality and that all spontaneous processes occur in the direction of decreasing energy quality.

This means that heat can only be raised from a low temperature to higher temperature by introducing some form of higher quality energy.

Not so many practical end applications where energy at a low temperature can be utilized, which means that there is a need for a technical solution that can, without violating the second law of thermodynamics, bring low temperature energy up to a higher temperature. A heat pump or a heat transformer is used to do just that.

A common application for a heat pump is to recover low temperature waste heat that cannot, due to the low temperature, be used for other purposes within the process and bring it to a more useful temperature.

The cycle can also be inverted in order to accomplish cooling (e.g. refrigerator) when the heat is removed from a cooled space and delivered to the higher temperature surrounding.

There are several different types of heat pumps and heat transformer technologies. The most common type of heat pump is the compression heat pump that accomplishes an increased temperature due to the fact that the working medium has a different boiling and condensing temperature at different pressures. A compressor powered by some form of work, for instance electricity, is then used to create the pressure difference within the heat pump.

Another approach is to use steam to raise the temperature up to a desired level, which is the case in an absorption heat pump. An absorption heat pump will instead utilize the increased boiling point of the solution compared to the boiling point of the pure substance. Steam is then used instead to drive the heat pump cycle as describe bellow.

Absorption heat driven heat pumps can be divided into two different types. Type I is a cycle where heat is introduced at both a high and a low temperature and the heat deliver from the heat pump has an intermediate temperature.

In the type II heat pump, heat will be introduced at an intermediate level and output heat will have both higher and lower temperatures *([6] Herold, Keith 1996)*

The absorption heat pumps considered here are of the first type.

Absorption heat pump cycle can be divided in one two or several effects meaning that the generator and condenser have been divided into two or more stages with an additional solution heat exchanger, in order to improve the heat pumps efficiency *([6] Herold, Keith 1996).*

The basic components included in the absorption heat pump are made up by heat exchanges with some alterations (2.4.1 and 2.4.2) and are described below.

Weak solution referrers to as dilute solution with a relatively low concentration of lithium bromide compared to the strong solution that has a high concentration of lithium bromide.

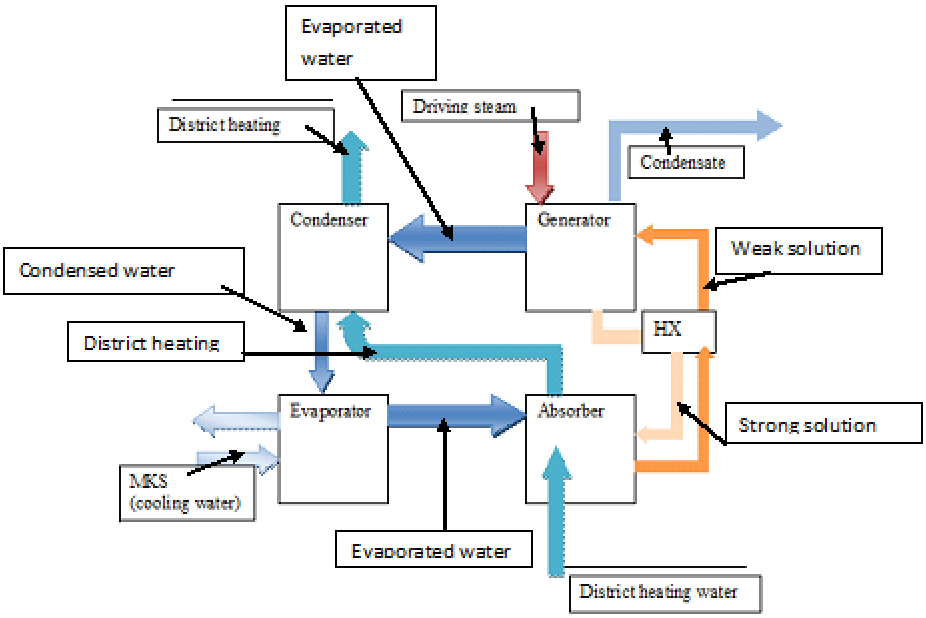


Figure 2: A single effect absorption heat pump

Evaporator

The saturated water coming from the condenser into the evaporator will evaporate into steam due to the low pressure within the evaporator by use of a low temperature heat source. The temperature of evaporation will then be around 30-40 °C and the heat required to evaporate can, due to the low pressure (around 0, 08-0, 04 bar), be made up by low temperature waste heat.

Absorber

The steam entering the absorber will be absorbed by a salt solution in an exothermic reaction. The week solution will have a higher boiling temperature than the boiling point of water and will therefore condensate. The heat realized in the condensation will be delivered to the district heating network.

Generator

The generator requires low pressure driving steam to evaporate the water out of the weak salt solution in an endothermic process. The strong salt solution is returned to the absorber while the evaporated water solved from the solution continues to the condenser.

Heat exchanger (HX)

A heat exchanger placed between the generator and the absorber will utilize the heat in the strong solution returning to the absorber to pre-heat the week solution before it enters the generator.

Condenser

The steam coming from the generator is condensed and heat is delivered to the district heating network. The condensed water is then returned to the evaporator and the cycle is completed.

2.3.1 Working medium

The working medium includes both the solution and the pure component and consists of one volatile and one non-volatile component. The working medium is most often made up by water as a volatile component together with the non-volatile component consisting of lithium bromide or ammonia as a volatile component and the non-volatile water**.**

All heat pumps at Renova operate with a working medium consisting mainly of water and lithium bromide. Lithium bromide is chemical compound of Lithium and Bromide and has a melting point of 547°C. Lithium bromide is soluble in water, alcohol and glycol and looks like a white powder *([7] Chemland,21)* .

Using Lithium bromide/water solution as a working medium compared to the use of water / ammonia solution has advantages and disadvantages. The efficiency of a single stage heat pump is increased when using lithium bromide/water compared to using water/ammonia.

The specific heat of water/ammonia solution is also about twice as high as for the lithium bromide/water solution causing any higher losses (irreversibility) as a result of inefficiency in the heat exchangers *([6] Herold, Keith 1996)*. The latent heat of water/ammonia solution is also about half the heat released as the lithium bromide solution changes phase resulting in a higher flow of the refrigerant to achieve the same heat or cooling effect *([6] Herold, Keith 1996)* .

A mayor disadvantage when using a salt solution like Lithium bromide salt solution is the risk of crystallization (se section below) that will restrict the operation of the absorption heat pump. Ammonia and water solutions is most often used when temperatures goes under zero in cooling heat pumps.

In order to prevent both corrosion and crystallization and also to increase the heat transfer giving an enhancement of the performance of the absorption heat pump, some additives have been included in the working fluid.

2.4 The heat pumps at Renovo waste incineration plant

The six heat pumps installed at Renova are of two different brands with a design that deviate somewhat from the general description above. Four of the heat pumps are of an older brand, manufactured by a company called SANYO. They have a nominal cooling effect of 4MW i.e. they can cool 4MW from the low temperature heat source.

The two more recently installed heat pumps are of a brand called Weir/ Entropie and have a more modern design and construction. They have a nominal cooling effect of about 6MW each.

2.4.1 Description of the four Sanyo machines

The four Sanyo heat pumps had identical design at the time of installation year 1988.

The heat pump machines are based on the absorption heat pump cycle but include one additional condensate after cooler heat exchanger and utilize a solution consisting of Lithium bromide and water as a working medium.

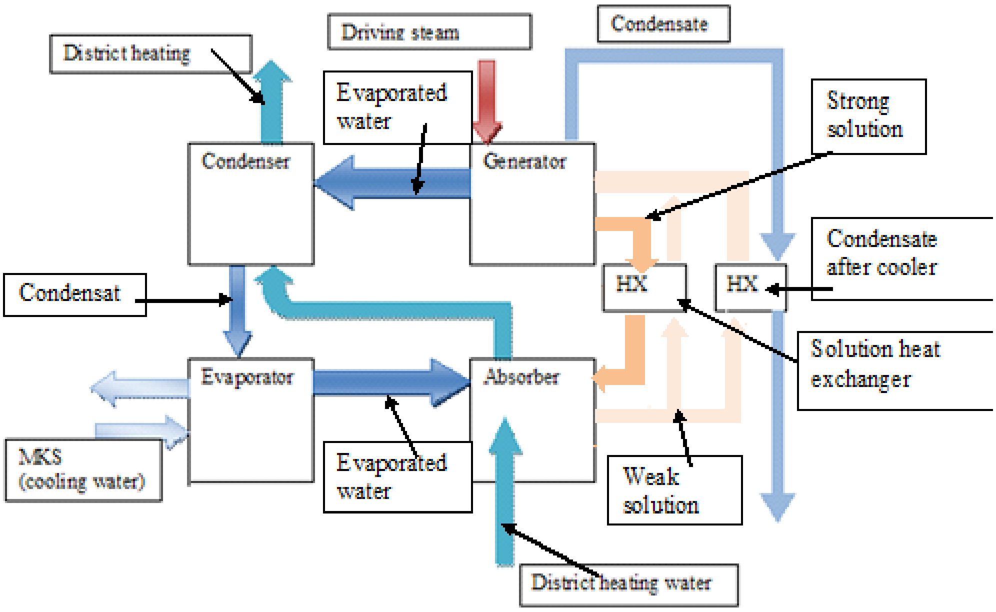


Figure 3: Sanyo absorption heat pump

Construction

The bottom of the heat pump consists of a large tube shaped vessel that contains both the evaporator and the absorber. The working medium is separated between the evaporator and the absorber by use of a fluid separator. The liquid that have not been vaporized as it passed the tube bundle in the evaporator is collected in the bottom below the tubes and circulated up to the top again. (See APPENDIX Figure 25-26)

The low temperature heat source from the flue gas condensation and the district heating water is lead through a tube bundle in the evaporator and absorber respectively. Located above the tube bundles are distribution tubes that have the objective to distribute the working fluids evenly over the tube bundle.

The evaporator can be considered as a four pass tube heat exchanger and the absorber as a two pass heat exchanger.

The driving steam is condenses as it is circulated within the tubes in generator order to separate the water from the Li/Br solution. The generator can be considered as a one pass evaporator where pool boiling occurs. (See APPENDIX Figure 27)

Evaporated water from the generator is entered in the condenser above the one pass tube bundle and the condensed water from a film on the tubes surface. The district heating water circulates within the tubes and will in that way recover the heat from the condensing steam. (See APPENDIX Figure 28)

Some heat in the strong solution leaving the generator is recovered as it is circulated in the tubes of a tube and shell heat exchanger (solution heat exchanger). The heat is transferred to the weak solution as it passes through outside the tubes on the shell side of the heat exchanger. Both the strong and the weak solution pass through the heat exchanger three times. (See APPENDIX Figure 29)

The condensed steam leaving the generator has a substantially high temperature and needs to be sub cooled in order to be handled in the collection container.

This is done through heat exchanging with the weak solution before they are entered to the generator .This is done in parallel with the weak-strong solution heat exchanger in the condensate after cooler. The condensate after cooler heat exchanger is a tube and shell heat exchanger where the condensed steam circulates inside the two pass tubes and the weak solution outside the tubes *([8] Sanyo manufacturing files).*

Control system

The Sanyo heat pumps have been equipped with two different control systems connected via a min selector, in order to control the pressure on the driving steam.

The first system controls the level of working fluid inside the evaporator.

More condensate from the condenser enters the evaporator than can be vaporized in the evaporator lead to an increased liquid level in the bottom of the evaporator.

When steam pressure of the driving steam entering the generator decreases, less water will be evaporated out of the solution and will leave the generator as steam. The decreased flow of evaporated water from the generator will result in a decreased flow of condensate from the condenser into the evaporator. This decreased flow will in turn lead to a lowered water level at the bottom of the evaporator.

An increased level of water in the bottom of the evaporator will mean that more water is on the water side (condenser and evaporator) and less water will be on the solution side (absorber and generator). The decreased amount of water on the solution side will mean that the medium concentration of Lithium bromide between the strong and the weak solution increases. A conclusion will therefore be that the liquid level in the evaporator is in proportional to the medium concentration of lithium bromide.

The second system regulates the steam demand depending on the desired outgoing temperature of the cooling water (MKS water).

The control system with the lowest control value will be the one that is selected. The system controlling the fluid level have been designed in order to minimize the steam consumption and will most often be the one giving the lowest signal and therefore be the prioritized signal. This is an important feature since the plant will always will be regulated in order to maximize the steam through the turbine and therefore maximize the production of electricity *([8] Sanyo manufacturing files).*

2.4.2 Description of the two Entropie machines

The two more modern heat pumps have a more advanced design and control system than the four older heat pumps, but are based on the same absorption heat pump cycle. However the design of the Entropie heat pumps includes, as in the Sanyo heat pumps, a condensate after cooler that is in this case mounted on the district heating water. The design also includes an additional heat exchanger in parallel to the absorber.

These heat pumps were installed 2003 and are identical in their initial design. These heat pumps also utile a solution manly consisting of lithium bromide and water as a working fluid.

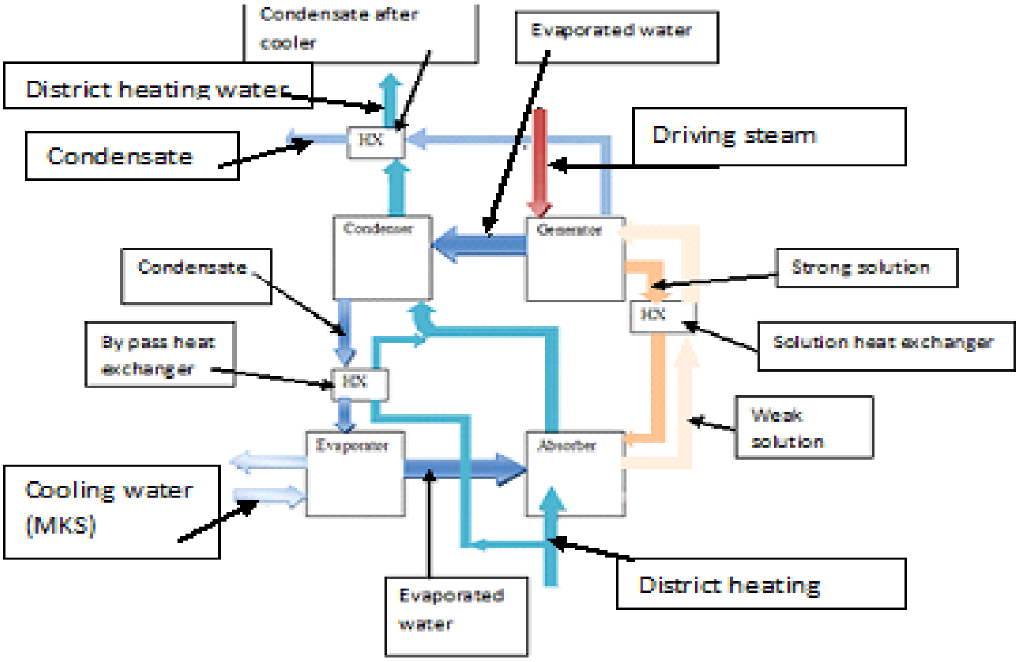


Figure 4: Entopie heat pump

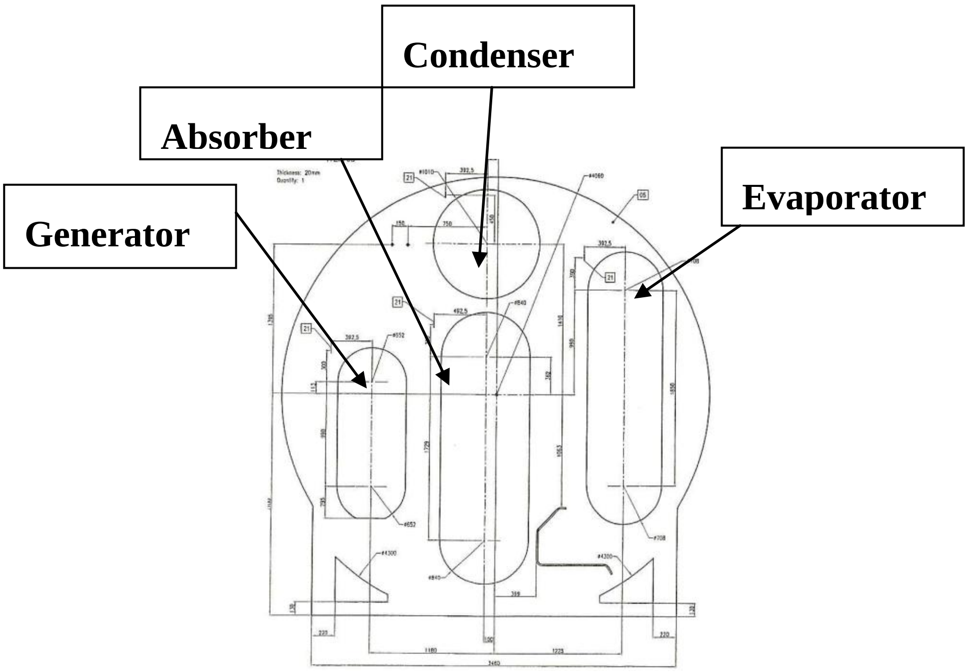


Figure 5: The front side of the Entropie absorption heat pump [9]

The tube bundle in the evaporator is divided in three parts as a three pass heat exchanger. The evaporated steam passes leaves the evaporator for the absorber and the water that still is in a liquid phase is collected in the bottom of the evaporator where it is mixed with the water coming from the condenser.

A pump is used to pump the collected liquid up on top of the evaporator where the water is sprayed down on the tube bundle from pressurized nozzles located above the tube bundle.

The steam from the evaporator enters the absorber from above the tube bundle and the steam is absorbed by the strong Lithium bromide solution. The district heating water circulates inside the two pass tube bundle containing aligned tubes.

A circulation pump is used to bring the solution from the low pressure in the absorber to the high pressure in the solution heat exchanger, generator and condenser.

The solution heat exchanger is designed as a cylindrical plate and shell heat exchanger that will transfer heat from the warm strong solution coming from the generator to the weak solution coming from the absorber. As the strong solution coming from the absorber enters the generator it is sprayed down over a one pass tube bundle circulated with condensing steam. In this way the Lithium bromide and the water is being resolved from each other.

The condenser can be seen as an ordinary one pass tube and shell condenser where the tube bundle is constructed in a cylindrical and staggered manor. The district heating water is circulated inside the tubes as the steam coming from the generator is condensed outside the tubes. The steam is entered above the tubes and the condensate is collected underneath the entire tube bundle.

Some of the reaming heat in the condensate is recovered and transferred to the district heating network after the condenser. The condensate after cooler is also constructed as a cylindrical plate and shell heat exchanger.

A plate and shell heat exchanger is mounted parallel to the absorber on the district heating side and is used to recover heat from the condensate coming from the condenser and make sure that it is in a saturated state.

Control system

The Entropie heat pumps have been equipped with an advanced control and regulations system in order to optimize the operation and to avoid crystallization.

The system includes several temperature and pressure measurement and rupture disk to protect against high pressure and temperature in the condenser as well as in the district heating water.

A main concern when designing a control system for an absorption heat pump is the risk of crystallization (see explanation below). The protection against crystallization includes measures to sustain a god circulation and to keep a sufficient low level of Lithium bromide concentration ([9] Entropie manufacturing files).

The temperature difference between the strong solution after the generator and the temperature of the weak solution before it enters the generator corresponds to the boiling point elevation of the weak solution. The boiling temperature elevation is, as described in a section below, connected to the concentration of the solution and need therefore be monitored in order to control the risk of crystallization.

The liquid level in the bottom of the evaporator and generator is controlled by the circulation pumps circulating the liquid faster or slower in order to decrease or increase the heat transfer and an therefore decrease or increase the level.

The liquid level in the condenser is controlled by the opening of two valves in the bottom of the condenser that regulates the outflow of condensate.

2.5 The overhead control system

The district heating water is a part of the wide spread integrated district heating network that delivers heat to many customer at different geographical location and from several heat producing units. This system is owned and controlled by an energy company called “Göteborg Energi” that manages this complex network. The water flow and ingoing and desired outgoing temperature is therefore predetermined and controlled by “Göteborg Energi” as the water enters the plant at Renova.

The district heating pipes to the heat pump units are designed in a parallel design to the big district heating pipe so that they take water from the district heating network and return it at a higher temperature.

The four Sanyo heat pumps and the two Entropie heat pumps have separate pipes connected to the district heating network.

The district heating flow to the two Entropie Heat pumps are regulated by frequency regulated pumps that is controlled from the control room.

In case the heat pumps demand more water from the district heating network than the incoming district heating water flow into the plant the water will move backward in the district heating lines and be lead through the heat pumps once more. This will result in a higher temperature of the district water coming into the heat pumps and therefore a lower efficiency of the heat pumps.

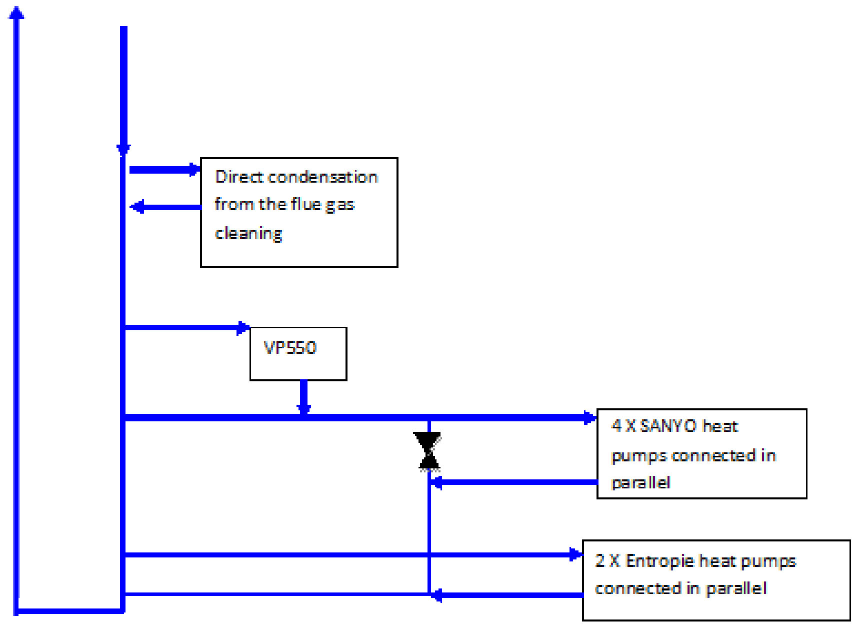


Figure 6: The six absorption heat pumps connected to the district heating network

3 Theoretical framework

An absorption heat pump is a very thermodynamically complex machine that can be described by chemical, mechanical and heat interactions.

As any system, the heat pump must obey by both the first and second law of thermodynamics, meaning that no energy or mass can be created or destroyed within the system and that no heat can directly be transfer to a higher temperature without addition of some external work.

Since the system mainly consists of heat exchangers in different executions, a parameter that strongly influencing the performance of the heat pump is the heat transfer area and heat transfer coefficient between the two fluids of each heat exchanger.

3.1 Models in the literature

Absorption heat pumps have been modeled by several different authors for the purpose of simulating different aspects of the heat pump.

According to Qin many of the developed models and investigations have been focused on a finite-time thermodynamics .The finite time -thermodynamics (or finite-size thermodynamic or endoreversibility or entropy generation) analysis is most often used in order simulate the optimization of operation of the absorption heat pump including the internal irreversibility, heat leakage and losses with heat resistance. *([10] Qin et.al 2007).*A finite time analysis results in many advanced equations and require a powerful calculation tool and will therefore not be used in the model created here.

Using the second law of thermodynamics to model the entropy generation or to perform an exergy analysis on the heat pump in order to evaluate the performance of the absorption heat pump, have been done by several researchers including Sencan and Wolf *([11] Sencan et.al 2004), ([11] Wolf .et.al 1988)*

Many of these analysis focus on locating the sources of losses and finding design possibilities to decreases these losses by an optimal design of the absorption heat pump system and the area distribution, and might therefore be more used in a design aspect of the heat pump where the heat pump configuration and area distribution in the included heat exchangers are design variables.

A model that is frequently mentioned and referred to in the literature is the “ABSIM” model (ABsorption SIMulation) developed by Gershon Grossman. This model is used to investigate different cycle configurations of absorption systems in steady state conditions *([13] Grossman et.al 2001).* The code in this model is constructed in a modular way as subroutines for each component solving the governing equations. The ABSIM model is used to evaluate different cycle configurations and working fluids and simulate absorption systems at off design operation.

A less complex approach is to model the absorption heat pump by considering the mass, species and heat balances. This approach was taken in the modeling of absorption heat pump cycles by Keith Herold who made in a special version of the Engineering Equation solver (EES) software. *([6] Herold, Keith 1996)*

The approach taken in this model is suitable for simulating a steady state operation found in the modeling of an existing heat pumps operation and will therefore serve as a base for the modeling in this master thesis work.

3.2 Coefficient of performance

When describing the efficiency of a process it is common to refer the required energy input divided by the useful energy generated by the process as a way to describe how efficient the process convert the input to desired output.

When referring to a heat pump cycle it is more common to use the coefficient of performance that will compare the work or driving steam required to produce the useful output.

The coefficient of performance will often exceed one and can therefore not be considered to be a true efficiency. Efficiency that is higher than one would mean that more useful energy output would be created in the machine than is required as energy input which would violate the first law of thermodynamics.

What is considered as the required energy output differs if the heat pump is used as a cooling machine or as a heating machine i.e. when a cooling machine is considered the required output is the amount of heat that you cooled off and in a heating machine the amount of the heat delivered.

$$COP_c = \frac{Q_c}{Q_{steam\ or\ W}} \quad (3.2.1) \quad ([6]\ Herold, Keith\ 1996)$$

$$COP_h = \frac{Q_h}{Q_{steam\ or\ W}} \quad (3.2.2) \quad (\ [6]\ Herold,\ Keith\ 1996)$$

As the heat pumps considered here is used to heat the district heating system, only the last expression of the COP will be considered in this report.

3.3 Mass and energy analysis

A common approach when conducting energy and mass balance of an absorption heat pump system is to consider each component as a separate control volume where inlet and outlet steams flows cross the system boundary of each control volume .

Mass and energy balance is based on the first law of thermodynamics meaning that the mass and energy flow into a system is equal to the change of mass and energy inside the system and the mass and energy flow out of the system.

The same goes for all individual components in the flow when no reaction occurs between the components.

The mass and energy balance equations for each individual component was stated, and all components was connected together to form the entire heat pump unit by linking inputs and outputs between the different control volumes. (See figure 15)

3.4 Absorption and desorption

The water vapor entering the absorber is absorbed into the strong solution.

The result of mixing two different species like water and Lithium bromide is not that the enthalpy of the solution will be the mass-weighted average of enthalpies.

When water is absorbed into the strong Lithium bromide solution, it will result in an enthalpy of the diluted solution that is less than the weighted average enthalpy meaning that the absorption process is exothermic i.e. heat is released to the district heating water (heat of solution). *([6] Herold, Keith 1996).*

In adiabatic system this would result in an increased temperature but as the absorber in the heat pump is constructed as heat exchanger .i.e. not an adiabatic system, the released heat will be delivered to the district heating water.

3.5 Heat transfer within the heat exchangers

Heat is transferred from a stream with higher temperature to a stream with lower temperature in heat exchangers.

The process of heat transfer occurs in three different ways, conduction through the tube wall, convection outside the tube wall and radiation. As the temperature difference the working fluid and the tube wall in heat exchanger tend to be small the heat transferred by radiation can be neglected.

The heat transferred between the two streams can be described as:

$$Q = U * A * (T_h - T_c) \quad (3.5.1) \quad ([14]\ Incropera, et. al\ 2005)$$

U is the overall heat transfer coefficient which describes the above mentioned heat transfer processes by including coefficient for the conduction and convection as well as the fouling resistance both inside and outside the tube.

The heat transfer is not constant throughout the whole heat exchanger and U will therefore be seen as an average value over the entire heat exchanger. The fouling resistance is the heat transfer resistance caused by dirt that is stuck on the tube wall.

$$U = \frac{1}{\frac{D}{d * h_i} + \frac{D}{d * h_{fi}} + \frac{ln(\frac{D}{d})}{2 * \pi * k * L} + \frac{1}{h_{fo}} + \frac{1}{h_o}} \quad (3.5.2)$$

$$([14]\ Incropera, et. al\ 2005)$$

The amount of heat that can be transferred per area unit is determined by the shape of the heat exchanger as well as the temperature and velocity of the fluids, the boundary layer surrounding the heat exchanger area and the process occurring in the heat exchanger.

3.5.1 The logarithmic mean temperature difference method (LMTD)

The temperature changes within the heat exchanger will result in a varying driving force throughout the length of heat exchanger as a result of varying temperature difference between the hot and the cold fluid. As the hot fluid heats the cold fluid and the temperature difference between them decrease resulting in logarithmic temperature profile through the length of the heat exchanger. (Figure 14)

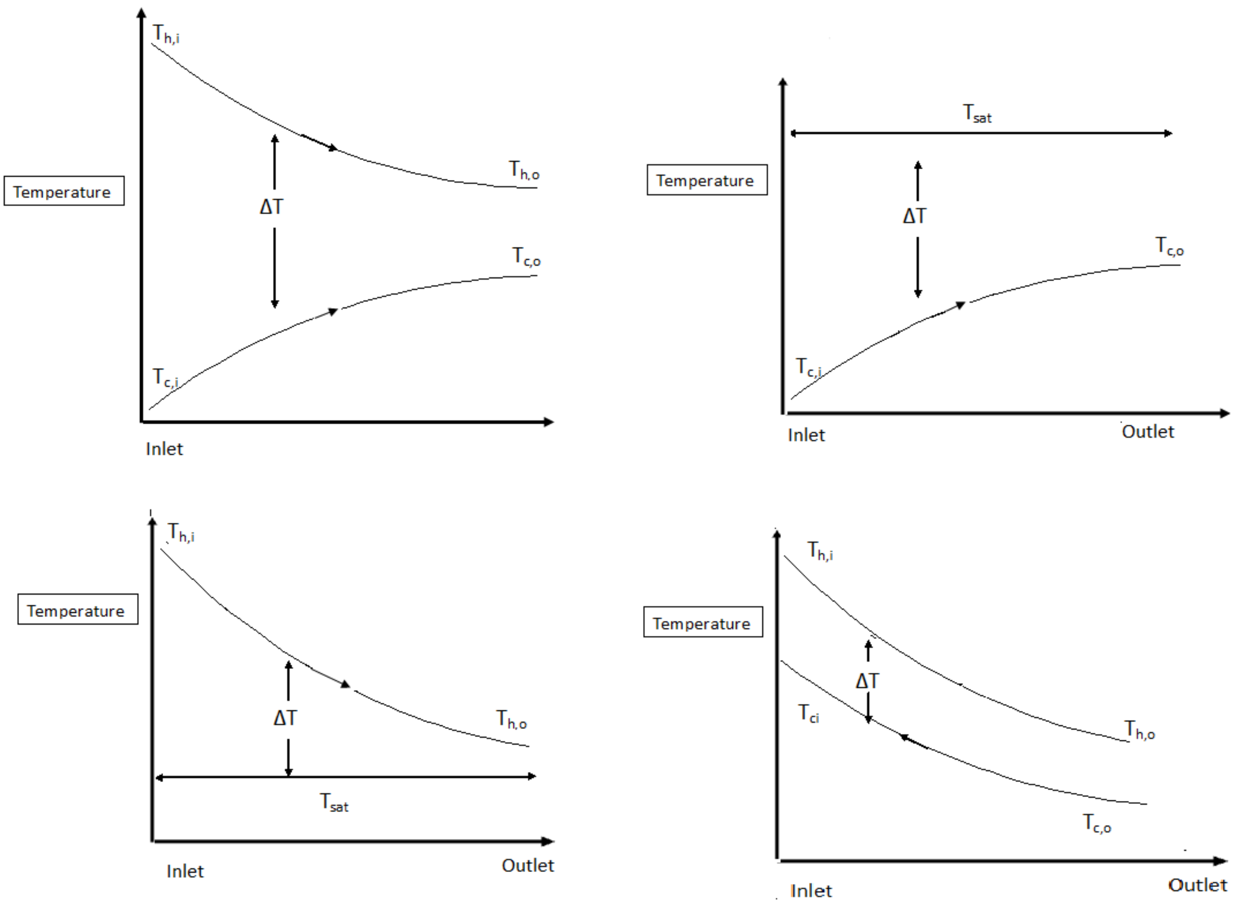


Figure 7: Temperature variation, Top row left parallel flow, Top row right condensation, bottom row left evaporation, bottom row counter flow

As the driving force and the temperature difference between the hot and the cold stream changes along the heat exchangers length it is not representative measurement of the driving force to take the difference between the two inlet streams.

In order to formulate a better estimation of the mean driving force throughout the heat exchanger, an integration of the differential equation describing the change in the temperature difference between the hot and the cold streams within a counter current is made.

$$\Delta T_{LM} = \frac{(T_{h,o} - T_{c,i}) - (T_{h,i} - T_{c,o})}{\ln\left(\frac{T_{h,o} - T_{c,i}}{T_{h,i} - T_{c,o}}\right)} \quad (3.5.1.1) \quad ([14]\ Incropera, et. al\ 2005)$$

To represent other heat exchange designs a correction factor Ft is used to calculate the corresponding logarithmic mean temperature difference.

$$\Delta T_{LM,corrected} = \Delta T_{LM} * Ft \quad (3.5.1.2)$$

The logarithmic mean temperature difference is used instead of the temperature difference in the above expression of the heat transferred. (Equation 3.5.1)

3.5.2 The number of transfer units - NTU method

In calculations where only the inlets temperatures are known, the logarithmic mean temperature difference method requires a sometimes very big iteration process.

A simplified method to evaluate the heat exchangers performance is called the Number of Transfer Unit method or the NTU method.

This method is based on the maximal amount of heat that theoretically could be transferred between the two streams (Q_{max}) and a dimensionless parameter called the heat transfer effectiveness ($\varepsilon = \frac{Q_{max}}{Q}$) *([14] Incropera 2005)*

The heat transfer effectiveness is the relation between the actual heat transferred and the theoretical maximum heat transferred.

The heat transfer effectiveness depends on the type of heat exchanger and the flow pattern and is calculated from the heat capacity rate (C=Cp*$\dot{m}$)) of the two streams and also the number of heat transfer units ($NTU = \frac{U*A}{Cmin}$).

The maximal theoretical heat that can be transferred between the two streams will be calculated based on smallest of the heat capacity rates (C_{min}) and the difference between the two inlet temperatures. *([14] Incropera 2005)*

As the temperature of the stream changes the specific heat will be calculated at a mean value of the inlet and outlet temperatures. This simplification will only be valid when the specific heat is fairly constant i.e. no big changes in temperature or concentration.

3.6 Boling point elevation

As the concentration of Lithium bromide decreases by the absorption of water in the absorber, the saturation temperature will increase.

This increased saturation temperature will in the low pressure result in the condensation of the water in the solution.

The opposite occur in the generator where the water is boiled of resulting in a decreased saturation temperature. The difference between these two temperatures is the boiling point elevation. The boiling point elevation is measured as the difference between the temperature of the strong solution out from the generator and the weak solution coming from the absorber.

This change in the boiling temperature with different concentrations is the principle that enables the absorption heat pump to move heat from a low temperature heat source up to a useful temperature.

Since the boiling point is a function of the lithium bromide concentrations, the boiling point elevation can be used as a measure of the concentration in the solution in order to prevent crystallization (see section 3.8).

3.7 Crystallization

Crystallization is, when the concentration of the salt exceeds the solubility limit, causing precipitation of the salt component.

The solubility is a strong function of the mass fraction and temperature and a weak function of the pressure.

Once the salt has started to precipitate it forms crystals that enhance the possibility for further creation of crystals. The formation of crystals will accelerate and continue even though the satiety has declined. The crystals can the clog the system and create blockages in the flow *([6] Herold, Keith 1996)*

To reduce the risk of crystallization in a Lithium bromide heat pump the temperature needs to be sufficiently high and the concentration be sufficiently low.

The highest risk for crystallization is when the strong solution has been cooled by the solution heat exchanger. This is the point where the concentration is highest and the temperature is lowest. (See point A in figure 15 below)

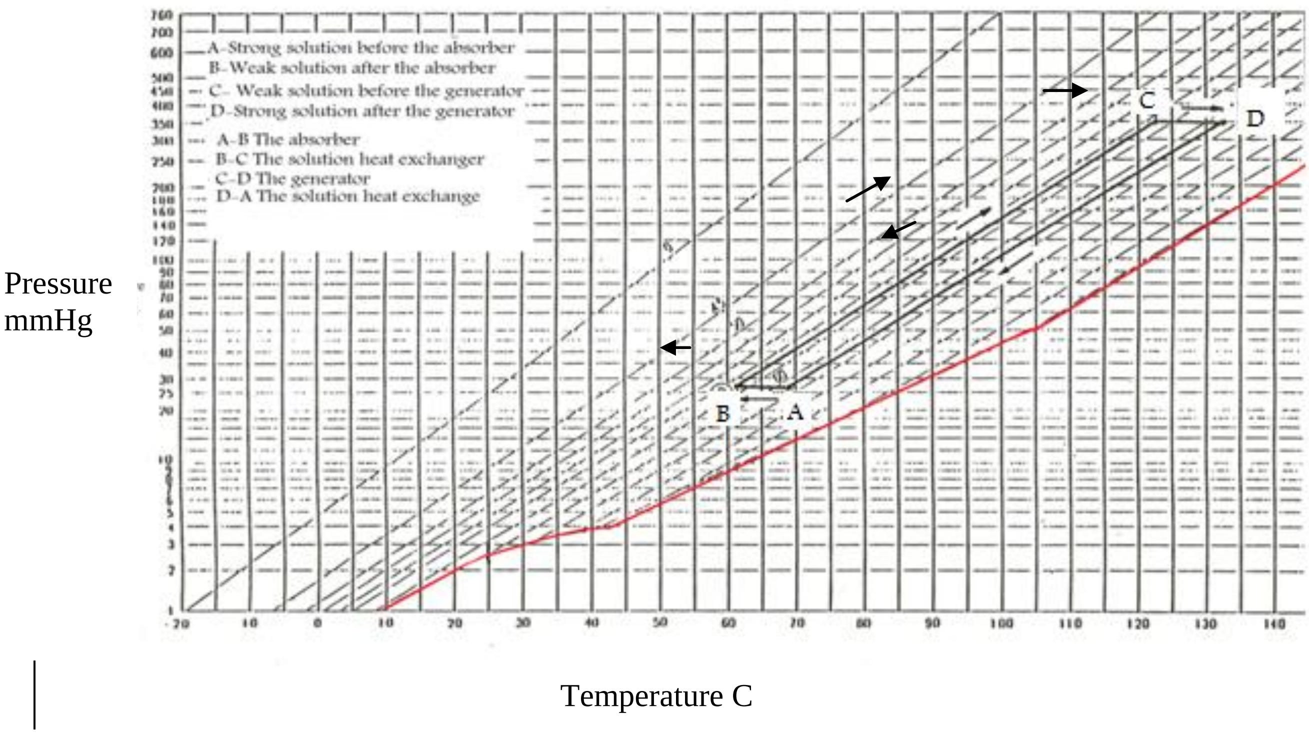


Figure 8: Crystallization line, absorption desorption ([8] SANY

3.8 Thermodynamic properties of lithium bromide and water solution

The thermodynamic properties of the lithium bromide solution are essential to get a god thermodynamic representations of the processes in the absorption heat pump.

The temperature of the working fluid in the considered heat pumps varies from the high temperatures in the strong solution to the low temperatures of the weak solution leaving the absorber and the concentrations varies from the weak concentration to the strong concentration. The thermodynamic properties must therefore be valid for a temperature range of about 30 °C to 130 °C and for mass fraction 0, 3-0, 8 (kg/kg).

Many articles have been made over the properties of lithium bromide/water solutions. An often used formulation of the properties of Lithium bromide is made by McNeely and has validity between 15 °C up until 166°C. (*[15] Klomfar J Patek 2006)*

The properties for the lithium bromide/water solution is calculated from an empirical correlations valid between 273K (unless the crystallization temperature is larger than 273K) and 500K and from 0 to 75% lithium bromide concentration, which was found in an article by J. Pa̓tek, JKlomfar *([15] Klomfar J Patek 2006).*

3.8.1 Enthalpy

The polynomial to find the enthalpy from the temperature and the concentration consist of 31 terms in a summation.

$$h(T,X) =$$

$$(1-X)*h' + h_c \sum_{i=1}^{30} a_i * X^{m_i} * \left(\frac{T_c}{T-T_o}\right)^{t_i}$$ (3.9.1.1) *([14] Klomfar J.Patek 2006)*

It can be seen in the figure below that the enthalpy of the solution will decrease as the concentration increases and obviously also decrease with decreasing temperature.

The enthalpy at zero concentration is the same as for pure water which could be expected (figure 16).

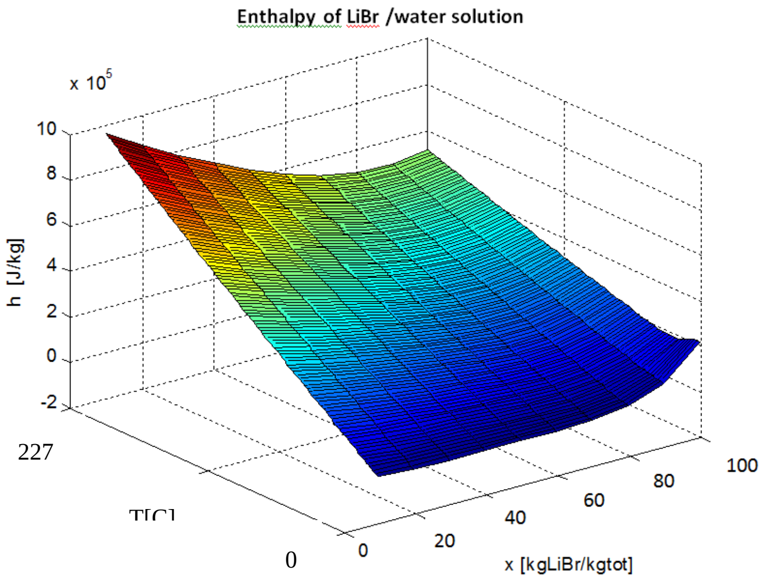


Figure 9: Enthalpy of the Lithium bromide/ Water solution

3.8.2 Saturation pressure

The saturation pressure of the solution is calculated from the polynomial bellow. In comparison to the polynomial to calculate the enthalpy the saturation pressure only includes 8 terms.

$$p(T,X) = P_\sigma(\theta) \quad \theta = T - \sum_{i=1}^{8} a_i x^{m_i}(0{,}4 - X)^{n_i}\left(\frac{T}{T_c}\right)^{t_i} \quad (3.9.2.1)$$

([15] Klomfar J Patek 2006).

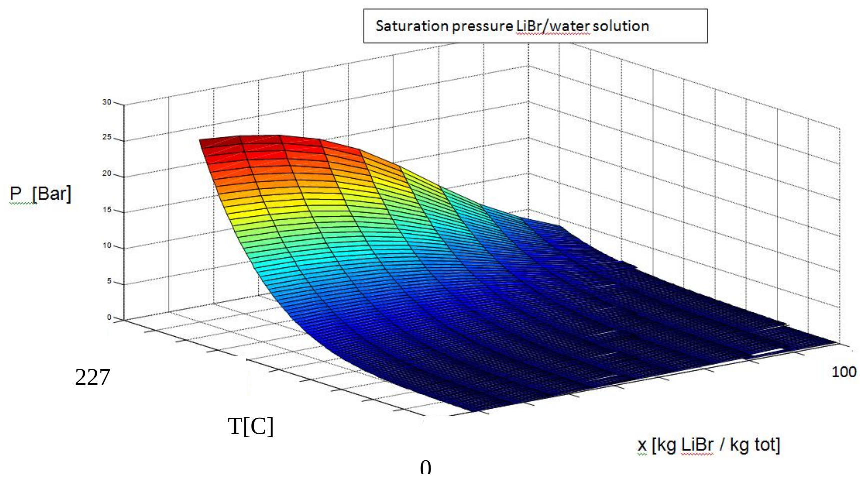


Figure 10: Saturation pressure of the Lithium bromide/ water solution

The saturation pressure will of course decrease with the temperature in the same way as for pure water, but will also decrease with decrease with increasing lithium bromide concentration.

This behavior is the reason for that the absorption heat pump can move heat from a low temperature to higher temperature

3.8.3 Specific heat and density

The equations used to calculate both the specific heat (Cp) and the density (ρ) for the lithium bromide and water solution are calculated from a similar polynomial as for the enthalpy includes fewer terms

$$Cp(T,x) = (1-x) * Cp(T) + Cp_{p,t} * \sum_{i=1}^{8} a_i * x^{m_i} * (0{,}4-x)^{n_i} \left(\frac{T_c}{T-T_o}\right)^{t_i} \quad (3.9.3.1):$$

([15] Klomfar J Patek 2006).

$$\rho(T,x) = (1-x) * \rho'(T) + \rho_c \sum_{i=1}^{2} a_i * x^{m_i} \left(\frac{T}{T_c}\right)^{t_i} \quad (3.9.3.2)$$

([15] Klomfar J Patek 2006).

3.9 The Model-Renova and its limitations

The Model-Renova have been implemented in Excel and covers the whole steam cycle from the generation of steam in the furnace, the turbine, and the steam headers to the point when the water return to the furnace as well as the heating of the district heating water and the thermodynamic aspects of the flue gas cleaning process. (Appendix V)

The model was built in a very simplified approach and has been gradually improved as the need for simulation of certain parts of the process has emerged.

The Model-Renova of the absorption heat pump represents the four smaller Sanyo machines as one unit and the two lager Entropies machines as one unit. (Appendix IV)

The model is based on process data collected for one of the Entropie heat pumps. This heat pump is then modeled by an empirical correlation of how the heat recovered from the flue gas cleaning system varies with the temperature in the outgoing district heating water, temperature of the incoming water from the flue gas cleaning and the driving steam pressure. The same procedure was used for a correlation of the coefficient of performance (COP) as a function of the incoming district heating temperature.

This correlation is then altered with respect to the nominal cooling power to be valid for the small Sanyo heat pump. The units representing the two Entropie heat pumps and the four Sanyo heat pumps are therefore described by multiplying this correlation with a scaling factor according to the nominal effects.

The COP value is then used to calculate the resulting heat delivered to the district heating and also the steam consumption.

The model is also equipped with two correction factors meant to describe the situations when not all of the heat pumps are running. One factor describing the division of district heating water between the two units and one factor that describes if the calculated COP value is expected to be under or over estimated in this particular operation condition.

3.9.1 Accuracy

The accuracy of the Model-Renova has been evaluated by comparing the results of the model with actual process data from the real heat pumps used as input. The available database located on sight (OPTIMAX) has been used to collect process data from every hour from 2009-07-03 to 2009-12-10 (3842 time samples).

The model of the absorption heat pumps has been isolated from the process model and all inputs from the surrounding model were identified. (Se Appendix IV)

The model of the absorption heat pumps was equipped with four factors

Factor S- Determine how many of the four Sanyo heat pumps currently in use (0, 5=2 heat pump are stopped). This factor was entered as how many of the heat pumps that were on, divided with 4.

Factor E- Determine how many of the two Entropies heat pumps currently in use (0, 5=1 heat pump are stopped).This factor was entered as how many of the heat pumps that were on, divided with 2.

Factor V- A factor that can correct for known overestimations of the COP in certain process cases. As this factor is an estimation done manually as the model is operated I put this factor to 1.

Factor F- Determine how much of the district heat flow that will be re circulated between the heat pumps.

The difference between the measured process data and the predictions from the model was divided with the measured value to get a relative error.

In order to see if there was any continuing trend in the errors i.e. that the model always predicts a too high or too low value, the frequency of the relative errors was plotted.

Table 1: Mean value and variance of the relative error

	Mean value of the relative errors (%)	Variance of the relative error
DH temperature	-0,0207	0,000522
Q Sanyo	-0,0157	0,0373
Q entropie	-0,003	0,0610
COP Sanyo	-0,0157	0,00538
COP Entropie	-0,0225	0,00572

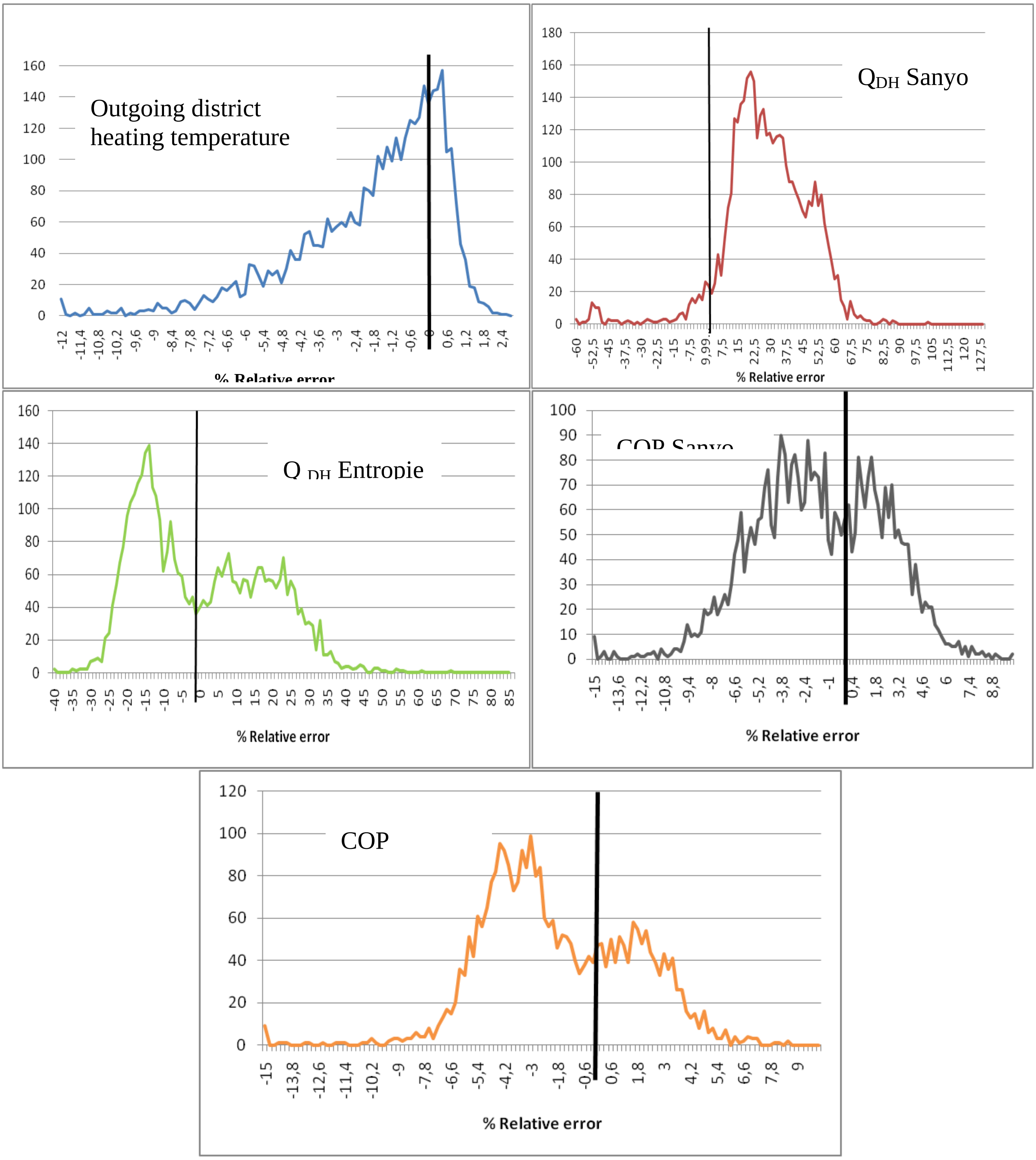


Figure 11: The frequency of the relative error

Based on figure 11 it can be noted that the relative errors are somewhat centered around a value on the right or left side of the zero value. A mean value of the relative errors has been calculated and is given in the table above (table 1). The variance of the relative errors has also been calculated as a measure of the distribution of the relative errors around the mean value. A high variance, meaning a high distribution of the errors, could mean that the result of the model is quite inaccurate even though the mean value is close to zero.

As the models errors are about 1-2 % of the measure process it can be concluded that the model are a god representation of the absorption heat pumps during the considered time span.

The relative errors of the district heating temperature have a peak near the zero, meaning that the model is a god prediction the outgoing district heating temperature in several of the process cases. However the distribution of the relative errors is quite broad and the prediction of the outgoing district heating temperature have been over estimated in many of the process cases.

The peak of the relative error on the calculated heat transferred to the district heating system shows, as can be seen in the figure 11 above, that the model over predicts and under predicts the heat transfer in the Sanyo and the Entropie heat pumps respectively with about 12-22 % .

It should also be noted that the process data have been collected as a time discrete sample where the process might not be in stable operation, and that the data was collected from measure equipment with unknown accuracy. This can explain some of the more extreme errors but since process data have be collected for an entire year the focus have to be on the bulk of errors.

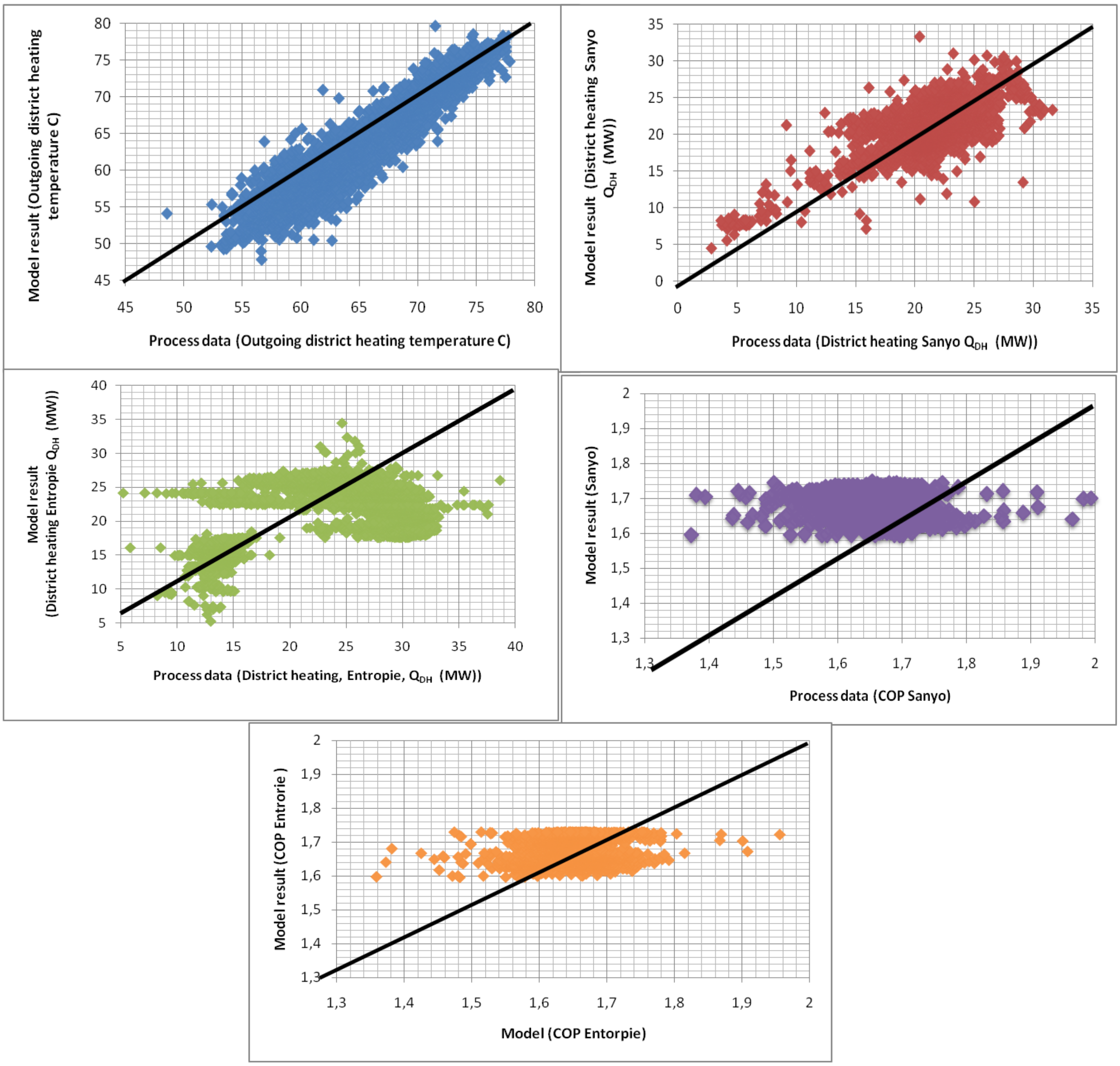


Figure 12: Scatter plot, Process data / Modeling results

The results from the Model-Renova were plotted as a function of the measured process data for the previous mentioned process parameters to form scatter plots (Figure 12).

All samples would end up at in alignment with the straight diagonal line (Y=X) if the model always predicted all properties to be equal to the corresponding measured value.

The scatter plots of the result of the tested made on the old model, during this period show that the Model-Renova of the absorption heat pumps predicts the temperature of the outgoing district heating water and the district heating effect fairly accurate.

However the heat transferred to the district heating water in the two Entropie heat pumps will, according to this plot, deviate from the measure value at higher heat transfers.

The estimations of the COP efficiency is more spread out in reality then the model is predicting for both the Sanyo and the Entropie heat pumps.

3.9.2 Advantages and disadvantages with the Model-Renova of the absorption heat pumps

The Model-Renova is a simplified representation of the reality as it basically represents the group of heat exchangers by one empirical correlation. The simplicity of this model makes it a robust representation of the reality and will, as it is based on analysis of real process data, incorporate all uncertainties in the process.

As the model basically only consist of a few main equations and some surroundings energy and mass balance equations, it can be modeled using only a limited computer size and can therefore easily be incorporated into the Model-Renova.

<u>Advantages</u>

- Simplicity
- God accuracy
- Small computer capacity needed
- Based on process data

Since the model is build up around process data, collected at discrete time points where the operation was assumed to be representative, there is an uncertainty concerning the range of validity. The model can therefore be expected to deviate from the reality in operations that differs considerable from the operation at the time when the process data was collected.

As the model represents the six heat pumps like two units there is no possibility to analyze how the individual heat pump operates and how changes and rebuilding in one or a few of the heat pumps affects the total operation in of the plant.

The exciting model simulates the operation when one or more of the heat pumps have been stopped by the previous mentioned factors (S and E). These factors only affect the heat transferred to the district heating network and do not take into account the reduced flow of district heating water to the heat pump units.

<u>Disadvantages:</u>

- Unknown range of validity
- Do not model each separate heat pump
- Hard to model when one or more of the heat pumps have been turned of

4 Creation of new the model

The absorption heat pump will, in reality, experience inputs from the surrounding system including temperature and mass flow of incoming district heating water and MKS water as well as steam data and flow. As the heat pump cools the MKS water and heats the district heating water, it will deliver “out data” to the surrounding system in form of temperatures of the outgoing district heating water, the MKS water and the condensate coming from the condensate after cooler.

The absorption heat pump can be seen as a network of heat exchangers assembled in such a way to achieve an indirect transfer of heat from one temperature level to another. The model is therefore created as separate heat exchangers with inputs as temperatures, pressures, mass flow rates and concentrations calculated in surrounding process models.

4.1 Mass balance and species balance

The mass balance in all components, except for the absorber and the generator, is trivial as the mass flow in to the component equals the flow out of the component at steady state conditions.

The mass flow of water vapor in to the absorber from the evaporator together with the mass flow of the strong solution coming from the generator will equal the mass flow of the weak solution leaving the absorber.

In the same way the mass flow of the weak solution entering the generator will equal the steam passing in to the condenser and the strong solution leaving in the bottom of the generator. The heat pump can therefore, with respect to the mass and species balance, be described by three streams seen in the figure below (figure 20).

The circulation pump mounted at the weak solution after the absorber will govern the mass flow of the weak solution based on the pump curve which is a function of the pressure difference between the low pressure and high pressure side. The pump curve is for now assumed to be fairly independent of the pressure, at the small pressure difference in which the absorption heat pump operates. The mass flow will be calculated from the commonly used value of the ratio between the weak solution mass flow and the steam leaving the generator 10.84 i.e. ($\dot{m_{weak}} = \dot{m}_{steam\ fromthe\ genertor} * 10{,}84$) (Herold, et al., 1996)

A-Water

B-Strong solution

C-weak solution

$$\dot{m}_A = \dot{m}_B + \dot{m}_C \quad (4.1.1)$$
$$\dot{m}_B * x_B = \dot{m}_C * x_C \quad (4.1.2)$$

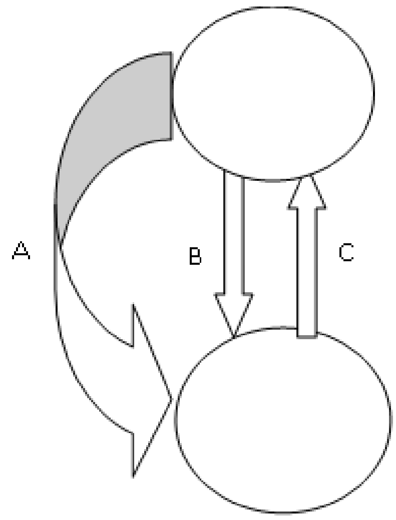


Figure 13: Mass balance

4.2 Energy balance

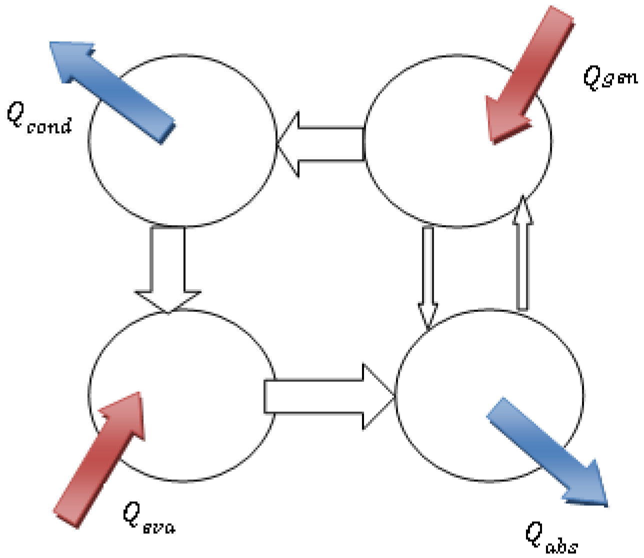
Energy into the heat pump in form of steam in to the generator and low temperature heat in to the evaporator must equal energy out from the absorber and condenser.

If the hot condensate, coming out from the generator, is cooled by after cooler heat exchange with a stream inside the heat pump (as is the case in the SANYO machines) this heat will have to be included as heat input along with the heat from the compensating steam. This is the case in the four Sanyo heat pumps considered here.

When considering each separate component the energy of streams into the component must be equal to the energy in the streams leaving the component regardless of the processes and reactions occurring inside the component.

The energy balance for each component is calculated as the enthalpy times the mass flow of the incoming streams minus the enthalpy times the mass flow of the outgoing which should streams add up to zero at steady state.

The heat of solution is included in the empirical formulations of the enthalpy of the water LiBr solution and is therefore indirect considered in the model (se absorption and desorption 3.4.)



$$Q_{eva} + Q_{gen} = Q_{cond} + Q_{abs} \quad (4.2.1)$$

Figure 14: Energy balance

4.3 Calculation procedure

The model is composed by separate calculation loops for each separate heat exchanger where one component ingoing properties i.e. temperature, are another components calculated outgoing results.

The equation system formed by the mass and energy balances and the property correlations is stated in the appendix VI and VII.

The equation system is compared with the equations found to be as base for the model created by Keith described above. (*[6] Herold, Keith 1996)*

The inputs to each component have been collected in the square located above the components calculations and the outputs are collected in a square bellow the calculations see figure 18.

This will become an iterative procedure and due to the fact that the model is sensitive to rapid changes the inputs will be connected to the calculated output from another component as:

$$Y_{i+1} = Y_i + \frac{\left(Y_{calculated\ (i)} - Y_i\right)}{\left(\frac{Faktor}{\left|Y_{calculated\ (i)} - Y_i\ \right|}\right)} \quad (4.3.1))$$

The factor that the difference between the input and the calculated value is divided with determines the speed of the iteration and will be set based on the order of magnitude on the input value. To make the iteration take smaller steps as the error decreases and the calculations approaches the right value, this speed factor is divided with absolute value of the difference between the input and the calculated value.

The calculation in the absorber the generator and the evaporator is an iterative procedure base on the logarithmic mean temperature difference described above.

The calculation in the absorber and the generator begins with a guess the temperature of the weak solution (T_4) and the strong solution outgoing (T_{11}) stream respectively. Energy and mass balances as well as calculations of concentrations and pressures can then be made based on the given guessed temperatures. The logarithmic mean temperature method is then used to recalculate the previous guessed temperature and a comparison is made between the guess and the calculated temperature, a new guess is then made based on the calculated temperature.

The new guess is made as (used in the iterations):

$$Y_{i+1} = Y_i + \frac{\left(Y_{calculated\ (i)} - Y_i\right)}{Faktor} \quad (4.3.2))$$

The iteration will be finished when the calculated temperature equals or almost the guess temperature.

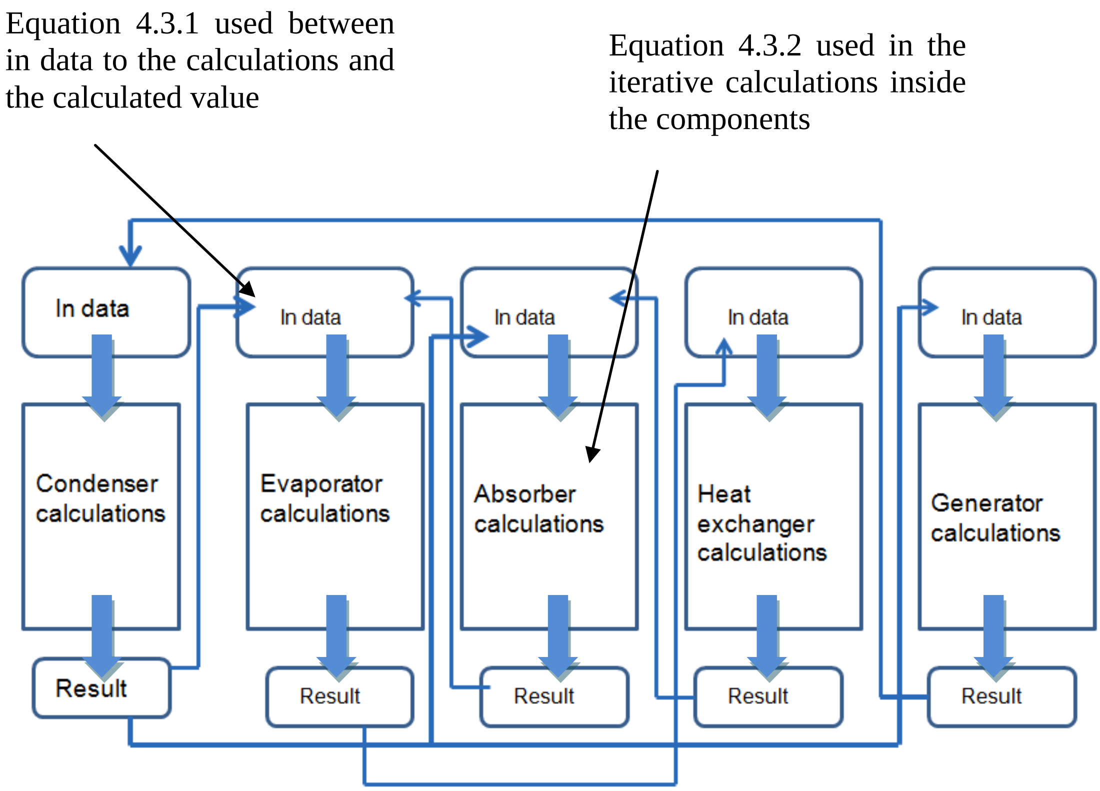


Figure 15 : Calculation procedure

The guess made in the evaporator will be on the low pressure (Plow). This guess enables calculations of the temperature on the outgoing MKS water and the saturation temperature within the evaporator. From these temperatures and the logarithmic mean temperature difference method it is possible to calculate a saturation temperature and therefore also the low pressure. The calculated pressure is then used in the same way as in the absorber and generator to create a new corrected guess of the pressure.

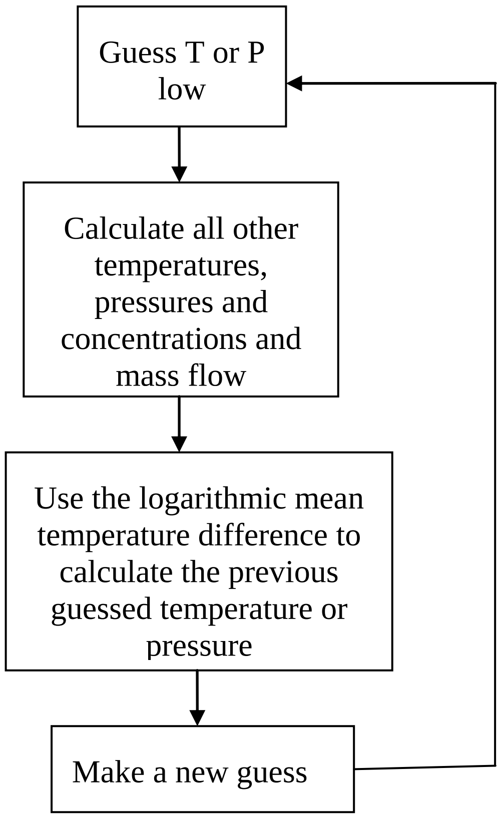


Figure 16: Logarithmic mean temperature, calculation procedure

The calculations in the condenser and the heat exchangers will instead be based on the NTU method described in a section above (3.6.2), where the heat transferred was determined. This method do not require any iteration and will therefore simplify the calculations in comparison to the Logarithmic mean temperature method (LMTD).

The remaining temperatures and mass flows are then calculated from the calculated heat transfer and the input temperature.

As the steam entering the condenser will have a temperature above saturation, upper part the off the condenser is used to bring the steam back to saturation. The condenser is therefore calculated as two heat exchangers where the first part is calculated through a simple energy balance and the rest through the NTU method.

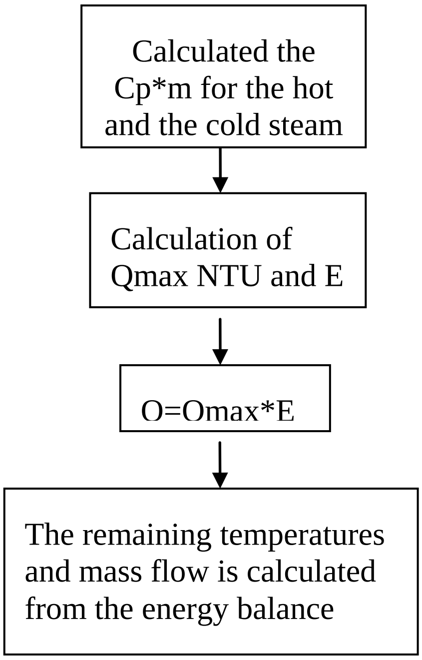


Figure 17: NTU method, Calculation procedure

One avantage with the NTU method is, as mentioned above, that it do not require a iterative calculation procedure as in the logarithmic mean temperature method and would therefore be ideal to use in the calculation of all components. The fact that the absorber and the generator include three flows and that the content of these flows change through the absorption and desorption process make an assumption of a constant specific heat unreliable.

The choice to use the logarithmic mean temperature method on the evaporator calculations is justified by the fact the only on temperature (the temperature of the incoming MKS water) is know at the start of the iteration, which would in any case lead to an iterative procedure since the NTU method requires two known temperatures.

4.4 Properties

Properties of enthalpy, pressure, density and specific heat are determined from polynomial found in the article by J. Pátek, JKlomfar.

These formulations of the properties cover a temperature range from 273K to 500K and a weight concentration from 40 to 70. (*[15] Klomfar J Patek 2006).*

4.4.1 Enthalpy

To calculate the enthalpy of the water/LiBr solution, the temperature and concentration needs be given. The summation part of the calculations is done and the calculated enthalpy in kJ/kg is then easily found. Equation (3.9.1.1)

In cases when the enthalpy and concentration of the lithium bromide/ water solution is known and the temperature is wanted this polynomial must be reversed.

An iterative procedure is then used starting with a guess of the temperature.

This reversing of this polynomial that includes as much as 41 terms in the exponential will result in some issue with the stability of the calculations. These instabilities can propagate through the calculations and create problem within the entire model.

4.4.2 Pressure

The polynomial to calculate the saturation pressure given the temperature and concentration is similar to the one used to calculate the enthalpy but will only have 8 terms in the summation equation (3.9.2.1).

This procedure is done to calculate the high pressure within the generator and condenser from the temperature of the strong solution (T_{11}) and concentration.

This polynomial can be reversed in order to calculate the saturation temperature or the concentration at saturation given the pressure and the concentration or temperature respectively.

The saturation temperature in the weak solution coming to the generator is calculated from the pressure and the concentration.

The calculation initiates with a guess of the saturation temperature. The saturation temperature for water at the given pressure is than calculated (θ). The summation is than calculated. The summation plus the saturation temperature for water (θ) equals the requested saturation temperature and a new guess is made.

The weak solution leaving the absorber will be in a state of liquid saturation and given that the temperature and pressure is know, the concentration can be calculated (x_{10}).

The calculation will once again be an iterative procedure, starting with the calculation of the saturation temperature (θ) and a guess of the molar concentration. The difference between the saturation temperature and the temperature of the solution will be the summation. The concentration is solved out of the summation by abstraction of one form the exponent over the concentration. The calculated concentration is compared with the initial guess and a new guess is made.

4.4.3 Specific heat and density

The calculations of the specific heat and the density of a solution is a straight forward calculation where the temperature and concentration is entered to the polynomial resulting in values for the specific heat and the density (equation (3.9.3.1) and (3.9.3.2)).

4.5 Heat transfer coefficients

Even though the design of the two absorption heat pump brands (Sanyo and Entropie) differ, the process inside the different heat exchangers can be assumed to be similar and the convection coefficient can therefore be calculated based on the same equations.

The convection coefficients in the tubes of the absorber, evaporator and the condenser can be calculated based on the formulation of the convection coefficient for water flow inside a tube (equation 12.17*) ([16] Sinnot R.K 2005)*

$$h = \frac{4200(1.35 + 0.02 * T) * u^{0.8}}{d^{0.2}} \quad (4.3.1)$$

These coefficients in the model have been calculated based for the flow inside the condenser, the evaporator and the absorber.

The condensation of the driving steam in the generator tubes is assumed to be film condensation occurring on the inside of the tube walls and the convection coefficient has been calculated based on the expression for film condensation in horizontal tubes. (Equation 10.47) (*[14] Incropera, 2005*).

$$h = 0.555 \left[\frac{g\ \rho_L\ (\rho_L - \rho_v) * {k_L}^3 h'_{fg}}{\mu_L (T_{sat} - T_s) D}\right]$$

$$h'_{fg} = h_{fg} + \frac{3}{8} Cp * (T_{sat} - T_s)$$

T_s= surface temperature (assumed to the equal to the bulk temperature)

The evaporated water entering the condenser will form a smooth flow of condensate on the outside of the tubes and the convection coefficient can then be calculated by an empirical equation using an equation based on Kern's method (Equation 12,50)

([16] Sinnot R.K 2005)

$$h = 0.95 * k_L \left[\frac{\rho_L(\rho_L - \rho_v) g}{\mu_L \Gamma_\mathrm{h}}\right]^{1/3} * \quad N_r^{-1/6} (4.3.2)$$

$$N_r = number\ of\ tubes\ inthe\ center\ row$$

$$\Gamma_\mathrm{h} = \frac{\dot{m}}{L * N}\ (kg/m\ s) \qquad (4.3.3)$$

As the water enters the evaporator and it is sprayed over the tube bundle and form a film around the outside out the tubes where the water is evaporated before it leaves the tube side as steam. This process is called falling film evaporation. The convection coefficients for this falling film evaporation in the evaporator will be guessed values.

The convection coefficient outside the tubes in the pool boiling process in the generator and the film absorption in the absorber are more complex processes due to the change in concentration and the simultaneous mass and heat transfer. The convection coefficient outside the tube in the generator and in the absorber will therefore be based on reasonable guesses of these coefficients. The same is done with the heat transfer coefficient in the solution heat exchanger and the condensate after cooler in both the Sanyo and Enrtopie heat pumps even thought the heat exchangers in the Entropie heat pumps are plate heat exchangers.

Due to the absence of oxygen inside the heat pumps the risk for corrosion on the tube walls is almost negligible and the clean water on both the tube side and shell side of the machine makes the fouling coefficients almost infinite high.

The fouling coefficients (*hf* in Equation 3.5.2 *)* have therefore been assigned a value of about 100000 W/ (K m) on the tube side and 50000 W/(K m) on the shell side.

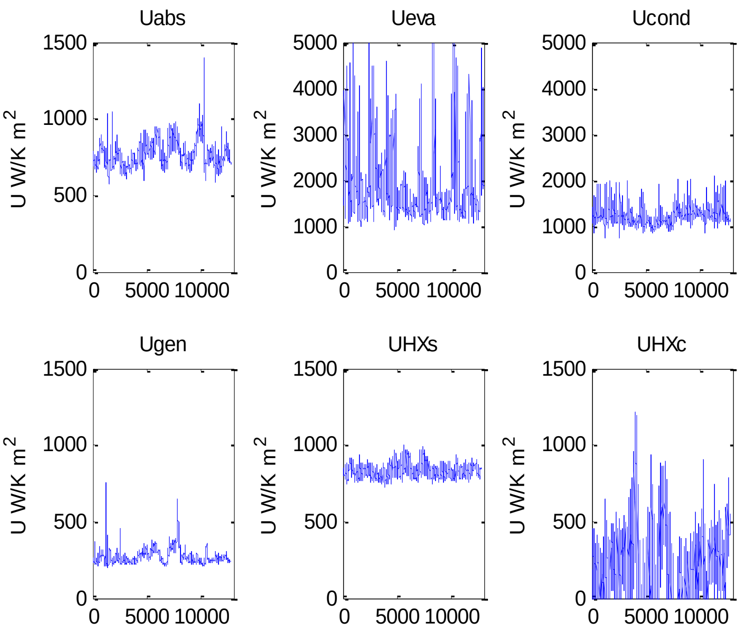
In order to be able to make these reasonable guesses of the heat transfer coefficients all available measured temperatures and pressures have been collected. The heat transfer coefficients inside the heat exchanger have then been calculated in the mathematical software Matlab based on these measured values and some complementing guess of the mass flow and temperatures inside the machines.

In the newer and more modern Entropie heat pumps many of the temperatures and pressures have been measured and process data have been collected during a time period 2010-02-23 to 2010-03-04 as discrete values every minute. The rest of needed values were guessed.

The results of these calculations are displayed in the figure bellow which shows that the heat transfer is fairly constant in the absorber, condenser and the generator but will vary more in the evaporator and the heat exchanger.

A mean value of all the heat transfer coefficient was calculated to give a base for the above motioned heat transfer coefficients.

Changes of the guessed value will only slightly affect the calculated heat transfer coefficients.



$Uabs_{\text{mean value}} = 766.2974\ [W/Km^2]$

$Ueva_{\text{mean value}} = 1.84 \cdot 10^3\ [W/Km^2]$

$Ucond_{\text{mean value}} = 1.02 \cdot 10^3\ [W/Km^2]$

$Ugen_{\text{mean value}} = 267.86 [W/Km^2]$

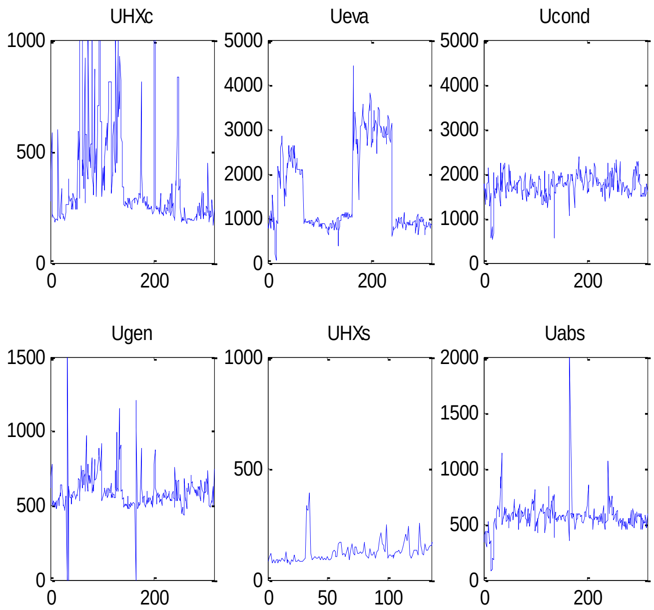
$UHXs_{\text{mean value}} = 830.5\ [W/Km^2]$

$UHXcond_{\text{mean value}} = 141.6\ [W/Km^2]$

Figure 18: The heat transfer coefficients, Entropie

The Sanyo heat pump machines have been equipped with a very limited number of measure points and the calculations of the heat transfer will therefore include several guesses of the temperatures, mass flow and the heat transferred in the solution heat exchanger and condensate heat exchangers. This means that the results of these calculations will only be a very rough estimation of the heat transfer coefficient and should not be taken as a definite value of the heat transfer coefficient.

Process data have been collected for the period 2009-01-01 to 2009-12-31 as a discrete value every day.



UHXcond $_{\text{mean value}}$=388.47[W/Km^2]

Ueva$_{\text{mean value}}$ = 1.6*10^3 [W/Km^2]

Ucond$_{\text{mean value}}$ = 1.73*10^3 [W/Km^2]

Ugen$_{\text{mean value}}$ =595.41[W/Km^2]

UHXs$_{\text{mean value}}$ =108.35 [W/Km^2]

Uabs$_{\text{meanvalue}}$ =564.04[W/Km^2]

Figure 19: Heat transfer coefficient, Sanyo 634

4.6 The heat transfer efficiency parameter (ε)

The dimensionless heat transfer efficiency parameter is calculated based on the NTU and the heating rates for the two fluids.

The efficiency parameter ε used in the condenser is given for any heat exchanger where the capacity rate for one of the fluid approaches infinite, as is the case in evaporators and condensers *([14] Incropera, 2005).*

The solution heat exchanger in the Sanyo heat pump models is designed with three pass on both the tube and the shell side. Since the number of passes is equal on both the shell and the tube side this heat exchanger can be seen as a three smaller one pass heat exchangers and the value of the heat transfer efficiency for that type of heat exchanger is therefore used in these calculations. *([14] Incropera, 2005*)

The condensate heat exchanger in the Sanyo machines is a two pass heat exchanger and the heat transfer efficiency correlation is given for this particular heat exchange design) (*[14] Incropera, 2005).*

4.7 Excel iteration

When a cell in excel is used to calculate the value in another cell, which in turn was used to calculate the value in the first cell a circle reference or iteration is created.

Excel will than take the calculated value to make it the new input until the calculated value equals the input.

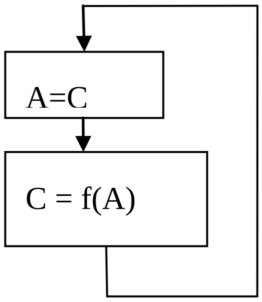


Figure 20: Excel iteration

The iteration in excel finish when no elements in the iteration changes more than a predetermined maximum value or when the number of iterations exceeds another predetermined value.

4.8 Assumptions

Process Assumptions:

- No pressure losses between the evaporator and the absorber or between the generator and the condenser.
- No pressure losses in the heat exchanger tubes
- Isentropic expansion in the expansion valves.

State assumptions:

- The condensate after the condenser (stream 17 in the Sanyo model) is considered to be saturated liquid.
- The condensate after the generator (stream 13 in the Sanyo model) is assumed to be saturated liquid due to the steam trap.
- The steam entering the absorber (stream 6 in the Sanyo model) from the evaporator is assumed to be saturated steam.
- The steam leaving the generator (stream 15 in the Sanyo model) will have a temperature that is equal to the saturation for the strong solution entering the generator (stream 10 in the Sanyo model).

Disregard:

- The influence off additives to the LiBr solution of the thermodynamic properties
- Heat losses to the surroundings (adiabatic)
- The influence of the circulation pump work
- The influence of the pressure difference on the mass flow through the pump (pump curve)

4.9 Accuracy of the iteration

The accuracy of the iterations in excel is determined by the entered value of the maximal change of any unit in the iteration.

As the calculations include both calculations of concentrations in centesimal and effects in thousands, the largest change is in the larger numbers i.e. the heat transferred.

A change of 0.1 in the concentration would make a big difference whereas the same change in the heat transferred in the order of 10^3 kW would make a neglect able difference. The entered value of the maximal acceptable difference should therefore be determined after considering that this is probably the maximum of the change in the highest value in the iteration.

As the calculations include properties in a wide range of order of magnitudes, the largest numbers have been scaled down i.e. the heat transfer and enthalpies are given in kW and kJ/kg. The properties with the smallest order of magnitudes i.e. the concentrations were given in percentage instead of in mass fractions. The scaling of the concentrations had a significant effect on the stability of the model since the concentrations is essential for the absorption and generator.

4.9.1 Iteration boundaries

In the beginning of the iteration some of the guessed and calculated temperatures, pressures and concentrations will start to oscillate to high and low value. This would cause the model to fail and the guessed values will therefore have to be restricted.

The new guesses off the temperature in the absorber, generator and iterations to calculate the temperatures is restricted to 273<T<500.

Example: A=IF((calculated value)<273;273;(calculated value))

B=IF((calculated value)>500:500;A)

New guess=B

4.9.2 IF ERROR (OM FEL)

If iteration fails it can only be started again by entering a new guess manually. Some of these iterations have instead been equipped with an if error functions so that the guess will take on a specific value and the iteration can be restated as soon as all other input to the iteration is recovered.

Example:
T_4=OMFEL (T_4; 320)

The speed of the iteration can sometimes be too fast for excel with failure as a result. To prevent this pervious mentioned function if error OM FEL is used to give a starting value to the iteration.

5 Adaption of the model to the individual heat pumps

The model is made as a theoretical representation of a heat pump with the same design and geometrical dimensions as the Sanyo and Entropies heat pump types.

The heat transfer coefficients in the heat exchangers are the result of guesses or based on some empirical calculations. The actual heat pumps will always deviate some from these theoretical calculations due to inaccuracy in equations and properties and also due to small differences in the construction and deteriorations due to ageing.

In order to adapt the model to the real heat pumps the heat transfer coefficient in each heat exchanger unit is multiplied with factor. The heat pump model can then be tuned in to fit collected process data for each of the six heat pumps.

The heat transfer coefficients will in these calculations include the entire heat transfer including correction factors when the flow deviates from a countercurrent flow, fouling and other progression affecting the heat transfer.

The model of the Sanyo heat pumps has been adapted to represent one of the four Sanyo heat pumps which were done by entering in data from some process cases and by a trial and error procedure find the corrected factors.

The heat transfer coefficient will after being corrected to fit the real heat pump deviate for both the calculated value and sometimes also from what you can expect. This can be explained by that fact that the heat transfer coefficient will, in this model, include the entire heat transfer like the correction factor for the flows that depart from a true counter current flow and also other inaccuracies in the calculations.

6 Results

6.1 Model: Sanyo

The model of the Sanyo heat pumps has been constructed and all components are connected through their inputs and outputs.

The model has, as mentioned above, been adapted to represent one of the four Sanyo heat pumps (Product name: Sanyo 634).

The model operates in that sense that you can enter values of the in data and after numerous iterations gets some resulting out data. The control of the heat pumps have also been included and is operating with the exception of the fact that the reference value on the liquid level in the evaporator needs to be entered as a medium concentration of the lithium bromide solution.

In order to evaluate the created model against the previous model some process data from 8 discrete moments in time was collected and inserted in both models.

The time samples have been taken during the last part of 2009 (2009-07-03 – 2009-12-09) at times where the incoming temperature and flow of district heating water, and the outgoing temperature of the cooled water where at its maximum and minimum as well as two randomly collected times where the operation can be assumed to be normal.

As the Model-Renova of the absorption heat pumps represents the four heat pumps as one unit and the new model represents each heat pump individually the results from the old model is scaled down to represent one of the heat pumps. The comparison will therefore not be completely accurate since the new model is adapted to fit one of the heat pumps but will be compared with a model that represents all four heat pumps.

The result of both the new and the Model-Renova at these eight times has been scaled against the measure process value in order to visualize the comparison (figure 27 and 28).

It can be noted that the accuracy of the new model is fairly good at the five of considered operation times but will deviate from the reality at three of these times.

This deviation can probably be improved by a better adaption of the heat pump model to the individual heat pumps or be explained by the fact that the considered measure point where taken at the highest and lowest temperate and flow into the absorption heat pump.

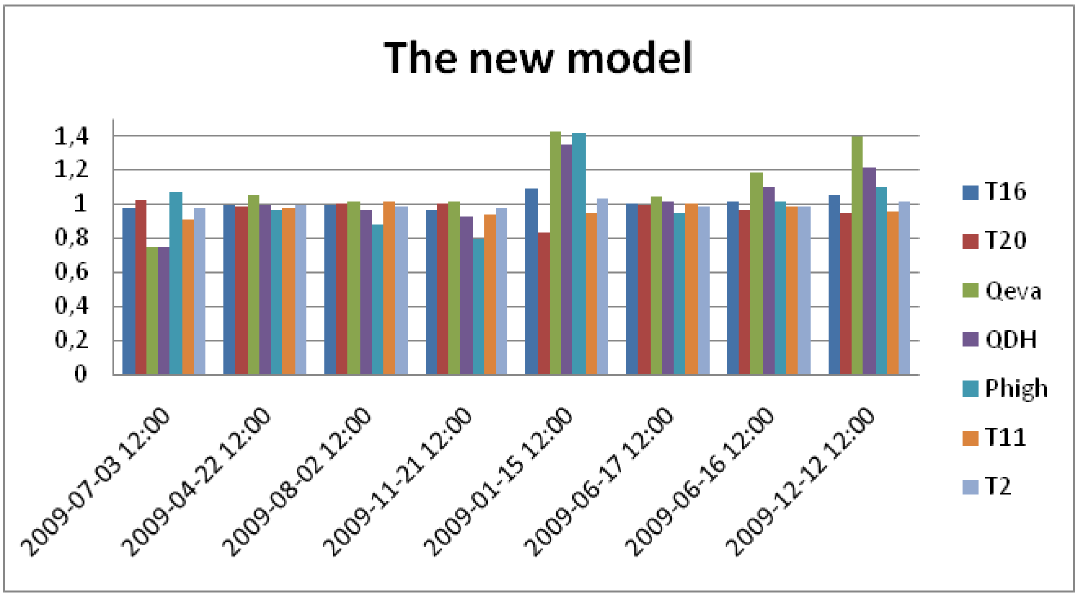


Figure 21: Result of the new model scaled against the measured value

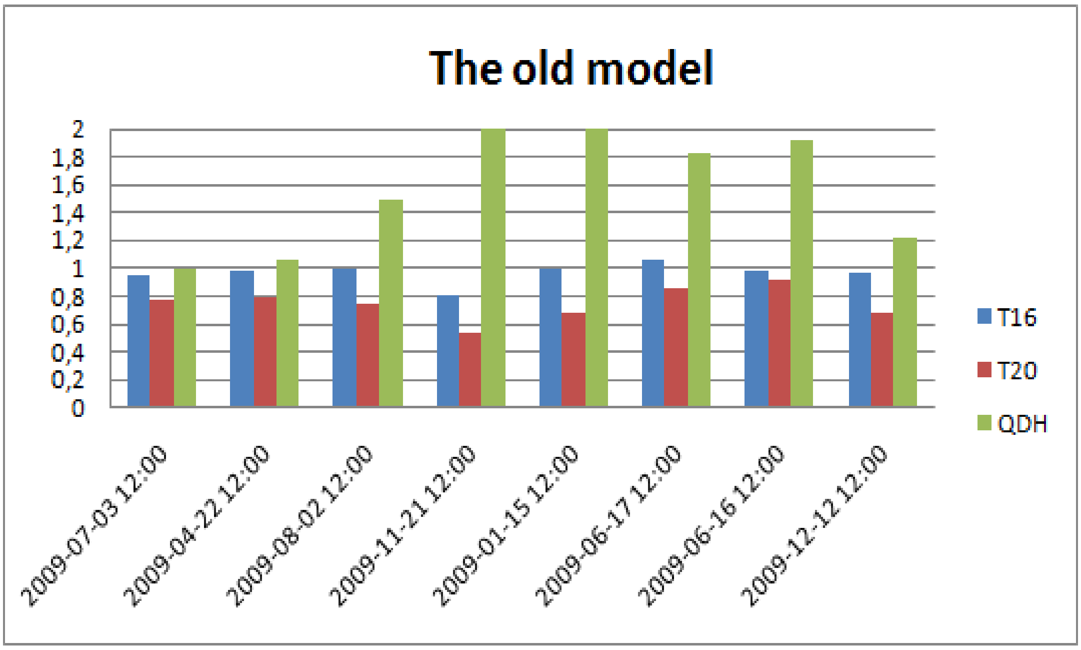


Figure 22: Result of the old model scaled against the measured value

Table 2: Test times

Date	
2009-07-03 12:00	Low incoming district heating temperature T1
2009-04-22 12:00	High incoming district heating temperature T1
2009-08-02 12:00	Low incoming cooling water temperature T19
2009-11-21 12:00	High incoming cooling water temperature T20
2009-01-15 12:00	Low district heating flow
2009-06-17 12:00	High district heating flow
2009-06-16 12:00	Medium conditions
2009-12-12 12:00	Medium conditions

The test result of the new model has been represented in a scatter plot (figure 23). This plot will show that some of the measure parameters are fairly god prediction with the measured process data for the few measure point evaluated here around .The accuracy seems also to be independent of the measured data, in that sense that the error will not increase or decrease as the process conditions change. It can also be noted that the model seams to give an accurate estimation of the high pressure at the high end of the measured values.

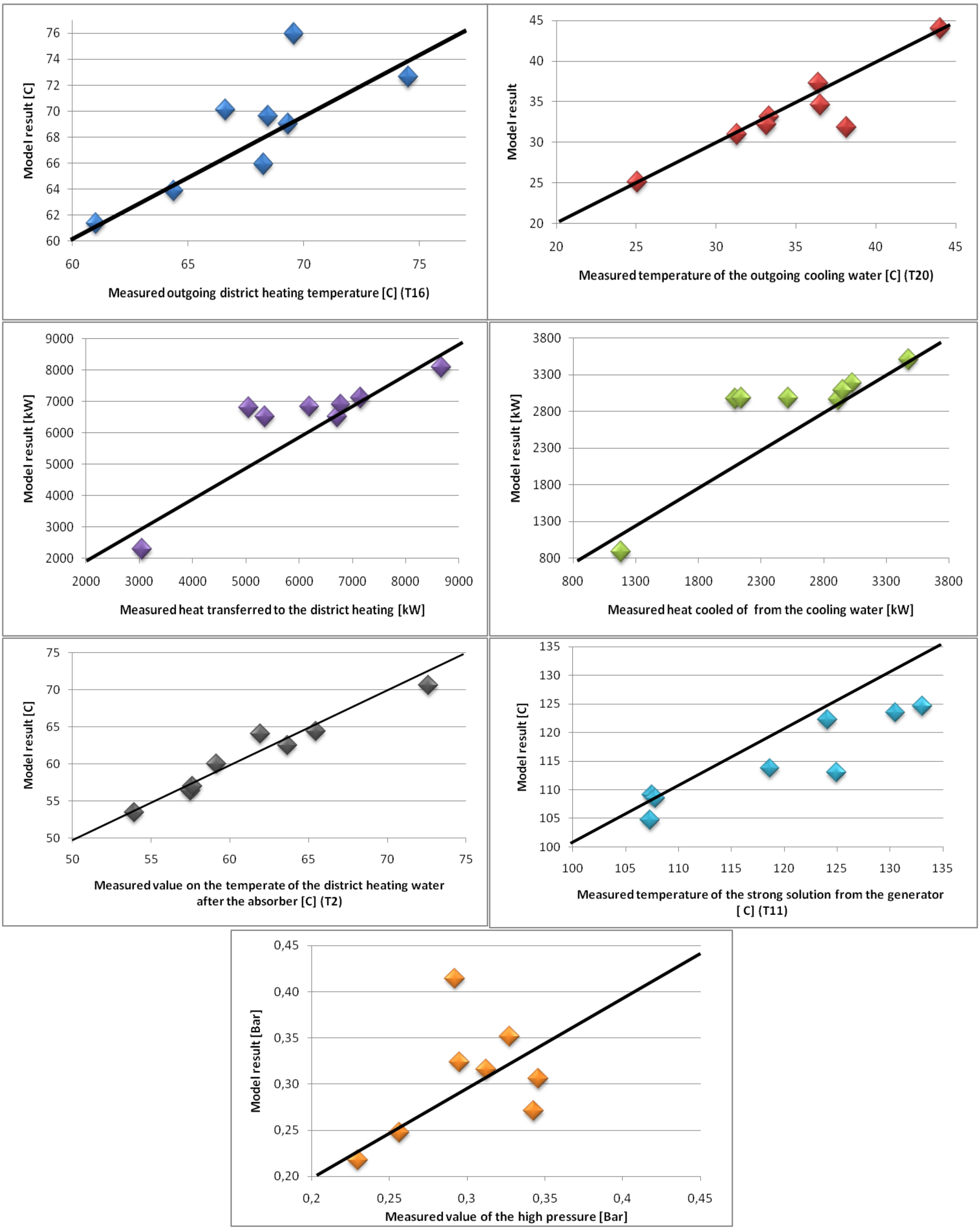


Figure 23: Scatter plot of the new model result

6.2 Model Entropie

The above mentioned problem with the modeling and the time constraint of this project resulted in that the model of the Entropie heat pumps do function to some extent but lacks a complete representation of the controls system and further improvements in the adaption to the real absorption heat pumps. But as in the models of the Sanyo heat pump, a set off outdate can be received by changing the models in data. More work needs to be done in the representation of these heat pumps with regard to the modeling of the advanced heat transfer control system.

6.3 Modeling issues and discussion

The iteration will continue until the total system of equations goes against a point where it converges. This would be when all guesses in the iterations equal the calculated values and when all inputs to each component equal its corresponding calculated output.

As the equations and correlations are interdependent on each other, a small change in the result of one equation can lead to a big change in another equation and vice versa. A small inaccuracy in one equation can result in much bigger inaccuracy in the rest of the model.

A difficulty with the model is to locate the point where the equation system formed by all the equations and variables, have been solved. The iterative procedure of the model operates by changing the values until the model finds the point where it converges. The changes of these values are governed by the connection between the iterated values and the factor that controls the speed of the iteration. The speed of the iteration and the steps taken can make the model miss the point where the correct value has been calculated. It is therefore essential to the model that the speed factor (Equation 3.7.1) used in the iteration is equal for the different units of the calculated properties.

If this factor is set to a to small value all the iteration will go too fast and the model becomes unstable as the differences between the guesses and the calculated values oscillates from positive to negative higher and higher. A to high factor will instead lead to that the iterations goes to slow and that the point of converged isn't found before the excel iterations stop due so small change of any value in the iteration that it will be bellow the inserted maximum allowed change.

In order to get the iteration to take smaller steps when it approaches the right value the difference between the guess and the calculated value have been multiplied with the absolute value of the difference between those two (see 3,9 calculation procedure). When the difference decreases the steps will be smaller and the risk that point where should converge will be passed.

The connection between the different component of the absorption heat pump as well as the calculations inside the absorber, generator and evaporator includes iteration loop.

The sequential iterations in Excel mean that the equations are calculated in cycles and the cells changes value each iteration cycle. This will be a problem when a value used is given from another components calculations or from inside an iteration loop. A calculation can then use values calculated from different iteration cycles, which mean that the result of this equation will not be equal to the result of the same equation calculated from value for the current calculation cycle and the exact balance in the equation system will therefore be more difficult to find.

The polynomial function of the enthalpy for the lithium bromide solution as a function of the concentration and the temperature includes 31 terms in order to increase the accuracy. When this function is inverted to get the temperature as a function of the enthalpy and the concentration instead these many terms bring about a problem with instability and in accuracy of the calculations.

The same phenomena is presented when finding the concentration as a function of the saturation pressure and temperature, and to find the saturation temperature before the generator but the pressure polynomial only includes 8 terms.

This instability can be a significant contribution to the instability and problems with converge in the models.

6.4 Excel

The complexity of the absorption heat pump have resulted in calculations and iteration procedures that is so demanding that Excel endlessly stop functioning (Not responding).

As Excel stop function the model freezes for a period of time unless Excel is being shut down and restarted. Excel will then create a restore a copy of the model as it was just before it stopped functioning but as the calculations in the model goes in loops this restored copy will use a defaulted value as an input to the calculation loop.

Excel do this as a result of that the computer can’t handle the model when it is constructed in excel and not because the excel program being too weak and the model might therefore work better in a very strong computer.

7 Conclusions

In accordance with the aim stated in the being of this report a thermodynamic representation of the absorption heat pumps have been developed.

The model will contains all components in the heat pump and therefore enables the possibility to alter the model to simulate changes to heat pump as well as deterioration of the heat transfer.

A disadvantage in the old model was that it depended on a curve fit to collected process data resulting in an uncertain validity range. The model created during this master thesis is also in some aspect using process data to estimate the heat transfer coefficient that adapts the model to the real heat pumps. But since the heat transfer coefficient can be assumed to differ less with different process conditions than the empirical relations used in the old model to represent the entire heat pump, the new model is an improvement.

The above mentioned problems experienced with the modeling of the absorption heat pumps might reduce the practical use of the models, at least in their existing condition. The models and the work done could therefore be considered as a base for more investigation in a more extensive model.

8 Further work

The constructed models should be seen as a base for further improvements and work that can improve the representation of the actual heat pumps and increase the practical use of the model.

The model of the Sanyo heat pumps have been adapted to one of the four heat pumps meaning that more work needs to be done in order to adapt the model to the three remaining heat pumps. Left to investigate, after the models have been adapted to the separate heat pumps, is how the heat pump works in combination with the rest of the processes and also gets a deeper understanding which parameters influence the process.

In order to make it possible to incorporate the six models of the absorption heat pumps into the model of the entire process, the problems with instabilities and coverage in the model needs to be improved, perhaps by changes to the iterative procedure and the speed of iteration.

The model of the Entropie heat pumps control system heat pumps has not been completely finished during this master thesis, and will therefore need additional work. The theoretical model will then have to be adapted to the two real Entropie heat pumps.

The control system controlling the liquid level in the evaporator in the model of the Sanyo heat pumps has for the moment been molded by controlling the medium concentration of lithium bromide between the weak and the strong side. Further work needs to be done in order to find the relation between the medium concentration and the liquid level in the evaporator.

More work can also be done to make better representations of the heat transfer coefficients in order to replace the guessed values. This will include studies in how the simulations mass and heat transfer within the absorber and generator affects the heat transfer.

The pump curve, has up until this point, been modeled by a constant value on the mass flow of the weak solution. Improvements can be made by representing the pump curve as an equation describing the dependence of the pressure difference on the mass flow through the pump.

The many problems with the created model make it appropriate to evaluate if it is better to uses a more simplified model or use the Model-Renova.

The Model-Renova could be improved by using the same method that was used to find the empirical formulations in the Model-Renova for each separate heat pump.

Another approach would be to use the equation system that constitutes the base for the created model and solve it by use of a computerized nonlinear solver e.g. in MATLAB.

It can also be concluded that the test of the Model-Renova showed that it differed some from the measured process data but the errors was fairly centered in some of the evaluated parameters and the model could therefore perhaps be slightly adapted to the real process data.

The limited possibility to individualize the model to the separate heat pumps in order to model changes in the separate heat pumps remains an issue in the old model.

References

Web sources

[1] EIA, U.S. Energy Infromation Administration, Independent statistics and analysis. *Inernational energy outlook 2009.*
Collected: 2010-04-14.
http://www.eia.doe.gov/oiaf/ieo/highlights.html

[2] Avfall Sverige Avfallsförbränning.
Collected : 2010-04-17.
http://www.avfallsverige.se/m4n?oid=849&_locale=1

[4] Renova AB. *Homepage Avfallskraftvärmeverket* . Götebrog;
Collected: 2010-04-14
http://www.renova.se/t/Page____302.aspx.

[7] Chemland, 21 Lithium broimide
Collected 2010-04-03 :
http://chemicalland21.com/industrialchem/inorganic/LITHIUM%20BROMIDE.htm,

[17] Engineering fundamentals (efunda) *Stainless steel stainless steel AISI 403*.
Collected : 2010-04-05.
http://www.efunda.com/materials/alloys/stainless_steels/show_stainless.cfm?ID=AISI_Type_403&show_prop=all&Page_Title=AISI%20Type%20403

[18] Theengineering Tool box. *Thermal conductivity*
Collected: 2010-04-05
http://www.engineeringtoolbox.com/thermal-conductivity-d_429.html

[19] Copper Development Association *Cu-Ni Alloys - Resistance to Corrosion and Biofouling*
Collected: 2010-04-18
http://www.copper.org/applications/cuni/txt_resistance_to_corrosion.html#mechanical

Manufacturing files

[8] SANYO. Sanyo Manufacturing files

[9] Entropie. Entropie Mannufacturing files 2002

Reports and journal articles

[13] Grossman, Greshon and Zaltash, Abdi.
ABSIM -modular simultion of advanced absosrption systems. Year 2001.
Faculty of mechanical engineering Technion-Israel institute of Technology Haifa, Israel
Energy division Oak Ridge National Laboratory Oak Ridge USA
International Journal of Refigeration 24. Page 531-543.

[15] Klomfar, J Patek.
A computationally effective formulation of the thermodynamic properties of LiBr-H2O solutions from 273 to 500 K over full composition range. Year 2006
Institute of Thermomechanics, Academy of Sciences of the Czech Republic,
Dolejškova 5, CZ 182 00 Prague 8, Czech Republic
International Journal of refigeration 29. Page 566-578.

[10] X Qin[1], L Chen[1], F Sun[1], L He[1].
Modeling of a real absorption heat pump cycle with a generlized heat transfer law and it's performance. Year 2007
Republic of China
Postgraduate School, Naval University of Engineering, Wuhan, Hubei, People's
Proceedings of the Institution of Mechanical Engineers, Part A : Journal of Power and Energy
Volume 221 Number 7/2007 Pages: 907-016

[3] Renova .
Miljörapport 2009 för avfallskraftvärmeverket och sorteringsanläggningen inklusive återvinningscentralen vid Sävenäs, Page 3 . 2009.
http://www.renova.se/upload/Pdf/Miljorapport_Savenas_2009_textdel.pdf

[5] Renova AB. (Pamphlet)
The Waste to energy plant in Göteborg, Sweden. s.l. :
http://www.renova.se/upload/Pdf/avfallskraftvarmeverket_i_savenas.pdf,.

[11] Sencan, Arzu, Yakut, Kermal A and Kalogirou, Soteris A.
Exergy analysis of Lithium bromide/water absorption systems. 2004
Cyprus : Department of mechanical education Technical education faculty Süleyman Demirel University,
Renewable energy 30 Pages 646-657

[12] Wolf J. Koehler and Warren E. Ibele and Joseph Soltes and Edgar R. Winter
Availability simulation of a lithium bromide absorption heat pump 1988.
Heat Transfer Laboratory, Mechanical Engineering Department, University of Minnesota, U.S.A.
Lehrstuhl C für Thermodynamik, Technische Universität München, München, West GermanyMünchen, West Germany : Elsevier Science Ltd. ,
Heat recovery systems and CHP, Volume 8, issue 2 1988 Pages: 157-171

BOOKS

[6] Herold, Keith E, Radermacher, Reinhard and Klein, Sanford A.
Absorption Chillers and heat pumps 1996 .
Boca Raton, New York, London Tokyo : CRC Press, 1996.
ISBN 0-8493-9427-9.

[14] Incropera, DeWitt, Bergman, and Lavine
Fundamentals of heat and mass transfer (page 675-678) Year 2005
Six edition. University of California, Los Angeles : John Wiley & Sons.Inc.

[16] Sinnott, R.K. 2005
Chemical engineering design Coulson and Rickardsom's Design text book
Heat-transfer equipment. Chapter: 12.5 Shell and tube heat exchangers: construction details (Pages 634-799)
Six edition
Butterworth Heinemann ISBN 0 7506 41428

Appendix I: System boundaries, Heat pump

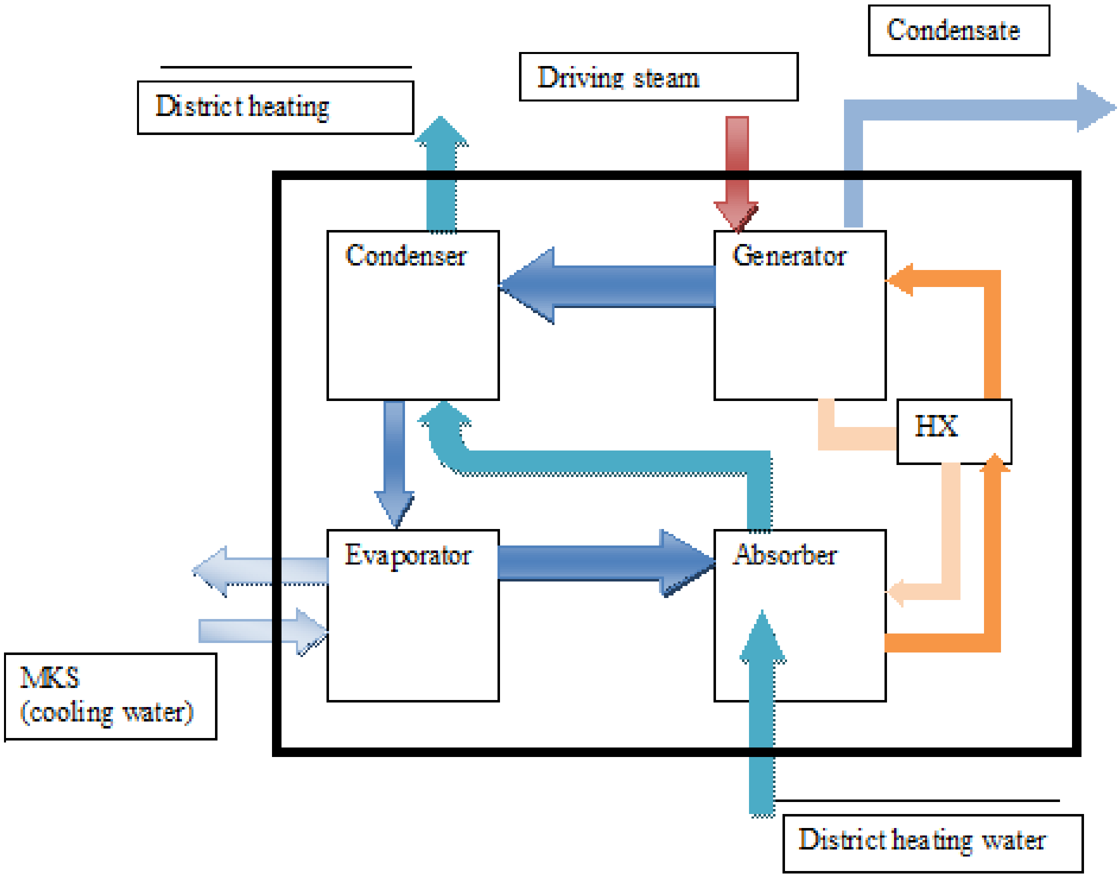


Figure 24: System boundaries

Appendix II: Absorption heat pump, technical data

Sanyo

Type: TSA-GH-1080XVS
Working fluid: Lithium bromide and water
Nominal power: 3800kW
Length: 8.8m
Width: 3.4m
Height: 4.6m
Evaporator-Absorber

Absorber and evaporator

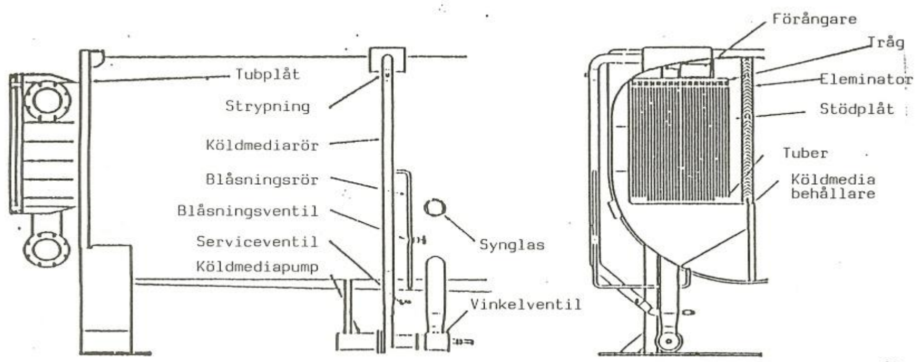


Figure 25: Evaporator [8]

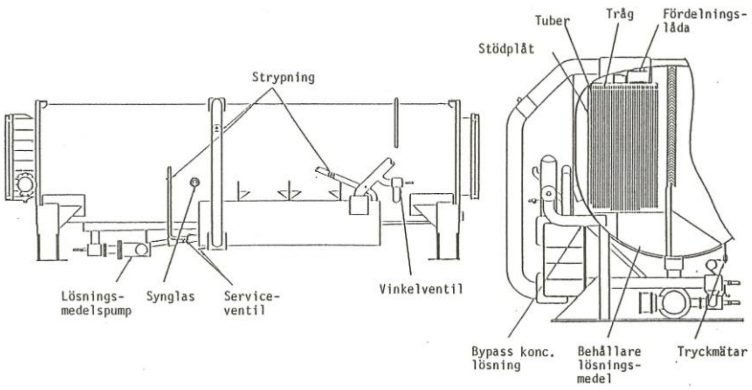


Figure 26: Absorber [8]

Material:
Evaporation tube: Copper C1201T-1/2H
k=401((W/m^2 K)   *([18] Theengineering tool box)*

Absorber tube: Stainless steel NSS430M3
k= 28, 7 W/m^2 K   *([17] Efunda)*

Number of tubes
Evaporator: 854 tubes
Absorber: 1002 tubes

Number of passes
Evaporator: 4
Absorber: 2

Tube dimensions (Outer diameter tube thickness)*
Evaporator: O 19*0,7mm
Absorber: O 19*1,0mm
L=8m (Assumption)

Design conditions
Design pressure in evaporation tubes: 10Bar
Design pressure in absorption tubes: 16Bar
Design pressure of working medium: 1.15Bar
Design temperature low temperature heat source: 60°C
Design temperature of the district heating water: 95°C
Design flow, evaporator: 75 Kg/s
Design flow, absorber: 108 Kg/s

<u>Generator</u>

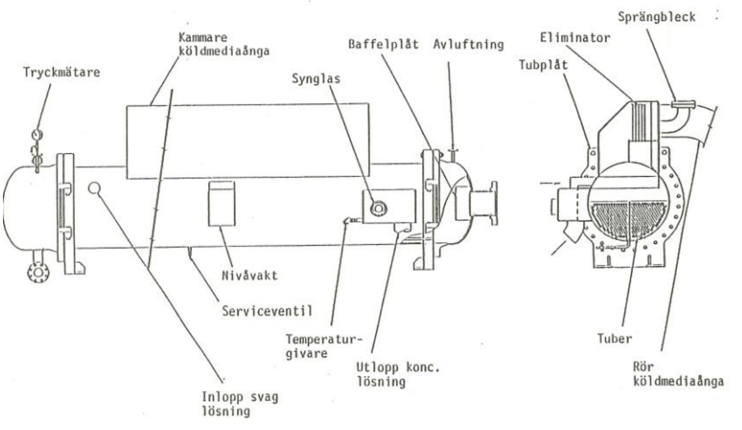


Figure 27: Generator [8]

Material:
The tubes: Stainless steel NSS430M3
k= 28, 7 W/mK *([17] Efunda)*

Number of tubes
Number of tubes: 427st

Tube dimensions (Outer diameter tube thickness)*
Tube dimensions: O 19*1mm
L=8m (Assumption)

Number of passes
1 pass

Design conditions
Design pressure in the tubes: 25Bar
Design pressure on the shell side: 1,15Bar
Design temperature on the tube side: 185°C
Design temperature on the shell side: 115 °C

Condenser

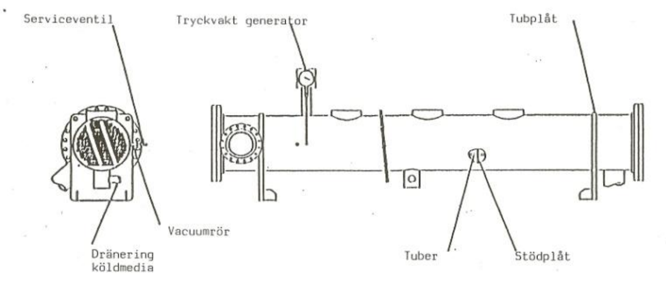


Figure 28: Condenser [8]

Material
The tubes: Stainless steel NSS430M3
k= 28, 7 W/m^2 K *([17] Efunda)*
L=8m
Number of tubes
Number of tubes: 299st

Tube dimensions (Outer diameter tube thickness)*
Tube dimensions: O 19*1mm
L=8m (Assumption)

Number of passes
1 pass

Design conditions
Design pressure in the tubes: 16Bar
Design pressure on the shell side: 1,15Bar
Design temperature on the tube side: 95°C
Design temperature on the shell side: 95 °C

Strong-weak solution heat exchanger

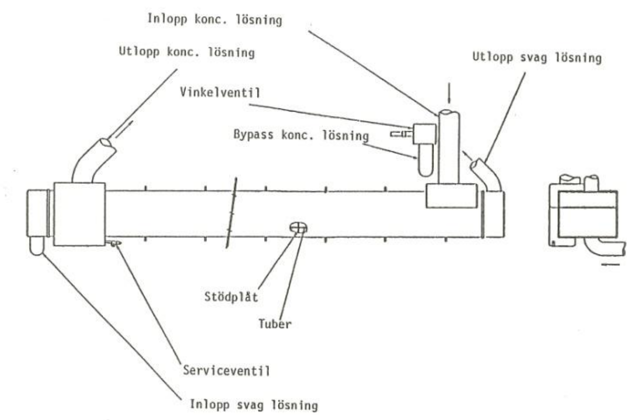


Figure 29: Solution heat exchanger / Sanyo [8]

Material
The tubes: Copper nickel 95-5
k=50 (W/mK)   (Copper-Nickel 90-10 [19] Copper org)

Number of tubes
Number of tubes: 1779st

Tube dimensions (Outer diameter tube thickness)*
 O 19*0,5mm
L=6m (Assumption)

Number of passes
3 pass on both sides

Design conditions
Design pressure in the tubes: 4Bar
Design pressure on the shell side: 1,15Bar
Design temperature on the tube side: 155°C
Design temperature on the shell side: 145°C

Condenser heat exchanger
Material
The tubes: Stainless steel
k= 28, 7 W/m^2 K   *([17] Efunda)*

Number of tubes
Number of tubes: 100st

Tube dimensions (Outer diameter tube thickness)*
Tube dimensions: O 19*0,1mm
L=4m (Assumption)

Number of passes
2 pass

Design conditions
Design pressure in the tubes: 25Bar
Design pressure on the shell side: 4Bar
Design temperature on the tube side: 185°C
Design temperature on the shell side: 155°C

Entropie

Evaporator

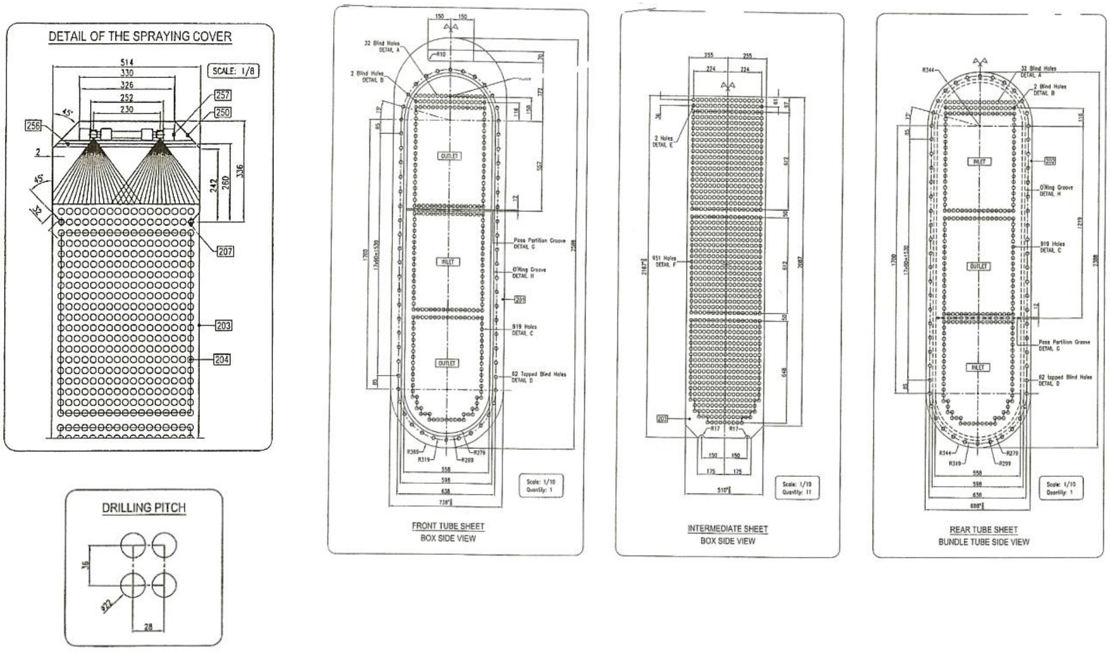


Figure 30: Evaporator [9]

Material
The tubes: Cu-Ni10Fe1Min
k=50 (W/mK)   (Copper-Nickel 90-10 [19] Copper org)
Number of tubes
Heat exchanger: 919st

Tube dimensions (Outer diameter tube thickness)*

Tube dimensions heat exchanger tubes: 22Ø *0, 7 length: 12040mm
Tube dimensions impact tubes: 22Ø *0, 7 length: 11980 mm

Number of passes
3

Absorber

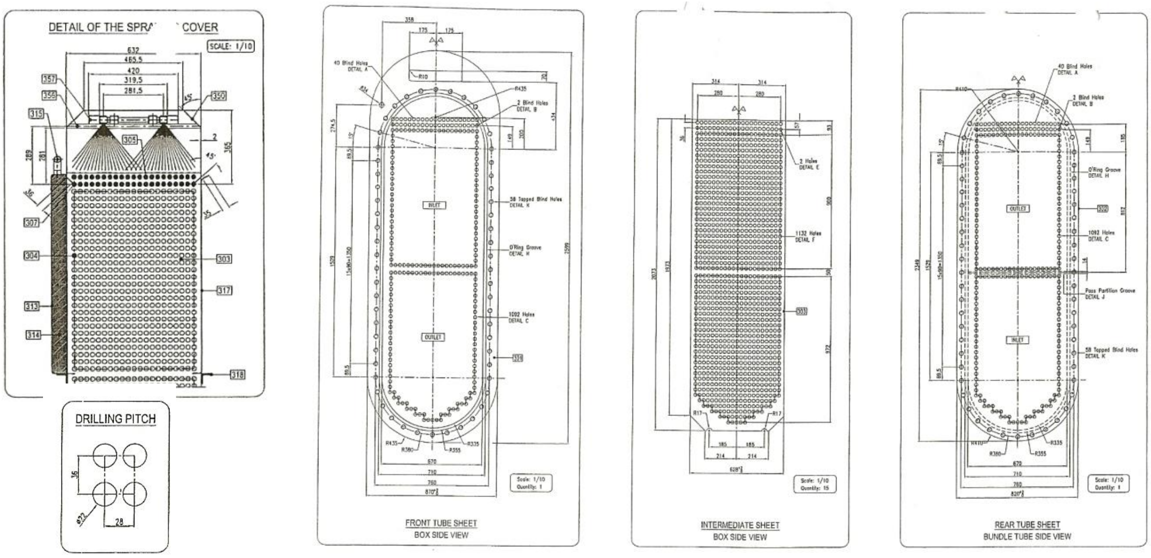


Figure 31: Absorber [9]

Material
The tubes: Cu-Ni10Fe1Min
k=50 (W/mK) (Copper-Nickel 90-10 [19] Copper org)

Number of tubes
Heat exchanger: 1029st
Impact tubes: 40st
Intermediate plate
Number of immersed plates: 15

Tube dimensions (Outer diameter tube thickness)*
Tube dimensions heat exchanger tubes: 22Ø *0, 7 length: 12140mm
Tube dimensions impact tubes: 22Ø *0, 7 length: 11980 mm

Number of passes
Two pass

Max conditions
Max pressure in the tubes: -1/1, 5 bar
Max pressure on the shell side: 16 bar
Max temperature on the tube side: 5/110 [℃]
Max temperature on the shell side: 5/110 [℃]

Condenser

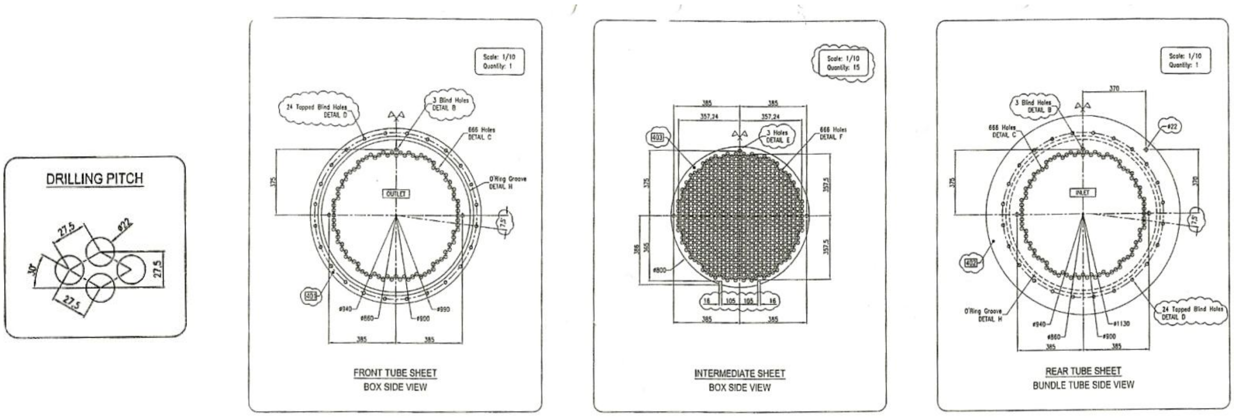


Figure 32: Condenser [9]

Material
The tubes: Cu-Ni10Fe1Min
k=50 (W/mK) (Copper-Nickel 90-10 [19] Copper org)

Number of tubes
Heat exchanger: 666 st
Intermediate plate
Number of immersed plates: 13

Tube dimensions (Outer diameter tube thickness)*
Tube dimensions heat exchanger tubes: 22Ø *0, 7 length: 1200mm

Number of passes
2

Generator

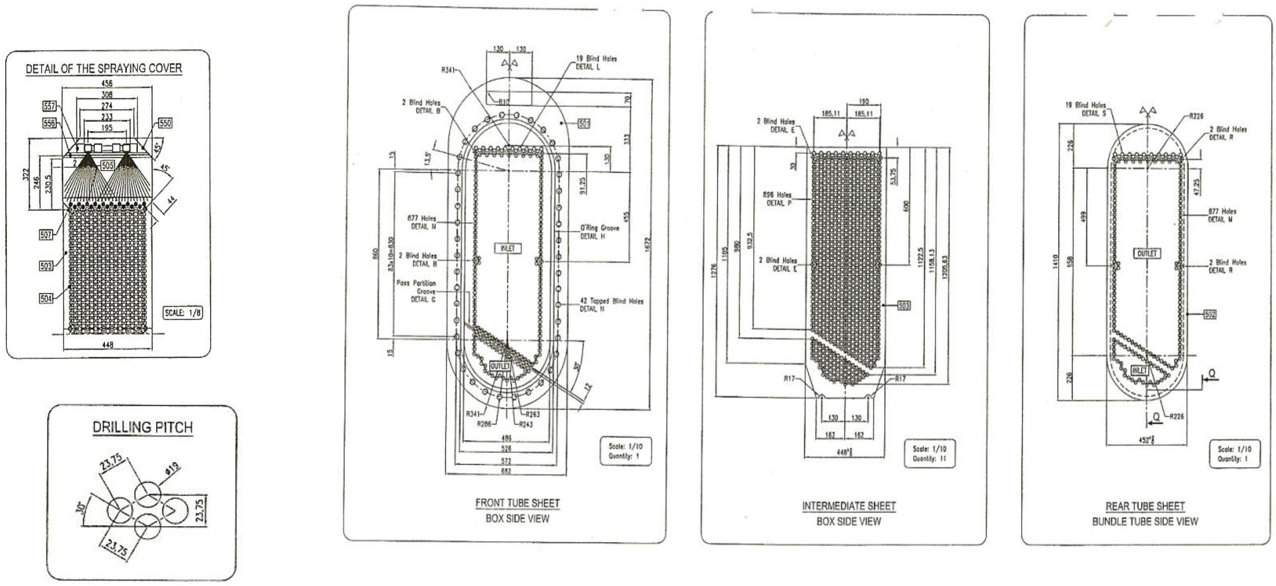


Figure 33: Generator [9]

Material

The tubes: Cu-Ni10Fe1Min
k=50 (W/mK) (Copper-Nickel 90-10 [19] Copper org)

Number of tubes
Heat exchanger: 877 st
Impact tubes: 19 st

Tube dimensions (Outer diameter tube thickness)*
Tube dimensions heat exchanger tubes: 22Ø *0,7 length: 11500 mm
Tube dimensions impact tubes: 22Ø *0, 7 length: 11980 mm

Number of passes
2

Appendix III: Drawing of the Entropie heat pump

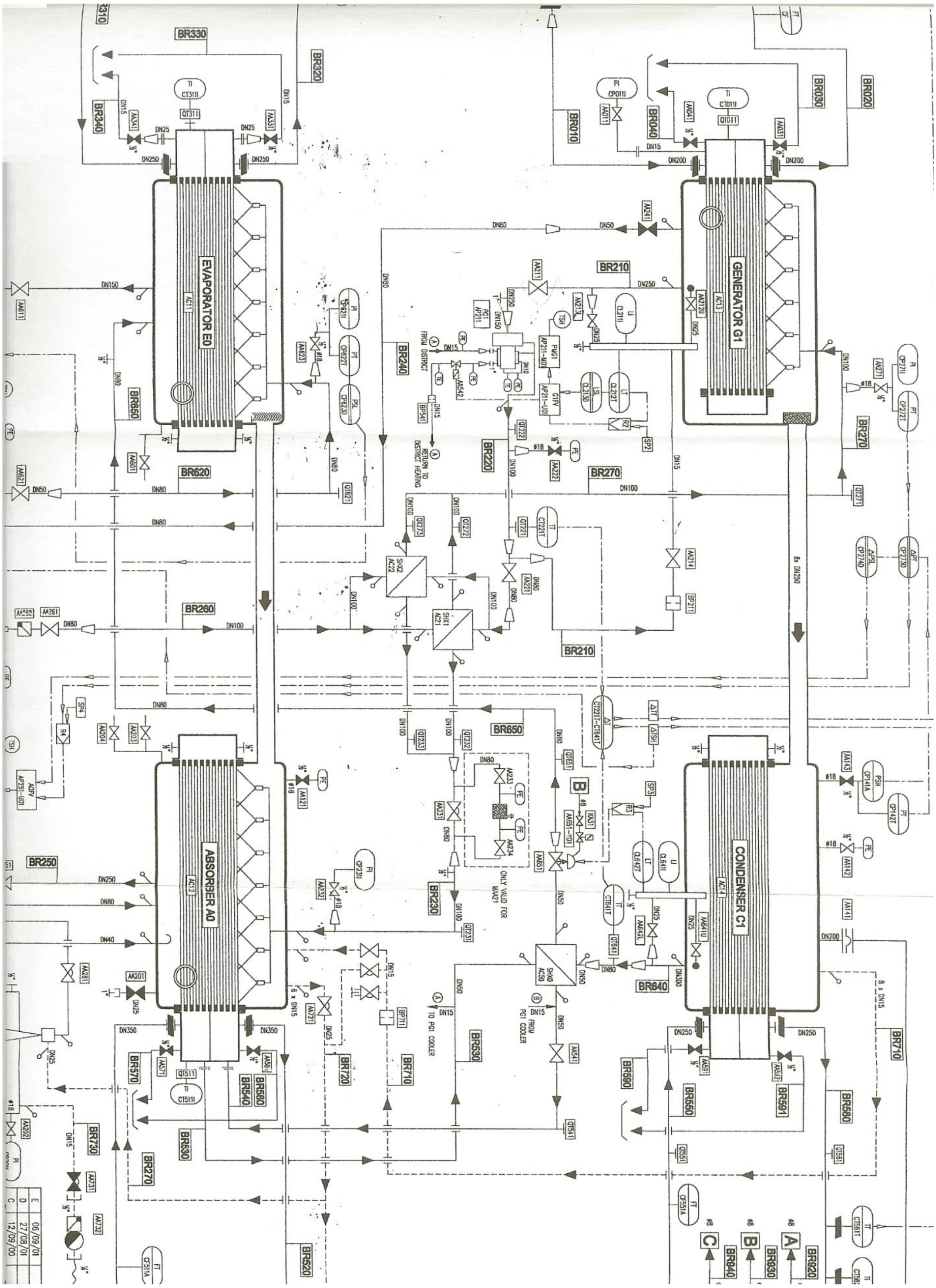
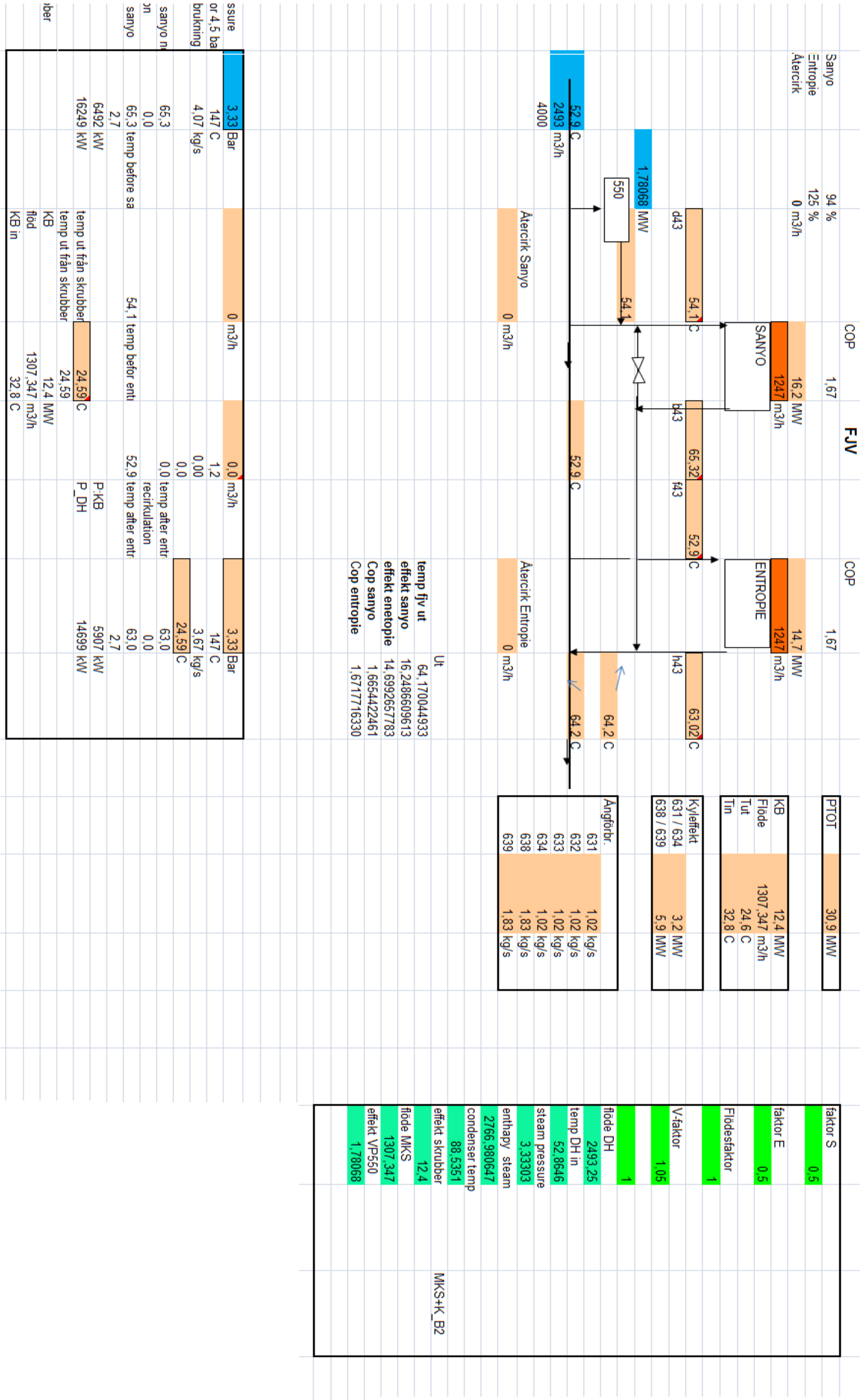
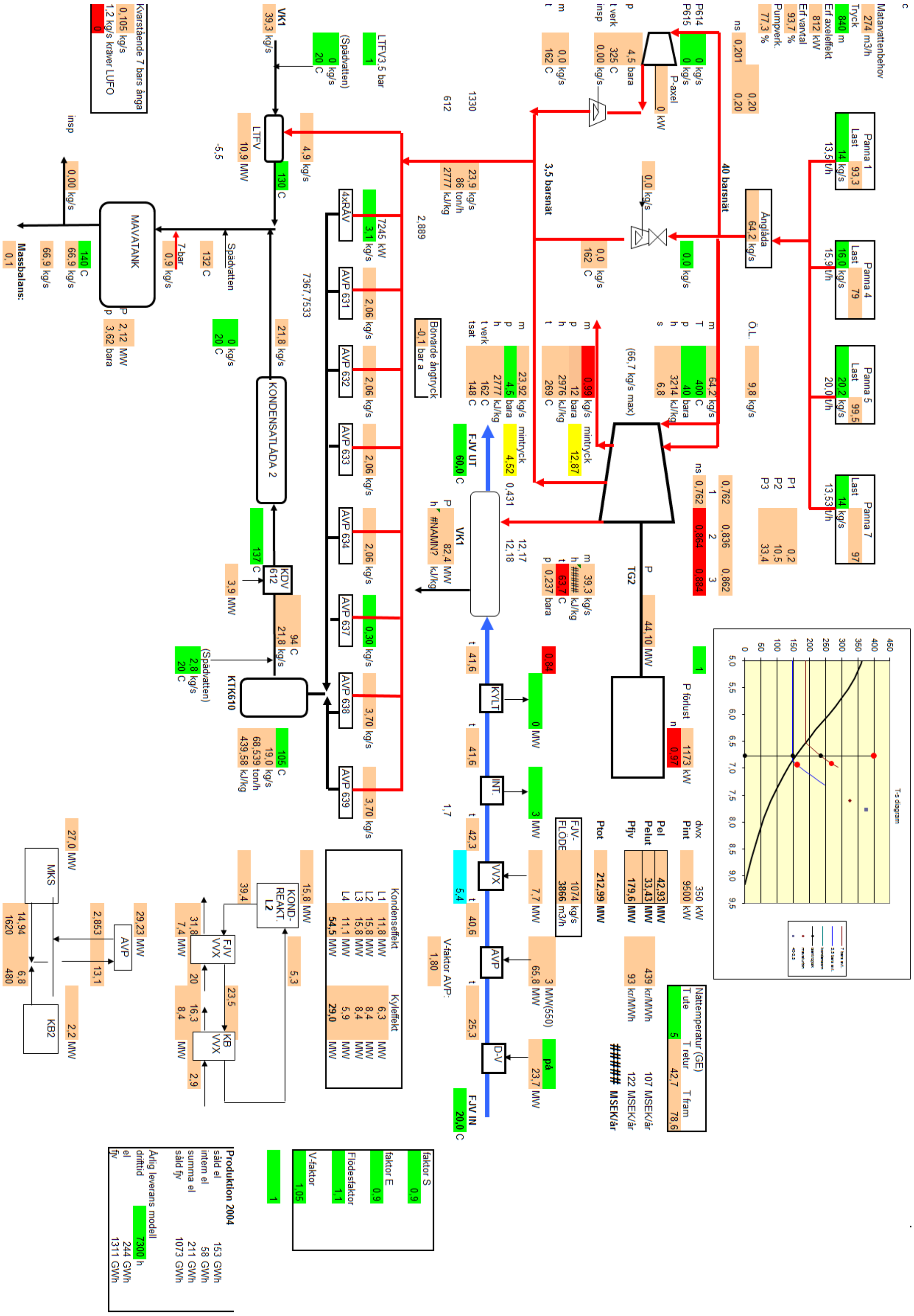


Figure 34: Condenser / Entropie (Entropie)

Appendix IV: Absorption heat pump, Modell-Renova

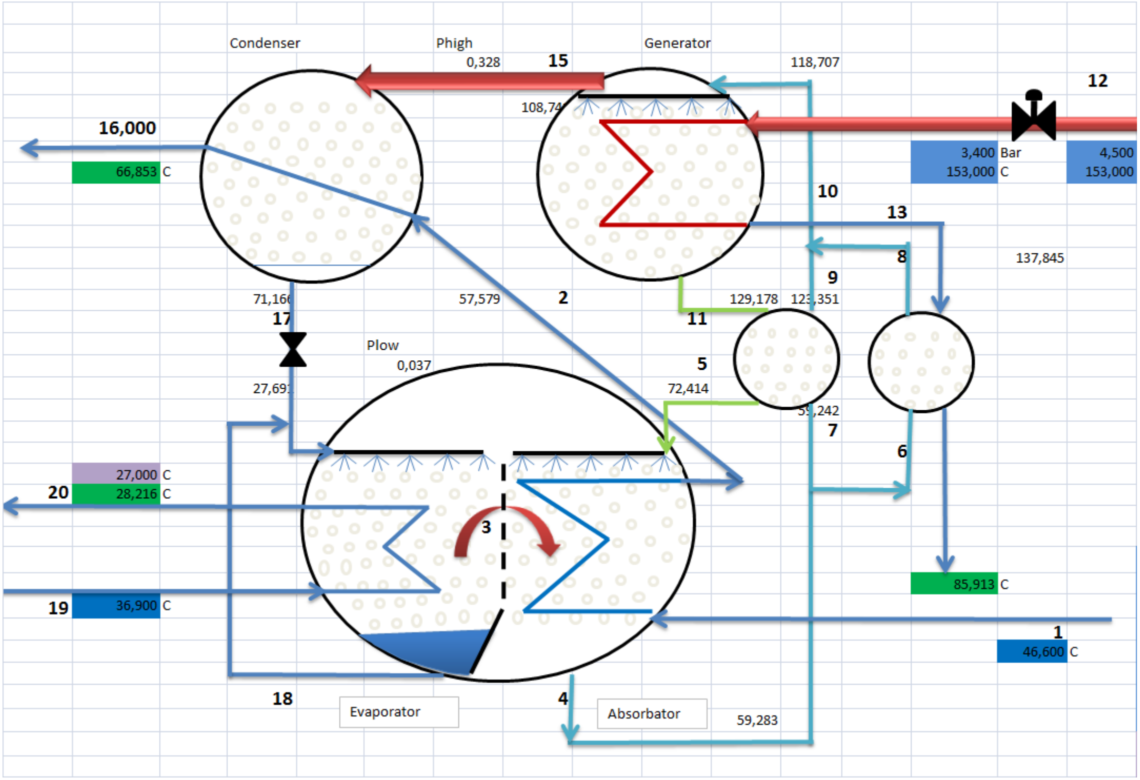


Appendix V: Model-Renova model of the entire plant

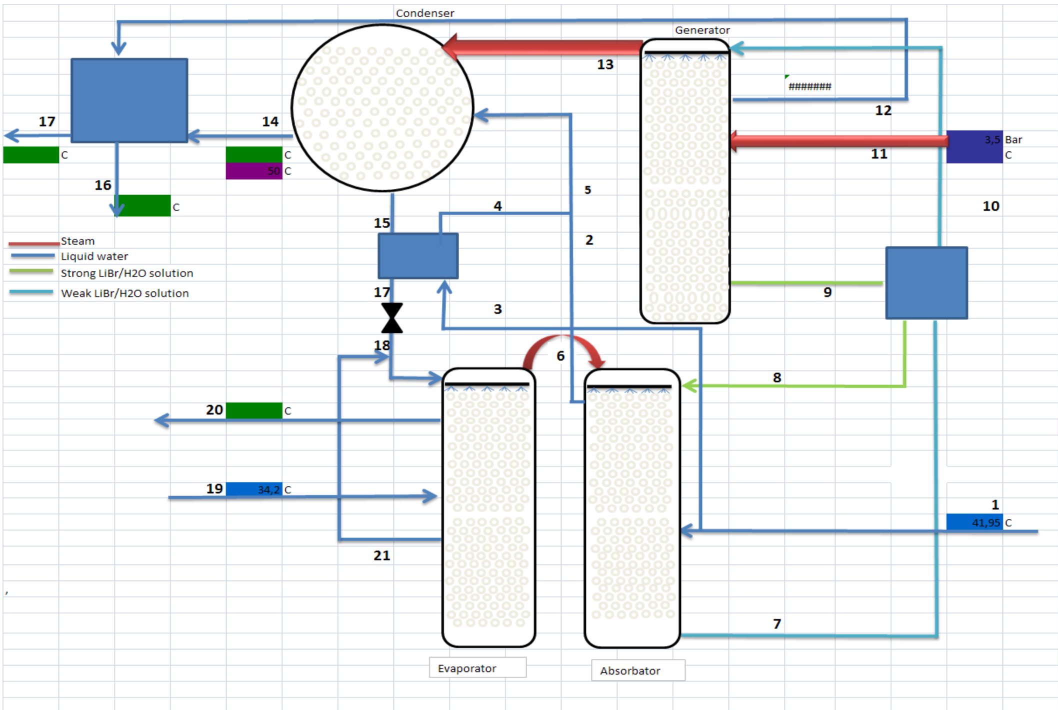


Appendix VI: The New model

Sanyo



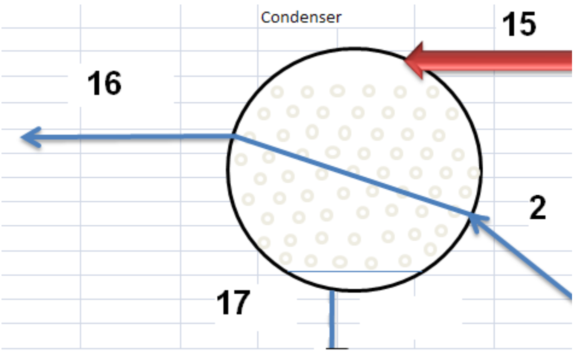
Entropie



Appendix VII: Calculation

The equations formulated here are used in the model of the Sanyo heat pumps. Calculation procedure for the Entropie heat pumps is similar except for the calculation of the bypass heat exchanger and the condensate after cooler.

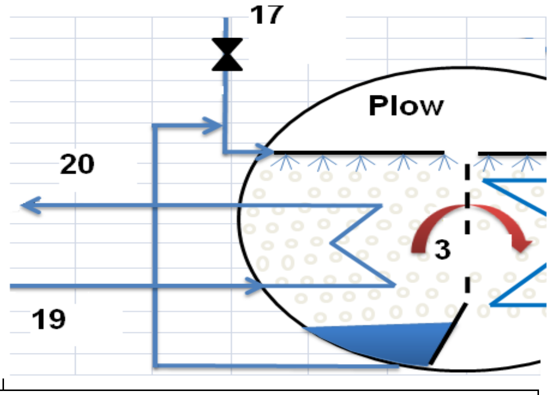
Condenser



Underlying equation	
$Q_{cond} = m_{DH}(h_{16} - h_2) = m_{17,15}(h_{15} - h_{17}) = Q'_{cond} + Q''_{cond}$	
$m_{17} = m_{15}$	
$T_{17} = T(P_{high}, saturated\ liquid)$	
$Q'_{cond} = E * q_{Max}$	
$Q''_{cond} = m_{15}(h_{15} - h'_{15})$	
Solved equation In data : T15 Phigh T2 m_DH	
$h15 = h(T15, Phigh)$	Enthalpy of the steam entering the condenser as a function of the temperature and pressure
$h'15 = h(saturated\ vapor\ , Phigh)$	Enthalpy of the steam when it has been cooled down to saturated vapor
$Q_{overheated} = m_{15}(h_{15} - h'_{15})$	Heat released when the steam is brought down to a saturated state
$h17 = h(sat\ liquid, Phigh)$	Enthalpy of the condensate leaving the condenser as saturated liquid at the pressure Phigh.
$T17 = Tsat.liquid\ (Phigh)$	Temperature of the condensate = saturation temperature at Phigh.
$C_{16,2} = \frac{Cp(sat.liduid, (T_2 + T_{16})}{2} * m_{DH}$ $C_{15,17} = Cp_{15,17} * m_{17}\ , \quad Cp_{15,17} = \infty$	The capacity rate C is calculated from the specific heat and the mass flow. The specific heat for the condensation steam is infinite since the temperature is constant during the phase transition.
$q_{max} = C_{16,2} * (T_{15'} - T_2)$ $T_{15'} = T_{17}$	The maximal heat that is transferred during the condensation. $T_{15'}$ is the temperature of the saturated steam which will be the same as the temperature for the saturated condensate.

$U = \frac{1}{\frac{D}{d * h_{tube}} + \frac{D}{d * hf\_tube} + \frac{D * LN(\frac{D}{d})}{2 * k} + \frac{1}{h\_shell} + \frac{1}{hf\_shell}}$	The overall heat transfer coefficient calculated from the convection and fouling coefficients at the shell and tube side and the conduction through the tube wall.
$NTU = UA/C_{16,2}$	The number of heat transfer units calculated from the Overall heat transfer coefficient and the heat transfer area and capacity rate of the district heating water.
$\varepsilon = (1 - e^{-NTU})$ (Incropera, et al., 2005)	The heat transfer efficiency calculated for a condenser.
$Q_{phase\ change} = q_{max} * \varepsilon$	The heat transferred during the phase change.
$Q_{cond} = Q_{overheated} + Q_{phase\ change}$	The total heat transferred is the heat released when the overheated steam is brought down to saturation and the heat released in the phase change.
$h2 = h(sat.liquid, T2)$	Enthalpy of the incoming district heating water as saturated liquid at the temperature T2.
$h16 = h2 + Qcond/m\_DH$	The enthalpy of the outgoing district heating water calculated from the above calculated heat transfer.
$T16 = T(sat.liquid, h16)$	Temperature of the outgoing district heating water calculated from the enthalpy h16.
$\mathrm{qcond} = (\mathrm{h15} - \mathrm{h17})$	The enthalpy difference between the incoming overheated steam and the outgoing condensate.
$m17 = \frac{Qcond}{qcond}$	The mass flow at the shell side of the condenser is calculated from the heat transferred.

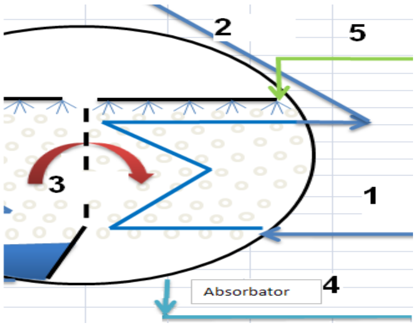
Evaporator



Underlying equation
$Q_{eva} = m_{MKS}(h_{19} - h_{20}) = m_{17,3}(h_{17} - h_3) = U * A * \Delta T_{LM}$
$m_3 = m_{17}$
$P_{low} = P(T_3, saturated\ )$
$T_3 = T_{17}$

Solved equation Indata : T17_cond (h17) m17=m3=m15 Phigh T19 $\dot{v}_{MKS}$	
$h17 = h(sat.liquid, Phigh)$	Calculate the enthalpy of the condensate when it exits the condenser as saturated liquid.
Guess : Plow	Make an initial guess of the pressure inside the evaporator.
$T17 = T(saturated, Plow\ )$	Calculate the temperature after the expansion valve (The temperature after the expansion valve will be lower than the temperature after the condensers Since the pressure is lower but the enthalpy is the same).
$T3 = T17$	Since the steam leaving the evaporator is assumed to be saturated vapor, the temperature will be equal throughout the shell side of the evaporator.
$h3 = h(sat.vapor, Plow)$	The enthalpy of the water vapor leaving the evaporator is calculated as saturated vapor at $P_{low.}$
$\mathrm{Qeva} = (\mathrm{h3} - \mathrm{h17}) * \mathrm{m17}$	The heat cooled off from the low temperature heat source (MKS water)
$h20 = h19 - Qeva/m\_MKS$	The enthalpy of the outgoing MKS stream
$T20 = h(liquid, h20)$	The temperature of the outgoing MKS stream as saturated liquid and a function of the enthalpy.
$U = \dfrac{1}{\dfrac{D}{d * h_{tube}} + \dfrac{D}{d * hf\_tube} + \dfrac{D * LN(\frac{D}{d})}{2 * k} + \dfrac{1}{h\_shell} + \dfrac{1}{hf\_shell}}$	The overall heat transfer coefficient calculated from the convection and fouling coefficients at the shell and tube side and the conduction through the tube wall.
$\Delta T_{LM} = \dfrac{Qeva}{U * A}$	The logarithmic mean temperature calculated from the heat transferred, the area and the overall heat transfer coefficient.
$T3 = T17 = (T19 - T20 * e^{(T19-T20)}/\Delta_{LM})/(1 - e^{(T19-T20)})$	The temperature of the evaporated vapor leaving the evaporator and the condensate entering the evaporator is calculated from the logarithmic mean temperature difference.
$PLow = Psat(T3)$	The pressure inside the evaporator can now be calculated from the saturation temperature.
$Plow = Plow$	The calculated pressure Plow is compared with the initial guess and a new guess is made.
Outdata Plow T19 T3 Qeva	

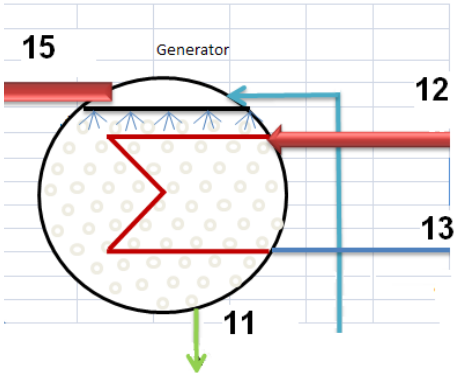
Absorber



Underlying equation
$Q_{abs} = m_{DH}(h_2 - h_1) = \dot{m}_3 h_3 + \dot{m}_5 h_5 - \dot{m}_4 h_4 = U * A * \Delta T_{LM}$
$\dot{m}_4 = \dot{m}_3 + \dot{m}_5$
$x_5 \dot{m}_5 = x_4 \dot{m}_4$
$P_{low} = f(T_4, x_4)$

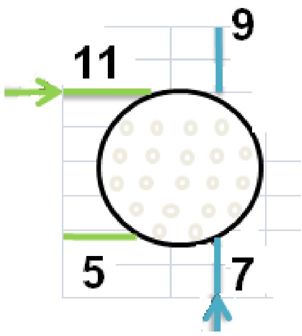
Solved equation Indata : T5 $\dot{m}5$ PLow T1 V_DH h3 $\dot{m}3$	
Guess : T4	Make an initial guess of the temperature of the outgoing weak solution.
$x4 = f(Plow, T4)$ $(equation\ 3.8.1.2)$	Calculate the LiBr weight concentration of the outgoing weak solution from the pressure and temperature. The concentration is calculated from the saturation pressure as a function of the temperature and the concentration.
$h4 = h(T4, x4)$ $(Equation\ 3.8.1.2)$	The enthalpy of the weak solution as a function of the temperature and the concentration.
$\dot{m}4 = \dot{m}5 + \dot{m}3$	From a mass balance over the shell side one can calculate the mass flow of the weak solution.
$x5 = \dfrac{\dot{m}4 * x4}{m5}$	The concentration of the strong solution is then given by the mass balance of the individual species (LiBr HO) .
$h3 = h(Sat. vapor, Plow)$	The enthalpy of the vapor entering from the evaporator.
$h1 = h(Liquid, T1)$	The enthalpy of the incoming district heating water.
$h5 = h(T5, x5)$ (Equation 3.8.1.1)	The enthalpy of the incoming strong solution.
$Qabs = h3 * m3 + h5 * m5 - h4 * m4$	The heat transferred to the district heating water.
$h2 = h1 + Qabs/m\_DH$	The enthalpy of the outgoing district heating water can now be calculated from the enthalpy of the incoming district heating water and the heat transferred.
$T2 = T(Liquid, h2)$	The temperature of the outgoing district heating water can now be calculated from the enthalpy.
$U = \dfrac{1}{\dfrac{D}{d * h_{tube}} + \dfrac{D}{d * hf\_tube} + \dfrac{D * LN(\frac{D}{d})}{2 * k} + \dfrac{1}{h\_shell} + \dfrac{1}{hf\_shell}}$	The overall heat transfer coefficient.
$\Delta T_{LM} = \dfrac{(T5 - T2) - (T4 - T1)}{LN(\frac{T5 - T2}{T4 - T1})}$	The logarithmic mean temperature difference.
$Qabs = U * A * \Delta T_{LM}$	The heat transferred can now be calculated from U, A and the logarithmic mean temperature difference.
$h4 = (h3 * m3 + h5 * m5 - Qabs)/m4$	The enthalpy of the outgoing weak solution can now be calculated from the energy balance.
$T4 = T(h4, x4)$ $(Equation\ 3.8.1.1)$	The temperature of the weak solution can now be calculated from the enthalpy and concentration.
$T4 = T4$	The calculated temperature is compared with the initial guess and a new guess is made.
Outdata : Qabs, T4, x4, m4, T2, x5	

Generator



Underlying equation	
$Q_{gen} = m_{12}(h_{12} - h_{13}) = m_{10}h_{10} - m_{11}h_{11} - m_{15}h_{15} = U * A * \Delta T_{LM}$	
$T_{13} = T(P_{12}, saturated\ Liquid)$	
$P_{high} = f(T_{11}, x_{11})$	
Solved equation Indata : P12 T12 m10 T10 x10 m15 m11 x11	
Guess : T11	An initial guess of the temperature of the outgoing strong solution.
$Phigh = P(x11, T11)$ $(Equation\ 3.8.1.1)$	The pressure inside the generator is calculated from the expression of the saturation pressure for a LiBr/H2O solution as a function of the temperature and the concentration.
$h11 = h(T11, x11)$ $(Equation\ 3.8.1.2)$	The enthalpy of the outgoing strong solution.
$T10sat = T(satliquid, Phigh)$	The saturation temperature of the incoming weak solution as a function of the pressure.
$h10 = h(x10, T10)$ $(Equation\ 3.8.1.1)$	The enthalpy of the incoming weak solution
$T15 = T10sat$	The temperature of the outgoing water vapor is the same as the saturation temperature of the LiBr solution entering the generator.
$h15 = h(Phigh, T15)$	The enthalpy of the water vapor as a function of the temperature and pressure.
$h12 = h(P12, T12)$	The enthalpy of the driving steam.
$h13 = h(saturated\ liquid)$	The enthalpy of the condensate leaving the generator (Assume to be saturated liquid).
$m12 = Qdes/(h12 - h13)$	The mass flow of driving steam m12
$U = \dfrac{1}{\dfrac{D}{d * h_{tube}} + \dfrac{D}{d * hf\_tube} + \dfrac{D * LN(\frac{D}{d})}{2 * k} + \dfrac{1}{h\_shell} + \dfrac{1}{hf\_shell}}$	The overall heat transfer coefficient.
$\Delta T_{LM} = \dfrac{(T12 - T15) - (T13 - T11)}{LN(\frac{T12 - T15}{T13 - T11})}$	The logarithmic mean temperature difference.
$Qdes = U * A * \Delta T_{LM}$	The heat transferred from the driving steam.
$h11 = Qdes - h10 * m10 + h15 * m15)/m4$	The enthalpy of the strong solution can be solved from the energy balance.
$T11 = T(h11, x11)$ $(Equation\ 3.8.1.1)$	The temperature of the strong solution can now be calculated from the enthalpy and concentration.
$T11 = T11$	The calculated temperature is compared with the initial guess and a new guess is made.
Outdata: Qdes, T11, T13, T15, m11, Phigh, m12	

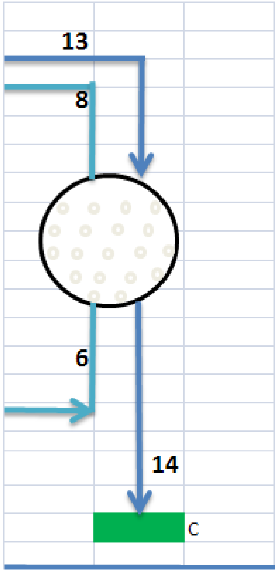
Solution HX



Underlying equation
$Q_{HXs} = m_7(h_9 - h_7) = m_{11}(h_{11} - h_5) = q_{max} * E$
$m_7 = m_9 = m_4 * (1 - F)$
$m_{11} = m_5$
$x_5 = x_{11} \qquad x_4 = x_9 = x_7$

Solved equation <u>Indata :</u> T11 m11 x11 T7 m7 x7 F	
$\boldsymbol{h_{11} = h(T_{11}, x_{11})}$	**Enthalpy of the strong solution from the generator.**
$\boldsymbol{h_7 = h(T_7, x_7)}$	**Enthalpy of the weak solution from the absorber (h_4=h_7).**
$\boldsymbol{Cp_{11,5} = Cp((T_{11} - T_9)/2, x11)}$	**Heating rate of the strong solution.**
$\boldsymbol{Cp_{10,7} = Cp(T_4 - T_7, x_7)}$	**Heating rate of the weak solution.**
$\boldsymbol{C_{min} = Cp_{11,5} * m_{11} \quad if \quad Cp_{11,5} * m_{11} < Cp_{10,7} * m_{07}}$ **else** $\boldsymbol{C_{min} = Cp_{10,4} * m_7}$	**The two heating rates are compared to find Cmin and Cmax.**
$\boldsymbol{Q_{max} = C_{min}(T_{11} - T_4)}$	**The maximal heat that could be transferred during the heat exchange.**
$\boldsymbol{U = \frac{1}{\frac{D}{d * h_{tube}} + \frac{D}{d * hf\_tube} + \frac{D * LN(\frac{D}{d})}{2 * k} + \frac{1}{h\_shell} + \frac{1}{hf\_shell}}}$	**The overall heat transfer coefficient.**
$\boldsymbol{NTU = UA/C_{min}}$	**The number of heat transfer units calculated from the Overall heat transfer coefficient and the heat transfer area and the minimum heat transfer rate.**
$\boldsymbol{\varepsilon = \left(1 - e^{-1/(C_{min}/C_{max})*(1-e^{(-NTU)})}\right) \text{ (Incropera, et al., 2005)}}$	**The heat transfer efficiency calculated for Counter current flow heat exchanger.**
$\boldsymbol{Q = q_{max} * \varepsilon}$	**The heat transferred.**
$\boldsymbol{h_9 = h_7 + \frac{Q_{HXs}}{m_7}}$	**Enthalpy of the weak solution leaving the heat exchanger calculated from the energy balance.**
$\boldsymbol{T_9 = f(h_9, x_9)}$	**Temperature of the weak solution leaving the heat exchanger.**
$\boldsymbol{h_5 = h_{11} - \frac{Q_{HXs}}{m_{11}}}$	**Enthalpy of the strong solution leaving the heat exchanger calculated from the energy balance.**
$\boldsymbol{T_5 = f(h_5, x_5)}$	**Temperature of the strong solution leaving the heat exchanger.**

Condensate HX



Underlying equation
$Q_{HXc} = m_8(h_8 - h_6) = m_{13}(h_{13} - h_{14}) = q_{max} * E$
$m_6 = m_8 = m_4 * F$
$m_{13} = m_{14}$
$x_4 = x_8 = x_6$

Solved equation Indata : m13, T13, m6, T6, x6 , P12	
$h_6 = h(T_6, x_6)$	Enthalpy of the weak solution from the absorber (h_4=h_6).
$h_{13} = h(T_{13}, satliquid)$	Enthalpy of the condensate from the generator.
$Cp_{6,8} = Cp(T6, x6)$	Heating rate of the weak solution.
$Cp_{13,14} = Cp(T13, satliquid)$	Heating rate of the condensate.
$C_{min} = Cp_{6,8} * m_6 \quad if \quad Cp_{6,8} * m_6 < Cp_{13,14} * m_{13}$ else $C_{min} = Cp_{13,14} * m_{13}$	The two heating rates are compared to find Cmin and Cmax.
$q_{max} = C_{min}(T_{13} - T_6)$	The maximal heat that could be transferred during the heat exchange.
$U = \dfrac{1}{\dfrac{D}{d * h_{tube}} + \dfrac{D}{d * hf\_tube} + \dfrac{D * LN(\frac{D}{d})}{2 * k} + \dfrac{1}{h\_shell} + \dfrac{1}{hf\_shell}}$	The overall heat transfer coefficient.
$NTU = UA/C_{min}$	The number of heat transfer units calculated from the Overall heat transfer coefficient and the heat transfer area and the minimum heat transfer rate.
$\varepsilon = \left(1 - e^{-1/(C_{min}/C_{max})*(1-e^{(-NTU)})}\right)$ (Incropera, et al., 2005)	The heat transfer efficiency calculated for a 2 pass heat exchanger.
$Q = q_{max} * E$	The heat transferred.
$h_8 = h_6 + \dfrac{Q_{HXs}}{m_6}$	Enthalpy of the weak solution leaving the heat exchanger calculated from the energy balance.
$T_8 = f(h_8, x_8)$	Temperature of the weak solution leaving the heat exchanger.
$h_{14} = h_{13} - \dfrac{Q_{HXs}}{m_{13}}$	Enthalpy of the condensate leaving the heat exchanger calculated from the energy balance.
$T_{14} = f(h_{14}, sat\ liquid)$	Temperature of the condensate leaving the heat exchanger.

Overall equation

$$h_{10}m_{10} = h_9 m_9 + h_8 m_8$$

Appendix VIII: Equation system

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	Unknown	Given	Equations	
1	T_2	T_1	h_1	enthalpystaLiq(T_1)
2	T_3	T_{19}	h_2	enthalpystaLiq(T_2)
3	T_4	T_{12}	h_3	enthalpysatVap(P_{low})
4	T_5	M_{DH}	h_4	entahlpyLiBr(T_4,X_4)
5	T_6	M_{MKS}	h_5	entahlpyLiBr(T_5,X_5)
6	T_7	P_{12}	h_6	entahlpyLiBr(T_6,X_6)
7	T_8	F	h_7	entahlpyLiBr(T_7,X_7)
8	T_9	U_{eva}	h_8	entahlpyLiBr(T_8,X_8)
9	T_{10}	U_{cond}	h_9	entahlpyLiBr(T_9,X_9)
10	T_{11}	U_{abs}	h_{10}	entahlpyLiBr(T_{10},X_{10})
11	T_{13}	U_{gen}	h_{11}	entahlpyLiBr(T_{11},X_{11})
12	T_{14}	U_{HXs}	h_{12}	entahlpy(P_{12},T_{12})
13	T_{15}	U_{HXc}	h_{13}	entahlpystaLiq(P_{12})
14	T_{16}	A_{eva}	h_{14}	enthalpystaLiq(T_{14})
15	T_{17}	A_{cond}	h_{15}	enthalpy(P_{high},T_{15})
16	T_{18}	A_{abs}	h_{16}	enthalpysatLiq(T_{16})
17	T_{20}	A_{gen}	h_{17}	enthalpysatLiq(P_{high})
18	m_3	A_{HXS}	h_{19}	enthalpysatLiq(T_{19})
19	m_4	A_{HXc}	h_{20}	entalphysatLiq(T_{20})
20	m_5		Q_{cond}	$M_{DH*}(h_{16}-h_2)$
21	m_6		Q_{cond}	$M_{17*}(h_{15}-h_{17})$
22	m_7		Q'_{cond}	ε*Qmax
23	m_8		Q''_{cond}	$M_{17}(h_{15}'-h_{15})$
24	m_9		Q_{Ccond}	$Q'_{cond}+Q''_{cond}$
25	m_{10}		m_{15}	M_{17}
26	m_{11}		h'_{15}	enthalpySatPW(P_{high})
27	m_{12}		T_{17}	TsatW(P_{high})
28	m_{13}		Q_{eva}	$M_{MKS}(h_{19}-h_{20})$
29	m_{14}		Q_{eva}	$M_{17}(h_3-h_{17})$
30	m_{15}		Q_{eva}	U_{eva}*A_{eva}*ΔTm_eva
31	m_{17}		m_{17}	m_3
32	Q_{abs}		P_{low}	psatW(T_3)
33	Q_{eva}		T_3	T_{18}
34	Q_{cond}		Q_{abs}	M_{DH} (h_2-h_1)
35	Q_{gen}		Q_{abs}	$m_3*h_3+m_5*h_5-m_4*h_4$
36	Q_{HXc}		Q_{abs}	$U_{abs}*A_{abs}*\Delta TM_{abs}$

37	Q_{HXs}	m_4	m_3+m_5
38	x_4	x_5	X_4*m_4/m_5
39	x_5	P_{low}	$PsatW(T_4,x_4)$
40	x_6	Q_{gen}	$m_{12}(h_{12}-h_{13})$
41	x_7	Q_{gen}	$m_{10}*h_{10}-m_{11}*h_{11}-h_{15}*m_{15}$
42	x_8	Q_{gen}	$U_{gen}*A_{gen}*\Delta TM_{gen}$
43	x_9	T_{13}	$TsatWP(P_{12})$
44	x_{10}	P_{high}	$Psat(T_{11},x_{11})$
45	x_{11}	Q_{HXs}	$m_7(h_9-h_7)$
46	P_{high}	Q_{HXs}	$m_{11}(h_{11}-h_5)$
47	P_{low}	Q_{HXs}	$Q_{HXsmax}*\varepsilon_{HXs}$
48	Q'_{cond}	m_9	m_7
49	Q''_{cond}	m_9	$m_4(1-F)$
50	h'_{15}	m_{11}	m_5
51	h_1	x_{11}	x_5
52	h_2	x_9	x_4
53	h_3	T_7	T_4
54	h_4	Q_{HXc}	$m_8(h_8-h_6)$
55	h_5	Q_{HXc}	$m_{12}(h_{13}-h_{14})$
56	h_6	Q_{HXc}	$Q_{HXcmax}*\varepsilon_{HXc}$
57	h_7	m_8	m6
58	h_8	T_6	T_7
59	h_9	m_8	m_4*F
60	h_{10}	x_8	x_4
61	h_{11}	x_8	x_6
62	h_{12}	m_5	$f(\Delta P)$
63	h_{13}	h_{10}	$h_8*m_8+h_9*m_9$
64	h_{14}	x_9	x_{10}
65	h_{15}	x_8	x10
66	h_{16}	m_{13}	m_{12}
67	h_{17}	m_{14}	m_{12}
68	h_{19}	m_{10}	m_9+m_8
69	h_{20}	T_{15}	$Tsat(P_{high},x_{10})$

Entropie

	Unknown	Given	Equations		
1	T_3	T_1	h_1	=	enthalpystaLiq(T_1)
2	T_4	T_{19}	h_2	=	enthalpystaLiq(T_2)
3	T_5	T_{11}	h_3	=	enthalpysatVap(Pl_{ow})
4	T_6	$m_{DH}=m_1$	h_4	=	entahlpyLiBr(T_4,X_4)
5	T_7	$m_{MKS}=m_{19}$	h_5	=	entahlpyLiBr(T_5,X_5)
6	T_8	P_{11}	h_6	=	entahlpyLiBr(T_6,X_6)
7	T_9	F	h_7	=	entahlpyLiBr(T_7,X_7)
8	T_{10}	U_{eva}	h_8	=	entahlpyLiBr(T_8,X_8)
9	T_{12}	U_{cond}	h_9	=	entahlpyLiBr(T_9,X_9)
10	T_{13}	U_{abs}	h_{10}	=	entahlpyLiBr(T_{10},X_{10})
11	T_{14}	U_{gen}	h_{11}	=	entahlpyLiBr(T_{11},X_{11})
12	T_{15}	U_{HXs}	h_{12}	=	entahlpy(P_{12},T_{12})
13	T_{16}	U_{HXc}	h_{13}	=	entahlpystaLiq(P_{12})
14	T_{17}	U_{HXby}	h_{14}	=	enthalpystaLiq(T_{14})
15	T_{16}	A_{eva}	h_{15}	=	enthalpy(P_{high},T_{15})
16	T_{19}	A_{cond}	h_{16}	=	enthalpysatLiq(T_{16})
17	T_{20}	A_{abs}	h_{17}	=	enthalpysatLiq(P_{high})
18	m_2	A_{gen}	h_{19}	=	enthalpysatLiq(T_{19})
19	m_3	A_{HXS}	h_{20}	=	entalphysatLiq(T_{20})
20	m_4	A_{HXc}	Q_{cond}	=	m_5*(h_{14}-h_5)
21	m_5	A_{HXby}	Q_{cond}	=	Q'_{cond}+Q''_{cond}
22	m_6		Q''_{cond}	=	ε*$Q_{maxcond}$
23	m_7		Q'_{cond}	=	m_{17}(h_{13}'-h_{13})
24	m_8		Q_{cond}	=	m_{17}*(h_{13}-h_{15})
25	m_9		m_{13}	=	m_{15}
26	m_{10}		m_5	=	m_{14}
27	m_{11}		P_{high}	=	Psat(T_{15})
28	m_{12}		Q_{eva}	=	m_{19}*(h_{19}-h_{20})
29	m_{13}		Q_{eva}	=	m_{15}*(h_{18}-h_6)
30	m_{14}		Q_{eva}	=	U_{cond}*A_{cond}*ΔTM_{eva}
31	m_{16}		m_{19}	=	m_{20}
32	m_{17}		m_6	=	m_{18}
33	m_{19}		T_6	=	TsatW(P_{low})
34	m_{20}		Q_{abs}	=	m_1*(h_2-h_1)
35	Q_{cond}		Q_{abs}	=	m_6*h_6+h_8*m_8-h_7*m_7
36	Q'_{cond}		m_7	=	m_6+m_8
37	Q''_{cond}		Q_{abs}	=	U_{abs}*A_{abs}*ΔTM_{abs}
38	Q_{eva}		P_{low}	=	Tsat(T_7,x_7)
39	Q_{abs}		x_8	=	x_7*m_7/m_8

40	Q_{HXby}	T_3	=	T_1
41	Q_{HXs}	Q_{HXby}	=	$m_3*(h_4-h_3)$
42	Q_{HXc}	Q_{HXby}	=	$m_{15}*(h_{15}-h_{18})$
43	Q_{gen}	Q_{HXby}	=	$U_{HXby}*A_{HXby}*\Delta TM_{HXby}$
44	x_7	m_{15}	=	m_{18}
45	x_8	m_4	=	m_3
46	x_9	m_3	=	m_1*F
47	x_{10}	m_{12}	=	m_{16}
48	P_{high}	m_{14}	=	m_{17}
49	P_{low}	Q_{HXc}	=	$m_{14}*(h_{17}-h_{14})$
50	h_1	Q_{HXc}	=	$m_{12}(h_{12}-h_{16})$
51	h_2	Q_{HXc}	=	$Q_{HXsmax}*\varepsilon$
52	h_3	Q_{HXs}	=	$m_9(h_9-h_8)$
53	h_4	Q_{HXs}	=	$m_7(h_{10}-h_7)$
54	h_5	Q_{HXs}	=	$Q_{HXsmax}*\varepsilon$
55	h_6	m_7	=	m_{10}
56	h_7	m_8	=	m_9
57	h_8	Q_{gen}	=	$h_{13}*m_{13}+h_9*m_9-h_{10}*h_{10}$
58	h_9	Q_{gen}	=	$m_{12}(h_{11}-h_{12})$
59	h_{10}	Q_{gen}	=	$U_{gen}*A_{gen}*\ \Delta T_{gen}$
60	h_{11}	m_{11}	=	m_{12}
61	h_{12}	P_{high}	=	$Psat(T_9,x_9)$
62	h_{13}	T_{13}	=	$Tsat(P_{high},x_{10})$
63	h_{14}	m_{10}	=	m_9+m_{13}
64	h_{15}	T_{12}	=	$Tsat(P_{11})$
65	h_{16}	h_5	=	$h_4*m_4+h_2*m_2$
66	h_{17}	m_7	=	$f(\Delta P)$
67	h_{19}	x_9	=	x_8
68	h_{20}	x_{10}	=	x_7

Appendix IX: Properties calculations

7			
X	0,54839154		Input Concentration kg/kg
T	332,928367	K	Input temperature in K
X	0,20120358		Mol concentration in mol LiBr/mol H20
Tc/(T-T0)	5,78134051		Tc/(T-T0)
H	4507,90921	J/mol	Enthalpy in [J/kg] for saturated water at T times the molar weight for water.
H	4283,58227	J/mol	(1-X)*hw+hc*∑(ai*(0,4-X)^ti ……
H	134,428806	J/kg	Enthalpy recalculated to J/kg

h Sum
0,45806874
-0,31978851
0,0009586
-0,00821623
-0,00111248
0,1321747
-0,39438343
………..

Summation $\sum_{i=1}^{30}(ai * X^{mi}\ (0,4-X)^{ni} * \left(\frac{Tc}{T-To}\right)^{ti})$