Modeling of Heat Transfer

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Abstract

In this master thesis heat transfer in a simulator developed at Solvina AB was investigated. The simulator was created in Dymola and the components to be taken under consideration were condenser, economizer, superheater and evaporator unit including riser tubes and steam drum. When modeling the condenser it was separated into three modules which handled desuperheating, condensing and subcooling respectively. Results showed that the condensing region needed to be discretized into two fractions to handle steeper temperature gradients on the coolant side of the condenser. The evaporator was modeled using temperatures as input to calculate heat transfer between water and flue gases. Dimensioning temperatures were used when calculating UA (a factor that describe the heat exchangers ability to transfer heat) to compensate for fouling in the evaporator and make the model more user-friendly. The economizer was modeled using the approach temperature to dimension UA. The superheater was also modeled using dimensioning temperatures to create a user-friendly model. Results showed that the HRSG model behaves realistic and capture the shrink-and-swell effect in the evaporator.
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# Nomenclature

## Chapter 2

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<td>$sat$</td>
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<td></td>
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<td>$W/m^2, K$ approach</td>
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## Chapter 3.1

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<td>pressure, bar</td>
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1 Introduction

1.1 Background
A simulator is a tool used to simulate numerous types of processes, from economics to thermodynamics. Dynamic modeling and use of simulators allow us to analyze complex systems in different situations. Simulators have many different applications and one is to simulate the process in a power plant. This type of simulator allows operators of a power plant to run tests in the simulator, instead of running tests in the actual power plant. Running tests in a simulator have economic, financial and environmental advantages. By using simulators operators can be exposed to many different types of scenarios which would be dangerous or even impossible to create in a power plant (Colonna & Vanputten 2007).

1.2 Problem Description
Questions investigated in this report include:

- How discretized should a water volume be to capture a steep temperature gradients without slowing down the simulation more than necessary?
- How should a condenser be modeled for being able to handle superheated steam as well as subcooled condensate?
- How can the evaporator be modeled to capture the behavior of a steam drum and which input parameters are relevant?
- How can the economizer be modeled to have relevant and user friendly input parameters?
- How can the superheaters be modeled to have relevant and user friendly input parameters?

1.3 Method
The simulator studied in this work is a combined cycle power plant simulator with a generator output of 390 MW in total active effect. The models to be developed and upgraded in the simulator were limited to the condenser, economizer, evaporation unit including riser tubes and steam drum and the superheaters. The models of these components were enhanced to behave more like a realistic process. The simulator is developed in Dymola – Dynamic Modeling Laboratory – based on the object orientated language Modelica.
2 Theory

2.1 Water and Overall Heat Transfer Coefficient

Water has the ability to change phase from liquid to steam through increased temperature or decreased pressure. The phase shift from liquid to steam is called evaporation. When the process is reversed it is called condensation. This change from liquid to steam and then back to liquid again is frequently used in the combined power plant. Water is heated and evaporates to steam in the evaporator. After being used in the turbine the steam is cooled and condensed back to liquid again in the condenser. The use of the same water in the system cut the costs of water treatment.

The overall heat transfer coefficient, $U$, describes how much heat a surface can transport and is frequently used in heat exchangers. $U$ differs very much depending on the phase of water. When water condenses $U$ can be as high as 10 000 W/m$^2$ K while $U$ can be 95 W/m$^2$ K for superheated steam cooled by water in a heat exchanger. More about the overall heat transfer coefficient and fouling in heat exchangers will follow in Chapter 3.2.

2.2 Combined Cycle Power Plant

Combined cycle power plants are dual cycle systems where the first cycle is used to burn natural gas or oil, and to generate electricity from an electric generator connected to a gas turbine. The second cycle is connected to the first cycles exhaust and generates steam from the hot flue gases. The steam flows through the steam turbine and is condensed in the condenser. The steam turbine is connected to an electric generator. The boiler in the second cycle has responsibility to recover heat energy from the gas turbine exhaust and are therefore named heat recovery steam generator (HRSG) (Figure 7). One of the main advantages with the gas turbine combined cycle is the rapid startup time, from cold plant to full load in about 5-10 minutes (Lammers & Woodruff 1998).

2.2.1 Condenser

The most commonly used condensers in steam power plants are surface condensers with the design of a closed vessel filled with a lot of tubes. This type of condenser is principally a shell and tube heat exchanger. The idea of surface condensers is to keep the steam separated from the cooling water and the reason for this is to reuse high purity water, without the need for water treatment. The amount of cooling water required to condense the incoming steam can be 50 to 80 times the steam flow depending on the cooling water temperature.
The condenser provides a low backpressure at the outlet from the turbine, generally in the range between 0.1-0.4 bar absolute pressure. The vacuum created gives the plant a good economy and maximizes the thermal efficiency of the plant. Another important role for the condenser is to deaerate the condensate in order to minimize the risk for corrosion. The steam entering the condenser is wet, generally 10-15 percents moisture (Lammers & Woodruff 1998). In the simulator studied in this work, the outlet steam from the turbine has 12.1 percent moisture when running in steady state. When saturated steam enters the condenser, temperature on the tube walls $T_w$ are lower than the saturation temperature of the steam $T_{sat}$. This temperature difference causes the steam to condensate at the tube surface (Zhou et al. 2010) (Figure 1). The presence of noncondensable gases in steam can severely reduce the performance of the condenser.

![Surface condenser](image)

**Figure 1: Surface condenser**

According to Minkowycz & Sparrow (1966) the negative effect from noncondensable gases is increasing as the system pressure level decreases. It can be shown that even a small bulk concentration of noncondensable gases gives a large reduction in heat transfer. The reason for the reduction is that vapor which is to be condensed is carried from the bulk to the interface by convective flow. The convective flow also carries the noncondensible gas and given that the noncondensible gas is impermeable to the liquid film, it accumulates at the liquid-gas interface (Zhou et al. 2010).
The buildup of noncondensable gases causes a reduction in the partial pressure of the vapor at the interface. Thus the saturation temperature $T_{\text{sat}}$ is reduced. The reduced saturation temperature lowers the thermal driving force ($T_{\text{sat}} - T_w$) and therefore decreases the heat transfer. Superheated vapor has on the other hand a reducing effect on the reduction of heat transfer caused by presence of noncondensable gases and thus increases the heat transfer compared to pure vapor. For example, Minkowycz & Sparrow (1966) report a reduction of condensation heat transfer of more than 50% for bulk mass fractions of noncondensable gases as small as 0.5% (Minkowycz & Sparrow 1966).

2.2.2 Heat Recovery Steam Generator (HRSG)

A heat recovery steam generators is useful to recover heat from hot flue gases. The temperature of hot flue gases entering the heat recovery steam generator is usually between 500-600°C. Heat recovery steam generators (HRSG) are commonly located next to hot flue gas exhaust of power plants. A HRSG is principally a boiler, without a furnace, that extracts heat from the hot exhaust flue gases from an externally burned fuel. The hot flue gases generally come from a gas turbine or a diesel engine and a common way is to combine the HRSG with a gas turbine. The HRSG contains three major components; an economizer, evaporator and superheater. The HRSG handles two streams, the hot flue gas stream and the cold steam/water stream and are therefore one of the simplest modern boilers. The heat exchanger model can be divided into three units (Figure 2) (Teir 2003).

HRSGs can be categorized into single pressure or multi pressure types. A single pressure HRSG has one steam drum and one pressure level. In multi pressure HRSGs there are two or more pressure levels with the hottest flue gases flowing through the highest pressure configuration first and then the lowest pressure configuration closest to the stack (Figure 3). In the multi pressure HRSGs there are as many steam drums as there are pressure levels.
To increase the efficiency of a power plant the heat of the exhaust gas from the boiler should be recovered. An economizer is located next to the evaporator in the HRSG (Figure 3). The feed water is flowing through tubes and these tubes are surrounded by hot exhaust gases from the combustion. The heated water then enters the steam drum inside the boiler. The highest efficiency is established when the flowing water and the hot gases flow in counter flow. An example of typical improvements of plant efficiency for conventional boilers can be from 74% for the boiler. If the boiler has an economizer the efficiency increases to 82%. This improvement results in lower fuel costs (Lammers & Woodruff 1998). The approach temperature is in this case defined as the temperature difference between the output temperature of the economizer and the saturation temperature in the evaporator (1). The approach temperature is usually designed to prevent steam formation and water hammer in the economizer (Taplin 1991). Normal approach temperatures are between 0-20°C.

\[ T_{sat} - T_{out} = T_{app} \]  

(1)
2.2.4 Evaporator
In the evaporator the steam generation occurs when water evaporates into steam. The heat transfer required is the amount needed to change the phase of water from liquid to steam. The *pinch-point* is the smallest temperature difference between two streams in a heat exchanger system and this occurs in the evaporator (Figure 4). A smaller pinch-point results in increased steam power and a larger area of the evaporator. The pressure drop of the flue gas side is usually 25-40 mbar (Teir 2003).

![Heat load graph for HRSG with a single pressure level. The outgoing cold flue gas meets the cold incoming feed water in the economizer.](image)

2.2.5 Superheater
Steam leaving the evaporator is saturated steam, and to heat the steam further a superheater is required. Superheated steam is steam heated above the saturation temperature. The steam is superheated to provide more energy for the turbine by letting the expansion in the first turbine stage be closer to isentropic expansion without causing condensation. The steam enters the superheater inside tubes with hot gas streaming along the tubes. A common configuration in conventional boilers is two types of superheaters in a row; the first superheater is of convection type and located inside the hot exhaust gas passage. The second superheater is located closely to the furnace and of radiation type (Lammers & Woodruff 1998).

The superheater model developed in this work is of convection type and in the HRSG configuration without supplementary burning there are no superheaters of radiation type. HRSG superheaters are located after the evaporator close to the hot flue gas inlet. Superheaters can be of parallel- or counterflow type and economic considerations must be taken depending on if a low temperature or high temperature superheater is used.
The metal alloy requirement in high temperature superheaters results in higher material costs since hot gas meets hot steam in the high temperature superheater. Therefore the most common configuration in high temperature superheaters is a mixture of parallel- and counterflow because of its economic benefits (Lammers & Woodruff 1998).

2.3 Dymola Modeling

Solvina have been working with simulation in Dymola for many years and therefore a component library called SteamPower has been created to simplify future development of models. The SteamPower library contains among many other things vessels and heat ports. Heat ports are used to transfer information of temperature and heat from one component to another and are for that reason frequently used in the models described in this report. The models in this work are mainly based on components from the SteamPower library.

Dymola is based on the object oriented language Modelica. Modelica have many benefits and is designed to be able to solve differential, algebraic and discrete equation systems. An advantage when entering equations into Modelica is its ability to solve the unknown variables in the equations as long as the number of equations and unknowns are equal. Another advantage is the independent execution order of the equations.

When modeling in Dymola the model is first created in the modeling window of Dymola. The modeling can be made graphically in a drag-and-drop window as in Figure 5 or by entering Modelica programming code as in conventional integrated development environments. Dymola grants the user access to Modelicas extensive medium library from where most medium properties can be extracted. Dymola supports hierarchical models and grants models the ability to inherit properties, which is a great benefit when creating models that interact and inherit properties (Dymola User Manual 2009).
When the model is ready to be simulated this is made in the simulation window (Figure 6). In simulation mode variables and constants can be plotted and analyzed. If simulation of the model fails to run, Dymola will try to give the user information of where and why the simulation went wrong.
3 Process Modeling

The process studied in this master thesis project is a combined cycle power plant with the schematic design of Figure 7. Components taken under consideration in this report are the condenser, economizer, evaporator and superheater.

![Rough schematic of the simulated combined cycle power plant](image)

**Figure 7:** Rough schematic of the simulated combined cycle power plant

3.1 Condenser Modeling

The outlet enthalpy and pressure from the turbine give the information needed to calculate the heat transfer in the condenser. In the condenser steam condenses into liquid as seen in Figure 8.

![Heat transfer during condensation in the condenser where heat is transferred from the hot stream (top) to the cold stream (bottom)](image)

**Figure 8:** Heat transfer during condensation in the condenser where heat is transferred from the hot stream (top) to the cold stream (bottom)
3.1.1 Model Overview

The condenser is divided into three separate zones (Hewitt et al. 1994). The first zone is the desuperheating zone where superheated steam is cooled to saturation temperature. In this zone the hot side wall of the tubes is simplified as dry, i.e. no condensate film is present. At the point where the steam reaches saturation temperature condensation begins. This is the second zone which is called the condensation zone. Here steam condensates and water drips down to the third and last zone, the subcooling zone. In this zone the condensate is subcooled.

The simulation of the condenser has the structure of a flowchart (Figure 9). Depending of the inlet flow state the calculation is handled in different ways. For superheated steam all three processes will be activated and if there is pure liquid entering the condenser, only the last subcooling process will be activated. If the inlet steam is superheated, the area needed to cool the steam to saturation temperature will be subtracted from the condensing area. The liquid level in the condenser is another factor that affects the condensing effectivity. A high level in the condenser leaves a reduced area for condensation and thus reduces the capacity of the condenser.

3.1.2 Model Discretization

The cooling water temperature will increase during the process (c.f. Figure 8). Thus a correct model of the condenser is differentiated in length. Since the model is already solved for time derivatives, the differentiation in length needs to be discrete. The simulation model for the entire power plant is required to be solved in real time. This requirement puts a restriction on the detail level of such a discretization. Hence, depending on how much the cooling water temperature increases, the least number of heat ports for accurate calculations of the heat transfer in the condensing region will change. A higher number of heat ports increase the discretization of the model.
Four models with one, two, five and six heat ports respectively in the condensing section of the condenser (c.f. Section 2.3) were compared and gave the following result (Figure 10). The conclusion is that two heat ports are accurate enough to handle steeper temperature differences of the cooling water. To simulate two heat ports requires half the calculation time compared to five heat ports. Therefore two heat ports is a good compromise between calculation speed and accuracy. In a condensing power plant the increase in cooling water temperature is small. The small temperature difference increases the accuracy of the calculation using two ports since the increase in temperature on the coolant side can be regarded as linear.

The temperature graph below (Figure 10) clarifies how the condenser behaves when used as a district heating condenser. In this figure the temperature of cooling water is increased by almost 45°C. Discretization of the condenser with just one heat port (blue) fails to capture the behavior of cooling water temperature. A model of two heat ports (red) on the other hand captures the behavior properly.

![Figure 10: Temperature rise of cooling water in the condenser depending on the number of heat ports (left) and increase in calculation time depending on heat ports used (right).](image)

The number of heat ports has a big impact on the liquid level in the condenser. The step from one to two heat ports make a large improvement to the accuracy of the condenser level, but from two heat ports to six heat ports the improvement is much smaller (Figure 11). Note that accuracy for the system is lower for larger temperature differences on the coolant side.
Figure 11: The number of heat ports compared to liquid level in condenser. More than five heat ports do not increase accuracy of the model.

3.1.3 Inflow

The condenser model must be able to handle different inflow qualities, from superheated steam to subcooled water.

Cooling of superheated steam will affect the area available for condensation and this must be included in the heat transfer calculations.

To calculate heat flow in a heat exchanger the following equation is used

\[ \dot{Q} = UA \Delta T_{lm} \]  

where \( \Delta T_{lm} \) for a counter flow heat exchanger is

\[ \Delta T_{lm} = \frac{(T_{hl} - T_{v}) - (T_{h0} - T_{ct})}{\ln \left(\frac{T_{hl} - T_{v}}{T_{h0} - T_{ct}}\right)} \]  

Equation 2 and 3 are taken from Incropera et al (2007).

If the incoming enthalpy is larger than the saturation enthalpy, i.e. \( h_{in} > h_{sat, vap} \), the heat transferred and area required in the desuperheating region are calculated as follows

\[ \dot{Q}_{SH} = U_{SH} A_{SH} \Delta T_{a} = \dot{m}_h (h_{in} - h_{sat, vap}) \]
where $\Delta T_a$ is the arithmetic mean temperature

$$\Delta T_a = \frac{T_{hi} + T_{ho}}{2} - \frac{T_{co} + T_{ci}}{2}$$  \hspace{1cm} (5)

For the condensing part the heat transfer is calculated using the remaining area with the desuperheating and subcooling areas subtracted.

$$\dot{Q}_{c1} + \dot{Q}_{c2} = U_c (A_c - A_{sh} - A_{sc}) \Delta T$$  \hspace{1cm} (6)

The condensate is finally subcooled in the subcooling zone at the bottom of the condenser, provided that the condensate level $> 0$

$$\dot{Q}_{sc} = U_{sc} A_{sc} \Delta T$$  \hspace{1cm} (7)
In Figure 12 the condenser is tested during different operating conditions. The medium entering the condenser is changed from subcooled liquid to superheated steam during the test. This change in properties of the inflow affects the distribution of heat flows in the condenser. For a subcooled inflow the condenser heat transfer takes place in the subcooling zone. For a saturated inflow the dominating heat flow is steam condensation. Finally, for a superheated inflow some of the condenser area is blocked for condensation due to desuperheating of the incoming steam. This can be seen as the condensing heat flow decreases rapidly.

Figure 12: Hot inlet stream (black) changing from subcooled to superheated affects the distribution of heat flows in the condenser. When the inflow contains steam condensation heat flow start to increase (blue and red). When the hot inflow turns superheated area is blocked for condensation and desuperheating heat flow increase (green). Note that the x-axis represents time in seconds.
3.2 Modeling of a Heat Recovery Steam Generator (HRSG)

The heat recovery steam generator model was separated into three models; economizer, evaporator and superheater. The separation made future modeling more flexible and simplified the development of the mentioned models.

3.2.1 Economizer Modeling

The economizer was modeled from two different approaches. The first approach used the area and heat transfer coefficients input parameters and the second was based on dimensioning conditions to calculate the heat transfer area. Both modeling approaches used two heat ports following the same argumentation as for the condenser model (Figure 13). The first approach with area and heat transfer coefficient as input parameters does not take into account the fouling or the approach temperature of the economizer. Dimensioning- or operating conditions are most likely known and are therefore user friendly inputs for the model. The second approach of the model was therefore to calculate the product of heat transfer coefficient and area, $UA$ that qualifies for given input parameters

$$\dot{m}_{fw}(h_{out} - h_{in}) = UA \left( \frac{T_{hi} - (T_{sat} - T_{app}) - (T_{ho} - T_{ci})}{\ln \left( \frac{T_{hi} - (T_{sat} - T_{app})}{T_{ho} - T_{ci}} \right)} \right)$$

(8)
Figure 13: Economizer was dimensioned using the approach temperature. This figure shows a model with two heat ports.

An economizer is designed for preheating water but is also constructed to allow boiling of water. If boiling occurs in the economizer, the outlet flow will contain fractions of steam. If the economizer volume contains steam, the level will decrease and this will affect the heat transfer. Steam has a lower $U$ than water and therefore heat transferred from flue gases to steam will be smaller than from flue gases to water. Note that boiling in the economizer will give a less accurate $U_{vap}A$ due to dependency between $U_{liq}$ and $A$. In the following equation, $level$ is percentage of the total height.

$$
\dot{Q} = \frac{(U_{liq} \cdot A \cdot \Delta T)_{level}}{100} + \frac{(U_{vap} \cdot A \cdot \Delta T)(1 - level)}{100}
$$

(9)
In Figure 14 the outlet temperature of the economizer is plotted. The approach temperature in this simulation was set to 3°C and the pressure of the feed water stream was 10 bar. Note that saturation temperature at 10 bar is 179.88°C (Wester 2008). As seen in the figure the actual approach temperature (red) is 3.5°C while the desired approach temperature is 3°C. The other curves show how the economizer behaves when it is operating outside the dimensioning conditions.

Figure 14: Outlet temperature of economizer. Approach temp is set to 3°C and the pressure is 10 bar which gives a saturation temperature of 179.88°C. The simulation is run with different hot inlet temperatures.
3.2.2 Evaporator Modeling
The evaporator can have an inflow of either subcooled water or steam and water at the saturation temperature. During the evaporation process the water and steam temperature is constant while the flue gas temperature is decreasing. Therefore discretization in length is necessary to gain accurate results of the heat transfer. In the evaporator, three heat ports have been used. Two of the heat ports were placed in the saturated region of the evaporator and the third heat port will be activated if the approach temperature is greater than zero. The area needed to heat the inlet fluid to saturation is subtracted from the available area for evaporation of water. To avoid problems during simulation as a result of division with zero, the logarithmic mean temperature is replaced with the arithmetic mean temperature.

3.2.2.1 Modeling Heat Transfer in Evaporator
The evaporator was modeled in two different ways. The first approach was to let the area of the evaporator govern the amount of heat that could be transferred to the fluid. The advantage with this approach is that the model becomes much easier to adjust. The downside is that the model doesn’t take into account the pinch-point or the fouling of the evaporator. In this model the fouling must be compensated.

To compensate for fouling the fouling factor \( R_f \) must be known:

\[
\frac{1}{U_{\text{Dirt}}} = \frac{1}{U_{\text{Clean}}} + R_f
\]  

(10)

For an evaporator that has been run for a while the heat transfer surfaces get fouled and this has a negative effect on the overall heat transfer coefficient \( U \).

\[
U_{\text{Dirt}} < U_{\text{Clean}}
\]  

(11)

Because an evaporator with a fouled surface have a lower capacity than one that is clean it can be difficult to insert the right \( U \) in the model. To get an accurate simulation \( U \) must be close to the correct value.

The heat flow is in both cases calculated with the arithmetic mean temperature

\[
UA \left( \frac{T_{hi} + T_{ho}}{2} \right) - T_{\text{sat}} = \dot{Q}
\]  

(12)
Another approach is to calculate the needed area and mass flow of flue gas from a few given parameters, including the pinch-point. If a real case is to be modeled $U$ is unknown for that specific heat exchanger because of fouling. It is often possible to measure the temperatures in the actual HRSG and the dimensioning pinch-point is probably known. Therefore this model is easier to get accurate results from. By calculating the area from the given parameters the factor $UA$ will be close to the real case evaporator. If $U$ has decreased because of fouling the calculated $UA$ will compensate for that. The $UA$ in the actual model will on the other hand change over time and the compensation for fouling is therefore limited.

The compensation for fouling occurs when the needed area is calculated

$$U_{\text{guessed}} \cdot A_{\text{calculated}} = (U \cdot A)_{\text{actual model}}$$  \hspace{1cm} (13)

The following equation gives the balance between the heat flow needed to evaporate the incoming water, and the area required. Note that the only unknown in this equation is the area

$$UA \left( \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \left( \frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}} \right)} \right) = \dot{m}_{fw}(h_{\text{sat.watp}} - h_{\text{in}})$$  \hspace{1cm} (14)

Where the pinch-point, PP, is defined in the saturated region:

$$PP = T_{ho} - T_{ci}$$  \hspace{1cm} (15)

Figure 15: Heat transfer in the evaporator for feed water heated by flue gases when a small part of the area ($A_1$) is used to heat subcooled water to saturation temperature.
If incoming feed water is subcooled, some of the available energy is needed to pre-heat the water to the saturation temperature. This means that the area in the evaporator is used both for evaporation and pre-heating of feed water. In calculation of the evaporator area this phenomenon has to be taken into account. This can be made in two different ways. The first approach is to use some of the available area to heat the incoming water to saturation temperature (Figure 15). Another way is to use the steam produced in the evaporator to heat the incoming feed water to saturation temperature (Figure 16).

![Figure 16: Heat transfer in the evaporator when feed water is heated by steam](image)

In the second case there is a jump in inlet temperature from subcooled to saturation temperature instantly. The outlet temperature of the flue gas remains the same in both cases. The results and comparison between the two approaches can be seen in Figure 17.

The difference between the two methods is very small and the biggest difference can be seen in the pinch-point of the system. The dimensioning pinch-point is set to 10 K and this is achieved in the model where a small part of the area is used to pre-heat the incoming water. In the model where steam is used to heat feed water the calculated pinch-point is unreliable because the exact flue gas temperature at the point of saturated water is not calculated (Figure 16). The actual pinch-point is equal in both cases because heat flow, stream temperatures and mass flows are equal in the two models. It is hard to make a clear choice between the two cases according to the results shown. Therefore both cases is said to work satisfactorily.
Figure 17: Difference between the two models where energy to heat the incoming feed water is taken from the steam (blue) in the evaporator or from the outgoing flue gas (red). At time 600 the incoming flue gas temperature increase from 300°C to 327°C. The first graph show how the level is affected by the increased flue gas temperature. The second graph plots the flue gas temperature out of the evaporator. The last graph plots the pinch-point in the evaporator.
3.2.2.2 Modeling Shrink-and-swell

In Figure 18 a schematic picture of the boiler is shown. When heat, $Q_{in}$, is transferred to the riser tubes it causes water to boil. The saturated steam produced rises through the riser tubes and causes a natural circulation in the riser-drum-downcomer loop. To create a realistic model of the steam drum in the evaporator, it must take the shrink-and-swell effect into consideration. To be able to capture this effect the distribution of steam and water in the system must be accounted for. The redistribution of steam and water causes the shrink-and-swell effect and this is an important phenomenon to describe in order to simulate a correct drum level behavior. When the steam valve opens the level in the drum will increase because drum pressure drops. The pressure drop causes steam bubbles below the liquid level to swell, resulting in initially increased level in the drum. Increased flow of feed water into the drum has opposite effect. All different parts of the system which are in contact with the saturated fluid will be in thermal equilibrium. This result in energy that is stored in the fluid is released or absorbed quickly when the pressure changes. Since energy is stored and released quickly all different parts of the boiler can be assumed to have the same temperature. This is the reason why models of low order can capture the dynamics in the boiler (Åström & Bell 2000).

The steam drum is modeled using the model created by Åström & Bell (2000). The model captures the shrink-and-swell effect in the drum very well and simplifies the complex behavior of the drum and has few input parameters. The model includes Equations 16-26. Equation 27 is from earlier published work (Åström & Bell 1988). The model created is pressure controlled and will therefore not regulate flows by using level.

Global mass and energy balances is first calculated

$$\frac{d}{dt} \left( \rho_s V_{vap,t} + \rho_{liq} V_{liq,t} \right) = \dot{m}_{fw} - \dot{m}_{vap} \quad (16)$$

$$\frac{d}{dt} \left( \rho_s V_{vap,t} h_{vap} + \rho_{liq} V_{liq,t} h_{liq} - PV_t + m_{met} c_p T \right) = Q + \dot{m}_{fw} h_{fw} - \dot{m}_{vap} h_{vap} \quad (17)$$

Figure 18: Increase in steam out from the drum will cause an increase in the drum level because of the pressure drop and swelling of steam bubbles below liquid level.
The total volume of the system is the total volumes of drum, downcomers and risers

\[ V_t = V_d + V_{dc} + V_r \]  \hspace{1cm} (18)

The total liquid volume in the system can be calculated from the total mass of the liquid-vapor mixture since all densities are known

\[ m = V_{liq,t} \cdot \rho_{liq} + (V_{tot} - V_{liq,t}) \rho_{vap}. \]  \hspace{1cm} (19)

Liquid and steam volumes in the drum are given by the total liquid volume in the system

\[ V_{liq,t} = V_{liq,d} + V_{dc} + (1 - a_r)V_r \]  \hspace{1cm} (20)

\[ \frac{d}{dt} (\rho_{vap} \cdot V_{vap,d}) = a_r \cdot \dot{m}_r - \dot{m}_{vap} \]  \hspace{1cm} (21)

and the total steam volume in the system is

\[ V_{vap,t} = V_{vap,d} + a_r \cdot V_r. \]  \hspace{1cm} (22)

Mass flow in the downcomer can be written as

\[ a_r \cdot V_r (\rho_{liq} - \rho_{vap}) = \frac{k \cdot \dot{m}_{dc}^2}{2}. \]  \hspace{1cm} (23)

Volumetric quality \( a_v \) describe how much a fluids volume expands according to increased temperature. The following equation (24) describe the relation between \( a_v \) and mass balance over the riser

\[ \frac{d}{dt} (\rho_{vap} \cdot a_v \cdot V_r + \rho_{liq} (1 - a_v) V_r) = \dot{m}_{dc} - \dot{m}_r. \]  \hspace{1cm} (24)

Steam quality in riser outlet \( a_r \) is dependent of the volumetric quality and can be written as

\[ a_v = \frac{\rho_{liq}}{\rho_{liq} - \rho_{vap}} \left( 1 - \frac{\rho_{vap}}{(\rho_{liq} - \rho_{vap})a_r} \ln \left( 1 + \frac{\rho_{liq} - \rho_{vap}}{\rho_{vap} a_r} \right) \right). \]  \hspace{1cm} (25)
Energy balance over the riser is

\[
\frac{d}{dt} \left( \rho_{\text{vap}} \cdot h_{\text{sat,vap}} \cdot \alpha_v \cdot V_r + \rho_{\text{liq}} \cdot h_{\text{sat,liq}} (1 - \alpha_v) V_r + \dot{m}_r \cdot c_{p,\text{met}} \cdot T \right) = \dot{Q} + \dot{m}_{dc} \cdot h_{\text{sat,liq}} - \\
\alpha_r \cdot \dot{m}_r (h_{\text{sat,vap}} - h_{\text{sat,liq}}) - \dot{m}_r \cdot h_{\text{sat,liq}}
\]

(26)

Finally, the drum level \(L\) can be calculated where \(A_d\) is area of the liquid surface in the steam drum

\[
L = \frac{V_{\text{liq,d}}}{A_d} + \frac{\alpha_v \cdot V_r}{A_d}
\]

(27)

The model was tested with a rapid change of heat flow to the evaporator. The initial heat flow to the evaporator was 40 MW with a change of \(\pm10\) MW. The increase of heat flow caused level in steam drum to initially rise and then drop (Figure 19).

![Figure 19: Increase in heat flow causes a swell effect that initially raises the liquid level in the drum.](image)

Because models in the simulator is pressure controlled the level is of less importance out of a controlling point of view, but is however interesting to capture the behavior of the drum.
Heat flow to the evaporator was decreased by 10 MW and resulted in an initial decrease in drum level followed by a rise (Figure 20).

Figure 20: Decrease in heat flow to the evaporator causes a shrink effect that result in initial drop of liquid level.
3.2.3 Superheater Modeling

The superheater was modeled out of two approaches. The first model was area as input, and was used for calculation of temperatures and heat flow based on the input area.

\[ \dot{Q} = UA \Delta T_a \]  \hspace{1cm} (28)

The arithmetic mean temperature was used for numerical reasons as in Section 3.2.2.

\[ \Delta T_a = \frac{T_{hi} + T_{ho}}{2} - \frac{T_{ci} + T_{co}}{2} \]  \hspace{1cm} (29)

Analogue to the other heat exchanger models, also this model is hard to adjust and a more user-friendly approach is to use temperatures as input and then calculate \( UA \) required fulfilling the conditions. Hence, in the second approach, the area necessary is calculated out of the logarithmic mean temperature \( \Delta T_{lm} \). The reason for using the logarithmic mean temperature here is that it is calculated only once using the input design temperatures from the user.

\[ \Delta T_{lm} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{ln\left(\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}\right)} \]  \hspace{1cm} (30)

Because the model only uses the input temperatures for calculating the area, possibility for division with zero is not a problem. The benefit on the other hand is a more accurate \( UA \). In this model two heat ports were used for increased accuracy (Figure 21).
Figure 21: Heat transfer in a superheater with two heat ports. The superheater was dimensioned using temperatures.

Input temperatures in the simulation shown in Figure 22 is $T_{ho}=540^\circ C$, $T_{co}=480^\circ C$ and $T_{hi}=640^\circ C$. Output temperatures is $T_{co}=488^\circ C$ and $T_{ho}=540,5^\circ C$. As seen in Figure 22 the models calculated $UA$ results in output temperatures close to the dimensioning.

Figure 22: Simulation of the superheater where the hot stream outlet temperature (red) and cold stream outlet temperature (blue) is plotted
The superheater is also tested with other operating conditions than the dimensioning conditions (Figure 23). Steam output temperature of the superheater drops when the hot inlet flue gas temperature drops.

Figure 23: The superheater is simulated with different operating conditions and the outlet temperature of steam is shown. The dimensioning condition is the pink graph. Output temperature of steam drops when flue gas inlet temperature drops.
4 Conclusions
In this master thesis a condenser, economizer, evaporator and superheater were enhanced. The heat transfer in the components were investigated and improved.

The condenser should be able to handle superheated steam and then subcool the condensate. The best approach when modeling the condenser was to separate it into three models designed to handle desuperheating, condensation and subcooling. The reason for this separation of volumes is that the medium undergoes three very different phases and Dymola handles volumes as perfectly mixed. Studies and tests show that discretization of the condensing region in two fractions capture the dynamic as well as static behavior well in terms of temperature rise on coolant side of the condenser. Discretization of the condensing region in further fractions does not increase the accuracy significantly according to the increased calculation time. This conclusion was applied also on the other models in terms of discretization.

The evaporator was created out of two different approaches and differences between them are very small. One of the models uses a small fraction of the evaporator area to transfer heat from the flue gas stream to heat incoming water to saturation temperature. The other model were steam is used to heat the incoming feed water to saturation temperature seems like the more realistic model but because of the small differences there is no clear choice between the two. Calculation time is slightly shorter in the model were steam is used to heat the feed water, but the pinch-point is calculated with lower accuracy. Using area and heat transfer coefficient as inputs in the evaporator model would not take into account the fouling of heat transfer surfaces and therefore a model were temperatures is used as input give a better and more accurate simulation. $U$ for the specific model is most likely unknown and to calculate $UA$ out of temperatures give a realistic simulation and more accurate output values. When the model was compared with Åström & Bell’s (2000) results the shrink-and-swell effect is smaller in the drum modeled in this work. The differences between the models were 5 cm change in level for Åström & Bell (2000), while the model designed in this thesis only had 1 cm change in level for the same dimensioning and operating conditions. The reason for this is because the drum does not take into account steam volume below liquid level in the drum. This is something that Åström & Bell implemented in their model. In this model the drum level is of less importance because flows are pressure controlled.
When simulating the economizer a model which takes into account the approach temperature was created. The approach temperature is of importance when simulating the HRSG because this affects the evaporator capacity. Another advantage by using temperatures is as mentioned before the often unknown heat exchange coefficient. The economizer was discretized into two fractions to increase the accuracy. According to the results when modeling the condenser the economizer was not discretized further. Discretization of a volume increase accuracy of the model but also increases calculation time. Therefore it is important to limit the amount of discretization in a model to prevent long calculation times.

The superheater also had temperatures as input to calculate $UA$. This resulted in a model that compensates for fouling and is easy to implement and dimension. There can however be difficulties in initialization of the model.
5 Future improvements

The condenser can be improved to take into account the presence of noncondensable gases. As described in this report, the presence of noncondensable gases has a large impact at the capacity of the condenser. To implement a vacuum pump that can be shut down to see how the condenser reacts when the pressure and amount of noncondensable gases is built up is one suggestion of further improvements.

Other improvements can be made in the steam drum of the evaporator. The drum used today does not describe the amount of steam below the liquid level in the drum, and this affects the level in the drum. If the steam volume below the liquid level in the steam drum would be taken into account the shrink-and-swell effect should be bigger, with higher level differences as a result.

The superheater is today somehow hard to initialize and dimension correctly. Input temperatures are not always intuitive and because of this it can be difficult to run a stable simulation. Ways to handle the problematic initialization is to let the user know when input temperatures are impossible or unlikely to fulfill during the present conditions.
References


