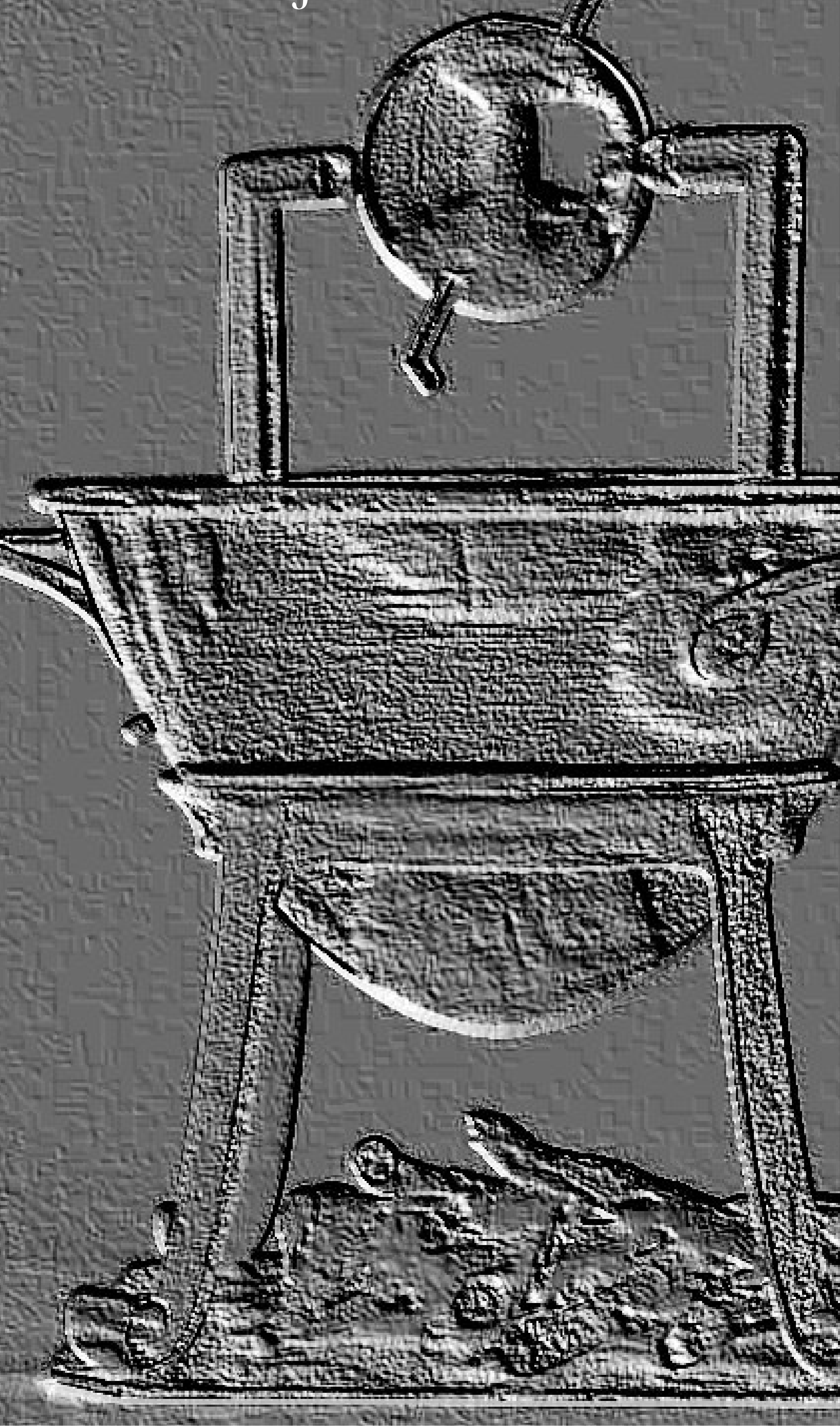
Óttar Kjartansson



Investigating the potential of ultra-super critical coal fired power plants

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Abstract

The main aspect of this project is to describe and model the water/steam cycle in super critical pulverised coal fired power plants. Then use that model to investigate the potential of a **ultra**-super critical pulverised coal fired power plant.
The Unit 3 in Nordjyllandsværket is chosen as base for the modeling of a steam/water cycle in super critical power plant because of the high thermal efficiency and it is regarded for many as the state of the art super critical coal fired power plant.
A model is developed and it is concluded that the model is fairly accurate, when comparing the solutions from the model with design data from unit 3 in Nordjyllandsværket. The model is also verified by investigating the tendency for the solutions when altering the temperature and pressure inputs. The tendency is as expected.
A investigation of the potential of ultra-super critical steam data is performed. The tendency in the solutions is as expected and shows increased thermal efficiency when the temperature and pressure of the steam is increased. By raising the temperature from 580 °C to 760 °C and the pressure out of the high pressure feedwater pump from 33 MPa to 42 MPa, the thermal efficiency improves by about 4%.
This improved efficiency is in accordance with resent literature on the subject and supplements further to the verification of the model.

Preface

This report has been written under the *Fluids and Combustion Engineering* (FACE), graduate programme, 10th semester at the *Institute of Energy Technology - AAU*.

At his 8th semester, the author of this report was in a project group that did a investigation of a rotary regenerator at Vattenfalls power plant Nordjyllandsværket [Kjartansson and Nielsen 2007]. At his 9th semester, the author had a internship at the same power plant [Kjartansson 2007]. This project is greatly inspired from these two semesters.

The report consists of three parts; the main report, a set of appendixes and a CD-rom. On the CD-rom an electronic version of the report and a copy of the numerically developed model can be found.

Tables and figures have been enumerated with the number of the chapter and the number of the figure in that chapter, *e.g.* "Figure 3.1". This figure will be the first figure in chapter 3. Appendixes are indicated with letters, *e.g.* "Appendix A".
Citations in the report are made in squared brackets, and give the authors last name and year of publishing, *e.g.* [Jensen 1999].

This report is typeset in LaTeX and compiled with LEd as editor.

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Nomenclature

Latin Letters

h	Enthalpy	kJ/kg
$\dot{m}$	Mass flow rate	kg/s
m	Mass	kg
P	Pressure	kPa
$\dot{Q}$	Heat transfer rate	kJ/s
s	Entropy	kJ/kg·K
T	Temperature	°C
$\dot{W}$	Electric energy rate	MW

Greek Letters

η	Efficiency	-

Subscripts

b	Boiler
$cond$	Condenser
el	Electricity
fw	Feedwater
gen	Generator
hp	High pressure
i	Inlet
o	Outlet
reh	Reheating
$turb$	Turbine

Abbreviations

CCGT	Combined cycle gas turbine
CHP	Combined heat and power
EES	Engineering Equation Solver
HHV	Higher heating value
HP	High Pressure
LHV	Lower heating value
LP	Low Pressure
IAPWS	International Association for the Properties of Water and Steam
IGCC	Integrated gasification combined cycle
IP	Intermediate Pressure
NJV3	Nordjyllandsværket unit 3
PFBC	Pressurised fluidised bed combustion
USC	Ultra-supercritical
VHP	Very High Pressure

1 Introduction

1.1 Problem orientation
1.2 Problem definition
1.3 Problem delimitation

In this chapter, the background for the project is introduced. The energy situation in the past and the future is discussed with focus on production of electricity from burning coal.

1.1 Problem orientation

As our society develops, the demand for energy is constantly growing. The energy supply is expected to be constant and reliable. There is also a requirement that the energy is environmental friendly or as little polluting as possible.
In Europe the energy comes mainly from two sources; from nuclear power plants and burning fossil fuels (coal, oil and natural gas), but there is increasing focus on renewable energy sources such as wind power and solar energy.
For environmental reasons, nuclear power and fossil fuels are not popular as energy source, but it seams like the the western world is stuck with it. At least until the renewable energy has developed to a stage, where it can provide enough energy so the other less environmentally friendly methods can be taken out.

In Europe, most countries (inclusive Denmark) have signed the Kyoto protocol. Those countries have agreed to cut down CO_2 equivalent emissions by 8%, expressed in relation to emissions in a predefined base year or period. This reduction of CO_2 equivalent emissions is supposed to happen in the period of 2008 to 2012. [Kyo 2007]

In Denmark, the majority of the electricity production is based on coal fired power plants [Dal and Zarnaghi 2007]. Efforts are constantly made to improve the efficiency of the plants. In the period from 1990 to 2004 has the total efficiency of the Danish heat and power plants risen from 60% to 73% [Pedersen 2005]. Figure 1.1 gives overview over how the energy sources for production of electricity in Denmark is divided between oil, natural gas, coal, waste burning and self sustainable sources from 1990 to 2006. Figure 1.2 gives overview over how the energy sources for production of electricity in Denmark are assumed to develop until 2030.
In Europe, solid fuels (hard coal and lignite) are projected to decrease somewhat by 2020 and to come back almost to the level it was in 2006 by the year 2030. [Mantzos

and Capros 2006]

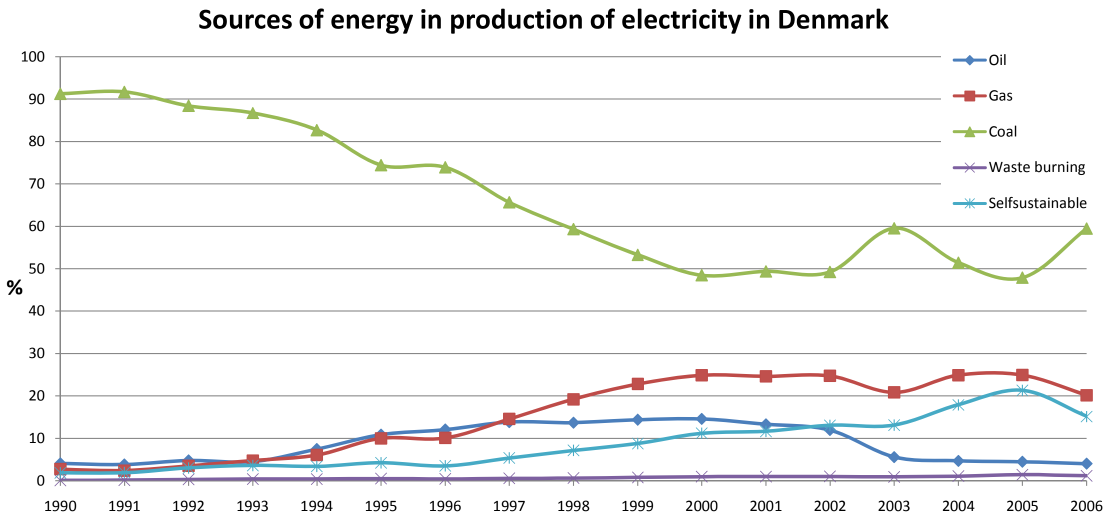

Figure 1.1: *Overview over the main sources of energy in production of electricity in Denmark.[Dal and Zarnaghi 2007]*

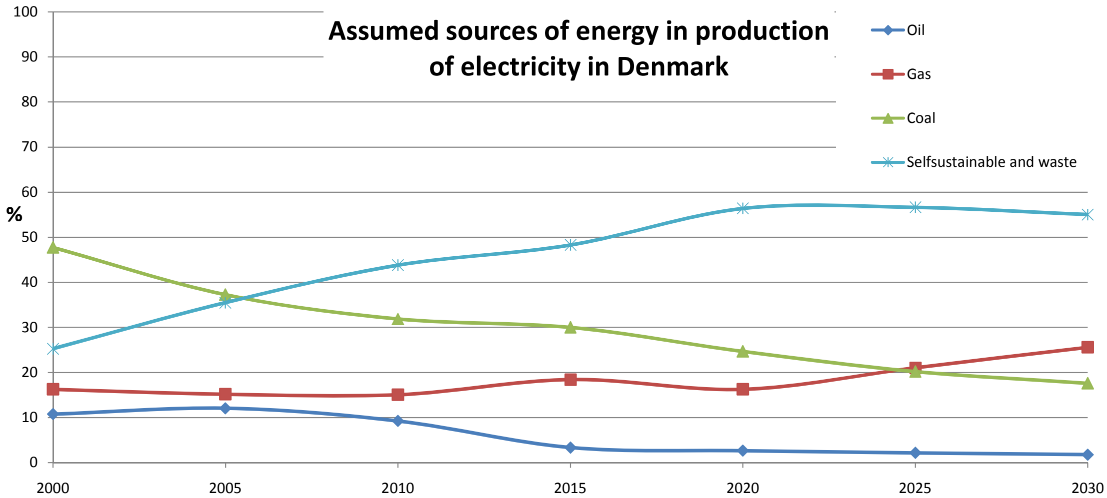

Figure 1.2: *Overview over assumed main sources of energy in production of electricity in Denmark.[Mantzos and Capros 2006]*

The advantage of using coal as fuel is e.g. that it can be acquired from many locations around the world. This becomes valuable when compared to oil, of which a great deal comes from areas where political instability has been long lasting.
A big disadvantage of coal as a fuel is that the use of it involves a large amount of CO_2 emissions. Carbon dioxide is one of the greenhouse gases that contributes to the greenhouse effect.

At the present stage, it is very likely that coal vil continue to be number one energy supply in production of electricity in Denmark. Therefore it is of outmost importance to improve the efficiency of coal fired power plants and reduce the CO_2 emissions. The tendency is clearly moving in that direction.

1.2 Problem definition

The main aspect of this project is to describe and model the water/steam cycle in super critical pulverised coal fired power plants, and to investigate the future potential for optimization of the process.

1.3 Problem delimitation

This project will have its main focus on the water/steam cycle in a super critical power plant. More specific, it will be based on how the efficiency can be increased by improving the steam data. The areas of interest within this project are as follows:

- Create a model of water/steam cycle in a super critical pulverised coal fired power plant
- Utilise the model to investigate how a change into a ultra super critical power plant, affects the efficiency.

2
Steam temperature and pressure

2.1 Introduction
2.2 Improvements on boiler materials
2.3 Comparing efficiencies
2.4 Summary

2.1 Introduction

The increased cost of fuel along with the need to reduce CO_2 emission, has provided an additional incentive to increase the efficiency of power plants. This chapter contains a literature study which has focus on what others have worked on, according to improvements on super critical pulverised coal power plants.

2.2 Improvements on boiler materials

When improving the efficiency of a super critical pulverised coal power plant, the main focus has been on raising the temperature and pressure of the steam leaving the boiler into the turbine. But this is not very simple. At temperatures over 400°C, there is increased risk of damage on the piping from creep, cycle fatigue, creep fatigue and erosion-corrosion.[Wilcox 1992] The upper limit in temperature and pressure in super-critical power plants has been around 600°C$/30MPa$. But there is a great deal of effort made to push these limits higher.

R. Viswanathan, K. Coleman and U. Rao have done a study on materials needed for the construction of the critical components of ultra-supercritical coal-fired boilers capable of operating with 760°C$/35MPa$ steam. It is estimated that by raising the temperature and pressure to these levels, will increase the efficiency from the average of 37% to approximately 47% for a double reheat configuration in the USA. [Viswanathan et al. 2006]

The European Union financed a AD700 project which Elsam coordinated. This project involves about 40 companies representing actors in the European power industry. The aim of the project is to raise the efficiency of a USC power plant above 50%. [Bugge et al. 2006]

At Oak Ridge National Laboratory and University of Cincinnati, long-term tests of mechanical properties of nickel-based alloys have been performed. These tests are made with temperatures up to 800°C. [Shingledecker et al. 2006]

In China, experiments have been made on a nickel-based super-alloy. The purpose of this experiment was to find a suitable alloy for application in USC superheater tubes above 750°C. [Zhao et al. 2006]

F. Tancret and H.K.D.H. Bhadeshiab have worked on a nickel based super alloy that would be affordable for power plant applications. The design requirements are a lifetime of 100000 hours at 750°C under 100*MPa*. The design requirements also included forgeability, weldability, corrosion resistance, and microstructural stability over long exposure at service temperature. This work resulted in a design of a nickel based alloy. A bar of this alloy has been fabricated and tested. Figures 2.1 and 2.2 show both modelled and measured results from the tests.

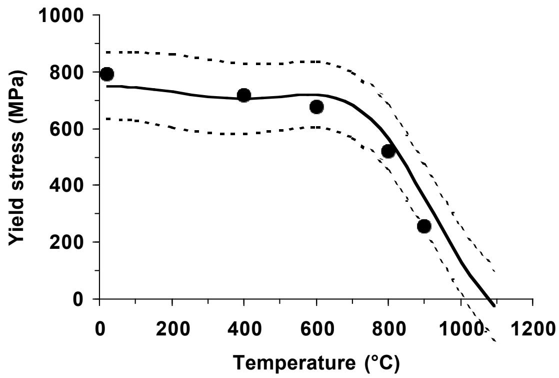

Figure 2.1: *Evolution of yield stress as a function of temperature: solid circles indicate measurements; solid line indicates mean Gaussian processes predictions; broken lines indicate predicted error bounds.[Tancret and Bhadeshia 2003]*

Notice in figure 2.1 how the strength is at first insensitive to temperature. It is first at about 700°C the strength begins to drop.
In figure 2.2 the circles represent five tests for creep at 320, 290, 260, 230 and 200 *MPa*. The test at 200 *MPa* was unloaded before fracture but the rupture time at 200 *MPa* is extrapolated to about 5000 hours. The measurements are all well over the 100 *MPa* limit and the modelling proposes that the alloy should withstand 100 *MPa* at 750°C.
According to [Tancret and Bhadeshia 2003], is the target of a lifetime of 100000 hours at 750°C under 100*MPa*, well attainable.

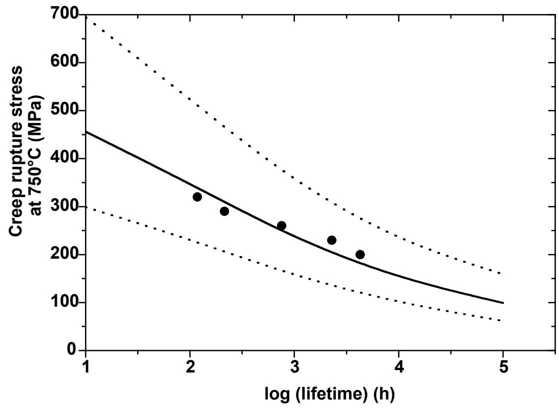

Figure 2.2: *Creep rupture stress as a function of lifetime. Circles: measurements; solid line: mean Gaussian processes predictions; dotted lines: predicted error bounds.[Tancret et al. 2003]*

2.3 Comparing efficiencies

In the work of improving the steam data, efficiency is the key word. But caution must be taken when comparing efficiencies between power plants. The difference in efficiency between American and European power plants, lies often around 3% in favour of the European ones. How to explain these differences is not a simple task as it depends on many factors. There are factors such as the quality of the coal, age of the power plant, cooling facilities, reheat or no reheat and many others. In the following quotation, American scientists explain the differences in the efficiencies of American and European power plants from their point of view.

> *"Plant efficiency is a function of a variety of factors such as coal quality, auxiliary power needs, condenser pressure and discharge temperature, flue gas exit temperature (function of coal sulfur content) assumed heating value of fuel (higher heating value (HHV) vs. lower heating value (LHV)), number of re-heat stages, steam turbine design and many other factors. Due to differences in these factors, European plants generally cite values of efficiency that are higher by 3–5% points compared to US plants for similar steam parameters. The US practice is to express efficiency in terms of HHV, while the European practice is to express efficiency based on LHV This difference alone causes the latter to be higher by about 2% points. Sulfur content of the European plants is generally lower enabling flue gas discharge at lower temperatures. Auxiliary power consumption is generally lower in European plants. Condenser discharge pressures and temperatures are often much lower. Turbine efficiency values assumed in the calculations are higher compared to US plants. All of these factors mentioned above further exacerbate the differences in calculated efficiencies. In comparing efficiencies, these considerations must be kept in mind, since a direct comparison is often not possible."*[Viswanathan et al. 2006]

When the literature on the subject is studied, it must be kept in mind that people often tend to favourite their ”home” technology, but whatever the reasons are for the difference in the efficiency are, it is not simple to compare the efficiency of two power plants.
Because of this difficulties in comparing efficiencies between power plants, it is probably more appropriate to focus on how each power plant can be improved rather than try to figure out which one is ”best”. It must though be kept in mind that competition is often a substantial drive in developing new techniques.

2.4 Summary

To sum up, there is a lot of work done in the field of increasing the efficiency of a super critical pulverised coal fired power plant. From these studies it seems likely that the boilers in the nearest future will be build of a nickel-based super-alloy. The maximum temperature/pressure will be around 700°C$/35MPa$ and about 760°C reheat temperature. Higher reheat temperature is possible because of lower pressure. This improvement of the steam data should be noticeable in better efficiency. One should though be careful in comparing efficiencies between power plants, because of different situations and fuel.

3
Nordjyllandsværket

3.1 Introduction
3.1.1 Nordjyllandsværket
3.2 Unit 3
3.3 Steam/water system
3.4 Combustion air and flue gas
3.5 Summary

3.1 Introduction

This chapter contains a brief description of the power plant Nordjyllandsværket Unit 3 which is the base for the model in this project. The main technical parameters for the unit are discussed, followed by explanations of the water/steam cycle and the air and fluegas system.

3.1.1 Nordjyllandsværket

Nordjyllandsværket is owned by Vattenfall, a Swedish multinational energy company. Vattenfall has operations in Denmark, Sweden, Poland and Germany and produces and delivers electricity and heat to its customers.

Nordjyllandsværket is a combined heat and power plant which produces electricity to the Danish power grid and district heating to Aalborg and part of the minor local towns in the area. There has been a power plant on the location since 1967 and in 1992 decision was made to build a new supercritical coal fired unit. The building of this unit called Unit 3, was finished 1998 and is now the main unit in the power plant, see picture 3.1. In table 3.1 the main technical parameters are given for Unit 3.

Figure 3.1: *Overview over the power plant Nordjyllandsværket. The largest building is Unit 3.[Elsam 1998]*

3.2 Unit 3

Unit 3 in Nordjyllandsværket, shown on figure 3.1 is the coal fired power plant in the world with the highest thermal efficiency (about 47%, relative to the heating value of coal). This exceptional performance of unit 3 is largely due to favourable cooling conditions in Denmark where the cold seawater from the Limfjord can be used in the condenser. If unit 3 would be situated at an inland location in the UK where a cooling tower would be required, it would operate with a maximum efficiency of approximately 44-45%. [Watson 2005]
The thermal efficiency is determined from the electric net output divided by the heat transfer rate provided to the boiler by the fuel, see equation 3.1.

$$\eta = \frac{\dot{W}_{electric}}{\dot{Q}_{fuel}} \tag{3.1}$$

The heat transfer rate is found from the lower heating value and the mass flow rate of the fuel (eq; 3.2).

$$\dot{Q}_{fuel} = \dot{m}_{fuel} \cdot LHV_{fuel} \tag{3.2}$$

Technical parameters for Nordjyllandsværket Unit 3		
Total thermal efficiency with full heating output	90	%
Thermal efficiency (condensation operation)	47	%
Output, no district heating	411	MW_e
Output, with district heating	340	MW_e
Steam cycle type	Double reheat cycle with advanced regeneration	
Temperature	580	°C
Pressure	290	bar
Fuel	Bituminous coal	
Boiler throughput of coals	117	t/h
Air preheater	Regenerative	
Cooling	Sea water	

Table 3.1: *Main technical parameters for Nordjyllandsværket.[Elsam 1998]*

This unit is a double reheat, supercritical power plant designed to supply both electricity and district heating. It has been in commercial operation since 1998. The plant is fired with bituminous coal as the main fuel and heavy fuel oil as the backup fuel. A similar plant, built in Skærbæk in southern Jutland, is identical to NJV3 but is fired with natural gas as the main fuel and oil as the backup fuel. The unit in Skærbæk has been in commercial operation since 1997. [Elsam 1998]

3.3 Steam/water system

The steam turbine is an impulse design which allows expansion from 28,500 kPa (boiler outlet) to 3 kPa (condenser inlet). A representation of the water/steam-cycle is given in figure 3.2. In figure 3.2, numbers for bleeding of steam from the turbine to the feed water preheaters are equivalent to the numbers on the T-s diagram in figure 3.3. The same applies to the letters shown on both figures.

From the boiler, the steam is lead to the VHP turbine. From the VHP turbine the steam is lead back to the boiler to be reheated for the first time. After the first reheating the steam expands trough the HP turbine and is reheated for the second time. From the latter reheating the steam is lead to the IP0 turbine. After the IP0 turbine the steam expands trough the IP1 and IP2 turbines. From the IP turbines the steam is either lead to the LP turbines or the district heaters. If the steam is lead to the district heaters it condenses there and from the district heaters it is lead to the feed water system. If the steam is lead to the LP turbine it continues from there to the condenser. Depending on the need for district heating, the amount of steam that is lead through the district heaters can be controlled so that the flow is partially through the district heaters and partially to the LP turbines. After the condenser, the feed water is pumped back to the boiler trough a number of feed water heaters which get their heat from steam that bleeds from the turbine.

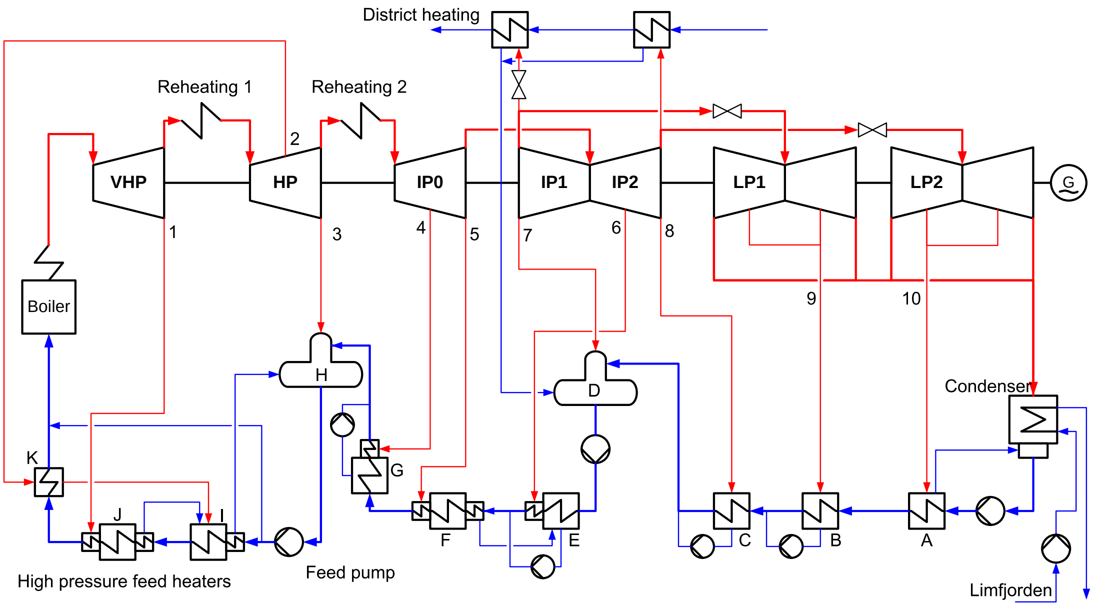

Figure 3.2: *A representation of the water/steam cycle in Unit 3 in Nordjyllandsværket.*

After the high pressure feedwater pump, there is a option of bypassing the last three feedwater heaters. Bypassing the feedwater heaters results in a less bleeding of steam from the turbine and therefore more power to the generator, but leads to slightly less overall efficiency of the power plant.
In table 3.2 the main technical parameters are given for the turbine.

Technical parameters for turbine

Stage	Pressure [kPa]	Temperature[°C]
VHP	28,500	580
HP	7,400	580
IP0	1,900	580
IP	720	429
LP1	150	233
LP2	70	154

Table 3.2: *Steam data for turbine inlets.[Elsam 1998]*

In figure 3.3, a T-s diagram over the process in NJV3 is shown. The letters represent the feed water preheaters and are also presented in the diagram in figure 3.2. The numbers stand for bleeding of steam from the turbine to the feed water preheaters and the same numbers are also shown in figure 3.2.

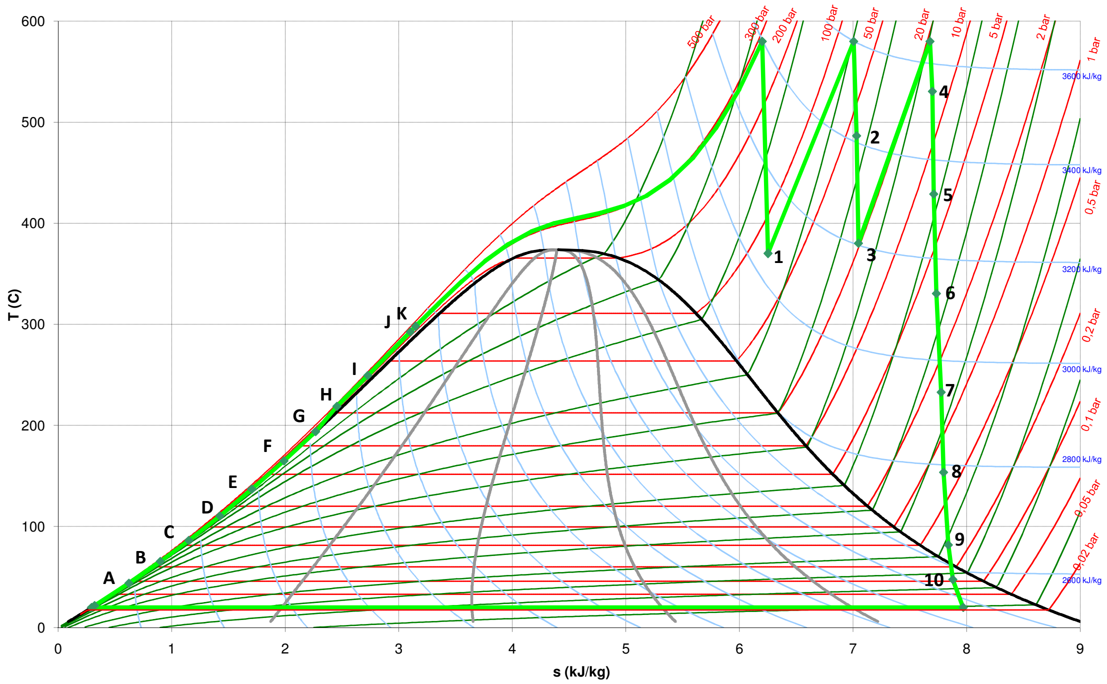

Figure 3.3: *A T-s diagram over the water/steam cycle in Unit 3 in Nordjyllandsværket.*

3.4 Combustion air and flue gas

Ewen though the main focus in this project is on the water/steam circulaton, the air and flue gas system can not be left unmentioned. To get a high efficiency from a power plant of this type, these systems must be designed to fit each other. On its way from the boiler, the flue gas heat exchanges with the feed water coming into the boiler. Because of the feedwater heaters the feedwater has a temperature close to 300 °C. That results in a high temperature of the fluegas flow leaving the boiler (about 370 °C). To utilise the energy in the flue gas, a regenerative air preheater is used to transfer heat from the flue gas to the combustion air. A flow diagram over the air and fluegas system in NJV3 is presented in figure 3.4.

The flue gas flows from the boiler to the air preheater through NO_x removal and from there it is lead through a cleaning system before it is released into the atmosphere through the stack. This cleaning system consists of electrostatic precipitator and flue gas desulphuration by wet scrubbing.
Between the electrostatic precipitator and the induced draught fans, a part of the flue gas is recirculated to the boiler via the air preheater. The flue gas recirculation is primarily used at low loads to maintain the volume flow through the boiler to ensure evenly spread heat exchanging in the boiler.

The air intake is at the top of the boiler building in the warm zone around the boiler. The air is lead through forced draught fans and splits up into primary air and combustion air, also called secondary air.

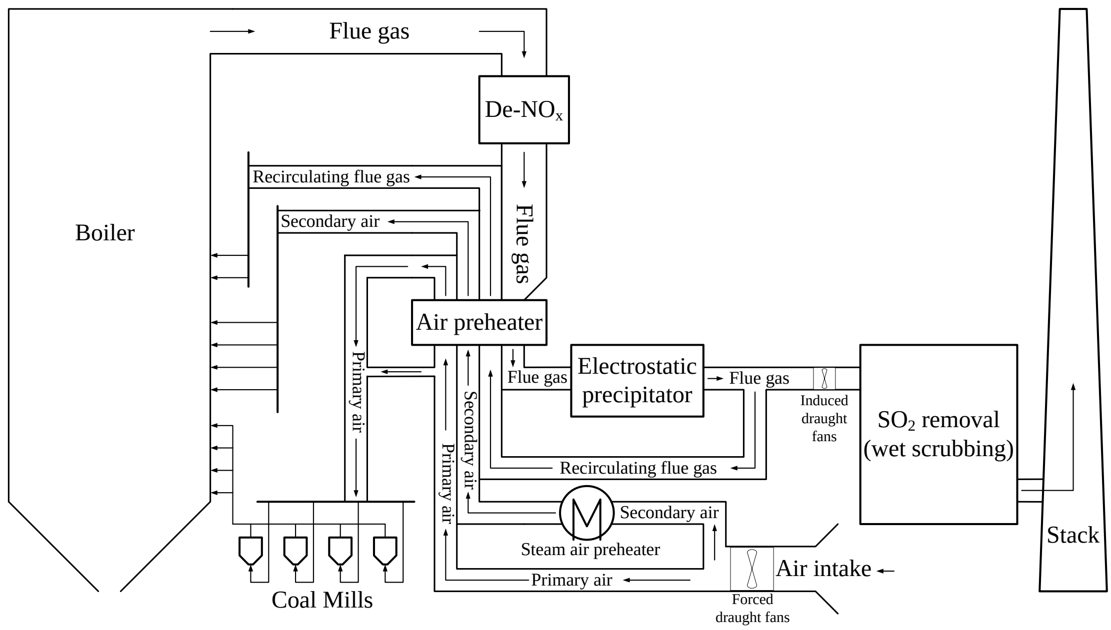

Figure 3.4: *Flow diagram of the channels where the flue gas and air are lead before and after the air preheater.*

The primary air is lead to the air preheater where it is heated up before it is lead to the coal mills where it is used to blow coal dust from the mills into the boiler. To prevent the coal dust from exploding, some of the primary air bypasses the air preheater to limit the temperature of the air.
The secondary air flows through a steam/air preheater on its way to the regenerative air preheater. The steam/air preheater is only used at low loads. At low loads, the temperature in the regenerative air preheater drops a little. A lower temperature in the regenerative air preheater could create conditions where sulphur in the fluegas condenses and forms a sulphuric acid. That would create erosion problems in the regenerative air preheater. Therefore the steam/air preheater is used at low loads to maintain the temperature in the regenerative air preheater over the sulphuric acid dew point. From the regenerative air preheater the secondary air is lead directly to the boiler to be used as combustion air. [Elsam 1998]

3.5 Summary

The Unit 3 in Nordjyllandsværket is chosen as base for the modeling of a steam/water cycle in super critical power plant. NJV3 is chosen mainly because of the high thermal efficiency and it is regarded for many as the state of the art super critical coal fired power plant.

4
Modelling the water/steam cycle

4.1 Introduction
4.2 Formulating the model
4.3 EES
4.3.1 Fluid properties
4.4 The foundation of the model
4.5 Summary

4.1 Introduction

In this chapter the development of the numerical model is explained. Engineering Equation Solver is introduced as well and the background of the model is explained.

4.2 Formulating the model

Programming this model is a highly iterative process. In the beginning it is defined what the model is supposed to do. This decision can change during the work on the model.
Deciding the variables and writing the equations is fairly straight forward but adjustment and improvement is constantly done through out the modeling progress.
In a model of this complexity, the model is very sensitive to the choice of start guesses of the modelling variables. These must be carefully chosen and often updated, even after minor changes in the model or the inputs to the model.
A flowchart over the modelling process can be seen in figure 4.1.

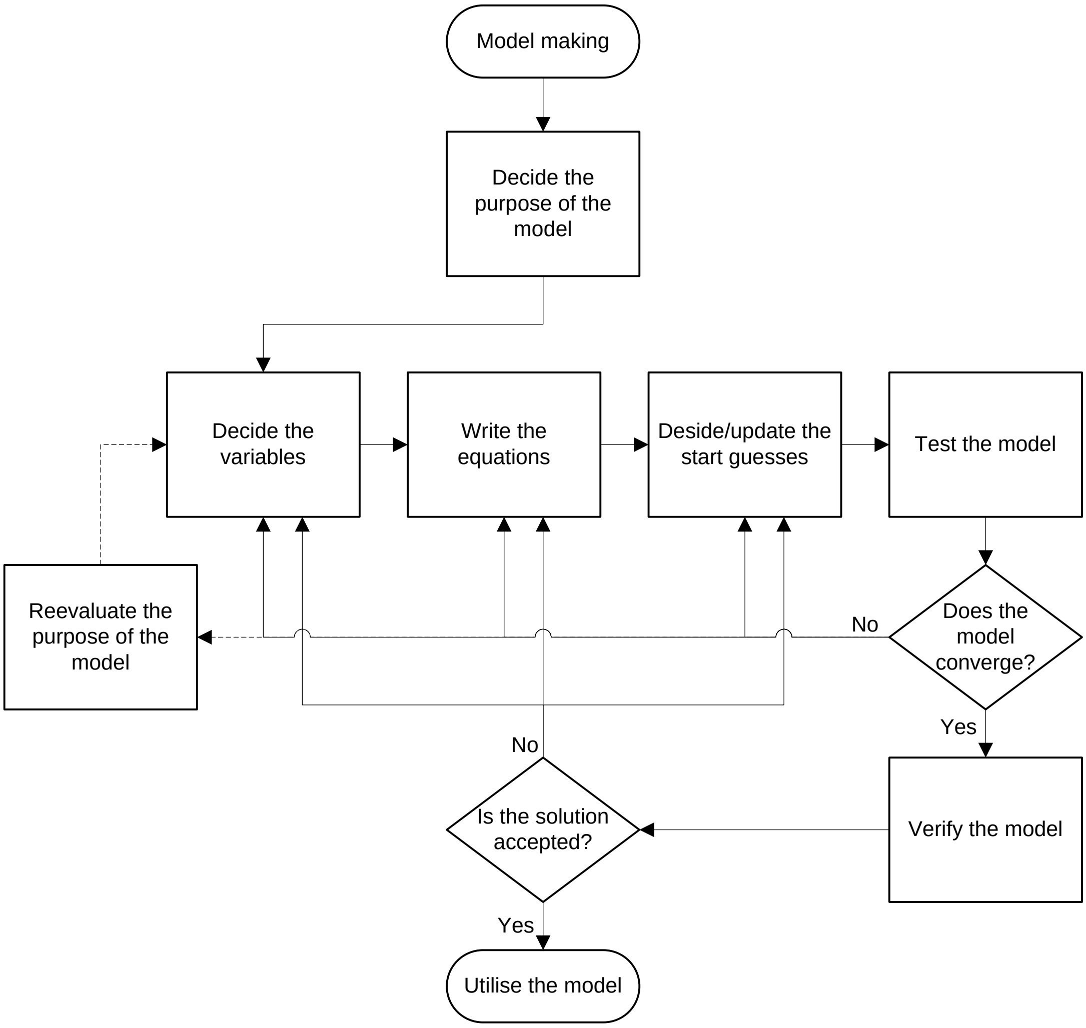

Figure 4.1: *Flowchart that shows the process of building the model in EES.*

4.3 EES

The model is built in EES (Engineering Equation Solver). The basic function provided by EES is the solution of a set of non-linear algebraic equations. EES can also solve differential equations, equations with complex variables, do optimization, provide linear and non-linear regression, generate publication-quality plots, simplify uncertainty analyses and provide animations. EES uses a variant of Newton's method to solve systems of non-linear equations. [Klein 2004]
When building a model such as this one in EES, great care must be taken when choosing start guesses. If the start guesses are not properly chosen the model will not converge.

4.3.1 Fluid properties

For the properties of water and steam, EES has several different reference tables to chose between. In this case Steam_IAPWS is chosen because it implements particularly high accuracy thermodynamic properties of water at super critical conditions with the 1995 Formulation for the Thermodynamic Properties of Ordinary Water Substance for General and Scientific Use, issued by The International Association for the Properties of Water and Steam. Steam_IAPWS uses slightly more computer power than the other tables, but its advantages are that it is more accurate at higher temperatures and pressures.

4.4 The foundation of the model

In order to make the model realistic, technical parameters for the turbine in NJV3 are used to calculate the isentropic efficiency of the turbine. These parameters are temperature and pressure of the steam at the inlet of each stage of the turbine and are acquired from a poster published by Elsam 1998 at the beginning of commercial operation of NJV3. [Elsam 1998] Table 3.2 lists those parameters for the steam.
It was also necessary to make a few other ”qualified guesses”. One example, is the pressure loss over the boiler. Those numbers are taken from T-s diagram made by Jeppe Grue. [Grue 2007] This T-s diagram can be seen in figure 3.3.
The model is then verified by comparing data from the model with data for NJV3. The verification of the model is done in section 6.2.

4.5 Summary

Building this model is an highly iterative procedure. During this process, the earlier work is constantly evaluated and reevaluated. The model consists of more than 250 equations and there are many opportunities to make mistakes. But finally it is assumed that the model must be fairly accurate when comparing the solutions from the model with design data from NJV3.

5
Presentation of the model

5.1 Introduction
5.2 Mass balance
5.3 Energy balance
5.4 Modules
5.4.1 Boiler
5.4.2 Turbine
5.4.3 Generator
5.4.4 Condenser
5.4.5 Feedwater pumps
5.4.6 Feedwater preheaters
5.4.7 Bypass
5.5 Summary

5.1 Introduction

In this chapter the structure of the model is explained.

5.2 Mass balance

In the model, the energy balance over the boiler is applied to determine the mass flow rate of water/steam through the boiler. Then from that energy balance, and from energy balances over heat exchangers in the feedwater flow, all other mass flow rates in the system are deduced. This is complicated task because this is not a simple cycle. There are two reheat stages and ten steam outlets on the turbine for the purpose of preheating the feedwater.
Apart from the energy balances, there are more than 20 equations of mass balances in the model. Each mass flow rate has one variable name in the model and the same mass flow rate variable name can thereby appear in several mass balances. By that, the mass balances are connected.
In figure 5.1 the boundaries for the mass balances are shown. In appendix B the equations for the mass balance can be viewed, as they are grouped together at the beginning of the code.
Additionally, there are restrictions on the mass flow rate variables in the model, primarily to ensure that the flow is in the right direction.

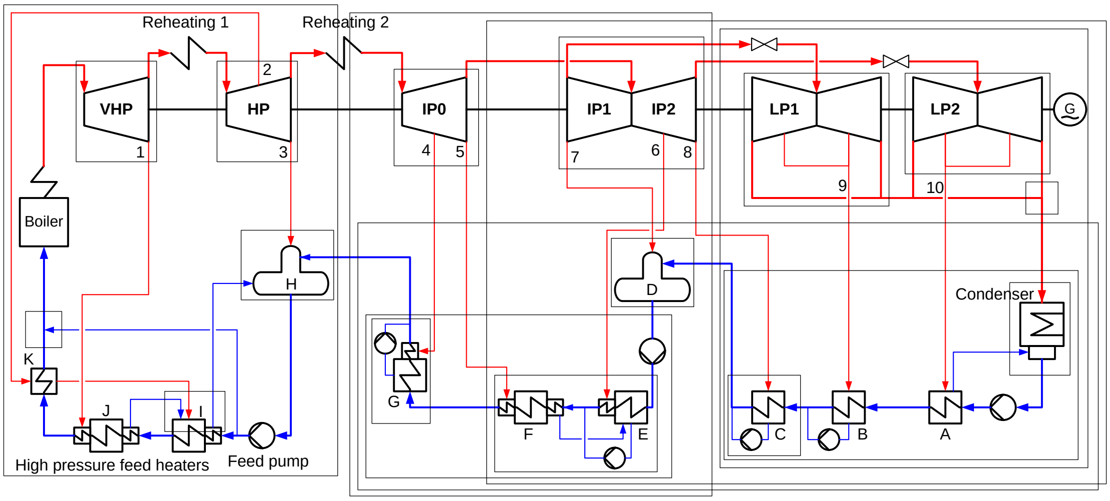

Figure 5.1: *Diagram that shows overview over mass balances in the model.*

5.3 Energy balance

The energy balances in the model set the mass flow rate trough various components in the model. It can also work the other way around, the mass flow rate sets the value of enthalpy of the flow in question by the energy balances. The boundaries for the energy balances are shown in figure 5.2. In appendix B the equations for the energy balances can be found in the code in the modules they belong to.

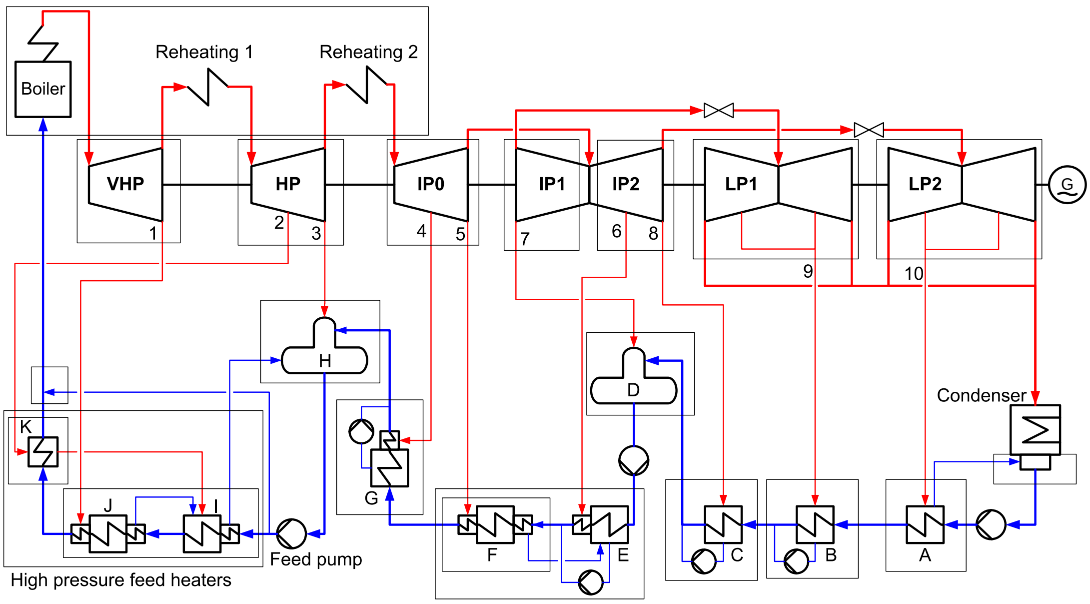

Figure 5.2: *Diagram that shows overview over energy balances in the model.*

5.4 Modules

The model is built up of modules that are connected together. Each module represents a component in the water/steam cycle. In the diagram in figure 5.3 it is shown how the modules are connected. Note that the arrows in figure 5.3 do not indicate the direction of the flow but show where from, the modules get their inputs from. It should also be noted that many arrows for mass flow rate and enthalpy are not drawn in order to simplify the diagram. For mass and energy balances figures 5.1 and 5.2 can be viewed.

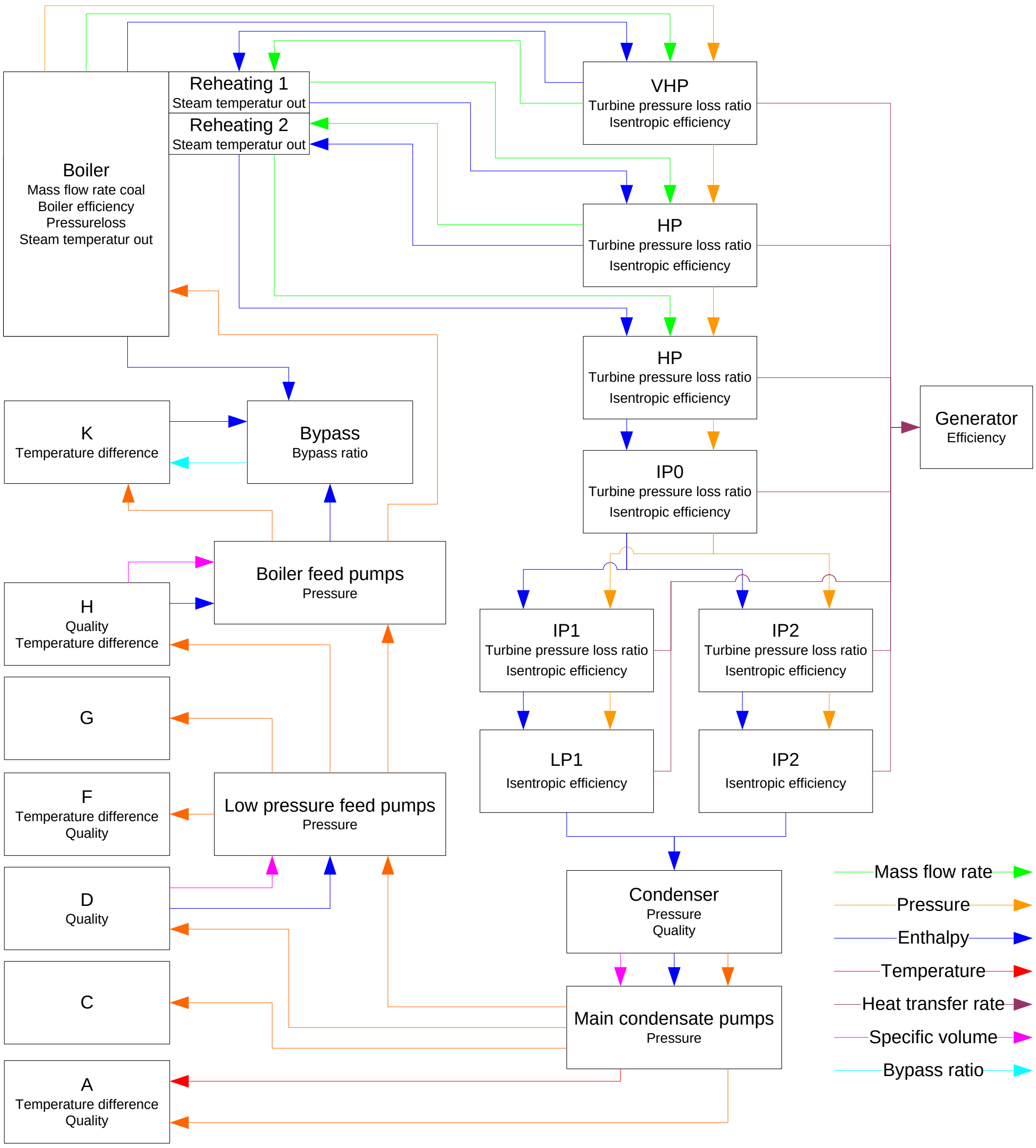

Figure 5.3: *Diagram that shows how the modules in the model are connected. Under the name of the module are listed those variables that are input to the model.*

5.4.1 Boiler

In this model, the temperature of the steam from the boiler, the throughput of coal, the heating value of the coal and boiler efficiency is given as input. The pressure of the steam from the boiler is set by taking the pressure from the high pressure feedwater pump and assuming about 14% pressure loss through the boiler. From the values of temperature and pressure, the enthalpy of the steam is acquired.
The heating value of the coal is set to be 28700 kJ/kg and that is within the limits for bituminous coal given by [Perry and Green 1997].

With these parameters it is possible to establish an energy balance for the boiler and calculate the mass flow of water/steam through the boiler. In equation 5.1 the energy balance as it is in the model is given and from that energy balance the mass flow rate can be isolated and found.

$$\dot{Q}_{coal} \cdot \eta_b = \dot{m}_{steamb} \cdot (h_{ob} - h_{ib}) + \dot{m}_{reh1} \cdot (h_{iHP} - h_{oVHP}) + \dot{m}_{reh2} \cdot (h_{iIP0} - h_{oHP}) \quad (5.1)$$

The boiler module does not include variables for energy flow in form of air and flue gas. Instead these energy transfer rates are encountered for in the form of boiler efficiency.

5.4.2 Turbine

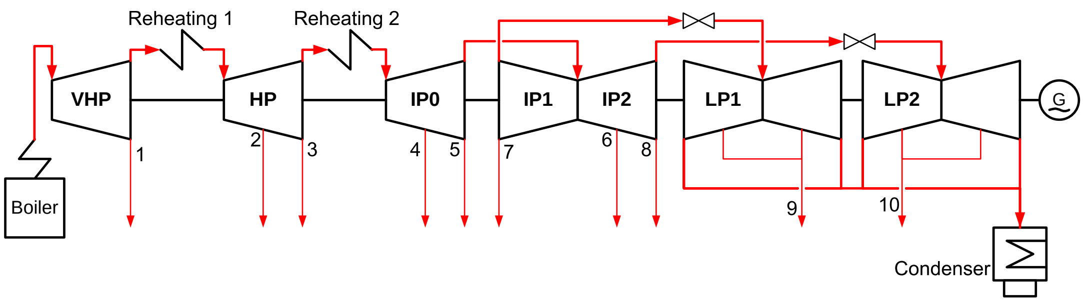

Figure 5.4: *Diagram over the turbine and its steam outlets.*

Each stage of the turbine is handled as an independent module in the model. Inputs into the turbine modules are mass flow rate into the turbine stage, enthalpy of the steam flowing into the turbine stage, isentropic efficiency, pressure loss ratio and pressure of the steam flowing into the turbine stage. Pressure loss ratio is the ratio of the pressure loss over the whole turbine.
For each stage of the turbine, the heat transfer rate is calculated by multiplying the mass flow rate with the change in enthalpy of the steam that flows through the turbine stage, equation 5.2.

$$\dot{Q}_{turb} = \dot{m}_{steam} \cdot \Delta h_{steam} \quad (5.2)$$

Then the heat transfer rates for each stage, are added to get the heat transfer rate for the whole turbine.

For the isentropic efficiency, a qualified guess is taken. There were four types of criteria that had to be fulfilled in the decision of the isentropic efficiency.

- The net energy output from the system should match NJV3 at full load.
- The mass flow rate out of the boiler should match NJV3 at full load.
- The thermal efficiency should match NJV3 at full load.
- The T-s diagram from the model is to look like the T-s diagram for NJV3 in figure 3.3.

The values for the first three items are listed in table 6.1 and are about 1% or less from the target. Regarding the last item, the two T-s diagrams in figures 3.3 and 6.2 can be compared to judge how similar they are.
Table 5.1 shows the isentropic efficiencies for each stage of the turbine system.

Isentropic efficiency		
VHP	93	%
HP	93	%
IP0	91	%
IP1	90	%
IP2	90	%
LP1	90	%
LP2	90	%

Table 5.1: *Isentropic efficiency for each stage of the turbine in the model.*

5.4.3 Generator

The output from the generator is calculated from total heat transfer rates from the turbine and the efficiency for the generator (equation 5.3).

$$\dot{W}_{el} = \eta_{gen} \cdot \dot{Q}_{turb} \tag{5.3}$$

5.4.4 Condenser

In the condenser module, the pressure and the quality of the flow is determined from a T-s diagram provided by Jeppe Grue [Grue 2007].
The mass flow rate and the temperature of the seawater that flows through the condenser is not accounted for in the condenser module. It is assumed that there is a balance between the heat produced by condensing the steam and the heat transfer from the condenser by the seawater.

5.4.5 Feedwater pumps

There are three modules for feedwater pumps in the model. Their pressure is set as it is in NJV3 but it can be altered to a value of choice.

5.4.6 Feedwater preheaters

The feedwater preheater modules calculate the increase of heat rate over the feedwater preheater. There are several variations and figure 5.5 and equation 5.4 give an example of the most simple one.

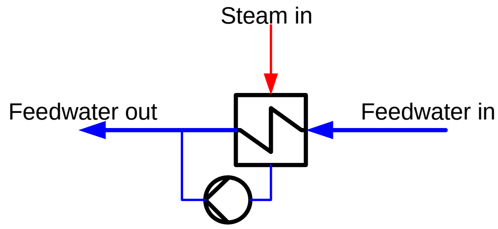

Figure 5.5: *Feedwater preheater.*

$$\dot{m}_{fw\ in} \cdot h_{fw\ in} + \dot{m}_{steam\ in} \cdot h_{steam\ in} = \dot{m}_{fw\ out} \cdot h_{fw\ out} \tag{5.4}$$

There are also cases where the flows through the heat exchanger do not mix. In those cases the assumption is taken that the temperature difference between the hot flow out of the heat exchanger and the cold flow into the heat exchanger is 5°C.

5.4.7 Bypass

There is a possibility of simulating bypass over the last three feedwater heat exchangers in the model (figure 5.6). The value for bypass can be set as 0 and 1 and anywhere there between for partial bypass. 0 is no bypass and 1 is full bypass.

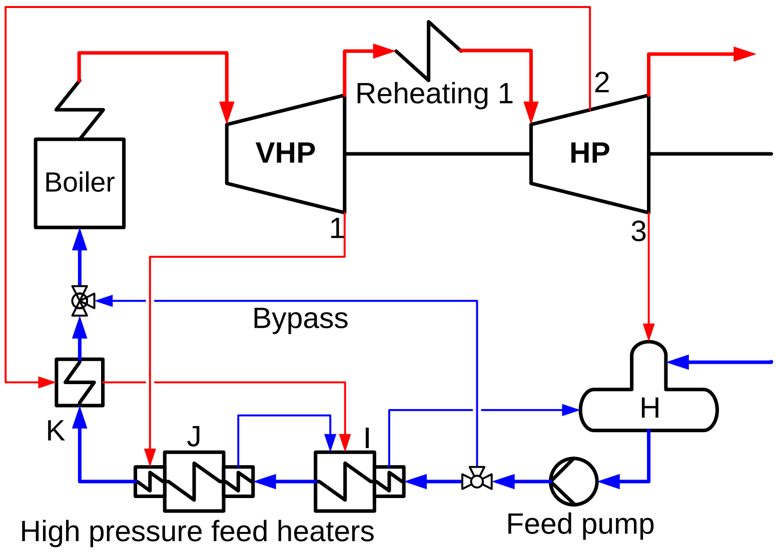

Figure 5.6: *Bypass over the last three feedwater heat exchangers.*

5.5 Summary

The model is made up of modules. Each module bases its calculations on inputs from other modules and constants set by the user of the model. Between the modules are mass and energy balances. These balances are connected when they involve the same flow by using the same variable name for that flow.

6
Verification of the model

6.1 Introduction
6.2 Comparing design data for NJV3 with solutions from the model
6.3 Comparing T-s diagrams from the model and NJV3.
6.4 Stability of the model
6.5 Summary

6.1 Introduction

In this chapter the model is verified by holding its solutions against known parameters from NJV3. Also the stability in the solutions is analysed to see if it is as expected.

6.2 Comparing design data for NJV3 with solutions from the model

To verify the validation of the model, it can be compared with design data for NJV3. When building the model, a effort was made to program the model to give results for certain chosen variables as close as possible to design data for NJV3 as they are presented in [Elsam 1998]. For this purpose, three variables were chosen; electric efficiency, steam mass flow from boiler and electric output from the generator. These variables are listed in table 6.1.
In diagram 6.1 the results from the model are displayed when the steam temperature from the boiler is set to 580 °C, and the pressure from the feed pump is 33.000 kPa.

Verification of the model

	NJV3	Model	Difference
Efficiency, electric generation only	47%	48.13%	2.40%
Max steam output from boiler	270 kg/s	272 kg/s	0.74%
Output from generator	411 MW	407.3 MW	0.90%

Table 6.1: *Comparison of design data for NJV3 at 100% load and calculations from the model. The difference is relative difference.*

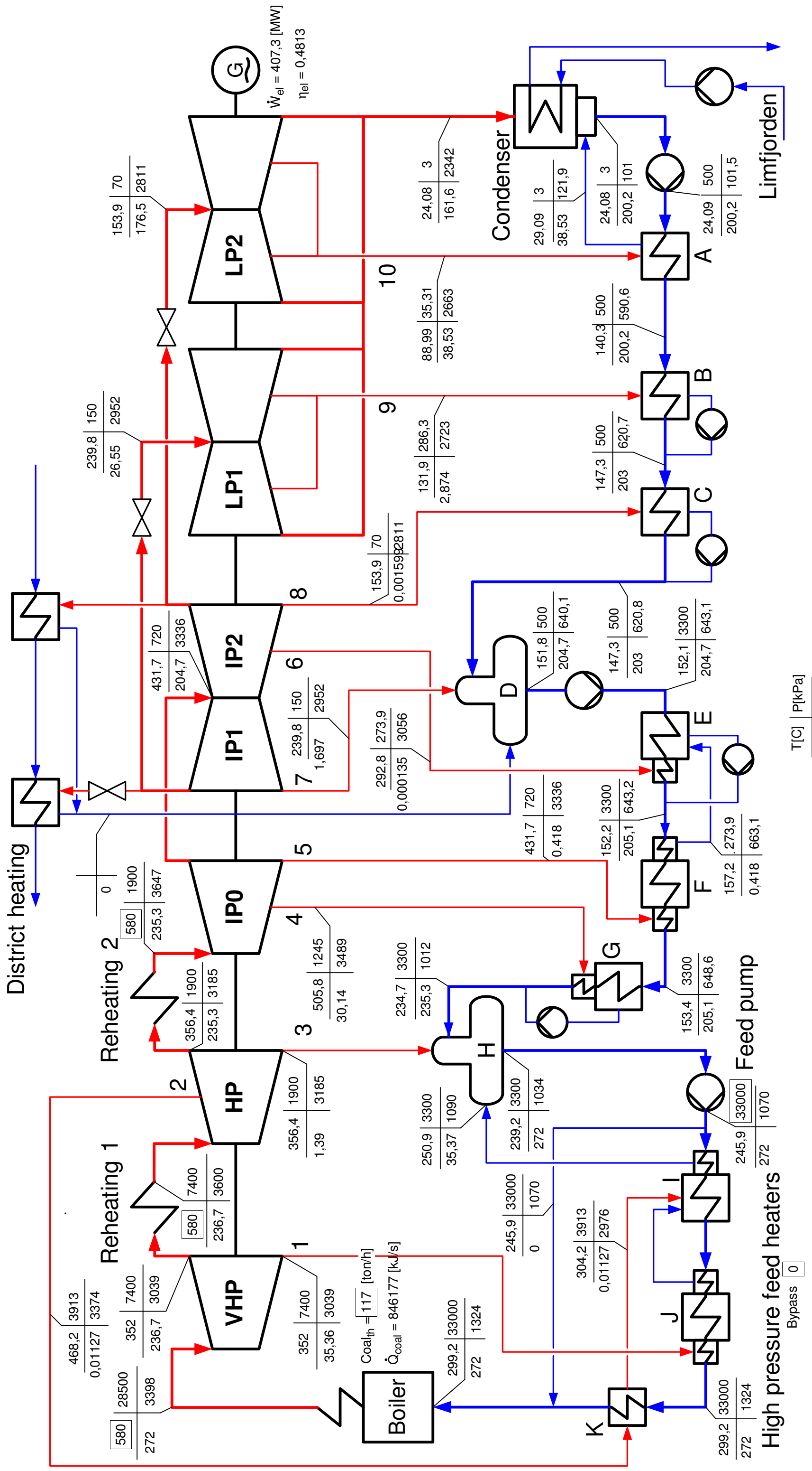

Figure 6.1: *Key numbers in the water/steam cycle from the model when simulating NJV3 at full load.*

6.3 Comparing T-s diagrams from the model and NJV3.

The model generates a T-s diagram which is also helpful in determining if the model is returning feasible results. The T-s diagram matching the diagram 6.1 is shown in figure 6.2. The T-s diagram in figure 6.2 can also be compared with the T-s diagram from the power plant Nordjyllandsværket shown in figure 3.3.

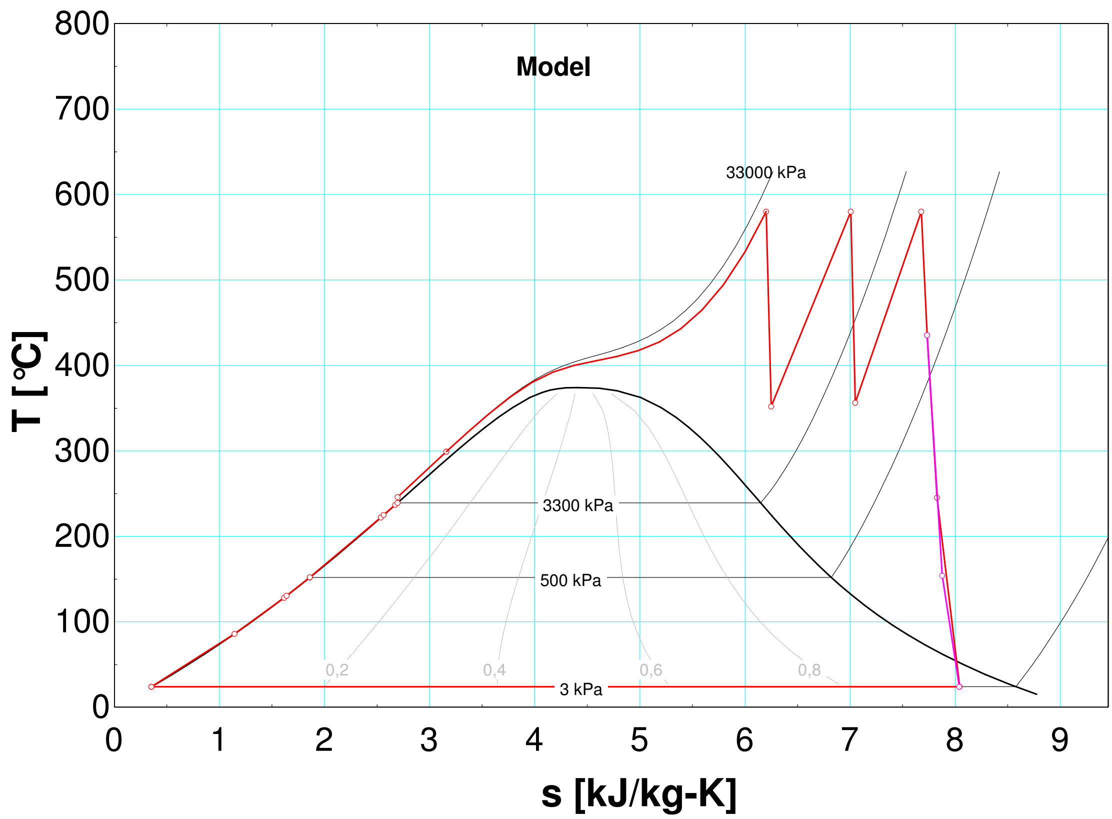

Figure 6.2: *T-s diagram over the water/steam cycle when simulating NJV3 at full load.*

Notice that there are two lines on the T-s diagram representing the steam flows through the IP and LP turbine stages.

6.4 Stability of the model

Because EES is an numerical solver, the solutions from the model can vary depending on the start guesses. To evaluate this phenomena in the model, a stability test was performed.

In table 6.2 solutions for six variables are logged. These variables are; electric output, thermal efficiency, mass flow rate out of the boiler, mass flow rate into the condenser, mass flow rate into the feedwater preheater D and mass flow rate into the feedwater preheater H. The model is set to simulate NJV3 at full load and every time the model converged, the start guesses were updated.

When updating the start guesses, the guess value of each variable is replaced with the value determined in the last calculation. This should improve the computational accuracy of the model since it ensures that a consistent set of guess values is used in the next calculation.

Stability test

$\dot{W}_{el}$ [MW]	η [%]	$\dot{m}_{o\ b}$ [kg/s]	$\dot{m}_{i\ cond}$ [kg/s]	$\dot{m}_{i\ D}$ [kg/s]	$\dot{m}_{i\ H}$ [kg/s]
407.4	48.15	272	158.6	198.3	235.5
408.2	48.24	272	159.9	199.5	235.5
408.7	48.3	272	161.1	170.1	235.5
408.4	48.26	272	160.5	177.8	235.5
404.9	47.85	272	162.8	189.8	235.5
404.2	47.77	272	163.2	190.3	235.5
408.9	48.32	272	162.7	189.7	235.5
409.4	48.38	272	162.7	189.7	235.6
409.6	48.41	272	162.7	189.6	235.5
403.9	47.73	272	163.8	190.9	235.5
405.3	47.9	272	164.7	169.8	235.5
407.7	48.18	272	164.4	169.8	235.5
409.7	48.42	272	164.1	168.8	235.5
409.3	48.37	272	163.1	169	235.5
410.8	48.55	272	160.1	194.3	235.5
410	48.45	272	158.8	193	235.5
411.1	48.58	272	158.3	192.5	235.5
410.6	48.53	272	158.8	190.1	235.5
1.76	1.76	0.00	3.96	16.58	0.04
The range as a ratio of the mean value [%]					

Table 6.2: *Stability test of the model when updating start guesses after every solution. When a solution is acquired, values for the chosen variables are logged, the start guesses are updated and the process is repeated.*

At the bottom of table 6.2, the range of each column is given as percentage of the mean value, see equation 6.1. By that it is possible to compare the stability for different variable solutions regardless of different units and size of the scale.

$$Fluctuation = \frac{x_{max} - x_{min}}{mean\ value} \cdot 100 \tag{6.1}$$

When viewing the solutions in table 6.2 it seems like the modules around the boiler (see fig. 5.3) give the most steady solutions. This is not surprising because it is in the boiler module where the governing inputs to the model are.
There is always the same mass flow rate through the boiler, but this mass flow is not distributed through the system in the same way in every calculation. The mass flow

rates through the bleeds from the turbine to the feedwater preheaters varies and that results in a varying mass flow rates of feedwater through the first feedwater preheaters. It is interesting to see so big fluctuation in the solution for the mass flow rate into feedwater preheater D. This happens in spite of low values for the maximum relative residual and the maximum change in a variable value from one iteration to the next. These are respectively set to 10^{-6} and 10^{-9}.

This variation in the mass flow rate through the first feedwater preheaters does not seem to affect the overall efficiency of the system much and there are also no major influences on the T-s diagram. On the other hand one must be cautious when making assumptions related to the mass flow rate in the system furthest from the boiler module, because of fluctuations in these values.

6.5 Summary

In this chapter the verification of the model has been discussed. The solutions from the model are compared with design data and T-s diagram from NJV3 at full load. There is consistency between the model and the data for NJV3 at full load and that gives confidence to work further with the model.
The stability of the model is tested by evaluating the range of the solutions. The thermal efficiency which is regarded as one of the most interesting variables fluctuates within 1% while other variables can vary much more. Therefore it is important to evaluate the fluctuations in those variables one wishes to investigate.

Another way of verifying the model, is to alter some variables and see if the results have the same tendency as expected. This is done in following chapter.

7
Ultra-super critical steam

7.1 Introduction
7.2 Altering the steam data
7.2.1 670°C and 42 MPa
7.3 Summary

7.1 Introduction

In this chapter a investigation of the potential of ultra-super critical steam data is performed. Results are presented for increased temperature from the boiler and pressure from the high pressure feedwater pumps.

7.2 Altering the steam data

In order to investigate the influences of increasing the temperature and pressure of the steam from the boiler, these values are increased gradually up to 760 °C and 42 MPa. In table 7.1 values are given for chosen inputs and outputs from the model. The inputs are temperature from the boiler and pressure from the high pressure feedwater pump. The outputs are output from generator, thermal efficiency, mass flow rate through the boiler and the mass flow rate through the condenser.
The results were obtained by altering the values for temperature and pressure in the order; $T_{out\ reh2}$, $T_{out\ reh1}$, $T_{in\ VHP}$ and $P_{hp\ fw}$. Full table of results is given in appendix C.

When gathering the data in table 7.1 it became clear that a better convergence of the model would be a great improvement of the model, as it took two and a half day to get the solutions in the table.

In figure 7.1 it is shown that the results for the efficiency rises about 4% by elevating the temperature and pressure to respectively 760 °C and 42 MPa. These results comply with what was expected and are also comparable to what other studies expect in efficiency when raising the temperature and pressure.

Increasing temperature and pressure

$T_{o\ b}$ [°C]	$P_{hp\ fw}$ [kPa]	$\dot{W}_{el}$ [MW]	η [%]	$\dot{m}_{i\ b}$ [kg/s]	$\dot{m}_{i\ cond}$ [kg/s]
580	33000	407.4	48.15	272	158.6
600	34000	412.8	48.79	267.4	159.6
620	35000	412.6	48.76	261.6	158.8
640	36000	418	49.4	255.8	154.5
660	37000	423.9	50.1	249.3	154
680	38000	424.6	50.18	246.7	152.4
700	39000	426.2	50.37	241.8	148.8
720	40000	429.2	50.72	237.1	146.3
740	41000	435	51.41	232.2	147
760	42000	441	52.12	227.6	143

Table 7.1: *Change in results when increasing pressure from the high pressure feedwater pumps and temperature from the boiler. Values for the efficiency are shown in figure 7.1. The values in this table are acquired from a more detailed table in appendix C.*

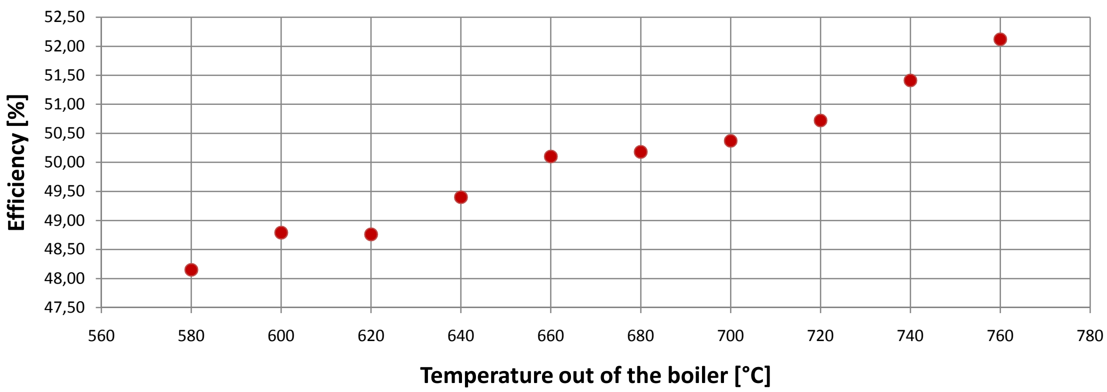

Figure 7.1: *Efficiency as a function of temperature from the boiler and pressure from the feedwater pumps.*

7.2.1 670°C and 42 MPa

In figure 7.2 the overall results are given for 760 °C from the boiler and 42 MPa from the high pressure feedwater pump. In figure 7.3 a T-s diagram for the same condition is shown.

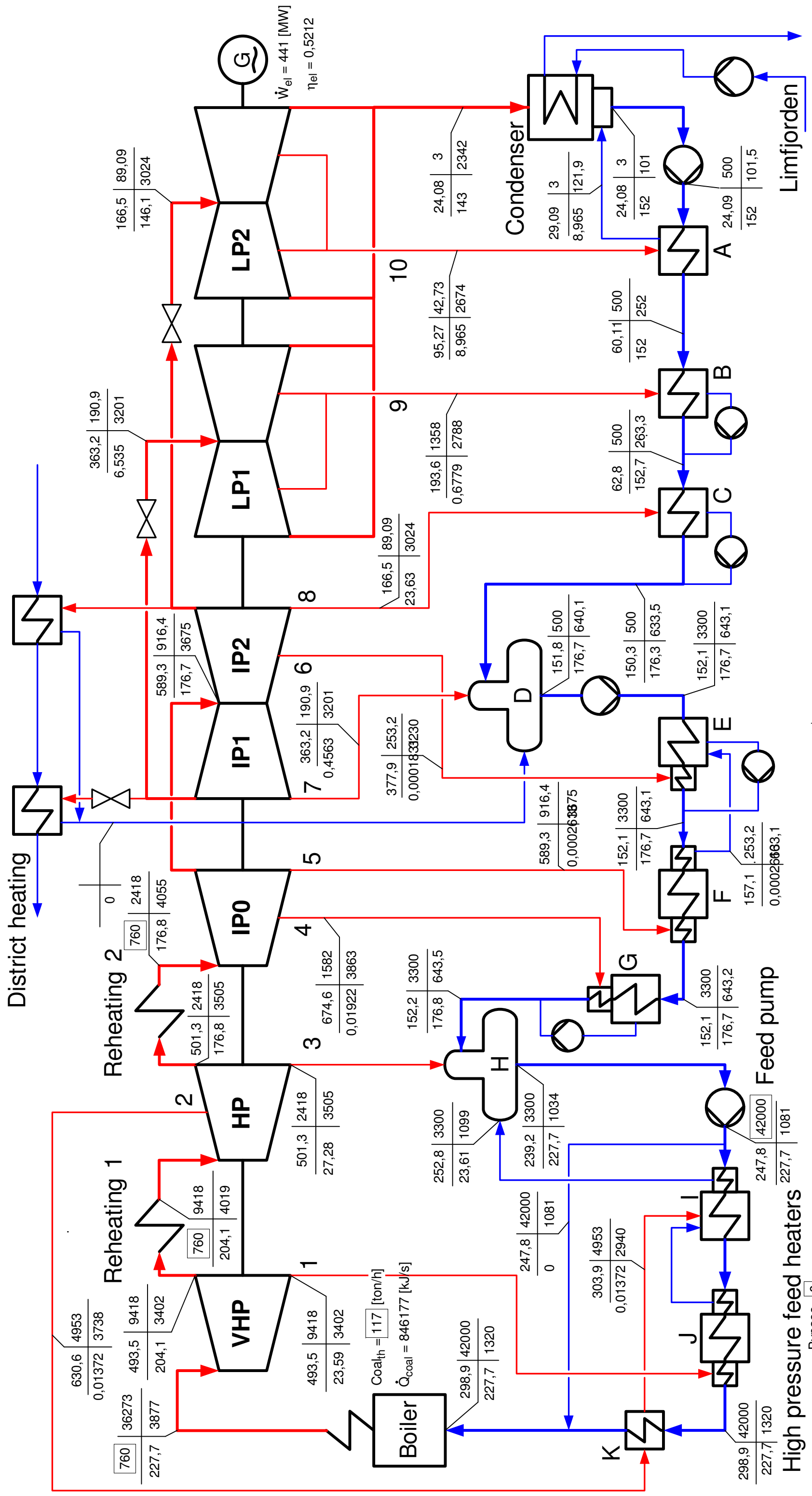

Figure 7.2: *Key numbers in the water/steam cycle from the model when simulating 769 °C from the boiler and 42 MPa from the high pressure feedwater pump.*

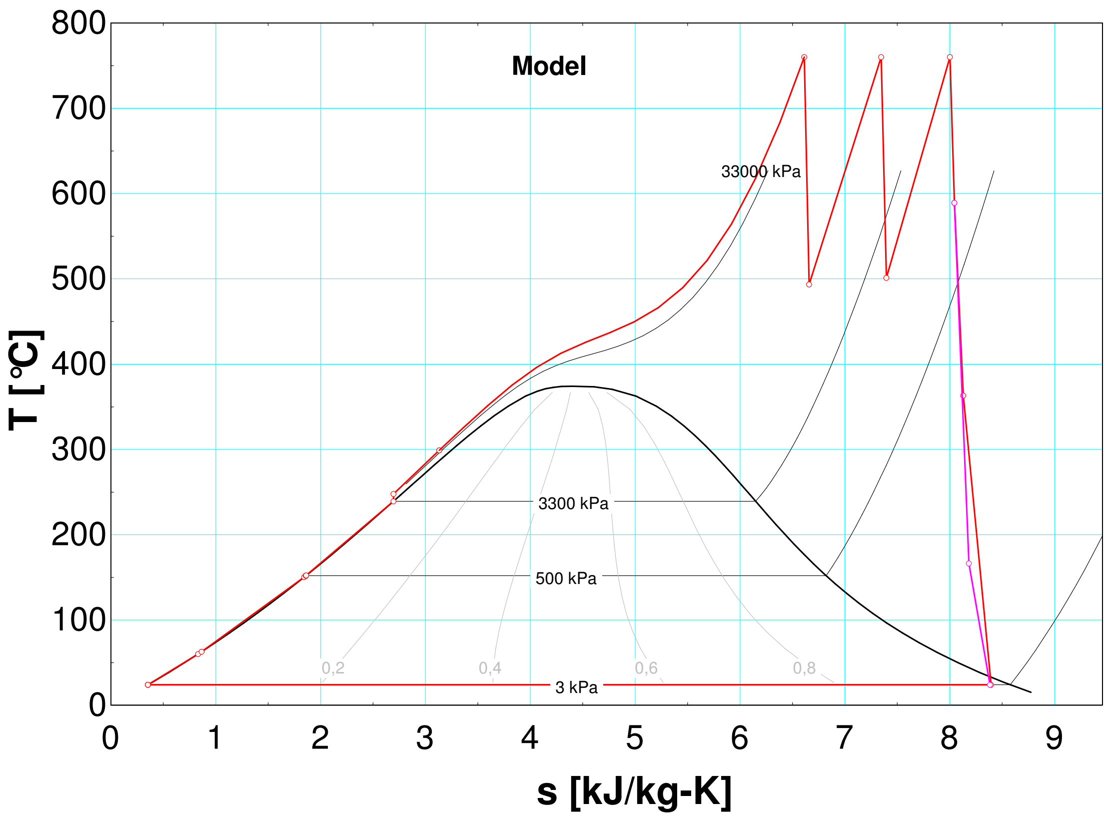

Figure 7.3: *T-s diagram over the water/steam cycle when simulating 769 °C from the boiler and 42 MPa from the high pressure feedwater pump.*

7.3 Summary

In this chapter a investigation of the potential of ultra-super critical steam data is performed. This investigation is done by making use of the model developed in this project. The tendency in the solutions is as expected and shows increased thermal efficiency when the temperature and pressure of the steam is increased. By raising the temperature from 580 °C to 760 °C and the pressure out of the high pressure feedwater pump from 33 MPa to 42 MPa, the thermal efficiency improves by about 4%.
This improved efficiency is in accordance with resent literature on the subject and supplements further to the verification of the model. It is though very time consuming to work with the model and if the model would converge more easily, more studies could be made on this system in the same amount of time.

8 Conclusion

8.1 Primary conclusion
8.2 Future work

8.1 Primary conclusion

The main aspect of this project is to describe and model the water/steam cycle in super critical pulverised coal fired power plants. Then use that model to investigate the potential of a **ultra**-super critical pulverised coal fired power plant.

There is a lot of work done in the field of increasing the efficiency of a super critical pulverised coal fired power plant. From these studies it seems likely that the boilers in the nearest future will be build of a nickel-based super-alloy. The maximum temperature/pressure will be around 700°C/35MPa and about 760°C reheat temperature. Higher reheat temperature is possible because of lower pressure. This improvement of the steam data should be noticeable in better efficiency.

The Unit 3 in Nordjyllandsværket is chosen as base for the modeling of a steam/water cycle in super critical power plant. Unit 3 in Nordjyllandsværket is chosen mainly because of the high thermal efficiency and it is regarded for many as the state of the art super critical coal fired power plant.

Building this model is an highly iterative procedure. During this process, the earlier work is constantly evaluated and reevaluated. The model consists of more than 250 equations and there are many opportunities to make mistakes. It is though concluded that the model is fairly accurate when comparing the solutions from the model with design data from unit 3 in Nordjyllandsværket.
The model is made up of modules. Each module bases its calculations on inputs from other modules and constants set by the user of the model.

The model is verified by comparing the solutions from the model with design data and T-s diagram from unit 3 in Nordjyllandsværket at full load. There is consistency between the model and the data for NJV3 at full load and that gives confidence to work further with the model. The model is also verified by investigating the tendency for the solutions when altering the temperature and pressure inputs. The tendency is as expected.

When altering the inputs to the model, it can get very difficult to get the model to converge. This has a great influence on the time required to get solutions from the model and has limited the number of analyses done with the model.

A investigation of the potential of ultra-super critical steam data is performed. The tendency in the solutions is as expected and shows increased thermal efficiency when the temperature and pressure of the steam is increased. By raising the temperature from 580 °C to 760 °C and the pressure out of the high pressure feedwater pump from 33 MPa to 42 MPa, the thermal efficiency improves by about 4%.
This improved efficiency is in accordance with resent literature on the subject and supplements further to the verification of the model.

In the introduction of this report the two areas of interest within this project are stated. These are modeling the water/steam cycle in a super critical coal fired power plant and utilise the model to investigate the thermal efficiency of a such power plant at a ultra-super critical conditions. It is concluded that these tasks where achieved.

8.2 Future work

There are many more interesting studies that could be done with the model, beside those described in this report. These studies could be done with the model either directly or with minor changes in the code. Few of these studies are mentioned in the following.

On the turbine there are ten steam outlets for the purpose of preheating the feedwater. It would be interesting to investigate if some of these steam outlets could be removed with out substantial loss in efficiency. Either by blocking the steam outlet or lead the steam out of the system for other purposes. Bypassing the feedwater preheaters is one version of this investigation.
The performance of the system at part load could be investigated.
The number of reheat stages could be investigated to determine how much each reheat stage influences the thermal efficiency.
The cooling conditions could be investigated with focus on the difference between summer and winter conditions or difference in climates and placement in the world.

From the above, it is clear that there is no shortage of useful exploitation of such a model. The imagination is the only limit.

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A
Model user guide

A.1 About the model
A.2 User guide

In this appendix a short user guide to the model is presented.

A.1 About the model

This is a model over steam/water cycle in a super critical power plant, made by Óttar Kjartansson at his 10th semester under the *Fluids and Combustion Engineering* (FACE), graduate programme, in the *Institute of Energy Technology - AAU*.
The model is based on the structure of unit 3 in the power plant Nordjyllandsværket described in chapter 3
For educational purposes, the model can be used without charge. Just make sure that the authors name is mentioned. For commercial use of the model, a permit from Óttar Kjartansson is required.

A.2 User guide

If you have little experience with EES it would be a good idea to have a look at the manual, e.g. [Klein 2004] to get started. It can be downloaded at http://fchart.com.

The model can be applied to investigate the consequences of different temperatures and pressures from the boiler on such a system described in this report. It is also possible to simulate bypass over the last three feedwater heat exchangers.
To do this it is most suitable to have the diagram window open and alter the values in the squared boxes that can be found in the diagram, and then press the ”calculate” button.
In table A.1 the variables are listed. It is recommended to stay within the restrictions given in table A.1. It is possible to enter values that are lower or higher than those given in table A.1 but it will probably be difficult to get the model to converge.

In the diagram window the solutions for temperature, pressure, mass flow rate and enthalpy are given in crosses for most of the flows (figure A.1). In the solution window all solutions are displayed.

Temperature [ºC]	Pressure [kPa]
Mass flow rate [kg/s]	Enthalpy [kJ/kg]

Figure A.1: *Explanation of the crosses in the diagram window.*

It is also possible to make changes or improvements on the code in the equations window. These changes could be e.g. altering the heating value of the fuel or isentropic efficiency of the turbine. New modules or equations can also be added.

It is also possible to get a T-s diagram over the process by pressing the ”T-s diagram” button.
The equation set can be visualised or modified by pressing ctrl+e and a formatted and easier readable version is seen by pressing ctrl+f. To return to the diagram window, press ctrl+d.

Variables in the model

Variable	Unit	Lower	Upper
Temperature	°C	400	1000
Pressure	kPa	10,000	50,000
Bypass	-	0	1
Coal throughput	ton/h	30	200

Table A.1: *Variables in the model that can be altered and their recommended lower and upper values.*

When altering the variables, an error message like the one in figure A.2 may appear. In that case, try again until the model converges.

Figure A.2: *Error message that is likely to come after changes in variables.*

If a solution is then obtained, use Update Guesses in the Calculate menu to set the guess values of all variables to their current values. Repeat until the model is stable. To speed up this process it might be a good idea to do the changes in smaller steps. Still, this can be very time consuming and patience is essential in this work.

B
Model code

B.1 Introduction
B.2 Code

B.1 Introduction

In this appendix the model code is presented as it is in the Equations Window in EES.

B.2 Code

```
"This is a model over steam/water cycle in a super critical power plant,
made by Óttar Kjartansson at his 10th semester under the Fluids and
Combustion Engineering (FACE), graduate programme, at the Institute
of Energy Technology - AAU.
For educational purposes, the model can be used without charge. Just
make sure that the authors name is mentioned. For a commercial use
of the model, a permit from Óttar Kjartansson is required."

"Mass balance"
m_dot_steamboiler=m_dot_reh1+m_dot_bleed1
m_dot_steamboiler=m_dot_out_H
m_dot_reh2=m_dot_in_H
m_dot_bleed1+m_dot_bleed2=m_dot_bleed_1_2
m_dot_reh1=m_dot_bleed2+m_dot_bleed3+m_dot_reh2
m_dot_in_H+m_dot_bleed3+m_dot_bleed_1_2=m_dot_out_H
m_dot_in_H=m_dot_bleed4+m_dot_bleed5+m_dot_bleed6+m_dot_out_D
m_dot_reh2=m_dot_bleed4+m_dot_bleed5+m_dot_out_IP0
m_dot_in_G+m_dot_bleed4=m_dot_in_H
m_dot_out_D+m_dot_bleed5+m_dot_bleed6=m_dot_in_G
m_dot_in_G=m_dot_bleed5+m_dot_out_IP0
m_dot_out_D=m_dot_in_D+m_dot_bleed7
m_dot_in_C+m_dot_bleed8=m_dot_in_D
m_dot_in_D=m_dot_bleed8+m_dot_bleed9+m_dot_bleed10+m_dot_out_LP
m_dot_out_IP0=m_dot_bleed6+m_dot_bleed7+m_dot_bleed8+m_dot_out_IP1+m_dot_out_IP2
m_dot_out_IP1=m_dot_bleed9+m_dot_out_LP1
m_dot_out_IP2=m_dot_bleed10+m_dot_out_LP2
```

```
m_dot_out_LP=m_dot_out_LP1+m_dot_out_LP2
m_dot_in_H=m_dot_bleed4+m_dot_bleed5+m_dot_bleed6+m_dot_bleed7+m_dot_bleed8+
   m_dot_bleed9+m_dot_bleed10+m_dot_out_LP
m_dot_in_D=m_dot_bleed8+m_dot_out_IP1+m_dot_out_IP2
m_dot_in_D+m_dot_reh2=m_dot_bleed8+m_dot_out_IP1+m_dot_out_IP2+m_dot_in_H
m_dot_out_cond=m_dot_out_LP+m_dot_bleed10

"Boiler"
{Coal_th=117} "Boiler throughput coal t/h"
m_dot_coal=Coal_th*convert(ton/h;kg/s) "Coal massflow"
eta_boiler=0,952 "Boiler efficiency"
HV_coal=28700 "Heating Value coal"
Q_dot_coal=HV_coal*m_dot_coal "Rate of fuel energy input"
{T_out_boilersteam=580} "Steam temperature[C]"
{h_in_boiler=Enthalpy(Steam_IAPWS;T=T_out_K;P=P_fw_hp)} "Enthalpy"
h_out_boiler=Enthalpy(Steam_IAPWS;T=T_out_boilersteam;P=P_in_VHP) "Enthalpy"
Q_dot_coal*eta_boiler=m_dot_steamboiler*(h_out_boiler-h_in_boiler)+m_dot_reh1*
   (h_in_HP-h_out_VHP)+m_dot_reh2*(h_in_IP0-h_out_HP)
{m_dot_steamboiler} "Steam massflow trough boiler"
{P_out_boiler=28500}
P_out_boiler=P_fw_hp-P_fw_hp*0,13636
T_in_boiler=Temperature(Steam_IAPWS;P=P_fw_hp;h=h_in_boiler)
Q_dot_boiler_out=m_dot_steamboiler*h_out_boiler+m_dot_reh1*h_in_HP+m_dot_reh2*
   h_in_IP0-m_dot_steamboiler*h_in_boiler-m_dot_reh1*h_out_VHP-m_dot_reh2*h_out_HP

"VHP turbine"
P_in_VHP=P_out_boiler
{P_in_VHP=285*convert(bar;kPa)} "Inlet pressure[bar]"
s_in_VHP=Entropy(Steam_IAPWS;T=T_out_boilersteam;P=P_in_VHP) "Entropy"
{T_out_VHP=370} "Temperature"
{h_out_VHP=Enthalpy(Steam_IAPWS;T=T_out_VHP;P=P_in_HP)} "Enthalpy"
h_s_out_VHP=Enthalpy(Steam_IAPWS;P=P_in_HP;s=s_in_VHP) "Isentropic enthalpy"
{eta_s_VHP=(h_out_boiler-h_out_VHP)/(h_out_boiler-h_s_out_VHP)} "Isentropic effici
eta_s_VHP=0,93
h_out_VHP=-eta_s_VHP*(h_out_boiler-h_s_out_VHP)+h_out_boiler
T_out_VHP=Temperature(Steam_IAPWS;P=P_in_HP;s=s_in_VHP)
Q_dot_VHP=m_dot_steamboiler*(h_out_boiler-h_out_VHP)

"Reheating 1"
{T_out_reh1=580}

"HP turbine"
{P_in_HP=74*convert(bar;kPa)} "Inlet pressure[bar]"
P_in_HP=(P_in_VHP-P_cond)-(P_in_VHP-P_cond)*0,740323542829070
h_in_HP=Enthalpy(Steam_IAPWS;T=T_out_reh1;P=P_in_HP) "Enthalpy"
```

```
{h_out_HP=Enthalpy(Steam_IAPWS;T=T_out_HP;P=P_in_IP0)} "Enthalpy"
s_in_HP=Entropy(Steam_IAPWS;T=T_out_reh1;P=P_in_HP) "Entropy"
{T_out_HP=380} "Temperature"
{T_bleed2=480} "Temperature"
T_bleed2=(T_out_reh1+T_out_HP)/2
h_bleed2=Enthalpy(Steam_IAPWS;T=T_bleed2;s=s[3]) "Enthalpy"
P_HPbleed2=Pressure(Steam_IAPWS;T=T_bleed2;s=s[3]) "Pressure"
eta_s_HP=0,93
h_s_out_HP=Enthalpy(Steam_IAPWS;P=P_in_IP0;s=s_in_HP)
h_out_HP=-eta_s_HP*(h_in_HP-h_s_out_HP)+h_in_HP
T_out_HP=Temperature(Steam_IAPWS;P=P_in_IP0;s=s_in_HP)
Q_dot_HP=(m_dot_reh1-m_dot_bleed2)*(h_in_HP-h_out_HP)

"Reheating 2"
{T_out_reh2=580}

"IP0 turbine"
{P_in_IP0=19*convert(bar;kPa)} "Inlet pressure[bar]"
P_in_IP0=(P_in_VHP-P_cond)-(P_in_VHP-P_cond)*0,933326315050707
h_in_IP0=Enthalpy(Steam_IAPWS;T=T_out_reh2;P=P_in_IP0) "Enthalpy"
{T_bleed4=530}
T_bleed4=(T_out_reh2+T_in_IP)/2
h_bleed4=Enthalpy(Steam_IAPWS;T=T_bleed4;s=s[5])
P_IP0bleed4=Pressure(Steam_IAPWS;T=T_bleed4;s=s[5]) "Pressure"
h_s_out_IP0=Enthalpy(Steam_IAPWS;P=P_in_IP;s=s[5])
h_out_IP0=-eta_s_IP0*(h_in_IP0-h_s_out_IP0)+h_in_IP0
eta_s_IP0=0,91
Q_dot_IP0=(m_dot_reh2-m_dot_bleed4)*(h_in_IP0-h_in_IP)

"IP1/IP2"
{T_in_IP=429} "Temperature"
{T_in_IP=Temperature(Steam_IAPWS;P=P_in_IP;s=s[6])}
T_in_IP=Temperature(Steam_IAPWS;P=P_in_IP;h=h_out_IP0)
{P_in_IP=7,2*convert(bar;kPa)} "Pressure"
P_in_IP=(P_in_VHP-P_cond)-(P_in_VHP-P_cond)*0,974734182545531
{h_in_IP=Enthalpy(Steam_IAPWS;T=T_in_IP;P=P_in_IP)} "Enthalpy"
h_in_IP=h_out_IP0
{T_bleed6=330}
T_bleed6=(T_in_IP+T_in_LP2)/2
h_bleed6=Enthalpy(Steam_IAPWS;T=T_bleed6;s=s[6])
P_bleed6=Pressure(Steam_IAPWS;T=T_bleed6;s=s[6])
eta_s_IP1=0,9
eta_s_IP2=0,9
h_s_out_IP1=Enthalpy(Steam_IAPWS;P=P_in_LP1;s=s[6])
h_s_out_IP2=Enthalpy(Steam_IAPWS;P=P_in_LP2;s=s2[6])
```

```
h_out_IP1=-eta_s_IP1*(h_in_IP-h_s_out_IP1)+h_in_IP
h_out_IP2=-eta_s_IP2*(h_in_IP-h_s_out_IP2)+h_in_IP
Q_dot_IP1=(m_dot_out_IP1+m_dot_bleed7)*(h_in_IP-h_in_LP1)
Q_dot_IP2=(m_dot_out_IP2+m_dot_bleed8)*(h_in_IP-h_in_LP2)

"LP1 turbine"
{P_in_LP1=1,5*convert(bar;kPa)} "Pressure"
P_in_LP1=(P_in_VHP-P_cond)-(P_in_VHP-P_cond)*0,994736288030319
{T_in_LP1=233}
T_in_LP1=Temperature(Steam_IAPWS;P=P_in_LP1;s=s[7])
{T_bleed9=81} "Temperature"
T_bleed9=(T_in_LP1+T_in_cond)/2
h_bleed9=Enthalpy(Steam_IAPWS;T=T_bleed9;x=1)
{h_in_LP1=Enthalpy(Steam_IAPWS;T=T_in_LP1;P=P_in_LP1)} "Enthalpy"
h_in_LP1=h_out_IP1
P_bleed9=Pressure(Steam_IAPWS;T=T_bleed9;h=h_bleed9)
eta_s_LP1=0,9
h_s_out_LP1=Enthalpy(Steam_IAPWS;P=P_cond;s=s[7])
h_out_LP1=-eta_s_LP1*(h_in_LP1-h_s_out_LP1)+h_in_LP1
Q_dot_LP1=(m_dot_out_IP1-m_dot_bleed9)*(h_in_LP1-h_out_LP1)

"LP2 turbine"
{P_in_LP2=0,7*convert(bar;kPa)} "Pressure"
P_in_LP2=(P_in_VHP-P_cond)-(P_in_VHP-P_cond)*0,997543601080816
{T_in_LP2=154} "Temperature"
s_out_LP2=Entropy(Steam_IAPWS;T=T_in_LP2;P=P_in_LP2)
T_in_LP2=Temperature(Steam_IAPWS;P=P_in_LP2;s=s_out_LP2)
{h_in_LP2=Enthalpy(Steam_IAPWS;T=T_in_LP2;P=P_in_LP2)} "Enthalpy"
h_in_LP2=h_out_IP2
{T_bleed10=46}
T_bleed10=(T_in_LP2+T_in_cond)/2
h_bleed10=Enthalpy(Steam_IAPWS;T=T_bleed10;s=s_out_LP2)
P_bleed10=Pressure(Steam_IAPWS;T=T_bleed10;s=s_out_LP2)
eta_s_LP2=0,9
h_s_out_LP2=Enthalpy(Steam_IAPWS;P=P_cond;s=s2[7])
h_out_LP2=-eta_s_LP2*(h_in_LP2-h_s_out_LP2)+h_in_LP2
Q_dot_LP2=(m_dot_out_IP2-m_dot_bleed10)*(h_in_LP2-h_out_LP2)

"Generator"
Q_dot_turbine=Q_dot_VHP+Q_dot_HP+Q_dot_IP0+Q_dot_IP1+Q_dot_IP2+Q_dot_LP1+Q_dot_LP2
eta_el=W_dot_el*convert(MW;kJ/s)/Q_dot_coal "Thermal efficiency condensation opera
{W_dot_el=411} "Output"
W_dot_el=eta_gen*Q_dot_turbine*convert(kJ/s;MW)
eta_gen=0,9341 "Power factor"
```

```
"Condenser"
T_in_cond=Temperature(Steam_IAPWS;P=P_cond;x=x_in_cond) "Temperature"
P_cond=0,03*convert(bar;kPa) "Pressure"
h_in_cond=Enthalpy(Steam_IAPWS;x=x_in_cond;P=P_cond) "Enthalpy"
h_out_cond=Enthalpy(Steam_iapws;x=0;P=P_cond) "Enthalpy"
x_in_cond=0,917 "Quality"
v_out_cond=Volume(Steam_IAPWS;P=P_cond;x=0)

T_out_cond_mix=Temperature(Steam_IAPWS;P=P_cond;h=h_out_cond_mix)
m_dot_out_LP*h_out_cond+m_dot_bleed10*h_bleed10_water=
   m_dot_out_cond*h_out_cond_mix

"Main condensate pumps"
P_fw_cp=5*convert(bar;kPa) "Feed water pressure"
h_fw_cp=h_out_cond+w_in_fw_cp "Enthalpy"
w_in_fw_cp=v_out_cond*(P_fw_cp-P_cond)
T_out_fw_cp=Temperature(Steam_IAPWS;h=h_fw_cp;P=P_fw_cp)

"Preheater A"
T_bleed10_water=T_out_fw_cp+5
h_bleed10_water=Enthalpy(Steam_IAPWS;T=T_bleed10_water;x=0)
T_out_A=Temperature(Steam_IAPWS;P=P_fw_cp;h=h_out_A)

m_dot_out_cond*h_fw_cp+m_dot_bleed10*h_bleed10=
   m_dot_out_cond*h_out_A+m_dot_bleed10*h_bleed10_water

"Preheater B"
m_dot_out_cond*h_out_A+m_dot_bleed9*h_bleed9=m_dot_in_C*h_in_C

"Preheater C"
T_in_C=Temperature(Steam_IAPWS;P=P_fw_cp;h=h_in_C)
m_dot_in_C*h_in_C+m_dot_bleed8*h_in_LP2=m_dot_in_D*h_in_D

"Feed water tank D"
v_out_D=Volume(Steam_IAPWS;P=P_fw_cp;x=0)
h_out_D=Enthalpy(Steam_iapws;x=0;P=P_fw_cp)
T_in_D=Temperature(Steam_IAPWS;P=P_fw_cp;h=h_in_D)
T_out_D=Temperature(Steam_IAPWS;P=P_fw_cp;h=h_out_D)
m_dot_in_D*h_in_D+m_dot_bleed7*h_in_LP1=m_dot_out_D*h_out_D

"Low pressure feed pumps"
P_fw_lp=33*convert(bar;kPa) "Feed water pressure"
w_in_fw_lp=v_out_D*(P_fw_lp-P_fw_cp)
h_fw_lp=h_out_D+w_in_fw_lp
T_out_fw_lp=Temperature(Steam_IAPWS;P=P_fw_lp;h=h_fw_lp)
```

```
"Preheater E"
m_dot_out_D*h_fw_lp+m_dot_bleed5*h_in_IP+m_dot_bleed6*h_bleed6=m_dot_in_G*h_in_G

"Preheater F"
{T_in_F=T_in_G-30}
h_in_F=Enthalpy(Steam_IAPWS;T=T_in_F;P=P_fw_lp)
T_bleed5_water=T_in_F+5
h_bleed5_water=Enthalpy(Steam_IAPWS;T=T_bleed5_water;x=0)

m_dot_in_G*h_in_F+m_dot_bleed5*h_in_IP=
   m_dot_in_G*h_in_G+m_dot_bleed5*h_bleed5_water

"Preheater G"
{T_in_G=T_in_H-15}
h_in_G=Enthalpy(Steam_IAPWS;T=T_in_G;P=P_fw_lp)
m_dot_in_G*h_in_G+m_dot_bleed4*h_bleed4=m_dot_in_H*h_in_H

"Feed water tank H"
v_out_H=Volume(Steam_IAPWS;P=P_fw_lp;x=0)
h_out_H=Enthalpy(Steam_iapws;x=0;P=P_fw_lp)
T_out_H=Temperature(Steam_IAPWS;P=P_fw_lp;h=h_out_H)
T_Hbleed_1_2=T_out_fw_hp+5
h_bleed_1_2=Enthalpy(Steam_iapws;T=T_Hbleed_1_2;x=0)
h_in_H*m_dot_in_H+h_out_HP*m_dot_bleed3+h_bleed_1_2*m_dot_bleed_1_2=
   h_out_H*m_dot_out_H
T_in_H=Temperature(Steam_IAPWS;P=P_fw_lp;h=h_in_H){T_in_H=230}

"Boiler feed pumps"
{P_fw_hp=330*convert(bar;kPa)} "Feed water pressure"
w_in_fw_hp=v_out_H*(P_fw_hp-P_fw_lp)
h_fw_hp=h_out_H+w_in_fw_hp
T_out_fw_hp=Temperature(Steam_IAPWS;P=P_fw_hp;h=h_fw_hp)

"Bypass"
{b=0}
m_dot_in_K=m_dot_out_H*(1-b)
m_dot_out_H*(1-b)*h_out_K+m_dot_out_H*b*h_fw_hp=m_dot_out_H*h_in_boiler
m_dot_bypass=m_dot_out_H*b
"HP heaters"
m_dot_out_H*(1-b)*h_fw_hp+m_dot_bleed2*h_bleed2+m_dot_bleed1*h_out_VHP=
   m_dot_out_H*(1-b)*h_out_K+m_dot_bleed_1_2*h_bleed_1_2
m_dot_out_H*(1-b)*h_fw_hp+m_dot_bleed2*h_out_K_steam+m_dot_bleed1*h_out_VHP=
   m_dot_out_H*(1-b)*h_in_K+m_dot_bleed_1_2*h_bleed_1_2
```

```
"Preheater K"
{T_in_K=292} "Temperature"
T_in_K=Temperature(Steam_IAPWS;P=P_fw_hp;h=h_in_K)
{v_1=Volume(Steam_IAPWS;x=0;P=P_fw_hp)}
{h_in_K=Enthalpy(Steam_IAPWS;T=T_in_K;P=P_fw_hp)} "Enthalpy"
T_out_K_steam=T_in_K+5 "Temperature"
h_out_K_steam=Enthalpy(Steam_IAPWS;T=T_out_K_steam;P=P_HPbleed2) "Enthalpy"
{T_out_K=300} "Temperature"
T_out_K=Temperature(Steam_IAPWS;P=P_fw_hp;h=h_out_K)
h_out_K=Enthalpy(Steam_IAPWS;T=T_out_K;P=P_fw_hp) "Enthalpy"
m_dot_out_H*(1-b)*(h_in_K-h_out_K)=m_dot_bleed2*(h_out_K_steam-h_bleed2)

"Arrays"
T[1]=T_out_boilersteam
T[2]=T_out_VHP
T[3]=T_out_reh1
T[4]=T_out_HP
T[5]=T_out_reh2
T[6]=T_in_IP
T2[6]=T_in_IP
T[7]=T_in_LP1
T2[7]=T_in_LP2
{T[8]=T_in_LP2}
T[9]=T_in_cond
T2[9]=T_in_cond
T[10]=T[9]
T[11]=T_out_cond_mix
T[12]=T_out_fw_cp
T[13]=T_out_A
T[14]=T_in_C
T[15]=T_in_D
T[16]=T_out_D
T[17]=T_out_fw_lp
T[18]=T_in_F
T[19]=T_in_G
T[20]=T_in_H
T[21]=T_out_H
T[22]=T_out_fw_hp
{T[23]}
T[24]=T_in_K
T[25]=T_out_K
T[26]=Temperature(Steam_IAPWS;P=P[26];s=s[26])
T[27]=Temperature(Steam_IAPWS;P=P[27];s=s[27])
T[28]=Temperature(Steam_IAPWS;P=P[28];s=s[28])
T[29]=Temperature(Steam_IAPWS;P=P[29];s=s[29])
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```
T[30]=Temperature(Steam_IAPWS;P=P[30];s=s[30])
T[31]=Temperature(Steam_IAPWS;P=P[31];s=s[31])
T[32]=Temperature(Steam_IAPWS;P=P[32];s=s[32])
T[33]=Temperature(Steam_IAPWS;P=P[33];s=s[33])
T[34]=Temperature(Steam_IAPWS;P=P[34];s=s[34])
T[35]=Temperature(Steam_IAPWS;P=P[35];s=s[35])
T[36]=Temperature(Steam_IAPWS;P=P[36];s=s[36])
T[37]=Temperature(Steam_IAPWS;P=P[37];s=s[37])
T[38]=Temperature(Steam_IAPWS;P=P[38];s=s[38])
T[39]=Temperature(Steam_IAPWS;P=P[39];s=s[39])
T[40]=Temperature(Steam_IAPWS;P=P[40];s=s[40])

s[1]=s_in_VHP
s[2]=entropy(Steam_IAPWS;h=h_out_VHP;P=P_in_HP)
s[3]=s_in_HP
s[4]=Entropy(Steam_IAPWS;h=h_out_HP;P=P_in_IP0)
s[5]=Entropy(Steam_IAPWS;T=T_out_reh2;P=P_in_IP0)
s[6]=Entropy(Steam_IAPWS;h=h_out_IP0;P=P_in_IP)
s2[6]=Entropy(Steam_IAPWS;h=h_out_IP0;P=P_in_IP)
s[7]=Entropy(Steam_IAPWS;h=h_out_IP1;P=P_in_LP1)
s2[7]=Entropy(Steam_IAPWS;h=h_out_IP2;P=P_in_LP2)
s[8]=Entropy(Steam_IAPWS;T=T_in_LP2;P=P_in_LP2)
s[9]=Entropy(Steam_IAPWS;h=h_out_LP1;P=P_cond)
s2[9]=Entropy(Steam_IAPWS;h=h_out_LP2;P=P_cond)
s[10]=Entropy(Steam_IAPWS;x=0;P=P_cond)
s[11]=Entropy(Steam_IAPWS;T=T_out_cond_mix;x=0)
s[12]=Entropy(Steam_IAPWS;T=T_out_fw_cp;P=P_fw_cp)
s[13]=Entropy(Steam_IAPWS;T=T_out_A;P=P_fw_cp)
s[14]=Entropy(Steam_IAPWS;T=T_in_C;P=P_fw_cp)
s[15]=Entropy(Steam_IAPWS;T=T_in_D;P=P_fw_cp)
s[16]=Entropy(Steam_IAPWS;x=0;P=P_fw_cp)
s[17]=Entropy(Steam_IAPWS;T=T_out_fw_lp;P=P_fw_lp)
s[18]=Entropy(Steam_IAPWS;T=T_in_F;P=P_fw_lp)
s[19]=Entropy(Steam_IAPWS;T=T_in_G;P=P_fw_lp)
s[20]=Entropy(Steam_IAPWS;T=T_in_H;P=P_fw_lp)
s[21]=Entropy(Steam_IAPWS;x=0;P=P_fw_lp)
s[22]=Entropy(Steam_IAPWS;T=T_out_fw_hp;P=P_fw_hp)
{s[23]}
s[24]=Entropy(Steam_IAPWS;T=T_in_K;P=P_fw_hp)
s[25]=Entropy(Steam_IAPWS;T=T_out_K;P=P_fw_hp)
s[26]=s[25]+(s[1]-s[25])/15
s[27]=s[26]+(s[1]-s[25])/15
s[28]=s[27]+(s[1]-s[25])/15
s[29]=s[28]+(s[1]-s[25])/15
s[30]=s[29]+(s[1]-s[25])/15
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s[31]=s[30]+(s[1]-s[25])/15
s[32]=s[31]+(s[1]-s[25])/15
s[33]=s[32]+(s[1]-s[25])/15
s[34]=s[33]+(s[1]-s[25])/15
s[35]=s[34]+(s[1]-s[25])/15
s[36]=s[35]+(s[1]-s[25])/15
s[37]=s[36]+(s[1]-s[25])/15
s[38]=s[37]+(s[1]-s[25])/15
s[39]=s[38]+(s[1]-s[25])/15
s[40]=s[39]+(s[1]-s[25])/15

P[26]=P_fw_hp-(P_fw_hp-P_out_boiler)/15
P[27]=P[26]-(P_fw_hp-P_out_boiler)/15
P[28]=P[27]-(P_fw_hp-P_out_boiler)/15
P[29]=P[28]-(P_fw_hp-P_out_boiler)/15
P[30]=P[29]-(P_fw_hp-P_out_boiler)/15
P[31]=P[30]-(P_fw_hp-P_out_boiler)/15
P[32]=P[31]-(P_fw_hp-P_out_boiler)/15
P[33]=P[32]-(P_fw_hp-P_out_boiler)/15
P[34]=P[33]-(P_fw_hp-P_out_boiler)/15
P[35]=P[34]-(P_fw_hp-P_out_boiler)/15
P[36]=P[35]-(P_fw_hp-P_out_boiler)/15
P[37]=P[36]-(P_fw_hp-P_out_boiler)/15
P[38]=P[37]-(P_fw_hp-P_out_boiler)/15
P[39]=P[38]-(P_fw_hp-P_out_boiler)/15
P[40]=P[39]-(P_fw_hp-P_out_boiler)/15

$SHOWWINDOW Diagram
{$SHOWWINDOW Equations}
{$SHOWWINDOW Plot}
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C

Investigating change in temperatures and pressure

C.1 Introduction
C.2 Increasing the temperature and pressure

C.1 Introduction

In this appendix a investigation of the potential of ultra-super critical steam data is performed. Results are presented for increased temperature from the boiler and pressure from the high pressure feedwater pumps.

C.2 Increasing the temperature and pressure

In order to investigate the influences of increasing the temperature and pressure of the steam from the boiler, these values are increased gradually up to 760 °C and 42 MPa. In table C.1 values are given for chosen inputs and outputs from the model. The inputs are temperature from the boiler and pressure from the high pressure feedwater pump. The outputs are mass flow rate through the boiler, output from generator, thermal efficiency, the mass flow rate through the condenser and the temperature of the feedwater into the boiler.
The results were obtained by altering the values for temperature and pressure in the order; $T_{out\ reh2}$, $T_{out\ reh1}$, $T_{in\ VHP}$ and $P_{hp\ fw}$.

Increasing temperature and pressure

Variable	Change	$\dot{m}_{i\ b}$ [kg/s]	$\dot{W}_{el}$ [MW]	η [%]	$\dot{m}_{i\ cond}$ [kg/s]	$T_{i\ b}$ [°C]
		272	407.4	48.15	158.6	299.2
$T_{out\ reh2}$[°C]	600	269.9	407.4	48.14	157.8	299.2
$T_{out\ reh1}$[°C]	600	268.9	407.7	48.19	158.2	299.2
$T_{in\ VHP}$[°C]	600	267.1	412.6	48.76	159.1	299.2
$P_{hp\ fw}$[kPa]	34000	267.4	412.8	48.79	159.6	299.2
$T_{out\ reh2}$[°C]	620	264.2	411.4	48.61	160	299.2
$T_{out\ reh1}$[°C]	620	263.2	413.6	48.88	157.2	299.2
$T_{in\ VHP}$[°C]	620	261.4	414.3	48.96	157.8	299.2
$P_{hp\ fw}$[kPa]	35000	261.6	412.6	48.76	158.8	299.2
$T_{out\ reh2}$[°C]	630	260	413.1	48.82	158	299.2
$T_{out\ reh2}$[°C]	640	258.5	414.3	48.96	158.3	299.2
$T_{out\ reh1}$[°C]	640	257.5	415.3	49.08	157.9	299.2
$T_{in\ VHP}$[°C]	640	255.6	418.4	49.44	155.6	299.2
$P_{hp\ fw}$[kPa]	36000	255.8	418	49.4	154.5	299.2
$T_{out\ reh2}$[°C]	650	254.3	420.1	49.65	156.1	299.2
$T_{out\ reh2}$[°C]	660	254.5	419.6	49.59	156.2	299.2
$T_{out\ reh1}$[°C]	660	253.5	420.2	49.66	156.2	299.2
$T_{in\ VHP}$[°C]	660	249.5	423.5	50.05	153.9	299.1
$P_{hp\ fw}$[kPa]	37000	249.3	423.9	50.1	154	299.1
$T_{out\ reh2}$[°C]	680	249.4	422.7	49.96	154	299.1
$T_{out\ reh1}$[°C]	670	249.4	423.3	50.03	153.9	299.1
$T_{out\ reh1}$[°C]	680	248	423.9	50.1	153.8	299.1
$T_{in\ VHP}$[°C]	680	246.6	424.1	50.12	152.4	299
$P_{hp\ fw}$[kPa]	38000	246.7	424.6	50.18	152.4	299
$T_{out\ reh2}$[°C]	700	244.3	425	50.23	151.1	299
$T_{out\ reh1}$[°C]	700	243.3	425.8	50.32	150.7	299
$T_{in\ VHP}$[°C]	700	241.6	425.8	50.32	148.6	299
$P_{hp\ fw}$[kPa]	39000	241.8	426.2	50.37	148.8	299
$T_{out\ reh2}$[°C]	720	239.3	428.3	50.62	149.5	299
$T_{out\ reh1}$[°C]	720	238.4	429.2	50.73	149.2	299
$T_{in\ VHP}$[°C]	720	236.8	430.6	50.89	148.5	298.9
$P_{hp\ fw}$[kPa]	40000	237.1	429.2	50.72	146.3	299
$T_{out\ reh2}$[°C]	740	234	429.7	50.78	144.9	299
$T_{out\ reh1}$[°C]	740	233.1	430.5	50.88	144.8	298.9
$T_{in\ VHP}$[°C]	740	231.7	431.5	51	143.9	298.9
$P_{hp\ fw}$[kPa]	41000	232.2	435	51.41	147	298.9
$T_{out\ reh2}$[°C]	750	231.1	433.4	51.22	144.8	298.9
$T_{out\ reh2}$[°C]	760	229.8	432.4	51.1	143.1	298.9
$T_{out\ reh1}$[°C]	760	228.8	435.4	51.45	146.1	298.9
$T_{in\ VHP}$[°C]	760	227.6	440.1	52.01	143.9	298.9
$P_{hp\ fw}$[kPa]	42000	227.6	441	52.12	143	298.9

Table C.1: *Change in results when increasing pressure from the boiler feed-water pumps and temperature from the boiler.*

D
Nickel based superalloy

Figure D.1: *This is the guy who makes it possible for ultra super critical power plants to reach steam temperatures and pressures of* $750°C/35MPa$. *Courtesy Franck Tancret*