Oskarshamn 3 – Optimization after power uprate

- A heat balance deviation analysis

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Preface

First, I would like to give special thanks to my supervisors for enabling this thesis: Carl Möller at OKG and Magnus Genrup at LTH. Your inspiration and great knowledge have been invaluable. Second, I like to thank all colleagues at OKG who have been to a great help whenever I have had a problem and also for all the fun discussions in the fikarum. Especially I would like to thank Berth Arbman, Bertil Persson and Fredrik Sturek at OKG for help and discussion about the thesis.

I have learned a lot writing this thesis, and I will take it with me in the future.

Lund 6th May 2013

Robert Kronblad

Abstract

Oskarshamn 3 has undergone a power uprate from 110% to 129.1% of original reactor thermal output in recent ended project PULS (Power Uprate with Licensed Safety). The main objective of this thesis have been to identify deviations in the power plant, explaining the differences in power output between designed and achieved power output after PULS. The investigation has been conducted with the use of classical thermodynamic heat balance calculations using computer software Probera.

Prior to the heat balance calculations, a literature study has been done to investigate how wet steam turbines can be modeled with a special focus on the turbine capacity or wideness. The literature study led to an implementation of capacity model Beckmann in Probera which gave a better description of the plants performance and behavior off-design.

The heat balance calculations showed that two major differences in the plant has occurred in comparison to what was designed:

- A wide HP-turbine which leads to a power loss in the region of 9-14 MW
- Higher condenser pressure than designed which leads to a power loss of 4-8 MW. The loss in power is however variable with a variable temperature in the cooling water, i.e. the Baltic Sea. At higher temperatures, around 15°C the loss in power output is none.

The high condenser pressure can be explained by a variable blockage of tubes in the condenser with more tubes blocked with decreeing CW-temperature, which lead to the following reasoning about possible explanations:

- Problems with ejector system, off-design operation or design flaw.
- Choking of the steam and/or disturbed flow pattern on shell side.
- Loss of siphon effect on tube side which leads to a lower water level in water boxes and hence no water will be present in upper tubes.
- Air flow on tube side causing plugging two-phase flow and decreeing heat transfer coefficient.

The explanations don’t stand alone but are even more plausible if one takes all or some of them into account.

To confirm or to rule out some of the explanations, suggestions have been made to measure temperature and pressure over the intercondensers in the ejector system and to measure outlet temperature of strategically located tubes in the condenser.

Keywords: Heat balances, Power output, Deviations

Sammanfattning

Oskarshamn 3 har nyligen genomgått en effekthöjning från 110 % till 129.1% av ursprunglig reaktor effekt i det nyligen avslutade projekt PULS. Huvudmålet med uppsatsen har varit att konstatera avvikelser i kraftverket som kan förklara skillnaden i uttagen eleffekt mellan designad och erhållen effekt efter PULS. Undersökningen har i huvudsak genomförts med klassiska termodynamiska värmebalansberäkningar med hjälp av datorprogramet Probera.

Före de faktiska värmebalansberäkningarna genomfördes en literaturstudie med inriktning mot hur våta ångturbiner kan modeleras med särkilt fokus på vidheten hos turbinen. Literaturstudien ledde till en implementering av vidhetsmodellen Beckmann i Probera vilket gav en bättre beskrivning av hur kraftverket betedde sig off-design.

Värmebalansberäkningarna visade att det i huvudsak är två stora avvikelser i kraftverket jämfört med vad man designade för:

- En vid högtryckturbin vilket ger ett effekttapp kring 9-14 MW
- Högre kondensortryck än förväntat, vilket ger ett effekttapp kring 4-8 MW. Effekttappet är dock variabelt med varierande temperatur på kylvattnet, alltså Östersjön. Vid cirka 15°C och högre är effekttappet obefintligt.

Det högre kondensortrycket kan förklaras med en variabel blockering av tuber i kondensorn med högre andel tuber blockerade vid fallande kylvattentemperatur. Detta ledde till följande förklaringsmodeller:

- Problem med ejektorsystemet, antingen av oväntade driftsfall eller konstruktionsfel.
- Chokning eller ändrad strömning av ångflödet genom tuberna på mantelsidan.
- Tapp av häverteffekten på tubsidan som ger en lägre vattennivå i vattenkamrarna och följaktligen får man inget kylvatten i de övre tuberna.
- Luftflöde på tubsidan vilket ger två-fas flöde av plugg-typ och som följd erhålls en lägre värmeöverföringskoefficient än förväntat.

Förklaringsmodellerna kan förklara det högre trycket enskilt, men ännu mer troligt är att en eller flera av dessa samverkar.

För att bekräfta eller för att utesluta några av förklaringarna föreslås en uppmätning av tryck och temperatur över interkondensorna i ejektorsystemet samt uppmätning av temperaturen i utloppet av strategiskt valda tuber i kondensorn.

NYCKELORD: Värmebalanser, Effekt, Avvikelser

Nomenclature

Arabic Symbol	Unit	Description
A	[m^2]	Area
C_T	[m^2]	Turbine constant
c	[m/s]	Speed
H	[J/kg]	Enthalpy
$\dot{m}$	[kg/s]	Mass flow
n	[-]	Polytropic coefficient
P	[Pa]	Pressure
Q	[W]	Heat flux
T	[K]	Temperature
U	[$W/(m^2 \cdot K)$]	Overall heat transfer coefficient
u	[m/s]	Rotor speed
v	[m^3/kg]	Specific volume
x	[kg/kg]	Steam quality
Y	[kg/kg]	Moisture content

Greek Symbol	Unit	Description
α	[-]	Baumannfactor
Δ	[-]	Difference
η	[%]	Efficiency
Ρ	[kg/m^3]	Density

Subscript		
0		Stagnation
des		Design
dry		Dry
i		Inlet
j		Outlet
lmtd		Logarithmic mean temperature difference
wet		Wet

Abbreviations		
BPV		Bypass valve
BWR		Boiling water reactor
CW		Cooling water
HP		High-pressure
HPFH		High Pressure Feed heater
LP		Low-pressure
LPFH		Low Pressure Feed heater
MSR		Moisture separator and Reheater
RTO		Reactor Thermal Output

Contents

1. Introduction

1.1 Background

The nuclear power plant Oskarshamn 3 located in Oskarshamn, Sweden, is a boiling water reactor (BWR) which in the recent ended project PULS was designed for a power output of about 1465 MW electric power with a temperature of 5°C in the cooling water. Some time has passed since the project ended and the power output hasn't quite achieved the designed value. The deviation in power output can be explained by a number of reasons in the plant and the aim with the thesis is to highlight some of these reasons and give plausible explanations to the behavior of the plant.

1.2 Problem

Why is there a deviation in power output and what is the reason?

1.3 Purpose

Investigate differences in achieved power output after the power uprate in comparison to the power output designed. Suggest how the designed power output could be achieved.

1.4 Limitations

Only the Turbine system will be investigated, the nuclear reactor and the generator or electric substations will not be included.

Modeling – Only the turbine component model is thoroughly investigated and examined as well as somewhat improved. The other components in the heat balance model are only briefly examined regarding, for example, pressure losses. Only steady-state conditions are considered.

2. Methodology

The investigation was mainly made with the use of classical thermodynamic heat balance calculations with the use of computer software Probera[1] which was designed by Bertil Persson at OKG. Some hand calculations were necessary to give input values to the model. To be able to describe the Turbine and the surrounding systems as realistic as possible, some literature studies have been conducted to verify and improve the program's component models prior to the actual heat balance calculations. The literature study can further be examined in chapter 3 Theory – Turbine modeling.

After the heat balance calculations, the major deviations were then further investigated via reports, drawings, measurements, calculations and interviews. See section 5.6 Deviation Analysis.

2.1 Programs used

Probera has been used for heat balance calculations, Process Explorer [2] has been used to collect and retrieve data from measurements and Excel [3] has been used for data analysis.

2.2 Procedure

Four heat balance models were considered in this thesis were three were Probera models:

- Original Designed heat balance model retrieved from Turbine supplier
- Probera Design model, aiming to imitate the original designed model
- Adjusted model, retrieved with the adjustment tool in Probera
- Adjusted model with the assumption of a condenser with a fouling factor of $3.0 \cdot 10^{-5}$ ("clean")

The new Turbine component model Beckmann introduced in chapter 3 was developed and tuned accordingly to the Turbine supplier's heat balance model for steady-state conditions at 5°C in the cooling water. This model is denominated Probera Design model.

The adjustment calculating tool in Probera was used to fit the parameters to the measured data at 5°C. This was done both in the HP- and LP turbines. To make sure that the calculation was successful the opening percentage of the bypass valve was chosen as a parameter together with the HP turbines and for the LP turbines the fouling factor in the condenser was chosen as parameter. This heat balance model is denoted the Adjusted model.

For reasoning, the Adjusted model was also examined with the parameter fouling factor put to its initial value and this heat balance model is therefore denoted Adjusted model with a clean condenser.

To show how the power output or condenser pressure varies with the cooling water temperature, a so called Table calculation was made were the parameter was the inlet CW-temperature. This parameter was changed in steps and the heat balance model was adjusted accordingly. Steps of 0.5°C were used in this report with a span of 0°C to 16°C or 18°C if possible. This was been done for the Probera Design model and also the Adjusted model with clean condenser. The Table calculation also provided information about condenser pressure, exhaust losses, values on parameters at measurement points and so on.

To describe further differences in the production curve, the parameter blockage percentage in the component model of the condenser was used. This means that some of the tubes are “deactivated” or rather that some of the area is unavailable. Table calculations using a constant blockage of tubes were made for the Adjusted model with the assumption of a clean condenser.

The major differences found when the heat balance models were compared were further investigated through calculations and discussion about plausible reasons and causes.

2.3 Measurement treatment

The measurements have been retrieved with Process Explorer together with Probera. Process Explorer was used to evaluate suitable times for the collection of data, for example at part-load conditions. Probera was then used to collect the actual data from DRUS (DRiftens Underhållnings System) which basically is a server were data is gathered. The collected data was exported to Excel were it was further analyzed and for all measurements a 2nd order polynomial fit using a least square method of the data was used for comparison with modeled data. The measurements can further be studied in appendix 9.4 and 9.5.

3. Overview of the plant

O3 is the world's largest [4] boiling water reactor (BWR) with a recognized low CO_2-footprint [5] which today is more important than ever considering the climate change issue. According to UNs climate change panel (IPCC), the global temperature rise needs to be kept to a less than 2°C and electricity from nuclear power plants could be a part of the solution.

The reactor of O3 (constructed by Asea-Atom [4]) delivers steam to the turbine at a temperature of about 270°C with a pressure of about 70 bar. I.e the steam is not superheated but the quality is rather 99 % at the inlet to the high pressure turbine. This is special for nuclear power plants in general, the thermodynamic admission data are fairly low which leads to high moisture contents and therefore moisture separation is essential. The behavior of some components and also the modeling of said components are therefore more complicated or at least more extensive.

Oskarshamn 3 consists mainly of one High Pressure (HP) turbine and three Low Pressure (LP) turbines together with one condenser, 2 high pressure feed heaters (HPFH), 4 low pressure feed heaters (LPFH), one of which is a feed water tank/dearator, and two moisture separators and reheaters (MSR's). The bypass valve (BPV) keeps the pressure in the reactor to 70 bar which is essential for a good and stable moderation. In Figure 1, an overview of the plant's main components is presented.

The picture is somewhat simplified, the polishing plant between LPFH 1 and 2 is left out, as is the double HPFH flow chains after the feed water pumps. The gland system and ejector system is also left out. To keep the condenser clean on tube side, the so called Taprogge© system is installed, cleaning the tubes with small balls.

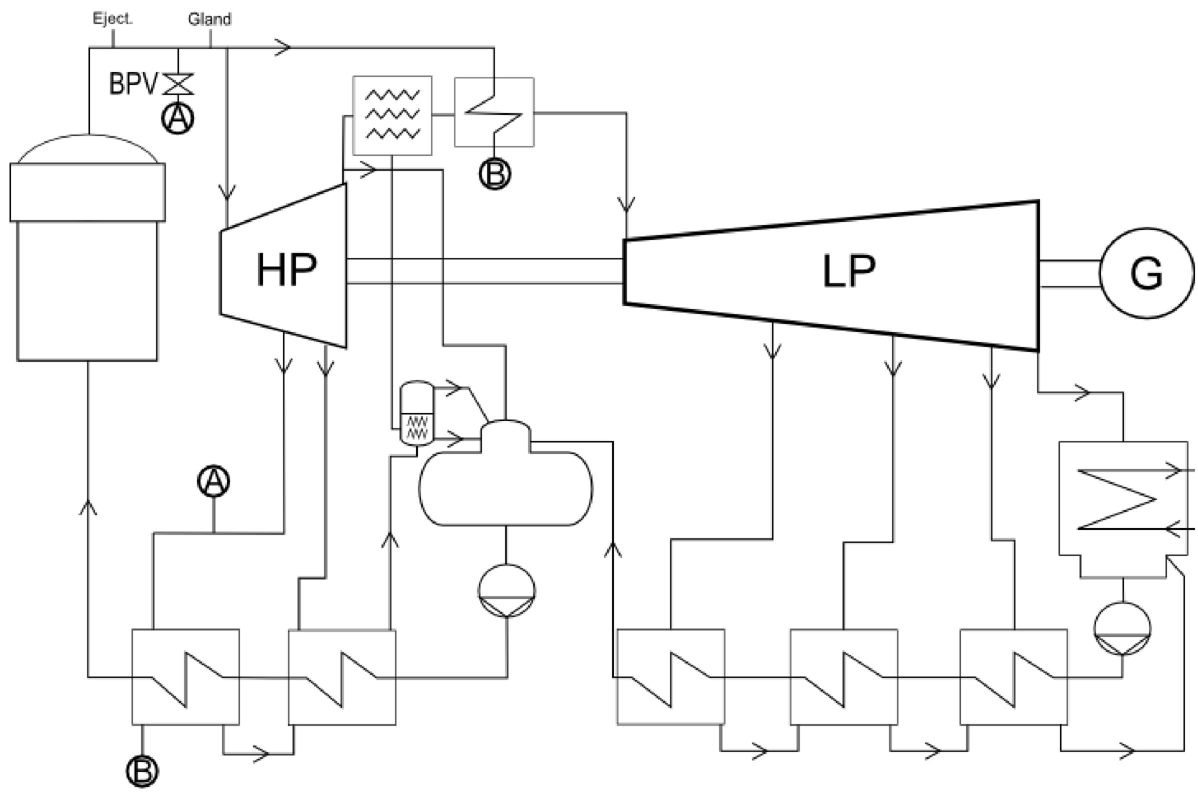

Figure 1, Schematic Overview of the main components in the plant.

A schematic view of the turbine sections and extractions can be seen in Figure 2.**Error! Reference source not found.** The turbines are dual-flow with steam inlet at the center of the turbine, and the extractions are all symmetrical except extraction 2 and 3 in the LP-turbine were they are unsymmetrical as seen in the figure.

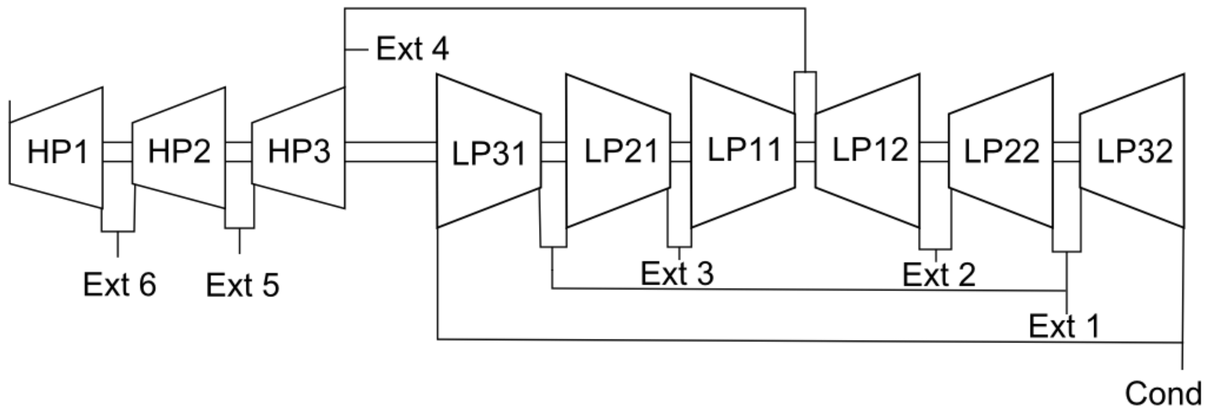

Figure 2, Schematic illustrations of Turbine sections and extractions as they are modeled.

For a good and stable operation of the reactor and for a stable moderation of the neutrons, the reactor needs to be kept at a pressure of 70 bar at all times. The pressure is kept by the BPV which is smaller than the conventional control valves to the turbine and also much faster in its operation. This allows the pressure in the reactor to maintain constant but at the same time the large and heavy control valves don’t need to operate under high speeds which means heavy loads. This is very beneficial, but the drawback is the “loss” of high valued steam which just passes the turbines without producing useful work.

4. Theory – Turbine modeling

To be able to estimate differences in the plant, such as power, pressure, temperature and mass flows, the type of component models in Probera are very important. The program has a very sophisticated modeling of the feed heaters, condenser and so on with correlations for heat transfer, pressure losses etc. However, the component model of the turbine is somewhat old-fashioned. The program is using Aurel Stodola to describe the swallowing capacity of the turbine, i.e. the connection between pressure and flow, which he empirically found in the 1920's. To achieve a better modeling, Beckmann is introduced instead in the turbine component model. The following chapter describes some of the features of a turbine and how it could be modeled.

4.1 Swallowing Capacity

In a turbine in general (steam or gas) there is a tight connection between the mass flow ($\dot{m}$) the pressure P and implicit the temperature T.

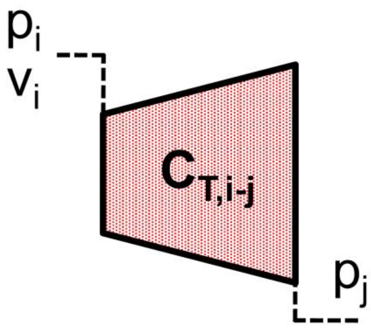

Figure 3, Schematic illustration of a turbine with inlet (i) and outlet (j) conditions linked with the swallowing capacity $C_{T,i-j}$.

Aurel Stodola[6] showed empirically that the connection between inlet and outlet conditions can be written:

$$\dot{m} = C_{T,i-j}\sqrt{\frac{p_i^2 - p_j^2}{p_i v_i}} = C_{T,i-j}\sqrt{\frac{p_i}{v_i}}\sqrt{1 - \frac{1}{\left(\frac{p_i}{p_j}\right)^2}} \ldots (1)$$

Where the constant C_T [m^2] is the so called turbine capacity or swallowing capacity which is a measure of the size of the turbine or the area of the flow channel. The equation is valid for all turbines including a turbine section, for example between extractions. The equation is fundamental to be able to describe how flows, pressures and temperatures vary off-design and how they are linked to each other and is an important tool to predict changes in the plants performance. The equation can be solved for the constant C_T and takes on the form of

$$C_{T,i-j} = \frac{\dot{m}}{\sqrt{\frac{p_i^2 - p_j^2}{p_i v_i}}} \dots (2)$$

Probera is currently using Stodola to describe the swallowing capacity of the turbine using the turbine constant as a parameter provided by the user. As a first approximation the turbine constant is calculated from the information provided by the turbine supplier and can then be adjusted accordingly to actual measurements.

Beckmann's [7] findings during the 60's led to an even improved and accurate formula:

$$\dot{m} = \left[C_{T,B} + K_\mu (\mu - \mu_{des})\right](1 + \mu) \sqrt{\frac{p_i}{v_i \mu} F} \dots (3)$$

Where $C_{T,B}$ and K_μ are constants, μ is calculated from

$$\mu = \frac{\int v dP}{u^2} \approx \frac{\Delta h_s}{u^2} \dots (4)$$

And therefore could be named the theoretical, dimensionless enthalpy drop through the stage or turbine. Δh_s [J/kg] is the isentropic enthalpy drop and u [m/s] is the blade speed on the average diameter. It has a somewhat connection to the stage blade loading and is quite similar, but should not be mixed up. The stage loading is written:

$$\Psi = \frac{\Delta h}{\mathrm{u}^2} \dots (5)$$

This dimensionless number, theoretically, is choosed by the designer of the turbine and in turn also decides the reaction of the stage. It can be shown that a lower reaction (or impulse-) turbine has a higher stage loading.

μ is then actually a function of the reaction of the stage and could be optimized accordingly by the designer, this is noted by the μ_{des} in equation (3). The factor left to explain in equation (3) is the factor F, which can be written

$$F = \frac{n}{n+1} \left\{ 1 - \left[1 + \frac{2\lambda}{n-1}\right] \left(\frac{p_i}{p_j}\right)^{-\frac{(n+1)}{n}} - \lambda \left[1 - \frac{n+1}{n-1} \left(\frac{p_i}{p_j}\right)^{-\frac{2}{n}}\right] \right\} \dots (6)$$

Where in turn n is the polytropic coefficient:

$$n = \frac{ln\left(\frac{p_i}{p_j}\right)}{ln\left(\frac{v_j}{vi}\right)} \dots (7)$$

And λ is a correction factor depending on the number of stages in the turbine, which Beckmann [7] introduced as:

Table 1, correction factor lambda as a function of number of stages in the turbine.

Number of stages	$\lambda = f(n_{stg})$
1	0.5 → 1.0
2	0.3 → 0.5
3	0.25 → 0.3
4	0.25
∞	0
Single Impulse	1.0

Magnus Genrup at Energy Sciences,LTH [8] has developed an equation for λ as follows:

$$\lambda = \frac{1}{1+n_{stg}}\left(8.0889 n_{stg}{}^{-0.1240} - 65.5910 n_{stg}{}^{-0.0132} + 58.8398\right) \dots (8)$$

Which is implemented in Probera and used in the analysis in this thesis.

To further improve the modeling of the swallowing capacity in the wet-area of the expansion in the turbine, Cotton [9] introduced yet another correcting factor which takes into account that the swallowing capacity decreases as the moisture content in the steam increases. The swallowing capacity should be corrected by the square root of the quality as:

$$\dot{m} = \left[C_{T,B} + K_\mu(\mu - \mu_{des})\right](1+\mu)\sqrt{\frac{p_i}{v_i \mu} F \sqrt{x_j}} \dots (9)$$

Where the quality at the outlet of the turbine (-section) is used.

Equation (9) is implemented in Probera and used in the analysis.

4.2 Moisture correction of the efficiency

The common method to adjust the turbines efficiency as the moisture content increases is to use the Baumann-rule, which was developed in the 20's. The turbines wet efficiency [10] could be written as

$$\eta_{wet} = \eta_{dry}\left(1 - \alpha\frac{Y_{in} + Y_{out}}{2}\right) \dots (10)$$

Where α is the Baumann-factor, which usually varies between 0.5-1.0. Y is the moisture content. The Baumann-factor is larger for a turbine section than for a single stage, for example could one droplet cause more losses through multiple stages than through just one. The losses caused by the moisture content could be divided into three parts [10]:

- Braking losses, the droplets hits the backside of the rotor, acting as a brake.
- Frictional losses, the friction between droplets and steam, i.e two-phase flow.
- Thermodynamic losses, sub cooling of the steam due to the rate of the expansion causes irreversibility [11].

Baumann describes how the efficiency decreases linearly, which is assumed, as the moisture content increases. However, only the total quantity of moisture is taken into account, not the structure [12].

For example, the last stage in the LP-turbines in Oskarshamn 3 is reinforced to better withstand the impact of the droplets, and the reinforcement is made near the tip where most droplets are present [12].

Probera is using the Baumann rule to take the moisture content's affect on the efficiency into account; however a slightly different methodology is used [13]. Probera calculates the fraction of how much of the expansion is taken place in the wet area (wetfrac) and then calculates the wet efficiency as

$$\eta_{wet} = \eta_{dry}\,(1 - \alpha * (1 - x_{out}) * wetfrac) \ldots (11)$$

For a wet fraction of 0.5 the methods are equal, regardless of the value of the Baumann-factor. However, when the wetfrac is higher (or lower), for example wetfrac=1, the methods differ as can be seen in the following picture:

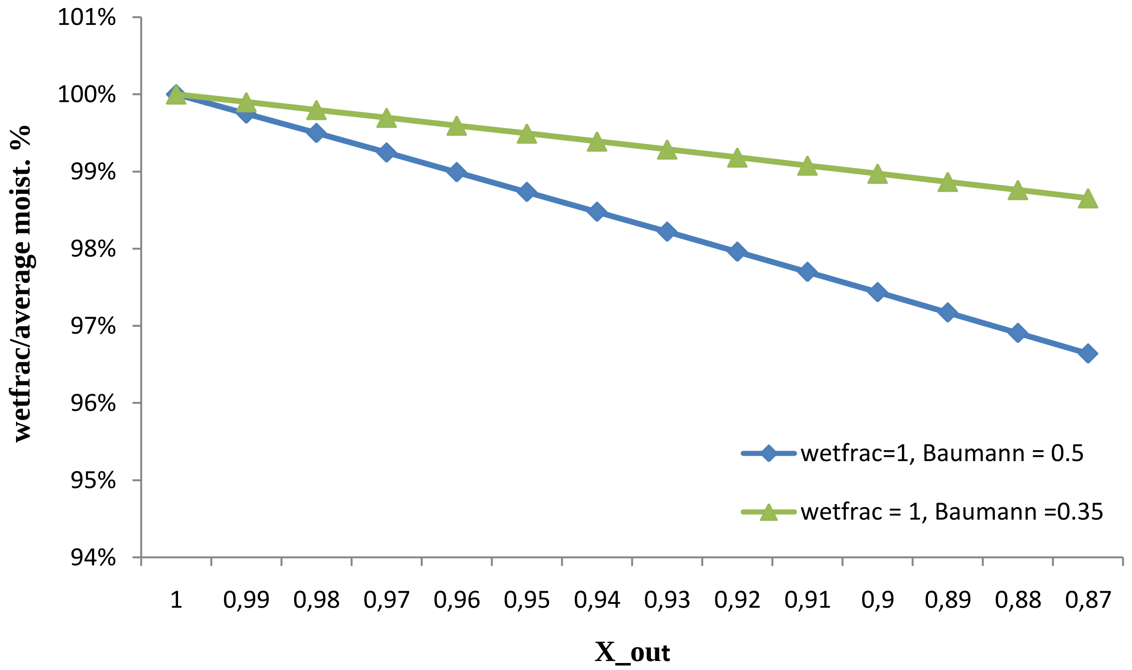

Figure 4 difference (%) between wetfrac- and average moisture methodology as a function of the steam quality at the outlet of the turbine with a wetfraction of 1 and a Baumann factor of 0.5/0.35.

The wetfrac methodology gives a slightly less penalty to the efficiency for a wetfraction of 1 (i.e the entire expansion takes place in the wet area) than the average moisture method. Note that every turbine section expands in the wet region except the first stage of the Low pressure turbine (LP11).

Some care when choosing the Baumann factor is therefore essential for a good result.

4.3 Stagnation properties and exhaust loss

The stagnation enthalpy is the enthalpy that a fluid with negligible potential energy, the steam in this case, would achieve if it was brought to rest with no work or heat transfer involved i.e adiabatically. This can be written [14]:

$$h_0 = h + \frac{c^2}{2} \ldots (12)$$

Where h is the static enthalpy and c is the speed, the term $c^2/2$ is often referred to as dynamic enthalpy. The exhaust loss in a turbine conducts mainly of the dynamic part of the enthalpy. For a multiple stage turbo machine the exhaust loss is only calculated at the last stage, one assumes for multiple stages that the following stage could recover the speed back into static properties or at least make benefit of the speed into useful work. The loss at the final stage could however be significant and must be considered for the last stage since the speed is quite high and also represent an additional load to the condenser. Probera is using a routine to calculate the exit loss using the inner- and outer diameter, the speed of the blade and a mean value of the blade angle. The mass flow to the condenser varies little with a varying cooling water temperature, but the specific volume of the steam and thereby the speed (or volumetric flow for a given area) varies quite a bit giving a non-linear connection between the exhaust loss and cooling water temperature. A typical exhaust loss variation can be seen below in Figure 5:

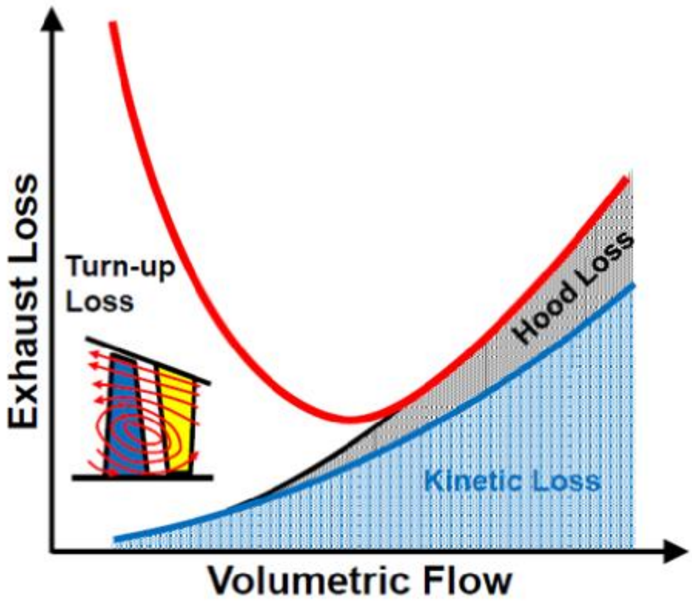

Figure 5, Schematic view over a typical exhaust loss. Courtesy to Magnus Genrup at Energy Sciences, LTH.

5. Results

5.1 Modeling – Heat Balance Differences

Four heat balance models are considered in this section:

- Designed model provided by turbine supplier, not a Probera model.
- Probera Design model, which tries to imitate the Designed model.
- Adjusted model
- Adjusted model with a clean condenser

The Designed model needed to be simulated in Probera since it's not provided from supplier how the plant actually behaves at varying cooling water temperatures but only at 5°C and 18°C. Approximate power output according to turbine supplier at 5°C and 18°C can be seen below in Table 1. A print screen of the entire Probera Design model and Adjusted model can be seen in Appendix 9.3 . For information about factors used in the plant, see Appendix 9.1.

Table 1, Approximate power output provided by turbine supplier.

Approximate power output [MW]	Cooling water temperature [°C]
1465	5
1386	18

In Table 2 differences between the designed plant (were data was gathered from turbine supplier) and the Probera model is presented. The biggest relative difference is an increased flow from extraction 6. The second biggest relative difference is a decrease in flow from extraction 4, thanks to the higher drainage from HPFH's (via flash tank 462 TD1) which in turn is a consequence of the increased flow from extraction 6 and 5. The slightly low condenser flow together with the lower flow from extraction 1 causes the flow from extraction 2 to increase. The pressures simulated are fairly equal, with the exception of extraction 2 and 4 pressures which are somewhat lower than designed. The power output is 2.3 MW lower than designed.

Table 2, Major differences at 5°C in cooling water between Designed plant and the Probera model of the designed plant. Relative error to Designed plant.

	Units	Probera Design	Relative error [%]
Condenser flow	[kg/s]	1137.0	-0.5
Condenser pressure	[kPa]	4.41	-0.2
Extraction 1 flow	[kg/s]	42.2	-7.9
Extraction 1 pressure	[kPa]	12.8	-3.6
Extraction 2 flow	[kg/s]	69.3	6.8
Extraction 2 pressure	[kPa]	62.0	-5.3
Extraction 3 flow	[kg/s]	107.1	0.0
Extraction 3 pressure	[kPa]	241.6	-4.1
MSR moisture flow	[kg/s]	237.7	3.6
Extraction 4 flow	[kg/s]	143.0	-11.2
Extraction 4 pressure	[kPa]	884.1	-6.1
Drainage from flashtank 462 TD1	[kg/s]	583.7	2.7
Extraction 5 flow	[kg/s]	101.5	3.4
Extraction 5 pressure	[kPa]	1774.5	2.0
Extraction 6 flow	[kg/s]	70.8	17.4
Extraction 6 pressure	[kPa]	2486.7	2.5
Bypassvalve (421 VB6)	[kg/s]	19.0	-0.1
Drainage from HPFH's to flashtank 462 TD1	[kg/s]	371.3	3.0
Admission flow to LP Turbines	[kg/s]	1356.0	-0.3
Admission pressure to LP Turbines	[kPa]	905.9	0.6
Admission pressure to HP Turbines	[kPa]	6460.3	0.0
Flow to Reheater	[kg/s]	183.3	-0.8
Total flow from Reactor	[kg/s]	2113.1	-0.1
Power output	[MW]	1462.7	-0.2

Differences between designed model and the adjusted model can be seen in Table 4. To be able to describe the behavior of the plant at the sea water temperature of 5°C, the model adjusted the fouling factor to $4.35044 \cdot 10^{-5}$ in the condenser. A clean condenser is assumed to have a fouling factor of $3.0 \cdot 10^{-5}$ according to Berth Arbman [15]. The adjustment on the turbine constants can be seen in Table 3. The biggest difference is in the HP-turbine were the HP1 and HP3 sections which are 9.0% and 9.3% wider respectively. The HP2 section is however somewhat narrower with a -3.1% decrease in wideness. The first sections in the LP-turbine are slightly narrower whereas the last sections in the LP-turbine are somewhat wider. See Table 3 for values.

Table 3, Change in Turbine capacity between Probera Design model and the Adjusted model. Relative difference to Probera Design.

Turbine Section	Probera Design $C_{T,B}$ [m^2]	Adjusted model $C_{T,B}$ [m^2]	Relative difference [%]
HP1	0.054381	0.059282	9.0%
HP2	0.176100	0.170583	-3.1%
HP3	0.190480	0.208275	9.3%
LP11	0.099469	0.097183	-2.3%
LP12	0.089200	0.086292	-3.3%
LP21	0.268000	0.265074	-1.1%
LP22	1.044810	1.049690	0.5%
LP31	5.315910	5.423210	2.0%
LP32	5.314530	5.574320	4.9%

Table 4, Major differences at 5°C in cooling water for the model adjusted to measurements. Relative error to Designed plant.

	Units	Adjusted	Relative error [%]
Condenser flow	[kg/s]	1149.0	0.6
Condenser pressure	[kPa]	4.71	6.6
Extraction 1 flow	[kg/s]	39.8	-13.2
Extraction 1 pressure	[kPa]	13.1	-2.0
Extraction 2 flow	[kg/s]	69.9	7.7
Extraction 2 pressure	[kPa]	62.0	-5.3
Extraction 3 flow	[kg/s]	110.4	3.0
Extraction 3 pressure	[kPa]	247.4	-1.8
MSR moisture flow	[kg/s]	228.1	-0.5
Extraction 4 flow	[kg/s]	150.9	-6.3
Extraction 4 pressure	[kPa]	913.2	-3.0
Drainage from flashtank 462 TD1	[kg/s]	562.5	-1.0
Extraction 5 flow	[kg/s]	86.0	-12.4
Extraction 5 pressure	[kPa]	1686.6	-3.0
Extraction 6 flow	[kg/s]	77.1	27.9
Extraction 6 pressure	[kPa]	2453.4	1.1
Bypassvalve (421 VB6)	[kg/s]	13.7	-28.2
Drainage from HPFH's to flashtank 462 TD1	[kg/s]	356.0	-1.2
Admission flow to LP Turbines	[kg/s]	1368.0	0.6
Admission pressure to LP Turbines	[kPa]	936.1	4.0
Admission pressure to HP Turbines	[kPa]	5989.3	-7.3
Flow to Reheater	[kg/s]	182.6	-1.1
Total flow from Reactor	[kg/s]	2108.9	-0.3
Power output	[MW]	1444.9	-1.4

On the assumption that the condenser in fact is clean, or has a fouling factor of $3.0 \cdot 10^{-5}$ instead, the change in flows, pressures and power output can be seen in Table 5. In comparison with the Adjusted model, the condenser pressure is lower and the flow is somewhat lower which gives a higher flow from extraction 1. Note that the extraction 1 pressure only changes a little bit from Adjusted model- the backpressure only has a minor impact on the flow through the stage, see equation (1). It is the wideness of the last stage of the turbine that governs what happens in the plant. The other results are fairly the same as for the Adjusted model, see Table 5.

Table 5, Major differences at 5°C in cooling water for the adjusted model with the assumption of a clean condenser. Relative error to Designed plant.

	Units	Clean Condenser	Relative error [%]
Condenser flow	[kg/s]	1146.4	0.4
Condenser pressure	[kPa]	4.45	0.6
Extraction 1 flow	[kg/s]	41.2	-10.0
Extraction 1 pressure	[kPa]	12.9	-2.9
Extraction 2 flow	[kg/s]	70.1	8.1
Extraction 2 pressure	[kPa]	61.9	-5.5
Extraction 3 flow	[kg/s]	110.6	3.2
Extraction 3 pressure	[kPa]	247.2	-1.9
MSR moisture flow	[kg/s]	228.1	-0.5
Extraction 4 flow	[kg/s]	151.0	-6.3
Extraction 4 pressure	[kPa]	913.0	-3.0
Drainage from flashtank 462 TD1	[kg/s]	563.5	-0.9
Extraction 5 flow	[kg/s]	86.0	-12.4
Extraction 5 pressure	[kPa]	1686.6	-3.0
Extraction 6 flow	[kg/s]	77.1	27.9
Extraction 6 pressure	[kPa]	2453.3	1.1
Bypassvalve (421 VB6)	[kg/s]	13.7	-28.2
Drainage from HPFH's to flashtank 462 TD1	[kg/s]	356.0	-1.2
Admission flow to LP Turbines	[kg/s]	1368.0	0.6
Admission pressure to LP Turbines	[kPa]	936.0	4.0
Admission pressure to HP Turbines	[kPa]	5989.3	-7.3
Flow to Reheater	[kg/s]	182.6	-1.1
Total flow from Reactor	[kg/s]	2108.3	-0.3
Power output	[MW]	1448.4	-1.1

5.2 Power Output as a function of cooling water temperature

A comparison between the Probera Design model, aiming to describe the designed production curve, the adjusted model with a clean condenser (fouling factor = $3.0*10^{-5}$) and the actual measurements (approximated with a 2^{nd} order polynomial) can be seen below Figure 6. For further information about the RTO, Power output and condenser pressure-measurements at full load, see Appendix 9.4, 9.5.

The difference between the adjusted curve with a clean condenser and the designed curve is -14.3 MW at 5°C. The difference is somewhat lower at higher CW-temperatures. At higher CW-temperatures the Adjusted model and measured data is coinciding, but diversifies as the temperature drops, only to converge again from about 4°C and lower temperatures.

At 18°C the power output is 1385.5 MW for the Probera Design curve. The Adjusted model has a power output of 1370.4 MW which gives a difference of -15.1 MW.

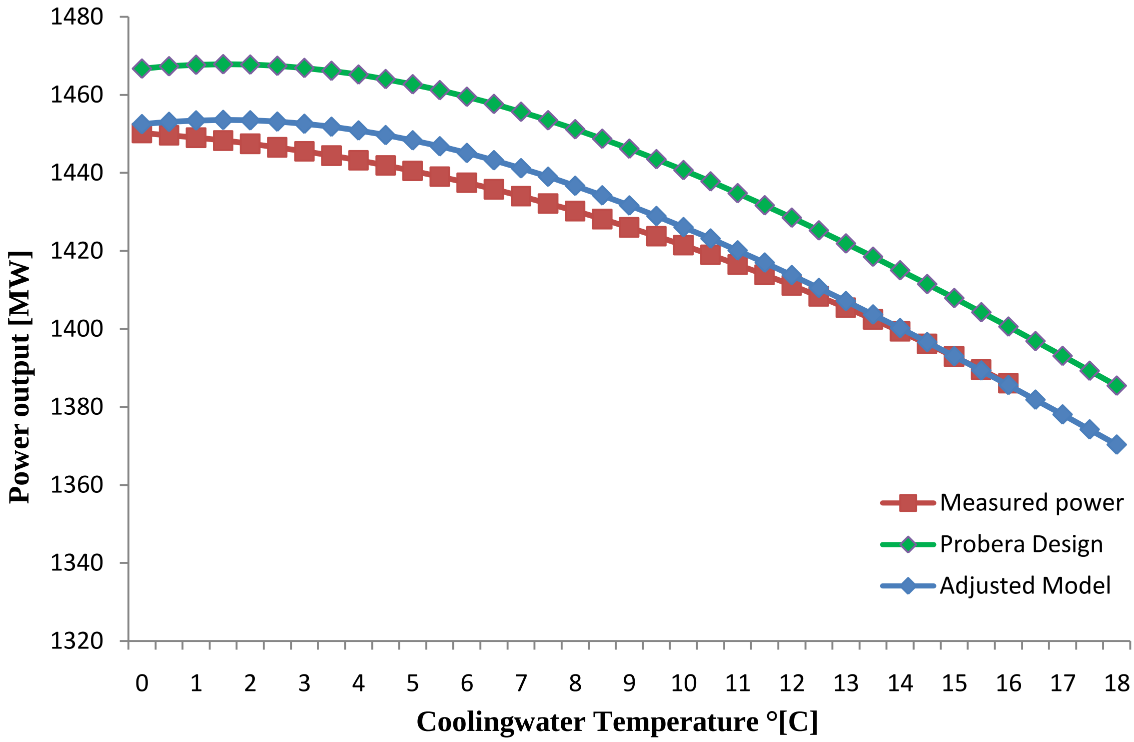

Figure 6, Power output as a function of cooling water temperature. Comparison between Probera Design model, Adjusted model and measured power approximated with a 2nd order polynomial fit.

5.3 Condenser pressure as a function of cooling water temperature

The condenser pressure as a function of cooling water temperature can be seen below in Figure 7 were both the modeled curve as well as the measured curve is presented. At higher temperatures the Adjusted model and measured curve coincides but diverge at lower temperatures, starting at about 15°C.

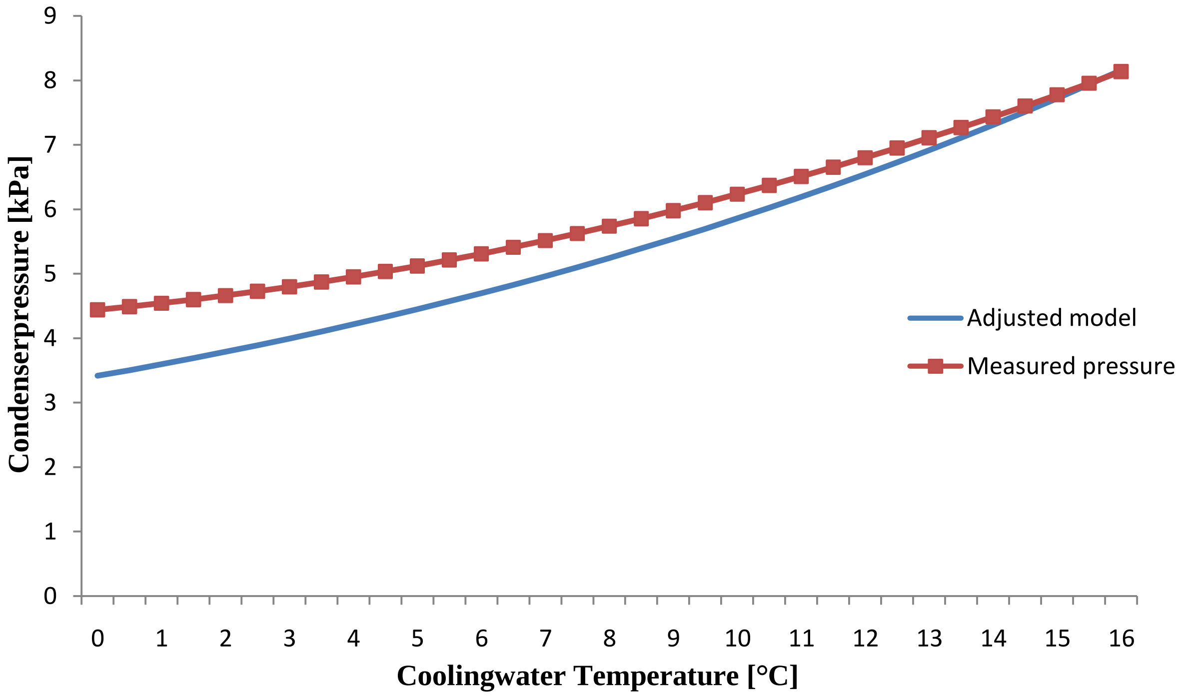

Figure 7, Condenser pressure as a function of cooling water temperature. Comparison between Adjusted model with clean condenser and measured pressure approximated with a 2nd order polynomial fit.

5.4 Modeling of blocked tubes in the Condenser

In Figure 8 it is shown how the power output varies with cooling water temperature as a constant blockage (%) of some tubes are present in the condenser, decreasing available heat transfer area.

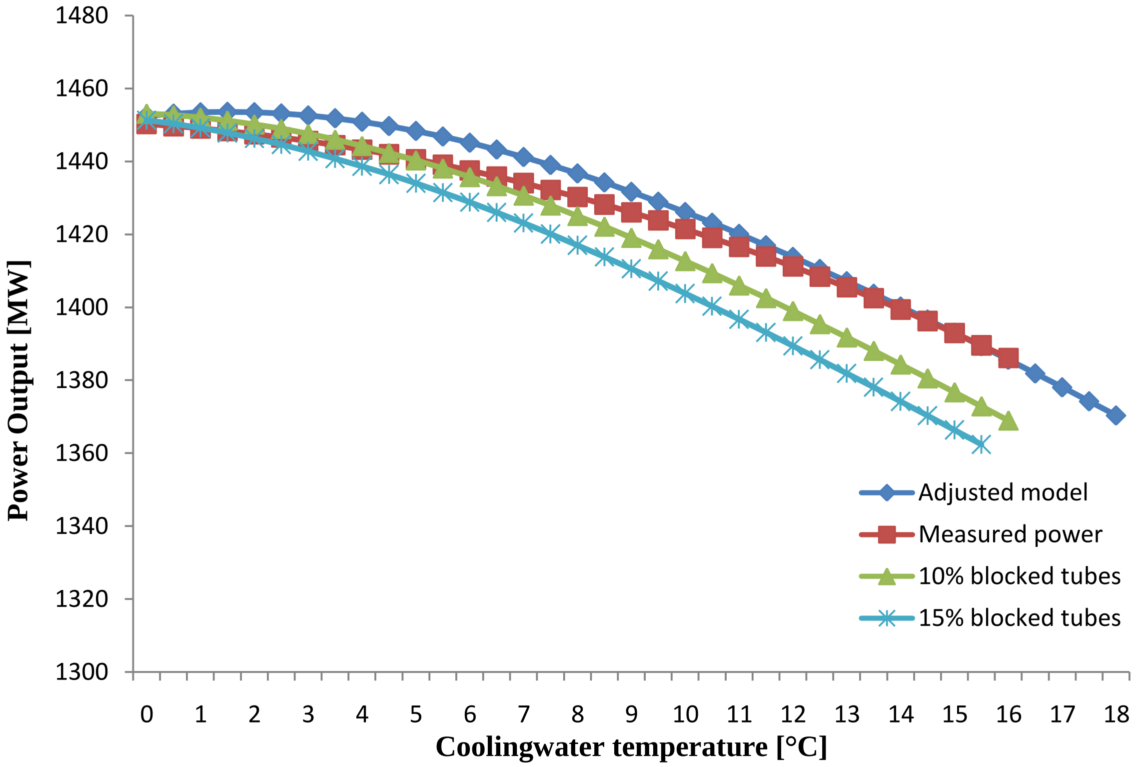

Figure 8, Power output as a function of cooling water temperature. Comparison between Adjusted model with a clean condenser, measured power output approximated with a 2nd order polynomial fit and adjusted model having a constant share of blocked tubes present.

If points from the curves with constant blocked tubes shown in Figure 8 are kept were the curve matches with the measured curve, one could derive a sort of variable blocking of the tubes in the condenser as a function of the cooling water temperature explaining the difference in power output. This can be seen in Figure 9 were the points have been placed together with a percentage of blocked tubes that matches the measurements. A greater percentage of tubes are blocked as the temperature of the cooling water decreases.

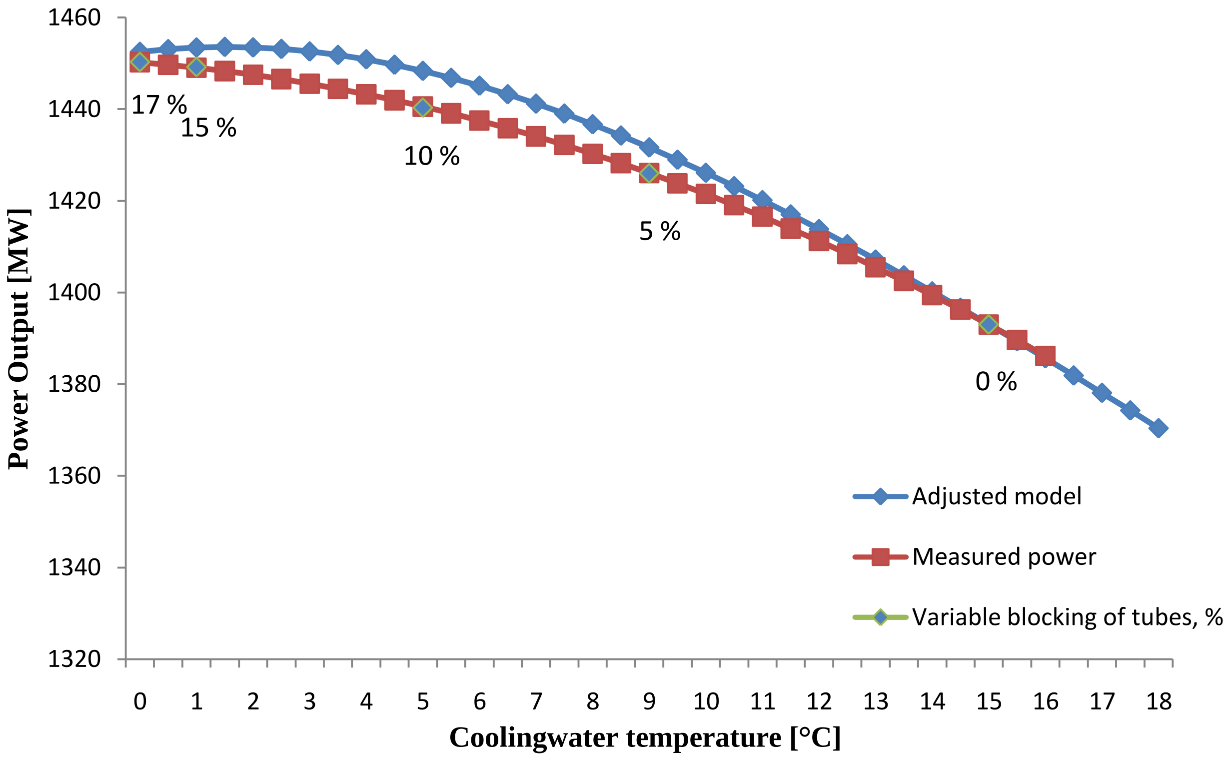

Figure 9, Power output as a function of cooling water temperature. Comparison between Adjusted model with clean condenser, measured power approximated with a 2nd order polynomial fit and Adjusted model with variable blocking of tubes.

5.5 Part load

In the summer of 2012 the plant had a RTO of about 105% during the 27th of July to the 11th of August. During this particular period the temperature varied between about 6.5°C to 11.5°C so the measured curves in Figure 10, Figure 11 are only presented in this span since they aren't valid for other temperature ranges. For more extensive values on the RTO, Power output and condenser pressure, see Appendix 9.4.

The power output as a function of cooling water temperature can be seen below in Figure 10. The difference between modeled and measured power output follows about the same behavior as the full load case with a lower power output than expected. It also seems that a variable blockage of tubes in the condenser matches the modeled curve to the measured one as it is in the case of full load.

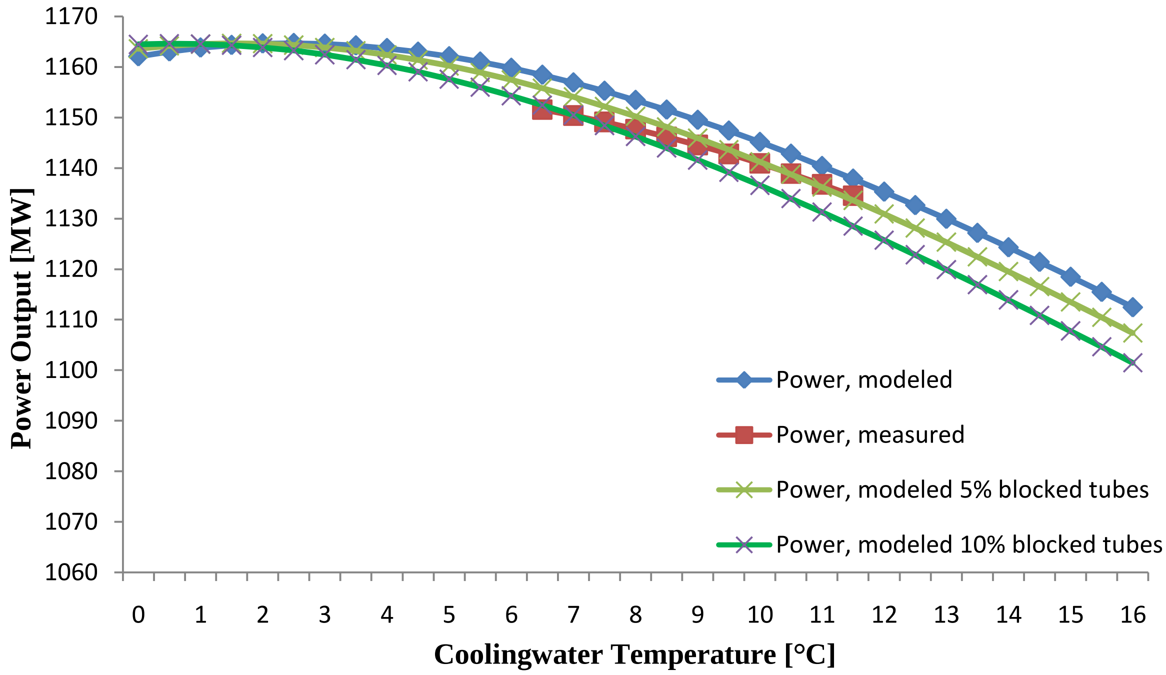

Figure 10, Power output as a function of cooling water temperature at 105.01% RTO. Comparison between modeled curve, measured curve approximated with a 2nd order polynomial fit and modeled curve with a constant blockage of tubes present.

The modeled condenser pressure and the measured condenser pressure are compared in Figure 11 were also the pressure at 129% RTO is presented. As seen in the figure, the condenser pressure is slightly high even at part-load conditions, although the difference to the modeled curve seems to be less. The part-load curve seems to also have about the same behavior with a diverging feature with falling temperature as the full load curve.

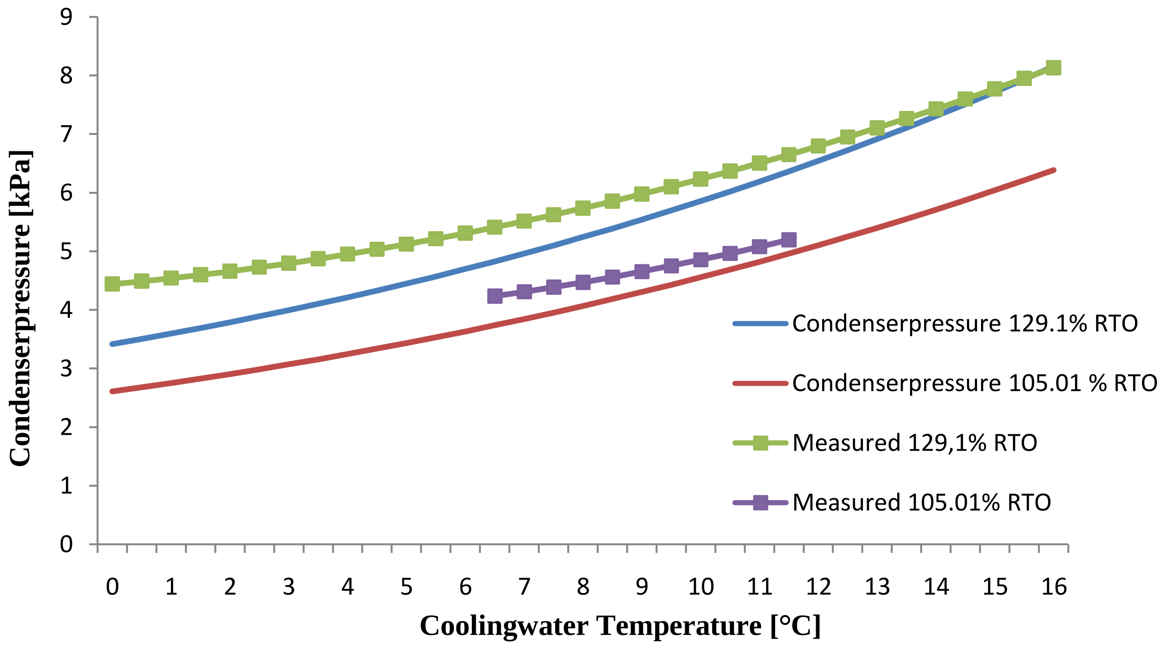

Figure 11, Modeled and measured condenser pressure as a function of cooling water temperature at full load (129.1 % RTO) and part load (105.01% RTO). The modeled curves have the assumption of a clean condenser.

5.6 Deviation Analysis

As stated previously, the two main reasons for the drop in performance of the plant and loss in power output is the wide HP-turbine and a higher than expected condenser pressure. The wide HP-turbine is a question for the turbine supplier, there isn't much to analysis further about that deviation. The high condenser pressure however could be explained by a number of things and for reasoning, the following equation which describes the heat transfer in a heat exchanger could be used [16]:

$$Q = U * A * \Delta T_{lmtd} \quad \dots (13)$$

Where Q is heat flux [W], U is overall heat transfer coefficient [W/ (m^{2}K)], A heat transfer area [m^2] and ΔT_{lmtd} [K] is the logarithmic mean temperature difference used for heat exchangers. A change in these three factors therefore contributes to a change in transferred heat and this lead to the following explanations regarding the high condenser pressure:

- Too small heat transfer area, simply not enough tubes.
- Too low overall heat transfer coefficient, caused by for example:
  - Dirty condenser, air in-leakage or a higher flow of radiological gases than expected.
  - Poor extraction of said gases could in turn be caused by for example a design flaw of ejectors or an off-design operation of the cooling chain of the ejector (inter-) condensers. Also an off-design operation of motive steam in the ejectors such as pressure, temperature, quality or flow leads to a poor extraction as well.
  - Lower heat transfer coefficient/heat conduction for the materials than expected. (Roughly a fifth of the tubes were changed to a steel alloy instead of titanium in PULS)

- Choked or disturbed flow pattern on shell side- some tubes doesn’t transfer enough heat.
- Air blockage on the inside of the tubes, released from the cooling water and/or air in-leakage.

Degasification - Ejector System

The ejector system is a really interesting system. Accordingly to the model, the affect on the power output if the mass flow of ejector steam increases with 1 kg/s is approximately -1MW. However, a flow different from the designed one could cause a worse extraction of radiological gases which in turn could block the heat transfer of the tubes in the condenser causing a higher condenser pressure and a large power loss. The reason is that the non-condensable (radiological) gases is present at the surface of the tubes, like a thin film, which forces the steam to first diffuse through this film thus reducing the heat transfer coefficient and increasing the thermal resistance [17].

Choked tubes or disturbed flow pattern

Since the power uprate not only ment a higher reactor thermal output but a higher load for the condenser, questions have been raised if the tubes somehow is blocked by the increased flow both on shell side and tube side. In Figure 12 the modeled exhaust loss is presented which in the model only consists of the kinetic loss, not hood loss, see Figure 5 in Chapter 4. As can be seen in Figure 13 the specific volume of the steam is quite high at lower CW-temperatures, i.e. lower condenser pressure. Even though the steam isn’t saturated at the outlet of the LP-turbines there should still be a significant increase in specific volume as the cooling water temperature decreases.

These effects combined, a high kinetic loss and high specific volume of the steam, together with the higher mass flow than before PULS, increases the load on the condenser considerably and makes it more difficult to condense the steam back into water.

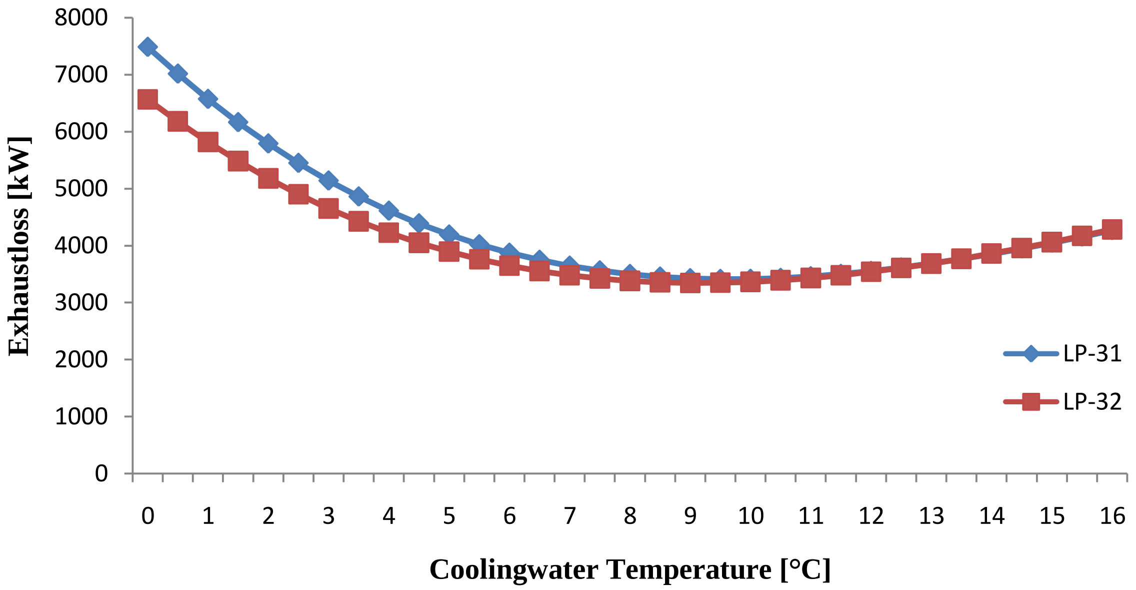

Figure 12, Exhaust loss for the model adjusted to measurements. Both LP-31 and LP32.

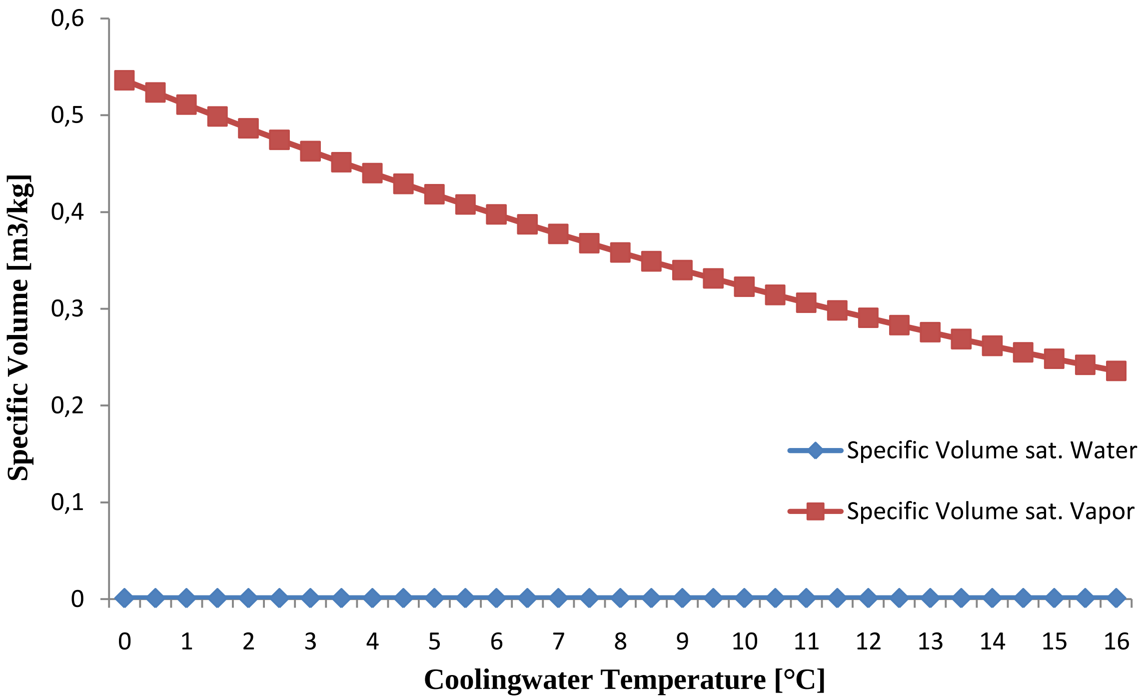

Figure 13, Specific Volume of saturated water and vapor as a function of cooling water temperature.

Siphon Effect

The cooling water is pumped, or rather sucked, through the condenser with the use of the siphon effect. In Figure 14, a schematic side view of the condenser is presented showing the water boxes, inlet to air pumps as well as the hot well.

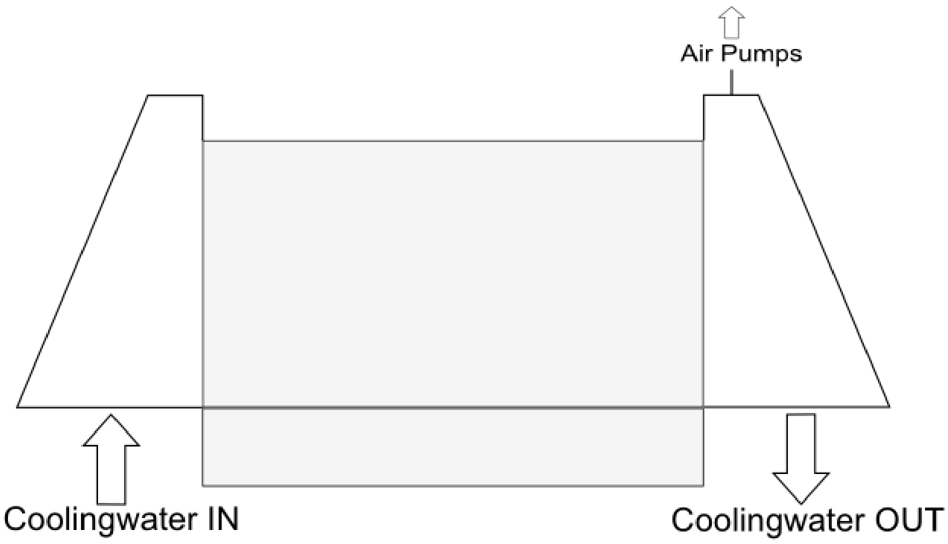

Figure 14, Schematic side view of the condenser.

The cooling water pumps provide a high flow rate but have a rather small pressure ratio. To be able to get the water through the tubes the gravity is used via the siphon effect. The lower pressure is achieved with air pumps which sucks the air out from the outlet water box. If the amount of extracted air isn't sufficient the water level in the condenser is too low and therefore the upper tubes won't have water on the inside. Some air constantly needs to be withdrawn, otherwise the pressure would increase and the level be lowered.

Air on the inside of the tubes

Following the previous discussion about the removal of air on the tube side, some air constantly needs to be ejected from the tube side in the condenser. Could the air then block the tubes even before it enters the outlet chamber? For calculations see Appendix 9.2.

The solubility of air in water is higher with a lower temperature and a higher pressure, according to Henrys law [18]. When the cooling water is passing through the condenser, the temperature increases and the pressure decreases and this leads to a separation of air from the water. This gives the following flow of air as the CW inlet temperature varies:

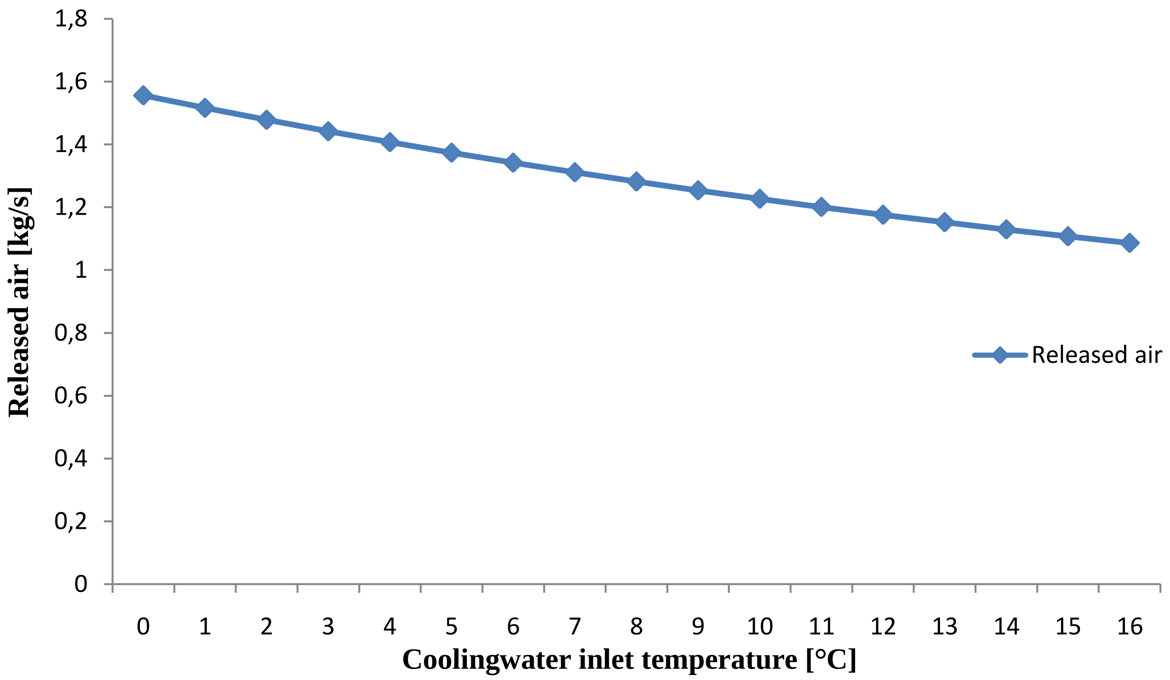

Figure 15, Released air on tube side of the condenser as a function of inlet cooling water temperature.

The released air in the tubes could in turn cause a two-phase flow in the tubes as seen in Figure 16.

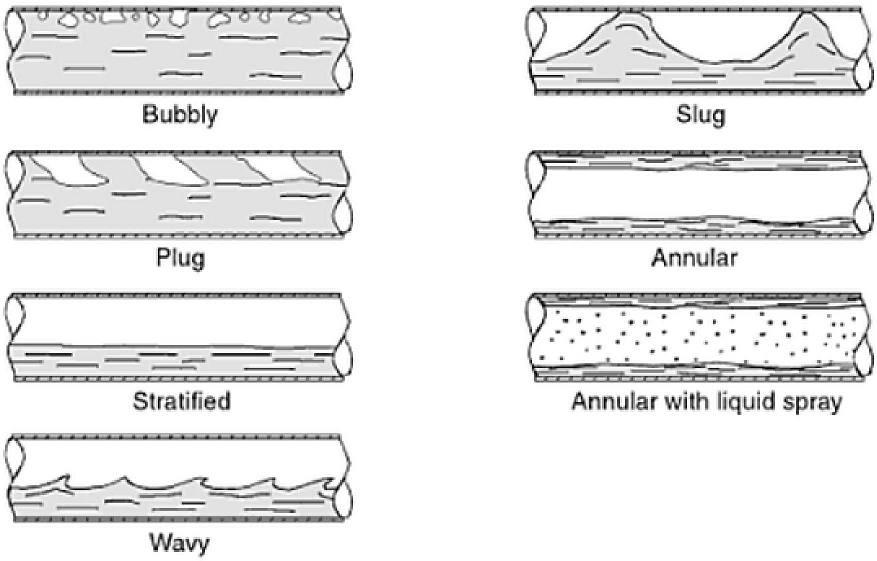

Figure 16, Different types of flow regimes for two-phase flow of gas and liquid. [19]

The type of flow can be determined with a so called Baker plot, shown in [20]. The type of flow regime can be calculated with parameters $G_f \cdot \psi$ and G_g/λ, for values see Table 6.

Table 6, values on $G_f \cdot \psi$ and G_g/λ for CW-temperatures of 0 and 16°C.

$Gf \cdot \psi$ [kg/(m²s)]	Gg/λ [kg/(m²s)]	Cooling water temperature [°C]
0.07	2484	0
0.05	2484	16

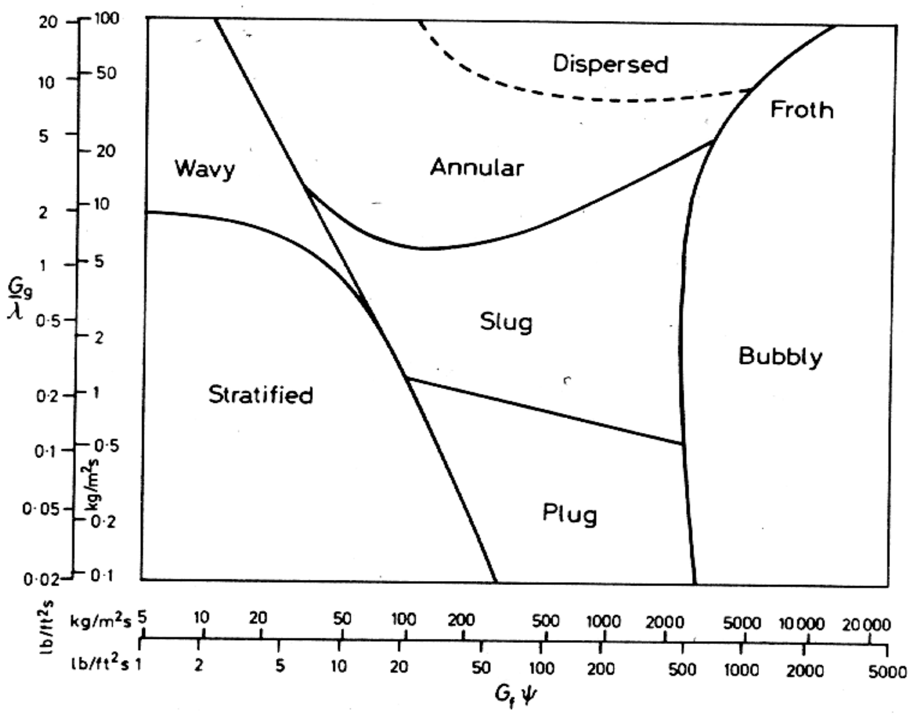

Figure 17, Baker-plot for two-phase flow regimes. [21]

Giving the indication that plugs could be present in the tubes, at least near the outlet of the condenser tube bundles.

6. Discussion

6.1 Discussion of Methodology

As an alternative, the adjustment calculation could be performed at a higher cooling water temperature since the results indicates that the measurements and model coincide. The plant is behaving more as expected and therefore the adjustment on turbine wideness would be somewhat different. It could however be more insecure since the measurements aren’t as extensive as they are with lower temperatures. Another alternative could have been to adjust the dry efficiencies instead of the wideness of the turbines and a somewhat different result would have emerged. Instead of using the BPV as parameter, perhaps fouling of the HPFH’s could have been used as well. A plausible argument is that all these factors somehow should be adjusted in order to fit the measurements even better. The measurements is however the next problem, one tries to adjust the model according to these, but they have in themselves an uncertainty. They are also in general located in the extractions close to the FH’s but the model used (and the one provided from turbine supplier) uses data in the extractions just after the turbine section which should give a difference.

The table calculation could have been done with shorter interval giving a smoother production curve, the calculating time has to be put against the increased accuracy, but the curve seems rather smooth and a decreased interval would probably not give that big a change.

6.2 Discussion of the Results

Probera Design

The biggest difference between the designed plant and the Probera Design model is the flow from extraction 6 which is higher in Probera. This indicates that the wideness of the HP2 should be higher to let more steam through; this would however lower the pressure in extraction 6, which to some extent could be good since the pressure in extraction 6 is slightly high, but it can’t be that wider since the pressure difference isn’t that big. Since the flow to the reheater and the flow through the Bypass valve is about the same as designed this means that the flow from extraction 4 is much lower than designed. To sum up: the feed water is heated about the same in the Probera Design as in the Design from turbine supplier, but more high-quality steam is used instead, causing a slight loss in power output. The efficiencies of at least some of the turbine sections are probably higher according to turbine supplier than assumed in the Probera model explaining the remaining difference in power output.

Adjusted Model

To fit the data accordingly to measurements, the adjustment tool had to increase the wideness of the HP1 and HP3 turbine sections together with an increased condenser pressure. These two changes contributes far more to the drop in power output compared to the other changes which are relatively low, or at least contributes much less. The wide HP turbine makes the bypass valve shut more, since the pressure in the reactor still needs to be constant. This means that a higher pressure drop occurs over the control valves and a loss in power output. The high condenser pressure affects mainly the last stage of the LP-turbine causing a lesser flow from extraction 1. On the assumption that the condenser is clean, the only factors that shows a significant change is the flow from extraction 1 proving the statement above about the condensers affect on the LP-turbine. The higher pressure also means that the expansion is reduced thus reducing the power output.

Power output and Condenser pressure at varying cooling water temperature

First, the Probera design curve needs to be discussed; it indicates a slightly less power output than provided by the turbine supplier at 5°C but seems to explain the behavior in a good way. It has almost the same power output as designed at 18°C with a difference less than 0.5 MW. It seems reasonable that the production curve has a maximum at somewhat lower temperature than designed (5°C), but it is not reasonable that the power keeps increasing when the condenser pressure/temperature drops. Somewhere the exhaust loss should start to be much more significant, as indicated in the section about choked flow. From a suppliers point of view you wouldn't want to sell a plant which has its maximum performance outside the normal operating area.

The difference between the Probera design and Adjusted model is 14.5 MW at 5°C which mainly is caused by the wide HP-turbine. The behavior over the temperature span is about the same, but in comparison to the measured curve it differs quite a lot; it doesn't really add up with the previous discussion about the performance. It's not until a variable blocking of the tubes in the condenser is implemented as the modeling matches the measurements in a good way. As the cooling water temperature decreases more and more of the tubes are blocked causing a higher condenser pressure than expected and a lower power output. But at about 2-3°C the production curve drops due to the exhaust losses, which actually makes the power output at 0°C almost the same with 17% blocked tubes as with 0%! The loss in power output varies, but at 5°C it is about 8 MW. If the condenser is replaced in the future, one has to take into account this variation and make a study on which temperature span is the most common one the plants operates in to make a legitimate economical calculation.

Part load

The difference in power output at part load is troubling; there should be enough tubes or heat transfer area to condensate the steam at a lower pressure. This doesn't however seem to be the case and as with the full load, the analysis shows that the condenser pressure is higher than expected. In absolute numbers the pressure is lower than at full load obviously, but it should still be lower given the lower load. As with the full load case, it looks like the loss in power output could be explained by a variable blocking of tubes in the condenser. One should however be careful with extrapolating the results to other temperature ranges, as indicated in the section about condenser pressure at part load. The polynomial fit doesn't seem perfect and is probably not valid at other temperature ranges.

Deviation Analysis

Quite a few reasons to the higher condenser pressure have been presented in this section, some perhaps more plausible than others. Trouble with the ejector system felt initially as the most plausible one, a higher condenser pressure increases the suction capacity of the ejector and hence more gases are ejected. This in turn gives a decrease in the thermal resistance and enables a better heat transfer and condensing which lowers the condenser pressure. Some sort of equilibrium should then be present and therefore the deviation at higher cooling water temperatures is none. Another reason could be an off-design operation of the motive steam. The discussion is however somewhat more troublesome when one takes into account the higher than expected condenser pressure at part load since it is a lesser amount of gases that needs to be ejected. It could however be a new equilibrium established at a lower pressure; the ejectors are once again unable to eject enough gases due to an off-design operation, lack in design or higher amount of gases than expected.

Choked tubes are a quite plausible reason for the condenser pressure when one takes into account the behavior at part load. The lower pressure than the one at full load increases the specific volume of the steam and even though the exhaust loss could be lower due to a lesser mass flow, the increase in specific volume counteracts the mass flow rate and gives a volume flow rate about the same and therefore the exhaust loss is about the same as well. To refer to the previous discussion about the ejector system, the flow pattern could have changed in a way that air pockets that before were ejected have moved away to a different location in the condenser making the degasification process much more difficult.

A lower water level than expected in the condenser could explain the behavior both at full load and at part load, since it is a problem on the tube side with fairly the same conditions. But why should the blocking of the tubes be reduced when the cooling water temperature rises? A warmer cooling water contains less air due to Henry's law and since the temperature rise is less at part load the amount of air released from the cooling water is lower and in turn less air needs to be extracted. The pressure variation would however be seen in the pressure measurement, although measurements in turn could be uncertain and it is also performed after a water separation tank that perhaps could affect the measurement? A measurement of the temperature at the inlet and outlet of the condenser could give an indication if the tubes are in operation or not. One should also take into account that it is the upper tubes that stand for a greater part of the condensation process; therefore a blockage here would be worse.

Air on the inside of the tubes could in itself act as a blockage if enough air is released as stated in the result section. One needs to bear in mind that the calculations are performed with the assumption of equilibrium which very well might not be the case. A lot of assumptions are made and the result shouldn't be regarded as an absolute truth but more of an indication. The release of air is non-linear and increases with decreasing CW inlet temperature and it seems to be indicated that plugs occur near the outlet of the condenser.

The plausible explanations to the higher than expected condenser pressure could stand both for themselves, but they are even more plausible together! Some choking or disturbed flow pattern of the condenser for example should give a worse extraction of gases which starts a build-up of gases as a film on the tubes increasing the thermal resistance. On the tube side the release of air could perhaps be higher giving a two-phase flow with an increase in thermal resistance and together with a water level for example caused by a lack of suction capacity in the air pumps, a variable blockage of the tubes as seen in the results seems reasonable. There is however a fair chance that other phenomenon could explain the behavior of the condenser with a variable blocking of the tubes.

6.3 General Discussion

The purpose with this thesis was to identify differences in the achieved power output after the power uprate in comparison to what was designed before the project. In this thesis two major differences have been found, a wide HP-turbine and higher condenser pressure than expected. The aim with the thesis is therefore considered to be achieved and the purpose fulfilled.

7. Conclusions and Recommendations

The high-pressure turbine capacity is too high which gives a loss in power output in the region of about 9-14 MW.

The condenser pressure is higher than expected, which gives a variable loss in power output, but at 5°C in CW-temperature the loss is about 8 MW.

The variable deviation in the condenser pressure could be explained by a variable blocking of the tubes in the condenser as the CW-temperature drops.

To rule out some of the suggested causes to the higher condenserpressure, the following is suggested:

- Measuring of the ejector inter-condensers, at least temperature but preferably also differential pressure to rule out or confim off-design operation.
- Temperature measurements on strategically tubes in the condenser, both tubes at the top and tubes at the center at least in the outlet. This could rule out or confirm both problems with water level in the waterboxes as well as choking of the tubes.
- If the condenser is replaced, an analysis on which temperature span is the most common is very important to be able to do an economical calculation. As stated the deviation varies with the CW-temperature.

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9. Appendix

9.1 Calculation of turbine capacity and other factors

To calculate the designed values of the turbine constant $C_{T,B}$, equation (9) is used. Therefore, the following must be known:

- Mass flow rate
- Assumed or calculated values of K_μ, μ_{des} and μ.
- Inlet condition (pressure and enthalpy in this case)
- Outlet condition
- Number of stages in the turbine section.

With the use of XSteam, an add-on to Matlab, which uses IAPWS IF-97 standard formulation for thermo hydraulic data on steam and water, specific volume and other conditions could easily be calculated. After use of equations (4, 6-9) The initial, designed, values on constant $C_{T,B}$ were calculated. Due to the fact that no disclosure to third-parties is allowed from the supplier, the exact data used in the calculations aren't presented. However, the turbine constants were calculated as:

Table 7, initial values of the turbine capacity, calculated from the designed values achieved from turbine manufacturer.

Turbine Section	Swallowing Capacity $C_{T,B}$ [m^2]
HP1	0,05448
HP2	0,16570
HP3	0,17948
LP11	0,09887
LP12	0,09041
LP21	0,27016
LP22	1,02481
LP31	5,31591
LP32	5,31453

Due to the unsymmetrical extractions in the LP turbine, some assumptions have been made regarding the mass flow rate, since it wasn't a known value. These values are also just the initial, approximately designed ones. Some adjustments need to be made in order to match the measured data to the model in a good way. Below is a table showing the factors used in the heat balance model, Baumann, η_{dry}, $C_{T,B}$, K_μ and μ_{des} are applied values whereas $\mu$and η_{wet} are calculated values . Note that the Baumann factor doesn't affect the LP11 section since it is not operating in the wet area.

Table 8, factors used in Designed plant.

Turbine Section	Baumann	η_{dry} [%]	η_{wet} [%]	$C_{T,B}$ [m^2]	K_μ	μ	μ_{des}
HP1	0.37	95	91.4	0.054381	1	1	1
HP2	0.32	93.25	89.6	0.176100	1	1	1
HP3	0.33	92	87.3	0.190480	1	1	1
LP11	0.35	92	92	0.099469	1	1	1
LP12	0.36	92	91.4	0.089200	1	1	1
LP21	0.34	90.5	87.5	0.268000	1	1	1
LP22	0.32	90.5	87.4	1.044810	1	1	1
LP31	0.32	85.5	82	5.315910	1.02	1.0664	1
LP32	0.31	85.5	82.1	5.314530	1.02	1.0424	1

Table 9, factors used in Modeled plant.

Turbine Section	Baumann	η_{dry} [%]	η_{wet} [%]	$C_{T,B}$ [m^2]	K_μ	μ	μ_{des}
HP1	0.37	95	91.6	0.059282	1	1	1
HP2	0.32	93.25	89.6	0.170583	1	1	1
HP3	0.33	92	87.5	0.208275	1	1	1
LP11	0.35	92	92	0.097183	1	1	1
LP12	0.36	92	91.3	0.086292	1	1	1
LP21	0.34	90.5	87.3	0.265074	1	1	1
LP22	0.32	90.5	87.3	1.049690	1	1	1
LP31	0.32	85.5	82	5.423210	1.02	1.04966	1
LP32	0.31	85.5	82	5.574320	1.02	0.99831	1

Below in Table 10 is measured temperatures for the extractions and coolingwater used to adjust the model to measurements. To be able to make comparisons the data had to be taken when the temperature in the coolingwater was 5°C, the chosed time was 17/1 -13 at 16:51 and a 10 minute average was used.

Table 10, Measured temperatures from measurementpoints on the 17/1 2013.

	Measurementpoint	Temperature [°C]
inlet coolingwater	112KB502	4.999
outlet coolingwater	112KB503	15.375
Extraction 1	423KA506	51.223
Extraction 2	423KB509	86.739
Extraction 3	423KB512	127.265
inlet MSR (extraction 4)	423KB503	178.592
Extraction 5	423KB502	202.757
Extraction 6	423KB501	221.607

9.2 Calculation of the solubility of air in water

Assumption: air with a composition of 79% nitrogen and 21% oxygen and with a molar weight of 32 g/mol (O_2) and 28 g/mol (N_2) respectively. With a use of an add-on to Probera which decides the Henry's constant as a function of molar weight and temperature, the mole fractions could be decided according to Henry's law [18].

$$y_{i,liquid\ \ side} = \frac{P_{i,gas\ side}}{H} \dots (14)$$

Were y_i is mole fraction, H Henry's constant and P_i is partial pressure (y_i=P_i/P for ideal-gas mixtures) The inlet pressure is assumed to be 1 atm for the air which gives a partial pressure of 0.21 for the oxygen and 0.79 for the nitrogen which is used in the equation. Below is a table of the constants used in equation (14) above.

Table 11, Henry's constant for oxygen and Nitrogen at a total pressure of 1 bar as a function of inlet CW-temperature.

Inlet CW-Temperature [°C]	Henry O2 [bar·kg/mg]	Henry N2 [bar·kg/mg]
0	0.014557	0.034841
1	0.014943	0.035665
2	0.015331	0.036491
3	0.015722	0.037319
4	0.016115	0.038149
5	0.016511	0.038979
6	0.016909	0.03981
7	0.017308	0.04064
8	0.017709	0.04147
9	0.018111	0.042299
10	0.018514	0.043126
11	0.018919	0.043951
12	0.019323	0.044773
13	0.019729	0.045593
14	0.020134	0.046409
15	0.02054	0.047221
16	0.020945	0.048029

The outlet pressure (pressure at the outlet water box) is assumed to be about 0.3 bar; Measurements indicate a pressure of about 0.23 bar but the pressure is measured after a water separation tank so 0.3 bar should be a conservative assumption. The amount of air released could now be calculated with the assumption of phase equilibrium and ideal-gas behavior. The cooling water is heated about 10.5°C regardless of the CW inlet temperature since it is the same amount of heat flux that needs to be cooled. A cooler water have a higher solubility however, as seen in the table with a decreasing Henry's constant as the temperature increases, which gives a non-linear relationship of the amount of air released. The result can be seen in Figure 15.

For the Baker-plot, some parameters needed to be calculated:

$$G_g = \frac{\dot{m}_g}{A_{section}} \dots (15)$$

$$G_f = \frac{\dot{m}_f}{A_{secti\,on}} \dots (16)$$

Were the mass flow for the gas (steam, see Figure 15) and for the fluid (water, approximately 55000 kg/s) is used. Using the area for one tube and making the assumption of same flow in all tubes this leads to values on G_g/λ and $Gf \cdot \psi$ as seen in Table 6. Below is the equations for λ and ψ [15], which both becomes equal to one since air and water is used.

$$\lambda = \left(\frac{\rho_g}{\rho_{air}} \frac{\rho_f}{\rho_{H_2O}}\right)^{\frac{1}{2}} \dots (17)$$

$$\psi = \frac{\sigma_{H_2O}}{\sigma}\left(\frac{\mu_f}{\mu_{H_2O}}\left(\frac{\rho_{H_2O}}{\rho_f}\right)^2\right)^{\frac{1}{3}} \dots (18)$$

9.3 Heat balance models

Figure 18 and Figure 19 below are print screens of the modeled plant in Probera.

Note that the factor “avvikelse” in the downright corner is a measure of the error from the actual measurements. For all measurement points one could supply a non-correlated factor which describes how much one could “trust” a measurement. The better the measurement method – the lower the factor and the trustworthiness of the measurement is higher. Probera takes all these factors into account when calculating the “avvikelse”-factor using a least-square method. According to Bertil Persson [22] a value less than 2 is very good.

Modeling of Designed plant

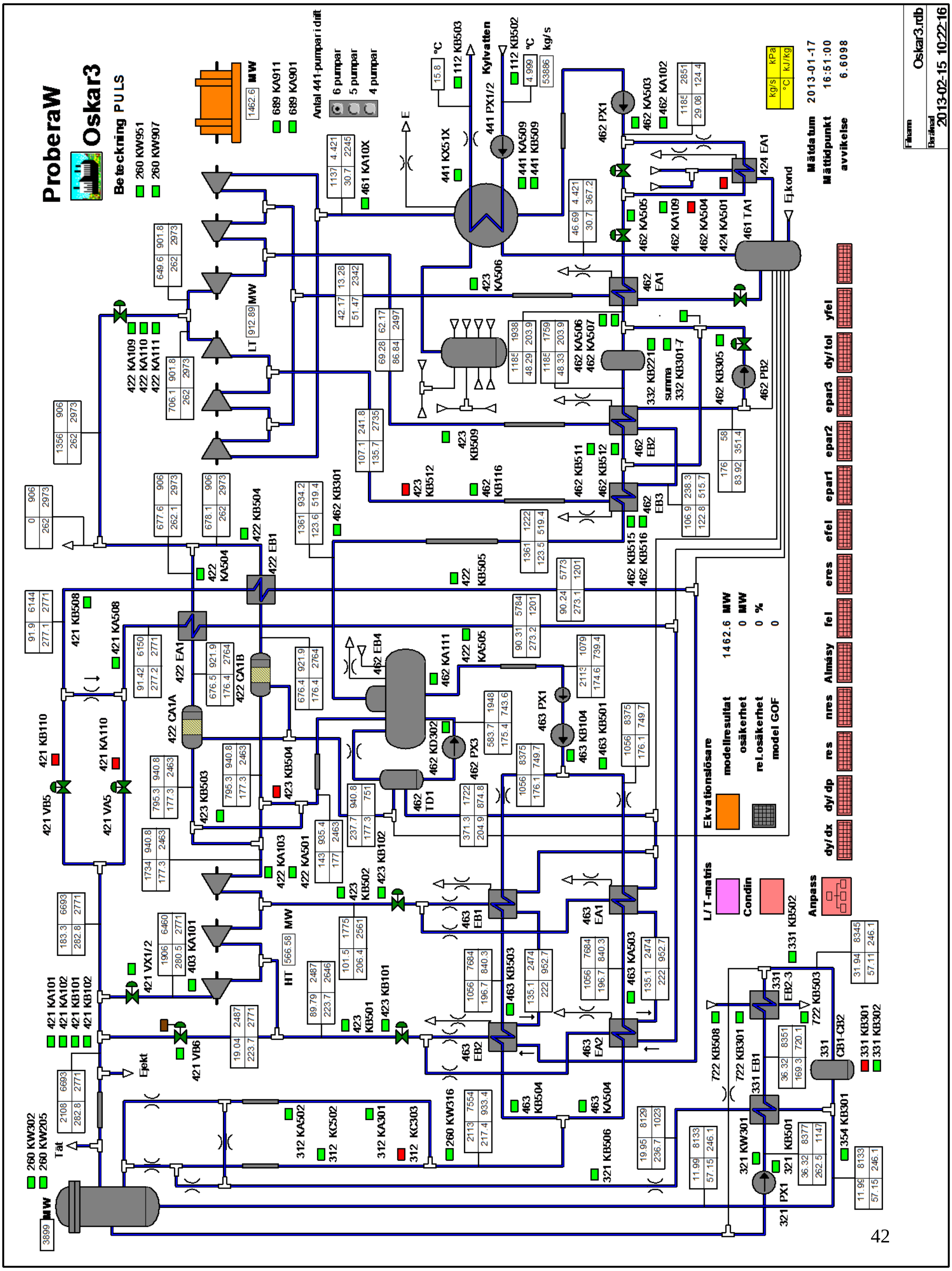


Figure 18, Approximation of the designed plant at a coolingwater temperature of about 5°C.

Adjusted heat balance model

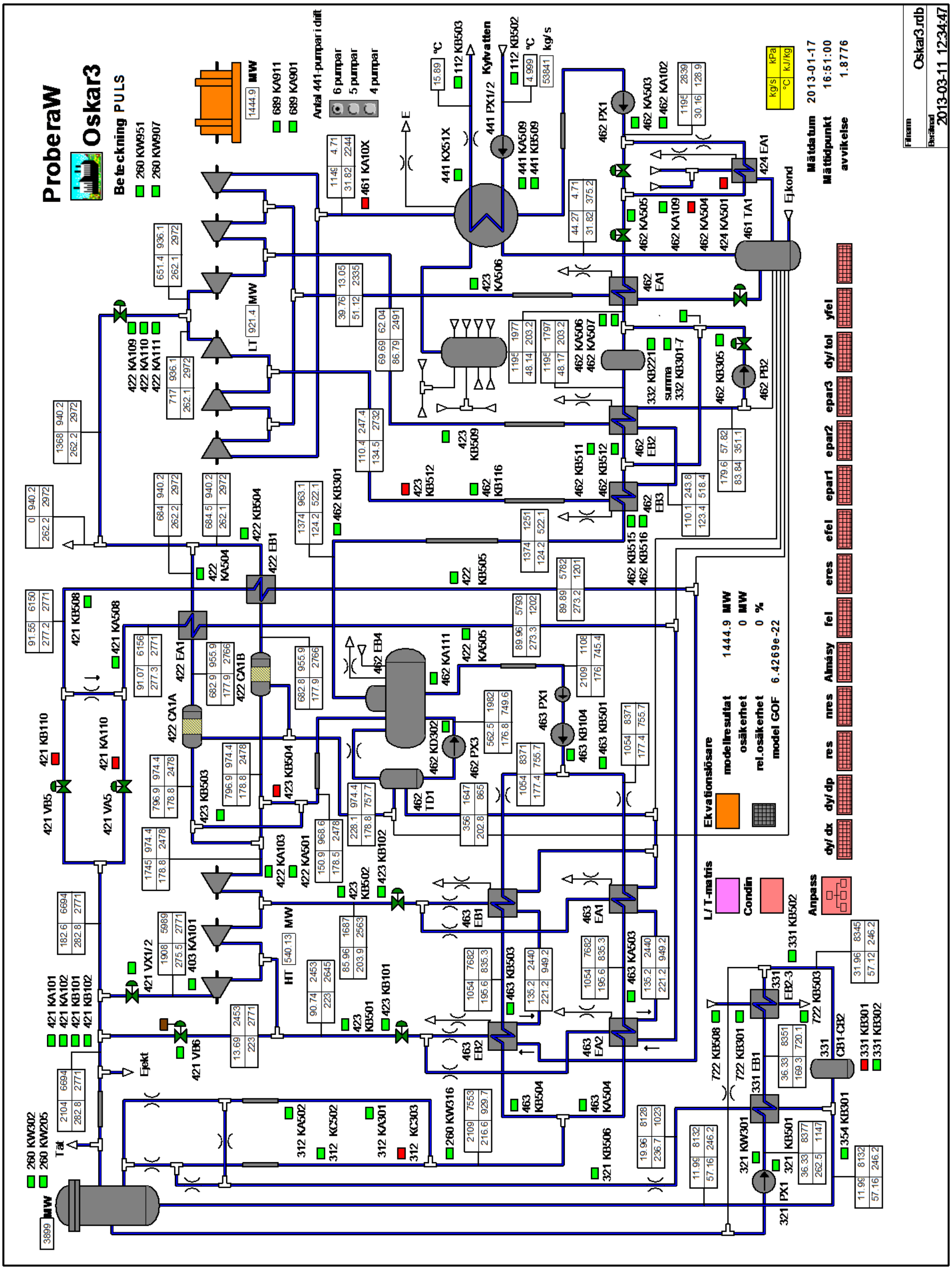


Figure 19, adjusted heat balance model according to actual measurements.

Non-wide HP1 Turbine section

Below is the HP1 Turbine section's affect on production curve presented. The wideness have been changed to its original value as was used in the Probera Design model. It seems like all power output difference between Adjusted model and Probera Design could be explained by a wide HP1-turbine.

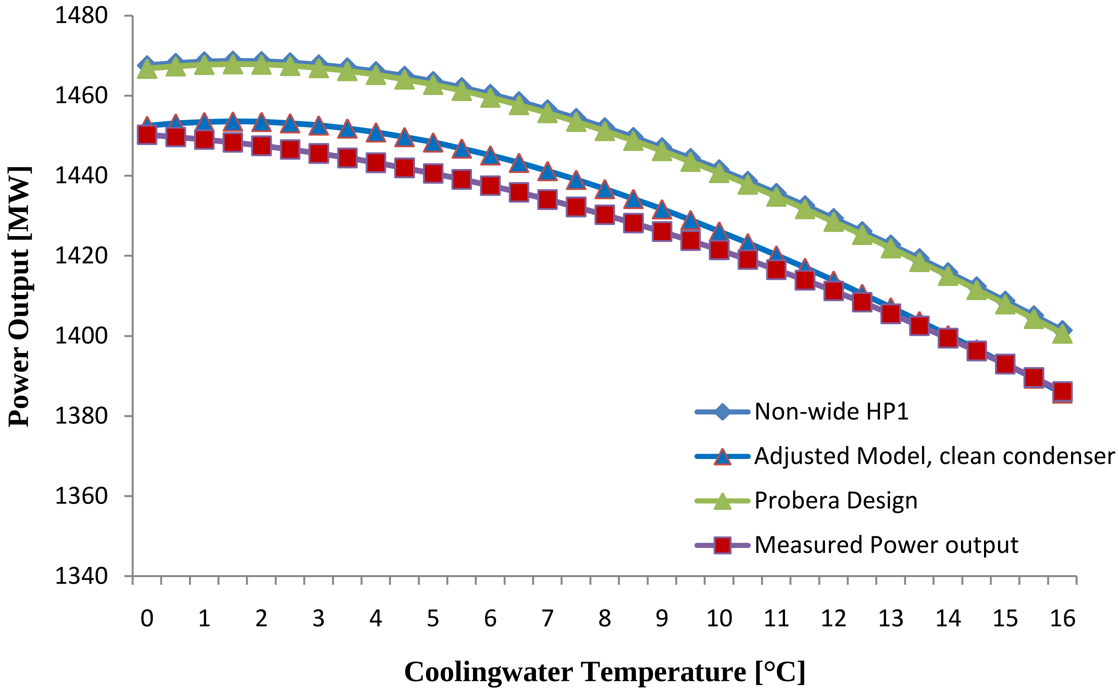


Figure 20, Non-wide HP-turbine section affect on production curve.

Differences between Beckmann and Stodola

Figure 21 shows a production curve presented with both Stodola and Beckmann together with a measured curve over the power output for comparison.

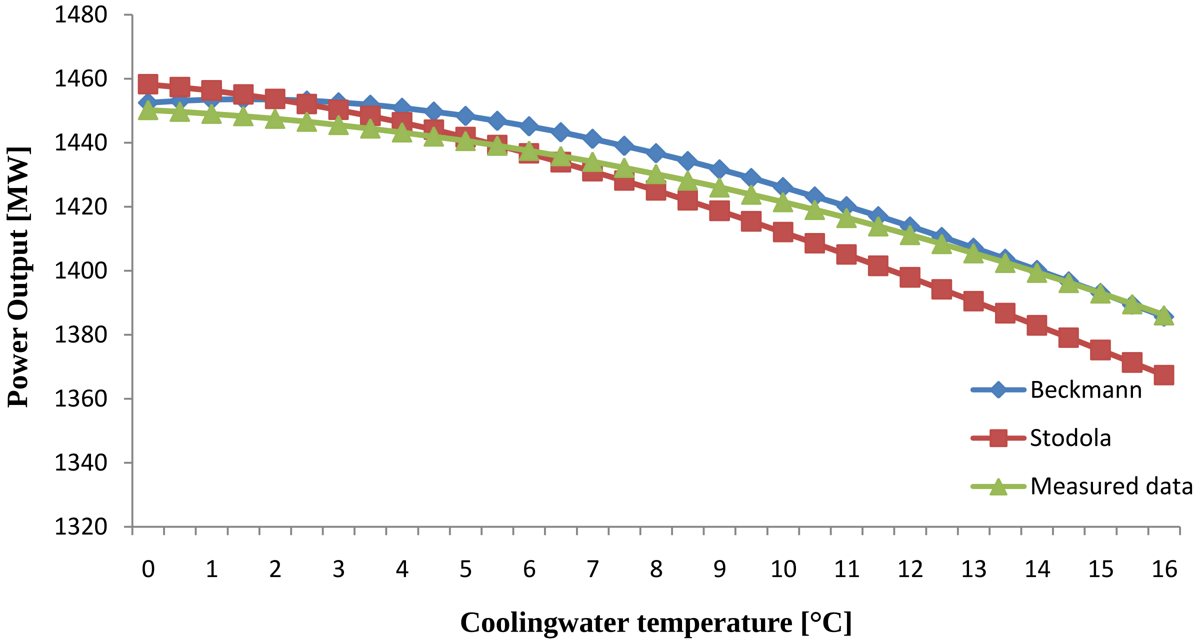


Figure 21, Comparison between Stodola and Beckmann modeling. Measured data with 2nd order polynomial fit.

9.4 Part load data

The following figures shows the raw material for the equations used in the part load analysis of 105.01 % (average) reactor thermal output. 130 data points of 10-minute average values is used from the period of 27/7-11/8 2012.

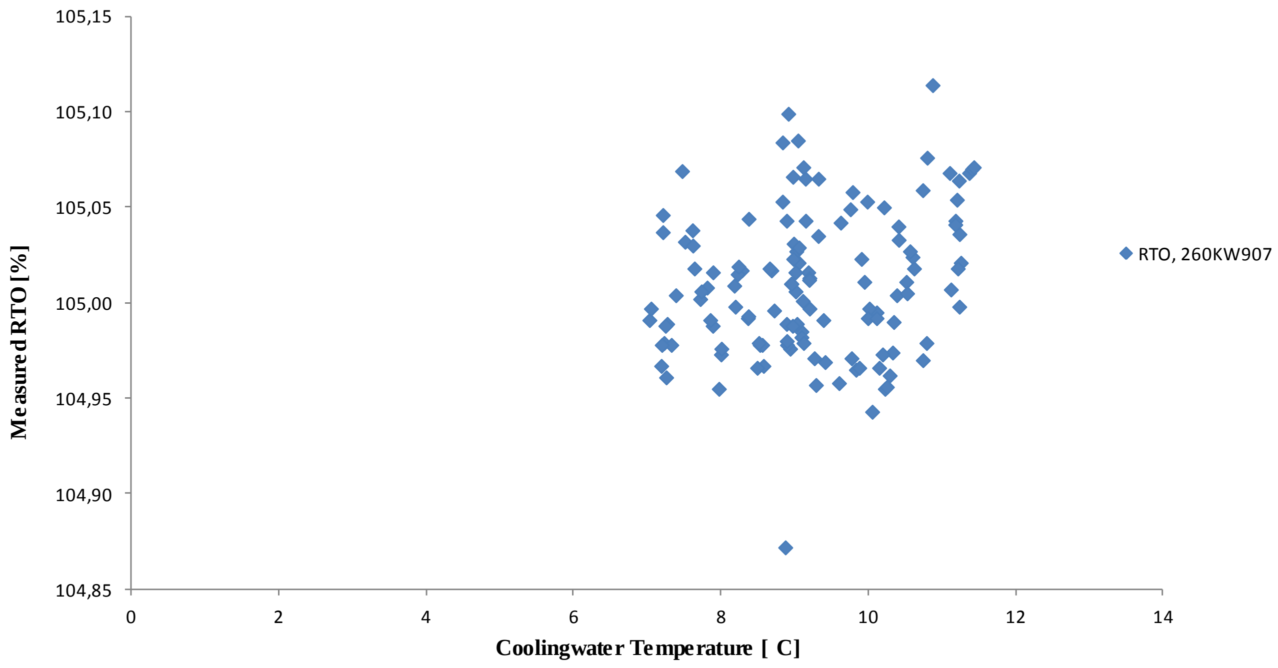


Figure 22, Measured reactor thermal output as a function of cooling water temperature. Part load.

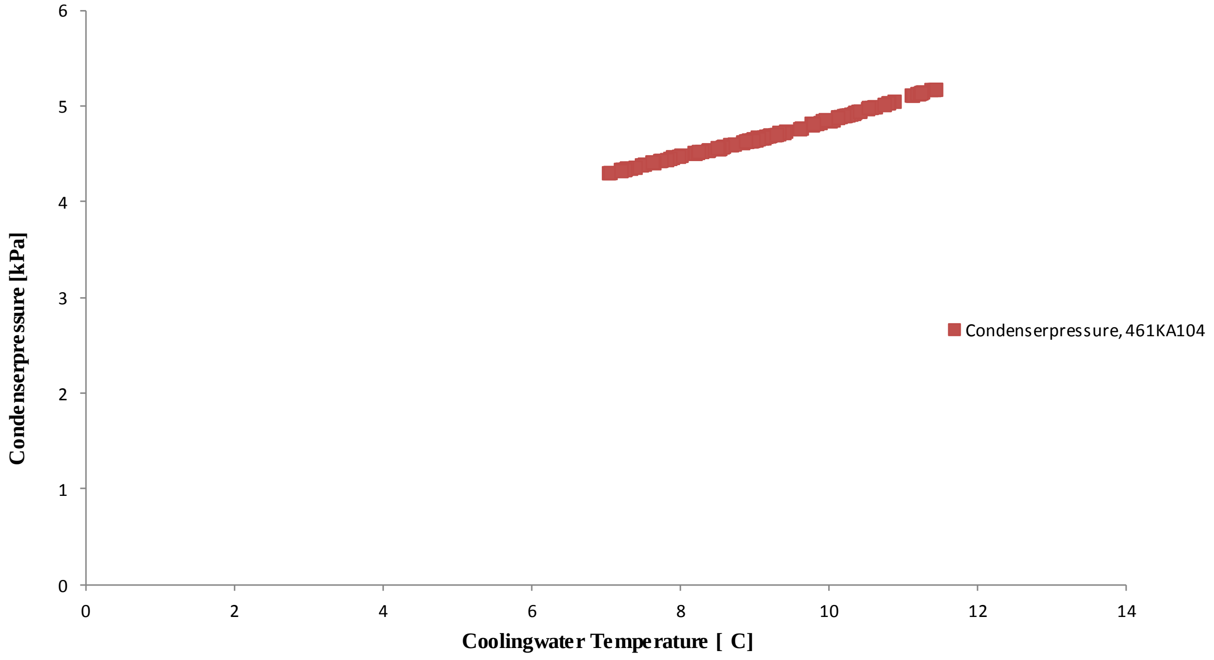


Figure 23, Measured condenser pressure as a function of cooling water temperature. Part load.

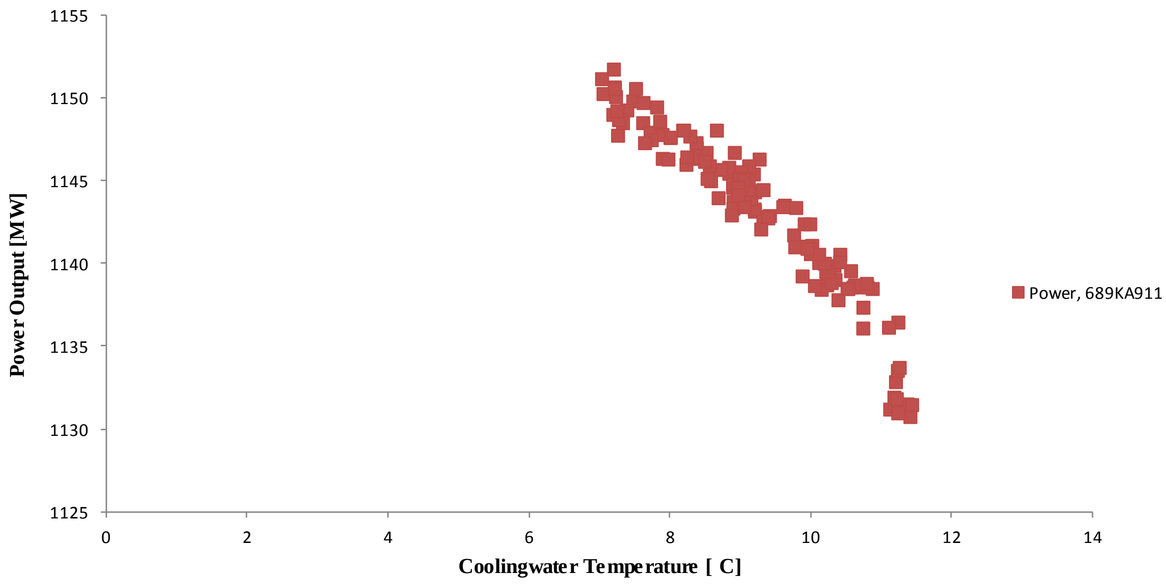


Figure 24, Measured power output as a function of cooling water temperature. Part load.

9.5 Full load data

The full load (129.1 % RTO) data consists of daily values from measurement points which is retrieved during the period of 17/6 -12 to 14/4 -13.

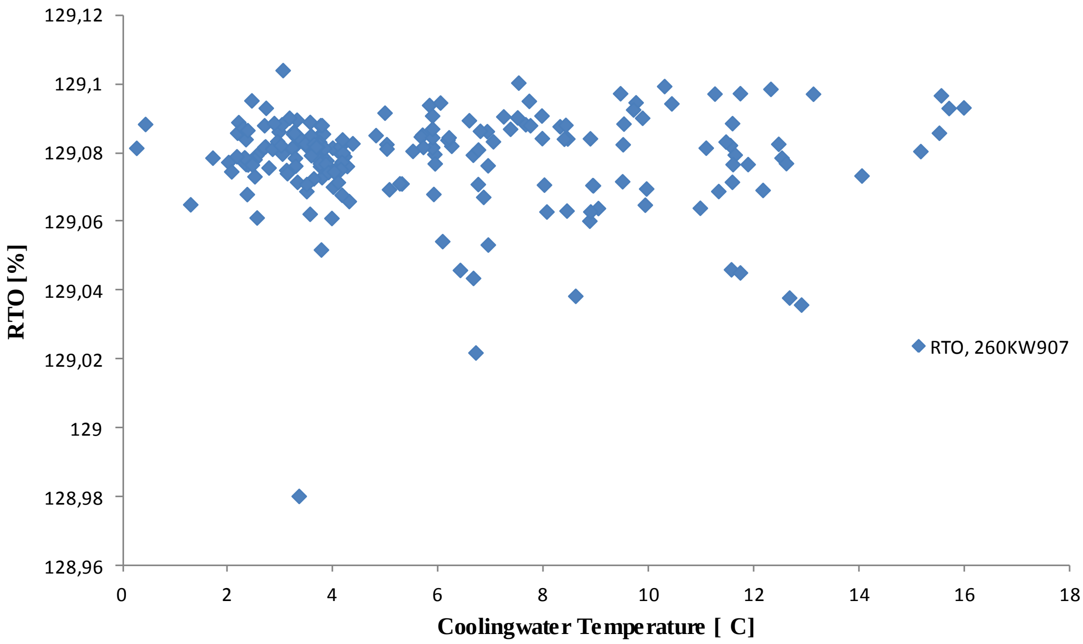


Figure 25, Measured reactor thermal output as a function of cooling water temperature. Full load.

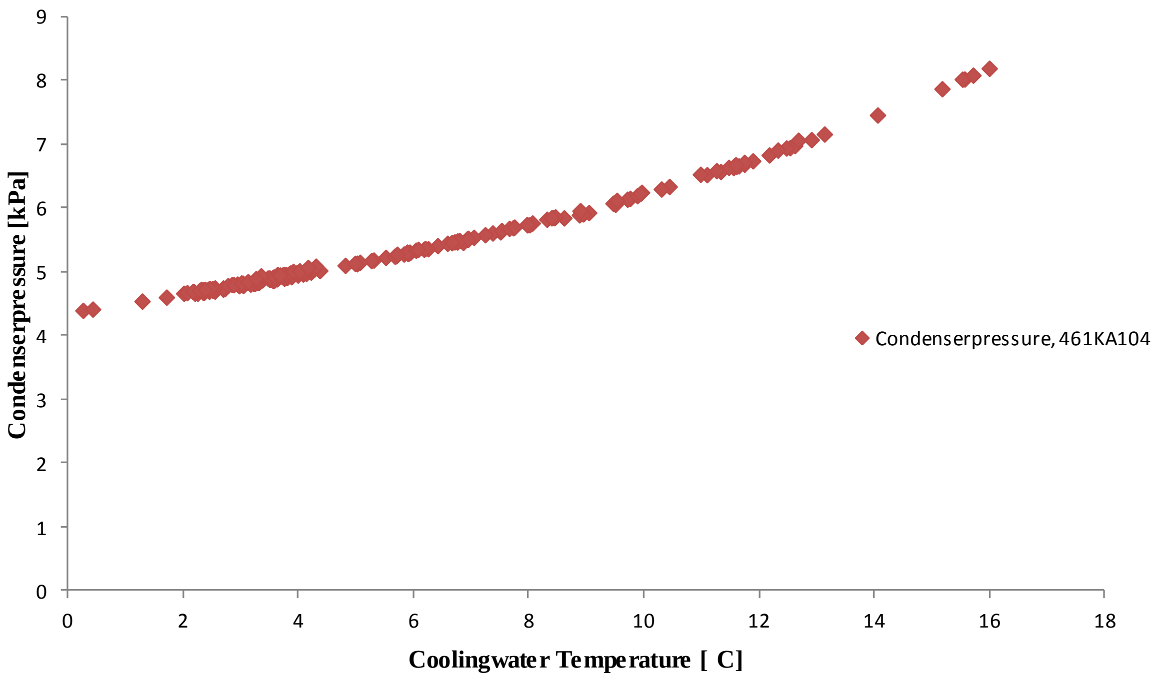


Figure 26, measured condenser pressure as a function of cooling water temperature. Full load.

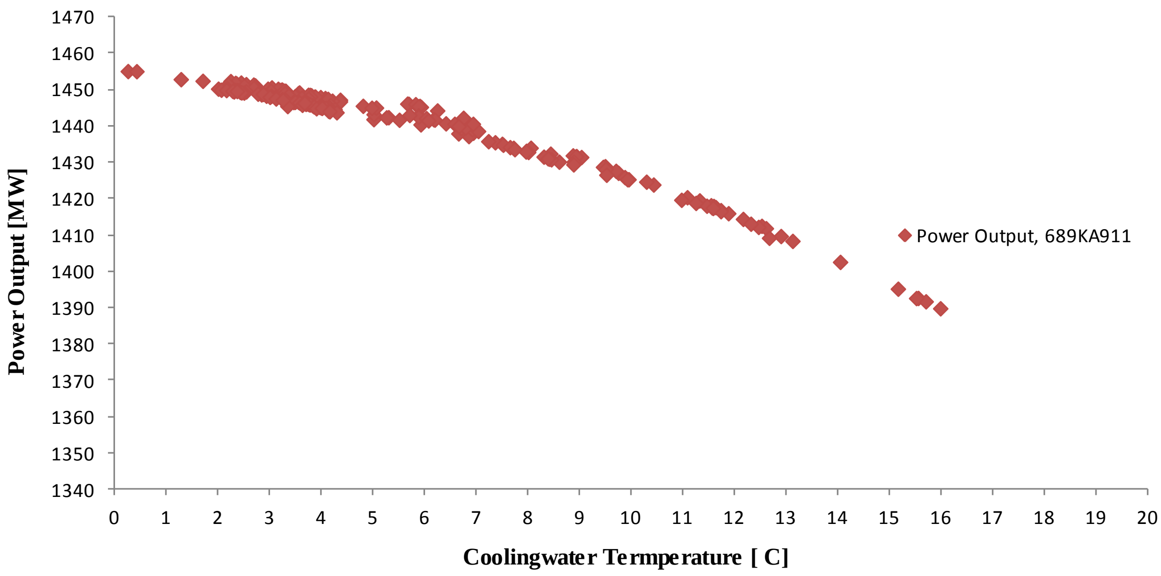


Figure 27, measured power output as a function of cooling water temperature. Full load.

Below is the difference shown between two power output measurements as a function of cooling water temperature. The difference is 3.7 MW at 5°C and varies between 3.3 MW at 0°C to 2.7 MW at 16°C with a peak on 3.7 MW.

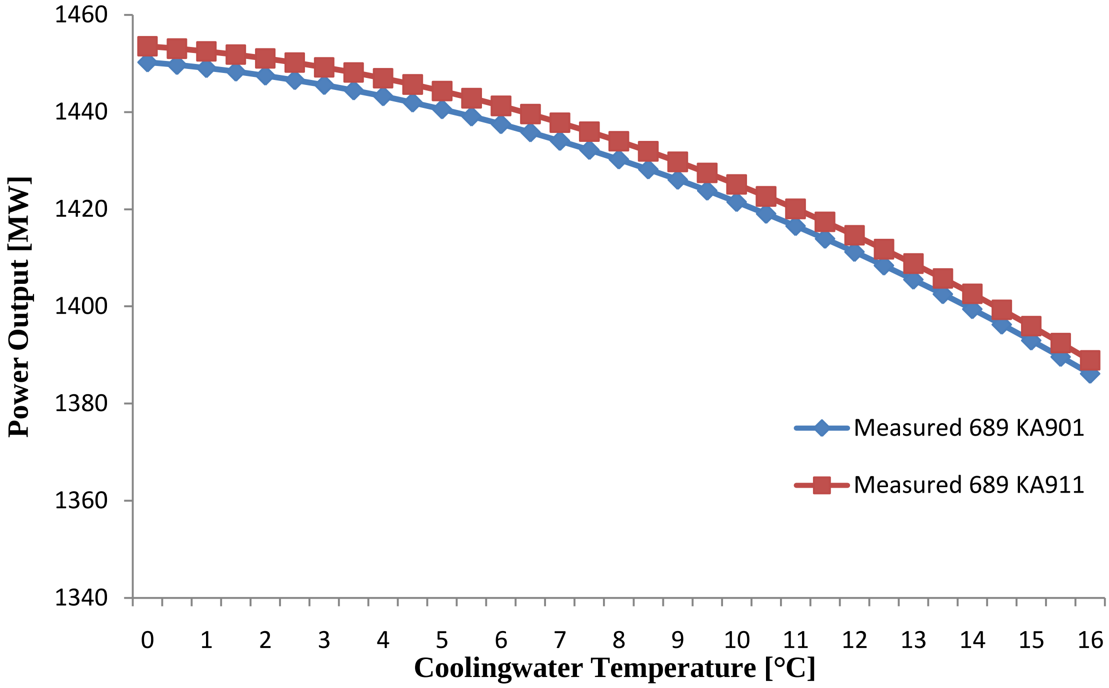


Figure 28, Differences between Power output measurement points, approximated with a 2nd order polynomial.

9.6 Sensitivity Analysis of modeled plant

This section contains a sensitivity analysis of the modeled plant, showing the most important factors in the turbine model's affect on the production curve. The affect on individual turbine sections are not included. In the end of the section a discussion of the analysis is presented.

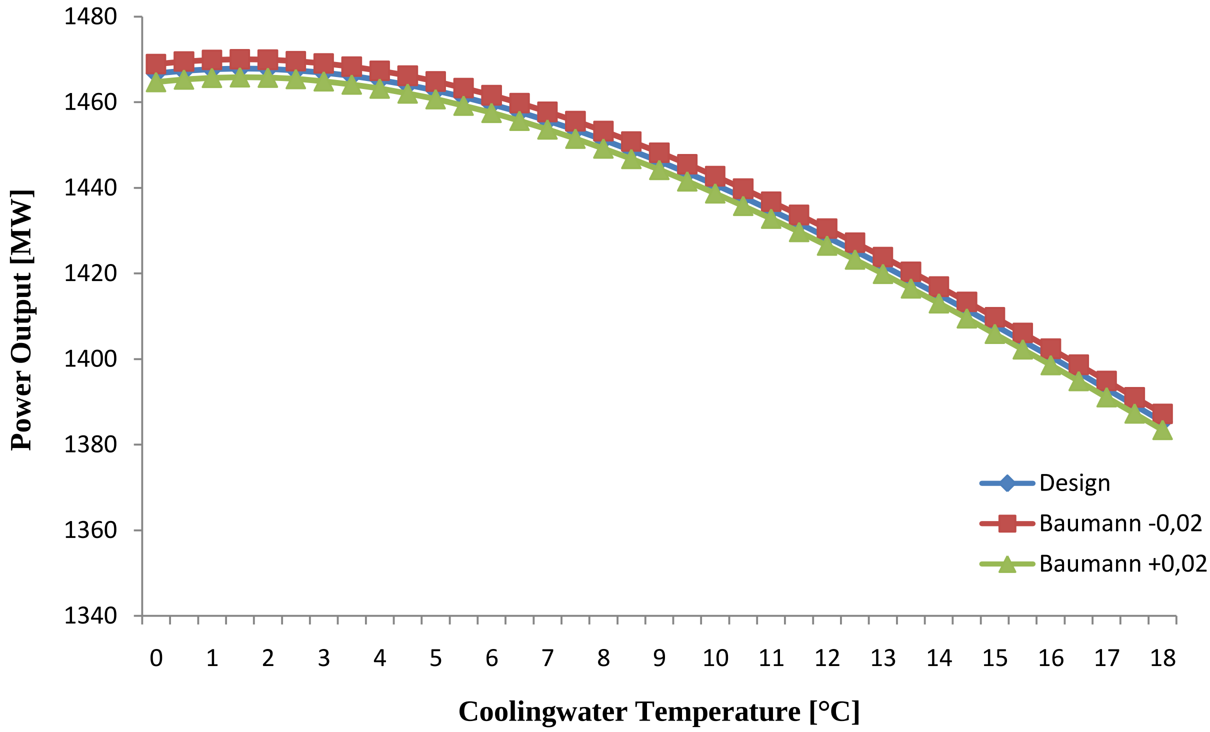


Figure 29, the Baumann-factors affect on power output as a function of cooling water temperature. Designed plant.

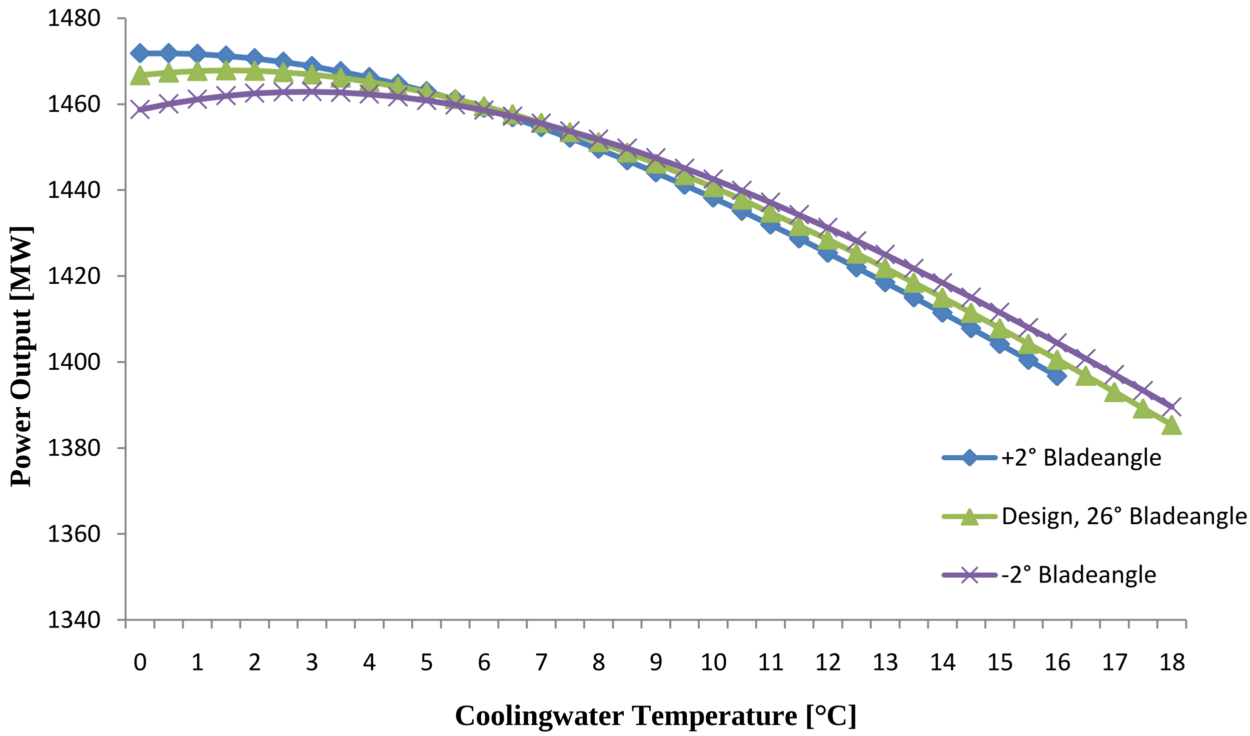


Figure 30, the Blade angles affect on power output as a function of cooling water temperature. Designed plant.

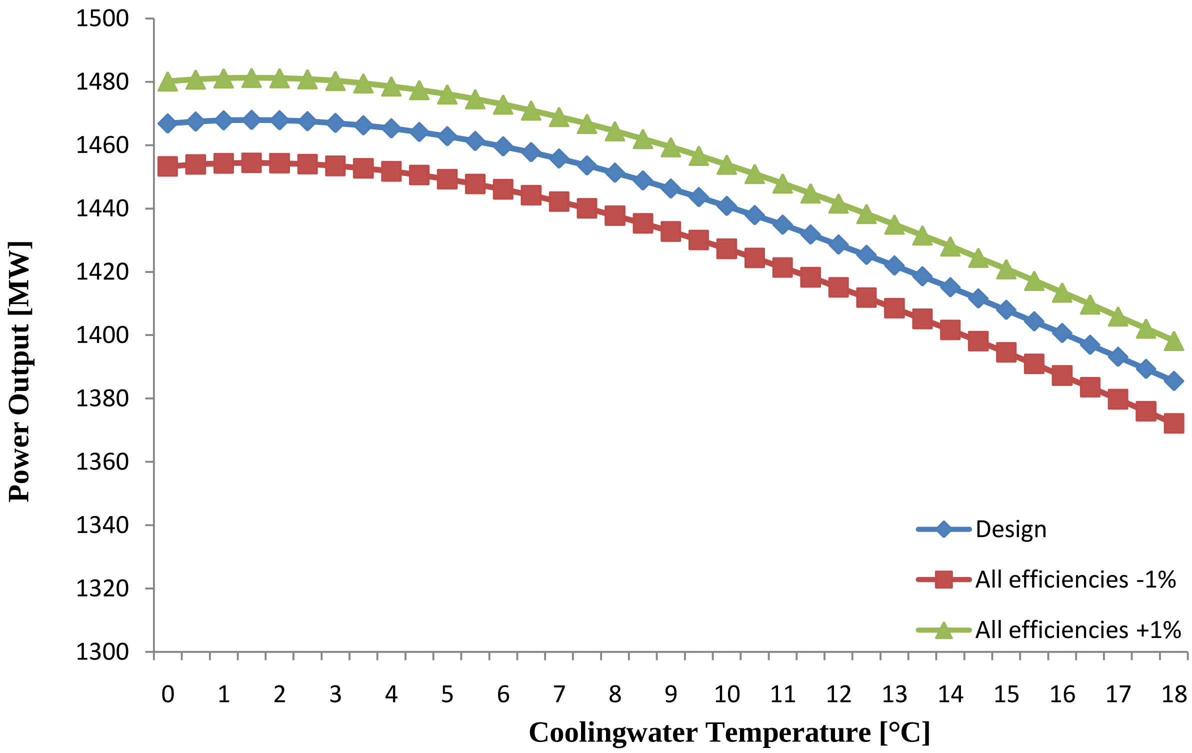


Figure 31, Dry efficiency's affect on power output as a function of cooling water temperature. All Turbine sections. Designed plant.

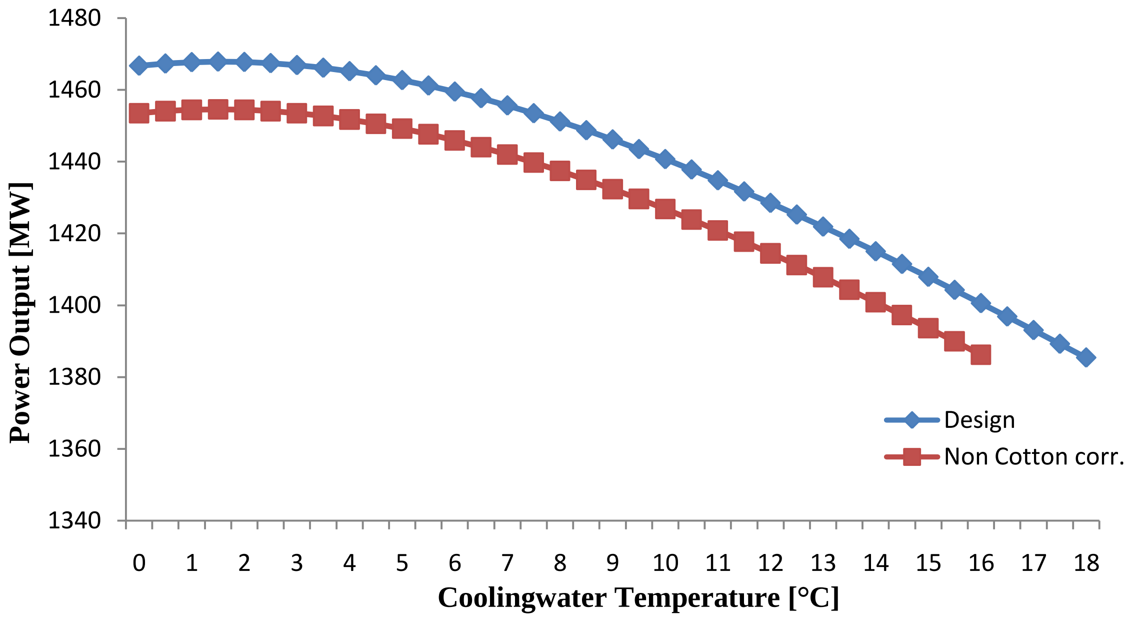


Figure 32, Cotton-correction of wideness' affect on power output as a function of cooling water temperature. All turbine sections. Designed plant.

Discussion

As seen in Figure 29-Figure 32 there is a lot of factors that affect the production curve. The Baumann-factor, dry efficiency and cotton-correction of wideness has a similar impact on the production curve throughout the entire temperature span. The affect on individual turbine sections is however not shown. Some turbine sections operate only in the wet-area and some don't which certainly affect both the cotton correction but also the Baumann affect on efficiency. A higher efficiency of a turbine section gives a lesser moisture content in the outlet thus affecting the previous factors impact on the result. The modeling is therefore somewhat complicated and extensive and especially individual turbine sections should perhaps be investigated further at other temperature ranges, part-load conditions etc. to be able to give a correct description of the behavior of individual parts in the plant.

The blade angles affect on the production curve is more interesting and isn't as entwined as the three factors discussed above. Just a slight change with two degrees affects the curve quite much. It must however be stated that the designed curve matches the measurements very well in the high-temperature region and a change in blade angle doesn't seem reasonable there. A combination however of a bigger blade angle together with a decree in, for example, Baumann-factor would give a different production curve. This would imply a somewhat higher power output than the model used at lower temperatures of the cooling water, but about the same as the used model for higher temperatures.

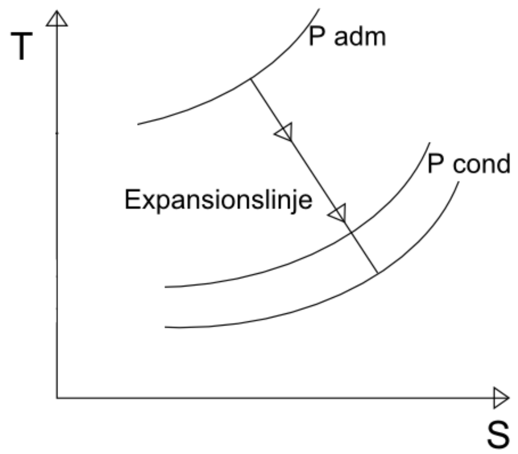
9.7 Maximera elproduktionen - Populärvetenskaplig artikel

För att klara av framtidens utmaningar om en klimatneutral elproduktion är det viktigt att producera så mycket el som möjligt från alla anläggningar i drift. I termiska kraftverk, som till exempel kärnkraftverk, kan tillståndet i anläggningen utvärderas med hjälp av värmebalanser. På så vis kan avvikelser upptäckas och åtgärder vidtagas för att hela tiden hålla anläggningen i gott trim.

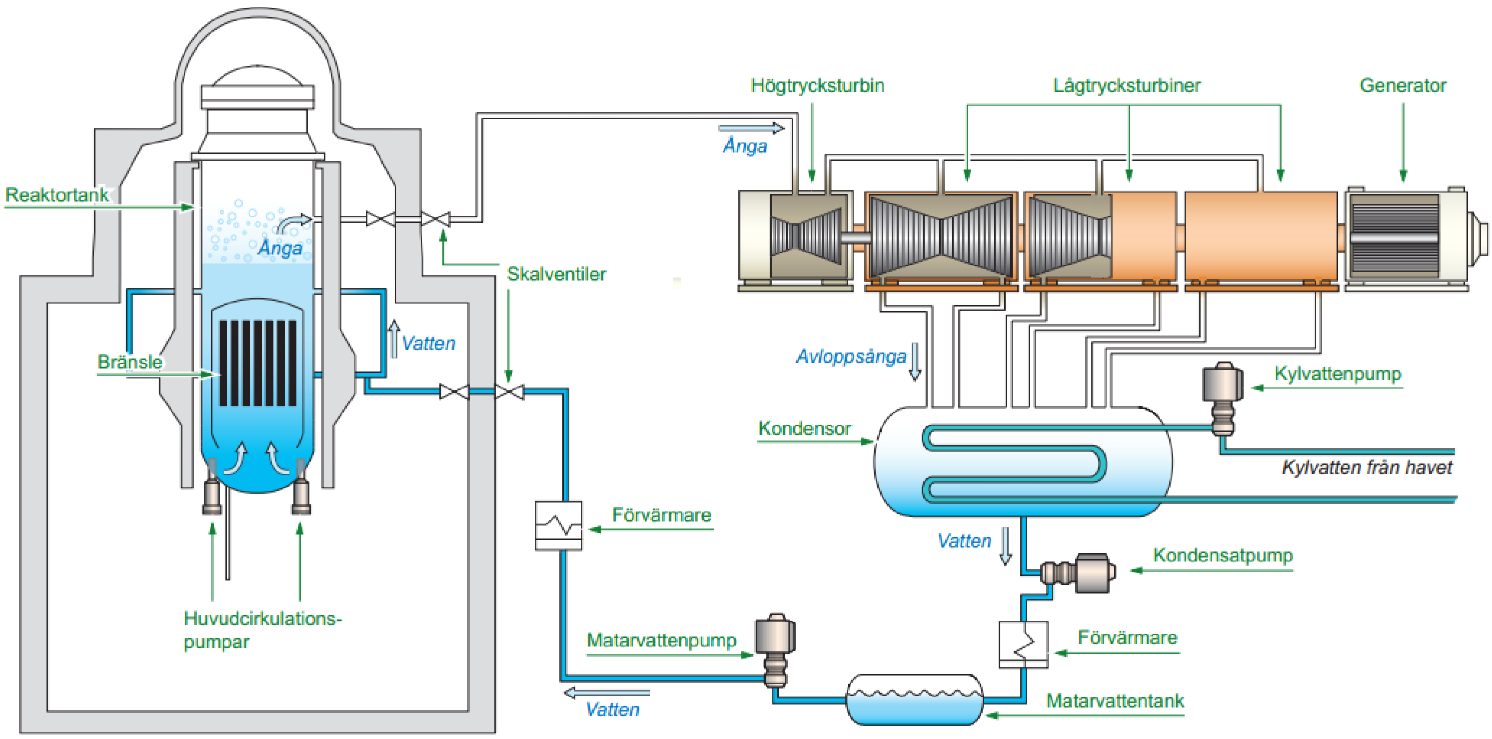
Den stora utmaningen världen står inför i och med klimatförändringarna är att minska växthusgasutsläppen men samtidigt kunna behålla levnadsstandarden för människor i den rika världen och dessutom lyfta folk ur fattigdom. Detta kommer att kräva massor med åtgärder i alla sektorer; byggnader, service, industri, transport och inte minst elproduktion. Sverige är lyckligt lottat i sammanhanget- vår elproduktion består till cirka hälften vattenkraft och knappt hälften kärnkraft som båda släpper ut väldigt lite växthusgaser i jämförelse med andra kraftslag, till exempel kolkraft. Elbehovet i framtiden kommer sannolikt dessutom öka, även om energieffektivisering också är en viktig åtgärd. Till exempel kommer transportsektorns behov av el att öka vid elektrifiering/hybridisering av sektorn för att få bukt med utsläppen. Att varje kraftverk producerar så mycket som möjligt är därför viktigt för hela samhällsekonomin, företagen och alla konsumenter. Arbetet bakom denna artikel handlar om just det- att maximera och optimera elproduktionen hos kärnkraftverket O3 i Oskarshamn som i dagsläget är världens största kokarreaktor.

O3 byggdes på 80-talet och har genomgått ett par effekthöjningar genom åren. Nu senast i ett projekt som kallas PULS (Power Uprate with Licensed Safety) höjdes reaktoreffekten från 110 % av ursprungseffekten till 129 %. Detta har gjort att man bytt en rad komponenter i anläggningen men samtidigt att man till exempel har komponenter som man valt att behålla som därmed lastas högre. Detta gör att det är svårt att säga exakt hur mycket elproduktionen kommer att öka när man höjer reaktoreffekten, leverantören av utrustningen säger en sak men utfallet kan bli ett annat. Arbetet bakom denna artikel har fokuserat på att utvärdera och utreda skillnader mellan det man har sagt att anläggningen kommer att leverera och det faktiska utfallet.

Hur får man då ut el i ett kärnkraftverk?Jo! Egentligen är det helt enkelt så att en reaktor är en stor vattenkokare som kokar vatten till ånga. Denna ånga har ett högt tryck och en hög temperatur (admissionstryck och –temperatur) vilket gör att värmeenergin är hög. Sedan låter man ångan passera en turbin som omvandlar värmeenergin till rörelseenergi vilket generatorn i sin tur omvandlar till elektrisk energi. För att få ut så mycket energi som möjligt ur ångan, försöker man expandera den ned till så låg temperatur och tryck som möjligt. Det låga trycket får man i en stor värmeväxlare som kallas kondensor som kommer efter turbinen och som kyls med havsvatten, det är alltså kylningen av ångan som gör att (kondensor-)trycket sjunker. Ångan får då alltså ett lågt tryck och en låg temperatur vilket gör att expansionen blir lång och elproduktionen hög.



Figur 33, Expansionslinje för en turbin.



Figur 34, Schematisk bild över O3[1].

Om till exempel kylningen av kondensorn med hjälp av havsvattnet fungerar dåligt- en mängd olika faktorer kan tänkas påverka! Så kommer man inte erhålla det låga tryck och den låga temperatur som man räknat med och följaktligen sjunker elproduktionen.

Det har i arbetet visat sig att just kondensortrycket är lite högre än man räknat med och därför skulle O3 kunna producera ytterliggare ca 4-10 MW mer el vilket ungefär motsvarar årsbehovet hos mellan 1000 till 2500 eluppvärmda villor[2]. Förklaringarna kan vara många, kondensorn består av ca 54000 tuber där havsvattnet passerar och kyler ångan och värmeöverföringen i dessa tuber fungerar troligen sämre än beräknat. Det visade sig att en variabel blockering av tuberna fick värmebalansmodellen att stämma bäst mot mätdata. Det innebär att ju kallare det blir i havet, ju fler tuber blir "blockerade" och därmed kan inte ångan kylas ordentligt, kondensortrycket ökar och elproduktionen minskar.

Modelleringsmässigt ser det alltså ut som att man får en ökad "blockering"av tuber ju kallare det blir i Östersjön. Vad kan detta bero på då rent fysikaliskt? Några tänkbara orsaker som presenteras är

- Förändrat/chokat flöde
- Ejektorproblem
- Luftbekymmer

Eftersom kondensorn inte har ersatts med en ny efter PULS och det trots allt går flera hundra kilogram mer ånga genom kondensorn nu än innan PULS, är det inte otänkbart att man fått ett förändrad strömning i kondensorn

1 http://okg.se/Documents/Karnkraft/Processchema_O3.pdf 2013-05-17

2 http://www.energiradgivaren.se/2011/09/elforbrukning-i-en-genomsnittlig-villa-respektive-lagenhet/ 2013-05-06.

som gör att ångan får svårt att kondensera tillbaka till vatten. Detta kan dessutom som följd få att ejektorerna inte suger ut gaser som de ska. En ejektor drivs av färskånga från reaktorn och genom ett lavalmunstycke uppnår ångan kraftig överljudshastighet. Denna hastighet gör att man kan suga ut gaser från kondensorn. Gaser följer alltid med från reaktorn och kan i sin tur förstöra värmeöverföringen. Har strömningen ändrats kan således dessa ”nästen” med ansamlingar av gaser har flyttat på sig och ejektorerna blir oförmögna att suga bort dom.

Luft som tar sig in i systemen måste man alltid beakta. Ett par bekymmer som skulle kunna uppstå är att man får för mycket luft i kondensorn på tubsidan vilket gör att vattennivån i vattenkamrarna sjunker och som följd finns inget vatten i de övre tuberna! Luften kan såklart ta sig in lite varsomhelst, men dessutom är det så att det frigörs luft från havsvattnet eftersom det värms upp. Lösligheten av luft i vatten är dessutom högre vid en lägre temperatur vilket gör att mer luft frigörs när havsvattentemperaturen sjunker vilket kan förklars det variabla beteendet som upptäcktes i modellen.

Självklart står inte alla förklaringsmodeller för sig själva utan snarare blir det mer troligt om man kombinerar ihop dom. Dessutom är det så att det blir mest en form av spekulation angående orsakerna. För att verkligen kunna ta reda på vad som kan tänkas ”blockera” tuberna, bör ett antal temperaturmätningar genomföras både i kondensorn och i ejektorsystemet för att bekräfta eller avvisa troliga orsaker.

En annan stor skillnad som förklarar effekttappet är att högtryckturbinen är för vid. Högtrycksturbinen är den turbin ångan passerar först efter reaktorn och levererar ca 40% av eleffekten. En för vid turbin innebär att turbinen släpper igenom för mycket ånga vilket bland annat gör att trycket innan turbinen sjunker och som tidigare påpekats är högt tryck och temperatur innan turbinerna viktigt för att producera så mycket el som möjligt. Följaktligen innebär detta en minskad eleffekt på troligen ca 9-14 MW vilket motsvarar årsbehovet hos 2500-3500 eluppvärmda villor.

De två stora avvikelser från beräkningarna, det vill säga ett högre kondensortryck och vid högtrycksturbin, innebär alltså en tappad produktion motsvarande årsförbrukningen hos 3500-6000 eluppvärmda villor! Det ska bli spännande att se om man kan komma tillrätta med vissa av bekymren framöver och kräma ut lite mer effekt ur turbin.