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# Once-through and reheater recovery boiler – concept studies

## Final report 15.2.2010

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## Summary

The starting point of this study was increasing kraft pulp mill electricity generation. It has already considerably improved though higher main steam pressure and temperature values.

This research focuses on a new pulp mill with a large recovery boiler at nominal capacity 5500 tds/24h (as fired liquor) which runs at about 200 kg/s steam flow. This study considers both the pulp mill with biomass boiler as well as pulp mill without biomass boiler.

In this study several recovery boiler concepts were compared with the whole pulp mill energy balance being considered. The studied concepts are

A. Natural circulation 82 %, 490 $^{o}$C, 9.0 MPa (reference Joutseno)
B. Natural circulation 85 %, 505 $^{o}$C, 10.2 MPa (reference Kymi)
C. Natural circulation 85 %, 515 $^{o}$C, 12.0 MPa (reference Yonago)
D. Assisted circulation 85 %, 540 $^{o}$C, 16.0 MPa (reference SoTu)
E. Natural circulation 85 %, 515/400 $^{o}$C, 12.0/3.4 MPa (SkyRec)
F. Once-through 85 %, 540/460 $^{o}$C, 26.0/5.4 MPa (SkyRec+)

Steam and electricity generation for each recovery boiler case was calculated. The steam production increases from Case A to Case B because of higher black liquor dry solids and more air preheating. The steam production increases from Case B to Case C because of high pressure preheating. The recovery boiler steam flow starts decreasing as further increases in main steam parameters require more heat.

As can be seen the modern recovery boiler Case C does produce about 20 % more electricity than roughly ten years ago, case A. Reheating cases E and E160 seem to give only marginally better electricity production. The only alternative seems to be to increase the main steam temperature to 540 $^{o}$C, Cases D and F. The pulping electricity usage is not constant. The main parameter that changes is the recovery boiler feedwater pump power requirement.

Electricity generation does not depend a lot on how the boiler steam side is configured. Reheating and once-through appear only marginally better when considering the recovery boiler electricity generation.

The increase in electricity generation seems very profitable up to case C. This confirms the rationality of design choices that have lead to the present recovery boiler. Case A costs more that it should were it built today. The reason is larger than required superheating surface and smaller than currently used superheater tube size. From cost of additional power, going to SoTu concept of 540 $^{o}$C steam seems desirable. Currently the corrosion issues have not yet been solved so in this study we assume that superheaters do not corrode. Reheater boiler concept seems not at all profitable. The additional electricity generation was only marginal. Once-through recovery boiler did produce as much additional electricity than the SoTu concept of 540 $^{o}$C steam. The corrosion issues still remain the same.

# Yhteenveto

Tutkimuksen lähtökohtana on sellutehtaan sähköenergian kehityksen parantaminen. Sellutehtaan sähköenergiaa ovat parantaneet erilaiset jo käytetyt tavat kuten päähöyryn lämpötilan tai paineen nosto.

Tässä tutkimuksessa on otettu lähtökohdaksi kokonaan uusi sellutehdas, jossa nimelliskooltaan 5500 tka/24h (polttolipeänä) soodakattila ajaa noin 200 kg/s höyrykuormaa. Tarkasteluna on käsitelty sekä soodakattilan että voimakattilan muodostamaa kokonaisuutta samoin kuin pelkästään soodakattilan muodostamaa kokonaisuutta.

Tässä tutkimuksessa tarkastellaan useita soodakattila vaihtoehtoja. Kussakin tapauksessa on energiataseena käytetty koko sellutehtaan energiatasetta. Tutkitut tapaukset ovat

A. Luonnonkierto 82 %, 490 $^{o}$C, 9.0 MPa (kuten Joutseno)
B. Luonnonkierto 85 %, 505 $^{o}$C, 10.2 MPa (kuten Kymi)
C. Luonnonkierto 85 %, 515 $^{o}$C, 12.0 MPa (kuten Yonago)
D. Avustettu kierto 85 %, 540 $^{o}$C, 16.0 MPa (kuten SoTu)
E. Luonnonkierto 85 %, 515/400 $^{o}$C, 12.0/3.4 MPa (SkyRec) Välitulistus!
F. Läpivirtaus 85 %, 540/460 $^{o}$C, 26.0/5.4 MPa (SkyRec+) Välitulistus!

Kullekin tapaukselle laskettiin höyryn ja sähkön kehitys. Höyryn määrä nousee kun mustalipeän kuiva-aine kasvaa ja ilman esilämmitys nousee Tapaus A ja Tapaus B. Höyryn kehitys nousee Tapaus B ja Tapaus C koska aletaan käyttää korkeapaine-esilämmitystä. Tämän jälkeen alkaa höyryvirta laskea kun päähöyryn arvoja parannetaan jolloin tarvittava lämpömäärä kiloa höyryä kohti nousee.

Kuten huomataan niin moderni soodakattila Tapaus C tuottaa noin 20 % enemmän höyryä kuin suurin piirtein kymmenen vuotta sitten Tapaus A. Välitulistustapaukset E ja E160 näyttää lisäävän sähkön tuotantoa vain hieman. Vaihtoehto on lisätä höyryn lämpötilaa 540 $^{o}$C, Tapaus D ja F. Sellutehtaan käyttämä sähkö ei pysy vakiona. Erityisesti syöttövesipumpun vaatima sähköteho nousee kattilan paineen noustessa.

Sähkön tuotanto ei näytä olevan merkittävästi riippuvainen siitä millaiseksi vesikiero on valittu. Välitulistus ja läpivirtaus näyttävät antavan vain hieman paremman lopputuloksen soodakattilan sähkön tuotannoksi.

Sähkön lisätuotanto näyttää hyvin kannattavalta aina vaihtoehtoon Tapaus C asti. Tämä vahvistaa että tähän asti on edetty järkevää polkua eteenpäin. Tapaus A maksaa enemmän kuin mitä se maksaisi tänä päivänä. Syynä tähän on suuri tulistinpinta ja pienehkö tulistimien putkikoko. Jos ajatellaan sähkötehoa, SoTu Tapaus 540 $^{o}$C höyry näyttää hyvinkin kannattavalta. Toistaiseksi korroosion vähentämisessä ei ole varmoja menetelmiä. Tässä tutkimuksessa oletettiin että korroosiota ei tapahdu. Välitulistus soodakattila ei näytä kovinkaan kannattavalta. Lisäsähkön tuotanto oli vain marginaalista. Läpivirtauskonsepti tuotti merkittävästi lisää sähköä samoin kuin Tapaus SoTu koska sähkö 540 $^{o}$C. Korroosioasiat olisivat läpivirtauskattilalla SoTu vaihtoehtoa vastaavat.

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## ABBREVIATIONS

| | |
|---|---|
| CHP | combined heat and power |
| CNCG | concentrated non condensable gases |
| DNB | departure from nucleate boiling. |
| DNCG | diluted non condensable gases |
| HP | high pressure steam (steam from recovery boiler) |
| LP | low pressure steam (3.5  - 6 bar) |
| MCR | maximum continuous rating |
| MP | medium pressure steam (9.5 – 14 bar) |
| NBSW | Northern bleached softwood pulp |

# 1 INTRODUCTION

The Finnish Recovery Boiler Committee has undertaken to study the future recovery boiler concepts. The aim is to increase the electricity generating potential and energy efficiency of recovery boilers. This is in line with the Finnish Government's long-term climate and energy strategy and the aims of the European Union and its objectives. This work has been supported by Tekes.

The work consists of once-through recovery boiler concept and study how this might increase the electrical efficiency. Necessary automation and safety changes are to be listed. Final report is a summary of the main tasks.

1. Calculate mass and energy balances to a recovery boiler of about 200 $kg_{steam}/s$ (~5000 tds/d).
2. Place heating surfaces at several typical pressures and temperatures along the flue gas flow
3. Show how water-steam circulation is done
4. Examine 100 % ja 80 % flows with concepts studied with APROS
5. Examine the effect of feedwater preheating
6. Examine the effect of air preheating
7. Look at how placing of reheater affects the recovery boiler

## 1.1 Comments to targets given 25.6.2009

SKYREC organized a meeting to get comments on different concepts studied. The following comments were received

- Maximum current superheating with long term experience is 515$^{o}$C
- Calculate once-through boiler with reheater (without reheater does not make sense)
- The need is to calculate new concepts 515$^{o}$C with reheating
- Choose main steam and reheated steam values so that for LP-steam desuperheating is maximum 30 $^{o}$C or minimum 2 % of moisture.
- Study effect of feedwater preheating for Case C include the resizing of superheaters
- Study effect of air preheating for Case C include the resizing of superheaters
- Do mill steam consumption and electricity steam consumption separate to recovery boiler
- Reheater boiler needs more Sootblowing steam
- Sanicro price is ~300 €/m
- Examine low pressure or medium pressure steam production from flue gases

Case C represents the state-of-the-art with respect of electricity generation. It utilizes the maximum main steam temperature that can be considered currently economically 515$^{o}$C. The mill is producing MP or LP steam from the recovery boiler flue gases 197$^{o}$C -> 155$^{o}$C.

To find from literature clear comparisons of current cases e.g. Swedish boilers and Finnish boilers.

The main steam pressure and temperature of concept D needs more thought.

When choosing the main steam values the low pressure steam the maximum moisture allowed for LP steam is 2 % and the maximum attemperating fro Lp steam is 30 $^{o}$C.

There is no reason to consider just once-though boiler but to consider once-through reheater boiler.

# 2 DEVELOPMENT OF RECOVERY BOILER

The pulp and paper industry faces a new era. New environmental expectations have appeared. Cutting down air emissions is not enough. Pulp and paper mills need to maximize their bioenergy potential and minimize their electricity consumption to produce green electricity.

Recovery boilers, which produce bioenergy, are built all over the world. The recovery boiler has developed in the past 70 years. It has culminated with units that are among largest biofuel boilers in the world. For pulp mills the significance of electricity generation from the recovery boiler has been secondary. The most important design criterion for the recovery boiler has been the high availability.

## 2.1 Main steam temperature

Maximizing electricity generation is driving increases in the main steam pressures and temperatures. The maximum steam temperature can be limited by the ash properties. The first melting curve at the superheater front should be taken into account. Increasing mill closure with high chlorine and potassium decreases the melting temperatures. The overall mill heat balance should be used to optimize the feed water and flue gas temperatures.

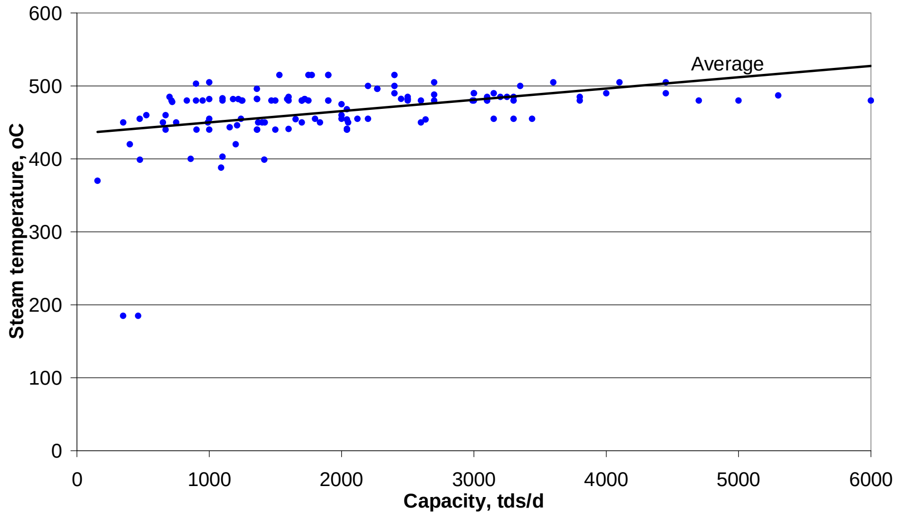


**Figure 2-1,** Main steam temperature as a function of recovery boiler capacity.

The main steam temperature of recent recovery boilers is shown in Figure 2-1 as a function of MCR capacity of that boiler. The average steam temperature increases with size. Small boilers tend to have lower pressures to reduce specific cost. There are many boilers with main steam parameters higher than 500 $^{o}$C. Most of them are in Japan.

## 2.2 Relationship between main steam values

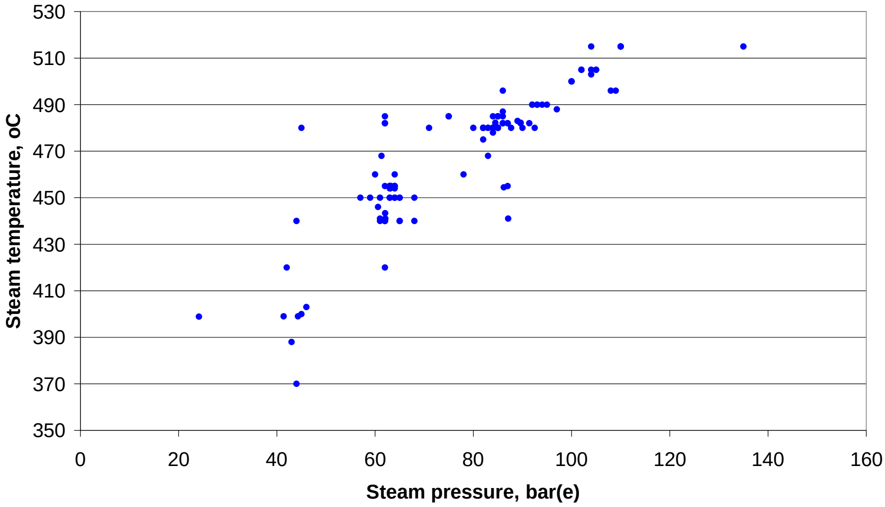


**Figure 2-2,** Main steam temperature as a function of recovery boiler main steam pressure.

The main steam temperature of recovery boilers is shown in Figure 2-2 with the corresponding main steam pressure. An increase in main steam temperature is usually accompanied with an increase in the main steam pressure. Increasing just either steam pressure or temperature alone has only a minor effect on back pressure electricity generation.

## 2.3 Black liquor dry solids

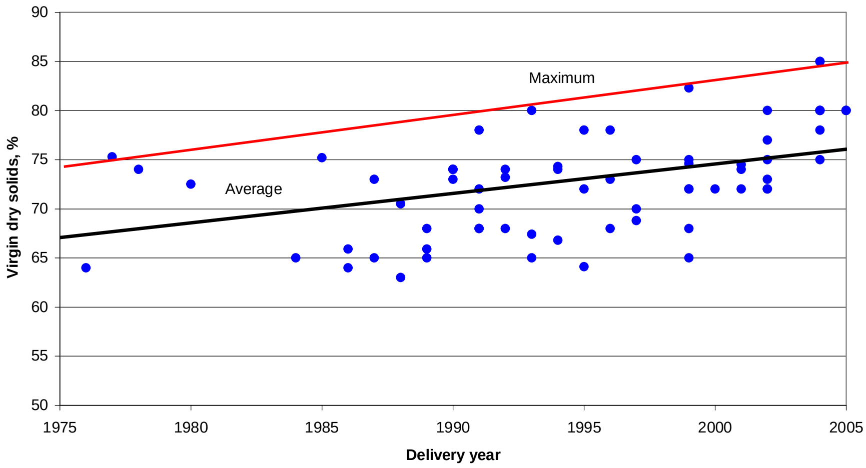


**Figure 2-3,** Virgin black liquor dry solids as a function of purchase year of the recovery boiler.

Black liquor dry solids has always been limited by the ability of available evaporation technology to handle highly viscous liquor. As technology has evolved so has the final black liquor dry solids. The virgin black liquor dry solids of recovery boilers is shown in Figure 2-3 as a function of purchase year of that boiler.

On average the virgin black liquor dry solids content has increased. This is especially true for latest very large recovery boilers. Design dry solids for the new green field mills and new recovery islands have been either 80 or 85 % dry solids. In Asia and South America 80 % (earlier 75 %) dry solids is in use. In Europe 85 % (earlier 80 %) dry solids is in use.

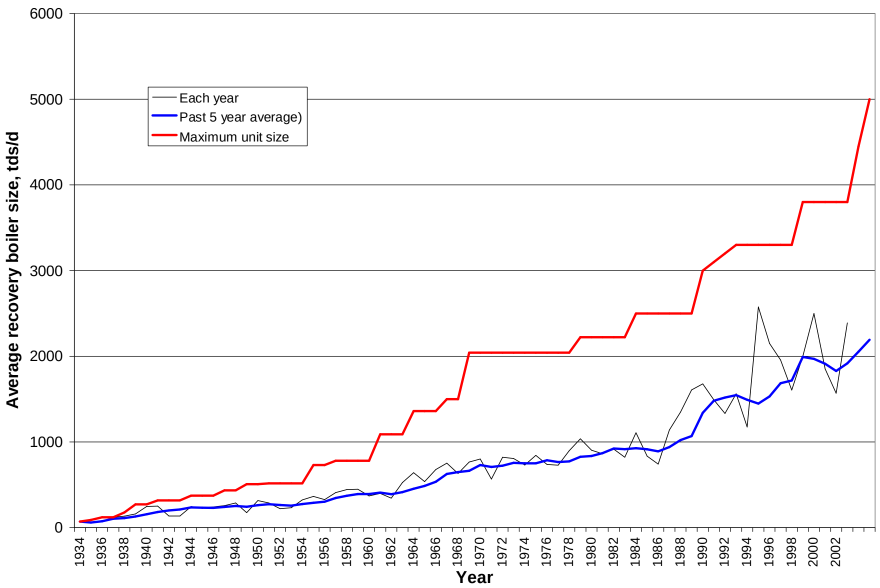


**Figure 2-4,** Size of the recovery boiler as a function of purchase year.

Recovery boiler size keeps increasing. The recovery boiler size doubles about every 20 years. Boilers with over 200 square meter bottom area are typical for the largest new greenfield mills. The largest recent proposals have been for a 6000 tds/d boiler. Average boiler size has typically been about half of the largest boiler bought.

The recovery boiler is now challenging circulating fluidized bed boilers for the title of largest bio-fuel fired boiler. Recovery boiler furnace size is about the size of the largest natural circulation coal fired boilers. This means that the existing mechanical and commercial limits of furnace size for natural circulation units have now been reached.

# 3 RECOVERY BOILER CONCEPTS

In the study some of the existing recovery boiler concepts are compared to future recovery boiler concepts. In the meeting 26.6.2009 it was agreed that the studied concepts are

A. Natural circulation 82 %, 490 $^{o}$C, 9.0 MPa (reference Joutseno)
B. Natural circulation 85 %, 505 $^{o}$C, 10.2 MPa (reference Kymi)
C. Natural circulation 85 %, 515 $^{o}$C, 12.0 MPa (reference Yonago)
D. Assisted circulation 85 %, 540 $^{o}$C, 16.0 MPa (reference SoTu)
E. Natural circulation 85 %, 515/400 $^{o}$C, 12.0/3.4 MPa (SkyRec)
F. Once-through 85 %, 540/460 $^{o}$C, 26.0/5.4 MPa (SkyRec+)

## 3.1 Case A - Modern high efficiency boiler

The modern recovery boiler is of a single drum design, with vertical steam generating bank and wide spaced superheaters. The most marked change was the adoption of single drum construction. The construction of the vertical steam generating bank is similar to the vertical economizer.

The effect of increasing dry solids concentration has had a significant effect on the main operating variables. The steam flow increases with increasing black liquor dry solids content. Increasing closure of the pulp mill means that less heat per unit of black liquor dry solids will be available in the furnace (Clement 1990).

The flue gas heat loss will decrease as the flue gas flow diminishes. Increasing black liquor dry solids is especially helpful since the recovery boiler capacity is often limited by the flue gas flow.

The most marked change in this case was that the mill (market NBSW pulp mill) was able to produce substantial amounts of electricity for sale even when selling all bark to neighbouring mills (Veitola, 2000).

In this concept the benefits of having a condensing tail were first realized. In addition this was one of the first boilers where biosludge with both DNCG and CNCG were burned with high dry solids black liquor (Vakkilainen, 2000)

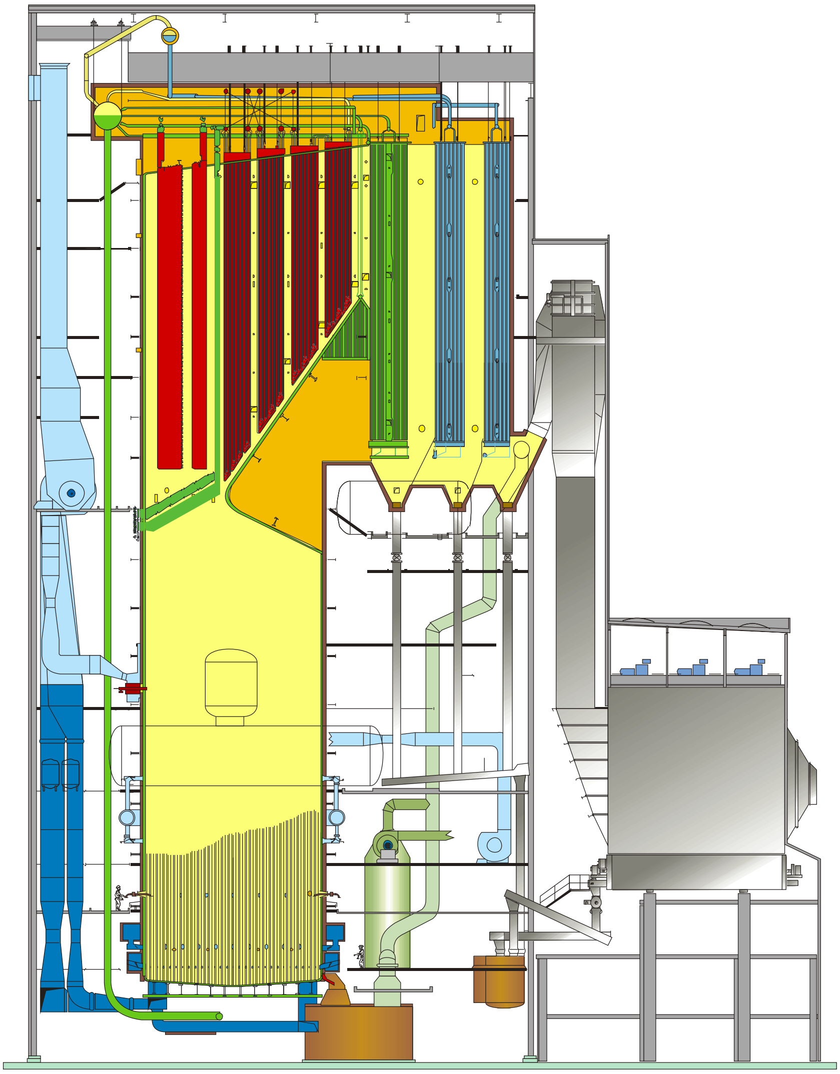

**Figure 3-1,** Modern recovery boiler, Vendor Andritz Oy, Capacity 3500 t ds/24h, Black liquor ds 82,3 % (80 %) Main steam 130 kg/s 93 bar(a) 490 °C.

### 3.2 Case B – High efficiency recovery boiler

During recent years the price of electricity has increased and especially the desirability of electricity produced from renewable fuels has risen. This has led to mills adopting strategies to increase the electrical generating efficiency (Raukola et al., 2002, Saviharju and Lehtinen 2005, Westberg 2007).

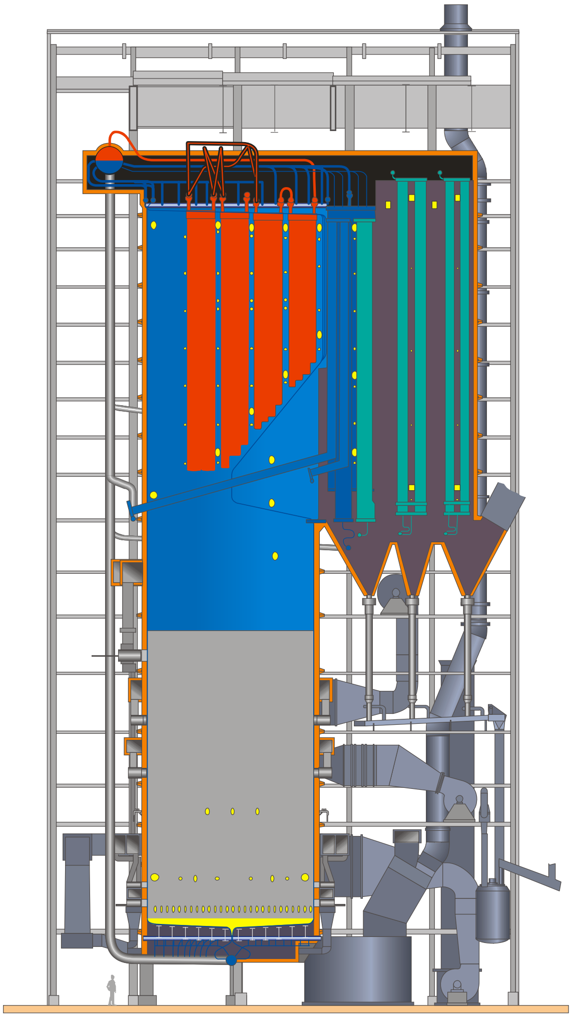

**Figure 3-2,** High efficiency recovery boiler, Vendor Metso Power Oy, Capacity 3600 t ds/24h, Black liquor ds 85 % (80 %) Main steam 170 kg/s 102 bar(a) 505 °C.

One of the aims of this project was to increase the share of electricity produced with biomass (Tikka, 2008). A special feature of this boiler is usage of 29 bar unregulated steam for sootblowing and high air preheating temperature to 190 °C. The feedwater inlet temperature has been raised ot 148 °C and there is an intermediate HP steam feedwater preheat stage between economizer I and Economizer II. Because of the high superheating temperature the last tubes of tertiary superhaters are of Sanicro 28 (Aikio, 2008)

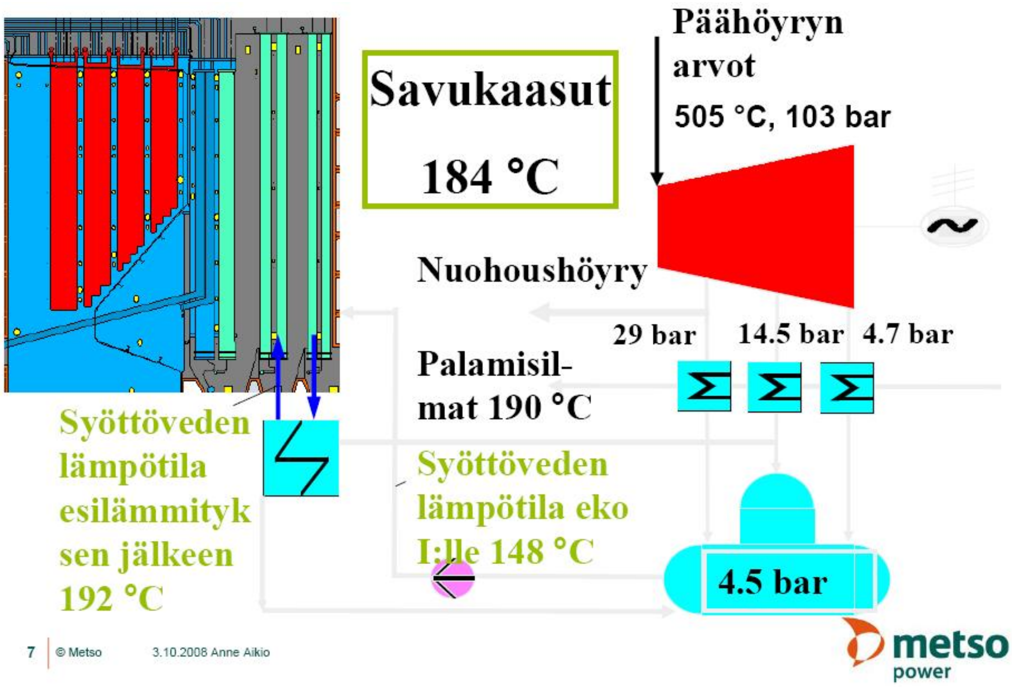


**Figure 3-3,** Some special features to increase electricity generating efficiency (Aikio, 2008).

### 3.3 Case C – High pressure and temperature recovery boiler

In 1998 the second generation high pressure and temperature recovery boiler started commercial operation in Japan (Arakawa et al., 2004). Figure 3-4 shows a second-generation high pressure and temperature recovery boilers. The second-generation recovery boiler is of single drum without evaporator design. Furnace outlet water or steam cooled screen tubes are not necessary with new 25%Cr special stainless steel for superheater.

Increasing superheating means that more heat transfer surface needs to be added. The higher main steam outlet temperature requires more heat to be added in the superheating section. Typically the furnace outlet gas temperature has increased. The alternative is to significantly increase superheating surface to decrease boiler bank inlet fluegas temperature. If boiler bank inlet gas temperature is reduced the average temperature difference between flue gas and steam is also decreased. This reduces heat transfer and substantially more superheating surface is needed. Low furnace outlet temperature design has been abandoned because of increased cost. With increasing dry solids content the furnace exit temperature can safely increase without fear of corrosion caused by carryover.

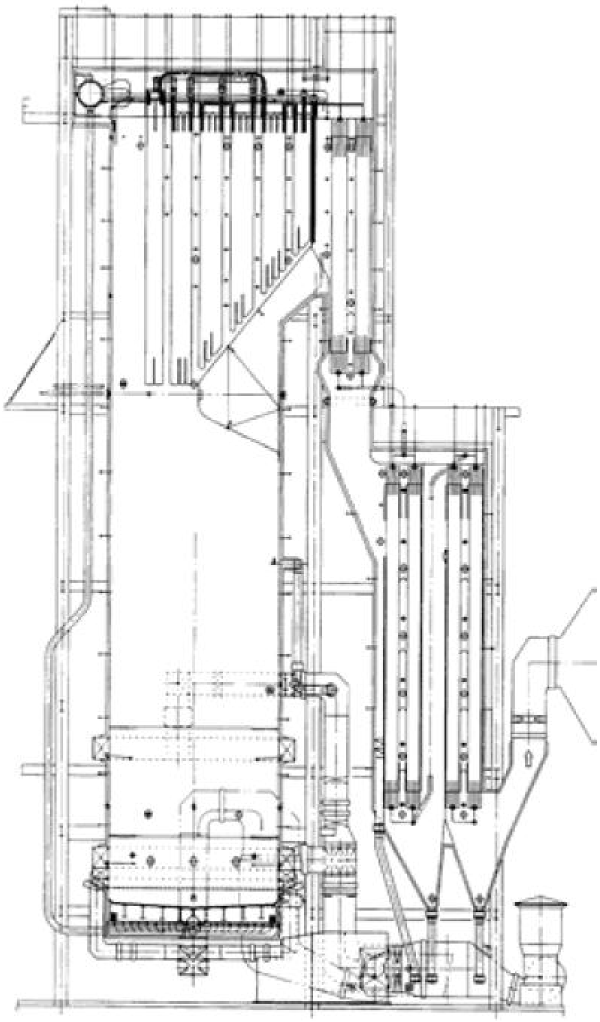

**Figure 3-4,** High pressure and temperature recovery boiler, Vendor Mitsubishi Heavy Industries, Ltd, Capacity 2500 t ds/24h, Black liquor ds 75 % Main steam 114 kg/s 109 bar(a) 515 °C**.**

### 3.4 Case D - Assisted circulation recovery boiler concept

Previously the target for higher electricity production has been to further increase temperature and pressure. previously the target concept has been maximum pressure with traditional design (Suomen Soodakattilayhdistys ry., 2007.)

The limit with natural circulation / assisted circulation design was chosen based on literature as 16 MPa (Steam 1992, Steam power engineering 1999). The target steam temperature was chosen as 540 $^oC$. Concept was however never detailedly studied.

## 3.5 Case E - Reheater concepts

From steam boiler literature we know that one way to increase electrical generation is applying reheating (Steam, 1992). That is after the steam has expanded partly in a turbine we take it back to the boiler and through another set of heat transfer surfaces called reheating surfaces, where the steam temperature is again raised. Reheating has been practiced in steam boilers for about 100 years.

### 3.5.1 Case E – High pressure and temperature recovery boiler with reheater concept

Babcock and Wilcox has recently promoted reheating recovery boilers (Hicks et al., 2009). One of the key features is the ability to use old turbine and just add new high pressure set with the new recovery boiler. This typically means rather high reheating pressures.

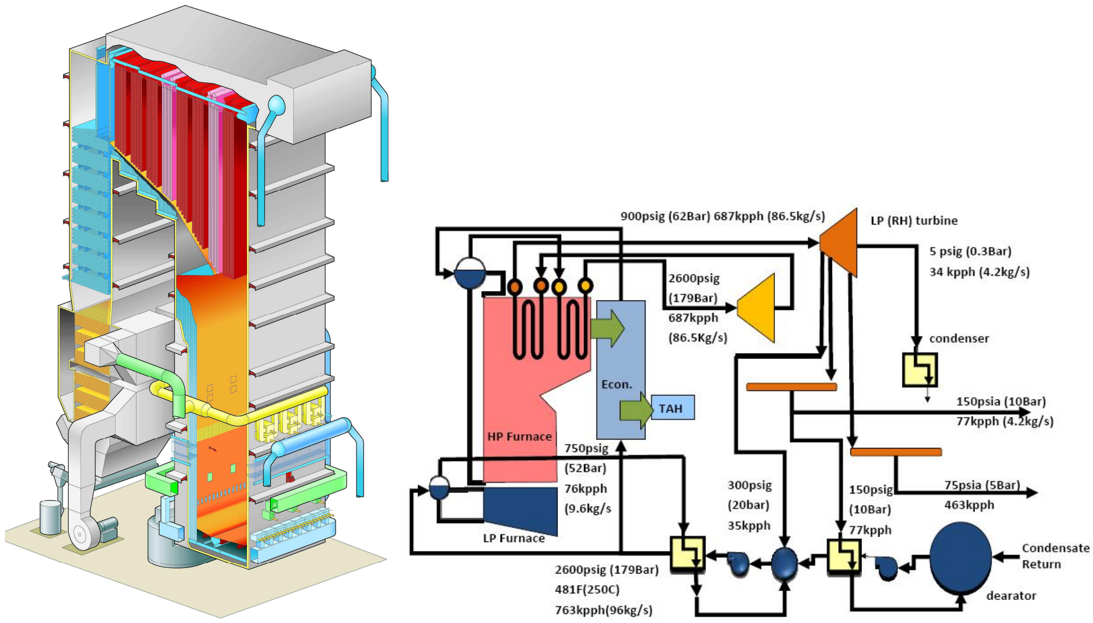


**Figure 3-5,** Reheater recovery boiler concept Vendor Babcock & Wilcox, Capacity 4500 t ds/24h, Black liquor ds 85 % Main steam 130 kg/s 179 bar(a) 510 °C reheated steam 62 bar(a) 443 °C.

One should also note the horizontal tube economizer. Running high dry solids i.e. practically no $SO_2$ creates non sticky conditions to economizer area.

### 3.5.2 First choice of reheat temperature

When choosing reheating pressure and temperature the requirement given (Chapter 1.1) was kept in mind. Choose main steam and reheated steam values so that for LP-steam desuperheating is maximum 30 $^{o}$C or minimum 2 % of moisture. In table 3-1 several different steam values are examined. The starting condition was chosen based on given instructions as 515$^{o}$C/12 MPa.

**Table 3-1** **4.5 bar(a) steam temperature and moisture with different reheats from 515 °C/12 MPa to different pressure and temperature combinations (bold values fullfill condition <30 °C desuperheat and < 2 % moisture).**

| | 480°C | 460°C | 440°C | 420°C | 400°C | 380°C | 360°C | 340°C | 320°C |
|---|---|---|---|---|---|---|---|---|---|
| 5.4 MPa | 182.7 | ***169.9*** | ***157.1*** | ***147.9*** | 147.9 | 147.9 | 147.9 | 147.9 | 147.9 |
| | 1.000 | ***1.000*** | ***1.000*** | ***0.994*** | 0.978 | 0.938 | 0.920 | 0.900 | 0.879 |
| 4.4 MPa | 202.4 | 189.3 | ***176.1*** | ***162.8*** | ***147.9*** | ***147.9*** | 147.9 | 147.9 | 147.9 |
| | 1.000 | 1.000 | ***1.000*** | ***1.000*** | ***1.000*** | ***0.984*** | 0.967 | 0.934 | 0.914 |
| 3.4 MPa | 228.2 | 214.5 | 200.8 | 187.0 | ***173.2*** | ***159.3*** | ***147.9*** | 147.9 | 147.9 |
| | 1.000 | 1.000 | 1.000 | 1.000 | ***1.000*** | ***1.000*** | ***0.995*** | 0.978 | 0.960 |
| 2.4 MPa | 264.8 | 250.3 | 235.8 | 221.2 | 206.6 | 192.0 | ***177.3*** | ***162.6*** | ***147.9*** |
| | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | ***1.000*** | ***1.000*** | ***0.998*** |
| 1.4 MPa | 326.8 | 310.8 | 294.8 | 278.7 | 262.6 | 246.5 | 230.4 | 214.3 | 198.1 |
| | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 |

Of the choices the combination of 3.6 MPa out and 3.4 MP in with reheating to 400 °C was chosen. This means that steam to be reheated can be used as sootblowing steam.

### 3.5.3 Comparison of chosen reheat temperature and pressure

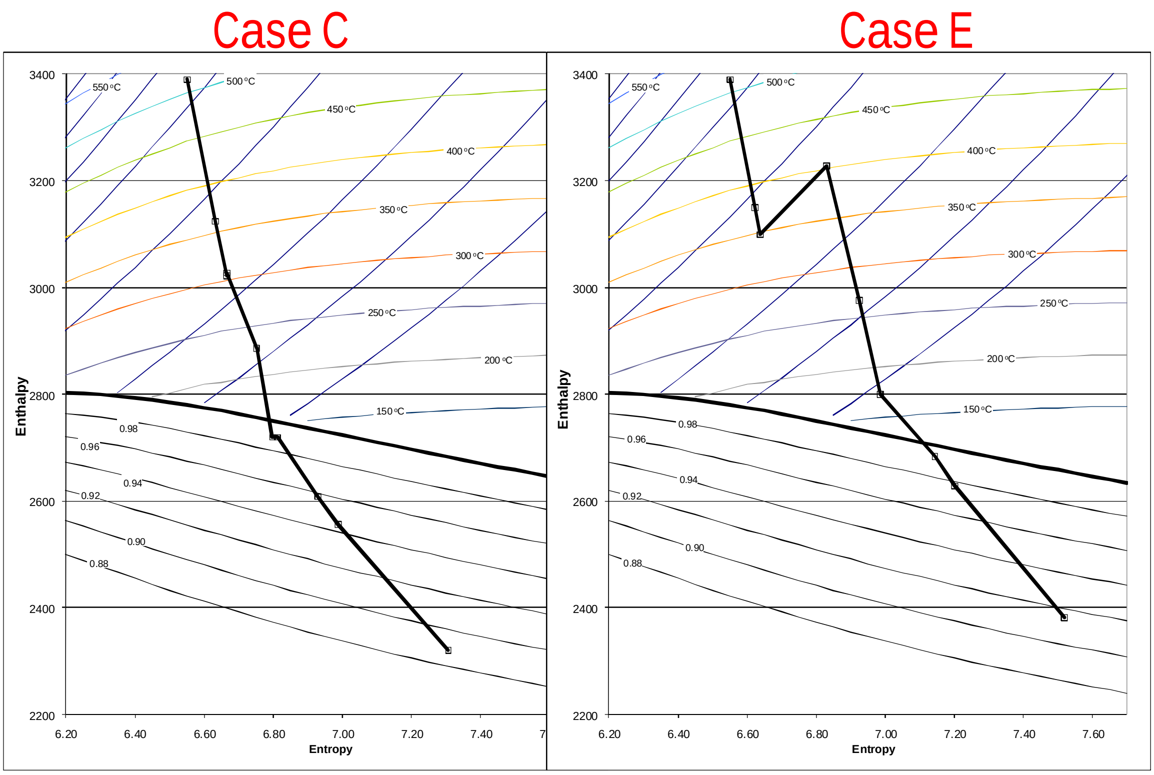


**Figure 3-6**, Comparison of case C and E

The chosen concept was calculated but the increase in electricity generation remained rather small. Therefore comparison with Case C was done.

Both start expansion at h=3389.2 kJ/kg at 514 $^{o}$C/118. After expansion to h=3098.9 kJ/kg at 348 $^{o}$C/34 reheater case E goes back for more heating and traditional case C continues expansion in the turbine. Case C expands to h=2726.1 kJ/kg at 149.5 $^{o}$C/4.5 while case E expands after reheat from h=3227.3 kJ/kg at 400 $^{o}$C/32 to h=2799.1 kJ/kg at 173 $^{o}$C/4.5.

In case C the turbine expands 3389.2-2726.1= 663.1 kJ/kg and in case E 3389.2-3098.9 + 3227.3-2799.1= 718.5 kJ/kg from main steam to low pressure steam. The difference in expansion is 55.4 kJ/kg. The additional expansion is only 55.4/(3389.2-2799.1) = 9.4 %. We can assume that condensing tail expansion enthalpy difference remains the same. If we reheat then each kg of steam expanding through turbine generates more electricity.

To generate reheated steam more heat per kg of steam is needed. In case C the heat addition is from 148 $^{o}$C = 623.5 kJ/kg to main steam = 3389.2 kJ/kg (=2765.7 kJ/kg). In case E more heat is added in reheat from 348 $^{o}$C = 3098.9 kJ/kg to 400 $^{o}$C = 3225.9 kJ/kg. So in Case C the heat addition is 3389.2 - 623.5 = 2765.7 kJ/kg and in Case E this is 2716.9 + (3227.3 - 3098.9) = 2894.1 kJ/kg. So in case C the heat that produces 1 kg of steam produces in case E only 0.956 kg steam with the same heat addition. The increase in electricity generation is therefore reduced to 0.956*718.5 = 686.6 kJ/kg or 3.4 %. If we take into account that the added sootblowing (+2 kg/s) reduces steam flow through the turbine and reheating causes poorer working of the condensing tail, the net benefit is reduced to almost zero.

### 3.5.4 Improving reheat from first premise

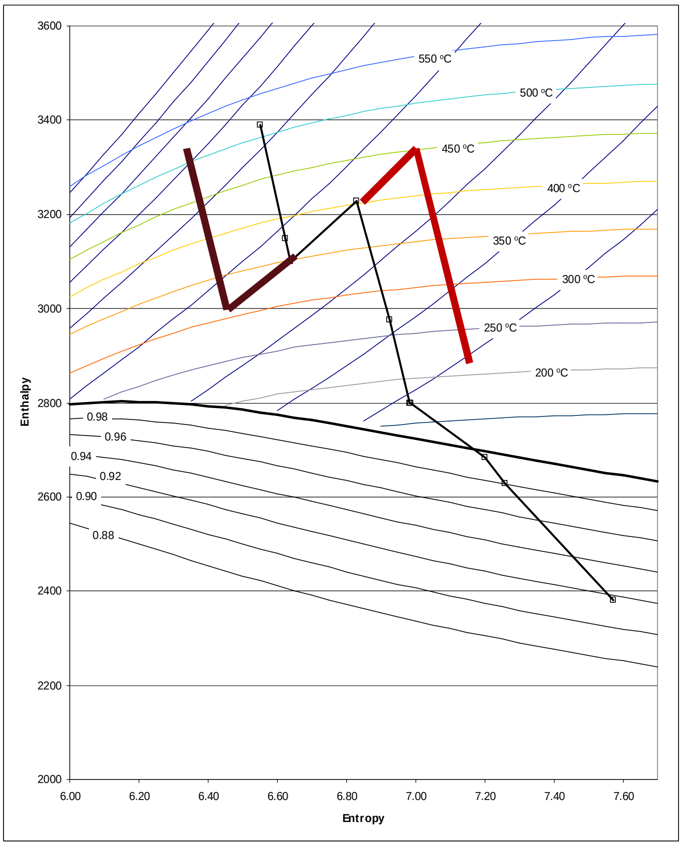


**Figure 3-7**, Options to improve reheat

One can increase reheating (the bright red line), but then the LP steam gets more and more superheated. A better alternative is to increase main steam pressure to e.g. 160 bar resulting in larger expansion, more heat addition and increased steam generation.

Therefore the alternative Case E with 160 bar furnace was calculated. The drawback is even more reheater surface.

### 3.6 Case F - Once-through recovery boiler with reheater concept

If we still want to increase the operating pressure we need to change to once-through boiler concept. Once-through boilers have been used since the 1930's and have been the main type for large utility boilers from the 1950's.

To improve cycle efficiency, utility boiler main steam pressures have increased to typically 248 bar. The steam temperatures have long remained conservative i.e. 540 $^{o}$C, but have recently started to climb close to 600 $^{o}$C.

Utility boilers use typically multiple feedwater heaters and have regenerative air heaters where flue gas heats air. These features add significantly to steam generation efficiency. To make comparison valid no additional heaters have been used for Case F. To choose main parameters again a comparison for end expansion conditions has been done, Table 3-2.

**Table 3-2 4.5 bar(a) steam temperature and moisture with different reheats from 540 $^{o}$C/26 MPa to different pressure and temperature combinations (bold values fullfill condition <30 $^{o}$C desuperheat and < 2 % moisture).**

| | 520$^{o}$C | 500$^{o}$C | 480$^{o}$C | 460$^{o}$C | 440$^{o}$C | 420$^{o}$C | 400$^{o}$C | 380$^{o}$C | 360$^{o}$C |
|---|---|---|---|---|---|---|---|---|---|
| 5.4 MPa | 208.2 | 195.4 | 182.7 | ***169.9*** | ***157.1*** | ***147.9*** | 147.9 | 147.9 | 147.9 |
| | 1.000 | 1.000 | 1.000 | ***1.000*** | ***1.000*** | ***0.994*** | 0.978 | 0.938 | 0.920 |
| 4.4 MPa | 228.7 | 215.6 | 202.4 | 189.3 | ***176.1*** | ***162.8*** | ***147.9*** | ***147.9*** | 147.9 |
| | 1.000 | 1.000 | 1.000 | 1.000 | ***1.000*** | ***1.000*** | ***1.000*** | ***0.984*** | 0.967 |
| 3.4 MPa | 255.5 | 241.8 | 228.2 | 214.5 | 200.8 | 187.0 | ***173.2*** | ***159.3*** | ***147.9*** |
| | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | ***1.000*** | ***1.000*** | ***0.995*** |
| 2.4 MPa | 293.8 | 279.3 | 264.8 | 250.3 | 235.8 | 221.2 | 206.6 | 192.0 | ***177.3*** |
| | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | ***1.000*** |
| 1.4 MPa | 358.9 | 342.9 | 326.8 | 310.8 | 294.8 | 278.7 | 262.6 | 246.5 | 230.4 |
| | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 |

Of the choices the combination of 5.6 MPa out and 5.4 MP in with reheating to 460 $^{o}$C was chosen. This means that the reheat lines remain smaller than with lower pressures. If a higher temperature was chosen (e.g. 540 $^{o}$C) then the expansion would be quite dry at around 5 bar.

# 4 WATER AND STEAM CIRCULATION

For a boiler to operate properly the steam water circulation must be designed for large variations of load, manageable temperature differences in parallel tubes and low possibility of tube inside erosion.

The main areas in steam-water side circulation design are choosing the right type of circulation, dimensioning downcomers and risers, dimensioning of superheater and dimensioning boiler banks.

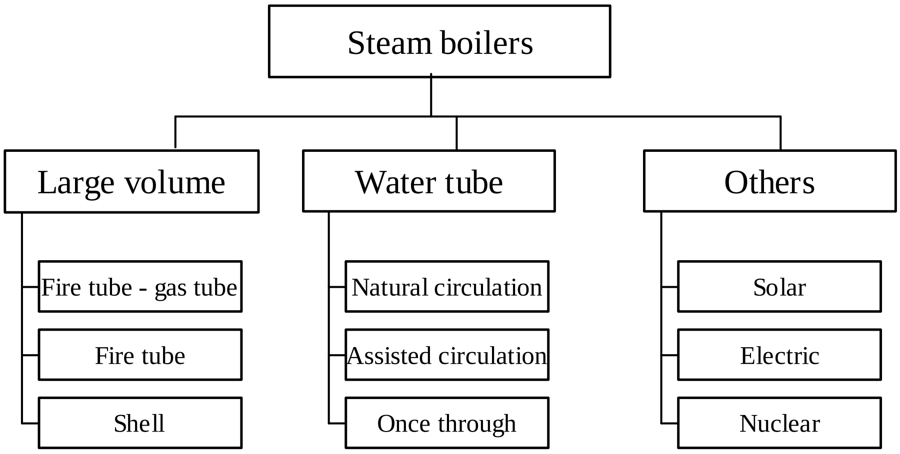


**Figure 4-1, Classification of steam water side**

There are two main groups of steam water circulation, Figure 4-1. The first group is the large volume boilers. In them the heating evaporates steam inside a large volume of water. For example a teakettle could be considered a large volume type.

Second group is the boilers where boiling occurs inside a tube filled initially with water. For example the coffee maker operates on this principle. Most of the large modern boilers belong to this group. In addition there is a group that could be called others of miscellaneous. They have features from both of these two groups. Many of the nuclear plant steam circuits as well as solar power circuits belong to this mixed group. All groups, in spite of the design are governed by the same laws.

## 4.1 Large volume boilers

Large volume boilers are usually fire tube boilers. Because of design they are limited by steam production (capacity) and operating pressure. The basic design has remained the same since the Scottish marine boilers of 1800.

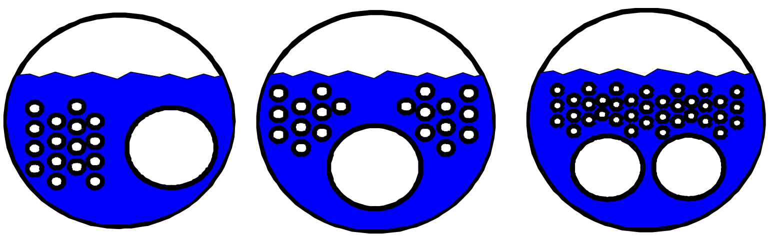

**Figure 4-2, Firetube – gas tube boiler designs (Effenberger, 2000).**

In large volume boilers water circulates downwards at the edges of the boilers, Figure 4-2. Steam bubbles rise, creating upward flow in the centerline of the boilers. Same mode of operation can be seen by watching a pot of water boiler on a stove.

## 4.2 Natural circulation boiler

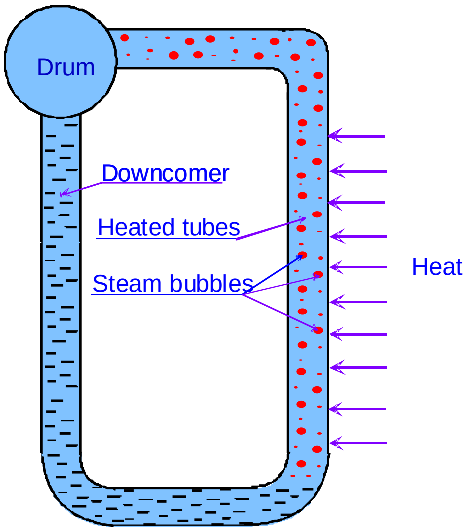


**Figure 4-3, Principle of natural circulation.**

Natural circulation is based on the density differences. The same principle can be seen in e.g. room in a cold winter day. Heated air rises on the wall where room is heated and subsequently cooled more dense air falls downwards on the opposite wall.

Natural circulation is caused by the density difference between saturated water and heated water partially filled with steam bubbles. In a natural circulation unit water tubes are connected to a loop, Figure 4-3. Heat is applied to one leg called raiser tubes, where water steam mixture flows upward. Denser saturated water flows downward in unheated leg called downcomer. Natural circulation means that the steam water movement in the evaporative tubes is achieved without use of external prime energy and mechanical means.

Driving force is static head difference between water in downcomers and steam/water mixture in furnace tubes.

$$\Delta P_{losses} = (\rho_{water} - \rho_{mixture})gh \qquad 4\text{-}1$$

where

$\Delta P_{losses}$ is the flow losses in the circulation
$\rho_{water}$ is the density of the water in the downcomers
$\rho_{mixture}$ is the density of the water steam mixture in the heated section
$h$ height of the circulation

Pressure increase decreases driving force. Natural circulation design is affected by several design factors. If we increase furnace height the driving head increases. We must try to avoid steam in downcomers. Efficient steam separators reduce the fraction of steam inside the drum. Inserting feedwater to drum at subsaturated state cools down and collapses remaining bubbles. Minimize axial flow inside drum helps creating equal flow in all parts of boilers. Natural circulation is also improved by higher heat flux in lower part of the tubes.

## 4.3 Assisted circulation

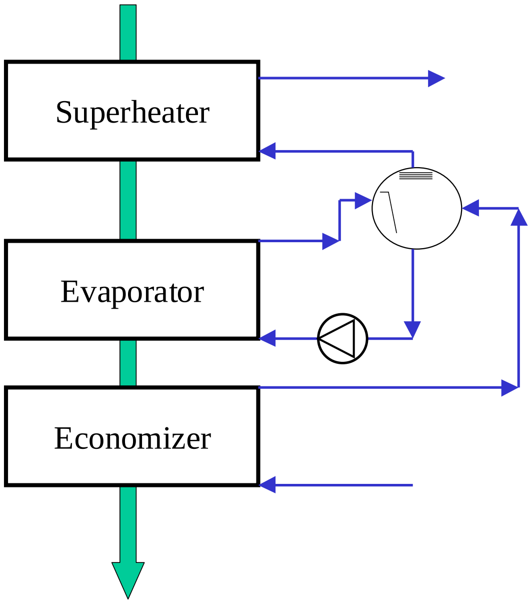


**Figure 4-4, Assisted circulation in a HRSG.**

Assisted circulation is typical in HRSG boilers and high pressure units. Water from the drum is pumped through evaporative surfaces. Synonyms for assisted circulation are forced circulation and controlled circulation.

### 4.3.1 Controlled circulation

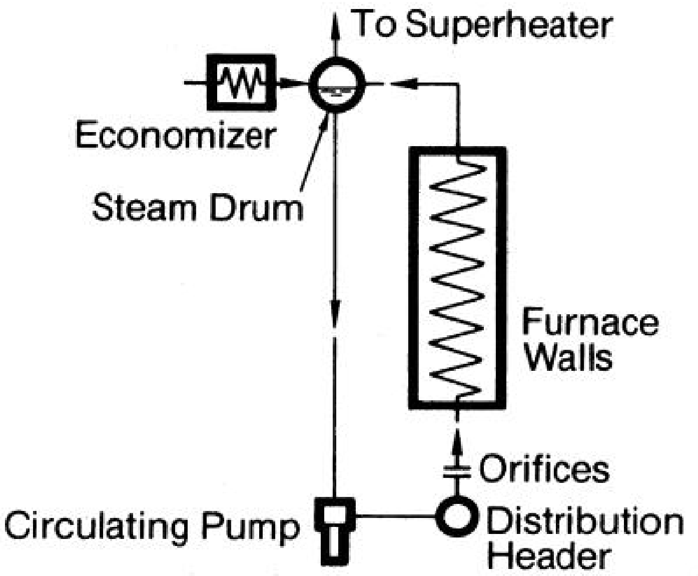


**Figure 4-5, Controlled circulation (Combustion, 1991).**

In controlled circulation a pump assists flow. Flow is regulated by orifices. This ensures even flow in all wall tubes. Controlled circulation is a trademark of ABB.

### 4.3.2 La Mont boiler

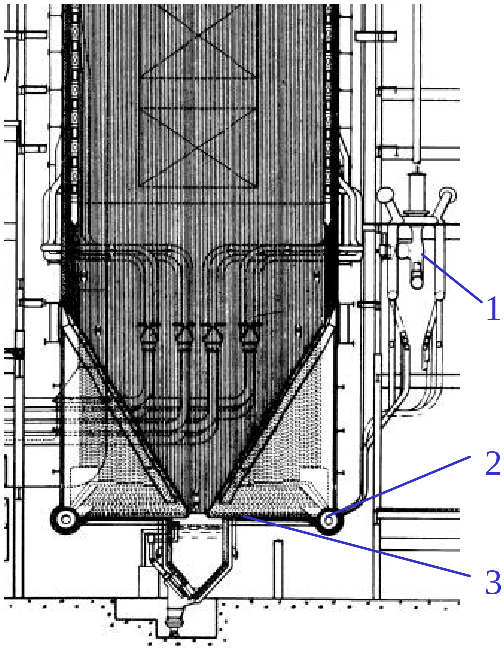


**Figure 4-6, La Mont boiler, 1 - HP circulating pump, 2 - header with orifices, 3 - individual wall tubes (Ledinegg, 1966).**

The most famous assisted circulation type is the La Mont boiler. The name comes from one manufacturer of these boilers. In La Mont boilers the drum pressure was usually below 19.0 MPa. The main advantage is that the designer can quite freely choose the tube pattern. Water flow through the tubes is controlled by appropriate sized orifices.

Circulation ratio in la Mont boilers is from 4 to 10. Pressure loss in a circuit was usually from 0.1 to 0.3 MPa.

## 4.4 Once-through boiler

In once-through boiler the water flows continuously through the boiler coming as 100 % steam at the main steam outlet. The circulation does not limit the pressure so boiler can be built at very high pressures. As positive circulation is kept up with the pressure difference the pressure loss through the boiler tends to be large. High pressure loss means high own power demand.

Feedwater purity must be very high as any contaminant tends to stay at the boiler walls. The feedwater must have about the same purity than the steam. The steam purity is dictated by the turbine requirements. Starting and shutting the boiler is problematic.

Because of pumps the mass flow densities in once through boilers are high, Table 4-1.

**Table 4-1, Mass flow densities.**

| Surface | Mass flow density |
|---|---|
| | kg/m$^2$s |
| Convective superheater | 1000 |
| Furnace tubes | 2000-3000 |
| Economizer | 600 |

The furnace tube arrangement is difficult. In natural and assisted circulation the furnace could be made of straight tubes of same temperature. Once-through boilers employ various tube patterns to cover the furnace walls. Based on the type of these circuits the once through boilers are divided into Sultzer Monotube boilers, Benson boilers and Ramzin boilers.

### 4.4.1 Sultzer Monotube boiler

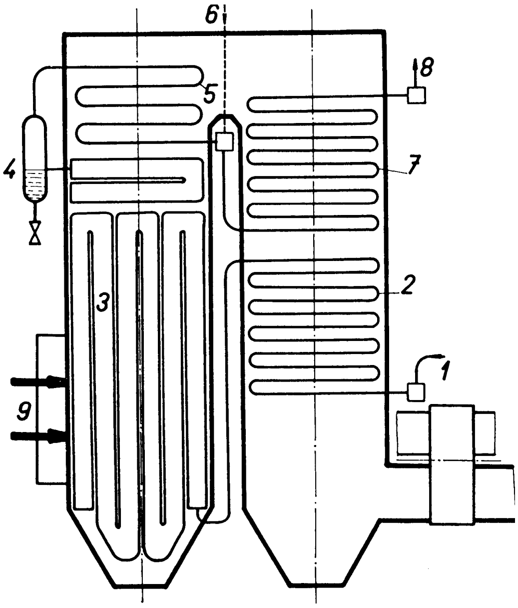


**Figure 4-7, Sultzer Monotube circulation, 1 – feedwater inlet, 2-economizing surface, 3-furnace tubes, 4-bottle, 5-superheating surface, 6-desuperheating spray, 7-superheating, 8-main steam out (Doležal, 1967).**

One of the first successful once through boilers was the Sultzer Monotube boiler. Its trademark is the use of a bottle, where remaining water droplets are separated from the steam. The flow in the furnace uses u-shaped tube bundles.

### 4.4.2 Benson-boiler

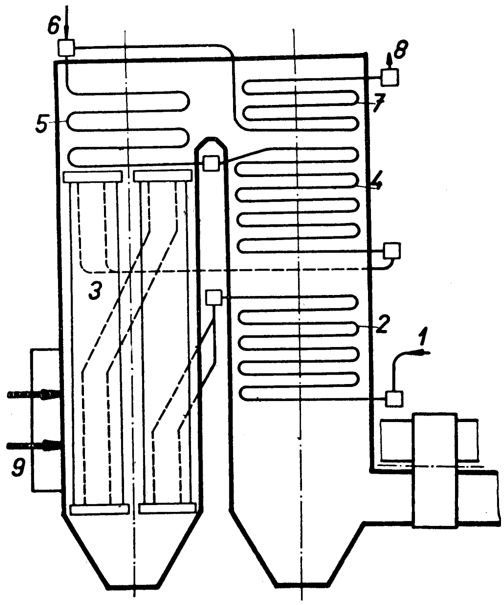


**Figure 4-8, Benson boiler circulation, 1 – feedwater inlet, 2-economizing surface, 3-furnace tubes, 4-backpass surface, 5-superheating surface, 6-desuperheating spray, 7-superheating, 8-main steam out (Doležal, 1967).**

A competitor to the Sultzer design the Benson boiler had a very similar tube pattern. Very rarely was the bottle or similar device used in a Benson boiler. To facilitate design Benson boiler uses straight heated up flowing parts and unheated down flow parts. In Benson boiler there is no definite point where the evaporation ends and the superheating begins, when operated under the critical pressure.

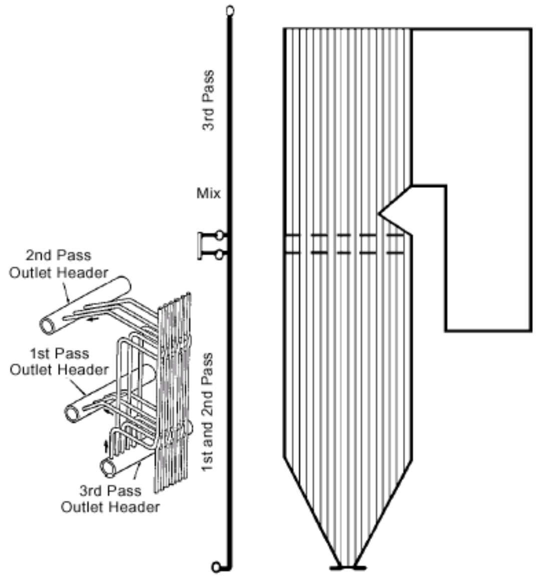


**Figure 4-9, Dividing wall passes to segments (Smith, 1998).**

Dividing each pass to several segments helps to maintain low temperature differences between adjacent tubes. Heat flux to tubes can vary because of localized fouling and tube placement.

### 4.4.3 Ramzin-boiler

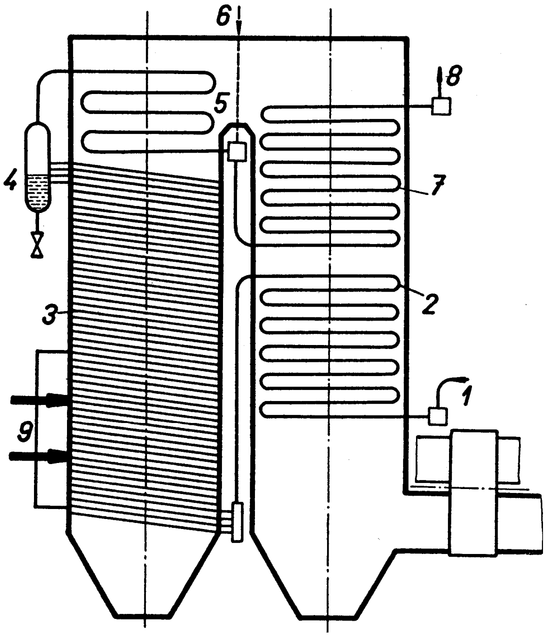


**Figure 4-10, Ramzin circulation, 1 – feedwater inlet, 2-economizing surface, 3-furnace tubes, 4-bottle, 5-superheating surface, 6-desuperheating spray, 7-superheating, 8-main steam out (Doležal, 1967).**

Ramzin boiler was developed in the Russia. It uses much the same operating principles that the Sultzer monotube boiler. The main distinctive feature of Ramzin boiler is that the tubes circle the furnace. It is expensive to manufacture. Separation (4) similar to Sultzer design, was added later. The main use of Ramzin boilers has been in former eastern block.

### 4.4.4 Tube selection for once through boiler

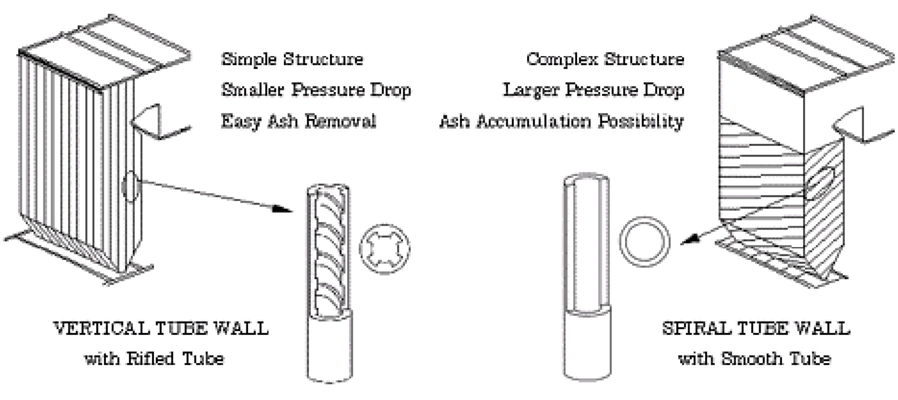


**Figure 4-11, Selection of tubes for once through boilers (Mitsubishi Heavy Industries Ltd, 2001).**

In once trough boilers the evaporation is brought to completion. Therefore parts of the tubes must be operated at very high vapor contents. This kind of operation stresses some tubes when water sometimes is separated from the tube walls and sometimes not. One helpful way is to make parts of the tubes from rifled or internally finned tubes. Water steam solution is brought into rotative motion and heavier water tends to stay longer at the tube walls.

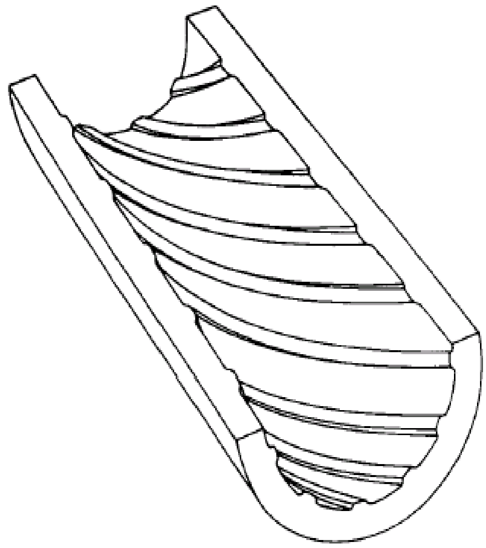

**Figure 4-12, Typical internally spirally finned tube.**

Spirally finned tube increases wall wetting, decreases the possibility of DNB and is more expensive than straight tube.

### 4.4.5 Combined circulation

Combined circulation is a trademark of former Combustion Engineering, now part of Ahlstom Power. Target in combined circulation is to improve ‘once-through’ in low loads. Includes pump for partial load operation. When low loads are run a part of the stam water solution can be recirculated to drum and pumped again through the boiler.

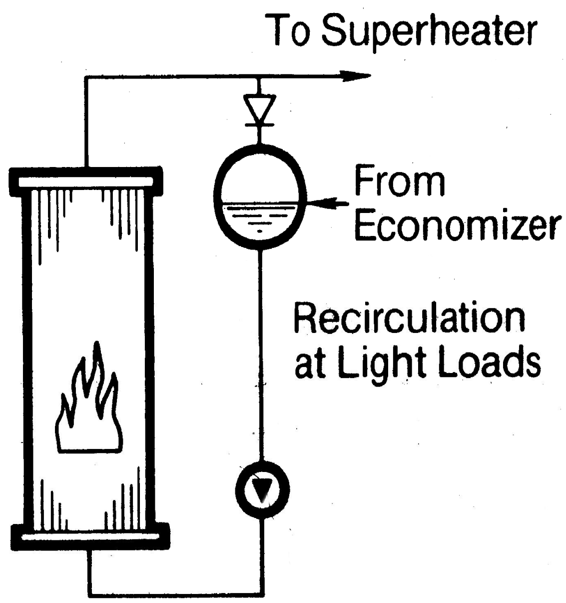


**Figure 4-13, Combined circulation (Singer, 1981).**

Headers were earlier of various shapes. Even square cross section was used. Nowadays only tubular headers are used.

### 4.4.6 Departure from nucleate boiling

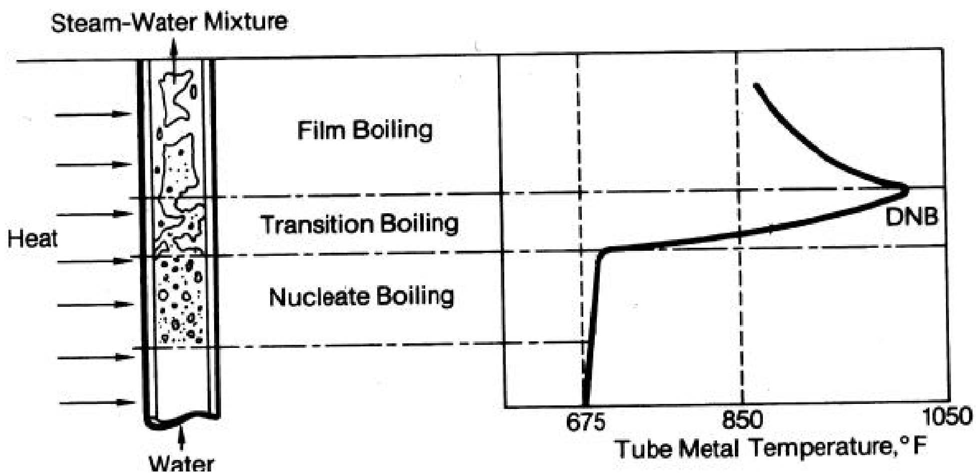


**Figure 4-14, Departure from nucleate boiling.**

The biggest design consern for once-through design is that DNB must be avoided. Furnace heat transfer must be arranged so that the furnace tube temperatures keep relatively constant even at partial load.

# 5 MILL BALANCES

Every recovery boiler case was considered in a mill setting. Two alternatives were used. The first is typical new mill with biomass boiler. The second was the same mill but with no additional boilers i.e. bark etc. are sold. This was done for not to confuse the electricity generating values with possible biomass boiler specific extra electricity generation.

## 5.1 Mill base dimensioning

| | |
|---|---|
| Target annual production, t/a | 1100000 |
| Softwood, ADt/a | 480000 |
| Hardwood, ADt/a | 620000 |
| Target operating days, d | 350 |
| Required average production, ADt/24h | 3143 |
| Mill debarking | |
| Additional wood for combustion, $m^3$sub/a | 500000 |

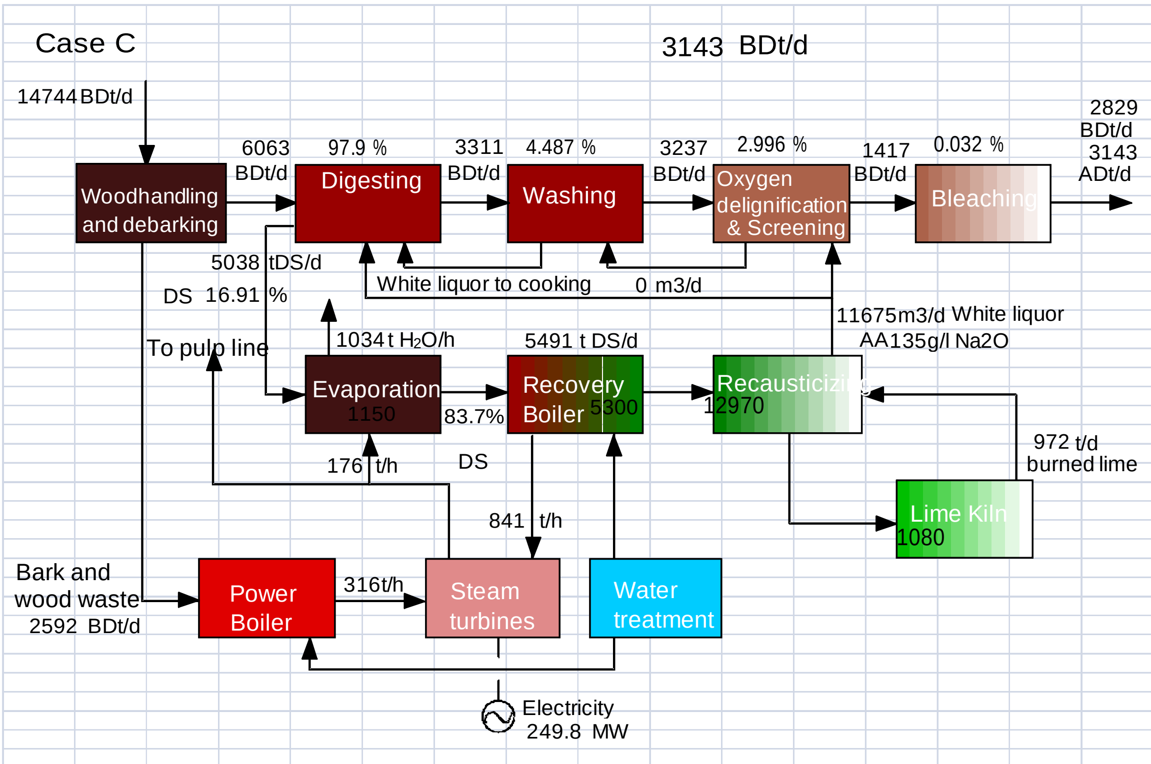


**Figure 5-1, Pulp mill overview.**

This mill corresponds to a new large mill at northern latitudes. The size was chosen large. The economical possibilities for extra electricity generation are better with large boilers.

### 5.2 Mill department sizing

| | |
|---|---|
| Woodroom, $m^3_{sub}$/d | 34300 |
| Digester, ADt/d | 3680 |
| Oxygendel., ADt/d | 3680 |
| Screening, ADt/d | 3680 |
| Pulp washing, ADt/d | 3680 |
| Bleaching, ADt/d | 3490 |
| Pulpdryer, ADt/d | 3490 |
| Evaporation, t $H_2O$/h | 1140 |
| Recovery boiler, tDS/d | 5300 |
| Causticizing, $m^3$WL/d | 20956 |
| Lime kiln, t lime/d | 1080 |
| CNCG-handling, $m^3$/h | 2300 |
| DNCG-handling, $m^3$/s | 40 |
| Power boiler, kg/s | 90 |
| Turbogenerators | Case |

The mill department sizing was done using LUT based mill balance program MILLFLOW.

### 5.3 Mill steam and electricity balances

Mill main balances were calculated to find out how changing the recovery boiler main steam parameters affect the kraft mill balances especially the recovery boiler electricity generation. The main balances were calculated using MILLFLOW excel spreadsheet as basis for balances.

When calculating the main balances the departmental steam usages correspond to needs for each case. So for each case the full mill steam and electricity balances are satisfied. The fiberline operation remains essentially unchanged as the mill production is kept constant. Recovery boiler preheating and pumping needs change from case to case. In addition in case A the evaporation is to lower solids. Main balance summary for recovery boiler is seen in Table 5-1.

**Table 5-1 Recovery boiler operating values for each case.**

| | | Joutseno | Kymi | Yonago | SoTu | SkyRec | SkyRec+ | SkyRec |
|---|---|---|---|---|---|---|---|---|
| **Case** | | **A** | **B** | **C** | **D** | **E** | **F** | **E160** |
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 82.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| HHV | MJ/kgds | 13.00 | 13.00 | 13.00 | 13.00 | 13.00 | 13.00 | 13.00 |
| LHV | MJ/kgds | 12.28 | 12.28 | 12.28 | 12.28 | 12.28 | 12.28 | 12.28 |
| $O_2$ in dry flue gas | % | 2.8 | 2.8 | 2.8 | 2.8 | 2.8 | 2.8 | 2.8 |
| Primary air percentage | % | 23.0 | 22.0 | 22.0 | 22.0 | 22.0 | 22.0 | 22.0 |
| Primary air temp. | °C | 150.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 |
| Secondary air percentage | % | 50.0 | 54.0 | 54.0 | 54.0 | 54.0 | 54.0 | 54.0 |
| Secondary air temp. | °C | 120.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 |
| Tertiary air percentage | % | 27.0 | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 |
| Tertiary air temp. | °C | 30.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 |
| Quartenary air percent. | % | 0.0 | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 |
| Quartenary air temp. | °C | 30.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 |
| Total air temperature | °C | 102.6 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 | 190.0 |
| Main steam pressure RB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 264.0 | 164.0 |
| Main steam temp. RB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 540.0 | 515.0 |
| Main steam pressure PB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 124.0 | 102.0 |
| Main steam temp. PB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 515.0 | 505.0 |
| Feedwater pressure | bar(a) | 110.0 | 121.0 | 146.0 | 182.0 | 146.0 | 290.0 | 290.0 |
| Feedwater temp. | °C | 120.0 | 148.0 | 148.0 | 148.0 | 148.0 | 148.0 | 148.0 |
| Reheater inlet pressure | bar(a) | | | | | 36 | 56 | 36 |
| Reheater inlet temp. | °C | | | | | 348 | 337 | 310 |
| Reheater outlet pressure | bar(a) | | | | | 34 | 54 | 34 |
| Reheater outlet temp. | °C | | | | | 400 | 460 | 400 |
| HP FW preheater inlet | °C | 200 | 200 | 200 | 200 | 200 | 200 | 200 |
| HP FWpreheater outlet | °C | 200 | 200 | 220 | 220 | 220 | 220 | 220 |
| Flue gas temp. (eco out) | °C | 155 | 197 | 197 | 197 | 197 | 197 | 197 |
| Flue gas temp. (to stack) | °C | | 155 | 155 | 155 | 155 | 155 | 155 |
| Sootblowing | kg/s | 6.0 | 6.0 | 6.0 | 6.0 | 8.0 | 8.0 | 8.0 |

The main balances were calculated using MILLFLOW excel spreadsheet as basis for balances. The mill condensing tail flow was set to balance the steam balance. The example pulp mill has condensing tail in its turbine. Most of the older pulp mills do not have a condensing tail, but run only backpressure operation. It should be noted that in the example mill there is no bark boiler. Recovery boiler is the sole steam producer in the mill. The bark is sold to outside of the mill.

In all cases the pulp mil produces more electricity than it consumes. This is because the mill is modern and thus energy efficient i.e. it has steam surplus. The second reason for high electricity generation is high steam parameters. The third reason is extra electricity generation done with condensing tail in the steam turbine.

**Table 5-2 Main balance values for each case, power boiler in operation.**

| | | Joutseno | Kymi | Yonago | SoTu | SkyRec | SkyRec+ | SkyRec |
|---|---|---|---|---|---|---|---|---|
| **Case** | | **A** | **B** | **C** | **D** | **E** | **F** | **E160** |
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 82.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| Main steam pressure RB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 264.0 | 164.0 |
| Main steam temp. RB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 540.0 | 515.0 |
| Main steam pressure PB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 124.0 | 102.0 |
| Main steam temp. PB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 515.0 | 505.0 |
| Feedwater pressure | bar(a) | 110.0 | 121.0 | 146.0 | 182.0 | 146.0 | 290.0 | 290.0 |
| Feedwater temp. | °C | 120.0 | 148.0 | 148.0 | 148.0 | 148.0 | 148.0 | 148.0 |
| Reheater inlet pressure | bar(a) | | | | | 36 | 56 | 36 |
| Reheater inlet temp. | °C | | | | | 348 | 337 | 310 |
| Reheater outlet pressure | bar(a) | | | | | 34 | 54 | 34 |
| Reheater outlet temp. | °C | | | | | 400 | 460 | 400 |
| | | | | | | | | |
| Sootblowing | kg/s | 6.0 | 6.0 | 6.0 | 6.0 | 8.0 | 8.0 | 8.0 |
| Steam flow RB | kg/s | 215.0 | 226.4 | 232.5 | 232.0 | 224.0 | 218.5 | 220.8 |
| Change in steam flow | % | 0.0 | 5.3 | 8.1 | 7.9 | 4.1 | 1.6 | 2.7 |
| Pulping usage total | MW | 87.6 | 88.1 | 88.9 | 90.2 | 88.2 | 93.2 | 90.0 |
| Mill total usage | MW | 95.5 | 96.1 | 96.9 | 98.2 | 96.2 | 101.2 | 98.0 |
| Electricity production | MW | 234.4 | 239.5 | 249.8 | 262.9 | 250.1 | 263.4 | 251.1 |
| Surplus electricity | MW | 138.9 | 143.3 | 153.0 | 164.7 | 153.8 | 162.2 | 153.1 |
| Efficiency to electricity | % | 23.2 | 23.1 | 24.1 | 25.4 | 24.1 | 25.4 | 24.2 |
| | | | | | | | | |
| Change in electricty | MW | 0.0 | 4.4 | 14.0 | 25.8 | 14.9 | 23.2 | 14.1 |
| | % | 0.0 | 3.2 | 10.1 | 18.5 | 10.7 | 16.7 | 10.2 |

We note that the steam production increases from Case A to Case B because of higher black liquor dry solids and more air preheating. The steam production increases from Case B to Case C because of high pressure preheating. The recovery boiler steam flow starts decreasing as further increases in main steam parameters require more heat.

The pulping electricity usage is not constant. The main parameter that changes is the recovery boiler feedwater pump power requirement, Table 5-3.

**Table 5-3 Recovery boiler own power usage for each case.**

| | | Joutseno | Kymi | Yonago | SoTu | SkyRec | SkyRec+ | SkyRec |
|---|---|---|---|---|---|---|---|---|
| **Case** | | **A** | **B** | **C** | **D** | **E** | **F** | **E160** |
| Air fan power | kW | 2275 | 2296 | 2296 | 2296 | 2296 | 2296 | 2296 |
| Flue gas fan power | kW | 2570 | 2534 | 2534 | 2534 | 2534 | 2534 | 2534 |
| Feedwater pumping power | kW | 3055 | 3556 | 4347 | 5719 | 4187 | 8606 | 5443 |
| Other power | kW | 1500 | 1500 | 1500 | 1500 | 1500 | 1500 | 1500 |
| Total power | kW | 9401 | 9886 | 10677 | 12049 | 10518 | 14937 | 11773 |

**Table 5-4 Main balance values for each case, no power boiler in operation.**

| | | Joutseno | Kymi | Yonago | SoTu | SkyRec | SkyRec+ | SkyRec |
|---|---|---|---|---|---|---|---|---|
| **Case** | | **A** | **B** | **C** | **D** | **E** | **F** | **E160** |
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 82.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| Main steam pressure RB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 264.0 | 164.0 |
| Main steam temp. RB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 540.0 | 515.0 |
| Main steam pressure PB | bar(a) | 91.0 | 104.0 | 124.0 | 164.0 | 124.0 | 124.0 | 102.0 |
| Main steam temp. PB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 515.0 | 505.0 |
| Feedwater pressure | bar(a) | 110.0 | 121.0 | 146.0 | 182.0 | 146.0 | 290.0 | 290.0 |
| Feedwater temp. | °C | 120.0 | 148.0 | 148.0 | 148.0 | 148.0 | 148.0 | 148.0 |
| Reheater inlet pressure | bar(a) | | | | | 36 | 56 | 36 |
| Reheater inlet temp. | °C | | | | | 348 | 337 | 310 |
| Reheater outlet pressure | bar(a) | | | | | 34 | 54 | 34 |
| Reheater outlet temp. | °C | | | | | 400 | 460 | 400 |
| | | | | | | | | |
| Sootblowing | kg/s | 6.00 | 6.00 | 6.00 | 6.00 | 8.00 | 8.00 | 8.00 |
| Steam flow RB | kg/s | 215.0 | 226.4 | 233.6 | 232.0 | 224.0 | 218.5 | 224.0 |
| Change in steam flow | % | 0.0 | 5.3 | 8.6 | 7.9 | 4.1 | 1.6 | 4.1 |
| Pulping usage total | MW | 87.6 | 88.1 | 88.9 | 90.2 | 88.2 | 93.2 | 90.0 |
| Mill total usage | MW | 91.1 | 91.7 | 92.4 | 93.8 | 91.8 | 96.8 | 93.6 |
| Electricity production | MW | 149.3 | 153.9 | 161.8 | 175.6 | 162.3 | 177.9 | 167.0 |
| Surplus electricity | MW | 58.2 | 62.2 | 69.4 | 81.7 | 70.5 | 81.1 | 73.4 |
| Efficiency to electricity | % | 20.4 | 20.4 | 21.4 | 23.2 | 21.5 | 23.5 | 22.7 |
| | | | | | | | | |
| Change in electricty | MW | 0.0 | 4.0 | 11.2 | 23.6 | 12.3 | 22.9 | 15.2 |
| | % | 0.0 | 6.8 | 19.3 | 40.5 | 21.1 | 39.4 | 26.2 |

As can be seen the modern recovery boiler Case C does produce about 20 % more electricity than roughly ten years ago, case A. reheating cases E and E160 seem to give only marginally better electricity production. The only alternative seems to be to increase the main steam temperature to 540 $^{o}$C, Cases D and F.

Electricity generation does not depend a lot on how the boiler steam side is configured. Reheating and once through appear only marginally better when considering the recovery boiler electricity generation.

## 6 BOILER DIMENSIONING

All described cases have been dimensioned. The dimensioning is based on typical recovery boiler dimensioning. The studied cases were

A. Natural circulation 82 %, 490 °C, 9.0 MPa (reference Joutseno)
B. Natural circulation 85 %, 505 °C, 10.2 MPa (reference Kymi)
C. Natural circulation 85 %, 515 °C, 12.0 MPa (reference Yonago)
D. Assisted circulation 85 %, 540 °C, 16.0 MPa (SoTu)
E. Natural circulation 85 %, 515/400 °C, 12.0/3.4 MPa (SkyRec) *Reheating*
F. Once-through 85 %, 540/460 °C, 26.0/5.4 MPa (Skyrec+) *Reheating*

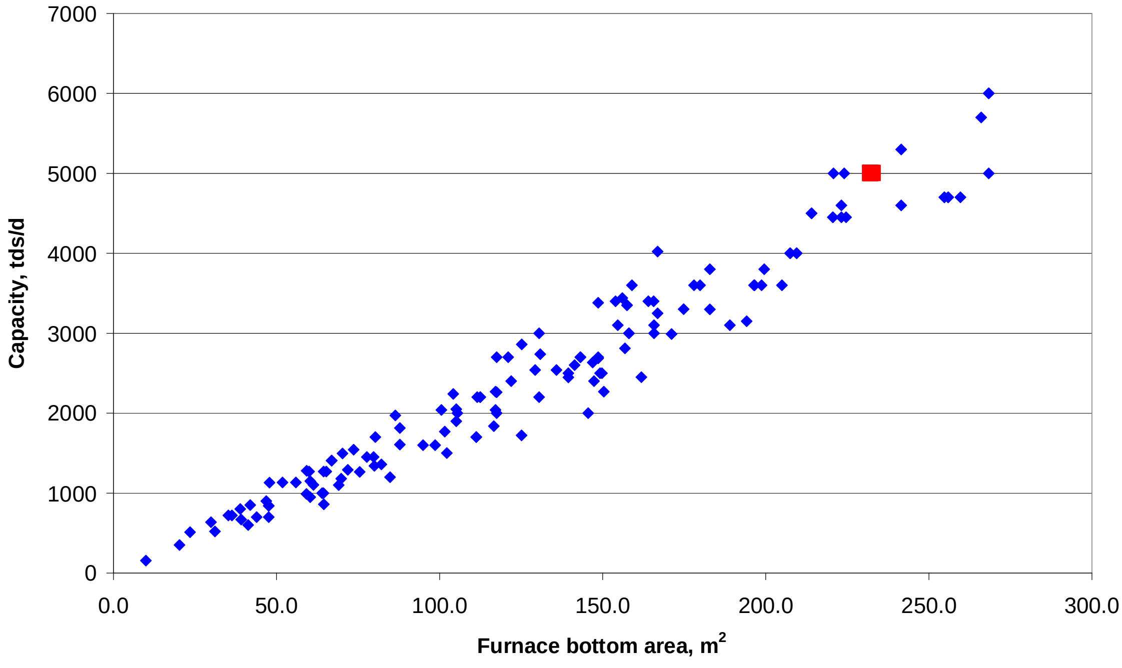


**Figure 6-1,** Boiler bottom area vs capacity, red squares indicate cases A-F.

Each boiler was dimensioned similar to commercial recovery boilers using recovery boiler specific dimensioning. The furnace bottom area size was constant for cases B to E and was 1.5 m$^2$ and 1.3 m$^2$ larger for cases A and F respectively. This was because of chosen furnace tube spacings. Furnace height varied, Figure 6-2. Furnace height for case A was 36.5 m and increased to 41.6 m to cases B and C. Furnace height then decreased to 39.6 m, 38.4 m and 33.6 m for cases D, E and F respectively. As can be seen the reheater cases E and F have rather low furnaces and thus high furnace nose temperatures. If we try to use traditional furnace nose temperatures the superheater surface area required quickly becomes quite large. Choice of variable furnace nose temperature meant that boiler bank entrance section temperature was kept reasonably unchanged.

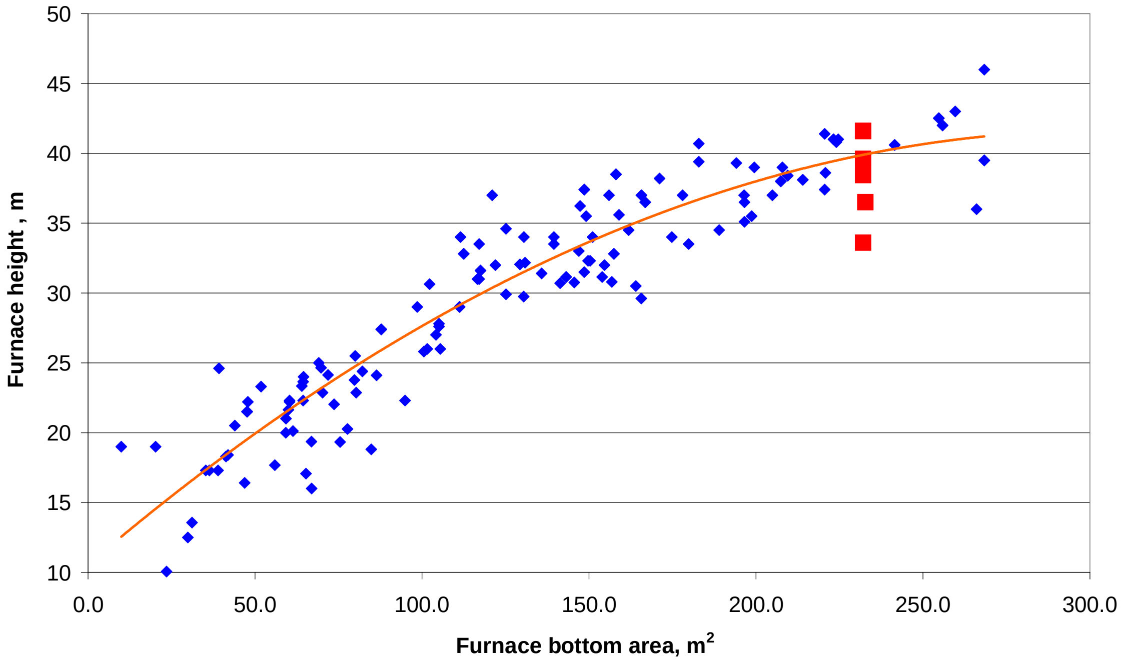


**Figure 6-2,** Boiler bottom area vs furnace height, red squares indicate cases A-F.

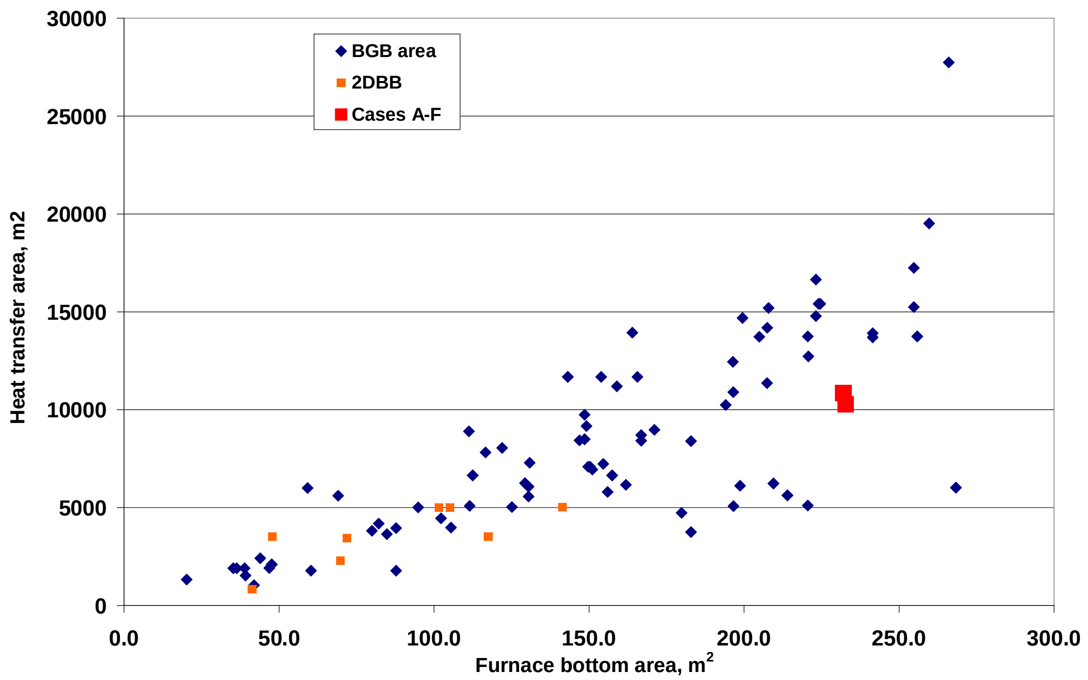


**Figure 6-3,** Boiler bottom area vs boiler bank area, red squares indicate cases A-F.

Because of equal boiler bank entrance section temperature the boiler bank area, figure 6-3 could be kept fairly equal. Similarly the economizer areas were fairly similar.

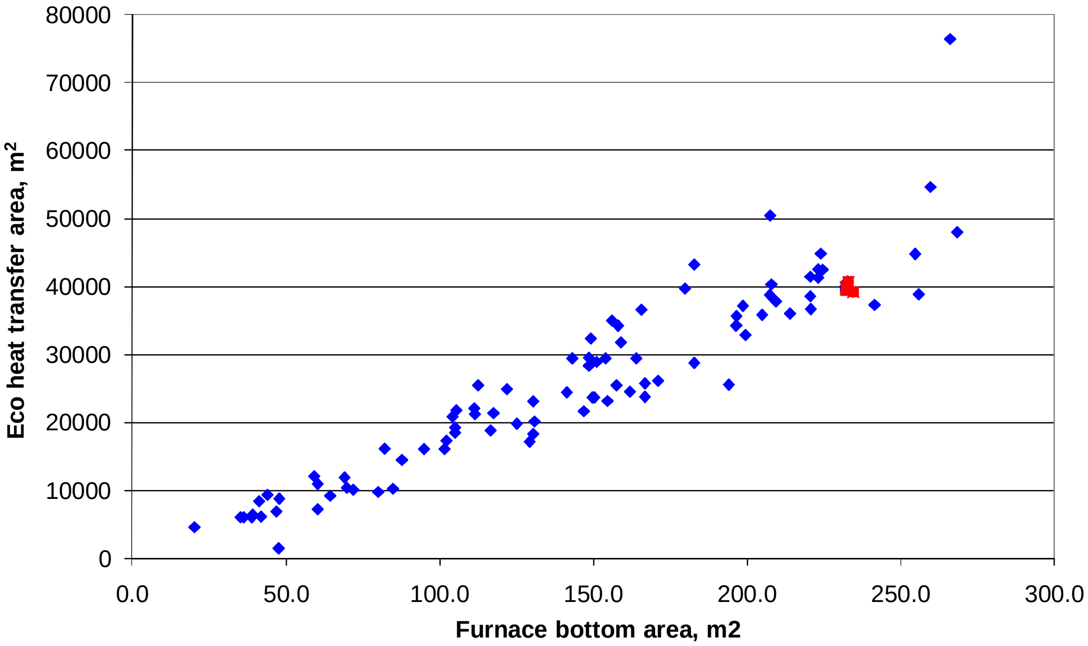


**Figure 6-4,** Boiler bottom area vs economizer area, red squares indicate cases A-F.

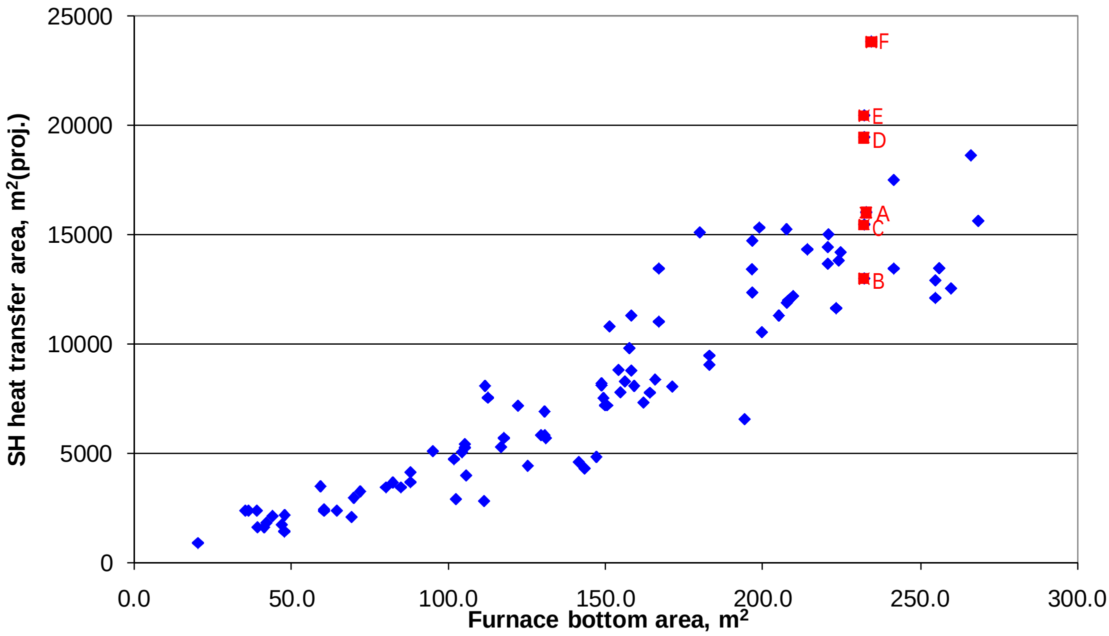


**Figure 6-5,** Boiler bottom area vs sum of superheater and reheater area, red squares indicate cases A-F.

The superheater and reheater area between different cases varied quite much. The area in case A could be smaller with different dimensioning. Otherwise the superheater area increases with higher main steam temperature and pressure. Each case dimensioning is discussed in more detail in the following sections.

## 6.1 Case A - Modern high efficiency boiler

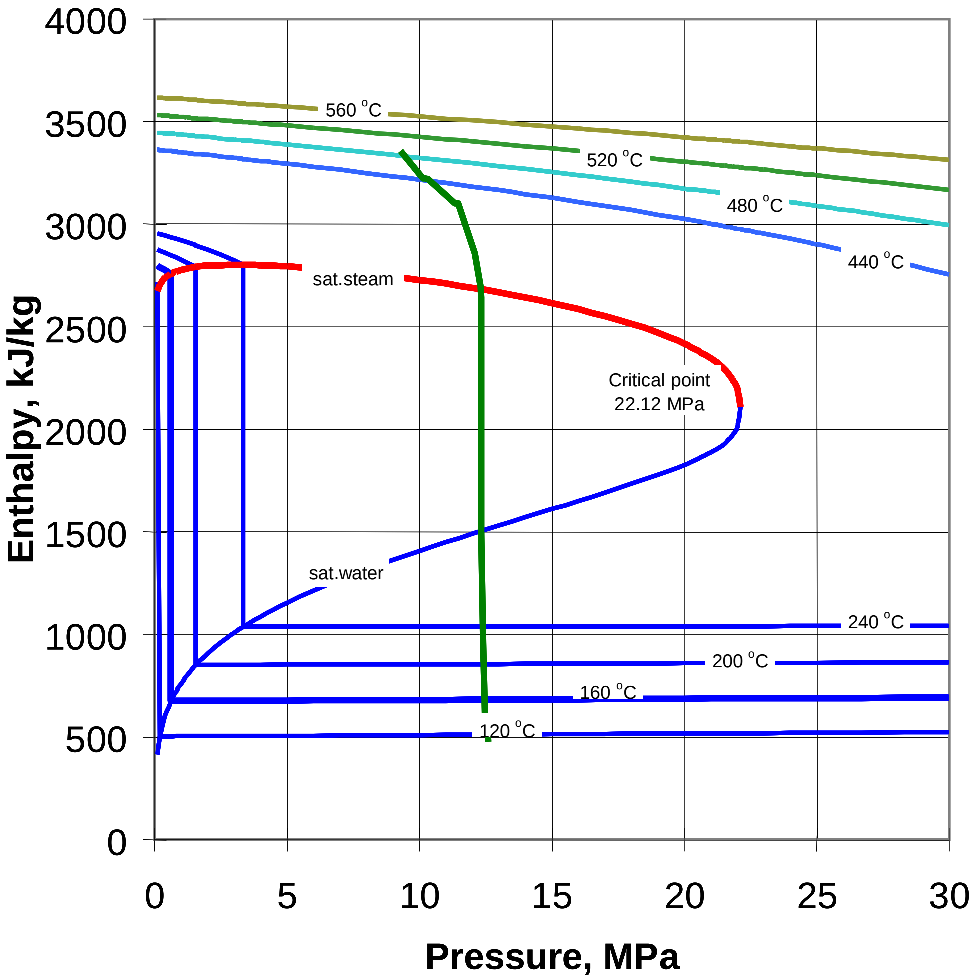


**Figure 6-6,** Modern recovery boiler in p-h-diagram, 215.0 kg/s, 9.0 MPa, 490 °C.

The heat addition diagram for Case A is presented as figure 6-1. As is typical the biggest pressure drop is in the superheaters. Because of low pressure there is fairly high requirement for economizing and evaporating.

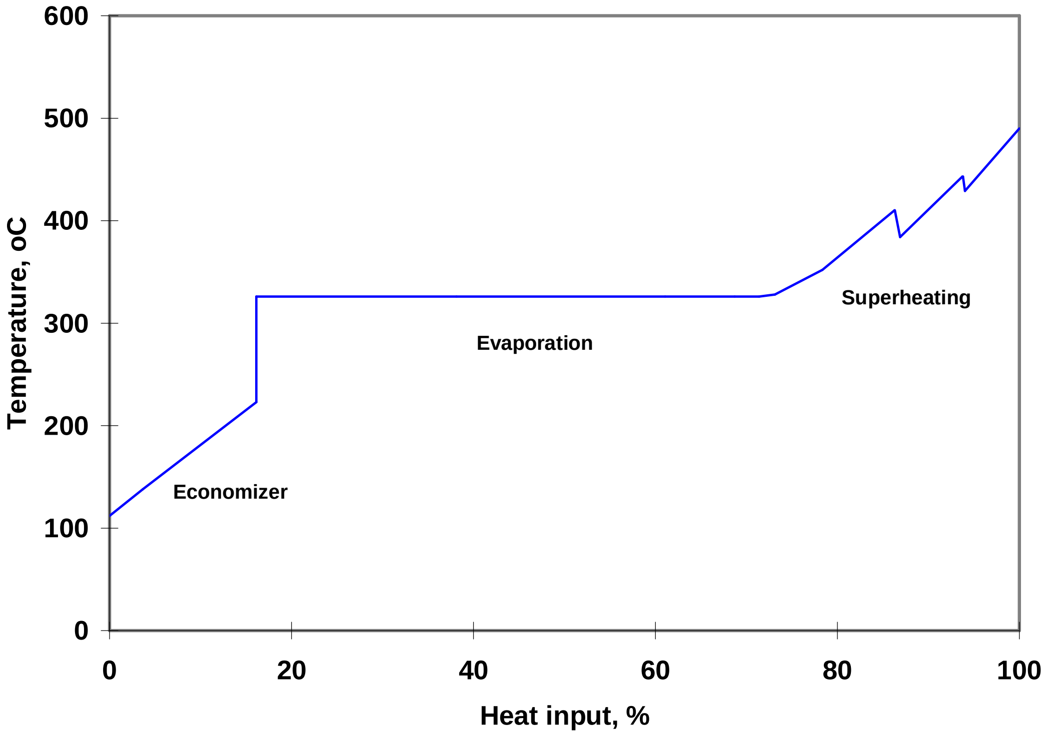


**Figure 6-7,** Modern recovery boiler in Φ-t-diagram, 215.0 kg/s, 9.0 MPa, 490 °C.

The heat-steam/waterside temperature diagram for Case A is presented as figure 6-2. The evaporation is overdimensioned as is typical and economizers are underdimensioned. This is because sizing boiler bank an economizers roughly of equal height gives savings in boiler house volume. The main dimensions for Case A are shown in Table 6-1.

**Table 6-1 Main dimensions of modern recovery boiler, Case A.**

| **Surface** | **unit** | |
|---|---|---|
| Furnace area | $m^2$ | 233.6 |
| Furnace height | m | 36.5 |
| Total boiler height | m | 75.5 |
| Furnace area | $m^2$ | 4417 |
| Screen area | $m^2$ | 0 |
| Superheater area | $m^2$ | 23712 |
| Superheater area (proj) | $m^2$ | 16004 |
| Boiler bank area | $m^2$ | 13600 |
| Economizer area | $m^2$ | 40720 |
| Total area | $m^2$ | 98454 |

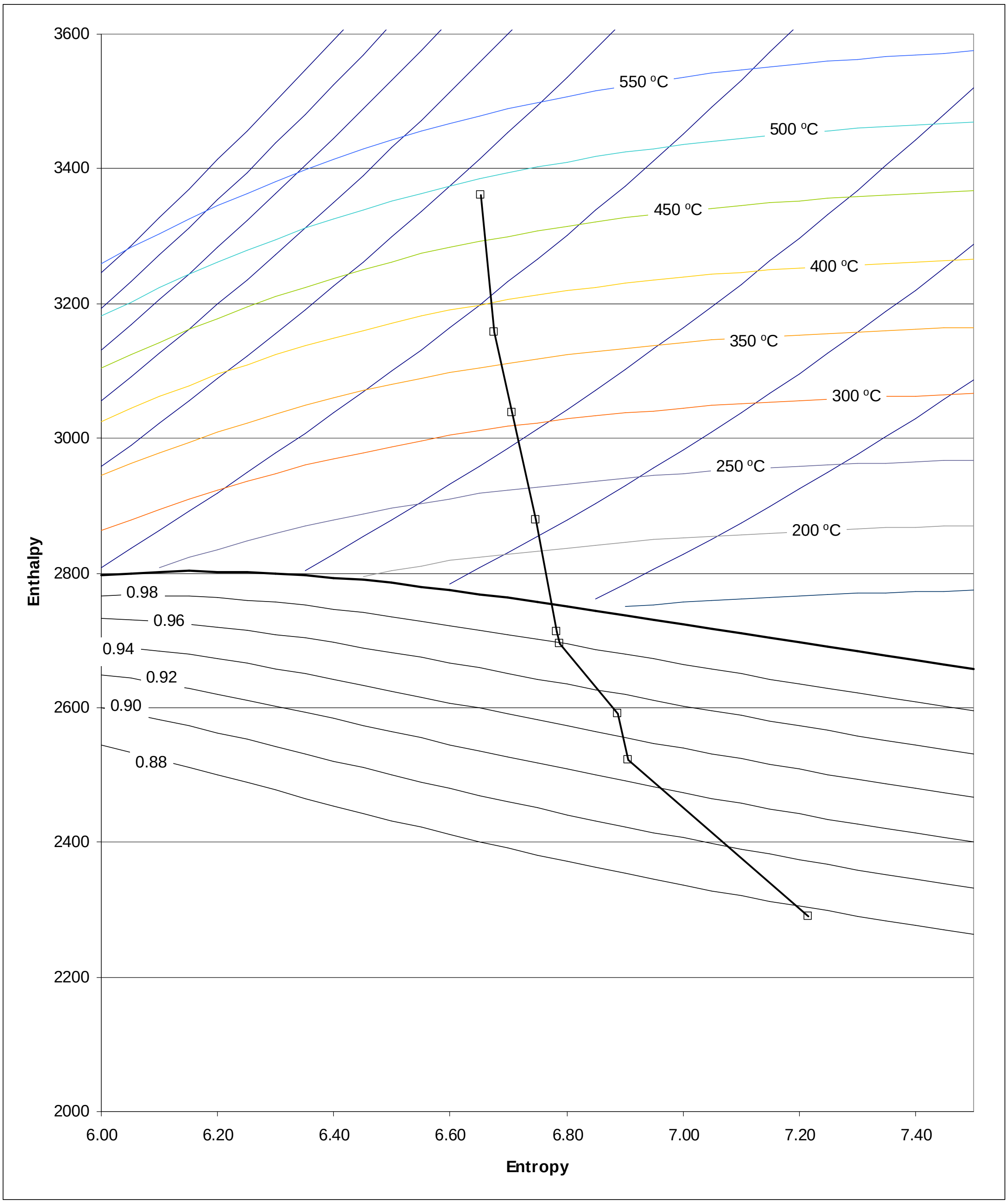


**Figure 6-8,** Modern recovery boiler turbine expansion in h-s-diagram, 215 kg/s, 9.0 MPa, 490 °C.

As seen in the figure 6-3 the turbine expansion tends to go to wet steam at about 5 bar(g) so this option requires higher low pressure steam values than others. In addition the turbine condenser end is at moisture which exceeds 12 %. Normally this would require special turbine design to be feasible. This temperature and pressure were used in the first recovery boiler turbines with condensing tail. If the turbine isentropic efficiency is lowered or turbine condenser end pressure is increased the turbine can be brought to more normal design conditions.

### 6.2 Case B – High efficiency recovery boiler

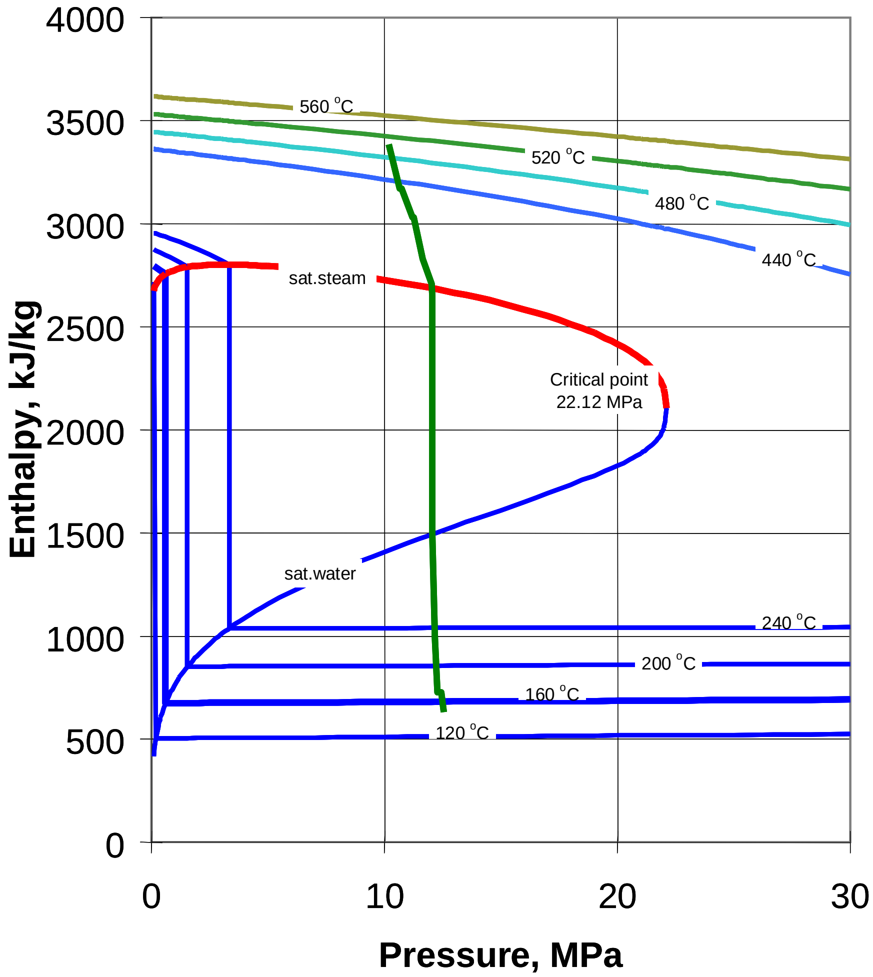


**Figure 6-9,** High efficiency recovery boiler in p-h-diagram, 226.4 kg/s, 10.2 MPa, 505 °C.

The heat addition diagram for Case B is presented as figure 6-9. This diagram does not markedly differ form the one for Case A. As can be seen the preheating of feedwater takes a fairly large portion of the heat flow. Increased economizer inlet temperature decreases economizing requirement.

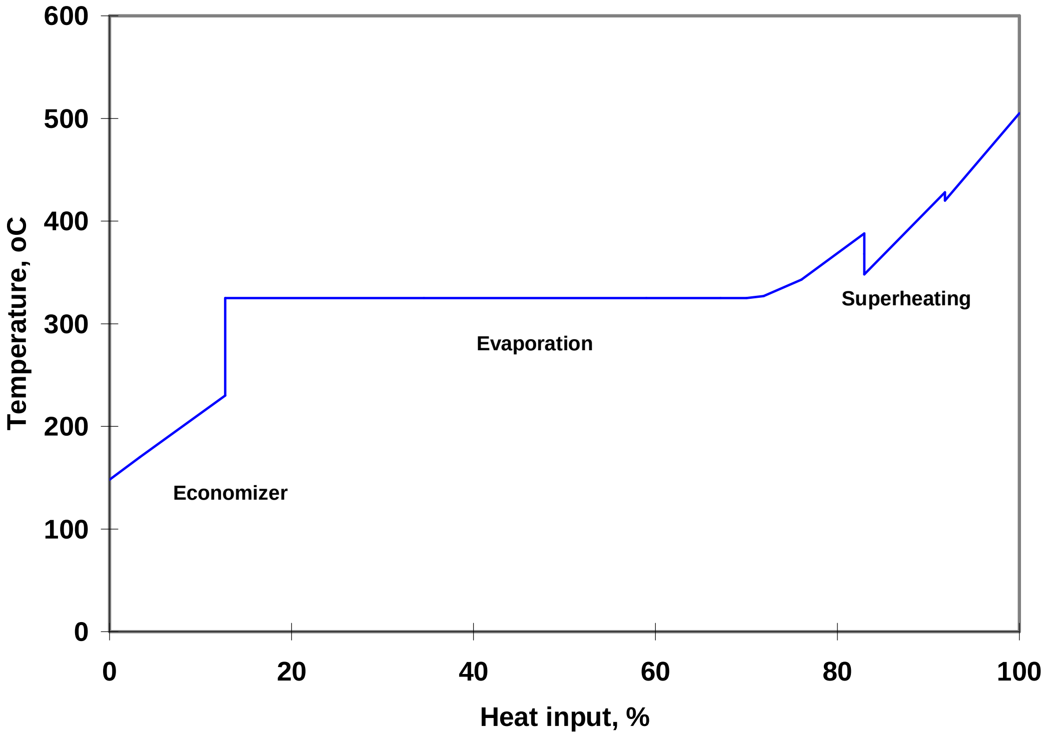


**Figure 6-10,** High efficiency recovery boiler in Φ-t-diagram, 226.4 kg/s, 10.2 MPa, 505 °C.

The heat-steam/waterside temperature diagram for Case B is presented as figure 6-10. The diagram is fairly similar to case A. The main dimensions for Case B are shown in Table 6-2.

**Table 6-2** **Main dimensions of high efficiency recovery boiler, Case B.**

| Surface | unit | |
|---|---|---|
| Furnace area | $m^2$ | 232.1 |
| Furnace height | m | 41.6 |
| Total boiler height | m | 75.1 |
| Furnace area | $m^2$ | 5056 |
| Screen area | $m^2$ | 796 |
| Superheater area | $m^2$ | 13536 |
| Superheater area (proj) | $m^2$ | 12984 |
| Boiler bank area | $m^2$ | 10262 |
| Economizer area | $m^2$ | 39676 |
| Total area | $m^2$ | 82310 |

The case B is dimensioned more modern than case A. Furnace is higher and superheater area has been brought to reasonable size. Because of this the total area required has decreased.

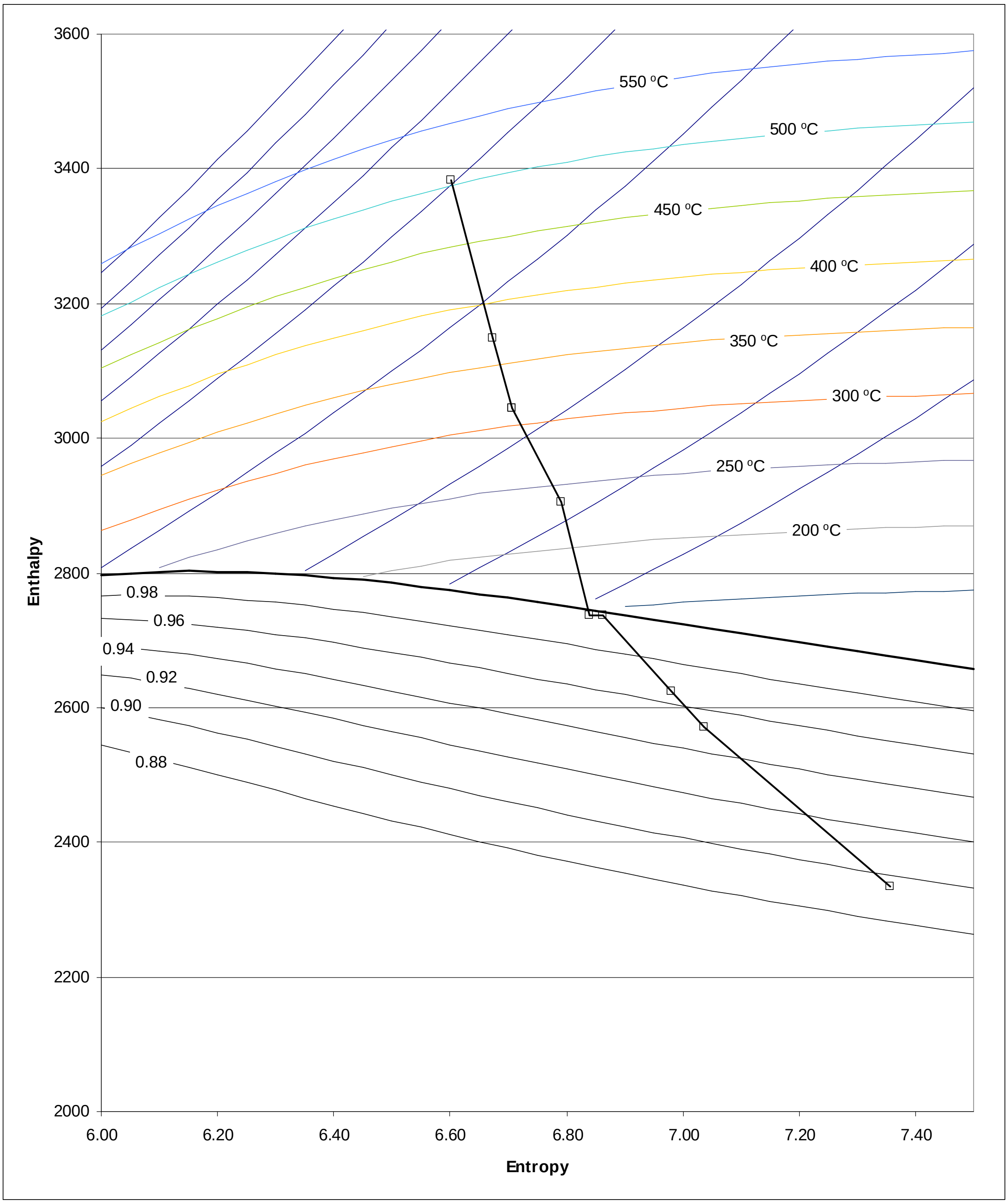


**Figure 6-11,** High efficiency recovery boiler turbine expansion in h-s-diagram, 226.4 kg/s, 10.2 MPa, 505 °C.

The high efficiency boiler turbine expansion seems to fit better the chosen LP pressure 3.5 bar(g). The end moisture is somewhat over 10 % which is fairly typical for condensing power turbines. Steam moisture of over 10 % requires better materials for the last rows of low pressure end turbine blades.

## 6.3 Case C – High pressure and temperature recovery boiler

The heat addition diagram for Case C is presented as figure 6-12. As is typical the biggest pressure drop is in the superheaters.

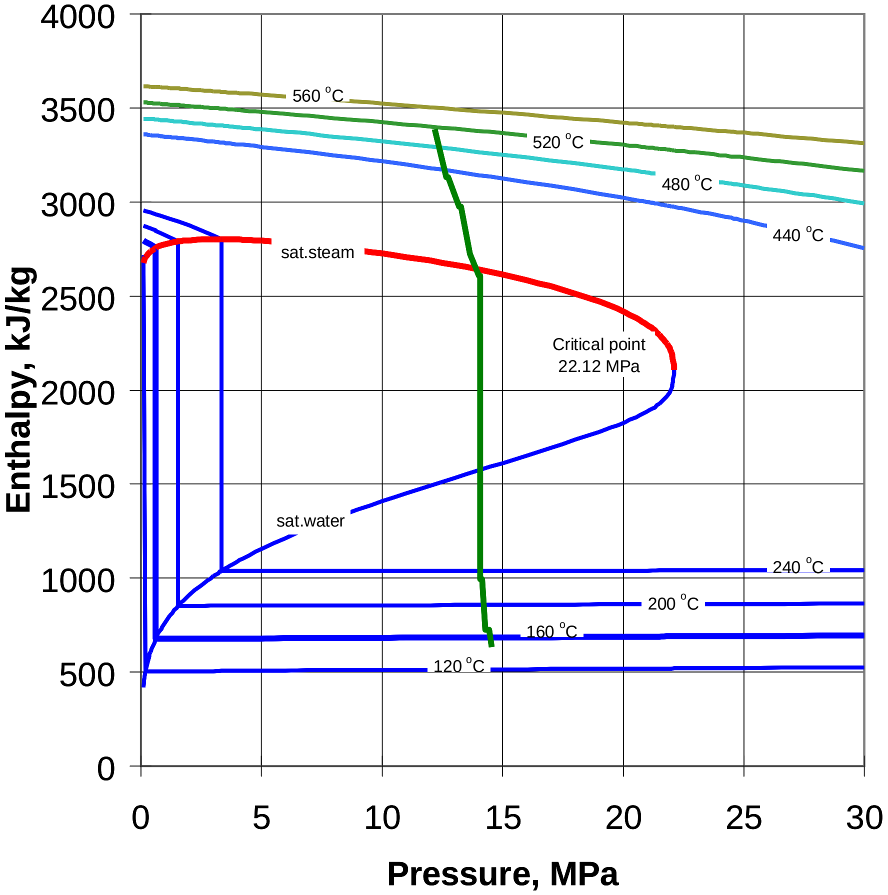


**Figure 6-12,** High pressure and temperature recovery boiler in p-h-diagram, 232.5 kg/s, 12.0 MPa, 520 °C.

There are no big changes in heat transfer surface placement nor in their sizing, therefore the p-h-diagram is fairly similar to cases A and B. As the pressure increases the portion of heat to evaporation decreases and the portion of heat to both superheating and preheating increases. This is the reason why different constructions from traditional are used in the boiler bank. Because the entrance to boiler bank has been kept fairly close to traditional values then the furnace inlet temperature has increased.

In the Φ-t-diagram one notices that the economizer outlet temperature starts to be fairly low compared to the evaporation temperature.

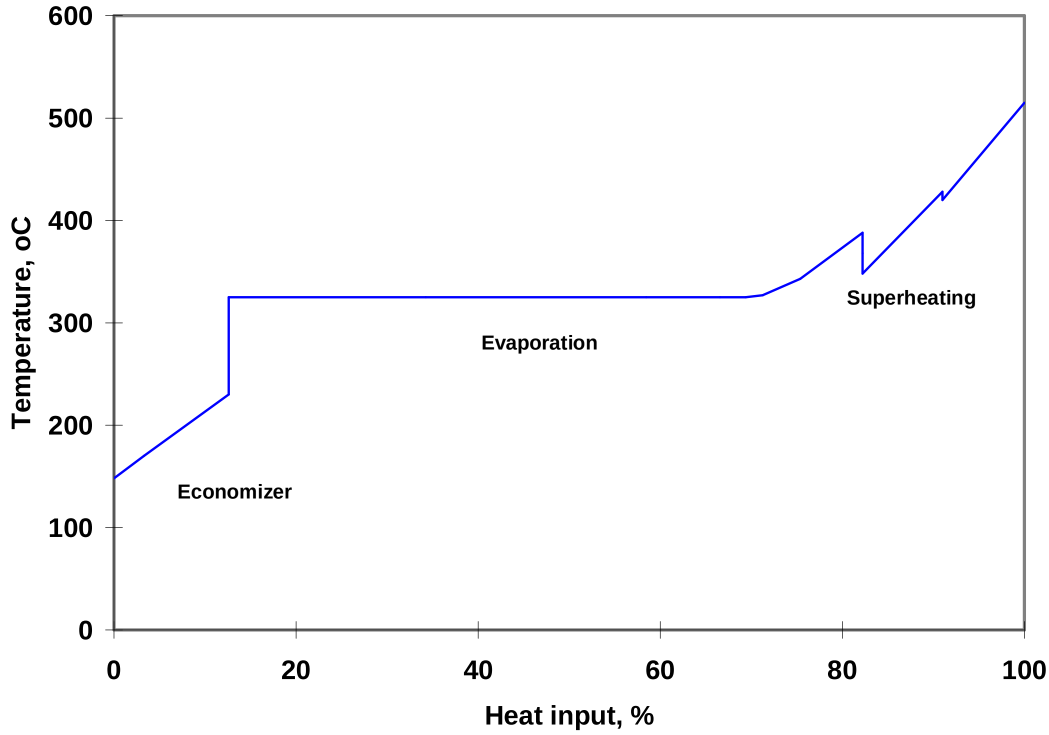


**Figure 6-13,** High pressure and temperature recovery boiler in Φ-t-diagram, 232.5 kg/s, 12.0 MPa, 520 °C.

Increasing pressure and temperature has required larger amount of heat used for superheating. This means that the superheating area has increased 19 %. This has also required more total boiler height meaning bigger boiler house. To maintain the end temperature more heat transfer surface area has been added to boiler bank area.

**Table 6-3 Main dimensions of high pressure and temperature recovery boiler, Case C.**

| **Surface** | **unit** | |
|---|---|---|
| Furnace area | $m^2$ | 232.1 |
| Furnace height | m | 41.6 |
| Total boiler height | m | 78.0 |
| Furnace area | $m^2$ | 5056 |
| Screen area | $m^2$ | 0 |
| Superheater area | $m^2$ | 16102 |
| Superheater area (proj) | $m^2$ | 15446 |
| Boiler bank area | $m^2$ | 11982 |
| Economizer area | $m^2$ | 39676 |
| Total area | $m^2$ | 88262 |

Superheater area has increased 19 % compared to case B because of the higher main steam temperature requirement. To keep the flue gas end temperature constant some more boiler bank surface has been added. This addition is about the equivalent of the screen area that was removed.

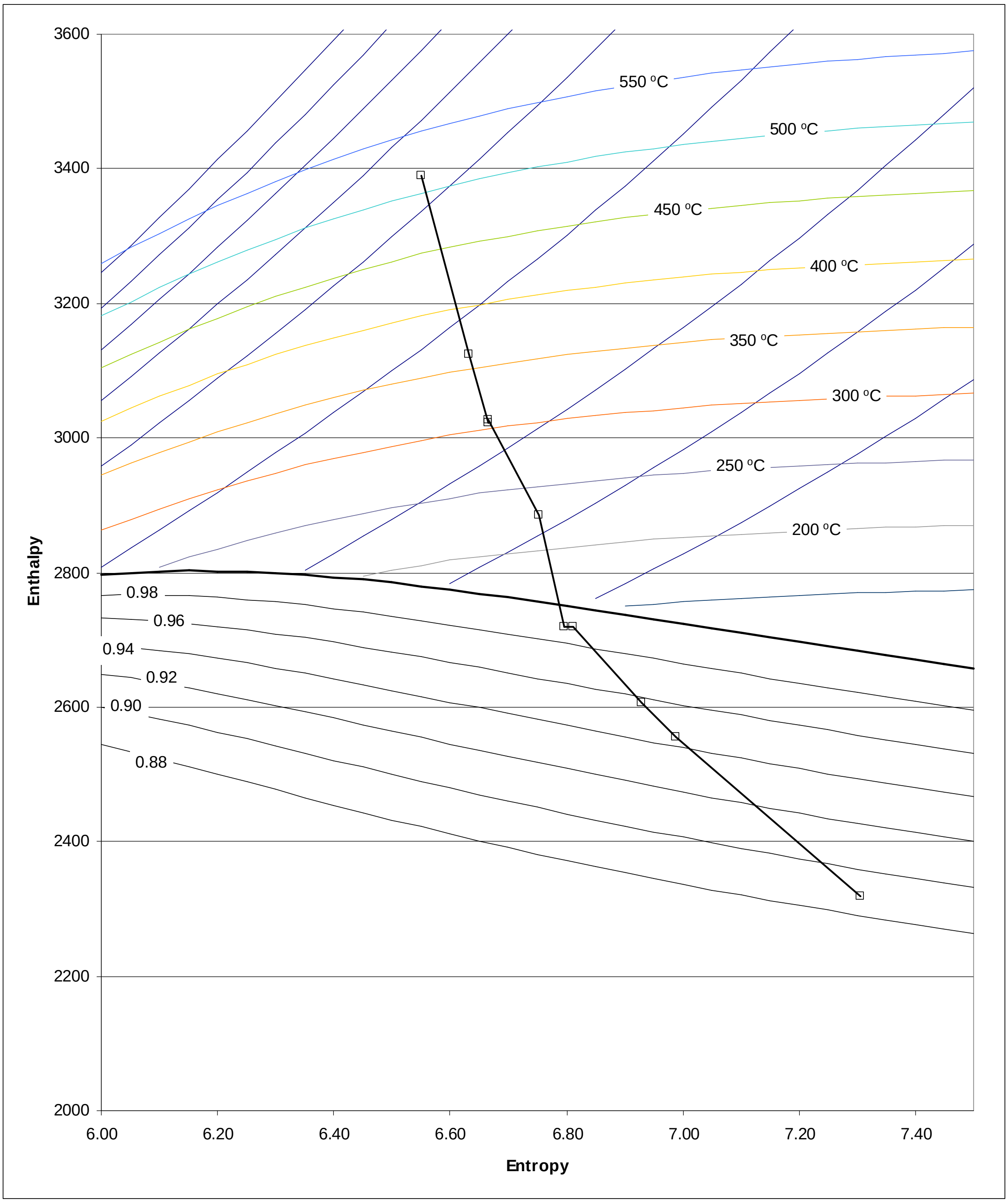


**Figure 6-14,** High pressure and temperature recovery boiler turbine expansion in h-s-diagram, 232.5 kg/s, 12.0 MPa, 520 °C.

High pressure and temperature recovery expansion goes clearly below saturation, Figure 6-14. This ensures high electrical output, but requires attention to be paid to condensate removal from low pressure lines.

### 6.4 Case D - Assisted circulation recovery boiler concept

The heat addition diagram for Case D is presented as figure 6-15. Increased pressure is decreasing the required evaporation and increasing the required evaporation.

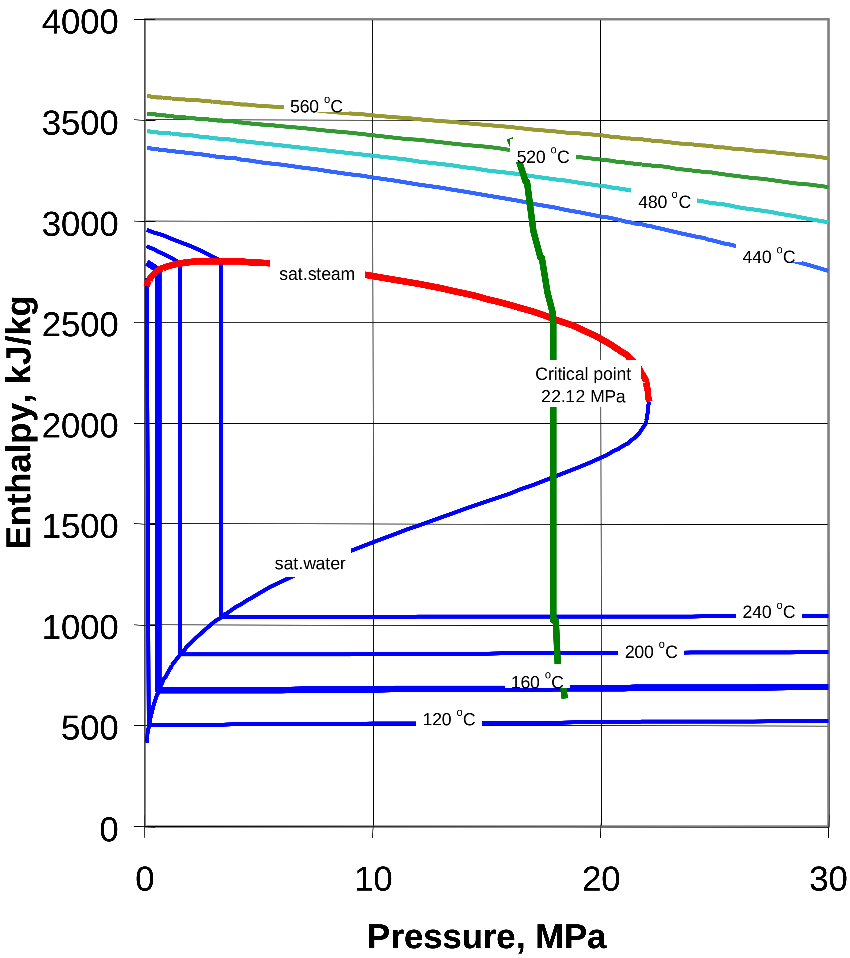


**Figure 6-15,** Assisted circulation recovery boiler in p-h-diagram, 232.0 kg/s, 16.0 MPa, 540 °C.

Because of lower evaporation requirement and addition of screen the furnace height and total boiler height are smaller. This saves in the boiler house costs. Higher pressure and temperature in main steam requires more superheating.

**Table 6-4 Main dimensions of assisted circulation recovery boiler, case D.**

| Surface | unit | |
|---|---|---|
| Furnace area | $m^2$ | 232.1 |
| Furnace height | m | 39.6 |
| Total boiler height | m | 76.0 |
| Furnace area | $m^2$ | 4813 |
| Screen area | $m^2$ | 796 |
| Superheater area | $m^2$ | 20255 |
| Superheater area (proj) | $m^2$ | 19429 |
| Boiler bank area | $m^2$ | 11982 |
| Economizer area | $m^2$ | 39529 |
| Total area | $m^2$ | 96804 |

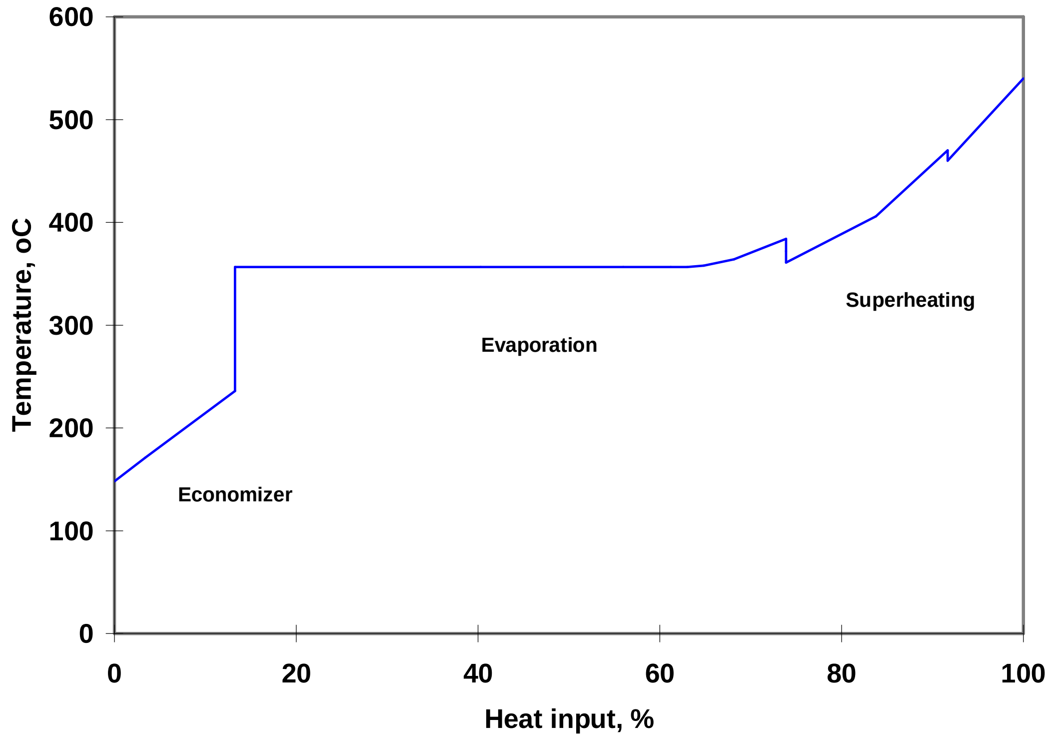


**Figure 6-16,** Assisted circulation recovery boiler in Φ-t-diagram, 232.0 kg/s, 16.0 MPa, 540 °C.

The heat-steam/waterside temperature diagram for Case D is presented as figure 6-16. One notices the large temperature difference between economizer outlet and drum saturated water temperatures. It could be possible to replace boiler bank with 3$^{rd}$ evaporator thus having a recovery boiler with no boiler bank.

Assisted circulation recovery expansion goes clearly below saturation, Figure 6-17. This is not acceptable and requires either changes to the turbine construction or higher low pressure values to be used in the mill. Also condensate end goes over 12 % moisture which would require much higher cost of turbine. An alternative approach would be to buy a turbine with lower efficiency and thus get the values to normal level. This would destroy any benefit achieved.

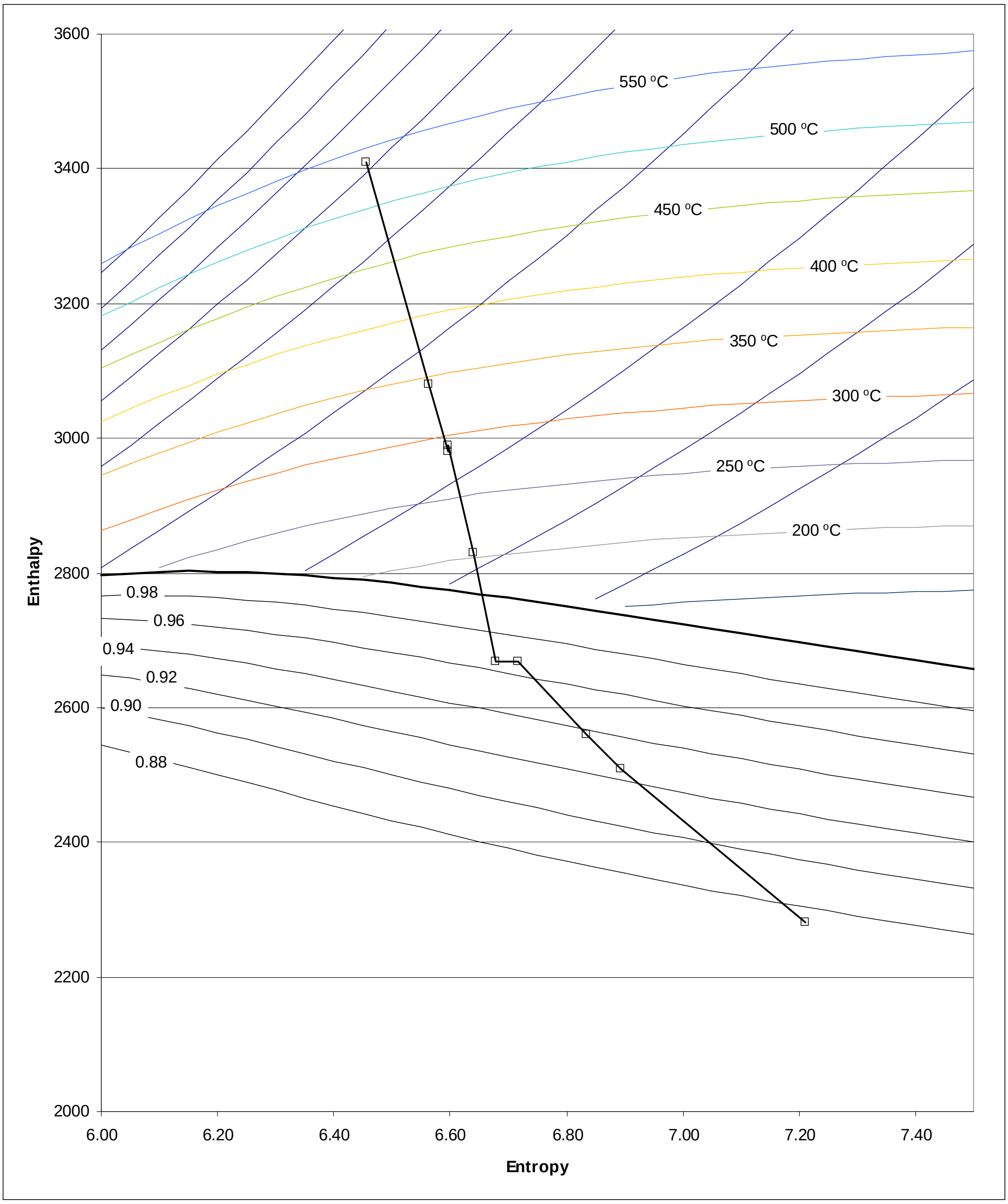


**Figure 6-17**, Assisted circulation recovery boiler turbine expansion in h-s-diagram, 232.0 kg/s, 16.0 MPa, 540 °C.

## 6.5 Case E – High pressure and temperature recovery boiler with reheat concept

This boiler equals the boiler of Case C but with reheating applied. Reheating can be seen in Figure 6-18 as additional superheating requirement, but at lower pressure.

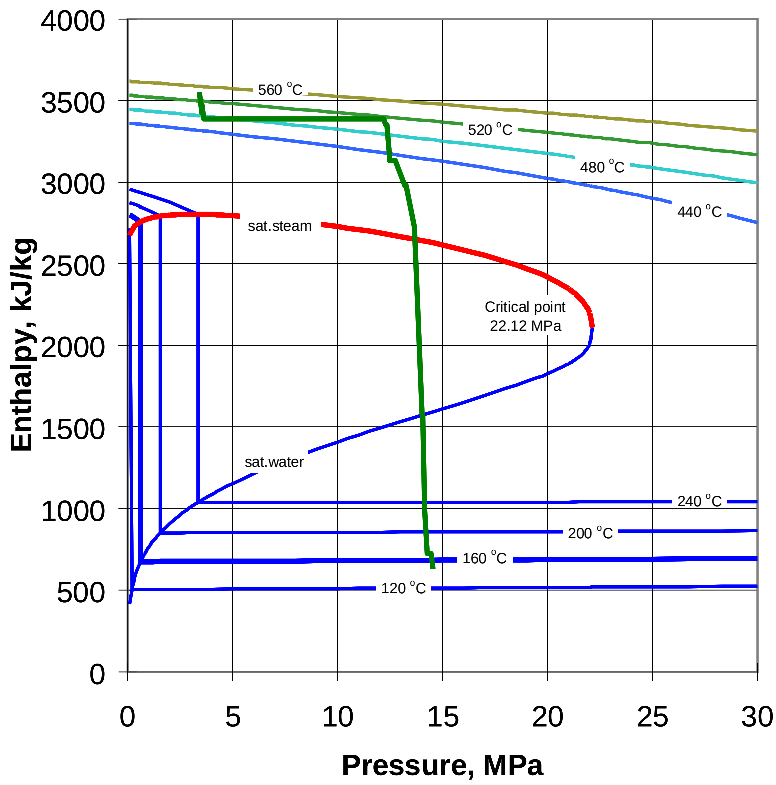


**Figure 6-18,** High pressure and temperature recovery boiler with reheat in p-h-diagram, 224.0 kg/s, 12.0/3.4 MPa, 515/400 °C.

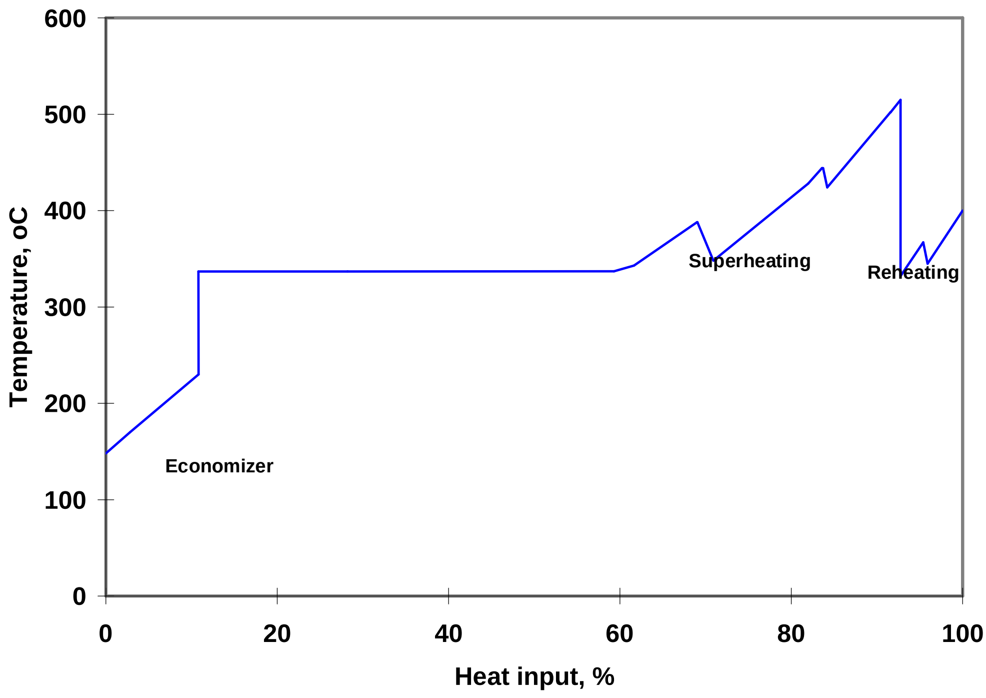


**Figure 6-19,** High pressure and temperature recovery boiler with reheat in Φ-t-diagram, 224.0 kg/s, 12.0/3.4 MPa, 515/400 °C.

Reheating does not change the basic boiler heat input much Figure 6-19. It just adds more superheating like heat requirement. Thus the percentage of economizing and evaporating decreases.

One again notices the large temperature difference between economizer outlet and drum saturated water temperatures. It could be possible to replace boiler bank with 3rd evaporator thus having a recovery boiler with no boiler bank.

**Table 6-5 Main dimensions of high pressure and temperature recovery boiler with reheat, case E.**

| Surface | unit | |
|---|---|---|
| Furnace area | $m^2$ | 232.1 |
| Furnace height | m | 38.4 |
| Total boiler height | m | 75.0 |
| Furnace area | $m^2$ | 4672 |
| Screen area | $m^2$ | 796 |
| Superheater area | $m^2$ | 16102 |
| Superheater area (proj) | $m^2$ | 15446 |
| Reheater area | $m^2$ | 5203 |
| Reheater area (proj) | $m^2$ | 4991 |
| Boiler bank area | $m^2$ | 11982 |
| Economizer area | $m^2$ | 39382 |
| Total area | $m^2$ | 98575 |

The total increase in superheater area compared to case D is not that big, only 5 %. If compare case C to case E, then the increase of required superheater area is rather large 32 %. This inspite of the fact that furnace height was reduced 3.2 m from case C to case E. If the furnace height would have been kept constant, meaning constant furnace exit temperature, the required superheating area increase would have been even larger.

The reheat boiler low pressure conditions are of high temperature. This could require spraying to get low pressure steam conditions closer to the saturated. This is mainly because the main steam pressure was chosen to correspond case C. If higher pressure were chosen the expansion could be closer to saturation or alternatively even more superheating could be used.

Condensate tail end moisture is between 8 and 9 % so fairly moderate and could be increased somewhat. Increasing would mean e.g. higher pressure. Higher pressure alone would not mean more electricity.

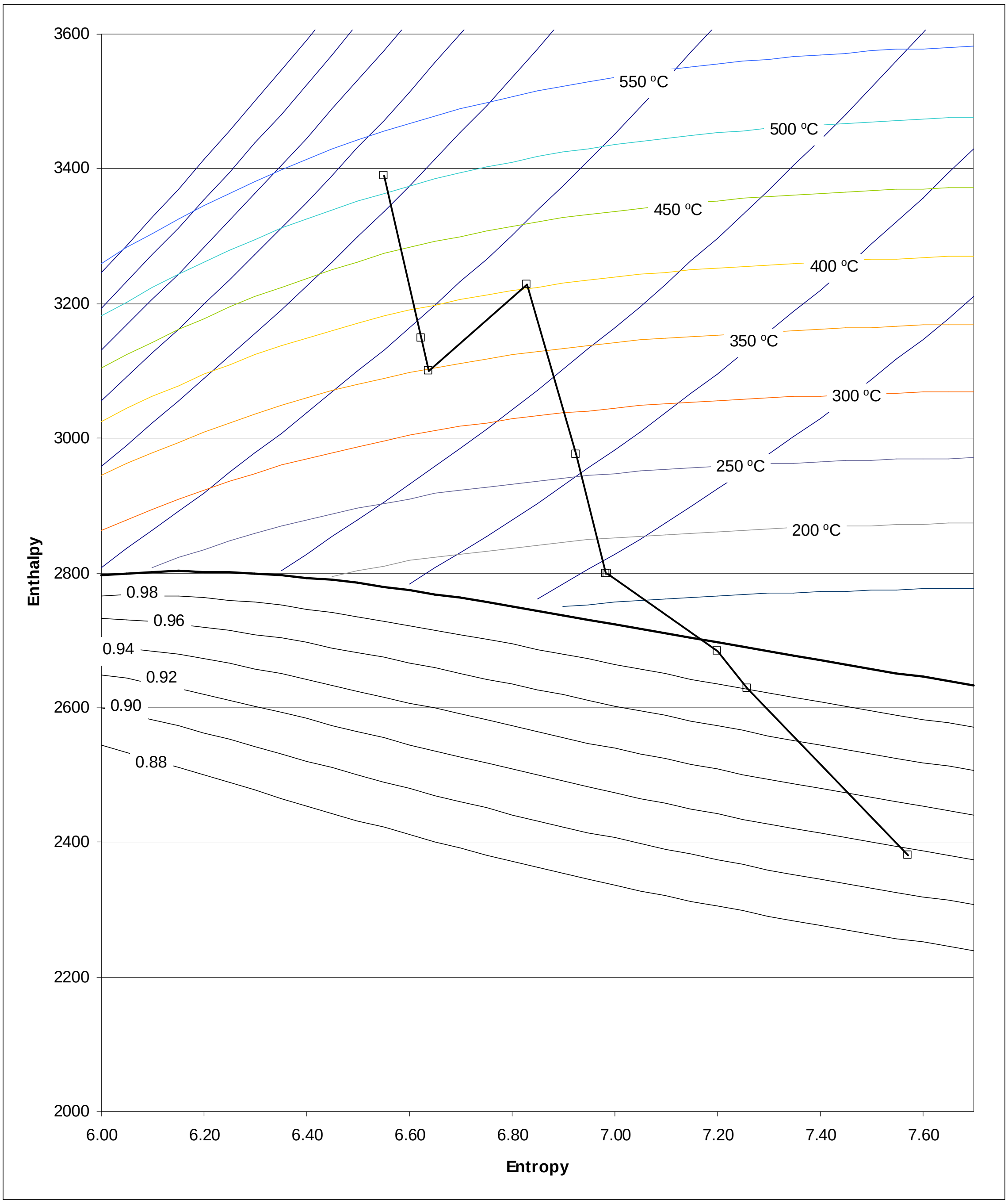


**Figure 6-20,** High pressure and temperature recovery boiler with reheat in h-s-diagram, 224.0 kg/s, 12.0/3.4 MPa, 515/400 °C.

## 6.6 Case F – Once-through recovery boiler with reheat concept

Once through boiler case F is both heat transfer surface tube size wise and heat absorption wise quite different from other concepts. High pressure means that traditional change from water to steam requiring latent heat does not happen. Instead the steam/water stays fluid because of the pressure.

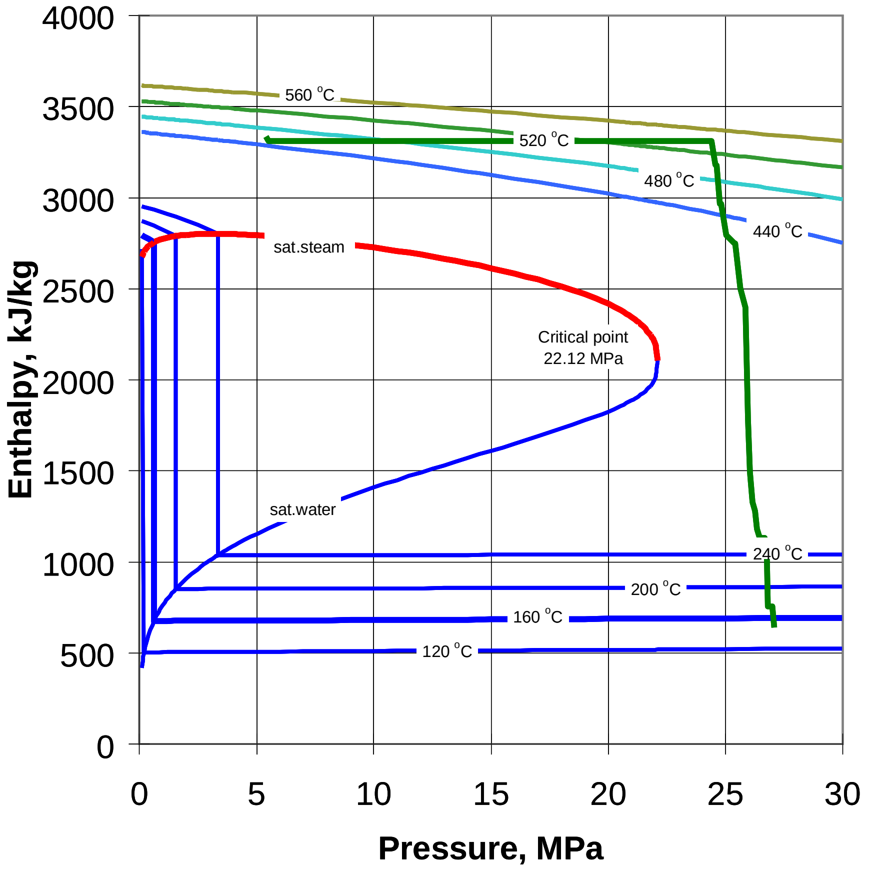


**Figure 6-21,** Once-through recovery boiler with reheat in p-h-diagram, 218.5 kg/s, 24.0/5.4 MPa, 540/460 °C.

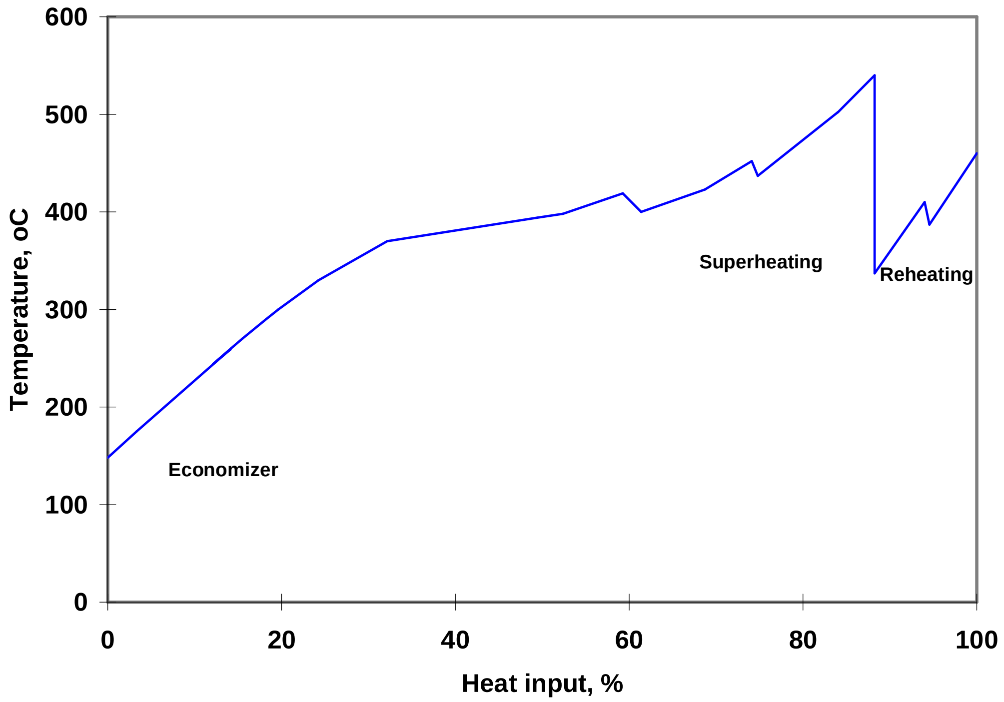


**Figure 6-22,** Once-through recovery boiler with reheat in Φ-t-diagram, 218.5 kg/s, 24.0/5.4 MPa, 540/460 °C.

The temperature of steam/water in once-through recovery boiler stays higher on average than in lower pressure boilers. This decreases the available temperature difference and heat transfer, thus requiring more heat transfer surface, Figure 6-21 and 6-22.

**Table 6-6 Main dimensions of once-through recovery boiler with reheat, Case F.**

| **Surface** | **unit** | |
|---|---|---|
| Furnace area | $m^2$ | 234.4 |
| Furnace height | m | 33.6 |
| Total boiler height | m | 72.0 |
| Furnace area | $m^2$ | 4084 |
| Screen area | $m^2$ | 796 |
| Superheater area | $m^2$ | 19160 |
| Superheater area (proj) | $m^2$ | 18713 |
| Reheater area | $m^2$ | 5212 |
| Reheater area (proj) | $m^2$ | 5090 |
| Boiler bank area | $m^2$ | 12786 |
| Economizer area | $m^2$ | 39235 |
| Total area | $m^2$ | 105075 |

Compared to case C once-through reheater recovery boiler has 19 % more total heat-transfer area and 54 % more superheating area. To keep flow velocities high enough, the tube size in furnace walls is much smaller than in natural circulation boilers.

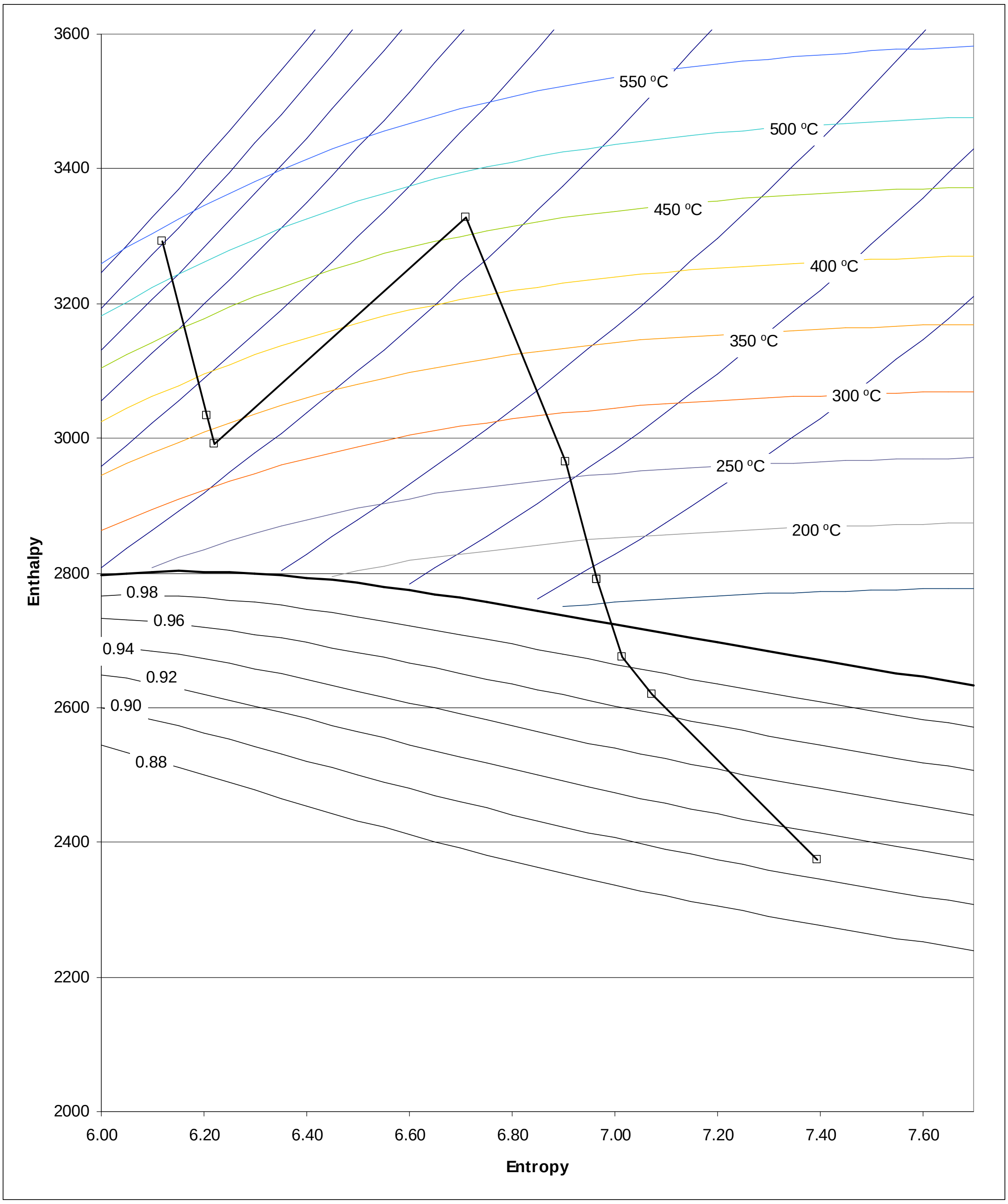


**Figure 6-23,** Once-through recovery boiler with reheat in h-s-diagram, 218.5 kg/s, 24.0/5.4 MPa, 540/460 $^{o}$C.

Because we start from much higher pressure we can have much higher reheating and still achieve about the same end conditions, Figure 6-23, than turbines for natural circulation boilers cases A-C.

## 6.7 Boiler size comparison

Recovery boiler pressure part weights for different options are shown in Table 6-7.

**Table 6-7 Main size values for each case.**

| | | Joutseno | Kymi | Yonago | SoTu | SkyRec | SkyRec+ |
|---|---|---|---|---|---|---|---|
| **Case** | | **A** | **B** | **C** | **D** | **E** | **F** |
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| Capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 82.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| Main steam pressure RB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 264.0 |
| Main steam temp. RB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 540.0 |
| **General data** | | | | | | | |
| Design pressure | bar(e) | 130 | 129 | 148 | 186 | 148 | 289 |
| **Areas** | | | | | | | |
| Furnace bottom area | m2 | 233.6 | 232.1 | 232.1 | 232.1 | 232.1 | 234.4 |
| Furnace height | m | 36.5 | 41.6 | 41.6 | 39.6 | 38.4 | 33.6 |
| Total boiler height | m | 75.5 | 75.1 | 78.0 | 76.0 | 75.0 | 72.0 |
| Furnace area | m2 | 4417 | 5056 | 5056 | 4813 | 4672 | 4084 |
| Screen area | m2 | 0 | 796 | 0 | 796 | 796 | 796 |
| Superheater area | m2 | 23712 | 13536 | 16102 | 20255 | 21306 | 24372 |
| Superheater area (proj) | m2 | 16004 | 12984 | 15446 | 19429 | 20437 | 23803 |
| Boiler bank area | m2 | 13600 | 10262 | 11982 | 11982 | 11982 | 12786 |
| Economizer area | m2 | 40720 | 39676 | 39676 | 39529 | 39382 | 39235 |
| Total area | m2 | 98454 | 82310 | 88262 | 96804 | 98575 | 105075 |
| **Pressure part weight** | | | | | | | |
| Furnace panels weight | tons | 1035 | 1085 | 1107 | 1157 | 1051 | 843 |
| Screen weight | tons | 0 | 465 | 0 | 588 | 503 | 543 |
| Superheater weight | tons | 2254 | 1463 | 1827 | 2490 | 2417 | 2569 |
| Boiler bank weight | tons | 813 | 740 | 1041 | 1041 | 1041 | 1080 |
| Economizer weight | tons | 1845 | 2080 | 2080 | 2072 | 2063 | 2057 |
| Total weight | tons | 5946 | 5832 | 6055 | 7348 | 7075 | 7091 |
| Total hanging weight | tons | 9250 | 8760 | 9270 | 10290 | 10170 | 10140 |

Case A weight and superheater size is clearly larger than for Case B. This is not because of steam side values. The bigger size reflects the trend at that time to use significantly oversized superheaters and smaller tube diameters than is currently used. Both of these increase pressure part weight.

Increasing design pressure and increasing superheating increase the pressure part weight from Case B to Case C and to Case D. Eco and boiler bank weights do not increase significantly. Weight of connecting piping does increase.

Reheater cases E and F add more superheating surface and increase the total weight. Because of smaller tube in the furnace and shorter height, the weight of furnace is decreased.

## 6.8 Boiler cost and profitability of electricity generating comparison

Recovery boiler prices, additional electricity and investment difference for various cases are shown in Table 6-8.

**Table 6-8 Prices, additional electricity and investment difference for each case.**

| | | Joutseno | Kymi | Yonago | SoTu | SkyRec | SkyRec+ |
|---|---|---|---|---|---|---|---|
| **Case** | | **A** | **B** | **C** | **D** | **E** | **F** |
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| Capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 82.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| Main steam pressure RB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 264.0 |
| Main steam temp. RB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 540.0 |
| **Cost** | | | | | | | |
| Cost difference | M€ | 9.2 | 0.0 | 6.5 | 17.1 | 24.3 | 27.0 |
| Electricity difference | MWe | -4.0 | 0.0 | 7.3 | 19.6 | 8.3 | 19.0 |
| Cost for additional | k€/MWe | -2310 | 0 | 890 | 875 | 2921 | 1422 |
| Price of extra power | €/MWh | -57.8 | 0.0 | 22.2 | 21.9 | 73.0 | 35.6 |

The The increase in electricity generation seems very profitable up to case C. This confirms the rationality of design choices that have lead to the present recovery boiler. Case A costs more that it should were it built today. The reason is larger than required superheating surface and smaller than currently used superheater tube size. From cost of additional power, going to SoTu concept of 540 $^{o}$C steam seems desirable. Currently the corrosion issues have not yet been solved so in this study we assume that superheaters do not corrode. Reheater boiler concept seems not at all profitable. The additional electricity generation was only marginal. Once-through recovery boiler did produce as much additional electricity than the SoTu concept of 540 $^{o}$C steam. The corrosion issues still remain the same.

# 7 APROS-STUDIES

A dynamic model has been developed for the once-through recovery boiler with reheat and supercritical steam properties. The boiler is of the Benson type where the location for evaporation varies with the load. Simulation results are presented for the load change from 100 % to 80 % under sliding pressure operation.

The component-level specifications for the boiler have been obtained from LUT in-house design and serve as a basis for the dynamic modeling. The dynamic model has been implemented using the simulation software APROS (Advanced Process Simulation Environment).

The Advanced Process Simulation Environment (APROS) provides tools, solution algorithms and mode libraries for fullscale modelling and simulation of dynamic processes, such as combustion power plants. The model libraries have been comprehensively validated against real physical process experiments. The modular and hierarchical approach of APROS allows unique flexibility of process analysis at various conceptual levels.

APROS allows the inclusion of the user's own models in the calculation as well as easy connection to external models, automation systems or control room equipment. A large process model can be divided into several flowsheet diagrams. This can be done both in a hierarchical and a horizontal way. At any time, the complete model information can be saved into a model snapshot file containing the full model configuration and its momentary state data at the time instant. Similarly, at any time, the user can backtrack to a snapshot once saved in the past.

## 7.1 Model Composition

The developed model includes all heat transfer surfaces of the boiler, with the following order on the water side:

- Economisers (1 and 2)
- Boiler generating bank
- Rear wall screen
- Lower furnace
- Upper furnace
- Superheaters (1A, 1B, 2A, and 3A)
- Reheaters (IA and IIA)

Two spray attemperators are employed between the superheaters while one is used for the reheaters. Water for the attemperators is taken from the feedwater that enters the boiler. For the completeness, the high-pressure turbine and bleed of the discharge steam are also included in the model. The model has a water-steam separator after the evaporator section to ensure the stability and water circulation at low load levels and under startups and shutdowns. For the studied load levels, the separator has no effect on the operation. The APROS flowsheet diagram for the developed model is presented in Figure 7-1.

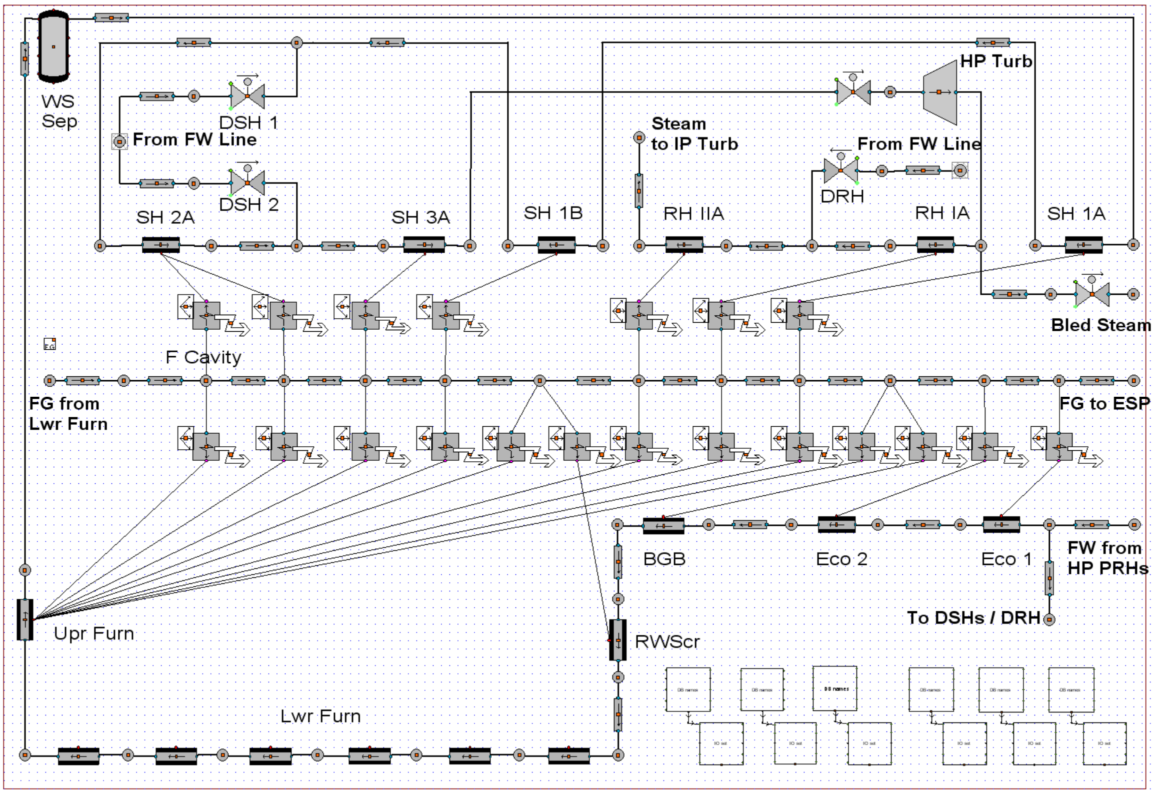


**Figure 7-1,** Once-through recovery boiler with reheat APROS model, 218.5 kg/s, 24.0/5.4 MPa, 540/460 $^{o}$C.

In the lower furnace below the nose, heat fluxes are given as input values for different heights and at varying load levels. In the upper furnace and onwards in the flue gas duct, heat transfer rates that are obtained from the in-house design for 100 % load form the basis for the heat transfer modeling. In the APROS model, the heat transfer is first tuned for the convection and radiation heat transfer rates to correspond to the design values at 100 % load. The tuning is performed using convection and radiation constants, $k_c$ and $k_r$, which in turn determine the convection and radiation heat transfer coefficients at the outer surface of the tube.

$$\alpha_c = k_c \, q_m^{0.6} \tag{7-1}$$

$$\alpha_r = k_r \, \sigma \frac{T_g^4 - T_w^4}{T_g - T_w} \tag{7-2}$$

In the equations $q_m$ is the mass flow rate and $T_g$ the mean temperature of the flue gas, $T_w$ the mean temperature of the tube outer surface, and $\sigma$ the Stefan-Boltzmann constant ($5.67 \cdot 10^{8}$ W/m$^2$K$^4$).

The heat transfer coefficient at the tube outer surface is formed as a sum of the convection and radiation coefficients, while at the inner surface the heat transfer coefficient is calculated by the software.

When simulating the operation below 100 % load, the heat transfer coefficients and consequently heat transfer rates are calculated by the software using the constants that have been determined during the tuning phase.

## 7.2 Input Values and Simulation Procedure

As the load is reduced, the following parameters are altered on the water side: feedwater mass flow rate and pressure, attemperator and bled steam mass flow rates, and inlet pressure to the intermediate turbine. On the flue-gas side, the varying parameters include the heat fluxes in the lower furnace, mass flow rate, and temperature at the nose.

At 80 % load, all water side input parameters have been reduced to 80 % of the corresponding 100 % load value. Heat generation on the flue-gas side has been adjusted so that the steady-state live steam temperature becomes equal to 100 % value 540 °C. As a result, the reheat steam temperature experiences a small change from 459 to 457 °C.

The water side and flue-gas side input values that have been used as a basis for the simulation from 100 % to 80 % load are given in Table 7-1. The corresponding heat flux distribution is presented in Figure 7-2.

**Table 7-1,** Water side and flue-gas side input values for 100 % and 80 % loads.

| | | **Load 100 %** | **Load 80 %** |
|---|---|---|---|
| ***Water side*** | | | |
| FW mass flow rate | [kg/s] | 223.1 | 178.5 |
| FW pressure | [bar] | 281.0 | 224.8 |
| DSH1 mass flow rate | [kg/s] | 20.0 | 16.0 |
| DSH2 mass flow rate | [kg/s] | 7.7 | 6.2 |
| DRH mass flow rate | [kg/s] | 4.6 | 3.7 |
| Bled steam mass flow rate | [kg/s] | 21.2 | 17.0 |
| IP turbine inlet pressure | [bar] | 54.0 | 43.2 |
| ***Flue-gas side*** | | | |
| Mass flow rate | [kg/s] | 322.1 | 274.6 |
| Temperature at nose | [°C] | 1401 | 1311 |

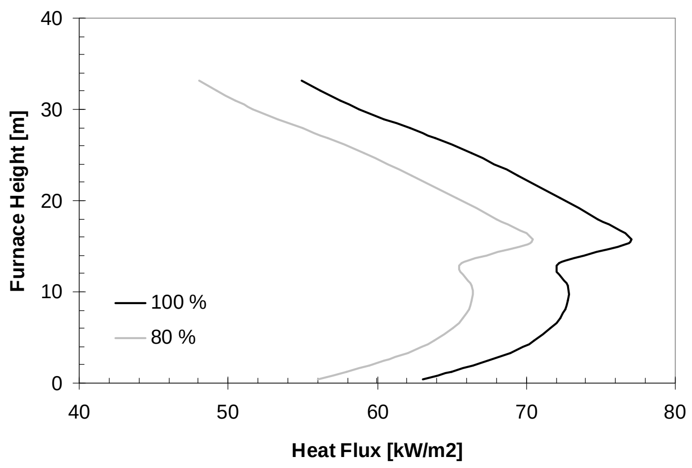


**Figure 7-2,** Heat flux distribution in the lower furnace for 100 % and 80 % loads.

The load change from 100 % to 80 % has been simulated assuming the input parameters to follow a decaying exponential function with given time constants. The time constant indicates the required time to reach approximately 63 % of the final value. A delay (dead time) occurs at the beginning of the simulation before any changes in input values. After the delay, the value for the input parameter is given by

$$y(t) = y_{100} + (y_{80} - y_{100})\left(1 - e^{-\frac{t-\theta}{\tau}}\right) \quad \text{7-3}$$

In the equation, $y_{100}$ and $y_{80}$ are the initial (100 %) and final (80 %) values of the input parameter, $t$ is time, $\tau$ time constant, and $\theta$ delay. The simulation starts at $t = 0$.

The selected time constant is 300 s for the heat fluxes, 300 s for the flue gas temperature at the nose, and 100 s for the flue gas mass flow rate. The time constant for the water side changes has been varied between 150 and 250 s. A delay of 100 s is used for all parameters.

Figure 7-3 provides the corresponding transient profiles of the input parameters, plotting the relative values of the parameter with different time constants and 100 s delay time.

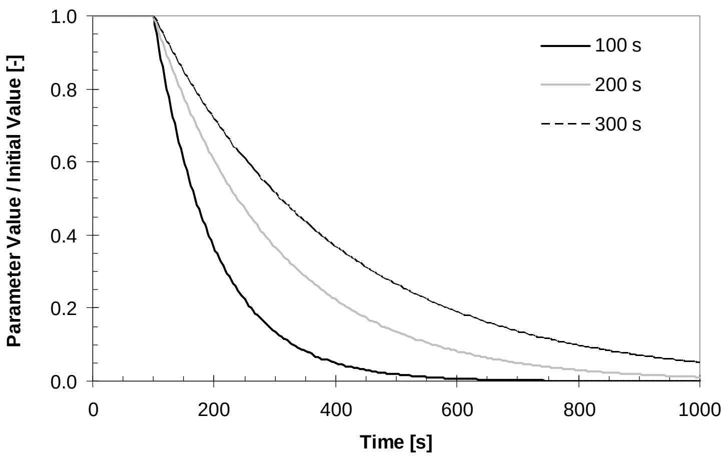


**Figure 7-3,** Transient profiles with different time constants.

## 7.3 Results and Discussion

Figures 7-4 and 7-5 present, as selected results from the simulation, the behavior of the live steam temperature and reheat steam temperature during 4000 s simulation for different values of water side time constant.

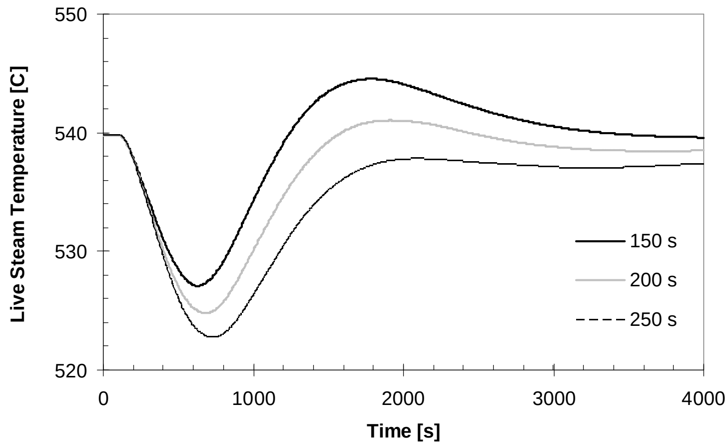


**Figure 7-4,** Variation of live steam temperature against time for different time constants on the water side.

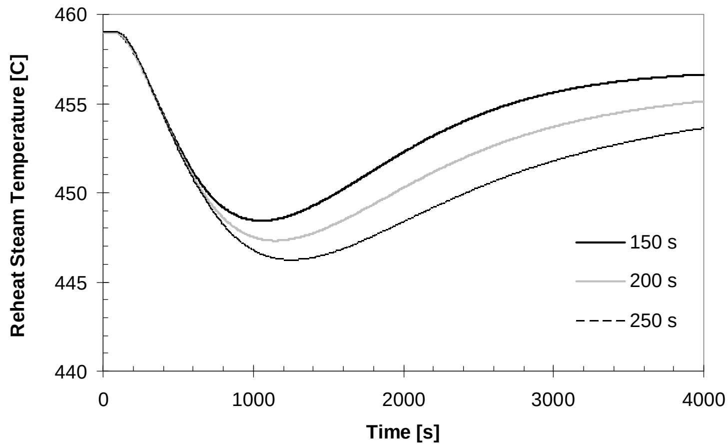


**Figure 7-5,** Variation of reheat steam temperature against time for different time constants on the water side.

As the figures show, the transient profiles on the water side have a significant effect on the steam temperature during the load change. Moreover, the temperature behavior can be affected via spray attemperators. In this concept-level study all water side mass flow rates are based on a common transient profile.

The simulation uses a simplified, time-constant based approach for describing the heat fluxes and flue-gas data during the load change. Increasing the accuracy of the dynamic behavior of the hot side would increase the accuracy of the simulation.

# 8 PREHEATER CONCEPTS

One of the most successful ways to increase electricity generation form recovery boilers has been the implementation of different preheating schemes (Raukola et al., 2002).

The preheating schemes have been selected based on typical practice.

## 8.1 Air preheater concepts

It is important to heat air. In Case C5 the air on average is preheated only to 121.2 $^oC$. This was typical to so called cold tertiary air systems. Cold tertiary was used to increase the mixing of air to flue gases in the furnace.

In Case C4 the air on average is preheated already to 150.4 $^oC$. This was typical after newer air systems became wider spread. In Case C3 the air on average is preheated to 190 $^oC$. This is the case for modern high electricity boilers (Aikio, 2009).

## 8.2 Water preheater concepts

In cases C5 – C3 the feedwater was preheated to 120 $^oC$. Flue gas temperature is then 155 $^oC$. In Case C2 the feedwater is preheated to 148 $^oC$ before the economizers. This means that flue gas exit temperature increases to 198 $^oC$. The flue gas is cooled down in feedwater preheater so the temperature to stack remains constant.

In case C1 the feedwater is additionally preheated between economizers from 200 to 220 $^oC$. This high pressure preheater concept is patented.

## 8.3 Sootblowing concepts

In cases C5 – C1 the sootblowing was taken from inside the recovery boiler. In Case C0 the Sootblowing is taken from turbine so some expansion in turbine occurs which increases the electricity generation. Case C0 equals Case C.

## 8.4 Electricty generation changes

Electricity generation in cases C5 – C0 is shown in the Table 8-1.

**Table 8-1 Main values and electricity generation for additional cases.**

| Case | | As case C<br>C0 | inside SB<br>C1 | No FW prh<br>C2 | Low FW<br>C3 | Air 150<br>C4 | Air 120<br>C5 |
|---|---|---|---|---|---|---|---|
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| Capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| Dry solids (virgin) | % | 83.8 | 83.8 | 83.8 | 83.8 | 83.8 | 83.8 |
| Recycle ash | % | 9.0 | 9.0 | 9.0 | 9.0 | 9.0 | 9.0 |
| HHV | MJ/kgds | 13.00 | 13.00 | 13.00 | 13.00 | 13.00 | 13.00 |
| HHV (virgin) | MJ/kgds | 14.29 | 14.29 | 14.29 | 14.29 | 14.29 | 14.29 |
| LHV | MJ/kgds | 12.28 | 12.28 | 12.28 | 12.28 | 12.28 | 12.28 |
| LHV (virgin) | MJ/kgds | 13.49 | 13.49 | 13.49 | 13.49 | 13.49 | 13.49 |
| $O_2$ in dry flue gas | % | 2.8 | 2.8 | 2.8 | 2.8 | 2.8 | 2.8 |
| Primary air | % | 22.0 | 22.0 | 22.0 | 22.0 | 22.0 | 22.0 |
| Primary air temperature | °C | 190.0 | 190.0 | 190.0 | 190.0 | **160.0** | **150.0** |
| Secondary air | % | 54.0 | 54.0 | 54.0 | 54.0 | 54.0 | 54.0 |
| Secondary air temp | °C | 190.0 | 190.0 | 190.0 | 190.0 | **160.0** | **150.0** |
| Tertiary air percentage | % | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 |
| Tertiary air temp | °C | 190.0 | 190.0 | 190.0 | 190.0 | **120.0** | **30.0** |
| Quartenary air | % | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 |
| Quartenary air temp | °C | 190.0 | 190.0 | 190.0 | 190.0 | **120.0** | **30.0** |
| Total air percentage | % | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 |
| Total air temperature | °C | 190.0 | 190.0 | 190.0 | 190.0 | **150.4** | **121.2** |
| Reduction | % | 96.00 | 96.00 | 96.00 | 96.00 | 96.00 | 96.00 |
| Main steam pressure RB | bar(a) | 124.0 | 124.0 | 124.0 | 124.0 | 124.0 | 124.0 |
| Main steam temp RB | °C | 515.0 | 505.0 | 515.0 | 515.0 | 515.0 | 515.0 |
| Feedwater pressure | bar(a) | 146.0 | 146.0 | 146.0 | 146.0 | 146.0 | 146.0 |
| Feedwater temperature | °C | 148.0 | 148.0 | 148.0 | **120.0** | **120.0** | **120.0** |
| HP FWpreh inlet temp | °C | 200 | 200 | 200 | 200 | 200 | 200 |
| HP FWpreh outlet temp | °C | 220 | 220 | **200** | **200** | **200** | **200** |
| Flue gas temp (eco out) | °C | 197 | 197 | 197 | **155** | **155** | **155** |
| Flue gas temp (to stack) | °C | 155 | 155 | 155 | 155 | 155 | 155 |
| Sootblowing | kg/s | 6.0 | **6.0** | **6.0** | **6.0** | **6.0** | **6.0** |
| Steam flow | kg/s | 232.5 | 228.6 | 221.0 | 217.1 | 213.6 | 211.1 |
| Change in steam flow | % | 0.0 | -1.7 | -5.0 | -6.6 | -8.1 | -9.2 |
| Pulping usage total | MW | 88.9 | 88.9 | 88.9 | 88.9 | 88.9 | 88.9 |
| Mill total usage | MW | 92.4 | 92.4 | 92.4 | 92.4 | 92.4 | 92.4 |
| Electricity production | MW | 161.8 | 160.8 | 158.4 | 156.5 | 154.5 | 153.6 |
| Surplus electricity | MW | 69.4 | 68.3 | 66.0 | 64.1 | 62.0 | 61.2 |

Changing Sootblowing from external to internal decreases electricity as the heat in steam does not produce electricity in the turbine, Case C0 to Case C1. High pressure feedwater preheating seems to add over 2 MW electricity, Case C1 to Case C2. About similar change is seen if we decrease the use of feedwater preheat, Case C2 to Case C3. Dropping air preheating decreases the electricity generation even more Case C3 to case C4 to case C5.

**Table 8-2 Prices, additional electricity and investment difference for each case.**

| Case | | As case C<br>C0 | inside SB<br>C1 | No FW prh<br>C2 | Low FW<br>C3 | Air 150<br>C4 | Air 120<br>C5 |
|---|---|---|---|---|---|---|---|
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| Capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| Primary air temperature | °C | 190.0 | 190.0 | 190.0 | 190.0 | **160.0** | **150.0** |
| Secondary air | % | 54.0 | 54.0 | 54.0 | 54.0 | 54.0 | 54.0 |
| Secondary air temp | °C | 190.0 | 190.0 | 190.0 | 190.0 | **160.0** | **150.0** |
| Tertiary air percentage | % | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 |
| Tertiary air temp | °C | 190.0 | 190.0 | 190.0 | 190.0 | **120.0** | **30.0** |
| Quartenary air | % | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 | 12.0 |
| Quartenary air temp | °C | 190.0 | 190.0 | 190.0 | 190.0 | **120.0** | **30.0** |
| Total air temperature | °C | 190.0 | 190.0 | 190.0 | 190.0 | **150.4** | **121.2** |
| Main steam pressure RB | bar(a) | 124.0 | 124.0 | 124.0 | 124.0 | 124.0 | 124.0 |
| Main steam temp RB | °C | 515.0 | 505.0 | 515.0 | 515.0 | 515.0 | 515.0 |
| Feedwater pressure | bar(a) | 146.0 | 146.0 | 146.0 | 146.0 | 146.0 | 146.0 |
| Feedwater temperature | °C | 148.0 | 148.0 | 148.0 | **120.0** | **120.0** | **120.0** |
| HP FWpreh outlet temp | °C | 220 | 220 | **200** | **200** | **200** | **200** |
| Flue gas temp (eco out) | °C | 197 | 197 | 197 | **155** | **155** | **155** |
| Sootblowing | kg/s | 6.0 | **6.0** | **6.0** | **6.0** | **6.0** | **6.0** |
| Steam flow | kg/s | 232.5 | 228.6 | 221.0 | 217.1 | 213.6 | 211.1 |
| Surplus electricity | MW | 69.4 | 68.3 | 66.0 | 64.1 | 62.0 | 61.2 |
| **Cost** | | | | | | | |
| Cost difference | M€ | 0 | -1.3 | -1.9 | -3.8 | -4.4 | -4.7 |
| Electricity difference | MWe | 0 | -1.1 | -3.5 | -5.3 | -7.4 | -8.3 |
| Cost for additional | k€/MWe | 0 | 1193 | 550 | 711 | 590 | 568 |
| Price of extra power | €/MWh | 0.0 | 29.8 | 13.8 | 17.8 | 14.7 | 14.2 |

Price of additional electricity for internal vs. external sootblowing does not seem very profitable. But other typically used means to generate additional electricity do seem to make a lot of sense. The key in internal vs. external Sootblowing is whether one needs to invest more in the additional openings and their control in turbine.

# 9 CONCLUSIONS

In this study several recovery boiler concepts were compared with the whole pulp mill energy balance being considered. The studied concepts are

G. Natural circulation 82 %, 490 $^{o}$C, 9.0 MPa (reference Joutseno)

H. Natural circulation 85 %, 505 $^{o}$C, 10.2 MPa (reference Kymi)

I. Natural circulation 85 %, 515 $^{o}$C, 12.0 MPa (reference Yonago)

J. Assisted circulation 85 %, 540 $^{o}$C, 16.0 MPa (reference SoTu)

K. Natural circulation 85 %, 515/400 $^{o}$C, 12.0/3.4 MPa (SkyRec)

L. Once-through 85 %, 540/460 $^{o}$C, 26.0/5.4 MPa (SkyRec+)

Steam and electricity generation for each recovery boiler case is shown in Table 9-1. The steam production increases from Case A to Case B because of higher black liquor dry solids and more air preheating. The steam production increases from Case B to Case C because of high pressure preheating. The recovery boiler steam flow starts decreasing as further increases in main steam parameters require more heat.

**Table 9-1 Steam and electricity generation for each case.**

| | | Joutseno | Kymi | Yonago | SoTu | SkyRec | SkyRec+ | SkyRec |
|---|---|---|---|---|---|---|---|---|
| **Case** | | **A** | **B** | **C** | **D** | **E** | **F** | **E160** |
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 82.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| Main steam pressure RB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 264.0 | 164.0 |
| Main steam temp. RB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 540.0 | 515.0 |
| **Steam generation** | | | | | | | | |
| Steam flow RB | kg/s | 215.0 | 226.4 | 233.6 | 232.0 | 224.0 | 218.5 | 224.0 |
| Change in steam flow | % | 0.0 | 5.3 | 8.6 | 7.9 | 4.1 | 1.6 | 4.1 |
| **Electricity** | | | | | | | | |
| Pulping usage total | MW | 87.6 | 88.1 | 88.9 | 90.2 | 88.2 | 93.2 | 90.0 |
| Mill total usage | MW | 91.1 | 91.7 | 92.4 | 93.8 | 91.8 | 96.8 | 93.6 |
| Electricity production | MW | 149.3 | 153.9 | 161.8 | 175.6 | 162.3 | 177.9 | 167.0 |
| Surplus electricity | MW | 58.2 | 62.2 | 69.4 | 81.7 | 70.5 | 81.1 | 73.4 |
| Efficiency to electricity | % | 20.4 | 20.4 | 21.4 | 23.2 | 21.5 | 23.5 | 22.7 |
| Change in electricty | MW | 0.0 | 4.0 | 11.2 | 23.6 | 12.3 | 22.9 | 15.2 |
| | % | 0.0 | 6.8 | 19.3 | 40.5 | 21.1 | 39.4 | 26.2 |

As can be seen the modern recovery boiler Case C does produce about 20 % more electricity than roughly ten years ago, case A. reheating cases E and E160 seem to give only marginally better electricity production. The only alternative seems to be to increase the main steam temperature to 540 $^{o}$C, Cases D and F. The pulping electricity usage is not constant. The main parameter that changes is the recovery boiler feedwater pump power requirement.

Electricity generation does not depend a lot on how the boiler steam side is configured. Reheating and once-through appear only marginally better when considering the recovery boiler electricity generation.

**Table 9-2 Cost difference and price of extra power for each case.**

| | | Joutseno | Kymi | Yonago | SoTu | SkyRec | SkyRec+ |
|---|---|---|---|---|---|---|---|
| **Case** | | **A** | **B** | **C** | **D** | **E** | **F** |
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| capacity (virgin) | tds/d | 5005 | 5005 | 5005 | 5005 | 5005 | 5005 |
| Dry solids | % | 82.0 | 85.0 | 85.0 | 85.0 | 85.0 | 85.0 |
| Main steam pressure RB | bar(a) | 94.0 | 104.0 | 124.0 | 164.0 | 124.0 | 264.0 |
| Main steam temp. RB | °C | 490.0 | 505.0 | 515.0 | 540.0 | 515.0 | 540.0 |
| **Cost** | | | | | | | |
| Cost difference | M€ | 9.2 | 0.0 | 6.5 | 17.1 | 24.3 | 27.0 |
| Electricity difference | MWe | -4.0 | 0.0 | 7.3 | 19.6 | 8.3 | 19.0 |
| Cost for additional | k€/MWe | -2310 | 0 | 890 | 875 | 2921 | 1422 |
| Price of extra power | €/MWh | -57.8 | 0.0 | 22.2 | 21.9 | 73.0 | 35.6 |

The increase in electricity generation seems very profitable up to case C. This confirms the rationality of design choices that have lead to the present recovery boiler. Case A costs more that it should were it built today. The reason is larger than required superheating surface and smaller than currently used superheater tube size. From cost of additional power, going to SoTu concept of 540 $^{o}$C steam seems desirable. Currently the corrosion issues have not yet been solved so in this study we assume that superheaters do not corrode. Reheater boiler concept seems not at all profitable. The additional electricity generation was only marginal. Once-through recovery boiler did produce as much additional electricity than the SoTu concept of 540 $^{o}$C steam. The corrosion issues still remain the same.

One of the most successful ways to increase electricity generation form recovery boilers has been the implementation of different preheating schemes (Raukola et al., 2002). Several preheating schemes have been selected based on typical practice. In Case C5 the air on average is preheated only to 121.2 $^{o}$C. In Case C4 the air on average is preheated already to 150.4 $^{o}$C. In Case C3 the air on average is preheated to 190 $^{o}$C. In Case C2 the feedwater is preheated to 148 $^{o}$C instead of 120 $^{o}$C before the economizers an additiona feedwater preheater is installed. In case C1 the feedwater is additionally preheated between economizers from 200 to 220 $^{o}$C. This high pressure preheater concept is patented. In Case C0 the Sootblowing is taken from turbine. Case C0 equals Case C.

**Table 9-3 Prices, additional electricity and investment difference for each case.**

| | | As case C | inside SB | No FW prh | Low FW | Air 150 | Air 120 |
|---|---|---|---|---|---|---|---|
| **Case** | | **C0** | **C1** | **C2** | **C3** | **C4** | **C5** |
| Capacity | tds/d | 5500 | 5500 | 5500 | 5500 | 5500 | 5500 |
| Total air temperature | °C | 190.0 | 190.0 | 190.0 | 190.0 | **150.4** | **121.2** |
| Main steam pressure RB | bar(a) | 124.0 | 124.0 | 124.0 | 124.0 | 124.0 | 124.0 |
| Main steam temp RB | °C | 515.0 | 505.0 | 515.0 | 515.0 | 515.0 | 515.0 |
| Feedwater temperature | °C | 148.0 | 148.0 | 148.0 | **120.0** | **120.0** | **120.0** |
| HP FWpreh outlet temp | °C | 220 | 220 | **200** | **200** | **200** | **200** |
| Flue gas temp (eco out) | °C | 197 | 197 | 197 | **155** | **155** | **155** |
| Sootblowing | kg/s | 6.0 | **6.0** | **6.0** | **6.0** | **6.0** | **6.0** |
| Steam flow | kg/s | 232.5 | 228.6 | 221.0 | 217.1 | 213.6 | 211.1 |
| Surplus electricity | MW | 69.4 | 68.3 | 66.0 | 64.1 | 62.0 | 61.2 |
| **Cost** | | | | | | | |
| Cost difference | M€ | 0 | -1.3 | -1.9 | -3.8 | -4.4 | -4.7 |
| Electricity difference | MWe | 0 | -1.1 | -3.5 | -5.3 | -7.4 | -8.3 |
| Cost for additional | k€/MWe | 0 | 1193 | 550 | 711 | 590 | 568 |
| Price of extra power | €/MWh | 0.0 | 29.8 | 13.8 | 17.8 | 14.7 | 14.2 |

Changing Sootblowing from external to internal decreases electricity as the heat in steam does not produce electricity in the turbine, Case C0 to Case C1. High pressure feedwater preheating seems to add over 2 MW electricity, Case C1 to Case C2. About similar change is seen if we decrease the use of feedwater preheat, Case C2 to Case C3. Dropping air preheating decreases the electricity generation even more Case C3 to case C4 to case C5.

Price of additional electricity for internal vs. external sootblowing does not seem very profitable. But other typically used means to generate additional electricity do seem to make a lot of sense. The key in internal vs. external Sootblowing is whether one needs to invest more in the additional openings and their control in turbine.

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# APPENDICES

# APPENDIX I

**APPENDIX II**