Lappeenranta-Lahti University of Technology LUT
School of Energy Systems
Degree Program in Energy Technology
Master's Thesis

Tommi Toikka

OPTIMIZING RESPONSE TIMES IN POWER GENERATION IN COMBINED CYCLE APPLICATIONS

Instructor:	M.Sc. (Tech) Matias Pessa
Examiner:	D.Sc. (Tech) Prof. Esa Vakkilainen M.Sc (Tech) Kari Luostarinen

ABSTRACT

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Optimizing response times in power generation in combined cycle applications

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74 pages, 35 figures and 6 equations

Examiner: D.Sc. (Tech) Prof. Esa Vakkilainen, M.Sc. (Tech) Kari Luostarinen
Instructor: M.Sc. (Tech) Matias Pessa (Alfa Laval Aalborg Oy)

Keywords: Start-up time, response time, waste heat recovery, combined cycle power plant

To allow higher shares of intermittent renewable energy generation flexible balancing power is required for back-up. Two significant flexibility parameters are the time it takes for a power plant to start-up and how quickly it can increase its load. In this work the restrictions and bottlenecks of a start-up and ramping of a combined cycle power plant are studied from thermodynamic point-of-view. A transient simulation tool is built to simulate the start-up process of the boiler and the results are analyzed with literature to find physical and sequence-related solutions for a flexible combined cycle power plant.

The main thermodynamic factor for the boiler start-up is its thermal inertia which mostly associated with the thermal capacity of the steam drum. The sequence-related factors usually try to protect the boiler, the piping and the steam turbine from damages with interlocks. For the boiler the steam line heating process related to the blow out sequence showed a big impact to the boiler start-up time. With the steam turbine the biggest bottlenecks were the heating process during cold start and the availability of auxiliary steam during hot start.

With the obtained results an analysis was done to figure out the most suitable technical solutions for different operational profiles and further questions for study were raised regarding standby operation, mechanical stress factors and drum level behavior.

TIIVISTELMÄ

Lappeenrannan-Lahden teknillinen yliopisto LUT
School of Energy Systems
Energiatekniikan koulutusohjelma

Tommi Toikka

Sähköntuotannon vasteaikojen optimointi kombilaitossovelluksissa

Diplomityö 2021
74 sivua, 35 kuvaa ja 6 yhtälöä

Tarkastaja: TkT Prof. Esa Vakkilainen, DI Kari Luostarinen
Ohjaaja: DI Matias Pessa (Alfa Laval Aalborg Oy)

Hakusanat: Käynnistysaika, vasteaika, lämmöntalteenotto, kombivoimalaitos

Vaihtelevan uusiutuvan energian tuotannon lisääminen kasvattaa joustavan varavoiman tarvetta. Kaksi merkittävää voimalaitoksen joustavuusparametria ovat, kuinka nopeasti laitos voidaan ajaa ylös ja kuinka nopeasti se pystyy nostamaan tehoaan. Tässä työssä tutkitaan termodynaamisesta näkökulmasta kombivoimalaitoksen ylösajon rajoitteita ja pullonkauloja käynnistyksen yhteydessä ja kuormaa lisätessä. Työtä varten rakennetaan aikariippuvainen laskentaohjelma simuloimaan kattilan ylösajoa ja tuloksia analysoidaan kirjallisuuden kanssa fysikaalisten ja sekvenssiriippuvaisten ratkaisujen löytämiseksi joustavalle kombivoiman tuotannolle.

Merkittävin termodynaaminen tekijä kattilan ylösajossa on sen terminen inertia, joka pääosin riippuu höyrylieriön lämpökapositeetista. Sekvenssivaiheiden tarkoitus on usein suojella kattilaa, höyryputkistoa ja turbiinia vaurioilta. Höyryputkistojen lämmitykseen liittyvällä ulospuhallussekvenssillä oli merkittävä vaikutus kattilan ylösajon kestoon. Höyryturbiinille merkittävimmät pullonkaulat olivat lämmitysprosessi kylmästartissa ja apuhöyryn saatavuus kuumastartissa.

Saaduilla tuloksilla analysoitiin eri ratkaisujen soveltuvuutta erilaisiin käyttöprofiileihin ja jatkokysymysehdotuksia annettiin liittyen alasajetun kattilan operointiin, mekaniisiin kuormituksiin ja lieriön pinnan käytökseen.

FOREWORD

This thesis started in the late spring and lasted all the way until ruska (the colorful time of Autumn) is at its prettiest here in my current hometown Lappeenranta. I would like to thank my instructor Matias Pessa for his efforts and constructive feedback throughout process from initial problem formation to finalization. I would also like to thank the many colleges in Alfa Laval Aalborg Oy who have helped me with sharing information and knowledge and been supportive. The same also goes for the external professionals who provided me insight in their respective fields of expertism.

With the finishing of this thesis my time as a student comes to an end. I had marvelous five and some years studying in Lappeenranta with many amazing people who I can call friends. Besides time with friends, I have also enjoyed the high-quality education. Throughout the journey the decision studying energy technology here in LUT has felt more and more like the best decision.

Lastly, I want to thank the people closest to me who have given me support during the thesis.

October 7th, 2021, Lappeenranta
Tommi Toikka

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LIST OF SYMBOLS AND ABBREVIATIONS

Roman alphabet

c_p	specific heat capacity	kJ/kgK
h	enthalpy	kJ/kg
m	mass	kg
P	power	kW
p	pressure	bar, Pa
q_m	mass flow rate	kg/s
T	temperature	°C/K
v	specific volume	m^3/kg

Subscripts

0	initial
1,2,3	state
ev	evaporation
nom	nominal
s	steel
TS	time step

Superscripts

$'$	liquid
$''$	steam

Abbreviations

ALA	Alfa Laval Aalborg
CCGT	Closed Cycle Gas Turbine
CCP	Combined Cycle Plant
CHP	Combined Heat and Power
E2D	Economizer to Drum
ECO	Economizer
EG	Exhaust Gas
EU	European Union
EVA	Evaporator

GHG	Greenhouse Gas
GT	Gas Turbine
HP	High Pressure
ICE	Internal Combustion Engine
IP	Intermediate Pressure
IWL	Idle Water Level
LAH	Level Alarm High
LAL	Level Alarm Low
LP	Low Pressure
NWL	Normal Water Level
OCGT	Open Cycle Gas Turbine
OEM	Original Equipment Manufacturer
PRDS	Pressure Reducing Desuperheater
SHL	Superheater Line
SUP	Superheater
WHR	Waste Heat Recovery
WHRB	Waste Heat Recovery Boiler

1 INTRODUCTION

Global warming is a threat and to combat this threat countries all over the world have agreed to limit the warming below 2 °C known as the Paris Agreement. Global warming is caused by greenhouse gas (GHG) emissions, mainly CO2, which forms an isolating layer around the earth preventing heat from radiating to space. Globally electricity and heat generation accounts to 31 % of total man-made GHG emissions (C2ES, 2019). The European Union (EU) has set a goal of reducing GHG emissions by at least 40 % by 2030 compared to the 1990 levels. To achieve this the union aims to increase its share of renewable energy to at least 32 % of the total energy demand and to improve energy efficiency by at least 32,5 % at the same time. However, these targets will likely increase in the future. (European Council, 2021)

To increase the share of renewables the countries in the union have installed a lot of wind and solar power which in their nature are highly intermittent and thus require flexible power generation to back them up. The annual capacity factors for solar and wind range between 10 - 21% and 23 - 51% respectively, depending on their location and technology (IEA 2020).

The factors measuring how well a power plant suits for flexible operation can be determined by many parameters such as start-up time, start-up cost, minimum load, part load efficiency, ramp rate, minimum uptime and minimum downtime (IRENA 2019). The importance of each factor is highly dependent on the operational profile. For example, if the power plant operates as load following entity the part load efficiency, the ramp rate and the minimum load play significant role, but the plant doesn't necessary require low start-up costs or short start-up time. The opposite is true for plants that run part time, for example, only operate during daytime with full load and are run down for the night. This kind of operational profile is the main scope of this work.

Table 1 lists state of the art flexibility parameters of conventional power plants. As it can be seen internal combustion engines (ICE) and open cycle gas turbines (OCGT) provide great flexibility parameters and especially ICE has great flexibility. However, closed cycle gas turbines (CCGT), while lagging on flexibility, offer nearly 50 % improvement in efficiency at full load and even more so on part load. Hence, we face a dilemma between renewable energy and energy efficiency. For maximum variable renewable energy penetration the open cycle operation is preferred, but to improve the energy efficiency the waste heat of a GT or ICE should be recovered.

Due to their greater efficiency the exhaust gases of an ICE are significantly cooler than for GTs meaning there is less recoverable energy available. According to an engine manufacturer Wärtsilä a steam turbine could boost the efficiency of an ICE by 20 % (Wärtsilä 2021). For an OCGT or ICE there isn't too much to be done to improve flexibility, but significant improvements can be done to the secondary cycle. (IRENA 2019)

Table 1. Flexibility parameters of conventional power plant types. (According to IRENA 2019) Note that start-up time is given for hot start (shut time less than 8 hours).

Type of plant	Hard coal	Lignite	CCGT	OCGT	ICE
Start-up time	2-10 h	4 - 10 h	1 - 4 h	5 - 11 min	5 min
Start-up cost [$€_{2021}$/MW]	> 90	> 90	50	1 - 65	< 1
Minimum load [% P_{nom}]	25 - 40 %	50 - 60 %	40 - 50 %	40 - 50 %	20 % (per unit)
Efficiency @ 100 % load	43 %	40 %	52 - 57 %	35 - 39 %	45 - 47 %
Efficiency @ 50 % load	40 %	35 %	47 - 51 %	27 - 32 %	45 - 47 %
Avg. Ramp rate [% P_{nom}/min]	1,5 - 4 %	1 - 2 %	2 - 4 %	8 - 12 %	> 100 %
Minimum uptime	48 h	48 h	4 h	10 - 30 min	< 1 min
Minimum downtime	48 h	48 h	2 h	30 - 60 min	5 min

Basically, the aims of flexibilization can be categorized into four:

- shorter start-up time and lower start-up costs,
- lower minimum load and improved part-load efficiency,
- higher ramp rate and
- shorter minimum uptime and downtime. (IRENA 2019)

Furthermore, in this work they are categorized in to two: start-up and load following properties. The plant shut down and start-up in particular cause great thermal and pressure related stresses in the boiler causing creep and fatigue on the steel structures increasing the likelihood of failures and unplanned outages and even lowering the technical lifetime of the plant

(Van den Bergh & Delarue 2015). This causes, together with factors like preheating fuel consumption etc., costs that are known as start-up costs and it determines the minimum up- and downtime of the plant. If the plant has low start-up costs it is profitable to start-up even for the shortest of times given the physical restriction of the start-up time and vice versa it is profitable to intentionally shut the plant down for shorter periods of time, so the plant isn't forced to operate during non-profitable market prices. Also ramping causes these same stresses to some extent.

The lower minimum load together with improved part-load efficiency and higher ramp rate offer the plant more variety in their business model. For example, the plant could participate in the day-ahead auction with its minimum load and then participate in the intraday market, which often has similar, yet more volatile, prices compared to day-ahead with its remaining capacity offering potential for higher revenue. The intraday market plays a big role for example, in the German and Dutch electricity market. In the Nordic countries a separate balancing market is available where producers can place bids for up and down regulations giving further incentive for flexible operation. The upward regulation has a requirement that the bid reserve must be fully activated within 15 minutes, but for day-ahead and intraday the time granularity with placed bids determines the required ramp rate. (Energinet 2018) In the Nordic countries the electricity market works with hourly orders (Nord Pool 2021) but there is on incentive for higher granularity (IRENA 2019).

The goals of this work are to identify the bottlenecks of a combined cycle plant (CCP) start-up sequence and find the most suitable physical solutions and optimized sequence to boost flexibility in the secondary cycle as well as to determine how those solutions fit to different operating profiles. The scope includes considerations for boiler, turbine and the balance of plant, but it is kept at pure power generation, so no combined heat and power (CHP) applications are considered. Any supplementary firing applications will be left out of scope and both one- and two-pressure level systems will be considered.

The outline of the thesis is that first the combined cycle power plant is introduced with its main parts and then its start-up sequence is explained and broken down to pieces for multiple scenarios. After that a built transient simulation tool is introduced it is validated by using real life data. The tool would then be used to evaluate boiler start-up times for different physical and sequence related solutions together with the help of literature. Thereafter the steam turbine and the balance of plant are studied with the aim of overcoming the bottlenecks of a steam turbine start-up using literature and the simulation tool to some extent. Finally, the boiler and turbine results are combined, and conclusions drawn to find the most suitable set-ups for different operational profiles.

2 WHR IN COMBINED CYCLE APPLICATIONS

Waste heat recovery is a way of improving energy efficiency in multiple thermal applications. In steam generation, WHRB and heat recovery steam generator (HRSG) are two established terms for the apparatus used for the generating steam from exhaust gas waste heat. Typically, the exhaust gas temperatures of gas turbines range from 450 to 520 °C in size range of 3 - 50 MW_e (Energy Solutions Center). For ICEs the exhaust temperature is often around 370 °C for gas engines (Wärtsilä 2020) and considerably lower for diesel engines around 330 °C (Wärtsilä 2019). Hence, there is a possibility for relatively high-grade heat recovery. Figure 1 shows an exemplary combined cycle power plant from the secondary cycle point of view. Red stream color means steam and blue water. The orange and black disc describes a steam blow out valve.

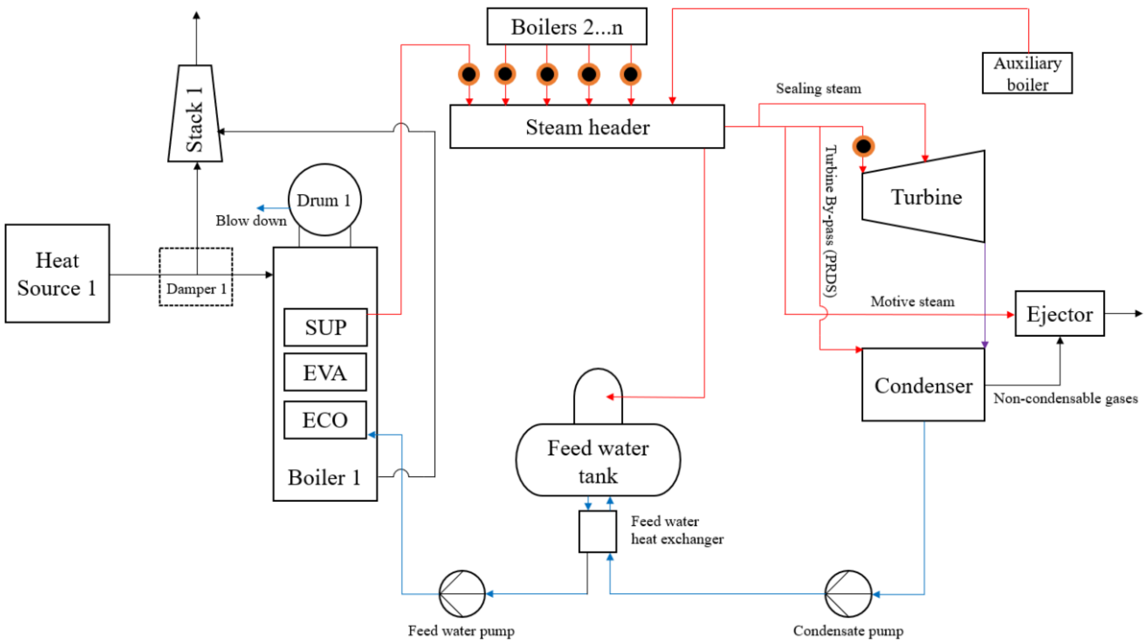

Figure 1. A combined cycle power plant.

Alfa Laval Aalborg Oy (ALA), to whom this thesis is done, is a market leader in WHR business. They offer WHRBs for combined cycle power plants amongst other applications. (Alfa Laval 2015a)

2.1 General WHR system overview

In a combined cycle power plant, the main steam generated by the HRSG drives a turbine which is connected to a generator. The steam expands in the turbine releasing thermal energy which is converted into mechanical energy. The turbine exhaust pressure is determined by a steam condenser and the two main parameters affecting to the pressure are the coolant flow and conductance of the condenser. In addition to the condensing performance the steam expansion is guided by the steam quality. As steam expands, it crosses the saturation curve and starts condensing spontaneously. These tiny water droplets hit the fast-moving turbine plates causing erosion and may potentially even break the turbine. The steam turbine will be covered in more detail in the following chapter.

Figure 2 shows the required exhaust gas temperature for different main steam pressures to prevent excess wetness in the turbine-end for one pressure level system. The calculation is done with turbine outlet steam quality x of 90%, condenser pressure of 80 mbar and isentropic turbine efficiency of 87 %. The required exhaust gas temperature is then calculated with sufficient 20 °C temperature difference to the main steam. As clear from the figure the required main steam temperature rises with the pressure. For one pressure level boilers the 300 °C temperature is required already for pressure of 6 bar(a). With a typical gas fueled ICE exhaust temperature this set-up could utilize around 11 bar(a) steam. The temperature of 520 °C, which encloses to the maximum possible temperature recoverable from the GT exhaust gases, would be required for around 30 bar main steam pressure. As the expansion enthalpy difference is greater with higher turbine inlet pressures there is more energy recoverable by the turbine. However, with one pressure level the pinch point at the exit of the evaporator dictates the achievable heat recovery from the exhaust gas. With higher evaporating pressures the exhaust gas outlet temperature would remain higher and thus less energy would be recovered due to higher evaporation temperature.

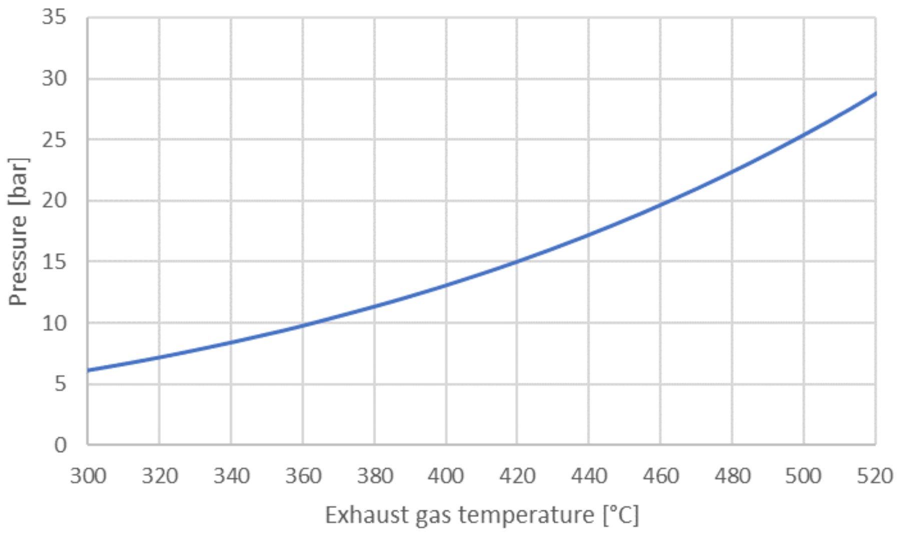

Figure 2. Required main steam pressure by exhaust gas temperature for one-pressure level system.

With reheating the steam, the main steam pressure can be increased. The reheating can be done in different ways, but with a combined cycle power plant an especially tempting opportunity is to add a second pressure level to the boiler and then inject the superheated low pressure (LP) steam to the turbine and mix it with the already expanded high pressure (HP) steam. Thus, the penalty of higher exhaust gas outlet temperature can be mitigated by further recovering the heat with the LP evaporator and LP economizer.

A similar calculation is conducted for a two-pressure level system. The LP steam pressure is set to vary such that the LP steam flow rate stays constant at 35 % of the HP steam flow. Thus, as the pressure and flow rate through the turbine are interconnected (Traupel 2001), the feeding pressure is determined. HP steam temperature is assumed to be 20 °C below the exhaust temperature and the LP steam is further 20 °C cooler than the HP steam. Figure 3 shows how the required main steam temperature relates to the HP drum pressure.

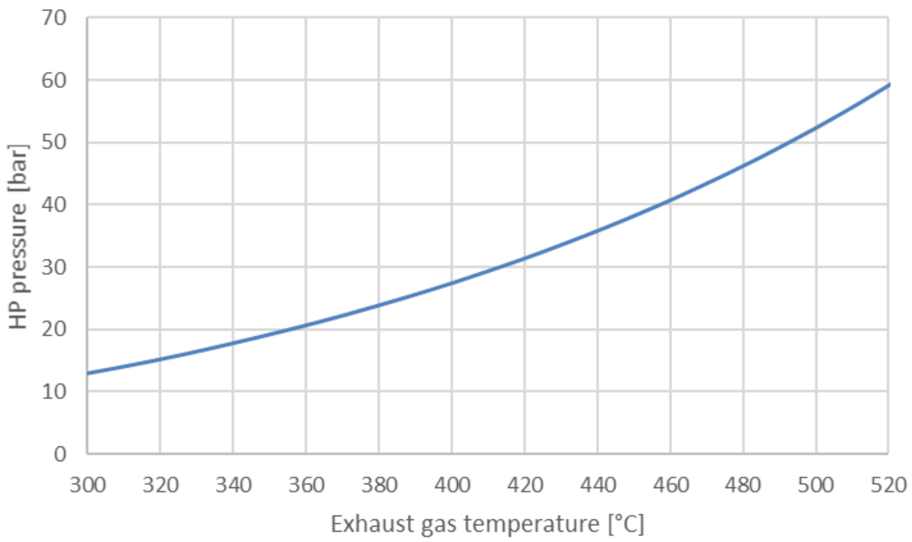

Figure 3. Required main steam pressure by exhaust gas temperature for two-pressure level system.

It can be noted that with adding a pressure level for 300 °C exhaust gas temperature the pressure can be increased from 6 to 12 bar. For typical ICE applications the pressure increase would be from 11 and 22 bars for one- and two-pressure systems respectively. For GT applications pressures as high as 60 bar(a) could be used. As for the recoverable heat the main steam pressure doesn't anymore have as significant effect as the low-pressure section predominantly dictates the outlet EG temperature. However, depending on the system design, the LP feeder steam pressure also varies a bit with the HP pressure, and it also has an effect to the total recoverable heat.

Yet, the thermodynamic process efficiency is often improved with the increased pressure as more energy can be extracted from the steam. However, to extract more energy out of the steam it must also contain more energy i.e. the steam should be hotter. Thus, the main steam pressure and temperature are interconnected. The total performance effect depends on the design of the system, so it is difficult to give a universal answer for the performance gain, but previous studies have been conducted about the issue and generally higher HP pressures and lower LP pressures correspond to greater efficiency as can be seen from Figure 4.

Figure 5 shows the calculated total efficiency of a combined cycle plant with one pressure level steam system. Although the exhaust temperature is left higher with higher pressures the gains with the process efficiency more than make up for it. Note that efficiency scales for figures Figure 4 and Figure 5 are not 0 - 100%, but are adjusted to showcase just the difference.

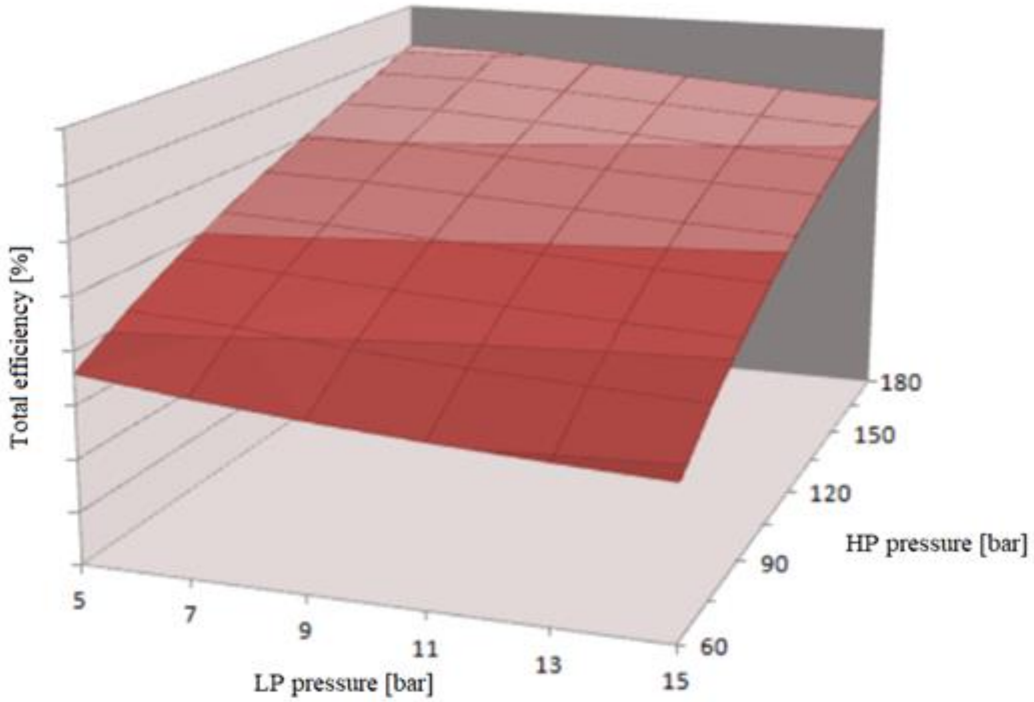

Figure 4. Pressure level effect to the total combined cycle plant efficiency with two pressure levels. (Llorca 2018)

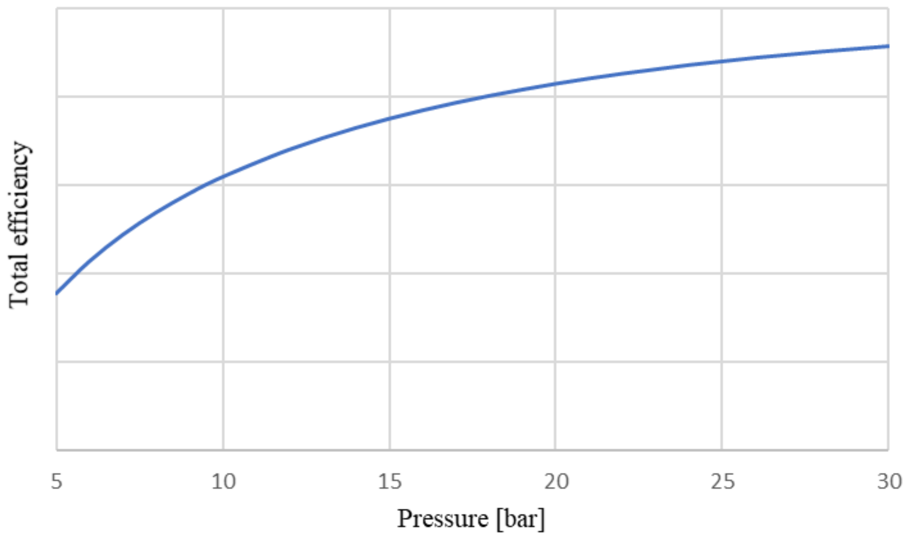

Figure 5. Pressure level effect to the total combined cycle plant efficiency with one pressure level.

Even though adding the second pressure level improves efficiency it also adds complexity and as the matter of fact the turbine requires the LP steam injection for reheating the mixture.

Hence, at the boiler start-up even in situations when the HP circuit would be ready to start ramping the turbine the system might have to wait for the LP circuit to be ready for ramp-up. In general, adding the second pressure level increases the main steam pressure. While great for efficiency, it means that the water in the HP circuit must be driven into higher temperature during start-up which would increase the start-up time.

Higher pressures also require thicker materials which are more prone for thermal stresses caused by uneven heating of the material. This could require tightening the interlocks regarding the boiler heating and thus expand the start-up time significantly. Also, the energy needed for heating the thicker materials increases the start-up time from pure thermodynamic point of view.

The European standard SFS-EN 12952-3 determines that the temperature difference through the wall of a drum or a header should not exceed 30 °C. For ferritic steels with material thickness below 32 mm this condition is met if the exposed heat flux is below 40 kW/m². (Finnish standards association SFS 2011) The same standard also determines that boiler components that undergo over 500 cold starts are considered to be exposed to cyclic loading. This brings an added restriction to the allowed heating rate as it provides an allowable temperature gradient [K/s]. (Finnish standards association SFS 2011). As the steam drum gains temperature fairly slowly these limitations don’t practically affect it, but the steam headers after the superheater might require a dampened warm up period. The boiler is considered cold after around 48 h of stand-still, but the steam header cools down considerably faster than the steam drum and thus undergoes a cold start more frequently.

2.2 Turbine overview

As told previously the role of the steam turbine is to transform the thermal energy of the steam to mechanical energy. This mechanical energy will thereafter be converted to electrical energy by the generator. The steam turbine is one of the most important parts of the total waste heat recovery.

The steam turbine efficiency is often measured with isentropic efficiency. This means the ratio of the actual enthalpy drop in the turbine compared to an ideal expansion. The losses inside the steam turbine cause irreversibilities so the resulting exhaust enthalpy will be greater as illustrated in Figure 6.

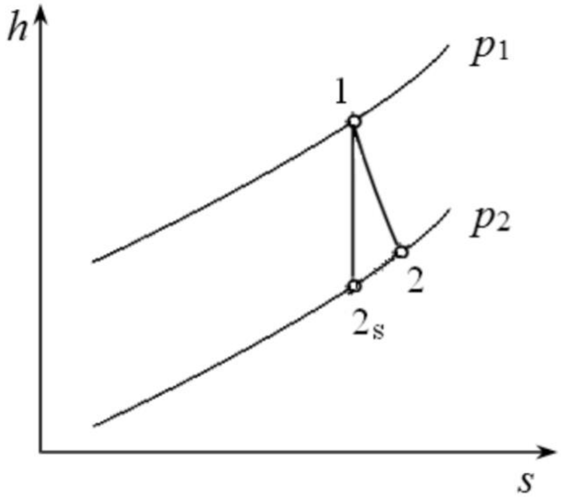

Figure 6. Steam expansion in a steam turbine (Ziolkowski et al. 2013)

The isentropic efficiency of a steam turbine is usually around 80 - 90% depending on the design. The turbine efficiency behavior on part-load situations, however, is difficult to assess as the scientific literature has rather mixed indications.

Tveit el al. (2015) claimed in their research that the turbine working blade efficiency stays rather constant until a relatively low load, but the regulating stage efficiency would start to decrease quite significantly after 75% load and the exhaust losses in the turbine end would also start to increase rapidly beneath the same load.

Dosa and Petrilean (2013) found that while the intermediate pressure turbine, which has similar inlet steam pressure as regarded in the scope of this work, suffers some efficiency loss during part load operation, but low-pressure condensing turbine gained some efficiency.

Thus, for the scope of this work, in absence of better knowledge the steam turbine isentropic efficiency is regarded constant at 87%. This assumption shouldn't have too big of an impact on loads above 50%, but most likely in reality the efficiency would start to drop significantly beneath that.

Besides the variations with isentropic efficiency the turbine part-load efficiency is affected more by the control scheme. The control scheme describes how the turbine power is governed. The most common alternatives are throttle control, nozzle group control, sliding pressure control and modified sliding pressure control.

Throttle control is the simplest of those alternatives. Before the turbine inlet a throttling valve governs the main steam flow by inducing a pressure drop. This isenthalpic expansion across the valve regulates steam flow through the turbine and thus the power. However, the throttling causes great throttling losses.

Another method is the nozzle group control also called partial admission control. Instead of single throttling valve the steam flow is controlled with a set of valves that operate sequentially. Through these valves the steam will enter the control stage known as the Curtis wheel through nozzles as shown in Figure 7. The Curtis wheel is an impulse turbine and the losses caused by governing are mainly due to the decreasing part load efficiency as found from Tveit et al.'s (2005) work.

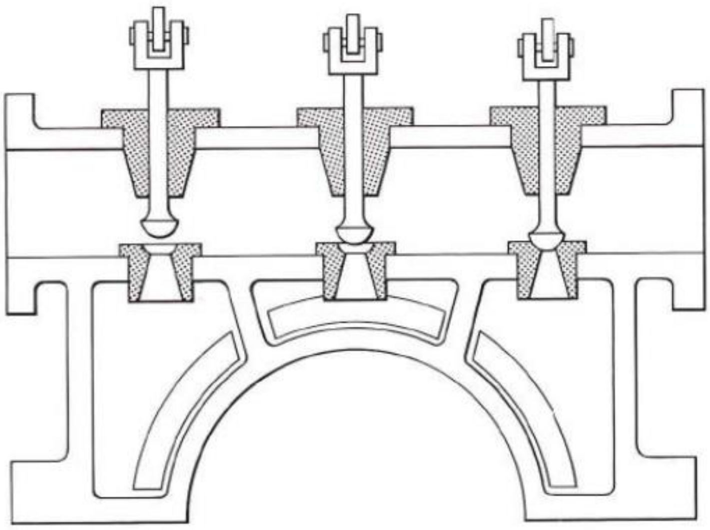

Figure 7. Illustration of a nozzle-group control set-up (British Electricity International 1990)

The third method is called the sliding pressure control. With pure sliding pressure the turbine doesn't have any regulating valves, but the steam flow through the turbine is adjusted purely by main steam pressure. Due to this the boiler operates under lower pressures on part load and as found out earlier this has the added benefit of increasing the boiler efficiency as more heat can be extracted from the exhaust gas. This is the reason why sliding pressure control is very common with HRSG applications.

Besides pure sliding pressure modified sliding pressure control mixes the characteristics of the pure sliding pressure and throttle or nozzle-group control. Up till certain load the turbine power is governed by the regulating valve and beneath that load the drum pressure starts to decrease until minimum threshold pressure from where the regulating valve(s) take back control as shown in Figure 8.

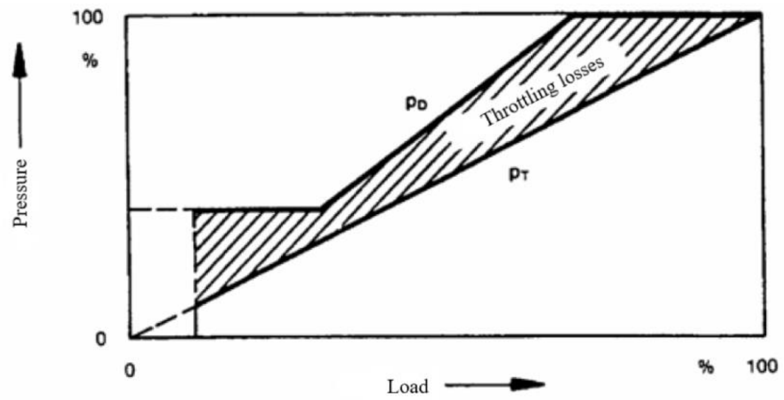

Figure 8. Control characteristics of a modified sliding pressure control scheme. (Kaikko 2019)

All different control schemes have different characteristics when it comes to part-load efficiency, response control and start-up features. Polsky (1982) has provided a graphical illustration to steam turbine part load power generation for a combined cycle power plant shown in Figure 9 where points B and C show constant pressure operation points and B' and C' the points for sliding pressure operation.

From the graph we can see that the steam turbine output is significantly higher at part loads with sliding pressure control thanks to increased steam production. The graph compares throttle governing to pure sliding pressure so constant pressure nozzle group control and modified sliding pressure methods would efficiency wise be somewhere in the middle of the two by mitigating the throttle losses with the nozzle group control or the slightly increased drum pressure for the modified sliding pressure control. Yet, for multiple pressure levels the sliding pressure control could cause problems as the LP section might not receive enough heat from the exhaust gases to provide sufficient steam for the injection (Polsky 1982).

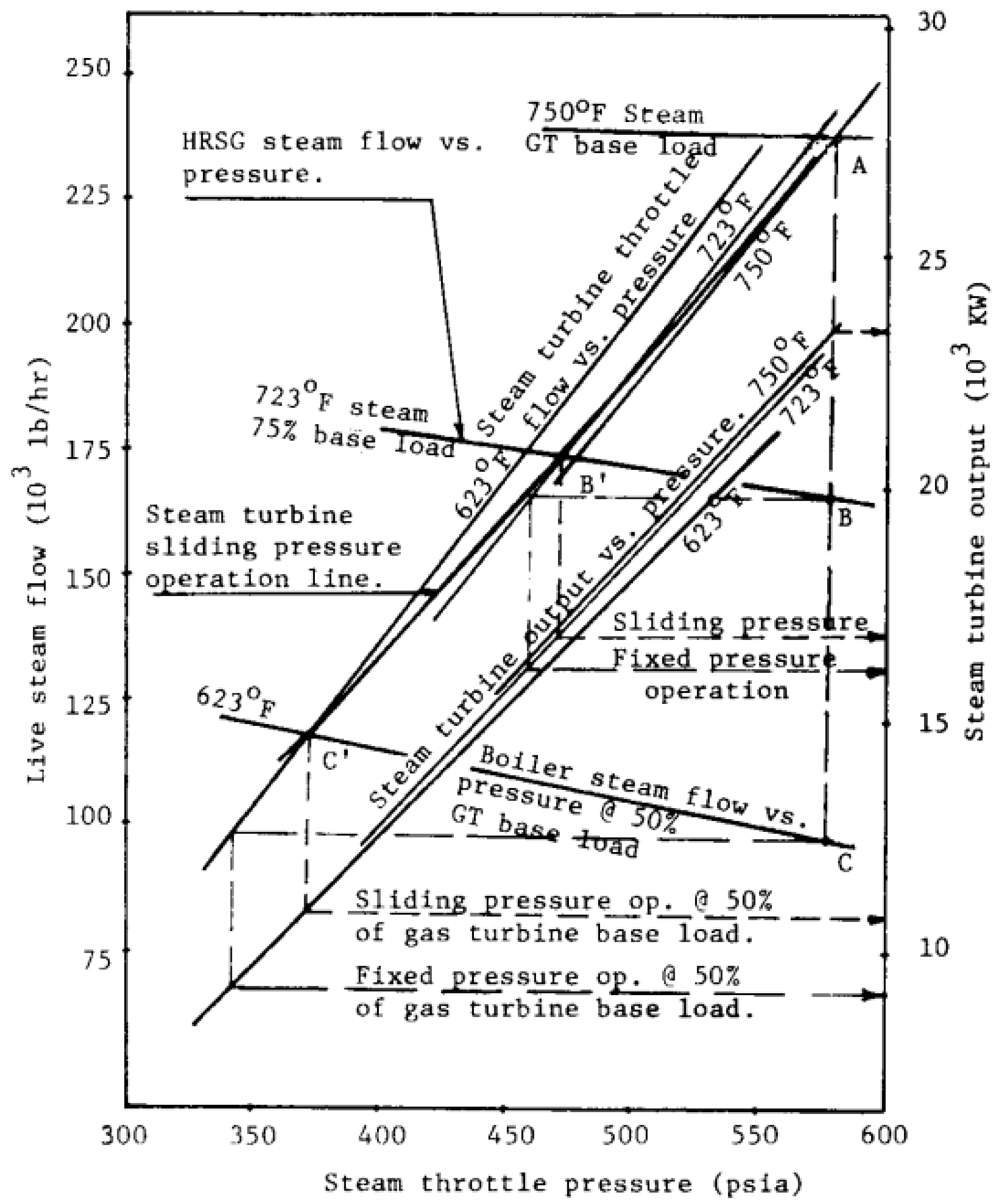

Figure 9. Steam turbine output in part load operation. (Polsky 1982)

The response affecting properties of the different control systems are shown in Figure 10. As clear the constant pressure operation has by far the best thermal response properties as it is mainly a question of the regulating valve position. The modified sliding pressure control can also achieve quick sudden changes by valve control, but as the boiler must increase its pressure to achieve the total change it takes time. With pure sliding pressure the response is naturally slowest as the change is only dependent from the drum pressure. The start-up properties will be assessed later in this work and a simulation is done regarding the rate of load increase in case of modular combined cycle block.

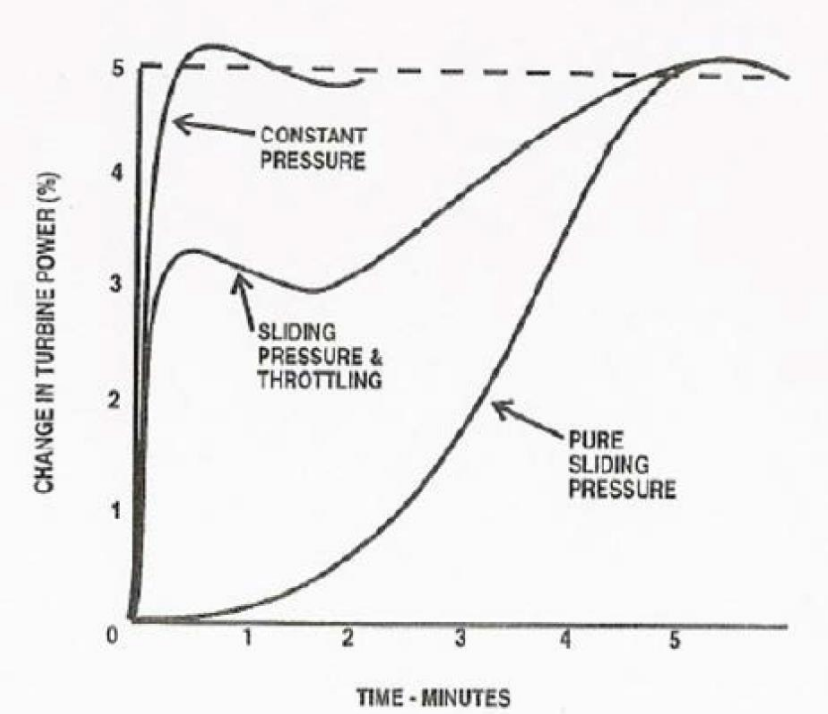

Figure 10. The response effect of turbine governing system. (Cotton 1994)

3 SECONDARY CYCLE START-UP SEQUENCE

Start-up procedure of the steam boiler follows a set sequence which is sort of a set of interlocks which need to be fulfilled before proceeding to the next step of the sequence. The sequence interlocks are set values which can be manipulated, and they affect the start-up qualities. Often the purpose of the interlocks is to protect the boiler and related piping from thermal stresses, but they can also be there to, for example, prevent excessive drum level fluctuations. The sequence differs depending on the state of the plant. For example, if the turbine is running the steam header is in its operating state a boiler is connected to it after it has gained its operative state. But if the turbine isn't operating and the header and main steam line are cold then the start-up sequence is altered to also heat up the whole main steam system.

3.1 General boiler start-up

The combined cycle power plant start-up sequence works like that first the primary cycle is turned up. The primary cycle provides hot exhaust gases, which are used by the HRSG, or it can be by-passed by an EG damper. With by-passing the HRSG the primary cycle can start-up without having any restrictions from the boiler. This, however, decreases efficiency, as the heat in the EGs is not utilized so ramping up the secondary cycle as quickly as possible is important for the total efficiency.

After the shutdown the boilers are isolated from the header inlet and economizer feed water control such that there is no flow in or out of the boiler, so it is in a closed state although in need of a pressure reduction a steam blow out is conducted. The drum pressure starts slowly decreasing due to heat losses which cause the steam inside the system to condensate to water that has smaller specific volume forcing the steam to expand and thus reducing pressure and keeping the saturated state. The boiler is also equipped with drains that drain the pipes from the condensed water. Through these drains some water is lost reducing the drum pressure furthermore as steam must fill the volume that has become vacant.

During start-up the boiler is as it is after the shutdown. If the shutdown has been prolonged the boiler usually goes into stand-by mode where the steam drum is heated to maintain an overpressure if such system is at place. But for hot starts the drum is still under higher pressure. When exhaust gases are admitted to the boiler the superheater might experience really high temperatures as it is the first heat surface the exhaust flow sees and because inside the superheater there is almost zero steam flow as the only stream out of the boiler steam system is through the drains. If the exhaust temperatures are high the steam blowout valve might have to be opened a bit to provide the superheater some cooling steam flow. This, however, increases the start-up time as the blowout has a pressure reducing effect. The blowout valve must in any case be opened when a set pressure is reached. This is done to preheat the superheater line and to also ensure that the line is free from any condensate.

As hot exhaust gases are fed to the evaporator the water inside the evaporator tubes starts to boil and due to buoyancy forces starts to rise towards the steam drum via the riser pipe. Initially the steam bubbles displace water in the evaporator and the riser causing the water level in the drum to rise, but more importantly the rising bubbles drag water upwards as well. To replace the ascended water more water is flown to the evaporator via the downcomer. This phenomenon is known as the natural circulation. As water evaporates in a closed system it causes pressure increase as the steam has higher specific volume than water forcing the steam to compress. However, as the pressure rises so does the saturated temperature of water. Hence not all the heat that is put into the system goes to evaporating the water but to also heat the water. With rising temperature so does the specific volume of steam and water increase boosting the pressure build up even further and rising the drum water level.

The drum level control is important as if the drum level rises too high it can cause damage with water carryover to the superheater or if the level dropped too low, it would expose the connected water tubes risking cracks and breakage. Due to interlocks the boiler would automatically trip if the water level went too high or low. Void fraction describes the percentage of steam in the evaporator and the riser volume wise and it is dependent on steam quality and steam pressure. Increasing the evaporator heat input initially increases the void fraction pushing the drum water level up. This effect is known as the swell and the opposite as the shrink. The lower the steam quality and steam pressure the bigger the impact of swell and shrink are. (Waite 2012) This swelling phenomenon is most clearly noticed before the natural circulation starts to act properly. Hence, upon the boiler start-up the damper position is usually restricted at the beginning to let the circulation balance and to prevent any excessive swelling effects. This also protects the boiler from abrupt thermal shocks. The swell could also happen during abrupt changes in steam flow such as a steam blow out.

During boiler shut down the drum water level is set to the idle water level (IWL) which is often between the normal water level (NWL) and the level alarm low (LAL). As initially the evaporator and riser are filled with water the steam quality is zero and as the drum pressure is low the swell effect is at its greatest. This may cause the water level to rise rapidly upon start-up and the drum should be dimensioned such that the resulting water level doesn't surpass the level alarm high (LAH). Besides swelling and shrinking the drum dimensioning should also take into consideration the surface vapor velocity and distance for gravity separation to avoid problems caused by droplet carryover (Yoshihara 1990). The maximum surface velocity is dependent on criteria such as pressure, geometry and most importantly the steam separation equipment (Campbell 2015). In practice this means the liquid-water interface area and the distance above the water surface. In other words, the steam space should be large enough. The drum should also be able to fit the internals and have sufficient space for maintenance actions etc. However, as the drum hosts a major part of the volume of the closed boiler cavity and thermal capacity and hence inertia it plays a major part in boiler ramp up properties and to allow faster start-ups the drum size should be kept as small as possible.

3.2 HRSG sequence with steam turbine running

In a case when connecting a boiler to an already operating header the sequence interlocks only consider the superheater line temperature and the drum pressure. Hence, to allow for fastest possible start-up the superheater line should achieve the required temperature simultaneously as the drum reaches minimum pressure, if connecting with an overlap between blow out and header inlet valve. However, the overlapping isn't necessary and often due to high blow out rates and low pressure gain the blow out is closed after reaching satisfying temperatures to let the boiler gain pressure more rapidly. Then after the drum pressure reaches the header pressure the connection happens. The header inlet valve can also be opened even before the header pressure is achieved, but then the resulting steam flow would be from header to the boiler. Even though it would pressurize the boiler very quickly it could also disturb the turbine steam flow and in the worst case even trip the turbine.

The beginning of the blow out sequence often coincides with the switch of drum level set point from IWL to NWL. As the set point switches the feed water control system starts to open the feed water control valve. As the drum is not yet up to pressure, but the feed water system is, there is a large pressure difference in the feed water piping and economizer pushing water through. This high flow rate of cool feed water causes the drum pressure to initially decrease as steam is used for heating the feed water instead of gaining pressure. The addition of the cold feed water collapses some of the steam bubbles in the water-steam mixture and cause a shrink effect (Liptak 2006) so it somewhat contradicts the possible swelling effect caused by the blow out start.

3.3 HRSG sequence with turbine shut

When firing up the first boiler the header inlet valve can be kept open to include all the main steam line and the header to the boiler cavity. The blow out will then be conducted through a valve in the main steam line. Thus, the superheater line, the steam header and the main steam line are all heated up in unison. Another option besides blowing out the steam is to by-pass the turbine and run the steam straight to the condenser through a pressure reducing desuperheater (PRDS). This has the added benefit that no water is lost from the system which would then need to be replaced with demi-water.

As the pressure increases in the drum the boiler is connected to the steam turbine. There are some interlocks which must be fulfilled before connecting. These could be like the steam piping must be above the saturated temperature by some margin and the drum pressure must be above a set limit. The boiler start-up sequence could also have some overlapping with the turbine sequence. For example, sealing steam could be fed to the turbine after a set pressure and thereafter also steam ejectors could be switched on to create a vacuum to the condenser. These two alone would already provide some heating steam flow through the steam piping.

3.4 Turbine sequence

If the steam turbine is already running the steam can be fed into the turbine to produce power. To increase the steam flow through the turbine its inlet pressure must also be increased according to the ellipse law which can be written as

$$q_m = q_{m0} \frac{\sqrt{(pv)_0}}{p_0} \frac{p_1}{\sqrt{(pv)_1}} \tag{4}$$

assuming that the turbine section outlet pressure is significantly lower than the inlet pressure. This is the case almost always as the result depends on the subtraction of the squared inlet and outlet pressures. (Traupel 2001) The rise in inlet pressure can be achieved depending on the control scheme either by increasing the drum pressure in all connected boilers or by adjusting the throttling. However, if the turbine also requires a start-up a sequence it must be carried out.

The turbine start-up is often the bottleneck of the whole secondary cycle as can be seen in Table 2. It is said that the steam turbine (ST) start-up is done in economic, normal and fast modes for hot, warm and cold starts respectively. If the hot start-up was done in the fast mode, then the GT would reach full load after 31 min instead of 47 min. In this example the GT waits for the secondary cycle instead of ramping to full power as soon as possible which is the reason why it is kept at part load for excessive amounts of time and why the ST start-up mode affects the GT start-up time. While good for efficiency a lot of GT capacity is wasted during the start-up sequence.

Table 2. Start-up sequence timeline of a CCP. Times in minutes. *) 14%, **) 2 min stop during speed up. (Kehlhofer et al. 2019)

	Hot start	Warm start	Cold Start
GT to FSNL	0 - 5,5	0 - 6	0 - 7
GT Load increase (0 - 8%)	5,5 - 8,5 *	6 - 8	7 - 9
GT Load held (8%)	8,5 - 10 *	8 - 21	9 - 26
GT Load increase (8 - 38%)	10 - 15 *	21 - 28	26 - 32
GT Load held (38%)	15 - 33	28 - 81	32 - 113
ST to FSNL	23 - 27	50 - 54	72 - 78 **
ST Load increase (0 - 58%)	27 - 33	54 - 81	78 - 113
GT Load increase (38 - 100%)	33 - 47	81 - 107	113 - 155
MS pressure increase	33 - 55	81 - 114	113 - 165
MS temperature increase	33 - 47	81 - 176	113 - 205
ST Load increase (58 -100%)	33 - 55	81 - 176	113 - 205

The ST start-up sequence explains why it is the restricting part of the CCP start-up. From a standstill first the rotor is rotated with a barring gear at slow speed e.g., 50 rpm. The barring is done to ensure that the rotor heats up axisymmetric and the thermal stresses do not cause any bending to it. The barring also creates heat inside the turbine casing due to a phenomenon known as turbine ventilation as especially the long last stage blades have already a significant peripheral velocity when they move though the stagnant fluid causing heat losses.

Unfortunately, this heat is created to the coldest part of the turbine so it doesn't really aid the start-up but can rather cause problems. (Banaszkiewicz 2014)

Next phase in the start-up is to admit sealing steam to the turbine labyrinth seals. The sealing steam is usually given some threshold values on pressure and temperature from the turbine manufacturer. In addition to sealing the turbine the steam also has a preheating purpose especially for the turbine rotor. After the turbine has been sealed the vacuum can be sucked to the condenser. The vacuum can be done either with a steam ejector or mechanical pumps. The steam ejectors too have a threshold limit for steam parameters depending on the design. As the turbine backpressure decreases so does the effect of turbine ventilation due to decreased fluid density inside the casing allowing higher rotational speed. Live steam can now be admitted to the turbine. (Banaszkiewicz 2014) If the sealing steam and ejector can operate with saturated or just slightly superheated steam it could then be provided by the boiler during its start-up sequence. However, to ensure that the line is free from any condensate some flow should be carried out through the PRDS or the blow out. Yet, if higher degrees of superheating are required it would increase the blow out time and thus the boiler start-up.

Prior to turbine start-up it must be checked that there is no eccentricity in the turbo-set nor any excess vibrations exist. When admitting live steam to the turbine the temperature gradients of different parts must be kept reasonable to prevent any excessive thermal stresses due to uneven heating. Table 3 shows exemplary warm up gradients which should not be exceeded to enable heat to conduct evenly and minimize thermal stresses. These limits are embedded to the turbine operating manual submitted by the OEM, and they can also be in other forms e.g., limiting the temperature difference between steam and metal etc. Also, during a hot start an inverse temperature gradient may occur as the main steam is cooler than the metal itself. (Banaszkiewicz 2014) The easiest way to control the temperature gradients is with the mass flow rate and quite often after synchronization the load increasing rate is restricted, but also things like desuperheating could be used. During hot start the easiest way to ensure appropriately high temperature is to conduct a long enough blow out or by-pass sequence so the main steam line is up to temperature before admitting main steam to the turbine. Usually, the larger turbines have slower start-up times as there is more metal to heat and due to higher material thicknesses also the thermal stresses associated with the process are higher.

As the turbine efficiency is often really low during minimum load and off-speed the exhaust steam quality is not an issue as the thermal energy of the steam isn't extracted. Thus, if the system would operate with two pressure levels and the lower pressure steam would be injected to reheat the resulting steam mixture the availability of that steam would only affect the very last moments of the turbine start-up providing that all auxiliary system steam could be taken from the higher pressure level.

Table 3. Permissible temperature gradients for turbine components [°C/min] (Banaszkiewicz 2014)

	Temperature range [°C]		
Component	< 200	200 - 400	> 400
Live steam pipeline	5	4	3
HP stop valve casing	3	2	2
HP control valves casing	6	5	3
HP inner casing	3	2	2
HP outer casing	4	3	3

The start-up timeline in Table 2 shows that in each start-up scenario the GT is held at 38 % load for a long time. For hot start this sequence lasts 18 minutes and it is said that by switching the economic ST operating mode to fast the start-up time could be enhanced by 16 minutes, but it is not determined how, but as the GT holding period is the bottleneck of the sequence it is the most likely source of improvements. In the economic mode the GT is kept at 38 % load to speed up, synchronize and to load the steam turbine to the maximum possible load attainable with the pressure and steam mass flow rate generated with that load. Only after that the GT ramp up is continued. Observing that the GT ramp rate is pretty much equal before and after the ST start-up and the fact that the live steam temperature is not tampered during the ramp up sequence most likely the time saving is done by not pausing the GT ramp up.

From Table 2 it can be noticed that only during hot start was the ST load limited by the mass flow rate and pressure the boiler could produce. On all other cases the live steam temperature had to be tampered to ensure turbine lifetime. This can be done by desuperheating i.e., spraying feed water directly to the live steam flow. Hence only during the turbine hot start did the boiler form the bottleneck of the total plant start-up. However, gas turbine and especially engine power plants are often arranged into blocks where multiple GTs and ICEs are connected to one header and even multiple headers can be connected to one steam turbine. Thus, often the primary cycle and the boiler are started up when the turbine is already in operation. In such cases the boiler forms the bottleneck for the start-up sequence.

4 THE START-UP SIMULATION TOOL

To simulate the boiler start-up procedure a start-up simulation tool is built with the Microsoft Excel. The tool combines dynamic thermodynamic formulas with a control sequence and in this chapter its main parts are presented.

4.1 Thermodynamic simulation

The simulation tool uses empirically proven ALA thermodynamic formulas to calculate the heat transfer between exhaust gas and the boiler surfaces. It furthermore compiles thermodynamic equations to simulate the dynamic boiler behavior. The essential boiler components are divided into sections and numerical calculations are conducted between these sections. Figure 11 shows how the boiler is divided.

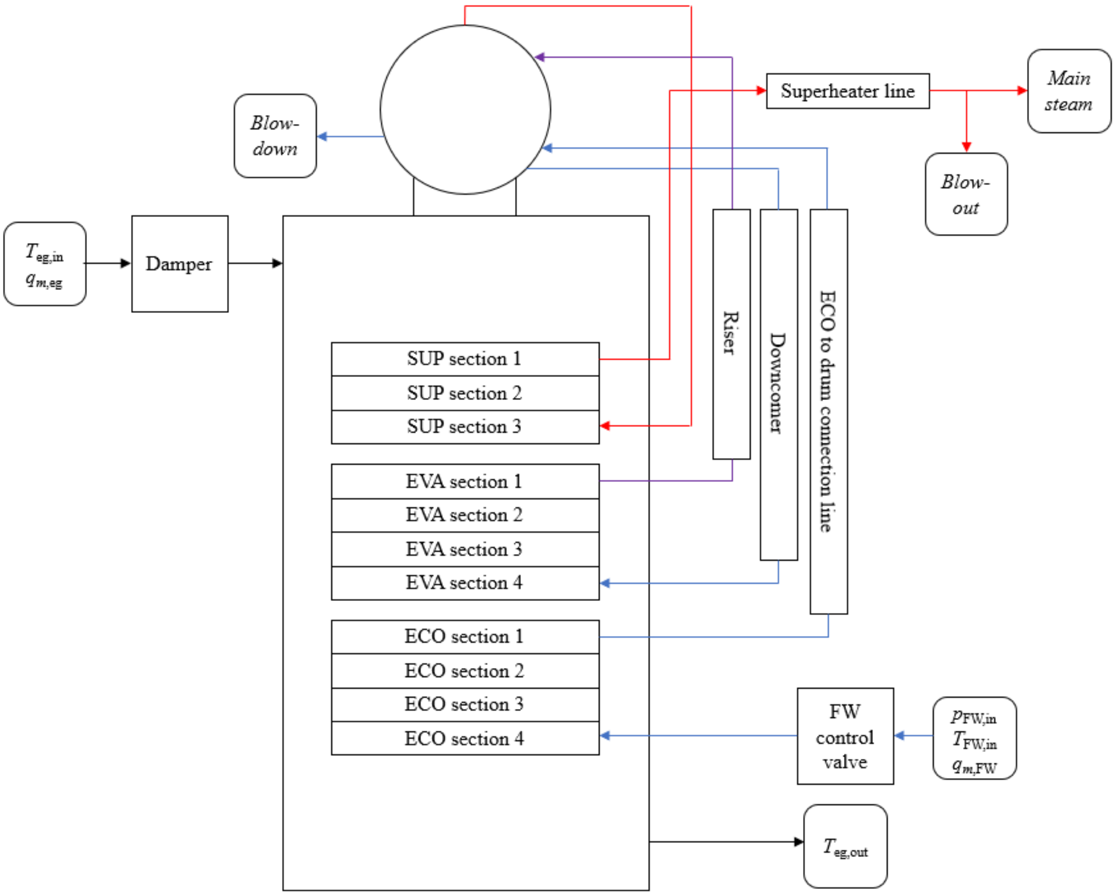

Figure 11. The boiler set-up for the simulator.

Each section forms a node and is assumed to have a uniform temperature and the same pressure though the static height is taken into account for the evaporator. Hence any pressure losses are not simulated in the system, but for feed water control and main steam flow the pressure difference between the feed water, header and drum are considered and the flow rates are adjusted accordingly.

From piping the economizer to drum (E2D) connecting pipe, downcomer, riser and the superheater line are included while the other parts are left out as they are deemed irrelevant. For example, the line connecting the drum to the superheater doesn't have a significant effect to the boiler as it doesn't host a significant thermal capacity and with its location before the superheater doesn't contribute to the superheater line heating process.

As an exhaust input values the tool requires mass flow rate, temperature and the composition of the fluid. The EG properties are calculated based on equations found from the Heat Atlas (VDI 2010). These values can be given through time in a matrix form in case of varying properties. The exhaust gas damper is thereafter simulated with a flow map based on data indications and discussions with the OEMs.

The superheater is the first surface to experience heat transfer. Figure 12 shows the heat and mass balance for a single section.

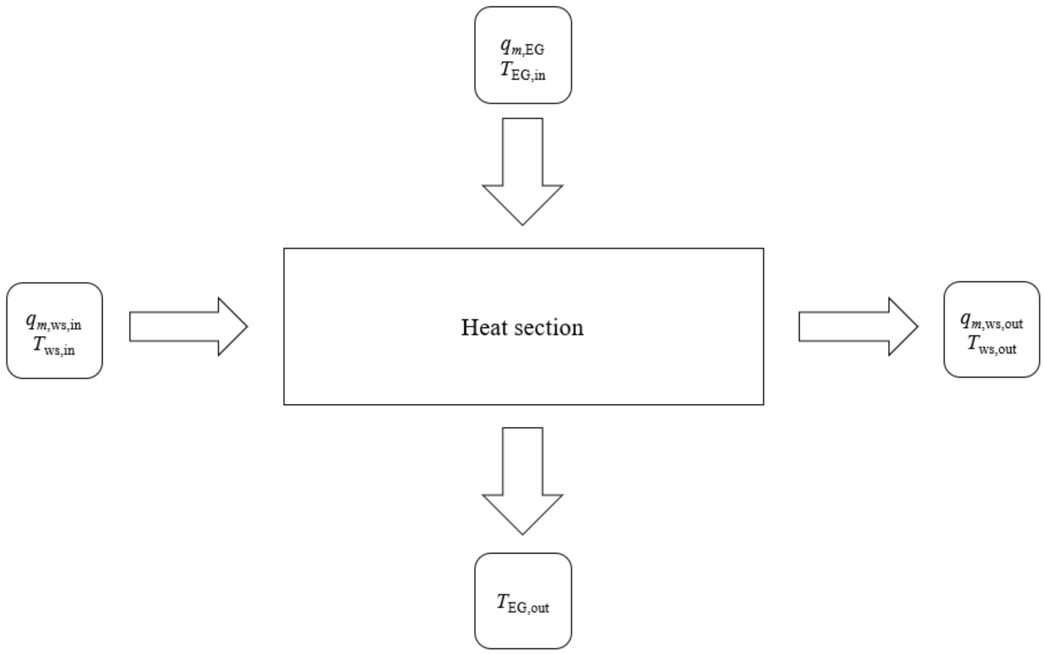

Figure 12. Illustration of the numerical method.

The simulator then solves the heat and mass balances. The EG mass flow through the boiler is constant, but for water and steam e.g. thermal expansion can cause non-equal mass flows between the inlet and outlet. This is especially notable in the economizer during the start-up sequence as the thermal expansion pushes water to the drum which can be seen as drum level rise. From the heat balance the average steam-water temperature is solved as well as the exhaust gas outlet temperature and they will thereafter be used as input for other sections.

To solve the incoming enthalpy of the steam or water so called first order upwind discretization scheme is used. As the heat balance only solves the temperature inside the node and not on the boundaries it is essential to determine the value to be used at the boundary. First order upwind means that the boundary value is taken from the node upstream meaning the inlet state is the same as inside the previous section and the outlet is same with the node's state.

Even though this might seem absurd but thinking of convection as a phenomenon, most of the heat transfer happens where the temperature difference is at its greatest. Hence it has a non-linear nature and for example, using average for discretization would create an even larger error. Figure 13 illustrates this. Many numerical software utilizes more civilized discretization method such as second order upwind or QUICK-scheme, but those require more calculative power and are more unstable, as the boundary value is dependent on multiple instead of one cell values.

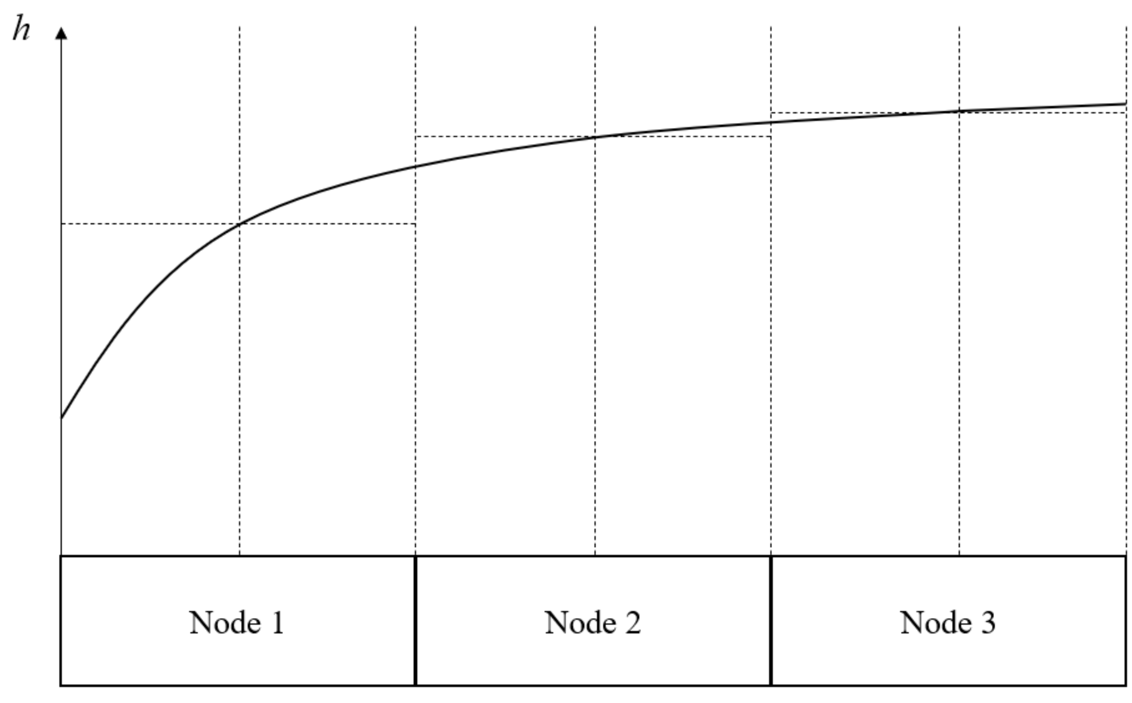

Figure 13. First order upwind discretization depicted.

The dynamic properties of the boiler are simulated in a semi-implicit fashion meaning that most equations use the values of the current time-step to and solve the set of equations iteratively. The opposite method is called the explicit method which utilizes the values of the last time to calculate the new time step. Semi-implicit means that while the tool utilizes implicit method primarily some sequence related values are taken from the previous time step as well as some key figures that are harder to converge such as main steam properties. The benefit of the implicit method is that it allows bigger time steps compared to the explicit method, but each step requires more calculation due to its iterative nature. Generally, the implicit method is a bit more accurate, but the best accuracy is often reached with the Crank-Nicolson method, which combines both implicit and explicit methods and uses the average to calculate the values of the new time step. This, however, is very calculative power consuming as it must calculate both the explicit and implicit methods and can’t benefit from larger acceptable time step of the implicit method.

The dynamic behavior is calculated with the heat balance as shown in Figure 12, but instead of being in a steady state the node has a thermal capacity depending on its steel weight and fluid that is inside. The tube wall and the fluid are assumed to have a uniform temperature. Although this is not ideal especially for the superheater, which is under minimal flow during start-up, it saves a lot of calculation power and should not form a drastic error when adjusted correctly. The dynamic heat balance is

$$T_{\mathrm{TS}} = T_{\mathrm{TS}-1} + \frac{\left(Q + q_{m,\mathrm{in}} h_{\mathrm{in}} - q_{m,\mathrm{out}} h_{\mathrm{out}}\right)\Delta t}{m_{\mathrm{f}} c_{p,\mathrm{f}} + m_{\mathrm{s}} c_{p,\mathrm{s}}} \tag{5}$$

Where T is the temperature,

TS is the current time step,

TS – 1 is the previous time step,

Q is the heat input from the exhaust gas,

q_m is the mass flow rate,

h is the enthalpy,

Δt is the time step,

m is the mass,

c_p is the specific heat capacity,

in symbolizes section inlet,

out symbolizes section outlet,

f symbolizes fluid and

s symbolizes steel.

The most important component of the tool is the drum as the system pressure is determined there. The steam pressure can be determined if its enthalpy and density are known. The steam drum is assumed to always be in equilibrium resulting in a saturated state. In reality the water might have a bit of subcooling at times, but the effect is neglectable. Hence the enthalpy of the water and steam is always assumed as saturated. The density of the steam depends on the volume of the steam and its mass inside the drum, evaporator and the riser pipe. The mass can be calculated from the mass balance while the volume can be calculated by subtracting the volume of the incompressible water from the total system volume. To calculate how much of the evaporator heat input goes to evaporating and how much to heating the steam, water and steel the resulting temperature of the heat balance has to be matched with the saturated temperature by adjusting the rate of evaporation $q_{m,\mathrm{ev}}$ through linear interpolation.

$$T_{TS} = T_{TS-1} + \frac{\left(Q - \left(q_{m,ev} - q_{m,ev,eco}\right)(h'' - h')\right)\Delta t}{m'c_p'(T_{TS-1}) + m''c_p''(T_{TS-1}) + m_s c_{p,s}} \qquad (6)$$

Where variables are as in equation (5) and,

$q_{m,\mathrm{ev,eco}}$ is the steam gain from the economizer,

' symbolizes water,

' ' symbolizes steam.

Figure 14 shows the system boundary of the steam drum. Like with other sections it is also expected to be in a uniform temperature. By setting the reference enthalpy to that of saturated water the blow down could be reduced from the equation. The $q_{m,\mathrm{ev,eco}}$ while described as the steam gain from the economizer can also have negative values if the feed water is below saturated temperature. Then it gives the amount of steam required to heat the feed water to saturated temperature. Together the addition $q_{m,\mathrm{ev}} + q_{m,\mathrm{ev,eco}}$ gives the net amount of produced steam.

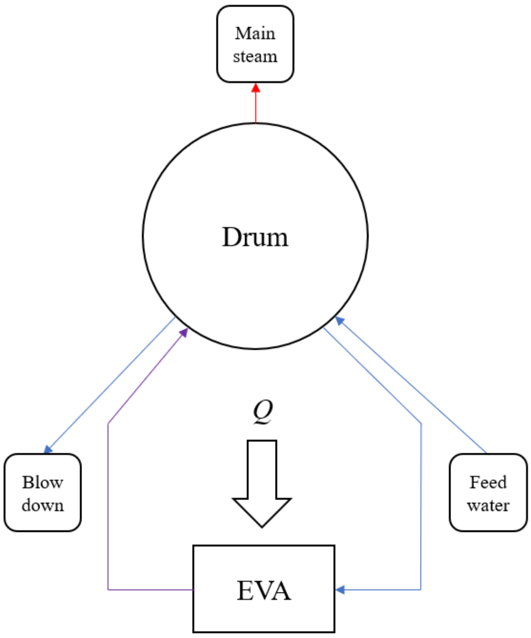

Figure 14. The drum system boundary.

To determine how much of the produced steam flows out of the drum and how much stays to increase the pressure is done by calculating the change in specific volume inside the superheater sections and the superheater line. As the pressure rises in the superheater line so does the saturated temperature and enthalpy and this energy has to come from the steam itself as there is no external source of heat. This means that during the start-up before the beginning of the blow out sequence some steam must condense to match the heat balance for the rising temperature. This water is then extracted from the system through drains and the vacant space is filled with steam creating a draft to the drum.

4.2 Sequence and control simulation

The boiler start-up is full of different sequences and to simulate the procedure they must be accounted. A simplified sequence was constructed, and it is shown in Figure 15. It is a simplified version of an actual sequence but has the same main parts. First row shows the action happening when entering the sequence. Second row shows the action happening during the sequence and the third row indicates the interlock for proceeding to the next sequence. First the damper position Y is limited to a set value to ensure that the boiler has warmed up appropriately before increasing the load and it has two interlocks with time T and pressure P.

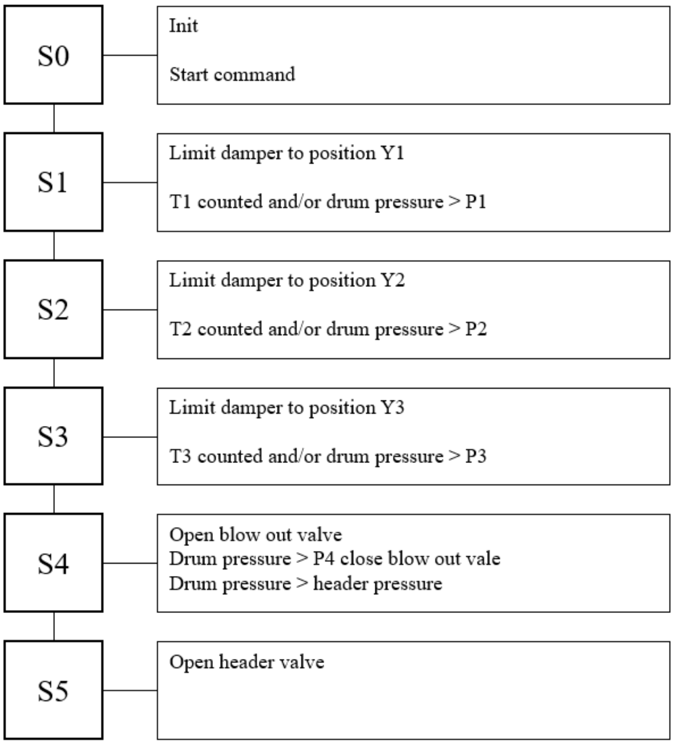

Figure 15. Simulated boiler sequence.

After the damper has been fully opened the boiler starts to gain pressure quickly and the blow out sequence begins at set P3 and carries on until set P4 is reached. Finally, after the header pressure has been reached, the boiler will connect to the header and start operating. These set values can be manipulated to match the sequence of a real boiler.

Once the boiler has been connected to the header the steam flow rate is calculated by the pressure differential between those two. When the drum and the header are at the same pressure there is naturally zero steam flow between and while the pressure difference matched with the nominal pressure loss of the superheater and piping a nominal steam flow goes through. These two points are the connected with a curve resembling a square root function as the pressure loss increases to square of the flow rate.

The feed water control is simulated with 1-point control scheme meaning it only follows the drum level. The drum level has two set points, IWL for the start and after a set pressure is reached the set point will switch to NWL. The feed water control valve will adjust itself based on three parameters, drum level tolerance, nominal control valve position and maximum control valve position. When the drum level is at the lower limit of the tolerance the control valve is at its maximum opening and when the level is at the upper limit the valve is closed. The nominal position will pass the amount of feed water equal to the nominal flow rate. The feed water flow is adjusted based on the control valve position, which also takes into account the travel time, and the pressure difference between the feed water line and the steam drum. Hence the feed water flow rate can exceed the nominal rate multiple times if the maximum valve opening isn't restricted and there is a large pressure difference.

The drum level is assumed to be only the factor of how much and how dense water is inside the control volume illustrated in Figure 14. This means that level fluctuations caused by events such as load increase or blow out opening aren't simulated as they would be almost impossible to calculate, and the empirical data shows very inconsistent yet rather small effect. In reality if the effect had a significant magnitude, it would also affect the feed water control system.

4.3 Validation

The tool validation process is conducted by comparing the achieved results to measured data and the total boiler heat balance is also evaluated by comparison to values calculated with iPro the ALA dimensioning tool.

The most important measurement is the drum pressure as it basically shows how much thermal energy is "loaded" inside the boiler. The drum pressure also dictates when the boiler can be connected to header and start producing steam or in case of the turbine sequence it controls when steam can be fed to the turbine sealing system and vacuum system. Figure 16 shows the calculated pressure curve with comparison to measured values.

During the initial curve the pressure gain is rather moderate due to limited damper position as well as the cold superheater tubes above the evaporator which take time to heat up. Thereafter a rapid gain in pressure can be noted until there is a clear dip. This dip coincides with the opening of the blow out valve, but in the matter of fact the dip is caused by the switching set point of the drum water level causing vast amounts of cold feed water be fed into the drum as illustrated in Figure 17.

After the dip the blow out sequence continues which can be seen as a rather straight pressure curve. The blow out is simulated with a constant volume flow rate as it happens through a choked valve and the results indicate that it is a correct assumption. Finally, the blow out is closed and the drum gains pressure a bit before connecting to the header.

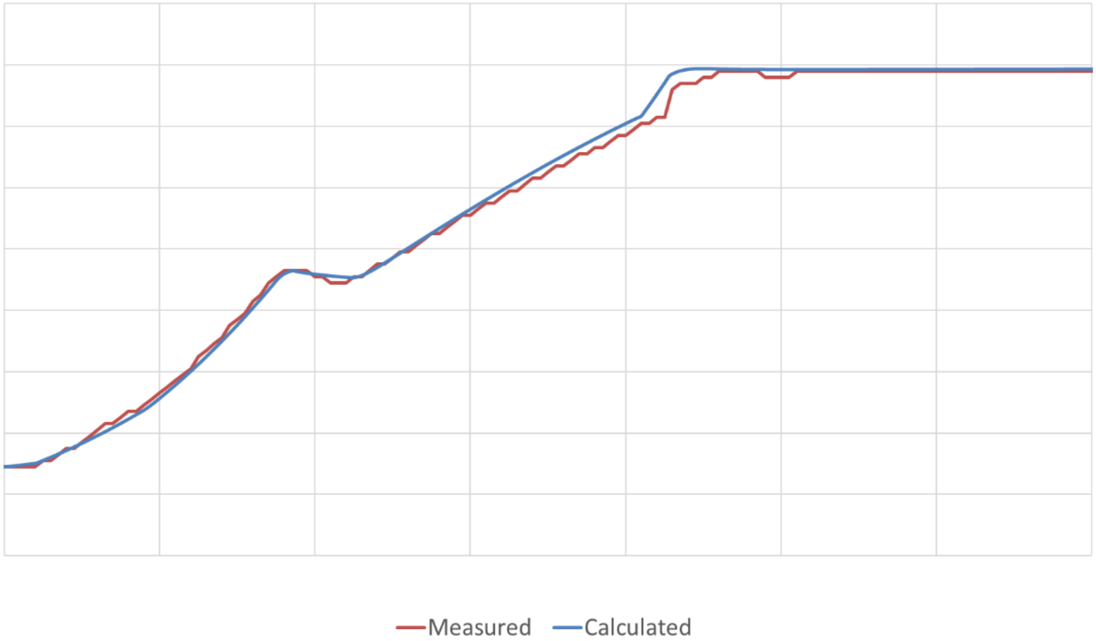

Figure 16. Drum pressure comparison.

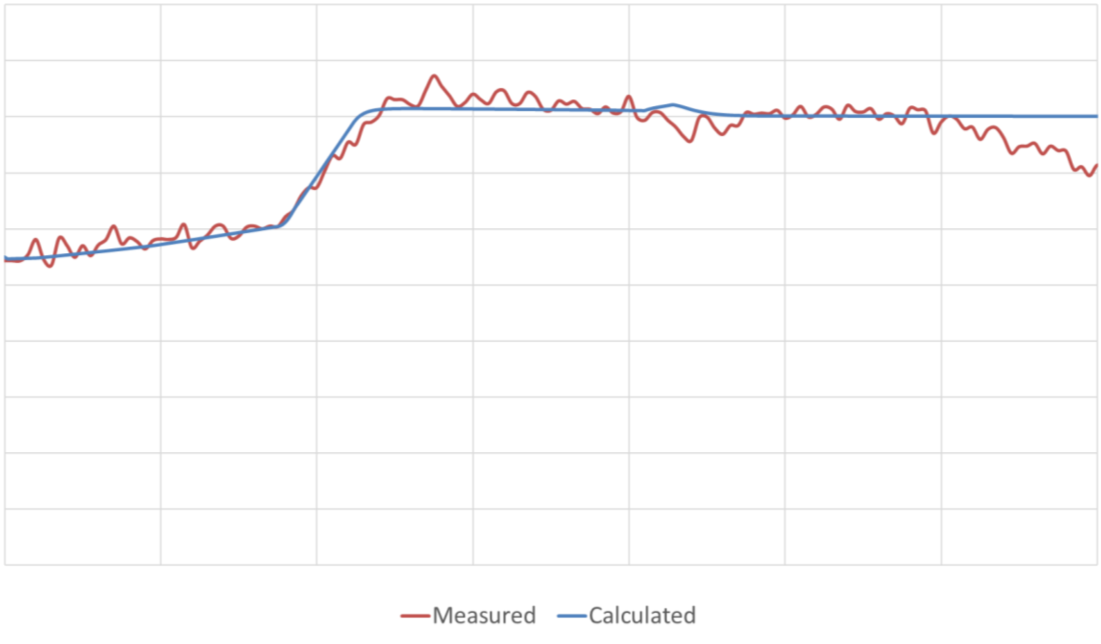

Figure 17. Drum level comparison.

Even though the drum level is a result of many factors but just the amount and density of water in the control volume if combining information between Figure 16 and Figure 17 it is clear that the tool excels at simulating the phenomena during the start-up period.

To evaluate the heat balance calculations the results are compared with the calculations conducted with iPro and measured values and the results are shown in Table 4.

Table 4. Heat balance comparison

	EG outlet temperature [°C]	Steam temperature [°C]
Simulator - iPro	-0,4	0,9
Simulator - measured	-19,1	-4,0

The iPro heat balance is calculated by matching the steam generation with the same input values by varying the effect of fouling. By doing this the differences caused by different methods of calculation can be inspected. As can be seen there are no drastic differences caused by these differences. However, by comparing the measured values a much greater difference can be seen. However, inspecting the measurement data it can be noted that the outlet temperature seems to be higher than what thermal heat balances would suspect. As the heat balance for exhaust gas and steam generation are quite simple to calculate it would indicate that probably the temperature sensor is poorly located, or it suffers some level of disturbance. Often the measured values are in line with the balance calculations so the big difference in outlet temperature shouldn’t be of great concern. Conducting the same research for another boiler gives a lot closer parameters between iPro and measurements.

5 EFFECTS OF BOILER SOLUTIONS

The build simulator is used to analyze the effect of different solutions regarding physical design of the boiler and the start-up sequence. The physical start-up properties are mainly depended on the thermal inertia. The thermal inertia is created by steam, water and steel inside the boiler cavity as they have heat capacity. The sequence has a more direct effect on the start-up as it controls what the boiler does.

5.1 Physical solutions

If the pressure generating section of the boiler as in Figure 14 is thought to compound of three parts, evaporator, piping, and the drum there isn't a lot to be done with regards to thermal inertia with the evaporator or piping. The evaporator needs a sufficient surface area for heat transfer. Generally, smaller tube sizes have more heat transfer area compared to weight than larger tubes, but with natural circulation the tubes can't be too tight to allow proper circulation. Same is true for the riser and to some extent the downcomer.

The tube finning has an effect on the thermal inertia of the boiler as fins improve heat transfer and thus enable smaller heat sections. Although, the fins also add weight the total steel mass is smaller with denser finning. However, the fin density is rather dependent on fouling than thermodynamic performance.

The steam drum serves a vital role for the total system operation. As explained earlier the drum needs a sufficient steam space to prevent droplet carryover, but it also enables safe room for swelling and shrinking phenomena. As told previously, those effects are strongest while drum pressure is low, and the natural circulation isn't yet flowing properly. Those situations usually happen during the early start-up and are mitigated by setting the drum level set point lower for idle situations. Thus, the boiler heat input is often restricted until sufficient pressure has been achieved prolonging the start-up significantly. However, it also hosts vast amounts of thermal inertia mainly with the water, so it has a significant effect on the total start-up time.

Figure 18 shows the start-up curves for the same boiler fitted with different sized drums. The last number of the name shows the drum volume in m^3. As can be seen, increasing the drum size increases the start-up time. However, on the figure the dots represent the start-up times if the blowout sequence had been carried on for the same amount of time rather than being pressure dependent. Thus, the effect of the drum size is rather small, and the figures indicate that doubling the drum size would increase start-up time by 10 - 15%.

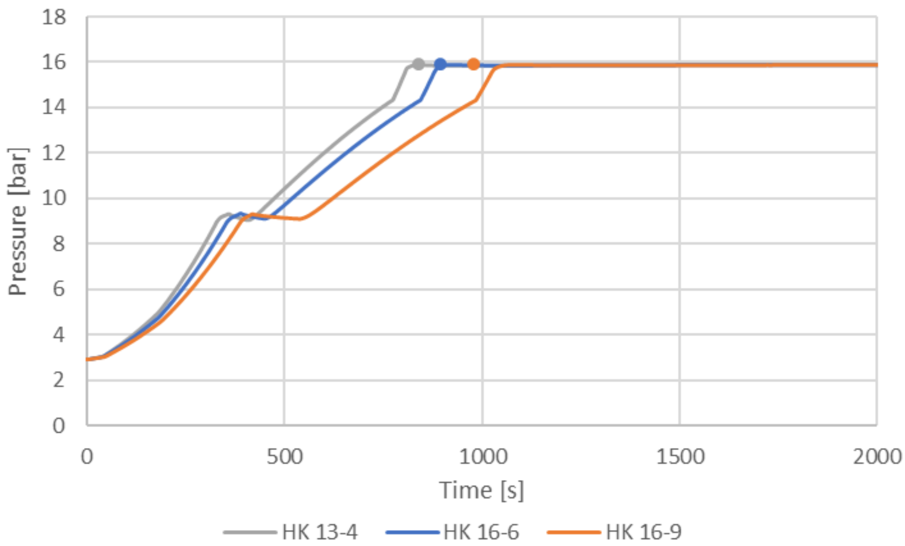

Figure 18. Start-up curves for different drum sizes.

In the figure a lot of the start-up time difference actually comes from the drum level set-point switch. As larger drums require more feed water to change the drum level the sequence carries on longer preventing the pressure gain. However, the higher the drum level, the more thermal inertia is in the boiler. On the other hand, larger drum size could allow for faster opening of the exhaust gas damper or ramping of the primary cycle as there is more space for swelling. Also, during shut down a boiler with a larger drum hosts more thermal energy to surface area than smaller drums and thus can preserve pressure longer. Thus, paying the price for higher thermal inertia could actually be beneficial regarding start-up time if the boiler frequently undergoes hot starts.

For a two-pressure level boiler the start-up is a bit more complicated as while the HP cycle is still gaining pressure it is extracting more heat from the exhaust gases than at running pressure. Thus, the low-pressure cycle lacks a sufficient supply of heat to gain pressure. Figure 19 shows the behavior of a two-pressure level WHR boiler during start-up. As can be seen the LP drum gains pressure really slowly initially. Only when the HP circuit starts approaching the blow out sequence the LP circuit starts to see proper pressure gain. However, as the steam generation is divided between two cycles so is the thermal inertia. Hence, the HP circuit can be ramped up in roughly similar time frame as a one-pressure level boiler despite the greater pressure level. The LP circuit requires some more time prolonging the total start-up time. For a combined cycle plant, if the LP steam is used as a reheating steam this might cause an issue.

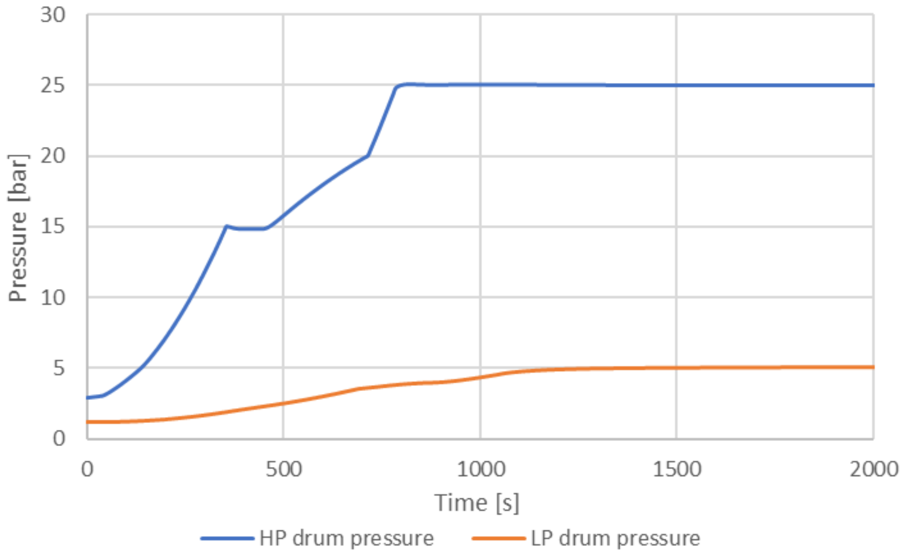

Figure 19. Start-up curve for a two-pressure level boiler.

However, increased drum pressure with an added pressure level brings another drawback. As higher pressures require thicker materials which are more prone to thermal stresses it could introduce new interlocks to the start-up sequence to reduce the temperature gain rate. If it is not the case and the swelling can be negotiated, the higher pressure allows for smaller steam drum as the required steam space is dependent on surface velocity and thus volumetric steam flow, reducing the thermal inertia.

5.2 Sequence solutions

Besides the physical structure of the boiler the sequence offers many parameters, which affect the start-up time. The most important being the blow out sequence as it is practically wasting energy and time to heat up the steam piping. A sufficient blow out sequence doesn’t run unnecessary long but makes sure that the piping is hot and free of condensate. At some cases if the exhaust gases are really hot a small blow out is required to protect the superheater from too high temperatures throughout the start-up sequence. However, usually with ICEs that is not required.

Another sequence dependent factor is the drum level set-point switch on the start-up as the cold feed water tends to drop the pressure. In theory keeping the water level low for a longer time period reduces the thermal inertia, and as the feed water in the economizer is also warming it should account for less dramatic pressure decline. Introducing cold feed water to the drum should in theory cause shrinking effect as the bubbles in the steam drum collapse due to subcooled water and it is often timed to act in synergy with the blowout sequence, which in theory causes swelling. Yet, as the drum level seems to act rather stable during this effect it is possible that both those effects are rather neglectable and could be adjusted independently.

The third significant sequence related factor is the EG damper control or the primary cycle ramp-up. Typically, the WHRB heat input is restricted at the beginning to ensure stable water level in the steam drum. If the heat input is increased too rapidly before the circulation is properly developed steam bubbles form inside the evaporator. Figure 20 shows such event. In that case the drum was initially near atmospheric pressure but saturated and the damper position was restricted to 20% until a set pressure was reached. At the beginning the water level was at IWL and in the graph it ends to NWL after the blow out sequence. As clear even with partial load the drum level was very unstable but stabilized after the drum had gained some pressure and the circulation was flowing as designed.

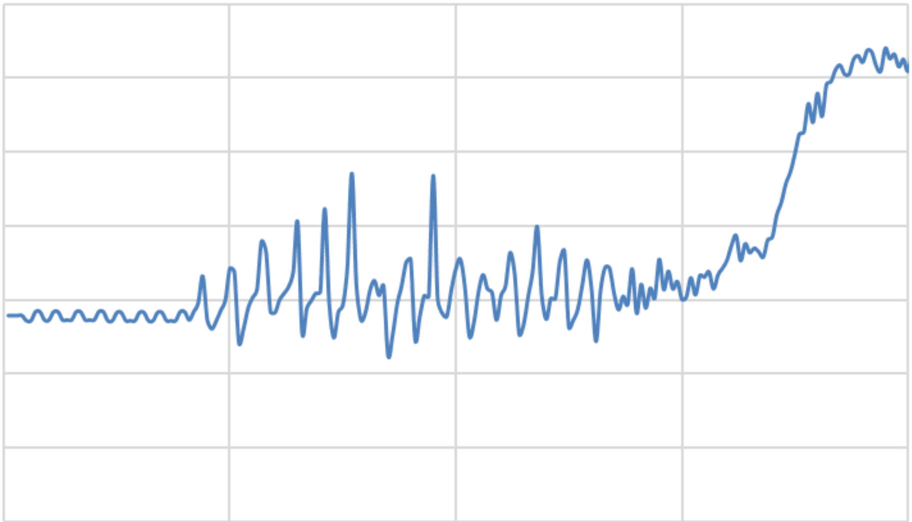

Figure 20. Drum level during initial start-up

Sequence wise there isn’t a lot to be done related to this event, but physically some things can be done. The most obvious solution would be to increase the drum size to allow more swelling, but that doesn’t fix the problem. Another thing that could be done is to tilt the horizontal evaporator tubes slightly upwards. This would ease the start of the circulation as steam bubbles wouldn't get stuck inside the tubes but would start rising upwards towards the drum and thus the drum level would be more stable.

5.2.1 Blow out sequence

As said the primary purpose of the blow out sequence is to heat the steam line. Yet, how to do it most efficiently regarding the start-up depends. Practically there are multiple ways to conduct the same sequence. The blow out could be conducted early and closed before connecting the boiler to the steam header or it could be started later and overlap the closing with the header valve opening. In theory this should not have a significant effect. Minor differences could come from differences in pressure over time. If the blowout is started earlier, the pressure would stay lower and thus more energy could be recovered from the EG. Another thing to consider with blow out timing is that the heating of the main steam line causes thermal stresses to the piping. Conducting the blowout at higher pressure could thus increase the mechanical stresses subjected to the piping as in addition to the thermal stress the pipe would also be under higher pressure stress.

Figure 21 shows start-up curves for late and early blow out sequences. The early blow out sequence begins after 7 bars and late after 12,5 bars respectively. For both cases the drum level set-point switches at 9 bars.

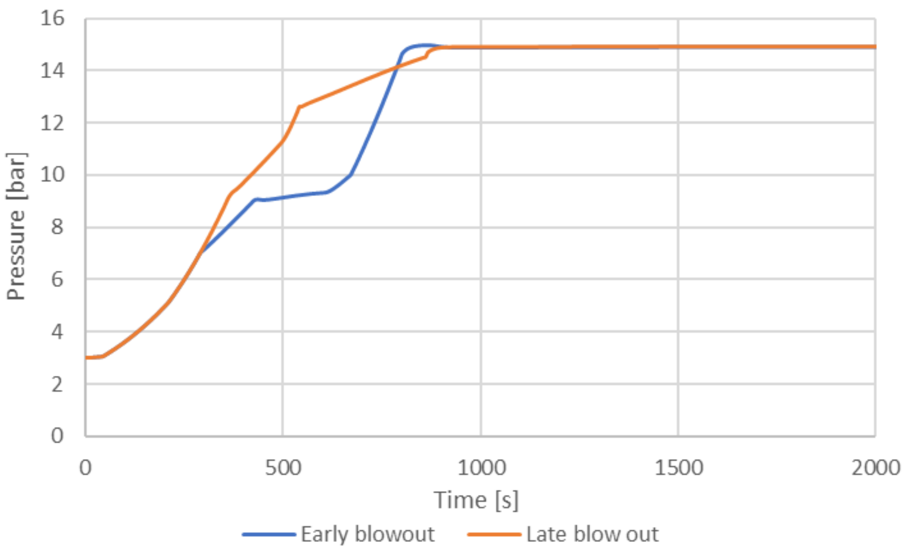

Figure 21. Blow out timing comparison.

Figure 22 shows the comparison between main steam line temperatures for the two blow out sequences. As the pressure is lower for the earlier blow out sequence the superheater inlet temperature is also lower which in turn reduces the steam temperature entering the main steam line. Also, as the blow out valve is choked, the volumetric flow rate is rather constant meaning the mass flow rate depends on the steam density i.e. pressure. Thus, with earlier blowout the steam line is heated much more gently. However, due to these reasons the blow out sequence carries on longer.

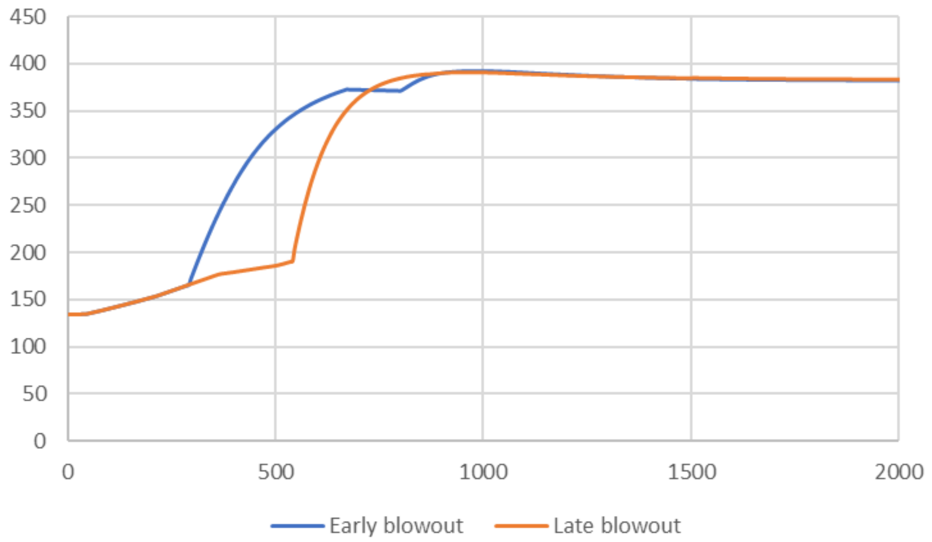

Figure 22. Main steam line temperature curves for different blow out timing.

Despite the longer blow out sequence the earlier blow out corresponds to greater HRSG efficiency during start-up thanks to lower pressure and thus results to faster start-up time. Hence starting the blow out early and closing it again after the line has been heated should be preferred sequence compared to overlapping the blow out and main steam valves. It results to faster start-up and reduced stress on equipment. The only concern is that if the gap between blow out closing and header opening is too big the superheater could experience too high temperatures if the exhaust gases are hot or at least cause sudden temperature changes.

Another factor is whether to do a small and long or big and short blow out. In theory increasing the flow increases the heat transfer between pipe and steam through increased Reynolds Number. However, with faster blowout also the thermal stresses in the piping are greater and this could have lifetime reducing effects, meaning that if the benefits are ineffable the smaller and longer blow out should be preferred.

Figure 23 shows the comparison between big and small blow out rates. The comparison has been done by stopping the blow out sequence after a set steam line temperature had been reached. As shown the smaller blow out corresponds to faster start-up as less steam is wasted thanks to smaller flow rate even though the blow out time has been increased significantly. This together with factors discussed earlier regarding thermal stresses suggest that the blow out is preferred to be conducted early with a flow that is just enough to heat the steam line before connecting boiler online. One think which could potentially cause problems with starting the blow out too soon is swelling as if the circulation flowing properly opening the blow out could cause problems with drum level control.

Accounting the heating rate restrictions given by the SFS-EN 12952-3 the simulation shows that generally the steam header is well below the set limits and thus regulatory wise there is no reason to avoid rapid heating. Yet, if there isn't a proper advantage of exposing the boiler components to abrupt heat shocks, they should be avoided to minimize wear.

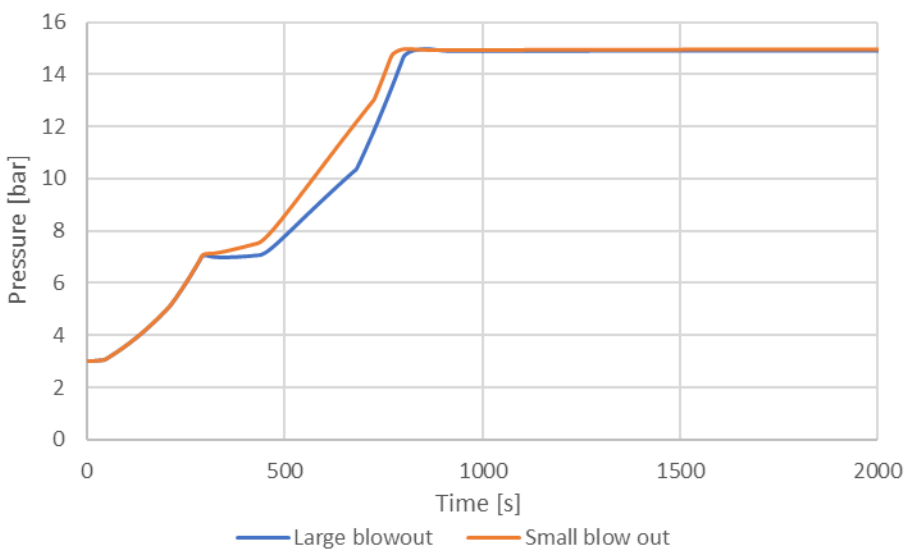

Figure 23. Start-up comparison for different blow out rates.

Regarding dimensioning of the blow out valve, if the valve is operated on-off, the valve size should be selected such that the flow through isn't wasting too much steam, but is must also be ensured that it can handle large enough steam flow to heat the steam line. For example, if the drum frequently undergoes hot starts, the main steam line could be cold even though drum still has significant pressure. Thus, to avoid a bottleneck the blow out valve should pass enough steam to heat the steam line in time. A controllable blow out valve would ensure optimum blow out rate for each start-up scenario.

5.2.2 Drum level control

As said, upon start up the drum level is kept lower than during operation and the set-point will switch after gaining enough pressure. Thus, it is worth investigating the differences of adjusting the timing. By switching the set-point early would mean that the pressure reducing effect would take effect earlier and thus boosting the evaporator performance, but it could cause an issue as there is less space for swelling if that were to take place during blowout. It would also mean that the boiler thermal inertia would be increased earlier. Another think to consider is that as the economizer is right after the evaporator and with no flow it could experience some level of boiling.

If the exhaust gases flow downwards, as typical with ICEs, the hydrostatic pressure would prevent the evaporation to some degree and as the economizer flow direction is upwards, the bubbles would flow to the drum without causing an issue. However, with GTs the EG flow direction is often upwards and as a result the pressure is lower than in the evaporator and the economizer flows downwards meaning that the bubbles would flow backwards in the economizer causing water hammering as they come to contact with colder water. Besides earlier set-point switch the water could be recirculated back to the feed water tank to ensure constant flow through the economizer and prevent boiling from taking place.

The benefit of the later set-point switch would be that, for example, during hot-start the drum is hot, the water inside the economizer could be cold. Thus, if the set-point switches too early really cold feed water would enter the drum and cause unnecessary pressure decrease. By switching the set-point later would ensure that the feed in water is properly up to temperature. Figure 24 shows the start-up curves for such case. As can be seen even in such situation where the drum is hot, but the economizer cold, retarding the set-point switch doesn’t provide any major benefit. Hence, matching it with the blow out sequence should be preferred to mitigate any swelling of shrinking effects.

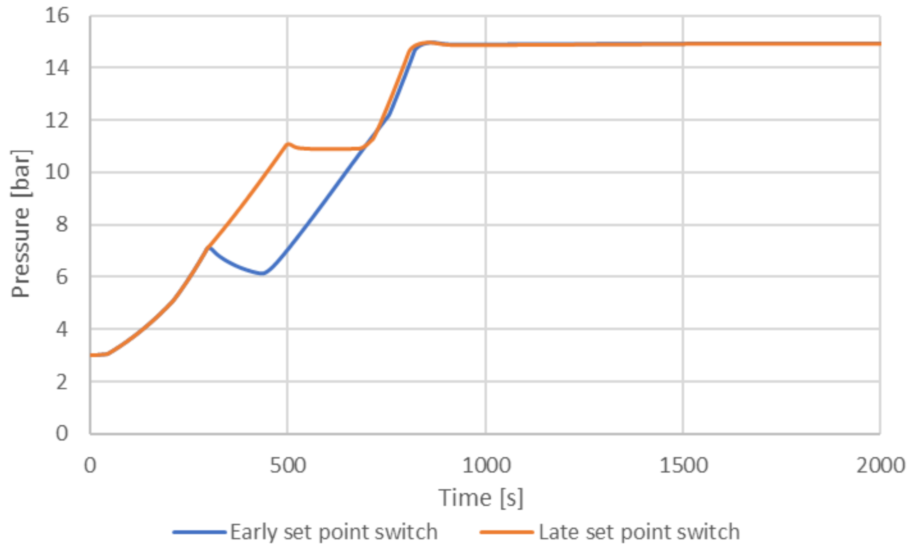

Figure 24. The start-up curves for drum level set-point switch alternatives.

Other think to consider besides timing is the feed water flow rate for elevating the drum level. Quick and short burst of feed water causes more abrupt disturbance for the boiler heating process, but it obviously takes less time. Figure 25 shows the results. As notable there doesn't seem to be any major effect regarding start-up time. However, with lower feed water flow, the thermal shocks for the feed water system are less severe and most probably the boiler control is also more stable when inducing smoother changes. Also, when restricting the feed water control valve opening there is lower valve travel time meaning the control system can react faster and provide greater stability.

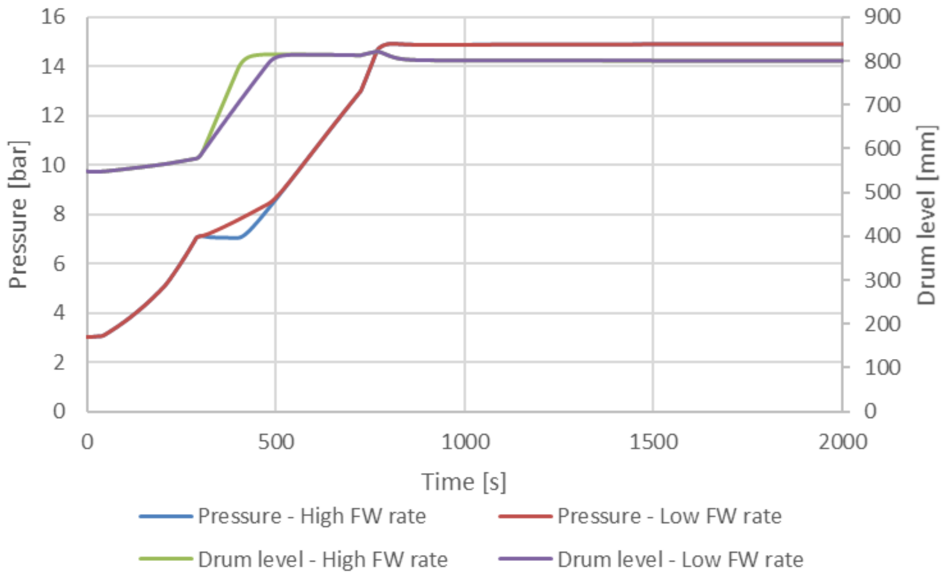

Figure 25. Feed water flow rate comparison.

Inconclusion, it seems that to ensure a fast and trouble-free start-up the blow out sequence should be started relatively early after gaining some pressure and the blow out flow should be kept relatively small in order to not waste precious energy. Preferably the blow out would continue until the boiler is ready to connect. The drum level set point should also switch simultaneously with the blow out to allow for swelling with the lower initial water level and to combat it by inducing shrinking phenomenon. To control the blow out rate a continuous adjustment of the blow out valve should be preferred as well as limiting the feed water valve opening to prevent excessive feed water flow rates.

6 EFFECTS OF TURBINE SOLUTIONS

The start-up and response properties of a combined cycle power plant aren’t just dependent on the boiler as with a condensing power plant the steam is only an energy carrier which must be conversed to other types of energy to provide revenue. This conversion is done first with the steam turbine to mechanical energy and thereafter to electrical energy by the generator. In addition to these the steam must travel from the boiler to the turbine through piping. As mentioned earlier this piping also takes time to heat up is therefore a potential bottleneck for the whole system. Hence it is preferable to study the effects it has.

6.1 Piping

A combined cycle power plant, especially with ICEs, often has a modular composition. This means that multiple primary cycles are connected to one secondary cycle. Two established piping solutions are to connect multiple boilers to a single steam header which is connected to the main steam line. Other alternative is to connect the boilers directly to the main steam line. These alternatives will be referred as the steam header layout and the collector line layout respectively and they are illustrated in Figure 26.

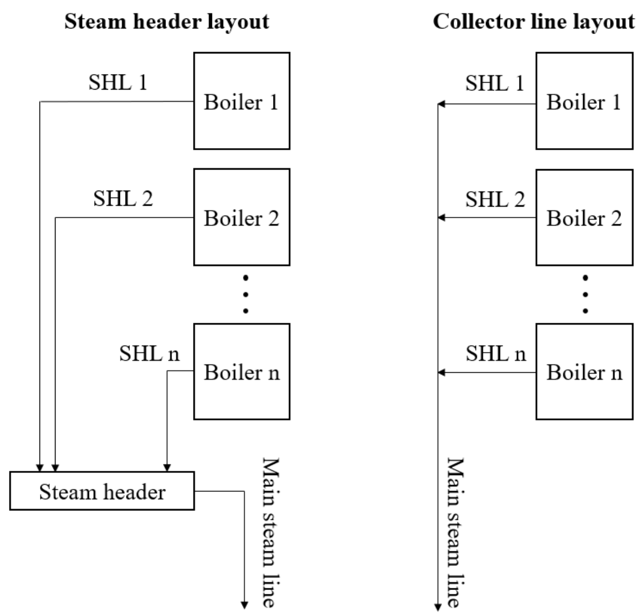

Figure 26. Main steam piping alternatives.

The benefit of the steam header layout is that it combines the smaller superheater lines (SHL) from the individual boilers closer to the steam turbine and all the valves are located in the same area allowing to use some shared components as well as providing a simple, pre-manufactured option for collecting the boilers. Hence the collector line layout option is potentially a bit more expensive, although it requires less piping.

Operatory wise these layouts have some differences related to the start-up. With the steam header layout, the superheater line can end up being very long. This excess length takes time to heat up and hence when connecting a boiler to the header the blow out sequence is longer. Yet, when starting up the turbine there is less piping to heat up and smaller steam cavity because the SHL is dimensioned only for the steam flow of one boiler while the collector line must pass through the steam generation of multiple boilers. Figure 27 shows the effect of the SHL length to the start-up sequence.

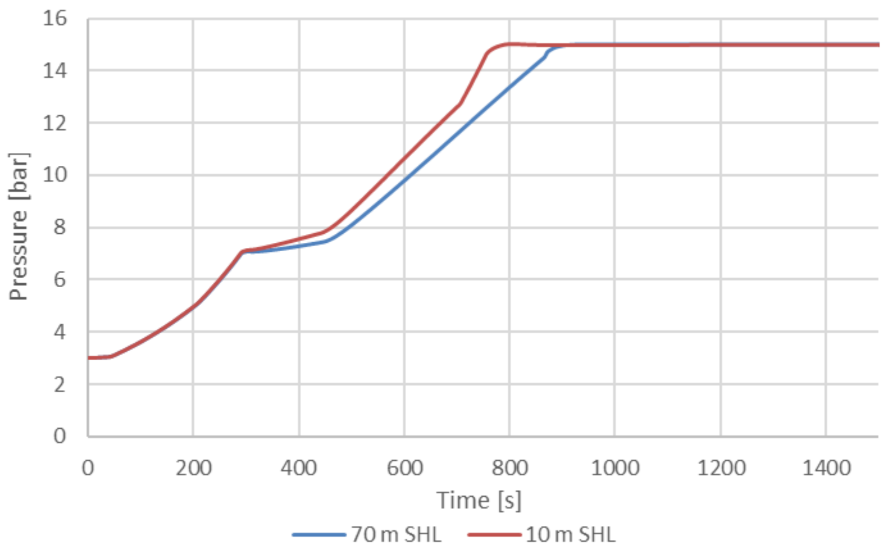

Figure 27. The effect of superheater line length to the start-up time.

In the figure the blow out times are adjusted to gain the same temperature to the end of the SHL. This together with the fact that colder and therefore denser steam through the blow out valve accounts for greater steam flow causes the start-up time to increase significantly with over 1,5 minutes meaning an increase of over 10 %. Differences in length like this are not farfetched as it corresponds to a layout where 7 boilers are located to a row with 10 m intervals.

Another effect of the SHL length is that it increases the heat losses. This effect is most notable during shut down as the boiler is insulated from the end of the superheater line. The added SHL outer surface area releases heat to the environment and this can be seen as an increased rate the pressure decline. Figure 28 shows how the measured pressure decline rates differ for four boilers with different SHL lengths. The difference of rate seems to roughly correspond to the added heat loss area.

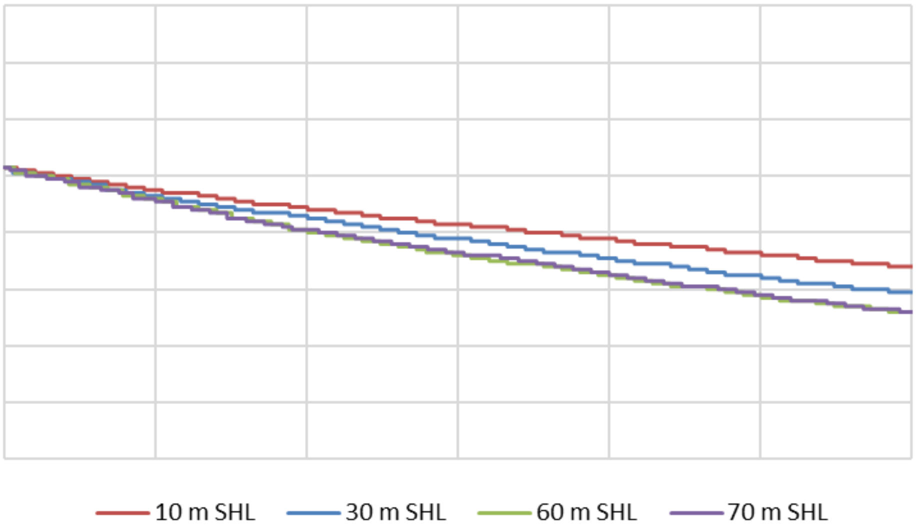

Figure 28. Pressure decline rates of four boilers.

When starting the whole plant, the steam header layout is really simple. The plants can start at any order with the only difference being the SHL length. Usually, the auxiliary steam outtake is located such that the whole header is heated no matter which boiler(s) is/are generating steam and as the header is only a few meters in length it happens quite quickly. On the other hand, the collector line has more things to considering upon start-up. In Figure 27 if the boiler 1 is started first it must heat the whole collector line, which could take a massive time. However, if the boiler n is started first, it only needs to heat up the piping from its joining to the turbine. The line from boiler 1 to boiler n will remain at saturated temperature without any real steam flow and due to heat losses, some condensation is bound to happen.

The problem with this approach is that when connecting more boilers, there will be some condensate in the pipeline and this water will mix with the steam flow. If designed properly this would only cause a bearable level of desuperheating, but there is also a danger of droplet carryover, which in the worst-case scenario could end-up in the turbine causing severe damage.

To prevent the risk of droplet carry-over the collector line must be designed appropriately. The condensate in the steam line can be thought as a desuperheating station although there are some differences. Mainly, the desuperheating nozzle, if acting correctly, spray the water into the steam flow speeding the mixing. In the line the condensation is collected to the bottom and the steam pushes it along. To ensure mixing the steam must have high enough velocity as it corresponds to more turbulent flow, which boosts mixing. Yet, with higher flow velocity the droplets may cause erosion in the steam line.

With a desuperheater unit the pipe must have a minimum straight line length downstream to enable mixing since the water droplets would hit the pipe walls in case of turns or change of diameter. The required straight line length is dependent of the degree of superheating, amount of condensate and pipe diameter, but is at minimum 3 m. The steam velocity in general should be above 10 m/s and below 76 m/s. (Flowserve Corporation) This could be a problem as the collector line should be in any situation be dimensioned for the steam flow of all the boilers. Hence, if there were in total seven boilers and the boilers were started-up in the described order, the steam flow of two boilers should account for the minimum of 10 m/s, which would result in to 35 m/s for the steam from all boilers. This could increase the main steam pressure drop significantly.

The required straight line length could cause issues with the piping design with aspects like placing the thermal expansion loops or if the site layout requires a turn. A steam trap should also be placed 6 m downstream of the point of mixation (Flowserve Corporation). One alternative for mitigating the risk of droplet carryover while adding boilers online is to connect the starting boiler to the collector line while the drum pressure is still lower than the line pressure and the blow out valve is open. Then the resulting flow would be from the downstream boiler towards the upstream boilers blow out valve carrying most of the condensate out. It is also worth noting that if the operating and connecting boilers are joined to the collector line not too far away from each other the heat of the hot steam would conduct and diffuse also towards the upstream boiler and thus reducing the expected amount of condensate in the line or even removing it altogether.

Upon the turbine start-up if the whole system is in hot state meaning the drum is under pressure and the steam lines are in saturated temperature there isn’t any significant differences with the heating rate if the starting boilers are located nearest to the steam turbine. Figure 29 shows how the heating process of the two layouts are really similar when starting from the boiler nearest to the turbine. Only notable difference comes from the fact that as the steam header has greater material strength it takes longer to heat up and that shows as colder steam to the downstream. It can also be noted that as the heated pipe length is rather short the main steam line temperature rises above the saturation line almost immediately meaning there is no risk for condensation.

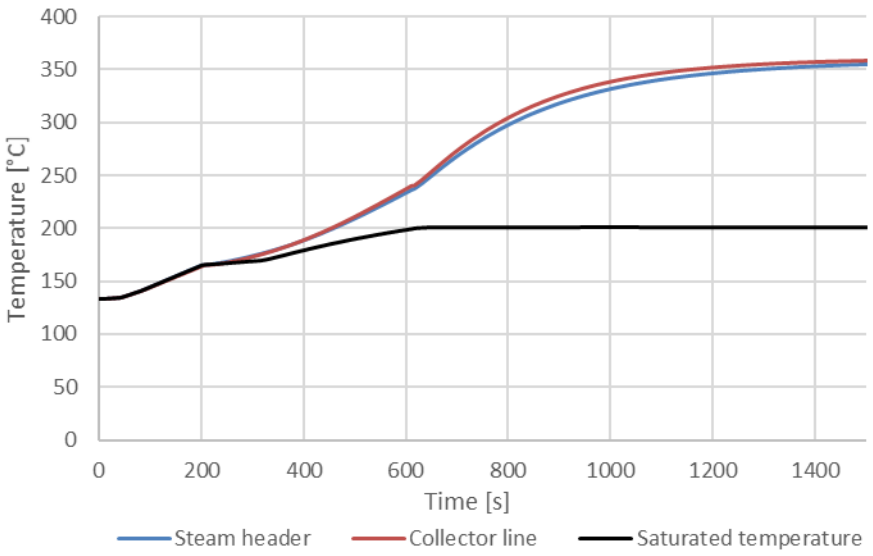

Figure 29. The temperature profile at the end of the main steam line for two layout alternatives.

Yet, if the start-up is conducted from the other end of the line there are more notable differences visible. Because the collector line is much bigger than the superheater line of the individual boiler it takes time to heat up as shown in Figure 30.

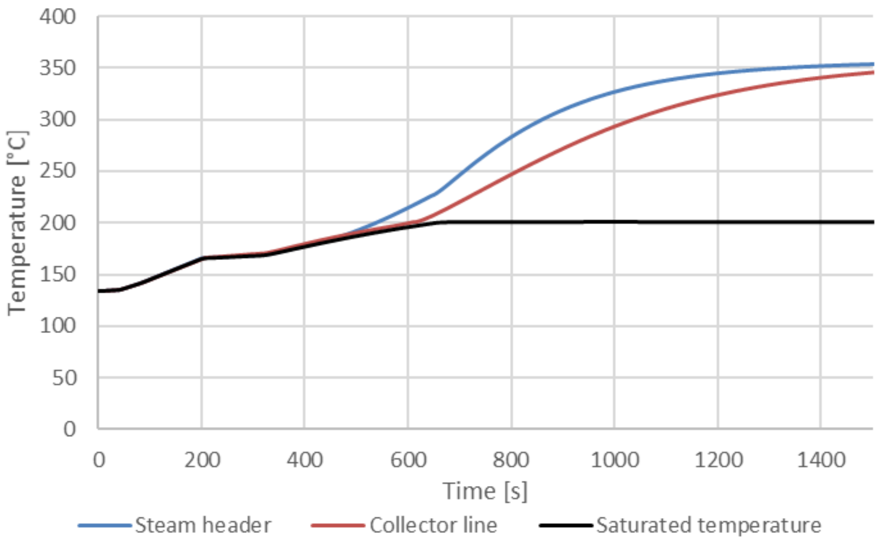

Figure 30. The temperature profile at the end of the main steam line for two layout alternatives.

In the figure we see that the temperature at the end of the line exceed the saturated temperature almost two minutes later for the collector line layout than it does for system with a steam header meaning there is a likelihood of water droplets which could potentially prevent the turbine sequence from proceeding. Another point is that the collector line has vastly greater steam cavity which has to be filled as the specific volume of steam increases with pressure, but it seems to have a neglectable effect to the start-up time. When operating with partial number of total boilers the steam cavity includes greater pipe surface area which accounts for greater thermal losses although the opposite is true when operating all boilers.

As conclusion for the piping solution, it seems that the collector line layout offers marginally better start-up properties as the superheater line length is rather constant for all boilers. Although, in this chapter the evaluation was done with a long blowout sequence which heated the line to almost operation levels even though it isn't necessary. Thus, the effect of SHL length is exaggerated. The steam header layout also gives more freedom to the running order of the boilers, but mainly suffers from greater thermal losses especially during standstill.

6.2 Turbine auxiliary systems

The steam turbine has numerous auxiliary systems ranging from lubrication to water drainage. However, regarding to the start-up sequence we will consider the sealing steam and vacuum systems. As described earlier a condensing steam turbine requires a condenser vacuum before it can speed up. Before the vacuum can be reached the steam turbine must be sealed to prevent air flow to the condenser through the rotor clearances. During operation, the high-pressure end of the turbine is above ambient pressure meaning it doesn't need external sealing steam, but some gland steam leaks from the labyrinth seal. This leaking steam can also be used as sealing steam for the low-pressure end. However, during start-up there is no high-pressure steam in the turbine to prevent air leakages an external source of sealing steam is required.

Figure 31 shows an exemplary sealing and gland steam system for a large steam turbine. For a smaller turbine the picture would be slightly different as there wouldn't be separate HP, IP and LP sections and the turbine wouldn't have double-axial-flow. Yet, we can see that during operation the HP and IP turbines don't utilize any external steam. The outermost seal is connected to gland steam ejector and the gland steam condenser where the air-steam solution is cooled and the condensate separated and returned to the cycle while the second outermost seal is connected to the same header that also receives external sealing steam at point 7. This sealing steam is then fed to the inner LP seal while the outer is connected to the same gland steam exhaust as the other turbines. During start-up the header would also feed to the HP and IP turbines meaning the required external steam flow is greater. Yet, the typical sealing steam flow rates are rather neglectable and are itself unsuitable for steam line heating.

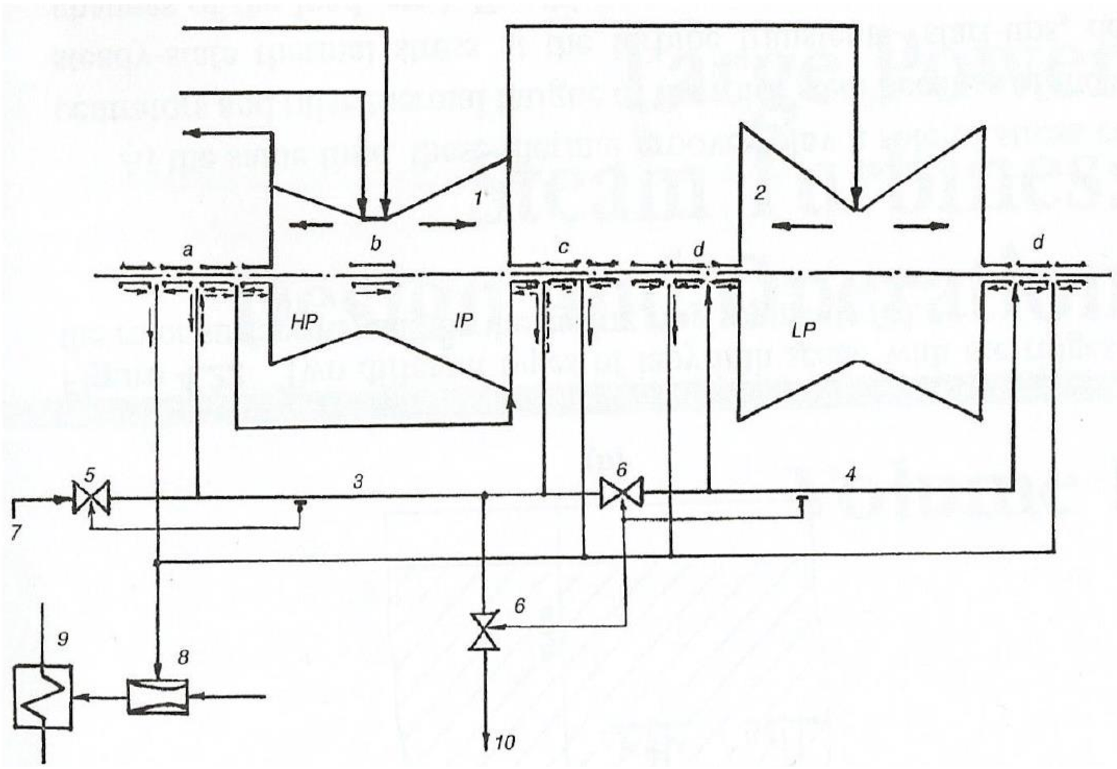

Figure 31. The sealing and gland steam system for a large steam turbine. (Leyzerovich 1997)

The condenser vacuum system is the other main auxiliary equipment that is in particular interest regarding the start-up procedure. The vacuum system can be described to have two operational profiles hogging and holding. Hogging means the process during start-up when the condenser is full of air, and it is removed while holding describes the online operation when the leakage air and other non-condensable gases are removed from the condenser as they can have a severe effect on performance. A quick raise of vacuum during a start-up is vital as steam isn't allowed to the condenser before some moderate level of vacuum is established. This means that in excess of forming a bottleneck for the turbine start it also restricts the start of the PRDS. If steam can't be admitted through the PRDS the steam line heating steam must be blown out wasting water or the heating must be retarded.

The vacuum system is mainly implemented in two ways, with a steam ejector or a mechanical pumping system. Steam ejector has multiple benefit as it has high efficiency and it doesn't have any moving parts requiring regular maintenance or even risking failures. Figure

32 shows the layout of a typical steam ejector. The steam ejector utilizes motive steam which goes through a laval-nozzle and according to Bernoulli's law the pressure decreases as the velocity increases. This low pressure forms a suction for the air-steam vapor in the condenser and the mixture then flows through a supersonic-subsonic diffuser. The steam is then condensed in a condenser and returned to the cycle while the non-condensable gases are usually released to atmosphere.

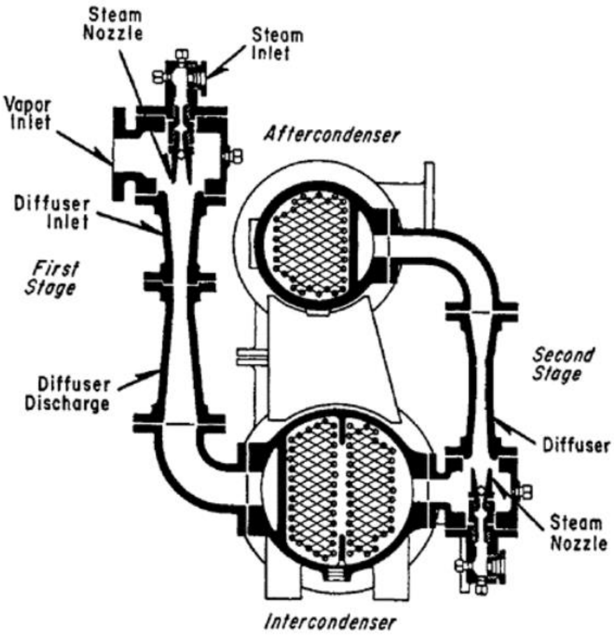

Figure 32. Description of a two-stage steam ejector system with surface condensers. (Coker & Ludwig 2007)

If discharging to ambient pressure the ejector performance i.e. capacity is mainly a question of motive steam properties and the suction pressure. The ejector design steam pressure is often selected based on the minimum expected available steam pressure, but usually if discharging to atmosphere the motive steam pressure should be above 3 bar(a). With motive steam pressures below the design pressure the ejector will not guarantee stable operation. However, pressures above the design pressure don't remarkably increase the suction capacity as the increase in pressure is linear with motive steam consumption. This added consumption increases flow rate through the diffusor and thus increases the steam consumption without increasing capacity significantly. (Coker & Ludwig 2007) In other words this means that the motive steam pressure only has a minor effect on the ejector capacity, but it is dependent on the design pressure.

The motive steam temperature also affects the ejector performance. If the motive steam is wet the droplets cause erosion to the ejector nozzle and they can also clog the nozzle causing instability (Coker & Ludwig 2007) With a CCP the wet steam is mainly an issue during the very initial start-up. It is unlikely that such short time frames could cause serious erosion and because the ejector is undergoing hogging the instability shouldn't be an issue. Yet, permitting wet steam operation is dependent on the recommendations of the manufacturer.

Usually, a few degrees of superheating are recommended for the motive steam, but highly superheated steam can be problematic due to its lower density which must be considered in the design (Coker & Ludwig 2007). The level of superheating also effects the condenser design as more heat must be erased from the vapor and as the heat transfer coefficient for superheated steam will significantly lower than for condensing flow it might be preferable to favor direct water injection instead of surface condenser.

During holding phase the required ejector capacity is dependent on the air leakage rate to the condenser. The condenser is never completely airtight and estimating the amount of the leakage air is difficult. Generally, the amount can be estimated by the condenser volume and whether the connections to the parts under vacuum are done with flanges or welds. A ballpark figure for a typical 10 MW_e turbine condenser could be around 4 kg/h of air. (GEA Wiegand) In addition to the air the ejector also sucks in some steam from the condenser. The ratio of steam and air is dependent on the location of the air extraction inlets in the condenser. Coker & Ludwig use on exemplary figure of 70 % steam (2007) which would in turn mean the capacity requirement of 13,3 kg/h of vapor for the aforementioned condenser. However, for a multistage ejector the condensable steam is mostly removed by the intercondenser and the exiting flow would ideally compound of air with 100 % humidity.

The effect of suction pressure to the capacity is highly dependent on the type of the vacuum system. Figure 33 shows typical capacity curves for different types. From the figure it can be seen that multistage ejectors have a rather narrow optimal operation window and the single stage ejector performance tends to collapse as the suction pressure decreases. In fact, it is recommended not to use single stage ejectors for suction pressures below 75 mbar (Coker & Ludwig 2007). Due to these reasons plants are often equipped with both running and start-up vacuum systems as single-stage ejector can't operate in the required vacuum and multi-stage ejector lacks capacity during hogging.

Mechanical vacuum pumps have quite stable performance rate over the suction range and are thus a great option for the start-up. Their downside is that they require more maintenance, increase the own consumption of electricity and might increase the capital costs. Yet, if they help to overcome a bottleneck for the steam turbine start, they might be preferable. Another alternative could be to use a single-stage ejector for the hogging so it could be optimally designed for the hogging process.

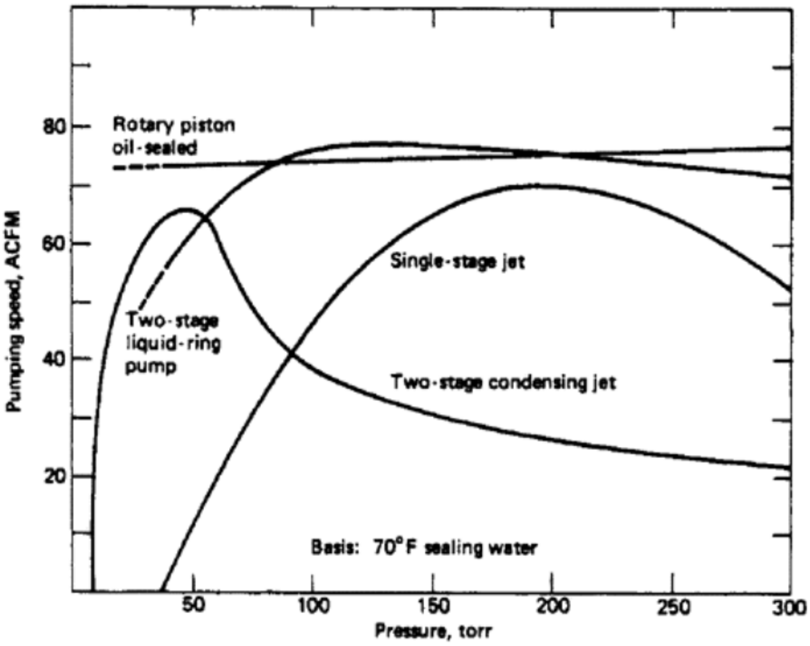

Figure 33. Vacuum system capacity by suction pressure. 1 bar = 750 torr and 1 m^3n/h = 0,589 ACFM. (Coker & Ludwig 2007)

Yet, the sealing steam and motive steam flow requirements are almost neglectable regarding the boiler start-up. The required pressure and degree of superheating are dependent on the OEM, but they should come available as soon as possible. The biggest concern is if the OEM requires a degree of superheating for the sealing steam, as the main steam line may take a while to heat. One solution to overcome this could be an additional start-up steam line from the boiler closest to the turbine. This line could be kept in the boiler cavity during shutdown and thus remain in saturated state without excessive heat losses. This line could be dimensioned for low steam flow rates and thus heat up quickly and start supplying the labyrinth seals. Ideally the control valve would also be placed to the upstream so the pressure and hence the saturated temperature in the pipe would remain low while the boiler gains pressure. If the hogging is also done with a steam ejector the same line could also be used for the motive steam and thus overcome the bottleneck of initiating the vacuum generation.

6.3 Two-pressure levels

If the secondary cycle utilizes multiple pressure levels it provides added challenges regarding the start-up process, because basically the added pressure must have its own steam system. In excess of only generating the pressure build up in the added pressure level, also the steam piping must be heated up. Often in combined cycle power plants, the low-pressure steam is used for also reheating the steam mixture before the LP turbine section instead of just adding the steam flow rate. This proposes further challenges for the turbine start-up and ramping.

Figure 19 shows the start-up curve for a two-pressure level boiler. As clear from the graph the LP section takes a lot longer time to reach operational pressure levels. Whether or not this forms a bottleneck for the system depends on the turbine. The turbine-end steam quality is a limiting factor on how much energy can be recovered from the steam and the exhaust steam quality is dependent on the inlet steam parameters and the condenser pressure. With two-pressure levels the superheated LP injection steam increases the exhaust steam quality, but if it is unavailable it might prevent the turbine from ramping, mainly because the inlet steam pressure must be regulated to prevent excess wetness.

Figure 34 shows the exhaust steam quality without LP steam injection for a steam turbine designed for 90% exhaust steam quality at full load with the injection. With regulating the steam flow either through throttling or sliding pressure the turbine load decreases and so does the exhaust wetness with the assumption of constant isentropic efficiency. From the figure we can observe that without the injection the steam quality would fall below the 90% value when the turbine load surpasses 60%.

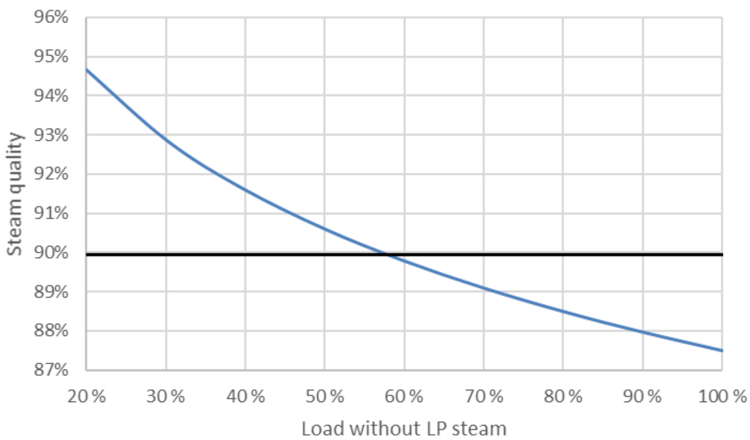

Figure 34. A two-pressure steam turbine exhaust wetness without LP injection.

In other words, the turbine load can’t exceed 60% without injecting the superheated LP steam. This doesn’t necessary mean that it is bottleneck as the LP steam is required only at a very late stage in sequence. If the steam turbine is connected to a block of boilers and the boilers are started sequentially the first starting boilers are already producing LP steam when the 60% load would be exceeded. However, if the boilers were started simultaneously or the steam turbine would only be fed with one bigger boiler, there would still be many parts in the sequence before the 60% load would be achieved.

Firstly, the turbine must be sealed and the condensator evacuated from air, steam lines must be heated, the turbine must be sped up and synchronized and the load increased. If we only consider from the Figure 19 the HP boiler would start to produce sealing steam at around 100 s and the LP steam is ready at around 1100 s it would mean that if the turbine were to exceed 60% of its injectionless load within the next 17 minutes the LP circuit would be a bottleneck. If we consider that evacuating the condensator takes 5 minutes, the speeding up and synchronization another 5 minutes and from the minimum load of 20% to the 60% with maximum 5 %/minute ramp rate this would take 18 minutes alone meaning that the LP circuit would be ready for the load increase. This is also a very quick turbine start and considering the 2 minutes prior to submitting sealing steam and the 8 minutes to increase the turbine load further to 100% the total start-time would only be 28 minutes. Considering that for combined cycle power plant post flexibilization start-up times range between 30 minutes to 3 hours (IRENA 2019) it seems that adding the second pressure level doesn’t remarkably hinder the power generation flexibility from the thermodynamic perspective.

However, as discussed before adding a second pressure level also increases the pressure in the HP circuit. This increase in pressure corresponds to thicker material strengths which are more prone to fatigue caused by thermal stresses. Even regulatory restrictions could come to play. Also, if using sliding pressure control it could prevent proper steam generation from the LP cycle during part-load operation, which could potentially increase the LP start-up time to cause issues during start-up. Lastly, it was considered that the auxiliary steam was provided from the HP circuit. With multiple pressure levels it is more common to use the LP steam for auxiliary needs as it is more energy efficient. In such cases the start-up times would increase remarkably unless the HP circuit is also connected to the auxiliary steam system. If needed the HP steam could also be used for heating purposes for lower pressure levels if that would prove beneficial.

6.4 Turbine control system

As discussed before, the turbine control system can be arranged generally in four ways, throttle control, partial admission control, pure sliding pressure and modified sliding pressure. In this chapter the throttle control, partial admission control and modified sliding pressure control will be discussed more in relation to start-up and load response. Pure sliding pressure is not considered as using modified scheme with a control valve or stage gives more control for the plant without drastic effects on efficiency.

As mentioned in chapter 3.4 the partial admission with nozzle group control has benefits compared to throttle control with efficiency, but due to partial arc admission in the control stage it results into uneven heating of the stator and causes thermal stresses and bending. Whether this is a problem depends. If the steam turbine regularly undergoes hot starts the thermal stresses opposed to it are only minor. Due to steam piping cooling down faster than the turbine it is even more likely to have negative temperature gradients meaning the steam is colder than the metal. This, however, can be mitigated by more aggressive heating sequence. The uneven heating can also be mitigated by constantly varying the open valves and thus sequentially heating the turbine casing.

Throttle control, on the other hand, doesn't suffer from uneven heating, but lacks in efficiency due to throttling losses. Modified sliding pressure, on the other hand, only has to do minimal throttling as the main steam pressure is adjusted in the boilers. Thus, depending on the operational profile, a constant pressure control system should be equipped with either throttle or admission control, while with sliding pressure the throttle control would enable easier and potentially faster start-ups without sacrificing efficiency. With the operational profile it is meant whether the turbine would operate excessively with partial load and how important is the start-up time. At full load the control system doesn't affect the efficiency. However, the choice for the individual projects should be discussed with the turbine supplier to understand their requirements and constrains and their suitability.

Regarding the load response parameters, it would seem that the constant pressure control has the advantage, but that isn't necessary the case. In case of modular combined cycle power plant, the load changes usually commission through starting or shutting down additional boilers. Thus, in case of constant pressure control all the operating boilers are up to pressure and the starting boiler must also gain that pressure before it can be connected to the turbine.

After that the load increase happens rapidly with the opening of the throttle valve with the rate permitted by the turbine supplier.

With sliding pressure, however, the steam pressure is lower so the connection can be done faster, but after that all the operating boilers must increase their drum pressure. Again, the pressure increase can’t be done too swiftly in order to prevent too rapid load changes for the steam turbine. In these cases, with all control systems the PRDS must control the header pressure. With constant pressure it is quite simple as the turbine control system governs independently the mass flow rate through the turbine and the PRDS only by-passes the steam to keep the header pressure constant. With sliding pressure, the turbine governor must account for the changing live steam pressure and the PRDS will account for the changing header pressure set point.

An exemplary study regarding the response times is done with an imaginary power plant with in total seven boilers. Three of which are operating and the fourth one is connected. With the sliding pressure scheme the drum pressure of the boilers will jump from 8,4 to 10,8 bar(a) when the fourth boiler is connected and for the constant pressure scheme the boilers will operate at 16 bar. With sliding pressure, the turbine load will jump from 31 % to 44 % and from 27 % to 41 % for the constant pressure scheme respectively. With the constant pressure the starting boiler must reach higher pressure, but after that the turbine can increase load with only restriction being the turbine, but the sliding pressure must gain higher pressure to increase power.

Figure 35 shows the results of this study. In both cases the fourth boilers are started simultaneously and carry on their respective sequences. After around 800 s the constant pressure boiler is ready to supply steam and the turbine can increase load. For the sliding pressure, it takes considerably longer, but 1200 s the turbine load should be roughly the maximum producible with four boilers. If we again consider a turbine ramp rate of 5 %/min this load increase would take a bit over 16 minutes for the constant pressure control and 20 minutes for the sliding pressure control. With the sliding pressure the maximum load ramping speed is roughly around 3 % per minute meaning the PRDS wouldn’t need to restrict the pressure gain rate at any moment. Thus, with modified sliding pressure it is possible to slightly improve the initial response rate, but in this situation increasing the turbine load also dampens pressure gain the total response time isn’t affected as much.

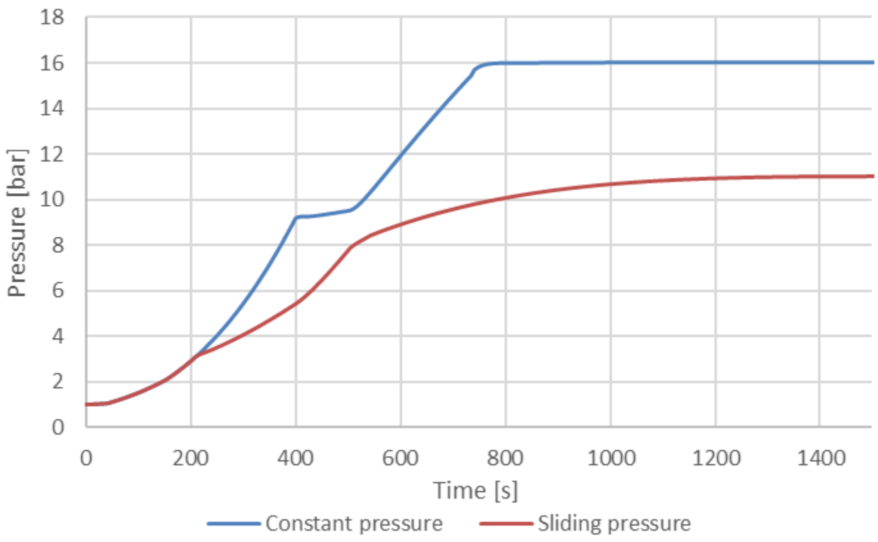

Figure 35. Response comparison between constant and sliding pressure turbines.

6.5 Turbine technical solutions

As we have learned the turbine start-up is much restricted on its heating process. Because the turbine operates with small tolerance gaps between the moving and stationary parts there isn't a lot that can be done to allow higher temperature gradients besides increasing these tolerances, which reduces the efficiency and thus total power. Another factor for the restricted heating is the low cycle fatigue, which develops over the turbine lifetime. This can be mitigated by the material properties. These, however, are the guidelines for the turbine OEM to set the start-up curves. Besides having higher temperature gradients, the turbine heating bottleneck can also be mitigated with technical solutions.

During standstill the turbine cools down and the colder it gets the longer it takes to get it up to temperature. These heat losses can be mitigated by using thicker turbine casing insulation. The turbine can also be kept hot with external energy during standstill. If the site has external superheated steam available, it could be used as sealing steam to heat the turbine rotor. In that case, the condenser could also be kept under vacuum during standstill and removing or shortening the hogging sequence during start-up. The profitability of this depends. In terms of capital investments, it would only require some piping and probably the hogging system could be downsized, but the value of consumed steam should be gained by more rapidly available steam turbine generation.

As the sealing steam mainly heats the turbine rotor it doesn't really aid the turbine start-up unless the casing is also heated. This could be done with electric heating blankets. Depending on the sealing steam temperature the optimal heating power of the blankets varies from zero to three kW/m². Regarding energy return on energy investment an exemplary 10 MW_e steam turbine with 10 m² of casing surface, 8 h of standstill and 1 kW/m² heating carpets would require only 30 seconds of full load running, but the profitability also depends on the electricity prices as during standstill the electricity would most probably be bought from the grid and the capital investment should also be covered. The heating blankets by themselves wouldn't practically provide any benefits unless hot sealing steam is available for the rotor, but the sealing steam would provide an improvement even without the blankets. (Spelling et al. 2012) If the site doesn't have a source of superheated steam and cheap electricity is available during standstill hours, also an electric boiler could be considered.

The blankets could also help with cold or warm start-ups by heating the casing instead of just maintaining temperature and as the steam heats the casing from the inside and the blankets do it from the outside, they diminish the temperature difference across the casing wall bringing additional benefit for the investment. Thus, higher ramping speeds could be allowed, but this again depends on the turbine supplier.

7 CONCLUSIONS AND DISCUSSION

The results obtained in chapters 5 and 6 indicate many possibilities for shortening the start-up and response times for the secondary cycle of a CCP. By combining these results and evaluating the requirements of the different operating profiles the most suitable solutions are drawn together with raising subjects needing more research.

7.1 Boiler solutions

For a plant that frequently undergoes hot starts, one important factor is how well the plant stores heat during standby. The start-up process can be accelerated significantly if the plant needs to do less work to gain operative state. From the boiler point-of-view, the main factors are the stored thermal energy and the rate of heat losses. The boiler can mainly store heat inside the steam drum and the bigger the drum the more energy can be stored. However, the larger the drum the longer it takes to heat during the start-up. The stand-by heat losses are related to the surface area and insulation. As was found, the effect of steam piping plays a big role in the rate the drum loses energy and pressure because it affects the heat loss surface area.

Another important factor for frequently hot starting boiler is the time it takes to reach operative state. During the start-up the main hindering interlocks are the damper position restriction and the blow out sequence. The damper position restrictions are in place for two reasons, to prevent excessive drum level fluctuations and to protect the boiler, mainly the superheater, from abrupt thermal shocks. During hot starts the drum level is often quite stable and the superheater is at least at the saturated temperature so damper can be opened wide quickly at least with ICE applications.

Thus, even though smaller drum would gain pressure faster, it doesn’t necessary improve operative qualities as the smaller drum loses pressure faster during standby and might require additional care in the very beginning of the start-up. With the absence of better knowledge, a bigger steam drum would ensure fast and trouble-free start-up with the minor penalty of increased thermal inertia.

Sequence-wise the biggest start-up affecting factor is the blow out sequence. The blow out should be conducted in a controlled manner to heat the piping and to remove condensate. A continuously adjustable blow out valve would ensure sufficient blow out for all situations to ensure safety, reliability, and performance as the mass flow rate, and thus the temperature gradients of the piping and the pressure gain rate of the drum, can be optimized. As the blow out sequence affects the total boiler start-up time significantly optimizing it provides major boiler start-up time reductions.

7.2 Balance of plant design

Regarding the balance of plant two steam piping alternatives were considered. The steam header layout has the benefit that the firing order of the boilers is free and the differences between boilers associate with the length of the pipe to be heated. This has the drawback that the boilers farthest from the header suffer greater heat losses during standby and slower start-up due to extended blow out period. With the collector line the individual boiler start-up times and heat losses are the same, but the firing order of the boilers significantly affect the steam turbine sequence as the main steam line takes extensive amount of time is heated by the boiler farthest away from the turbine. When starting up from the boiler nearest to the steam turbine, additional care should be taken when connecting additional boilers as it should be done sequentially to prevent carryover risk. Also, because the whole MSL is in the steam cavity piping heat losses are greater with part-load running, but the opposite is true with full load due to reduced piping surface area.

The balance of plant also includes its auxiliary system and with the start-up sealing steam system and vacuum systems are the most important. If saturated steam was allowed to be used as sealing steam, with proper steam traps it could be provided to the turbine instantaneously when steam with sufficient pressure is available and if the plant is equipped with hogging ejectors the same is true for them. But if the OEM requires superheated steam, a big bottleneck could be created if the steam piping required heating. This bottleneck could be overcome by a separate start-up steam line, which would heat up considerably faster. If saturated steam is allowed, it could also affect the preferred drum size as larger drums holds pressure longer meaning sufficient steam pressure could be available from the get-go or at least sooner.

The vacuum system also plays a vital part in the turbine start-up. Turbine back-pressure prevents it from speeding and even admitting steam to condenser is not allowed until a set vacuum is reached. The vacuum initiation takes a few minutes, but with a proper design there are no significant differences between steam ejector and mechanical pump solutions for hogging. Hence, the solution is mainly dependent on financial, and maintenance related reasons provided sufficient steam is available.

7.3 Turbine control scheme

The rate of the rotational speed increase of the turbine is dependent on the OEM and the boiler can only affect it by having sufficient live steam available when it starts. With a constant pressure control scheme, the OEM might require operating pressure steam for the running up, but technically speaking it is unnecessary as it is either way governed to lower pressure with the control system. Yet, if it is required the steam boiler could become the bottleneck, but due to aforementioned reasons it is not concerned.

The preferred turbine control system depends on the operational profile. Constant pressure with throttle control has poor part load efficiency due to throttling losses but has great load response parameters and a simple heating sequence. Nozzle group control gives significantly better part load efficiency, but the regulating stage is difficult and time consuming to heat uniformly. Modified sliding pressure control has the best efficiency and doesn't suffer from extensive throttling losses as less throttling is required due to lower main steam pressure but suffers from poor load response as it is dependent from main steam pressure and lower operating pressure could force larger steam drum sizes and piping increasing costs and possibly boiler start-up times slightly.

7.4 Solution fitting

For a plant that operates on-off, for example full load during day and standby during night, the best option would be a constant pressure turbine with throttle control equipped with the collector pipe layout as the plant doesn't require high part load efficiency but requires fast and simple hot starts frequently. The collector line layout ensures low pressure decline rates during standby and good piping efficiency during full load operation. After synchronization the turbine loading would happen quickly following the guidelines of the turbine supplier.

For a plant that operates part load and would endure start-ups frequently the modified sliding pressure would be preferred as it has significantly better part-load efficiency than constant pressure with similar start-up qualities. With a steam header layout, the running boilers could be more freely selected and thus align operating hours better with the maintenance schedules as well as providing better part load efficiency.

The nozzle-group control would be preferred for a turbine that rarely undergoes start-ups. The benefit of nozzle-group control compared to modified sliding pressure mainly comes with multiple pressure levels as lowering the HP circuit pressure affects the LP steam production and the LP steam could become useless. With such layout the load response is highly dependent on the LP circuit as it takes long time to heat. With one pressure level the modified sliding pressure would still be preferable as it improves part load efficiency, and the more expensive regulating stage would be less important.

From pure thermodynamic point-of-view the two-pressure system wouldn't cause issues with regular turbine start-ups. However, there are also financial and mechanical concern related to the subject. Two-pressure system would require a separate LP circuit with a steam drum, piping etc. so the profitability of the system with limited operating hours is a question as well as the mechanical factors. Two-pressure level WHR system utilizes roughly two times higher pressures in the HP circuit than a one-pressure level system. This higher pressure corresponds to thicker materials which are more prone to wear caused by cyclic loading and even regulatory restriction might demand a dampened start-up sequence.

7.5 Points for further research

As mentioned in the introduction, the start-up costs associated with a power plant are caused by the wearing of components during the starting process. Mechanical, structural and material investigations are required to determine the start-up costs and lifetime effects for the boilers and related components to understand better the restricting factors besides thermodynamic considerations and how they could be mitigated. The most important topics are the damper restriction requirements from structural point-of-view and the stresses opposed to the steam drum, headers and piping.

Also, more thermodynamic research should be done with the drum level behavior and the pressure and temperature decline in drum and piping during standby. The drum level behavior is an important phenomenon to understand properly as it is one of the restricting factors in the start-up especially when the drum is not hot. With proper understanding, solutions such as tilted evaporator tubes could be assessed whether they could allow for faster damper opening or primary cycle loading.

By studying the standby heating losses better comprehension could be obtained regarding hot starts and the piping efficiency as partly the same losses also affect during operation. A proper model to simulate the standby behavior could help with preserving heat inside the boiler and the critical components to boost hot starts and to help with topics like drum selection.

The topics discussed in this work were mainly done from the point-of-view of waste heat recovery from ICE exhaust gases. With these applications the pressures and temperatures are quite moderate and thus these results aren’t directly compatible for CCP with higher secondary cycle pressure levels and hotter exhaust gases. With higher pressures the steam drum size is typically smaller as denser steam has lower surface vapor velocity and thus enables smaller steam space. Due to swelling this could lead to tighter interlocks regarding damper control. The same is true with hotter exhaust gases as more heat-resistant materials might be more sensitive for temperature gradients and the superheating cooling during start-up could become a major concern.

8 SUMMARY

With the increasing requirements for flexible power generation the secondary cycle of a combined cycle power plant was studied with the aim of improving the start-up properties. The total combined cycle plant was reviewed from the point-of-view of waste heat recovery and the start-up and response related qualities were broken down to pieces.

The steam boiler’s potential for improved start-up times were associated with the boiler’s thermal inertia and the interlocks ensuring safe heating of the boiler components and stable behavior of the drum water level. With the steam piping the interlocks were associated with the line heating mainly to ensure condensate free lines and to prevent droplet carryover to the steam turbine. The turbine itself requires an excessive heating process for cold starts, but with hot starts the bottlenecks are more related to auxiliary systems, namely the sealing steam and the vacuum system.

A tool to simulate the boiler start-up process was build and it showed great accuracy when comparing to a real-life boiler. The tool was used to assess the effects of the steam drum size and the sequence to the total boiler start-up time. The drum size showed clear, yet rather minor effect to the rate of pressure gain, but other concerns related to standby pressure decline and the exhaust gas damper control meant a larger drum should be preferred until further research. With the sequence mainly the blowout and drum level set-point control were evaluated, and it was found that the blow out sequence has big impact on the total boiler start-up. An adjustable blow out valve would ensure safe and efficient line heating process. The drum level set-point switch doesn’t in itself effect the start-up significantly, but to ensure stability a good practice is to coincide the switch with the blow out start and to do it in calm manner.

With the steam turbine the main concern for hot starts is to ensure the availability of sealing steam for the labyrinth seals and motive steam for the ejector if it’s selected for hogging. The condenser evacuation takes a few minutes before steam can be admitted to condenser and the turbine sped up. It was found that the auxiliary steam system should enable operating with saturated steam as it can take a long time before superheated steam comes available for the turbine. The hogging system selection between mechanical pumps and steam ejector doesn’t affect the length of the evacuation process significantly if designed correctly, but again the question comes from the availability of steam.

The turbine governing systems affects the turbine start-up mainly because if a nozzle group control is used the regulating stage is time consuming to heat. Thus, for plants with increased part load running and regular start-ups a modified sliding pressure scheme should be selected as it mitigates the throttling losses also boosts the HRSG efficiency during part load.

More research is needed with the thermal stress factors exposed to boiler to determine the most profitable rate of start-up for the critical components. The feasibility of an added pressure levels also comes down partly to the low-cycle fatigue related to start-up. From thermodynamic perspective a two-pressure level system doesn't suffer from significantly increased start-up times. Other points of further research come from the start-up sequence interlocks, mainly the drum level behavior and superheater temperature gradients as well as heat losses during start-up.

The goals of the thesis were to identify the start-up related bottlenecks and the most suitable physical solutions for a secondary cycle experiencing frequent start-ups and to optimize the start-up sequence. Excluding the concerns of further research these goals were met and many sequence-related solutions were found and the guidelines for the effect of physical solutions were created. The created simulation tool also showed great potential to be used as a tool to find the best solutions for the future projects.

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