Integrated process and control modelling of water recirculation in once-through boilers during low load and transient operation

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01 2020

Submitted to the Department of Mechanical Engineering at the University of Cape Town in partial fulfilment of the academic requirements for a Masters of Science degree in Mechanical Engineering

Key Words: Boiler, once-through, recirculation, control, process model

Abstract

Power plant stability at lower loads is becoming ever more important, highlighting the increasing requirement for the development of advanced models and tools to analyse and design systems. Such tools enable a better understanding of the thermo-fluid processes and their dynamics, which improves the ability to specify and design better control algorithms and systems.

During low load operation and transients, such as start-up and shutdown, the required water flow rate through the evaporator tubes of once-though boilers must be significantly higher than the evaporation rate to protect against overheating of the tubes until once-through operation is reached. Controlling the minimum required water flow rate through the evaporator and economiser is notoriously difficult. Within industry, strong emphasis is placed on maintaining the minimum required flow through the economiser and evaporator without adequate consideration of the potential thermal fatigue damage on the economiser, evaporator and superheater components and the risk of turbine quenching incidents.

The purpose of this study was to develop an integrated process and control model that can be used to study transient events. The model developed in Flownex can simulate the complex thermo-fluid processes and associated controls of the feedwater start-up system. This includes the water-recirculation loop, and allows for detailed transient analysis of the complete integrated system. The model was validated using data from an actual power plant in steady state as well as a transient cold start-up, up to once-through operation. Transient results from the model are also compared to the power plant unit during start-up for the addition or loss of mills using the existing control strategy. The model results compare well with the actual process behaviour.

A new control strategy was then proposed and tested using the model. The results indicated significant improvement in control performance and overall controllability of the start-up system, and the large temperature fluctuations currently experienced at the economiser inlet during transients were significantly reduced. The new control strategy was also implemented on a real power plant unit undergoing commissioning. During all modes of start-ups (cold, warm and hot), as well as transients, the performance of the control system showed significant improvement, with a notable decline in instabilities of the feedwater flow. As predicted in the model, the large temperature fluctuations are significantly reduced.

The new model therefore enabled the development of an improved control strategy that reduces damaging thermal fatigue. The general controllability of transients is also significantly improved, thereby minimizing risks of water carry-over, quenching and unit trips during start-up.

Declaration

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Name Date

Acknowledgements

Firstly, I would like to thank and give honour to our Father in heaven for the opportunity. Without His grace, strength and divine intervention, this study would not have been possible.

Secondly, I thank my lovely wife for her patience and support and for the daily inspiration she provides me.

Thirdly, to my supervisor Prof. Pieter Rousseau. I actually do not have words to describe my gratitude. I would especially like to thank him for helping and supporting me through some very difficult personal circumstances faced during this study. His knowledge and wisdom is truly inspirational, not only technically and professionally but personally as well. I would not have completed this study without his guidance, patience and supervision.

Also, I would like to thank Mr. Philip du Toit from Eskom, for providing and helping with data extraction from the power plant (DCS), his design and implementation of the control system changes on the plant and for making use of his data extraction and graphing tools. Without his help, implementation of the controls would not have been possible. Another Eskom colleague that needs recognition is Mr. Gary de Klerk, who was also the Industrial Mentor for this study.

Lastly I would to thank the EPPEI (Eskom Power Plant Engineering Institute) programme for acceptance into the program and for providing the opportunity to conduct this study and to Eskom for partly funding this study.

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List of Nomenclature

Greek letters and Notation

α	Alpha (weighing factor between the new and old time levels)
A	Area
g	Gravtitational acceleration
h	Enthalpy
H	Height (elevation)
L	Length
$\dot{m}$	Mass flow rate
N_{pch}	Phase change number
N_{sub}	Sub-cooling number
p	Pressure
$\dot{Q}$	Heat rate
ρ	Rho - Density
t	Time
V	Volume
v	(No subscript) Velocity
v	(with subscript) specific volume
$\dot{W}$	Work rate
x	Differential length
z	Elevation

Acronyms and Abbreviations

ASME	The American Society of Mechanical Engineers
avg	Average
BCP	Boiler Circulation Pump
BMCR	Boiler Maximum Continuous Rating
CCPP	Combined Cycle Power Plants
Eco	Economiser
EPRI	Electric Power Research Institute
Evap	Evaporator
HRSG	Heat Recovery Steam Generator

PI	Proportional-Integral
PID	Proportional-Integral-Derivative
MCR	Maximum Continuous Rating
min	Minute or minutes
NPSH	Nett Positive Suction Head
sat	Saturated
SI	International System of Units
SH	Super-Heater
ss/SS	Steady State

Definitions

Benson Boiler	Typically refers to a once-through boiler, however the "Benson Boiler" is trademarked.
Benson Point	Typical phrase used within industry that refers to the point at which all feedwater is fully evaporated and exits the evaporator as either saturated steam with a quality close to or equal to 1, or slightly superheated. The point where there is no more recirculation (BCP is switched off). Typical values are around 40% BMCR.
Boiler	Steam Generator – These terms are used interchangeably throughout the text
Feedwater	Demineralised water in the power plant used for conversion to steam (typically downstream of the Deaerator storage tank up to and including boiler circulation system.
Mass flow	Refers to mass flow rate
Tank	In the context of this study refers to the two-phase tank used in simulations representing the collecting vessel.
Quality	Thermodynamic property of a two-phase mixture describing the mass fraction of vapour to the total mixture mass.

Units (SI)

GW	GigaWatt
J	Joule
K	Kelvin
kg	kilogram
kJ	Kilo Joule
kW	Kilo Watt
m	meter
mm	millimeter
MW	Mega Watt
MWth	Mega Watt thermal
s	second
W	Watt
°C	Degrees Centigrade

1. Introduction

1.1 Background

Coal fired boilers continue to play a vital role in the electricity generation sector. In South Africa, more than 93% of the installed electricity generating capacity is via coal fired power stations.

As more renewable energy sources are added to the national grid, the base load coal fired power stations require more operational flexibility for the system operator to balance the supply with demand. These enhanced requirements for flexibility in plant operation emphasize the importance of accurate modelling and dynamic simulation capabilities, not only as design and optimisation tools, but also to help prevent damage to plant components and ensure stable plant operation.

More than 50% of the installed capacity (including new build stations) of the Eskom fleet of coal fired stations now has once-through type boilers. These boilers offer many advantages over the older drum-type boilers in terms of operational flexibility, capacity and efficiency. However, these plants typically are specified with a minimum stable boiler load condition of around 40% of MCR (Maximum Continuous Rating). The minimum boiler load is determined by taking various factors into account such as furnace tube design (spiral vs. vertical), stable flow conditions through the economiser and evaporator, flame stability, draught group limitations, emission requirements (NOx) and coordinated load control. However, the minimum load of a once-through unit is predominantly defined at the once-through point to avoid recirculation and overheating of evaporator tubes.

Power plant stability at lower loads is becoming ever more important, thus highlighting the increasing requirement for the development of advanced models and tools to analyse and design systems. Such tools will enable a better understanding of the thermo-fluid processes and their dynamics, which should improve the ability to specify and design better control algorithms and systems. Advanced thermo-fluid models can also be a valuable tool to diagnose, analyse and to correctly determine the root causes of plant issues, such as unit trips and damage. With increased controllability of the plant, risk of component damage can be reduced, trip risks can be minimized and better load following characteristics obtained.

1.2 Problem Overview

During low load operation and transients such as start-up and shutdown the required water flow rate through the evaporator tubes of once-though boilers must be significantly higher than the evaporation rate to protect against overheating of the tubes until the 'Benson' point (once-through operation) is reached.

The downstream superheater heat exchangers are protected from liquid carry-over by water separation and collection vessels from which the liquid is re-circulated through the economizer and evaporator tubes. The water level in the collection (separator-) vessel must be controlled without negatively affecting the boiler feed water loop operation while minimizing the loss of expensive demineralized water via the drains. Figure 1 displays the main components of a typical once-through boiler start-up recirculation system.

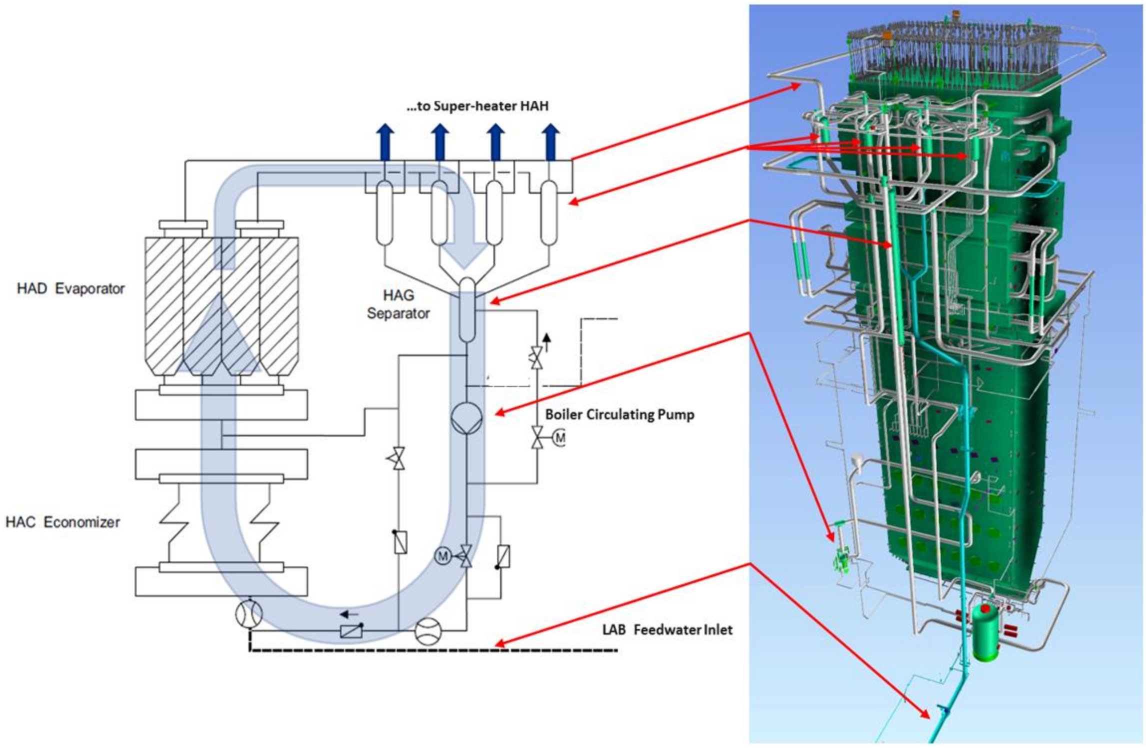


Figure 1: Typical start-up recirculation system (left: Process diagram, right: Component layout), [1]

Controlling the minimum required water flow rate through the evaporator and economiser is notoriously difficult and can become a nightmare to fine-tune (optimize controllers), especially for the inexperienced plant controller. However, this can also be a challenge for the experienced plant controller/tuner. The problem can also become more inflated with inadequate control narratives and a lack of information regarding the complete system and all the associated systems and component characteristics.

Within the industry, emphasis is placed on maintaining the minimum required flow through the economiser and evaporator. In some cases, the control strategies that are implemented will maintain the minimum flow, but can have long term consequences such as thermal fatigue damage on the economiser and evaporator. In some cases, inadequate feedwater control (collecting vessel level control) can also lead to turbine quenching incidents as well as thermal fatigue damage to superheater components when water is carried over to the superheater section. Root cause identification and subsequent control system modifications related to the feedwater system can also be problematic. Finding the root cause of unit/turbine trips and other plant issues on large scale power plants related to the feedwater system during start-up, can be a daunting task. More often than not- symptoms are identified from trip- and problem analysis and is then treated. This often leads to more problems, especially for the control system of the process. By making use of advanced simulation tools, simplified process models can be used to describe/re-run actual complex phenomena of a large-scale boiler system.

A review of literature show that a significant amount of work has gone into modelling of traditional fixed pressure drum type boilers. Most models that have been developed for once-through boilers tend to focus on steady state and dynamic characteristics after the once-through point has been reached. Accurate works that can predict and correlate to real world/actual plant transients and dynamic characteristics of the start-up system of once-through boilers seem not to be well documented in literature. Research also seems to be focussed on components and component damage downstream of the evaporator such as the superheater and reheater, as opposed to the economiser and the evaporator itself. This study aims to focus on the start-up thermo-fluid dynamics of the high-pressure feedwater system and the controllability thereof.

Two phase flow instability is another aspect that needs consideration. The phenomenon is well documented in literature in terms of the instability mechanisms as well as numerical modelling and prediction methods. However, the applicability to operational once-through boilers and suitable instability criteria do not seem to be defined. The two-phase region inside the once-through boiler can vary significantly during start-up as steam production is started in the evaporator. Modelling of the two-phase zones and predicting water levels inside the separator collecting vessel becomes even more complex during warm and hot starts as colder feedwater is introduced into hot components of the economiser, evaporator and separators.

It also seems that there are no well-defined standard or industry norm for feedwater control loops for once-through boilers during start-up and low load operation. Instead, different designers will specify different control narratives, which are not always well justified. With many different designers, different designs can be found for the control narrative, control logics and controller setups. This can create uncertainty and potential disagreement amongst different operators of these boilers.

1.3 Objective of this study

Being able to model the thermo-fluid processes and associated controls of the feedwater system (including water-recirculation loop) will allow for the detail transient analysis of the complete integrated system. This will enable analysis of anomalies and optimization to ensure stable operation at lower loads and start-ups. It will also support the development of procedures for quicker start-up and shut down transients while minimizing the risk of water carry over, component damage, unnecessary trips and wastage of demineralized water.

Modelling of the thermo-fluid processes and associated controls will also enable detailed analysis to be carried out of events such as unit trips, thereby greatly assisting in the root cause identification of these events. Changes to the control system narrative can also be tested on the model before implementation on the actual plant. This will greatly assist in finding robust solutions to seemingly complex problems in the start-up system.

The primary objective of this study is to develop an integrated system process model of a once-through boiler start-up forced recirculation system together with its controls in Flownex. The model should allow analyses of the operational procedures applied during low load operation, start-up, shut down and transients (such as unit trips). To achieve the primary objective, the following enabling objectives were defined:

i. Review literature of once-through boilers with a focus on operational/dynamic models of the start-up recirculation systems and controls. Conduct a focussed review of two phase flow instabilities and the applicability to a large scale modern once-through evaporator.
ii. Develop a process model in Flownex of a power plant evaporator and investigate numerical solver stability as well as thermo-fluid stability of the two-phase region.
iii. Develop a model of the collecting vessel with the aim to achieve simple level control.
iv. Develop a model of the complete start-up feedwater system. The model should include the economiser, evaporator, separators, collecting vessel and recirculation.
v. Create a simplified model of the boiler flue gas side.
vi. Verification and validation of the process model with the aid of an appropriate case study.
vii. Integration of the controls with the process model in Flownex.

viii. Analyse a range of low load and transient operational conditions with the aid of the new model. Validate and verify the results with a case study.

1.4 Scope of Study and Assumptions

This study will focus on the start-up recirculation system of a large scale once-through boiler. Emphasis is placed on the evaporator (two phase) dynamics during low load operation, the influence on the collecting vessel level and controllability of the complete process.

A simplified process model of a typical start-up recirculation system can be seen in Figure 2.

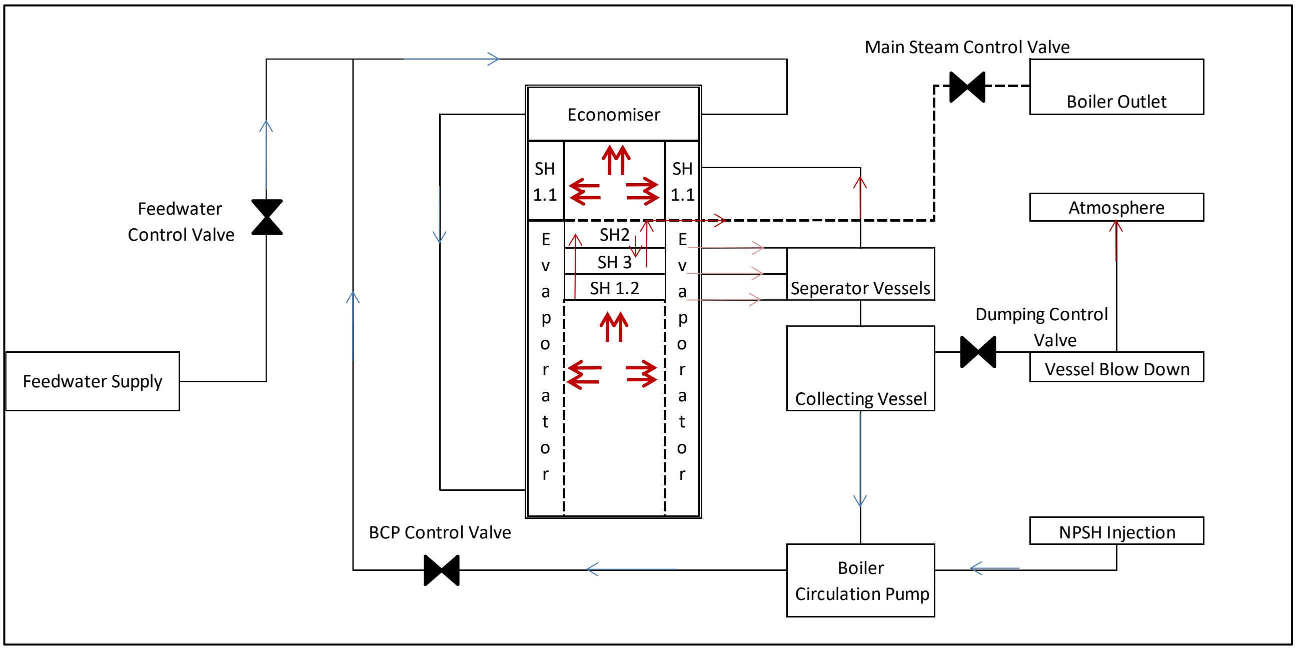


Figure 2: Simplified Process Model (Displaying System Boundaries)

Important Assumptions for this study:

i. Two-phase flow is a homogeneous mixture.

ii. Main Steam control valve can either be the HP-bypass control or turbine inlet control, this is modelled as a pressure boundary condition.

iii. Feedwater control valve is modelled as a mass source boundary condition.

iv. Boiler circulation pump and control valve is modelled as mass source boundary conditions.

v. Separator vessels are assumed to occupy volume in the system and two-phase separation occurs in the collecting vessel.

vi. Heat transfer in the furnace is modelled as fractions of the total heat input with all heat transferred to the tubes.

1.5 Dissertation Outline

Chapter 2 contains the literature review which starts off with a brief history of the once-through boiler technology. The low load characteristics of these boilers are then investigated, followed by a literature review of models that have been created for once-through boiler units. Typical component damage found in the start-up system is then reviewed followed by a survey of the control systems used for once-through boiler start-up. Traditional drum boiler controls are also investigated. The final part of the literature review contains a summary of two-phase instabilities regarding the mechanisms, predictability and application to large evaporator systems.

Chapter 3 contains the methodology that was followed to build the complete and integrated process model. Starting with the modelling of the evaporator and setting of boundary conditions. A fundamental steady state analysis is conducted followed by a summary of the transient solution methodology. This chapter also contains a detailed description of the basic process model, followed by a detailed description of the complete integrated process and control model.

Chapter 4 contains the validation of the model carried out based on an actual power station unit (case study) for steady state as well as a transient cold start-up.

In Chapter 5 the model is applied to conduct a steady state analysis followed by a study of the transient behaviour of the process without the controls. Next, the controls are activated and simulations are conducted to study changes in boiler load. Plant data from the case study is then used as boundary conditions and the results are compared to the actual plant behaviour. A proposal is made to change the control philosophy, followed by simulations of the new control and comparing the results to the previous control.

Chapter 6 contains the results of changed control strategy that was implemented on one of the units used for the case study. A general comparison of the control performance before and after implementation of the changed control strategy is presented.

Chapter 7 contains the conclusions and recommendations of the complete study with the main conclusion reached that the process and control model has led to the development of an improved control narrative.

2. Literature Review

2.1 Once-through boiler development history

Up to now, the most manufactured boiler in the world is the once-through type steam generator [2]. Development of the first coal fired steam generators/boilers built for power generation started in 1866, and the first power station was built in 1882 in New York City [3]. In South Africa, the first steam driven power stations were commissioned in the early stages of the 1890's. In these early stages the plants generally generated less than 1 [MW]. The early 1900's saw the development of larger units rated up to 10 [MW]. Developments in steam turbine technology, introduction of steam turbine extractions to preheat feedwater and the introduction of pulverised coal in the 1910's lead to the development of larger boilers. The once-through boiler concept was introduced in the 1920's, as well as the introduction of the reheat cycle. According to Franke [2], Mark Benson applied for a patent in 1922 titled; "process for the generation of working steam ready for use at any desired pressure". In 1924, Siemens acquired the rights to Mark Benson's patent and subsequently built the first Benson boiler in 1926/1927. The once-through boiler is also commonly referred to in industry as the Benson boiler, however "Benson Boiler" is trademarked and these boilers are manufactured under licence from Siemens.

In the 1930's the drum type boiler was the most popular and electrical output from these units had increased to 300 [MW] [3]. These units were still operating at subcritical pressures. The pressure limit for drum type boilers is below the critical pressure of water (around 22.1 [MPa]), as the drum serves as the evaporation endpoint. The world's first supercritical once-through unit was commissioned in 1954 [4]. Typical main steam conditions reached throughout the 1960's were around 650°C and 345 bar [3]. In the early 1960's the first 1000 [MW] unit went into operation and was followed by other units reaching 1300 [MW]. Even though the first Benson boiler was equipped with a spiral wound evaporator, most units constructed in those early years were equipped with vertical wall tubing. It was only in the 1960's that the spirally wound evaporator design matured in the form of parallel membrane walls and was incorporated to eliminate the undesired effects of uneven heating within the furnace. However, the spirally wound evaporator does carry high manufacturing and assembly costs. Despite these achievements, the once-through boiler was not the standard boiler of choice for utilities. The once-through boiler grew in popularity in later years due to the advantages of higher efficiencies and flexible operation.

The predominant advantages of the once-through boiler, compared to the drum boiler, is the higher thermal efficiencies that can be achieved with supercritical steam conditions, better load changing capabilities when operated at sliding pressure (variable pressure) operation, as well as the reduction of thermal fatigue on thick-walled components. The most recent advancements made to the once-through boilers are the internally ribbed or also called internally rifled tube evaporator designs. The concept was first proposed by Sulzer in the 1980's and was first installed at a power plant in Japan, and was available from other reputable boiler manufacturers in the late 1990', [5]. The advantages of a vertically arranged ribbed tube evaporator compared to a smooth tubed, spirally wound evaporator is lower fabrication and assembly costs and better heat transfer characteristics (explained in more detail below). Perhaps the biggest advantage offered by the internally ribbed tube evaporator is that the once-through boiler can maintain once-through conditions down to 20% load, compared to smooth tube evaporators typically rated around 38% load. The ribbed tube evaporator offers significantly improved low load characteristics for operators looking for maximum flexibility of coal fired plants.

2.2 South African Power Utility (Eskom) context

In South Africa the first large coal fired once-through boilers came online in 1976 at Kriel followed by Duvha in 1980, Tutuka in 1985, Matimba in 1987 and Mujuba in 1996 [6]. Although these units were once-through, they are operated at sub-critical pressures. The first supercritical units in South Africa are Medupi and Kusile with their first units that came online in 2015 and 2016 respectively. As in other parts of the world, the once-through boiler was not the only type that was built in the late 20th century. Forced and Natural circulation drum boilers were still popular and widely procured. The South African power utility, Eskom, currently owns and operates 42 (upon completion of Medupi and Kusile projects) once-through type boilers with a combined capacity of almost 28 GW. Table 1 provides an overview of the large-scale coal fired power stations of the Eskom fleet.

Station Name	Commisioned	Power Station AGE	Config	Installed Capacity [MW]	Avg Unit Capacity [MW]	Boiler Design
Arnot	1974	45	6 x 400MW	2400	400	Forced Circulation
Camden	1969	50	8 x 200 MW	1600	200	Natural Circulation
Duvha	1984	35	6 x 600 MW	3600	600	Once Through
Grootvlei	1977	42	6 x 220 MW	1320	220	Natural Circulation
Hendrina	1976	43	10 x 200 MW	2000	200	Natural Circulation
Kendal	1993	26	6 x 686 MW	4116	686	Natural Circulation
Komati	1962	57	5x100MW, 4x125MW	1000	111	Natural Circulation
Kriel	1979	40	6 x 510 MW	3060	510	Once Through
Kusile	2016 est.2024	0-3	6 x 800 MW	4800	800	Once Through
Lethabo	1990	29	6 x 618 MW	3708	618	Natural Circulation
Majuba	2001	18	3x657MW, 3x712MW	4107	685	Once Through
Matimba	1993	26	6 x 665 MW	3990	665	Once Through
Matla	1983	36	6 x 600 MW	3600	600	Natural Circulation
Medupi	2014 est.2022	0-5	6 x 794 MW	4764	794	Once Through
Tutuka	1990	29	6 x 609 MW	3654	609	Once Through

Table 1: Overview of large coal fired power stations in South Africa, [1] & [6]

2.3 Low load operation

Benson and/or once-through type boilers with smooth tube evaporators are typically specified with a minimum load of between 35-40% BMCR (Boiler Maximum Continuous Rating). This corresponds to the point where all water is evaporated in the evaporator, i.e. when the steam quality at the outlet of the evaporator is one, or slightly superheated and there is no liquid collected in the separators or collectors. This minimum load is also commonly referred to as the once-through point or Benson point, where all the sub-cooled liquid entering the boiler is fully generated to steam.

As mentioned previously, recent developments in evaporator design with ribbed tubes have made it possible for once-through units to maintain stable minimum load operation at around 20% BMCR [7]. The spirally wound evaporator tubes were initially developed as an improvement to the vertically tubed walls in once-through boilers. In a vertically tubed evaporator, individual tubes or sections of tubes that are subjected to excessive heat input can cause temperature variations within individual tubes as well as at the outlet of the evaporator. In the spirally wound evaporator tubes, each tube is essentially routed through all the heat transfer zones and thus compensates (to some extent) the maldistribution of heat within the furnace.

The mass flux is also an important parameter in the thermal-hydraulic design of the evaporator, as sufficiently high mass flux is required to avoid boiling crisis. Boiling crises occurs at certain steam qualities when wetting of the tube wall is no longer possible, and so called ‘dry-out’ occurs. Heat transfer to the fluid at this dry-out location is reduced and leads to an increase in the tube wall temperature. To ensure that the heat transfer does not lead to significant increases in tube wall temperatures the mass flux must be sufficiently high. The internally ribbed tube design compensates to a large extent the effect of dry-out as the ribs provide a swirl which has enough centrifugal force on the fluid to keep the tube internally wetted. The swirl that is created, forces the liquid to the wall and thus sufficient heat is transferred to the fluid to avoid a temperature increase of the wall. Hence there is a big difference between in the minimum loads that can be achieved by a smooth tube spirally wound evaporator (typically 40%) and internally ribbed tube evaporator (20%) as the minimum required mass flux can be reduced. The lowest value for once-through boiler load found in literature is 15%, as reported by Starkloff et al, 2015 [8]. However, verification of the source of this information could not be done.

2.4 Modelling of boiler start-up systems

From the Literature review, published work on Combined Cycle Power Plants (CCPP) and the Heat Recovery Steam Generators (HRSG) seem to be more popular than coal fired plant with regards to start-up dynamics. As the amount of renewable energy sources (specifically wind and solar) are added to grids worldwide, the response and dynamic characteristics of conventional power plants have received more attention as more operational flexibility is required from them. The quick start-up time that can be achieved by Gas Turbines and CCPP make them ideal to relieve/mitigate the negative effect that high penetration renewables has on the active power requirement of power grids. These plants are designed and operate under more frequent start and stop operations and are more flexible than the conventional coal base load power plants. For this reason, it seems that more focus has recently been placed on the starting behaviour and dynamics of the Heat Recovery Steam Generators. However, modelling of the once-through coal fired steam generators can be found in literature.

The Heat Recovery steam generators are traditionally fitted with natural circulation boilers. However, in the past two decades more of these plants are fitted with once-through boilers, due to their increase in operational flexibility [9]. The HRSG's are different to conventional coal fired steam generators. With advancement in technology and demand for higher efficiency and shorter start-up times, the newest HRSG's are fitted with multiple pressure stages consisting of combinations of natural circulation drums and once-through sections. This makes these steam generators quite different to once-through coal fired steam generators. However, the HRSG's do have similar aspects with regards to evaporator modelling, level- and feedwater controls. As such some of the modelling of these systems were included in the literature review.

N. Mertens *et al*. 2015 [9] used an existing, design validated model [10] of a once-through HRSG and compared the model to a drum type natural circulation HRSG. Both models were developed with the software package APROS [11]. Their model aimed to show the differences when replacing the once-through high pressure system with a natural circulation system during start-up and transient events. Their results show, and is also consistent with coal fired boilers, that the pressure build-up in the once-through boiler is generally quicker. Their results also showed a larger temperature differential in the thick-walled drum of the natural circulation unit when compared to the separator of the once-through unit. In their model, which is based on the same model and control of Alobaid *et al*. 2012 [10], the feedwater setpoint during start-up is not defined to sufficient detail to determine if a minimum flow is maintained, and to which extent feedwater is dumped or recirculated. However, the principle of control of the separator level is to control a fixed level until 30% load is achieved. The feedwater control then switches to Benson mode, where the feedwater setpoint is derived from the heat load and by taking various other parameters into account. These parameters include the available heat that can be absorbed by the evaporator and the enthalpy setpoint at the inlet of the separator. The signal is then corrected by a derivative element considering additional heat output of the metal masses of the evaporator, fluid side mass storage in the economiser, degree of sub-cooling at the evaporator inlet, degree of superheating at evaporator outlet (Benson mode) or steam quality (level mode) and attemperator flow as a function of feedwater flow. Although their model is focussed on the differences between natural circulation and a once-through section, an observation regarding the separator level is made. During both hot- and warm starts there are some level anomalies as depicted in Figure 4 (solid blue lines). The authors do not provide an explanation of the level anomalies, except for the relatively quick diminishing of the separator vessel level at around 26 minutes after start. They attribute the level change due to the small volume of the separator, where the water inventory is quickly evaporated and subsequently the temperature rises quickly (compared to the drum).

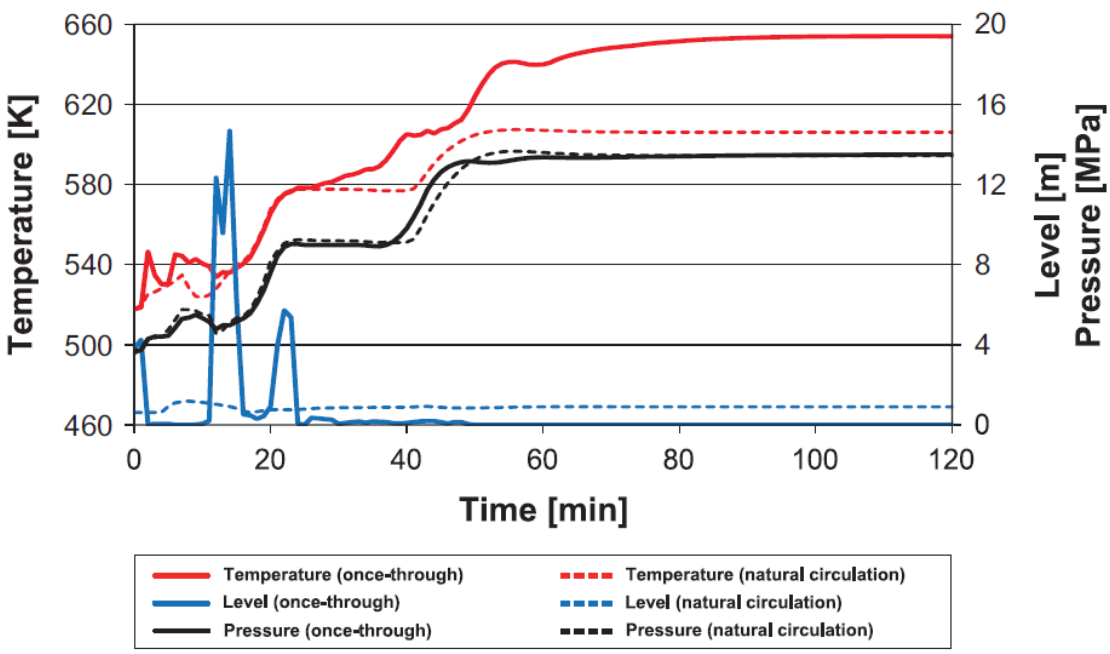


Figure 3: Dynamic behaviour of HP drum/separator during hot start-up. [9]

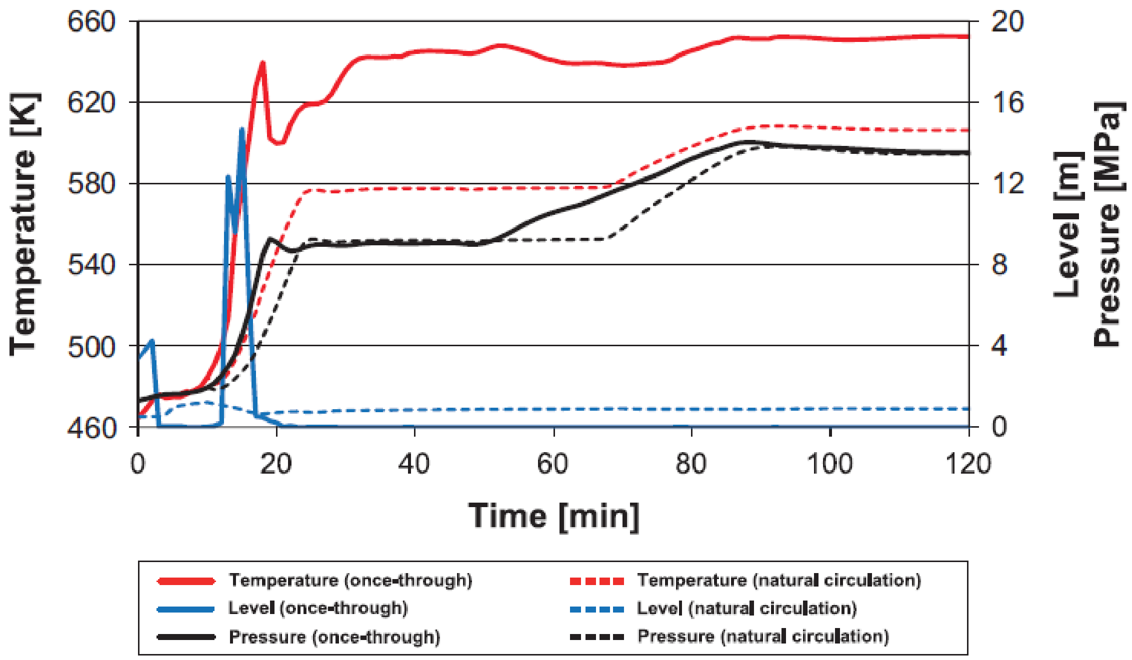


Figure 4: Dynamic behaviour of HP drum/separator warm start-up. [9]

Deng *et al.* 2017 [12] developed a numerical model of a supercritical once-through boiler start-up system. The system was also simulated by using the commercial software package, APROS. The superheater and economiser are modelled as homogenous fluid conditions to solve the mass, energy and momentum equations. The model also includes a six-equation solution for the two phase flow through the evaporator where one dimensional conservation of mass, energy and momentum were used. The evaporator also consists of three different heat transfer zones: liquid zone, transition zone for two phase flow and a vapour flow zone. Validation of their model was carried out by comparing the simulation results to design parameters. Steady state validation was performed at 40% load of the Turbine Heat Rate Acceptance data, which showed good correlation. The dynamic simulations were carried for a cold start and start-up after 72 hours. The model simulated the start-up system from initial firing up to 30 % BMCR (Boiler Maximum Continuous Rating). In both dynamic simulations there is only one spike in separator/collecting vessel level but is corrected by the controller elements nicely. The transition to once-through operation is also very smooth.

It must be noted however, that some plants can experience many transient level variations during start-ups, and other models have also shown some variations during the transition from circulation to once-through operation. The once-through HRSG of Mertens *et al*. 2015 [9] as example, showed variations in the level as well as steam flows during transition. Deng *et al.* 2017 [12] observed in their model a so called 'false water level phenomenon', when steam is generated in the evaporator. They do not offer a detailed explanation/description of the phenomenon, but it seems that they attribute a sudden increase of the separator vessel level as the 'false water level phenomenon', which is then controlled with a drain valve to a value below the alarm value. Figure 5 below displays the water level results they obtained during a start-up simulation.

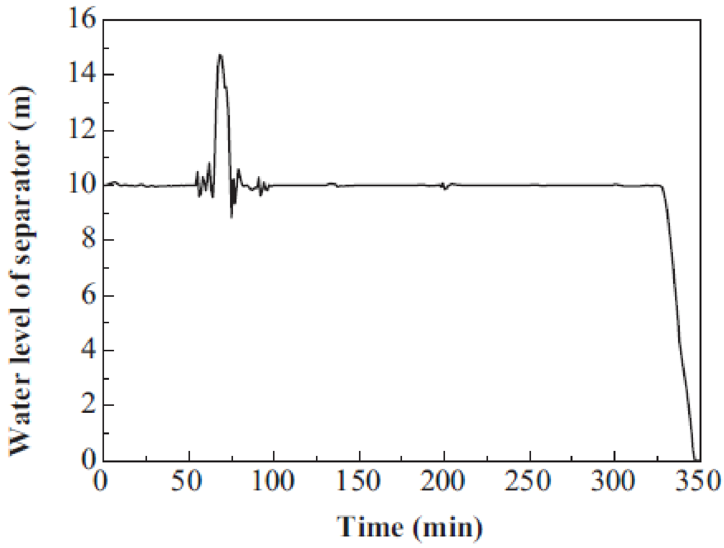


Figure 5: Water level of separator for cold start after 72 h shutdown, Deng et al. [12] .

Liu *et al*, 2015 [13] developed a dynamic model of a supercritical once-through boiler which is viable for the design of the overall control system. The model validation between 50% load and 100% load proved that the model was accurate enough for overall controller design. However, the model they developed was very simplistic as the heating elements (economiser, separator, evaporator and superheater) of the boiler was modelled as a heating tube and they only considered the state changes at the inlet-, outlet- and representative points of the boiler. The model structure was derived from fundamental energy and mass balance equations and data analysis. The model did not include the addition of momentum conservation equations. The model was extensively simplified by avoiding the phase transition zones and positions. Even with such simplifications they did observe that the separator enthalpy is sensitive to imbalances of mass and energy. The Boiler start-up system was not considered in the model, however they did also observe higher enthalpy deviations for validation at the separator outlet at lower loads.

Hubka, 2011 [14] presented a numerical model of the temperature dynamics of a once-through boiler. The model was created in Matlab/Simulink. He placed emphasis on making use of the model for control design and optimisation, however low-loads and start-up scenarios were not included in the model. The model was created during the design phase of a project and verification was based on measured data from a power plant. The model is mainly focussed on the main steam temperature control optimization following a disturbance. His results indicate that accurate tuning of the control system can be done based on modelling results. He also noted that the evaporator is one of the most difficult components to model due to the phase change that occurs. The phase change from water to steam causes rapid changes in the density and heat capacity of the fluid with resulting small changes in outlet temperature. The model of the evaporator is solved as a fully distributed model (where the heat capacity and density were not averaged across the heat exchange area). Figure 6 below displays the temperature array he obtained when changing the power level (heating) by 10%, demonstrating sufficient level of detail for the evaporator modelling.

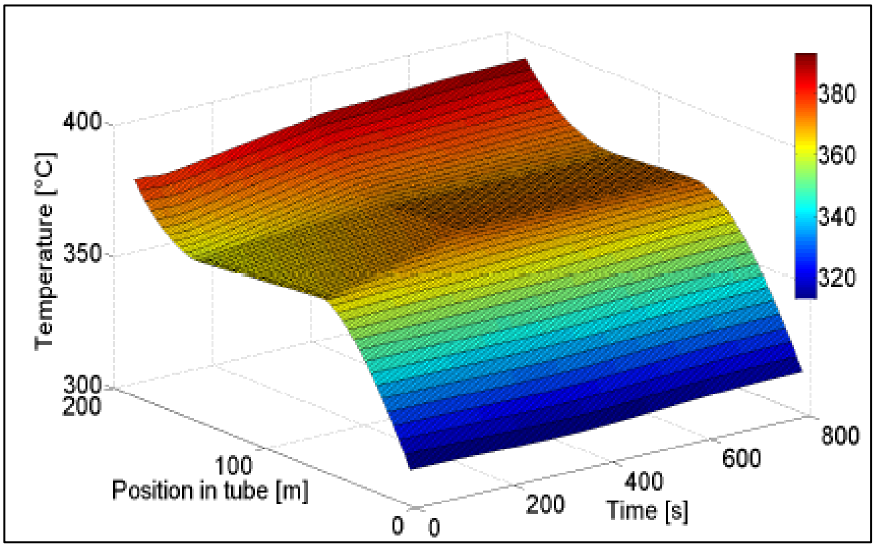


Figure 6: Temperature fields in the evaporator for 10% change in heating level, [14].

Although Hubka 2011 [14] did not present modelling results at low loads (<50%), he clearly demonstrated the density space distribution for different power levels. As depicted in Figure 7 below, the change in density along the tube length becomes more linear at higher loads (100% power level, yellow) and a lot less linear, becoming almost hyperbolic at lower loads.

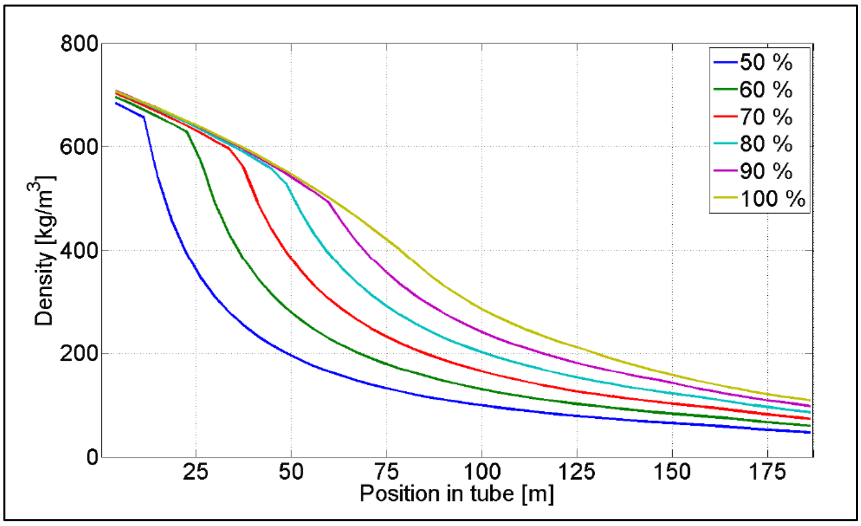


Figure 7: Density space distribution for different power/operational levels, [14].

The model created by Hubka demonstrates that evaporator modelling is an important aspect of modelling the once-through boiler, and that control optimization can be done based on modelling results.

One of the earliest models found of the once-through steam generator with emphasis on the evaporator dynamics is the mathematical model presented by Lausterer *et al*. 1979 [15]. One of their simulations included a feedwater flow rate disturbance, from their modelling results it is shown how the pressure transient behaviour of the evaporator is influenced by heat transfer characteristics. By implementing a delay of heat flux to the tube, they were able to simulate the delay in wetting of the hot walls (film boiling) to bring the simulated and measurement data in close agreement. However, the simulations were carried out at 50% load.

Trangbaek, 2006 [16] created a low load simulation model of a 400 [MW] once-through boiler including recirculation. His objective was to create a model for controller design. The simulation model was created by using physical parameters and measured plant data during closed loop operation. The model was developed to be of low order and reasonably fast. He achieved simulation times approximately four times faster than real time. The economiser, evaporator and superheaters were each modelled by lumped parameter control volume with three state variables – steam mass, steam enthalpy and wall temperature. A simple mass and energy balance is applied to each volume and a pipe across which the pressure drop occurs. Heat input from the furnace to the wall was simplified as the fuel flow multiplied by a constant gain for each of the control volumes. Simplifications were also applied to the separator and two-phase flow separation. The simplifications lead to ordinary differential equations that are then solved using a backwards difference type scheme. He used the feedwater flow reference, fuel flow, injection water flows and the turbine valve position as open loop inputs (boundary conditions) in the simulation when fitting data. He noted difficulties in modelling the evaporator, as different correlations are needed when the boiler operates in once-through mode and recirculation mode. He also notes that the bottle (collecting vessel) level measurements are noisy and are used to correlate the fraction of steam flows to the superheater.

In his remarks and experiences from the modelling he noted that the recirculation coolant flow can be quite cold and significantly affects the system behaviour. He also notes that the proposed model is not very flexible in terms of being able to modify the dynamic behaviour by adjusting the parameters and proposes splitting of the individual control volumes into smaller control volumes, however this would increase the simulation time. He attempted a local linearization approach to decrease simulation times but notes that the approach might not work in all situations (load ranges) due to the non-linear behaviour of the system.

Botha 2016 [17] presented a transient model of a once-through helical coil steam generator typically found in modular nuclear reactor designs. He created a homogenous two-phase flow model EES (Engineering Equation Solver) that was able to simulate steady states as well as transients. This model was then verified with a Flownex model and showed good correlation. He also showed that the coil geometry could be simplified and represented by vertical parallel pipes with an enhanced heat transfer coefficient. He verified his model created in Flownex by comparing his results to test data from the IRIS (International Reactor Innovative and Secure) steam generator, which showed good correlation. He also modelled the transient response of the two-phase boiling of the steam generator. From his study, two noteworthy observations were made. The first is that he observed a sudden increase in outlet mass flow of the steam generator when boiling starts. The sudden increase in mass flow (almost doubled) also coincides with a rapid drop in inlet pressure, after which the inlet pressure and mass flow recovers to new steady state values. The increase in mass flow is explained by the sudden decrease in fluid density at the onset of boiling. The second was his investigation into the numerical solver stability of his model. He studied convergence/divergence of the source terms by following the Explicit approach (α=0), Implicit approach (α=1) and Crank-Nichols approach (α=0.5). From his analysis, he concluded that for the given geometries and boundary conditions, an alpha(α) value of 0.7 and time steps larger than 0.01[s] yielded satisfactory results to ensure stability.

Meinke *et al*, 2011 [18] presented a model of a once-through boiler, validated on Rostock power plant in Germany. Their model makes use of the open source Modelica library ThermoPower. The Boiler considered in the model is very similar to the one that is the focus of this study with regards to the start-up system and related components. Their model includes components specifically related to the start-up system of the boiler such as the cyclone separator and start-up bottle (separator collecting vessel). Validation was carried out using the steam properties in the boiler and generator output. These values were used because the dynamic behaviour of the overall process is dominated by the fuel pulverisation in the mills and transient response of the boiler and measuring these is only possible with high effort and low accuracy.

From their measured and calculated (simulated) results, interesting behaviour is noticed during the start-up of the boiler which is presented below. The water inlet temperature to the economiser and outlet temperature of the evaporator is shown for the simulation and measured values during start-up in Figure 8. The boiler is in Benson mode after 50 min. The mean relative error recorded for the evaporator outlet temperature is approximately 3.3%, whilst the economiser inlet temperature has a deviation with a maximum error of 28% between 10- and 30 min. The authors explain the deviation as being an actual higher amount of hot re-circulating water, whilst the simulation has a higher proportion of colder feedwater entering the economiser, as can be seen from the mass flow results presented in Figure 9.

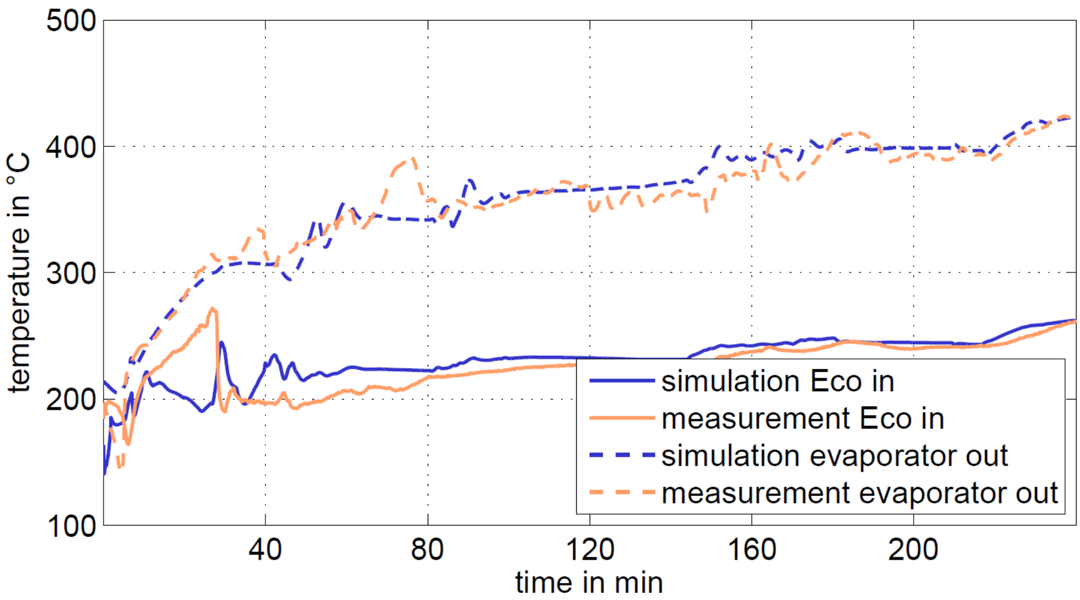


Figure 8: Comparison of calculated and measured fluid temperature at inlet of the economiser and after evaporator, [16]

Unfortunately, the start-up bottle (collecting vessel) level during start-up is not presented. From the measured mass flow results in Figure 9 there is a very quick increase in main steam mass flow, which can be associated with a sudden decrease in collecting vessel level as was the case with other models in literature, like Mertens *et al*. 2015 [9]. It is often the case that the increased steam mass flow will then lead to lower recirculation flow and higher feedwater mass flow, which lowers the eco inlet temperature quite significantly, as can be seen in Figure 8 at around 35 min. The eco inlet temperature drops by approximately 80°C – these type of temperature differentials experienced during start-up is explained in more detail under section 2.5 - Component Damage.

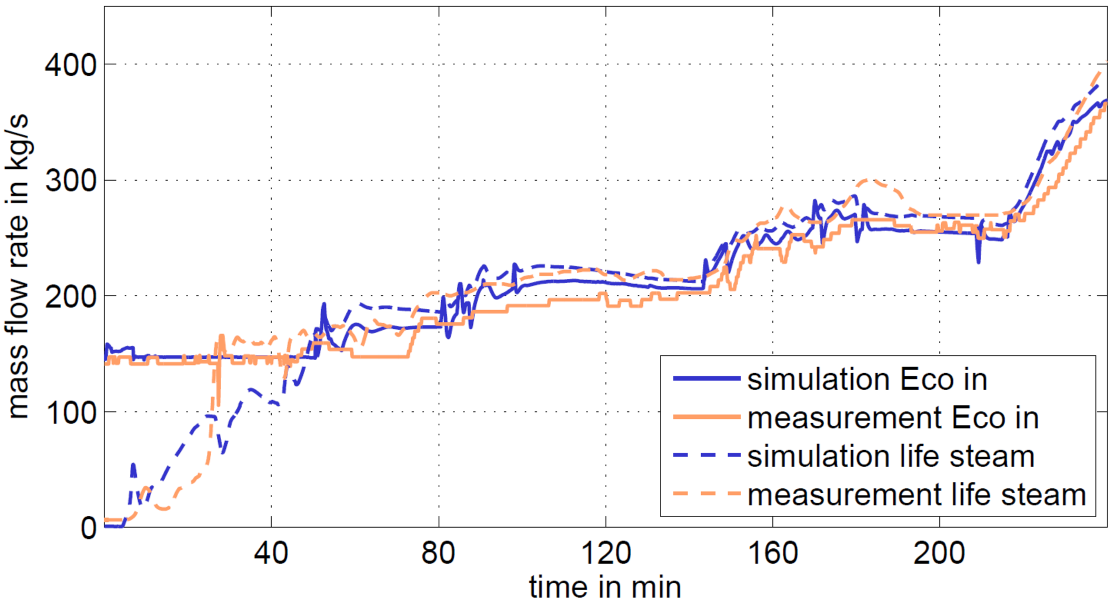


Figure 9: Comparison of calculated and measured economiser in and superheater out mass flow rates, [18].

Starkloff et al, 2015 [8], developed a full-scale dynamic model of a large scale hard-coal fired power plant in the process simulation program APROS which includes all power plant components and the associated control schemes. Their model is validated against operational data from a coal fired plant during part load transients and shows good agreement. They conducted a dynamic comparison between the model and an actual plant during a large load change, changing from once-through to forced circulation. The load range tested was an operator-initiated load change from 100% load to 22.7% load, which consisted of a series of load changes; from 100% to 67.5%, down to 42%, 33.4%, 29.7% and finally to 22.5%. The load changing scenarios are presented in Figure 10.

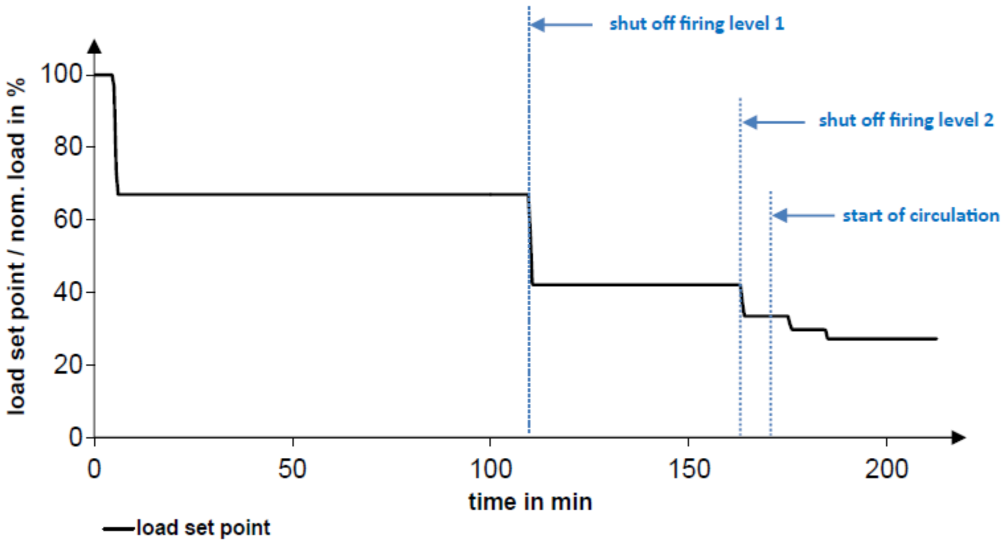


Figure 10: Load Changing Steps from 100% to 22.5%, [8].

They note that there is room for improvement as the start-up of the circulation system is too abrupt and recommend that it should be smoothened and is left for a further stage. Unfortunately, details of the abrupt starting and dynamics thereof are not presented. Their results for the start of circulation are presented in Figure 11, which displays the mass flows of the steam generator. The circulation pump is started at around 170 min and causes a slight increase in eco inlet temperature as depicted in Figure 12.

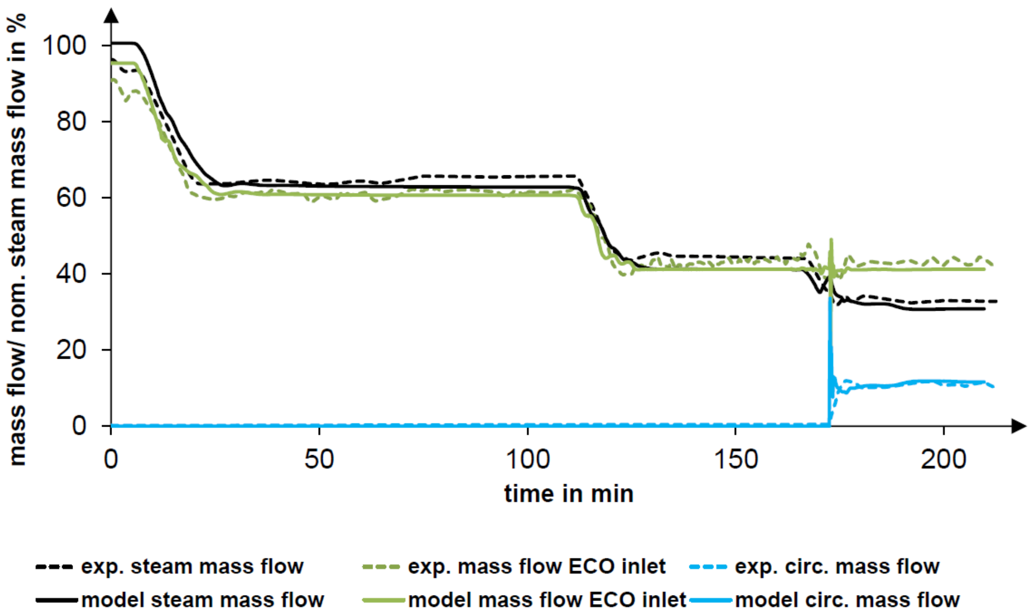


Figure 11: Mass flows in the steam generator, [8]

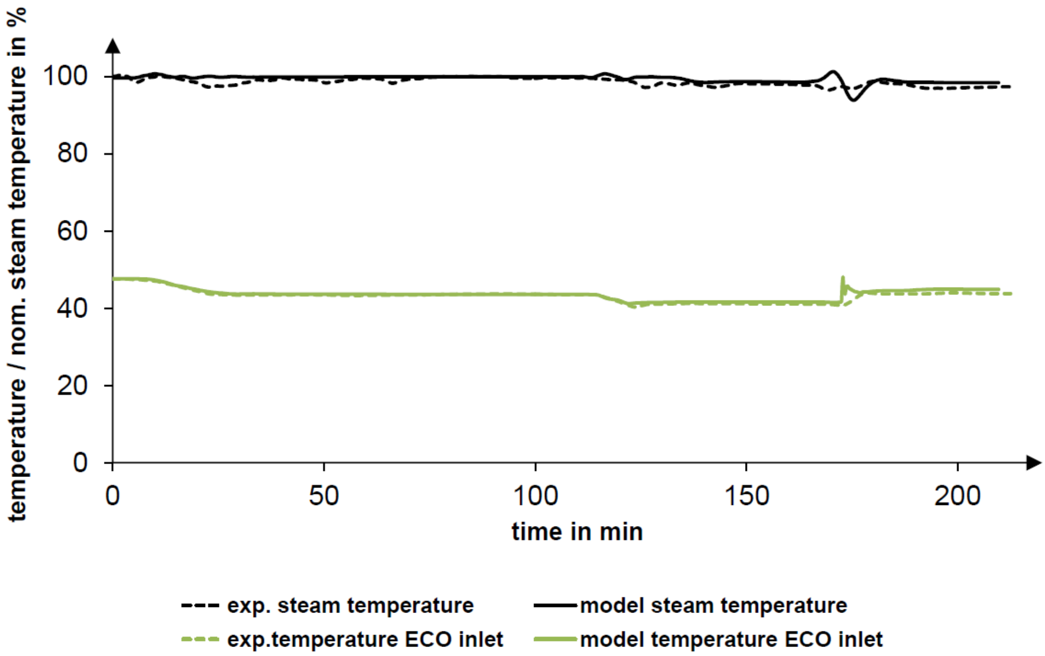


Figure 12: Temperatures at steam generator outlet and inlet, [8]

In their concluding remarks, they note that their model can be used to simulate the real power plant with high accuracy. They also mention that the model could be used to study new control concepts such as an economiser bypass, which has been studied by Boje, 2011 [19] which is reported in the next section.

2.5 Component damage

During start-up and shut down of a once-through boiler unit components of the start-up system can be subjected to various damage mechanisms. Thermal fatigue damage is one of the mechanisms that is of interest for this study. It can occur when colder feedwater is introduced into hot re-circulating water from the collecting vessel and flows through the economiser and evaporator. The rapid introduction of colder feedwater can occur during transients, shut-downs and start-ups.

According to research conducted by the Electric Power Research Institute (EPRI [20]–[23]), Thermal fatigue is mentioned as the leading tube failure mechanisms in the water walls of supercritical boilers. Thermal fatigue can have a wide variety of causes which can become complex (such as deposit build-up, thick weld overlays, higher heat flux from geometry, etc.) and is outside the scope of this study. However, major root causes of interest are the thermal transients from unit operations such as start-ups and unit/load cycling, and reduced tube flow rates originating from two phase flow instabilities (detailed in section 2.8), with resulting tube overheating.

One of the most common failures in the economiser inlet headers and tubes is also due to thermal fatigue, specifically caused by the rapid introduction of relatively colder feedwater, EPRI [20]. In coal fired units it is not uncommon to find damaging rapid temperature differentials (ΔT) of 80 °C, as seen in the actual results of Meinke *et al*, 2011 [18]. In the author's experience, ΔT's of over 100 °C have been observed. Figure 13 displays the temperature differential at the economiser inlet for a typical scenario/transient that can happen during a unit start-up. The collecting vessel level is high and the circulation flow is almost equal to minimum flow. As the level suddenly starts to decrease, the circulation flow is reduced and colder feedwater flow is increased to maintain the economiser minimum flow, resulting in ΔT's of over 100 °C. The scenario is hypothetical, as only a simple one-dimensional steady state mass and energy balance calculation is performed for the sole purpose of depicting the temperature differential mechanism at the economiser inlet.

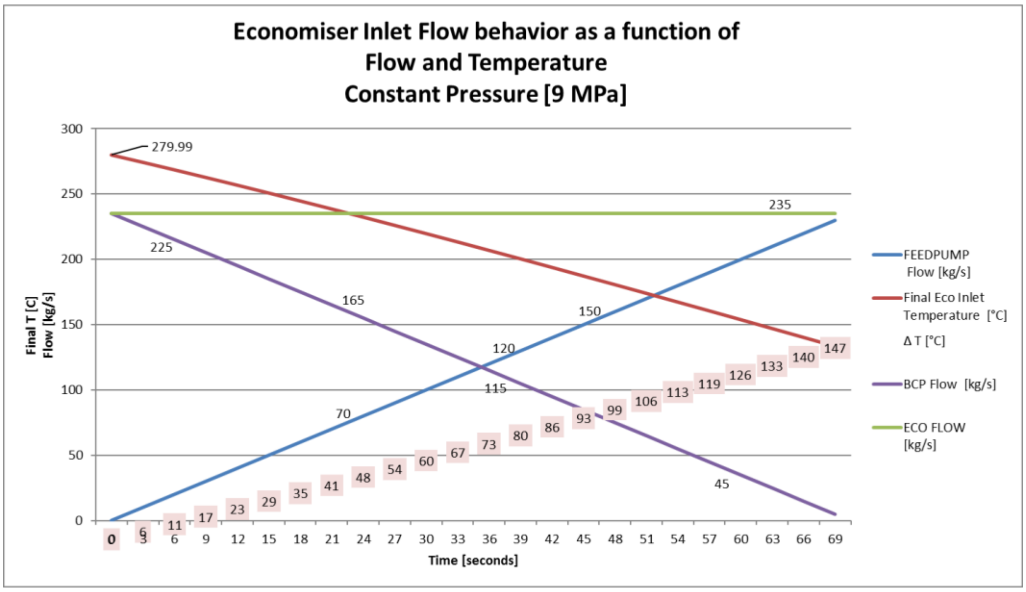


Figure 13: Hypothetical Linear Temperature Profile of the economiser inlet during Vessel Level Collapse.

Some of the operating recommendations from EPRI [20], which aims to minimize large temperature spikes is to introduce a trickle feed system, introduce intermittent feedwater flow by valving and/or increasing inlet feedwater temperature.

Other options to reduce thermal cycling within the start-up system of once-through boilers found in literature is an economiser bypass. Boje, 2011 [19] investigated the possibility of damping the thermal oscillations at the evaporator inlet by using a by-pass of the economiser. The by-pass valve investigated in his study is located upstream of the economiser and passes a fraction of the flow to the economiser outlet/evaporator inlet. The specific purpose of this valve is to reduce the temperature oscillations at the inlet of the evaporator thereby reducing the feedback temperature fluctuations to the economiser inlet caused by the recirculation system.

His simulation results for the change in evaporator inlet temperature is shown in Figure 14. He makes the conclusion that the controlled by-pass is effective in damping temperature oscillations in the evaporator inlet. Although dampening of the oscillations is achieved, large temperature differentials are still present, as can be seen between 40- to 70 min. Reduction of the economiser inlet flow could also lead to maldistribution of flow and thermal excursions However, he expects better results/performance could be achieved on the plant as the plant wide feedback effects are not included in the simulation; whereby the reduced temperature oscillations will reduce the oscillations of the hot re-circulated water mass flow rate.

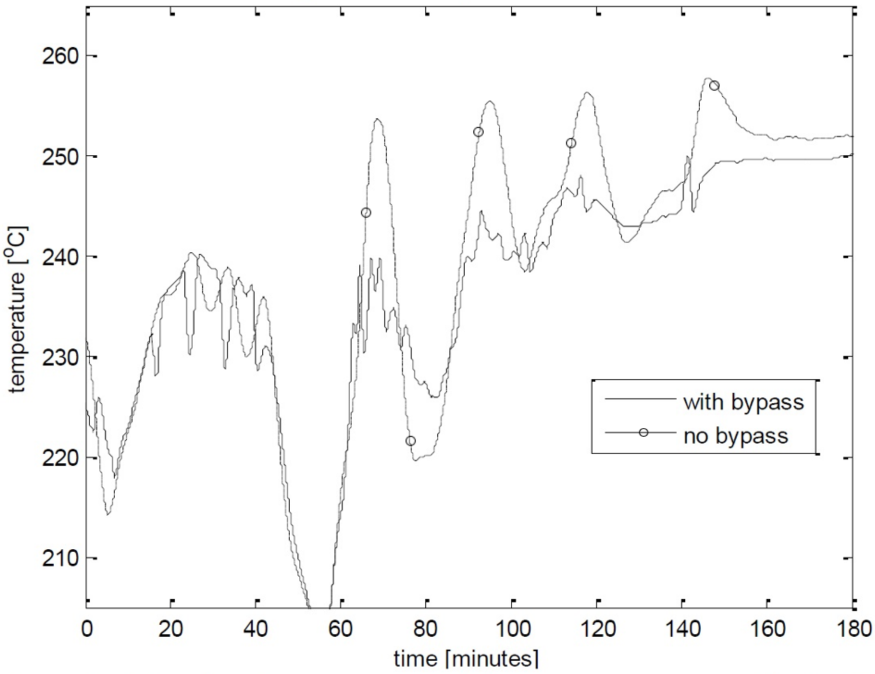


Figure 14: Simulated evaporator inlet temperature with and without control/bypass, [19]*, [circles were added for clarity and to establish distinction]*

Thermal fatigue damage related to the start-up system is not just limited to the economiser and evaporator but can extend beyond the separators into the superheaters, and high-pressure Turbine. The thermal fatigue of these downstream components is largely related to so-called 'over-feeding' or 'overflow' events. These occur when the collecting vessel level is not properly controlled or too much feedwater is introduced into the evaporator. This can cause saturated water to enter the superheater, resulting in a rapid drop in the main steam temperature. In extreme cases water and/or cold steam can enter the turbine and cause quenching. According to ASME TDP-1-2006 [24], experience has shown that once-through units hold a greater potential for water induction through the motive steam system during start-up and shut-downs than drum type boilers, because of their start-up systems. In a case study conducted by EPRI [25] of 55 quenching incidents, over-flowing of the collecting vessel is listed as one of the probable causes of the turbine quenching incidents. These incidents can also happen without the turbine quenching, as modern control systems have differential main steam temperature protection. A typical example of the protection is that the turbine would trip (closing of stop and control valves) if the main steam temperature drops by 30°C in less than one minute. This would protect the turbine from relatively colder steam, or so called 'quenching'. However, it does not eliminate or protect the thermal fatigue that the upstream superheater, evaporator and economiser systems are subjected to. It is however important to highlight that there is a difference between thermal quenching and thermal fatigue. Thermal quenching is the extreme case of thermal fatigue, characterised by a very large temperature differential that leads to ductile or brittle fracturing. Thermal fatigue on the other hand, results from repeated/cyclic thermal gradients causing localised yielding below the fracture strength of the material [26].

2.6 Control philosophies

Most, if not all, control systems are contemplated, designed, built and implemented according to a certain control philosophy. The control philosophy is usually a summarized description of the physics and fundamentals of a process/system and the intended operation of the system/process according to the physical- and within fundamental limits (conservation equations, component characteristics, fluid properties, material limits etc.) of the system. It goes without saying that any control concepts and objectives should be informed by a well understood fundamental philosophy.

A deeper understanding of the process behaviour and its translation into reliable process models will most certainly lead to significant improvements of control performance.

It is necessary to gain an understanding of the modern feedwater control philosophies and strategies and to compare the differences between drum- and once-through type boilers.

- Drum type boiler level control

Controlling the drum level is considered as one of the main control objectives in drum type boilers. Principally, the level of the drum is controlled by adjusting the feedwater flow into the drum. If the level is above the setpoint, feedwater flow is reduced. If the level falls below the setpoint, feedwater flow is increased.

Figure 15 displays a schematic of a typical drum type boiler layout. The feedwater is preheated in the economiser and is then admitted into the steam drum. Sub-cooled liquid at the bottom inside the drum is extracted via the downcomer and circulated through the evaporator inside the furnace, by either natural or forced circulation, and back into the steam drum where saturated vapour is then routed to the superheater section.

The fundamental processes that influence the level inside the drum can be described as follows: As more heat is added from the furnace to the evaporator, the fluid inside the circulation loop will heat up more (in the case of sub-cooled liquid) or produce more vapour (in the case of two phase water-steam mixture). As the quality of the mixture at the outlet of the evaporator increases with increased heat transfer, more vapour and less liquid will be fed back to the steam drum. This causes the liquid level to drop and more feedwater (sub-cooled liquid) must be added to make up for the imbalance and to restore the level to its set-point value. In order to maintain the pressure in the drum, more saturated vapour will now also have to be extracted to the superheater section. Therefore, as more heat is added to the evaporator, more steam is extracted to the superheater section and ultimately fed to the turbines.

The level in the drum must be maintained to prevent water carry-over (high level) or dry-out and overheating of evaporator tubes (lower level). Although the level control principles described above seem relatively simple, the dynamics of the process during transients render a simple single element level control to be insufficient. Drum level dynamics and control has therefore been the subject of many studies found in literature, including those described in the following references: [27][28][29][30][31][32][33][34][35][36].

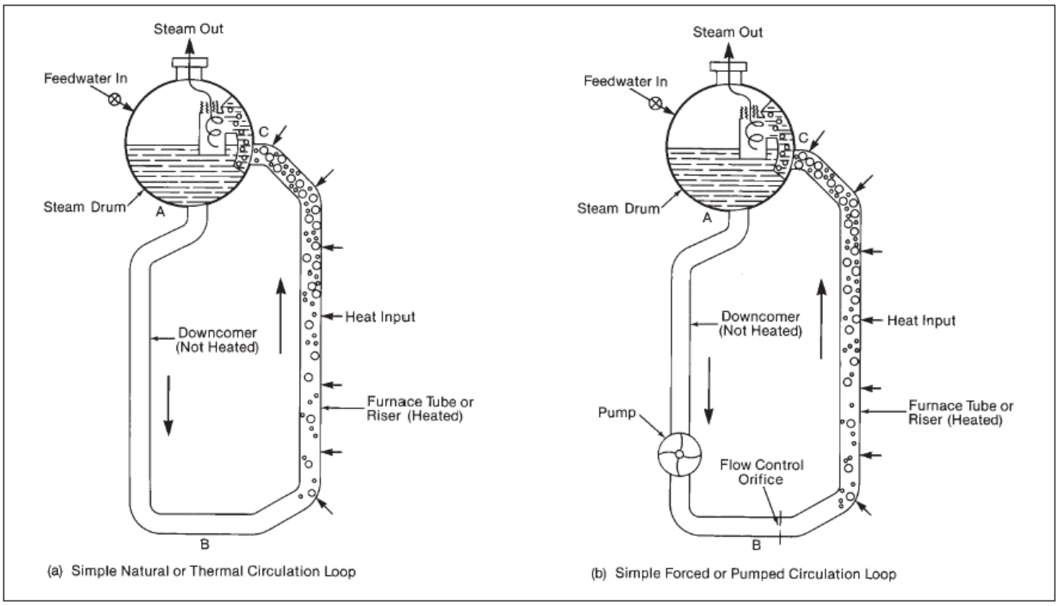


Figure 15: Drum type boilers, (a) Natural Circulation, (b) Forced circulation, [37]

A common occurrence/anomaly often cited is the so-called “false water level” phenomenon, which is also referred to as the “shrink-and-swell” effect. According to Kwatney & Berg [27] and Bell & Astrom [28], the shrink-and-swell effect is the cause of the nonminimum-phase behaviour of level dynamics. The phenomena is generally explained as follows: When the drum is operating in steady state a sudden demand for steam flow (load increase) will cause the main steam pressure (and thus the drum pressure) to decrease. The decrease in drum pressure results in the enlargement (“swelling”) of steam bubbles below the drum level. The swelling of steam bubbles then causes the level in the drum to increase. The controls will therefore call for the feedwater flow rate to be reduced. However, since more steam is produced, more feedwater is actually required to maintain the level. With the swelling effect and appearance of higher level, more feedwater is required, not less. The opposite occurs when the pressure of the drum is increased (load decrease, reduction of steam flow demand). The steam bubbles are compressed/”shrinked” and causes the level to decrease. The false water level phenomenon can thus be described as a fast-transient process whereby the level in a two-phase tank changes due to a change in pressure, not due to a change in mass flow balance. It is the change in pressure that changes the density and saturation temperature of the working fluid, which also changes the specific heat capacity of water and volume void fraction in which expansion or contraction leads to a temporary change in the vessel level. With the associated change in mass flow balance due to the pressure change, the level change is much more affected by the change in pressure, and opposite to the change in mass balance.

Maffezzoni, 1997 [34] reported from his fundamental analysis that the drum level is subject to three different kinds of variations, first (of integral type) is due to a mass flow imbalance between the feedwater (inlet) and main steam (outlet) mass flows. Second (of proportional type) is due to the dependence of mean fluid density on the evaporator pressure. Thirdly, the void fraction (volume of steam to total evaporator volume) that might be very quick, as any changes in the void fraction immediately changes the level. Figure 16 displays the results from Åström and Bell, 2000 [28], where their model and plant data correlate very well and the shrink-and swell can clearly be seen.

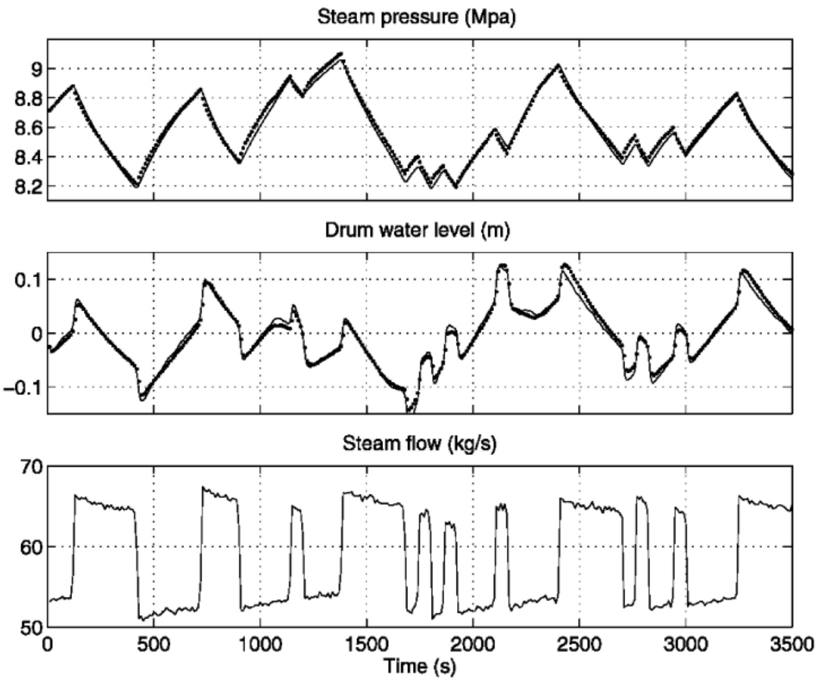


Figure 16: Comparison of model (solid line) and plant data (dots) for perturbations in steam flow rate at medium load, [28]

The single element controller is thus inadequate to control the transients. The modern control strategies widely applied in industry consist of a cascaded three element controller. The reason the control strategy is called a three-element controller is due to the fact that three different measurements are used in the control strategy: the drum level, feedwater flow and main steam mass flow. The three-element controller consists of two PID controllers, the drum level with PID and the feedwater controller as PI, the main steam mass flow is used as a feedforward signal added to the output of the drum level controller. However, in practice the system usually reverts to one element control (drum level) at loads lower than around 30% - which is attributed to inaccuracy of steam flow measurements at these low loads. However control studies have shown that the steam feedback to the controller at low loads has a destabilizing effect, with the inaccuracy of the measurement only contributing to instability, [27]. Essentially what the three-element controller does is maintain the feedwater mass flow rate equal to the main steam mass flow rate, which should keep the level constant, whilst the drum level controller provides minor adjustments.

- Once-through boiler feedwater control

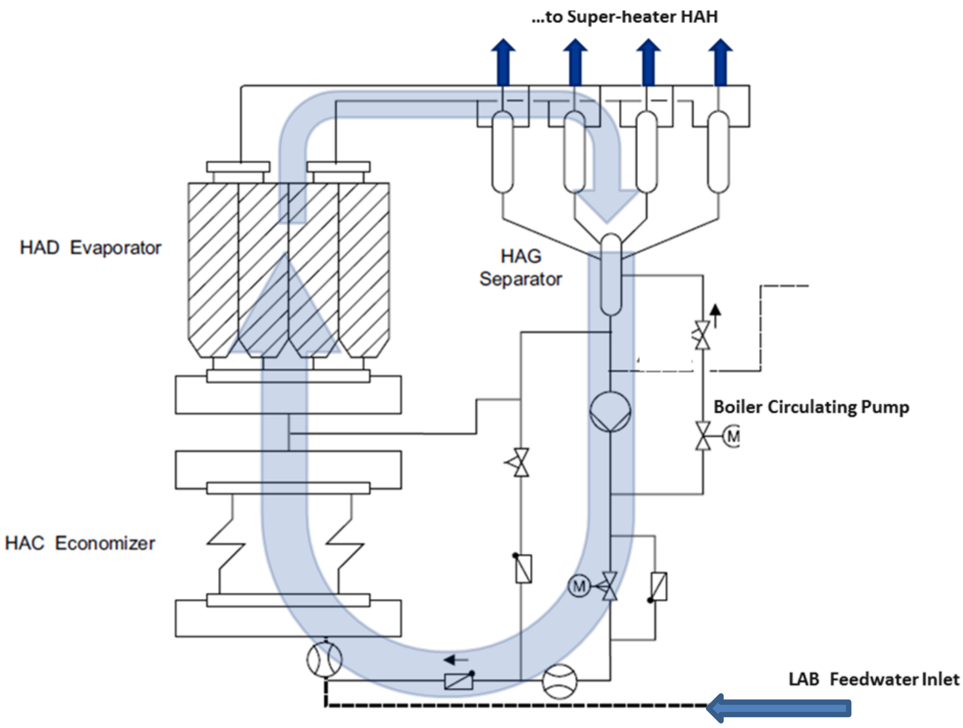


Figure 17: Once-through boiler start-up system, [1].

Figure 17 displays a process flow diagram of a typical start-up system of a once-through boiler. Feedwater control is achieved by a two-step process (set-point change). Up to the Benson point, there is not enough heat to turn all of the subcooled feedwater into steam and a certain minimum flow has to be ensured through the spirally wound evaporator to prevent overheating from boiling crisis and maldistribution of flow. The two-phase mixture leaving the evaporator is routed to separators where the liquid phase/water is routed to the collecting vessel and the saturated steam to the superheaters. To prevent the discharge of water, the water in the collecting vessel is recirculated through the economiser and evaporator. As more heat is added to the furnace, the quality of the two-phase mixture increases, and less water is recirculated as more steam is produced.

At the Benson point, there is enough heat in the furnace to evaporate all of the feedwater and the evaporator outlet is fully saturated steam. The circulation pump is then switched off, and the feedwater flow set-point is no longer constant but is now a linear function of the boiler fuel load, with a feedforward enthalpy controller. The feedwater flow set-point and corresponding feedwater inlet/pump flow and circulation flow as a function of boiler load can be seen in Figure 18.

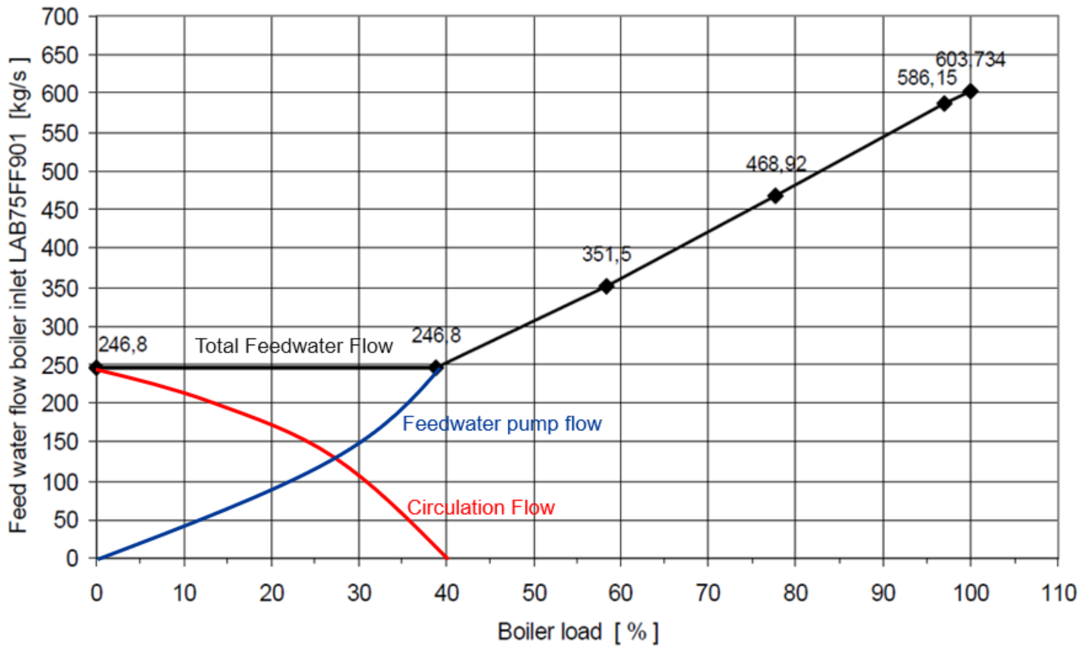


Figure 18: Typical Feedwater flow as a function of boiler load, in Red-Circulation flow, Blue-Feed water supply/pump flow, Black- Total Feedwater flow, [1]

As discussed previously, during start-up the total feedwater flow is controlled on a constant set-point by adding or removing feed pump flow with a PID controller. The circulation flow on the other hand is controlled as a function of the level in the vessel. Basically, the collector vessel level is controlled with a PID by changing the circulation flow rate. Thus, the circulation flow has an indirect impact on the feedwater supply flow. The vessel level also has discharge drain valves to atmosphere and will open if the vessel level is too high. As explained before and as noted by other authors such as Eitelberg and Boje, 2003 [38], the start-up system controls can be problematic in practical applications. It is here where many variations in control strategies and methods are found in practise. Many different and sometimes complex feedforwards are applied to the feedwater supply to prevent units from tripping during start-ups and low load scenarios, without cognisance of the possible resulting component damage, as described in section 2.5.

2.7 Control System

Modern power plants require advanced and robust control systems to enable safe and efficient operation. Advanced control systems consist of many individual components all linked together to form a complex system such as can be found in a DCS (Distributed Control System). A DCS typically consists, amongst others, of controllers such as the conventional PID (Proportional Integral Derivative) and variations thereof, control elements (valves, actuators), parameter monitoring, feedback loops, fuzzy logic, Human Machine Interfaces (HMI), PLC's (Programmable Logic Controller's), alarms etc. The purpose of a control system is to maintain the process conditions close to their set-points, as informed by the control strategy and logics. The control system requires tuning, optimization and performance testing to establish its adequacy to control the process. This would normally consist of making adjustments to the control parameters to compare or match the control system's characteristics to the process being controlled, [32]. The control system should be able to safely control transients during start-up, shut-down and normal operation. An example of a transient experienced on a real plant is provided in Figure 19, where the unit tripped. The level in the collecting vessel suddenly drops, the recirculation flow is decreased significantly and fast, which causes a sudden reduction in the economiser inlet flow. The feedwater inlet flow increases to maintain the minimum economiser inlet flow set point but overshoots the set point. The overshoot causes a large error, and the feedwater flow is decreased to such an extent that the unit is tripped due to low economiser flow.

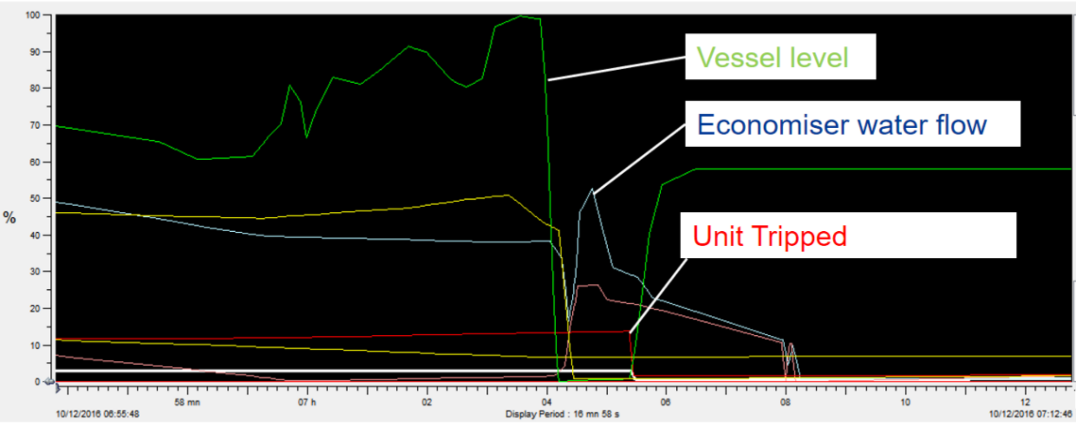


Figure 19: Typical Vessel Level collapse and unit trip

This situation is not ideal, not only because the unit tripped, but also due to the thermal fatigue experienced by the components during the transient, as explained in the previous section. Care must be taken to determine the root cause of such incidents or trips as these could be related to a problem in a mechanical process, control elements, control system parameters or control strategy. This highlights the need for accurate process modelling tools. This study will focus on basic control strategies, elements and analysis. As shown by [8], it is not necessary to model the complete control system, however the control schemes and logics of the plant must be taken into account to be able to compare the simulated results to measured data. As such, this study is focussed on the part of the control system responsible for the waterside start-up control of a once-through boiler, as described previously in Section 1.4.

Maffezzoni, 1997 [34] presented the principal dynamic phenomena which determines the structuring of the boiler-turbine control systems. He describes how to capture basic boiler dynamics by simple first principle models and specifies the limits of their use for control design. In his concluding remarks regarding drum boilers, he notes that the boiler dynamics exhibit drastic changes at low loads and as such the controls must be structurally changed due to the steam flow feedback which adopts a destabilizing effect. He further comments that once-through boilers require a completely different representation of the evaporator due to the enhanced interaction between the evaporator and superheater control. He refers to a simple once-through model by Maffezzoni (1989) and thorough dynamic analysis of Dolezal and Varcop (1970). However, these references could unfortunately not be found with existing resources.

Eitelberg and Boje, 2003 [38] analysed the multivariable water flow and collecting vessel-level control problems of a Benson type boiler during start-up by using a loop-to-loop approach. They investigated the difficulties of controlling the collector vessel level and circulating water flow fluctuations during start-up. They created a basic model, ignoring the drain valves (used in case the vessel level is too high), separator and cold injection water flow to the circulating pump, which are valid assumptions for the purposes of their investigation. They describe the start-up control system as follows: the feedwater flow is indirectly manipulated by the collecting vessel level whilst the variable component of the flow through the evaporator, the circulated water, is directly manipulated by the collecting vessel level. The circulating flow appears as a disturbance in the feedwater control loop and therefore is controlled/regulated by a corresponding variation of the feedwater flow rate, which they note as being significantly imperfect in practice. Unfortunately, the feedwater control loop is not discussed in terms of pumping equipment (fixed speed or variable speed pump) and the feedwater regulation valve control loop. In their analysis of real plant data during a cold start, they note large and erratic flow and level fluctuations which are strongly influenced by the collecting vessel drain valves during the first stages of the start-up below 6 [MPa]. They also note the serious impact of the difficult-to-control thermal processes related to the mixing of colder feedwater with hot re-circulated water on the flow transients. Their paper advocates the use of a loop-by-loop approach to multivariable feedback control design. By using this approach they conclude that to have stability (within realistic margins), the circulation control loop bandwidth must be less than the feedwater control loop bandwidth. For clarity, the control loop bandwidth is a measure of how fast the loop responds to a change in the input value. The Bandwidth can also be explained as a range of frequencies in which the controller exhibits satisfactory performance. Generally, higher bandwidth systems provide better performance but are prone to be disturbed by noise. In their case study they found that the circulation loop had been tuned to a slightly higher bandwidth than the feedwater loop. They also point out that the control loop bandwidth (and performance in general) is not fundamentally limited by the controller design, but by the non-minimum phase lag of the plant and instrumentation. As such, they claim that the entire circulation control system performance hinges on the effective dead-time in the feedwater pumping equipment.

Unfortunately, Eitelberg and Boje did not analyse any warm or hot starts, where the influence of the drain valves and increased thermal excursions become even more prevalent. In the author's experience, increased opening frequency of the drain valves and increased colder feedwater during warm and hot starts, lead to a reduction in boiler pressure, further amplifying instability of the start-up system. Also, the impact of significantly increasing the feedwater loop bandwidth is unknown. Considering that the once-through boiler start-up system is more likely to cause turbine quenching [24], and that thermal fatigue of the start-up system is largely caused by the introduction of colder feedwater by the feedwater loop [20], increasing the feedwater loop bandwidth might negatively affect these aspects. However, more stability in the start-up system characteristics might positively affect the damage aspects.

The loop-by-loop approach does offer an understanding of the control performance and the instabilities in the start-up system of the once-through boiler. Unfortunately, changing the feedwater loop bandwidth to be higher than the circulation loop bandwidth was not modelled or validated. This further amplifies the need for accurate dynamic process and control models. However, if the feedwater control bandwidth has been maximised from the controllers, but the bandwidth is still not significantly higher than the circulation loop bandwidth, the feedwater regulation valve sizing becomes a very important parameter in ensuring stability of the start-up system.

As the collecting vessel level has a direct and definite impact on the control of the feedwater system, the instrumentation used to measure the level must also be investigated in terms of providing reliable measurement of the actual level. The collecting vessel in once-through boilers is usually very small in volume (in the range of 4-8 [m^3]) and does not offer large storage capacity compared to the volume of the economiser and evaporator (in the range of 100-150 [m^3]). The most common method found in industry to measure the level of drums, tanks or vessels containing a two-phase mixture of water and steam is by differential pressure transmitters. These operate by measuring the pressure difference between a reference leg and the pressure inside the vessel, from which the level is calculated based on the fluid properties. Figure 20, from [31], provides a visual explanation of the level measurement principle typically found on steam drum boilers, but is also used in collecting vessel level measurement.

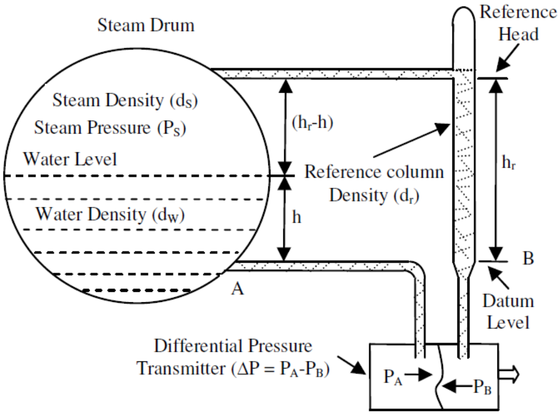


Figure 20: Pressure vessel level measurement principle, [31]

Although accepted and used extensively within the industry, rapidly changing pressures inside the vessel may compromise the accuracy of the measurements. Chakraborty et al, 2014 [31] provide the following practical design conditions to improve the accuracy of the level measurement.

- The reference leg temperature must be as close as possible to the vessel temperature. In the author's opinion, the reference leg temperature should ideally be measured.
- The pressure in the vessel should not deviate significantly from the working pressure from which the water and steam properties are taken. In the author's opinion – the water and steam properties should be calculated dynamically (on-line) from the measured pressure.
- The reference leg head must remain constant and stable under all operating conditions.
- The impulse lines from the plant tappings to the differential pressure transmitter should be at the same level as the plant connections, geometrically similar and the temperature maintained as close as possible to the temperature in the vessel.

Thus, the accuracy of the collecting vessel level measurement forms an integral part of the control system, as this level is directly controlled by the circulating flow.

2.8 Two Phase Flow Instabilities

For the purposes of modelling and accurate prediction of the system behaviour, it is of great importance to fully understand the internal characteristics of heated channel (tube) flow.

Heated water flow in tubes such as in the evaporator, where steam is generated through evaporation and boiling of the water, is susceptible to thermo-hydraulic instability. These instabilities are mainly due to the density difference of water and the steam that is generated, which is the major source of the nonlinearity of the system characteristics.

According to Nayak *et al*, 2008 [39] a system is considered stable if the system returns backs to its original steady state following a disturbance (perturbation). If the system oscillates at a constant amplitude, it is considered neutrally stable. When the oscillations have an increasing amplitude, the system is considered unstable.

Research on two-phase flow is mainly concentrated on modelling the instability mechanisms, with a specific focus on parallel flow in channels and tubes, natural circulation systems- especially in the nuclear industry and more recently research has also been done on direct steam generation in parabolic trough solar power plants [40][41].

Kakac and Bon, 2008 [42] provides a summarized review of the theoretical and experimental works carried out by many investigators, over a period of many years, demonstrating and explaining the three main modes of dynamic two-phase instabilities encountered in various boiling channel systems. Their review included references to 145 published works on the subject. Another comprehensive study on two-phase flow instabilities can be found in the Doctoral thesis of Ruspini, 2013 [43].

Instabilities in two phase systems can be caused by a wide variety of mechanisms and/or parameters. Two phase flow instabilities can be categorized into static and dynamic instabilities, as was first introduced by Bourè et al, 1973 [44]. The definition of the two types of instabilities are as defined by Kakac and Bon, 2008 [42]:

Static Instability occurs when one steady state operating point is changed/jumps to another stable operating point. Further these static instabilities are categorized as Ledinegg instability, Boiling Crisis and flow pattern instability. Ledinegg instability involves a sudden change in mass flow to a lower value when the slope of the pressure-drop versus flow curve is negative and steeper than the supply characteristic. In this case, it is widely accepted that operating outside of the negative slope region, or introduction of a throttle valve to modify the system internal pressure drop characteristic, stabilises a system [42] [43]. Prediction of the instability boundary is thus carried out from steady state calculations.

Dynamic Instabilities occur due to a wide variety of mechanisms, but a flow is said to be subject to a dynamic instability when there is sufficient interaction and delayed feedback between the inertia of flow and compressibility of the two-phase mixture. However, the instability can also occur due to multiple feedbacks between pressure drop, flow rate and the change in density due to vapour formation. The main types of dynamic instabilities considered applicable to this study are pressure drop oscillations (PDO) and density wave oscillations (DWO).

Pressure drop oscillations can only occur if there is a compressible volume downstream of the heated section and a negative pressure drop versus flow rate characteristic. The mechanism is described as a dynamic interaction between a heated section and the compressible volume and is characterised by low frequency oscillations [42].

Figure 21: Mechanism of Pressure Drop Oscillation, (a) System capable of sustaining oscillations, (b) Characteristic curves, *[43]*Figure 21 is directly taken from [43] and provides a visual representation of the pressure drop oscillation mechanism. The fully developed oscillation is composed by a compression in the surge tank, CD. Flow excursion from two-phase to liquid, DA. Decompression in surge tank, AB. Another flow excursion from a low quality- to a high quality two-phase state. It should also be mentioned that this form of instability can also be associated with other types of instabilities.

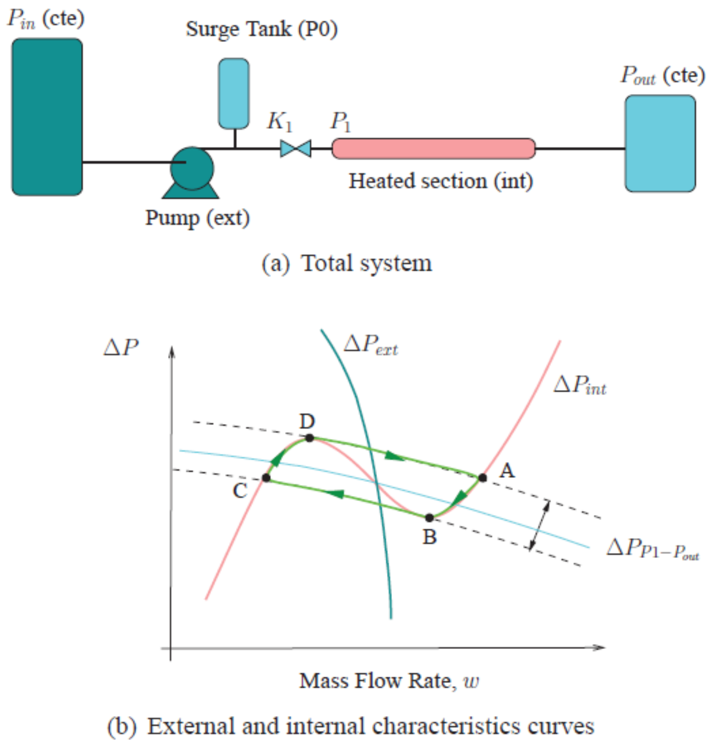


Figure 21: Mechanism of Pressure Drop Oscillation, (a) System capable of sustaining oscillations, (b) Characteristic curves, [43].

Guo *et al*, 2001[45]conducted experiments to obtain the critical conditions for the occurrence of pressure drop type dynamic oscillations in a closed circulation helically coiled tube steam generator. From the experimental results, they concluded and confirmed that the pressure drop oscillation only occurs with the presence of a compressible gas volume in the circulation loop, thus demonstrating the importance a compressible volume has on the occurrence of pressure drop oscillations. In their experiments, they also did not observe any oscillations above 3 [MPa] and that the mass flow rate disturbances resulting from pressure disturbances, are higher at lower pressure conditions.

Density wave oscillations can be defined as the delayed and feedback effects in the relationship between flow rate, density and pressure drop. The oscillation is characterised by the frequency of the transit time as a continuity wave [42]. Papini *et al*, 2011 [46] describes that the density wave type oscillation originates from the difference in density between a fluid entering as a subcooled liquid and exiting a heated channel as a two-phase mixture which triggers delays in the transient pressure drops along a tube/channel, which may include self-sustained oscillations. They also mention that a constant pressure drop, such as found in parallel channels, is a boundary condition than can excite the dynamic feedback effects which are at the source of the instability mechanism.

Nayak *et al*, 2008 [39] conducted a review of the characteristics of different instabilities as well as the effects of different operating and geometric parameters. Their research was focussed on the flow instabilities that occur in boiling natural circulation systems. An interesting conclusion was reached in their review that most instabilities observed in forced circulation systems are observable in natural circulation systems. However, natural circulation systems are more unstable due to the intricate relationship between pressure drop and the buoyancy force causing the flow.

According to Papini *et al*, 2011 [46], no universal stability maps exist. Predicting the stability threshold is popularly done by the two-dimensional stability map introduced by Ishii & Zuber, 1970 [47], who introduced the non-dimensional phase change number (N_{pch}) and subcooling number (N_{sub}). The phase change number can simply be described as a specific ratio of heat rate to mass flow where the onset of oscillations occur, which corresponds to a specific degree of subcooling, quantified by the subcooling number [48].

The following example (Figure 22) of a stability map was presented by Strømsvåg, 2011 [49] in which three theoretically determined stability thresholds are displayed. The three thresholds are the equilibrium theory of Ishii and Zuber, 1970 , the non-equilibrium theory of Saha and Zuber, 1978 , and the simplified stability criteria of Ishii, 1971 , based on the thermal equilibrium model.

Figure 23 displays the stability map presented by van Antwerpen *et al*, 2017 [48] in which the various methods used by Papini *et al*, 2011 [46] as well as their work in determining the onset of oscillation boundaries by using Flownex are compared.

Strømsvåg, 2011 [49] summarizes stability effects of operational parameters as presented by various authors, these operational parameters are further summarized below based on their work:

- Inlet Velocity Increase: has a stabilising effect regardless of subcooling or channel power input. Increased velocity leads to increased mass flux and a decrease of N_{pch}.
- Power Input Increase: Destabilizes the system as N_{pch} will increase into the unstable zone.
- Subcooling: When operating at intermediate or high subcooling, increased subcooling would stabilize the system. In contrast, when operating with low subcooling, increasing the subcooling would destabilize.
- System Pressure: Increasing the system pressure has a stabilizing effect due a reduction in void fraction and therefore the two-phase friction and momentum pressure drop.

- Restrictions: increased restriction at the inlet has a stabilising effect as the single phase pressure drop is increased. Increased restriction at the exit has a destabilizing effect as the two-phase pressure drop is increased, which also increases the time delayed pressure drop.

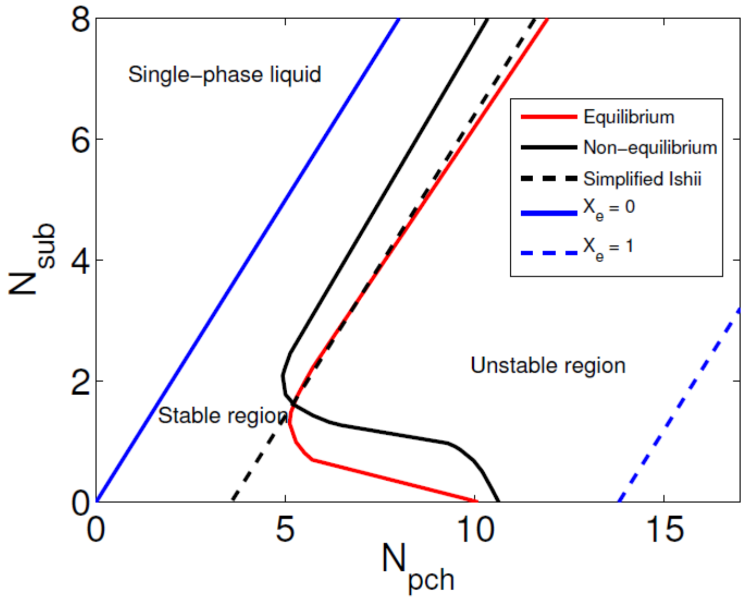


Figure 22: Stability map as presented by Strømsvåg, 2011 [49].

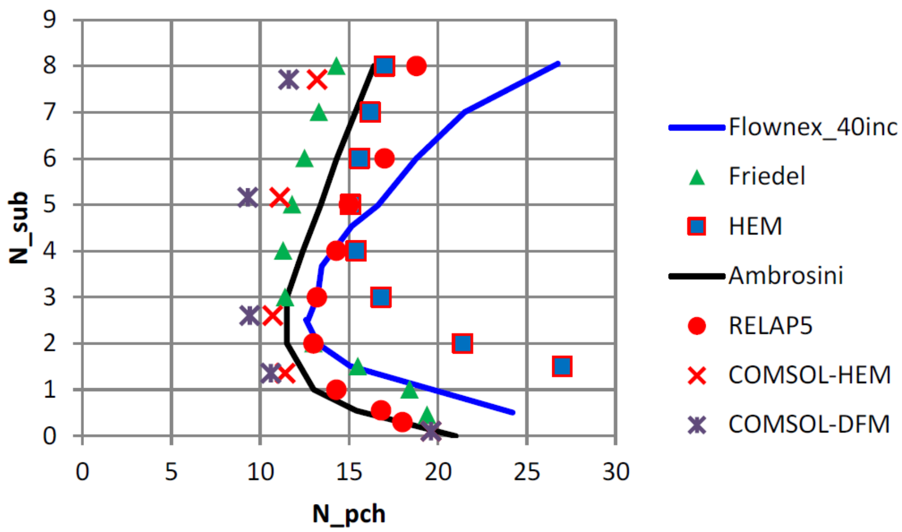


Figure 23: Onset of oscillation boundaries by various methods, [48].

3. Methodology

To study and analyse the boiler start-up system, the software package Flownex Simulation Environment is utilised to create a process model with control elements added later.

Geometric and component specific characteristics are chosen from a reference once-though boiler unit at a certain power station, which will be referred to as Plant X. As most of the geometry is available, an effort was made to accurately incorporate the actual plant geometry. Actual plant data from Plant X was used to validate the model and to compare specific transient cases with the simulated results.

The model was built in several stages/steps. Firstly, the evaporator and economiser sections were built, analysed and studied with regards to discretisation, numerical stability, two-phase instabilities/ stability boundaries, heat transfer and dynamics of the two-phase flow. Secondly, the circulation loop was modelled including the collecting vessel and boiler recirculation. The completed economiser-evaporator model was then added to the circulation loop to create the complete process model. The control elements were then added to form an integrated process and control model. The completed model was then used to simulate plant transients.

3.1 Evaporator - Geometry

The evaporator is modelled as a single tube (represented as a pipe component) with equally distributed heat transfer, however with the option that the heat transfer can also be modified to incorporate inequal heat transfer. Figure 24 displays a single evaporator tube in a typical spirally wound evaporator configuration in a tower type once-through boiler.

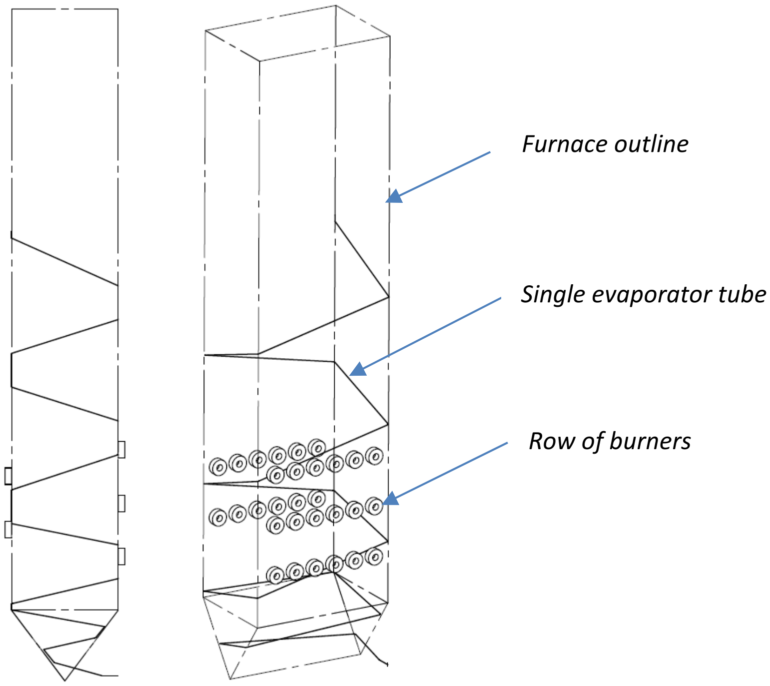


Figure 24: Typical view of a single evaporator tube in a once-through tower type boiler

The evaporator is split into three different sections with three distinct diameters, lengths and number of tubes. The geometry and design of the boiler can be very complex, Figure 25 displays the average length of single tubes and the amount of tubes through each distinct section of the boiler, from the economiser to the last superheating element, superheater 3 (SH3). The reheat section is not shown.

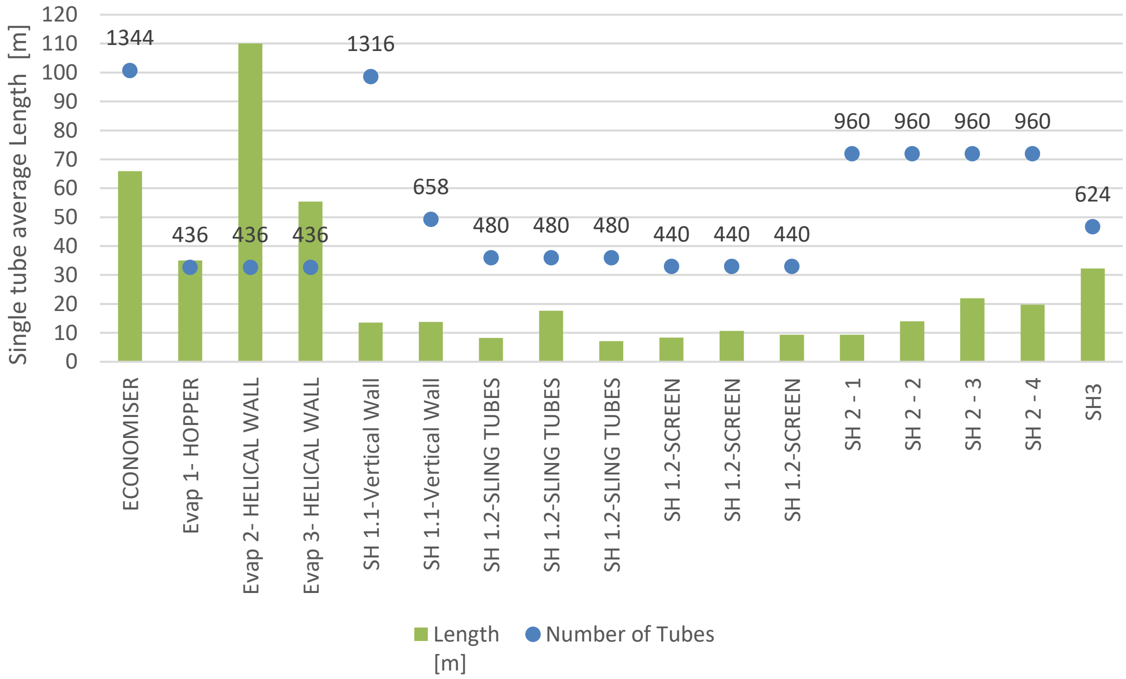


Figure 25: Once-through boiler geometric data from the economiser inlet to the last superheating element, [1].

3.2 Evaporator theory

As reported by [43], the stability of any two-phase flow system is highly dependent on the pressure drop losses across the system. As such, it is of utmost importance to correctly predict the pressure losses. However, predicting two-phase flow pressure drops is a whole study on its own as there are many correlations that describe the pressure drop and heat transfer characteristics of two-phase systems. Two-phase flow is governed by mass, momentum and energy conservation equations with several constitutive equations and correlations used to calculate the two-phase friction factors and heat transfer coefficients. Liquid and gases are arranged in certain flow regimes/patterns which significantly effect the pressure drop and heat transfer characteristics, [50]. The flow patterns and heat transfer regions typically found in forced convection vertical evaporator tubes is shown in Figure 26. A sub-cooled liquid (water) enters the tube at the bottom where convective heat transfer takes place to the liquid. Sub-cooled boiling occurs as the wall temperature of the tube exceeds the saturation temperature of the liquid. When the fluid reaches saturation temperature, saturated nucleate boiling occurs with an approximately constant heat transfer coefficient. With increasing steam quality, the flow reaches the annular flow zone and convective flow boiling occurs. In this zone the heat transfer coefficient is increased as the heat from the tube wall is increasingly being transferred by convection to the film of water. As the film of liquid on the tube wall is vaporized the heat transfer is decreased significantly, such that the tube wall temperature increases, this is called the dry-out and post dry-out regions and the phenomenon is commonly referred to as boiling crisis, [33].

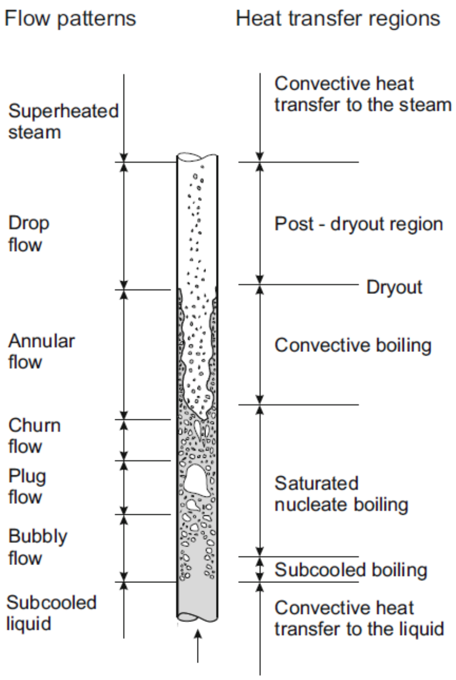


Figure 26: Flow patterns and associated heat transfer regions in a vertical heated tube, [32]

As the current model is developed in Flownex, a summary of the approach used in Flownex is described below. For more details on the calculations and methods, readers are referred to the Flownex Theory Manual, [50].

Flownex uses a one-dimensional homogeneous mixture model for two-phase flow. In this approach the liquid and gas phases are assumed to be distributed evenly over the cross-sectional area of the flow path and that the pressure, temperature and velocity of the different phases are equal. As there is no distinction in the flow patterns and regions (as depicted in Figure 26), single phase constitutive equations and correlations are implemented for the mixture. Thermo-physical properties are calculated using subroutines from the National Institute for Standards and Technology (NIST/ASME steam properties database). The effects of the two-phase flow are incorporated into the single phase pressure drop by an empirically correlated two-phase pressure drop multiplier. These multipliers are used for the frictional as well as loss components in the momentum conservation equation. The frictional component is calculated using the Lockhart-Martinelli or Beggs and Brill method. Critical flow is based on the homogeneous equilibrium model.

For the heat transfer correlations, the Dittus-Boelter correlation is used to calculate the forced convective heat transfer coefficients for both subcooled boiling and saturated boiling. However, other correlations are used to determine the critical heat flux, where the look-up table from Groeneveld is used with scaling techniques for larger diameter tubes. Linear interpolation is used in the transition boiling region between critical heat flux and minimum heat flux. The 2003 film boiling look-up table of Groeneveld is used to determine the film boiling heat transfer coefficient as a function of pressure, mass flux, flow quality, and wall superheating.

Heat transfer from the combustion chamber to the tubes is significantly simplified, as the flue gas subjects the evaporator to various heat transfer mechanisms such as convection, conduction and radiation. The evaporator section is largely subject to radiation heat transfer, whilst the economiser is predominantly subjected to convection heat transfer. Simple heating elements (distributed heat source element in Flownex) are used for the individual sections, with the option of unequally/non-uniformly distributing the heat transfer to the tubes.

3.3 Evaporator characteristics

To be able to accurately predict the system characteristics, it is necessary to discretise the piping elements into a finite number of elements. However, the choice of the discrete number of elements needs to be taken carefully as a balance needs to be found between simulation time and accuracy of the model. The objective was to achieve high accuracy of the prediction of the evaporator characteristics where high accuracy of the two-phase zone pressure drop is required, and less elements for zones of single-phase conditions, whilst maintaining reasonable simulation times. The methodology followed to make this decision was to use the pressure drop vs. flow rate characteristic curve as a basis to determine the number of elements required, whilst assuming that 1000 increments would provide the most accurate calculation. The Sensitivity Analysis function in Flownex was used to compare the effect of the number of increments on the pressure drop vs. flow rate curve. In this specific investigative case, an averaged diameter 200 [m] vertically inclined tube was used, with subcooled inlet condition, heat added to the tube to achieve a slightly superheated outlet condition at the outlet/exit at a mass flow corresponding to the once-through point. Figure 27 displays the results obtained from the different number of discretisations. Naturally, a single increment provides the fastest results, but the results are not accurate when compared to the 1000 increment base results. As the number of increments increases, the pressure drop values converge towards the 1000 increment results. The flow area of interest for this study is indicated with vertical lines (which correspond the minimum/nominal eco flow). The data from the results were analysed by comparing the total pressure drop (P) and exit quality(x) results to the base results when using 1000 increments. A summary of the statistical analysis is provided in Table 2.

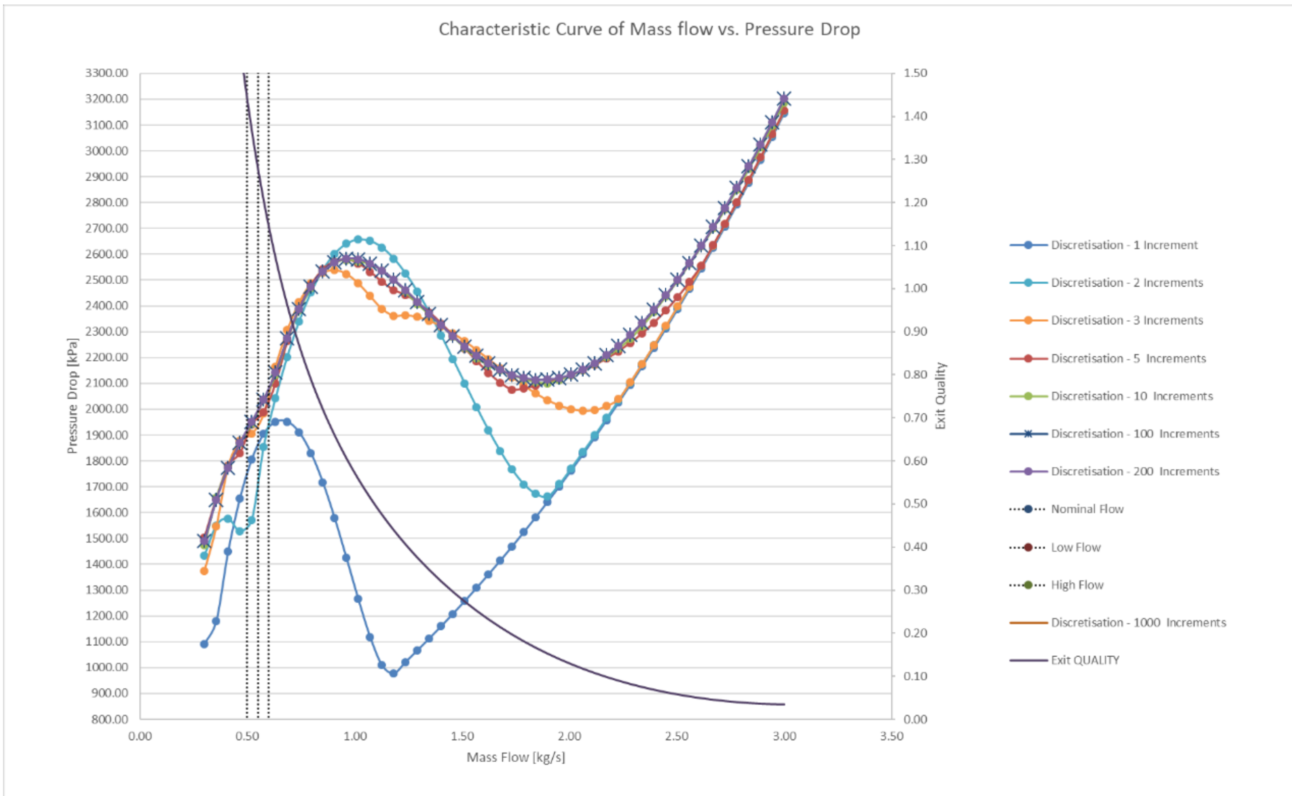


Figure 27: Mass flow versus pressure drop characteristic curves for different numbers of discretisation

Normalization of Data								
Number of Incriments	**1**	**2**	**3**	**5**	**10**	**50**	**100**	**200**
Standard Deviation (P)	44.774%	8.842%	3.231%	1.066%	0.234%	**0.010%**	0.003%	0.001%
Standard Deviation (x)	4.658%	4.687%	3.815%	2.772%	0.514%	**0.023%**	0.003%	0.000%
Max Error	-155.70%	-27.20%	-10.10%	-3.04%	-1.12%	-0.03%	-0.01%	-0.01%
Min Error	-1.80%	3.40%	1.34%	0.82%	0.37%	0.03%	0.01%	0.00%

Table 2: Data analysis compared to 1000 increment results

From these results, it is possible to determine the number of increments to be used for the model based on a qualitative assessment. By choosing 50 increments, a sufficiently accurate result can be obtained; using statistical analysis the standard deviation on pressure drop (P) is only 0.010% and exit quality 0.023%. However, if the need arises to reduce simulation running times - reducing the increments to 10 would still yield satisfactory results as the standard deviation for pressure drop is 0.234% and exit quality 0.514%. Correctly determining stability boundaries as noted by others [48][43][46], who used 40 to 50 increments to accurately model the phenomena, is also considered.

The effect of the tube roughness on the evaporator pressure drop was also investigated by the same methodology as described for a single tube above. Figure 28 displays the results and as is expected the pressure drop increases with increasing tube roughness. The dashed line indicates the area of interest during a once-through boiler start-up. The displayed values were calculated for an arbitrary boiler load of around 15% BMCR. The tube roughness selected was 80 µm and can be easily changed to suit plant tube conditions.

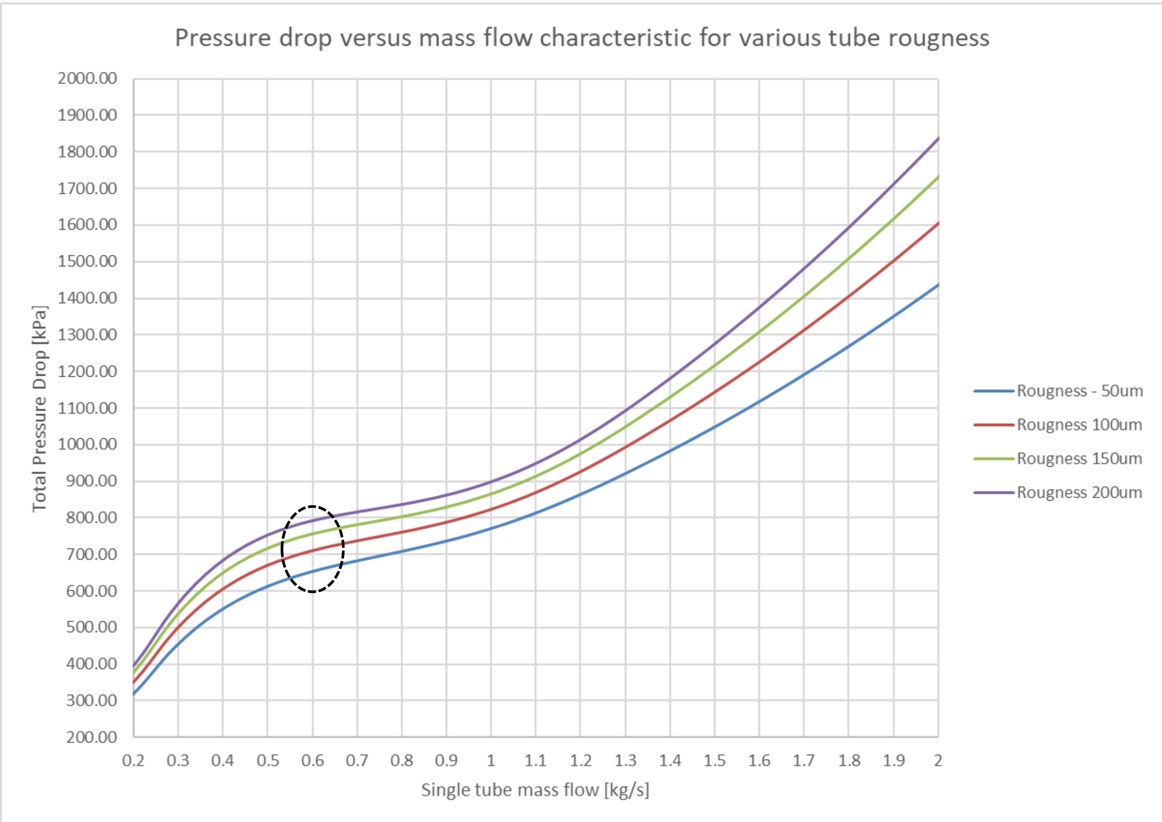


Figure 28: Pressure drop [kPa] and mass flow [kg/s] characteristic for different values of tube roughness

During the start-up of the boiler the pressure and heat transfer increases which also increases the pressure drop. To gain an understanding of the pressure drop characteristics at different loads, realistic fuel inputs and associated changes in inlet feedwater temperature and circulation flow rates are varied to calculate the pressure drop across the economiser and evaporator at steady state loads. The characteristic obtained is displayed in Figure 29.

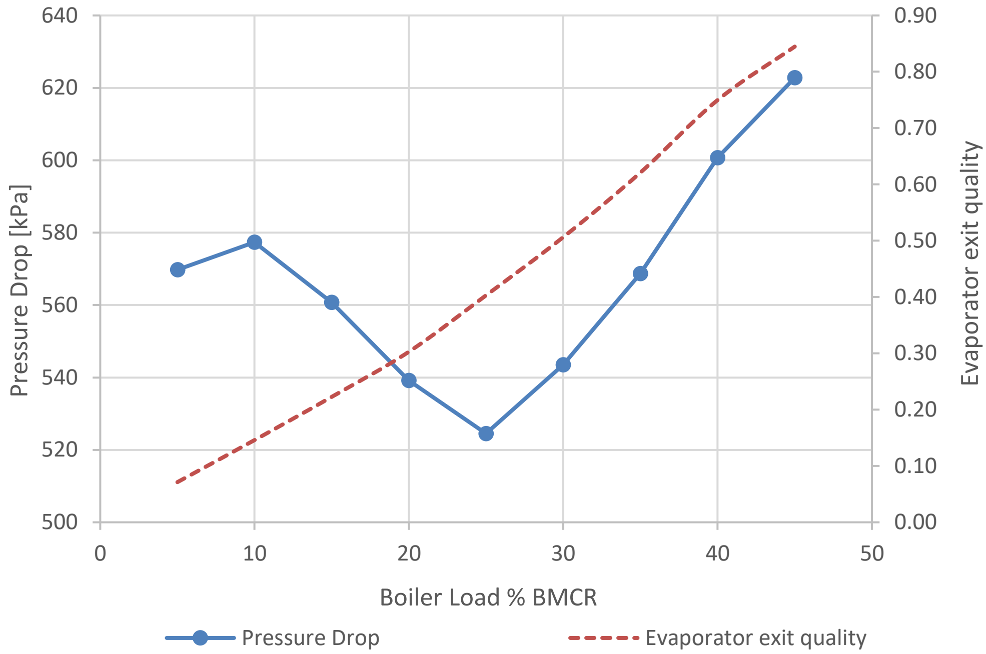


Figure 29: Economiser and Evaporator pressure drop as a function of Boiler Load

The characteristic obtained in Figure 29 shows a decrease in total pressure drop from around 10% BMCR, and increases again after 25% BMCR. This is due to the evaporation process – initially the pressure drop increases but then starts to decrease due to the reduction of the gravity term (as the water head decreases). As the steam production increases, the frictional pressure drop in the two-phase zone increases and starts to increase the total pressure loss from an exit quality of around 0.4. This finding could perhaps prove significant as the flow rate for these pressure drop calculations remains constant and as can be seen in section 3.4 below (equations *(3)* to *(6)*) the conservation of momentum (pressure balance) plays a definite roll on the level of the collecting vessel.

3.4 Fundamental steady-state analysis

Before starting the complete integrated modelling and analysis, a fundamental understanding of the process is required. First, consider only the collecting vessel as a one-dimensional control volume, with a fixed volume, as displayed in Figure 30.

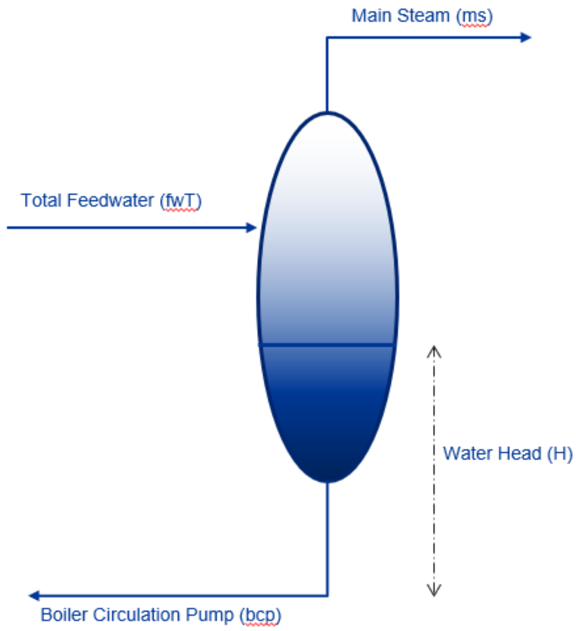


Figure 30: Collecting vessel control volume

From first principals and assuming steady state conditions, whereby nothing changes over time, the mass balance equates to:

$$\dot{m}_{fwt} = \dot{m}_{bcp} + \dot{m}_{ms} \tag{1}$$

For the conservation of energy, no heat source is added to the vessel and there is no work, kinetic and potential energy terms are ignored. Therefore:

$$\dot{m}_{fwt}h_{fwt} = \dot{m}_{bcp}h_{bcp} + \dot{m}_{ms}h_{ms} \tag{2}$$

The total mass flow at the inlet ($\dot{m}_{fwt}$) is known/specified, with all other variables unknown. However, we know the enthalpy at the boiler circulation pump (h_{bcp}) and main steam (h_{ms}) is at the saturated water and steam states respectively, which can be computed from the fluid property steam tables if the pressure is known.

For the Conservation of momentum, the kinetic energy terms are ignored as well as the potential terms of the steam path (ms) and (fwt). Assuming that there is a certain pressure p_{cv} inside the vessel, three equations can be obtained with Δp_{OL} equalling frictional pressure loss:

For the main steam exit section:

$$p_{cv} - p_{ms} = \Delta p_{OL\,ms} \tag{3}$$

For the feedwater total inlet section:

$$p_{cv} - p_{fwt} = -\Delta p_{OL\,fwt} \tag{4}$$

For the boiler circulation pump exit section

$$p_{cv} - p_{bcp} = \Delta p_{OL\,bcp} + \rho g H_{bcp} \tag{5}$$

Re-arranging the momentum conservation equation to solve for the head of the boiler circulation pump:

$$H_{bcp} = \frac{p_{cv} - p_{bcp} - \Delta p_{OL\,bcp}}{\rho g} \tag{6}$$

Writing the equation for head in this way is quite significant, as the head is then directly related to the pressure inside the collecting vessel minus the frictional losses. However, p_{bcp} is also unknown and will vary according to the value of H_{bcp}. The steady state fundamental equations offer valuable information on the system and the level of liquid inside the vessel. The following observations are made:

- The steady state mass and energy balances do not determine the level.
- The level in the vessel is directly related to the pressure inside the collecting vessel and the frictional losses up to the circulation pump.

Now Consider the Larger Control Volume in Figure 31, with some internal elements shown:

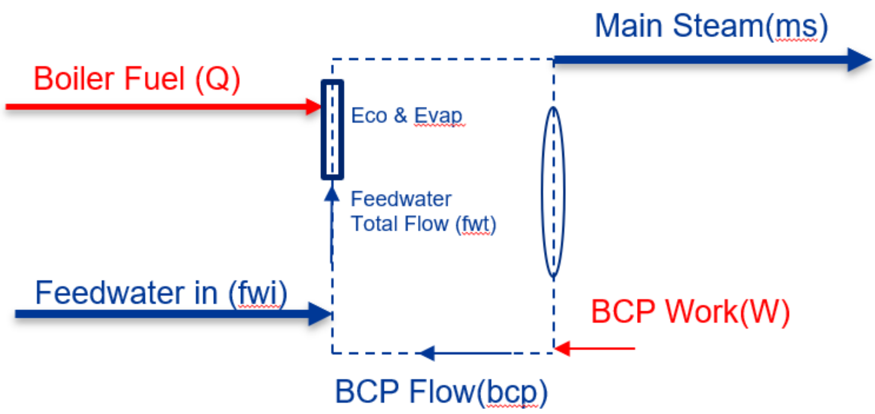


Figure 31: Larger Boiler Control Volume

From first principals and assuming steady state conditions, whereby nothing changes over time, the mass balance equates to:

$$\dot{m}_{fwi} = \dot{m}_{ms} \tag{8}$$

However, considering the mixture of flow inside the control volume, we can also write the mass balance of the total feedwater flow as:

$$\dot{m}_{fwt} = \dot{m}_{fwi} + \dot{m}_{bcp} \tag{9}$$

Re-arranging to:

$$\dot{m}_{bcp} = \dot{m}_{fwt} - \dot{m}_{fwi} \tag{10}$$

and substituting *(8)* results in:

$$\dot{m}_{bcp} = \dot{m}_{fwt} - \dot{m}_{ms} \tag{11}$$

Euaution *(11)* thus provides a fundamental and simple one-dimensional steady state expression for the circulating mass flow.

For the conservation of energy, ignoring the potential and kinetic terms:

$$\dot{Q} - \dot{W} = h_{ms}\dot{m}_{ms} - h_{fwi}\dot{m}_{fwi} \tag{12}$$

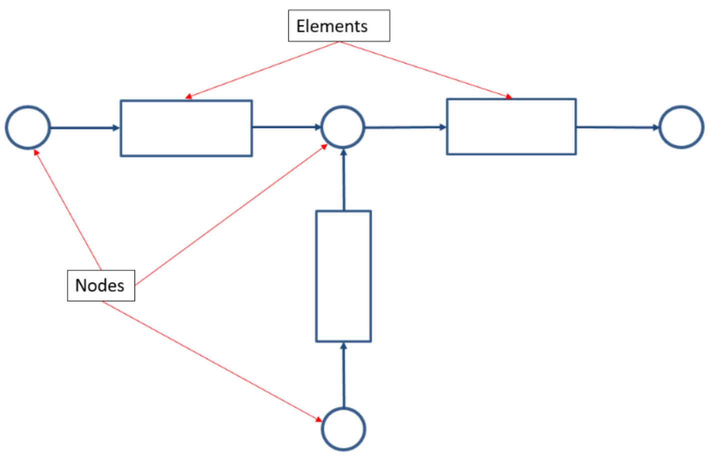
Re-arranging and substituting (8) we can get the following expression for the main steam mass flow:

$$\dot{m}_{ms} = \frac{\dot{Q} - \dot{W}}{h_{ms} - h_{fwi}} \tag{13}$$

From (13) the observation can be made that the main steam mass flow is directly proportional to the amount of boiler energy/fuel input. The main steam enthalpy is fixed at the vapour saturation enthalpy, which means that the feedwater inlet enthalpy is also proportional to the main steam mass flow. In practical terms, for the same pressure and heat transfer, different steady state main steam mass flows can be obtained with different feedwater inlet enthalpies.

3.5 Fundamental transient analysis

Steady state analysis provides a fundamental understanding of the system that is in thermodynamic equilibrium. However, the different steady states do not provide insight on how the system changes over time. To conduct the study of how the main parameters such as pressures, temperatures, mass flows and quality change over time, the transient form of the conservation equations must be solved using a suitable numerical solution method together with the appropriate fluid property relations, modes of heat transfer and component characteristics. As mentioned previously the software package Flownex is used to conduct the transient analysis of the complete system. A short summary of the solution methodology used in Flownex is provided below. Using the analogy as provided in [51], the solution method is described in terms of the network methodology using elements and nodes. Elements represent components such as pipes, turbines, vales, pumps etc. Elements are control volumes which can be sub-divided into increments of finite length. The fluid properties within the element are assumed to be the weighted average between the inlet and outlet of the elements. Different elements are connected with nodes which are assumed to be homogeneous and the fluid properties represented by a single averaged value of all the inlets and outlets.



For the Conservation of mass, the general one dimensional partial differential equation is given by:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho v A)}{A \partial x} = 0 \tag{14}$$

After spatial integration of equation *(14)* the discretised transient form of the mass conservation can be simplified to:

$$V \frac{\partial \rho}{\partial t} = \sum \dot{m}_i - \sum \dot{m}_e \tag{15}$$

Which states that the rate of change of the average density ρ within the element is equal to the difference in the sum of the total mass flow rate into ($\dot{m}_i$) the element and the total mass flow rate out ($\dot{m}_e$) of the element. Subscripts i refers to the inlets and e the outlets. V donates the volume and t is the time.

Applying the same spatial integration, the simplified conservation of energy equation is given by the spatially integrated transient differential equation:

$$\frac{\partial h_0}{\partial t} = \frac{1}{\rho V}\left(\sum \dot{m}_i (h_{0i} + g z_i) - \sum \dot{m}_e (h_{0e} + g z_e) + \dot{Q} - \dot{W} + V\left(\frac{\partial p}{\partial t} - h_0 \frac{\partial \rho}{\partial t}\right)\right) \tag{16}$$

With h_0 the total/stagnation enthalpy, z the elevation, g is the gravitational acceleration constant, p the static pressure. $\dot{Q}$ is the heat transfer rate into the control volume and $\dot{W}$ the work output from the control volume.

The transient conservation of momentum spatially integrated transient differential equation is given by:

$$\frac{\partial \dot{m}}{\partial t} = \frac{A}{L}\left((p_{0i} - p_{0e}) + \rho g (z_i - z_e) + \Delta p_{0W} - \Delta p_{0L} + L v \left(\frac{\partial \rho}{\partial t}\right)\right) \tag{17}$$

In this equation A is the area, L is the length, p_0 is the total pressure, Δp_{0W} the pressure increase or decrease due to work, Δp_{0L} the pressure drop due to friction and v is the weighted average velocity.

Equations *(15), (16) & (17)* represent the fundamental one-dimensional conservation equations that needs to be integrated and solved numerically. Flownex solves the fundamental equations by Euler integration over a discrete time step Δt, with a weighting factor (α) between 0.0 which is fully explicit, and 1.0 which is fully implicit. If α=0.5 the time integration is equivalent to the second order

Crank-Nicholson method. A good compromise between stability and accuracy is achieved with α of 0.6 reported by [52] and 0.7 reported by [17] and [51].

Flownex uses an implicit pressure correction solution algorithm that results in fast and accurate simulations, [50]. The steps of the algorithm are listed below:

(i) Guessing of the initial node pressures
(ii) Calculate the mass flow rates using pressure drop and volume flow relationships
(iii) Continuity is checked at all nodes
(iv) Continuity is maintained at all nodes by adjusting pressures
(v) Mass flow rates are adjusted using updated pressures
(vi) Repeat steps *(i)* to *(v)* until convergence
(vii) The energy equation is solved
(viii) Repeat *(i)* to *(vii)* until convergence is achieved
(ix) Move to the next time step and repeat *(i)* to *(viii)*

Essentially the network methodology reduces to the calculation of new values for the density and total enthalpy in each node based on the old/previous values together with the mass flow rates in and out of the nodes which are dictated by the elements connected to the nodes. The pressure in the node can be calculated with the fluid property relationship $p = f(\rho, h)$, as the density and enthalpy are known. The calculated pressures then dictate the mass flow rates of the elements. A total pressure correction solution matrix is produced from combining the mass and momentum conservation equations after a Newton-type linearization, [51].

3.6 Description of the basic model

Figure 32 displays the model that was created in Flownex for the once-through boiler start-up recirculation system. The recirculation loop is highlighted with blue arrows, the collecting vessel where two-phase separation occurs is indicated with a blue circle. The yellow arrow at the top is the main steam exit (inlet to the superheaters). The green arrow on the left is the feedwater inlet, and the green arrow at the bottom represents the NPSH-injection for the boiler circulation pump. Heat distribution in the furnace is indicated with red arrows.

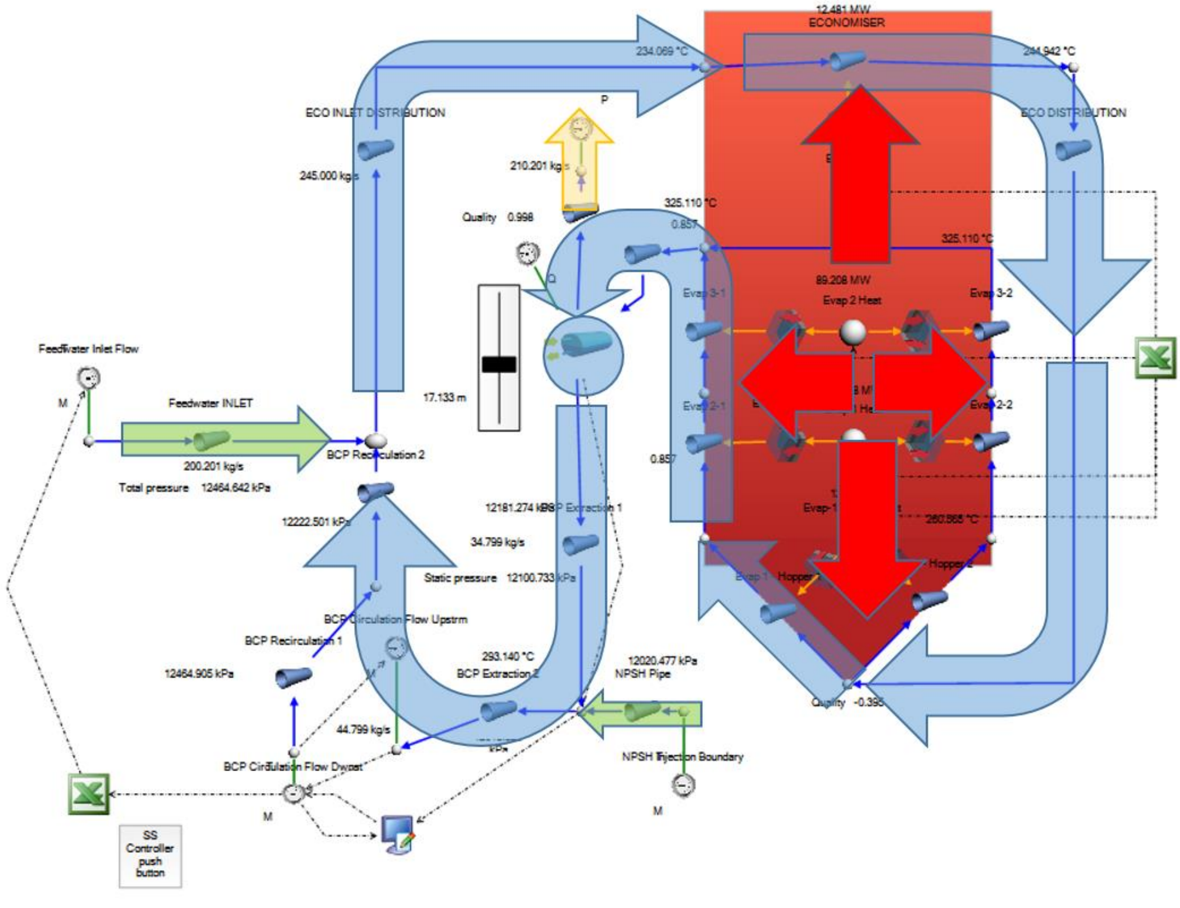


Figure 32: Process model of the once-through boiler start-up recirculation system

The model initially consists of the main components and boundaries listed below, with short descriptions and simplifying assumptions. Further details of the functions and calculations can be found in Appendix A.

- Simplified furnace

The furnace is modelled as a non-uniformly distributed array of heat source elements. Inside the furnace all heat is transferred with conduction through the tube walls, and convection heat transfer to the fluid using the Dittus-Boelter correlation. Radiation is ignored. The heat source is modelled with an Excel component that scales the heat transfer to the distributed heat source elements through a node as fractions of the overall boiler fuel input. The boiler fuel input fractions were derived from using the reference boiler design heat flux values. From reference geometry the tube areas were calculated to derive full load heating values which was then linearly scaled to lower loads. Arbitrary fractions based on the boiler geometry was used to fraction the heat transfer non-uniformly to the tubes. The final heat load can also be adjusted and manipulated in the model by entering a boiler efficiency in the Excel component, which can be applied to increase or decrease the heat transfer to all tube elements.

- Feedwater Inlet (Boundary Condition)

The feedwater inlet represents the outlet of the feedwater pump and associated feedwater control valve and high pressure heaters. Mass flow rate and temperature are the boundary conditions, with the inlet mass flow determined from the recirculation flow rate (equation *(9))* and the feedwater inlet temperature is a function of the boiler load that represents the increase in feedwater inlet temperature as the high pressure heaters enters service.

- Economiser

Modelled as a single tube (pipe element) in parallel with 1433 tubes.

- Evaporator

Modelled as single tubes (pipe element) in parallel with 436 tubes. The evaporator has three different sections with three different tube diameters to allow for larger velocities as steam is produced along the length of the tubes. The first section at the bottom of the boiler is called the hopper, followed by the second and third helical sections with increased inside diameters. The evaporator can also be split into two different sections, as displayed in Figure 32, representing 2 pipe elements each with 218 tubes in parallel to investigate further uneven heat distributions if the need arises.

- Collecting vessel.

The collecting vessel is modelled as a two-phase tank element, which emulates the two-phase mixture collecting in the vessel where phase separation occurs. Saturated steam is bled off from the top of the tank and the saturated liquid phase from the bottom. The two-phase tank is bound with a quality boundary condition to solve steady states. The volume and height of the collecting vessel is also based on the real plant (case study) geometry. Separators are not modelled as the two-phase tank element emulates the separation between the water and steam. The volume occupied by the separators are included in the piping elements connected to the two-phase tank.

- Boiler Circulation Pump (Circulation back to economiser)

Modelled as a boundary condition with a data-transfer link. Additional feedwater is also added to the suction/inlet of the boiler circulating pump to increase the available Net Positive Suction Head (NPSH) for the circulating pump. As the liquid is at saturated conditions, adding colder feedwater at this point in the system cools the fluid to become sub-cooled and reduces pump cavitation risk by increasing NPSH.

- Superheater Inlet

Modelled as a boundary condition including the volume of the separator vessels. In this study, the steam flow entering this point is referred to as the main steam mass flow, whereas the main steam mass flow on the actual plant is measured between superheater 1 and 2 and includes attemperation flow.

- Total Feedwater flow

The total feedwater flow (also referred to in industry as eco flow), is the total mass flow rate upstream of the economiser and consists of the feedwater inlet flow from the feedwater pump and boiler recirculation flow (as per the definition provided in equation *(9)* and in reference to Figure 18). For the purpose of this study, the total feedwater flow value is defined as 245 [kg/s]. The Excel component on the bottom left of Figure 32, enforces this value by changing the feedwater inlet flow boundary condition according to equation *(9)*:

$$\dot{m}_{fi} = 245 - \dot{m}_{bcp}$$

To solve the model in steady state the boiler load, main steam pressure, feedwater inlet temperature and tank quality boundary values are used as inputs. With both the circulation flow and tank quality being boundary conditions, there is an energy imbalance on the two-phase tank (collecting vessel). To solve the energy imbalance (to obtain energy balance = 0) an external solver routine (based on an iterative Newton-Raphson method) is used and executed with the "SS Controller" pushbutton, with SS meaning State State, script obtained from [53]. Essentially the script for "SS Controller" adjusts the circulation flow until the energy balance is achieved. The feedwater flow is automatically adjusted with an Excel component to keep the economiser and evaporator flow constant (at the specified minimum required flow of 245 [kg/s]). In all cases a steady state solution is found within 10 iterations. Realistic boundary conditions were used to solve steady states across a range of boiler loads and pressures during the start-up.

3.7 Complete process and control model

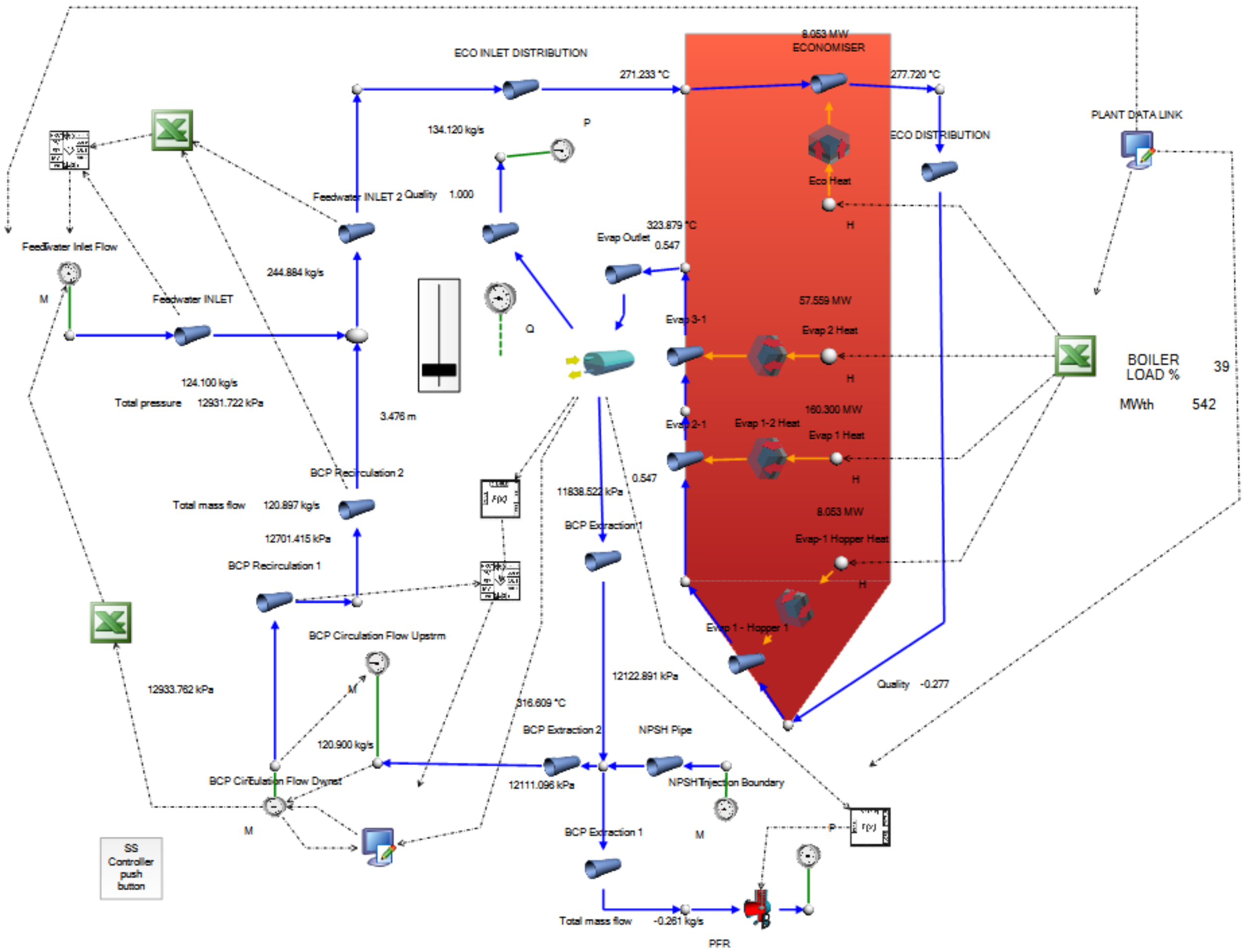


Figure 33: Complete process & control model

Figure 33 displays the completed process model together with the main control elements. The evaporator was reduced to a single tube as the non-uniform heat transfer was not of interest to the present study and results in faster simulations, but can easily be added if required. The collecting vessel dump valve was also incorporated and can be seen in the bottom right hand corner of Figure 33, which opens in case of a high level in the collecting vessel. The control system consists of two PID controllers, one to control feedwater inlet flow and the other to control the level inside the collecting vessel. The dump vale is controlled by a function generator. Additionally, real plant data can be imported (as recorded/measured) and used to change the boiler load and inlet feedwater temperature at the same time intervals as the recorded data. This is achieved through a script that was developed by Willie le Grange, [54]. The plant data link can be seen in top right hand corner of Figure 33.

In summary, complete control is achieved by the following objectives and assumptions which is directly based on the reference plant design:

- The pressure boundary condition at the outlet of the system (superheater inlet) is fixed. Thus, the main steam pressure is not connected to a controller, but the boundary acts as a pressure control device by keeping the pressure constant and allowing the mass flow rate to change.
- Feedwater inlet flow is adjusted with a controller to maintain the economiser minimum flow rate. When the controls are enabled a change in circulation flow rate results in changes the measured feedwater total flow. The resultant error from the minimum flow value (245 [kg/s]) and total measured flow are then routed to the feedwater controller to adjust the feedwater inlet flow.
- Collecting vessel level is controlled by manipulating the recirculation flow rate in proportion to the level in the collecting vessel.
- A function generator is used to open and close the collecting vessel dump valve based on the level in the vessel. The function is also directly based on the reference plant design, of which details can be found in Appendix A.

The model is first solved in steady state, the “SS controller” is then activated to calculate a circulation flow rate to achieve energy balance on the collecting vessel. The simulation is then started for only one time step, and deactivated to disconnect the collecting vessel quality boundary condition. The simulation can then be started. The controllers are activated by stopping and de-activating the simulation, connecting the controller outputs to the control elements and starting the simulation again.

The model was rigorously tested over a wide range of operating conditions, for both steady state and transient modes. The only adjustment made was a change of the alpha *(α)* integration weighting factor to 0.7. Time steps of $t = 0.1[s]$ and $\alpha = 0.7$ yields satisfactory results in terms of stability, accuracy and simulation time.

4. Model Validation

4.1 Steady state validation

The model was first validated in steady state by comparing steady state solution parameters to some of the measurements taken from the reference plant. Finding stable measuring points during the start-up proved to be a challenge as the start-up values vary significantly, however some stable points could be identified and was averaged ranging from 2 – 10 min periods. Minimal adjustments were made to the model for calibration to the plant data. Only small adjustments to the boiler efficiency (heat input), main steam pressure boundary condition (due to measurement location) and circulation flow were needed for more accurate modelling results. Important assumptions and notes on the validations are as follows:

- Tank quality in the model (for validation) is arbitrary as the circulation flow in the model is adjusted to match the actual plant circulation flow. This was done to enable comparable temperatures for the economiser inlet and outlet as the circulation flow has a large impact on these temperatures.
- Circulation flow on the power plant is considered unstable, as the flow changes directly with a change in the collecting vessel level. Higher flow before the measuring period results in higher than predicted economiser temperatures and lower flow preceding the measuring period results in lower than predicted temperatures. This is due to the large surface area and heat retention properties of the economiser.
- Pressure differentials are also influenced by the circulation flow and temperature as noted above.
- As the fluid inside the evaporator is in a saturated two-phase condition, the pressure is directly related to the evaporator outlet temperature.

Table 3 below contains a summary of the steady state results from the model compared to the measured plant data. Steady state results from three different loads are compared. Generally, good correlation is obtained between the model and plant data, with the largest errors observed in the economiser temperatures and pressure drops due to the circulation flow changes, as explained above.

	Boiler Load Case **Input Data**							
	% BMCR	**MWth**	**Feedwater Supply Temperature**	**Feedwater Supply Pressure [MPa]**	**Collecting Vessel Pressure [MPa]**	**dP (Total pressure drop) [kPa]**	**Economiser Outlet Temperature [°C]**	**Evaporator Outlet Temperature [°C]**
DCS Measured (avg)	20%	374	130	5.99	5.75	240	260	274
Model Result (ss)	20%	374	130	6.02	5.76	260	230	273
Difference	-	-	-	-0.03	-0.01	-20	30	1
Percentage Error				-0.50%	-0.17%	-8.33%	11.54%	0.36%
DCS Measured (avg)	29%	**539**	228	13.25	12.43	820	289	325
Model Result (ss)	29%	539	228	13.22	12.48	740	283	328
Difference	-	-	-	0.03	-0.05	80	6	-3
Percentage Error				0.23%	-0.40%	9.76%	2.08%	-0.92%
DCS Measured (avg)	45%	942	237	18.98	18.35	630	275	362
Model Result (ss)	45%	942	237	18.98	18.34	640	248	361
Difference	-	-	-	0	-0.01	-10	27	-1
Percentage Error				0.00%	-0.05%	-1.59%	9.82%	-0.28%

Table 3: Validation data comparing plant measurements to model results

4.2 Transient Validation

For the transient validation, a complete cold start-up of the case study (Plant X) was used to validate the model results. The main steam pressure, firing rate (boiler load) and feedwater inlet temperature of the case study measurements were used as inputs to the model. A short description of the cold-start-up process modelled is provided below.

The results cover a 6 hour start-up, up to once-through operation. The simulation was started with the boiler pressure around 1 [MPa] and a firing rate of 10% BMCR. The firing rate is increased from 10% to 16% BMCR in the first 60 [min]. The first mill is then started (76 [min]) and the load is increased to 28% BMCR, with the mill in service the feedwater inlet temperature is gradually increased from 130 [°C] to 210 [°C] as the HP heater enters service. The main steam pressure is also increased to around 12 [MPa] after which the load is kept constant. Another mill is started after 322 [min] where load is first increased to 43% BMCR, and then to 53% at 341 [min], after which the once-through point is reached as there is no more level inside the collecting vessel. A more thorough description and analysis of the complete cold start-up is provided in the next chapter under section 5.3.5.

During initial validation it was found that the modelled results for the main steam mass flows were lower than the actual plant, and the economiser inlet temperatures were higher. This was a direct result of the heat transfer from the furnace that was linearly scaled to lower loads, leading to an under prediction of the total heat transfer to the evaporator during start-up. The boiler efficiency function was changed to 103%, to enable more of the total furnace heat transfer to be allocated to the tubes of the economiser and evaporator.

For the transient validation, the results of the model are compared graphically to the measurement results for the economiser inlet temperature, main steam mass flow and collecting vessel level.

The economiser inlet temperatures are compared in Figure 34. The model predicts a slightly lower economiser inlet temperature (dark blue dotted line) than the actual plant measurement (solid red line). This is due to a slightly lower circulation flow rate emanating from the circulation flow rate control function created in the model.

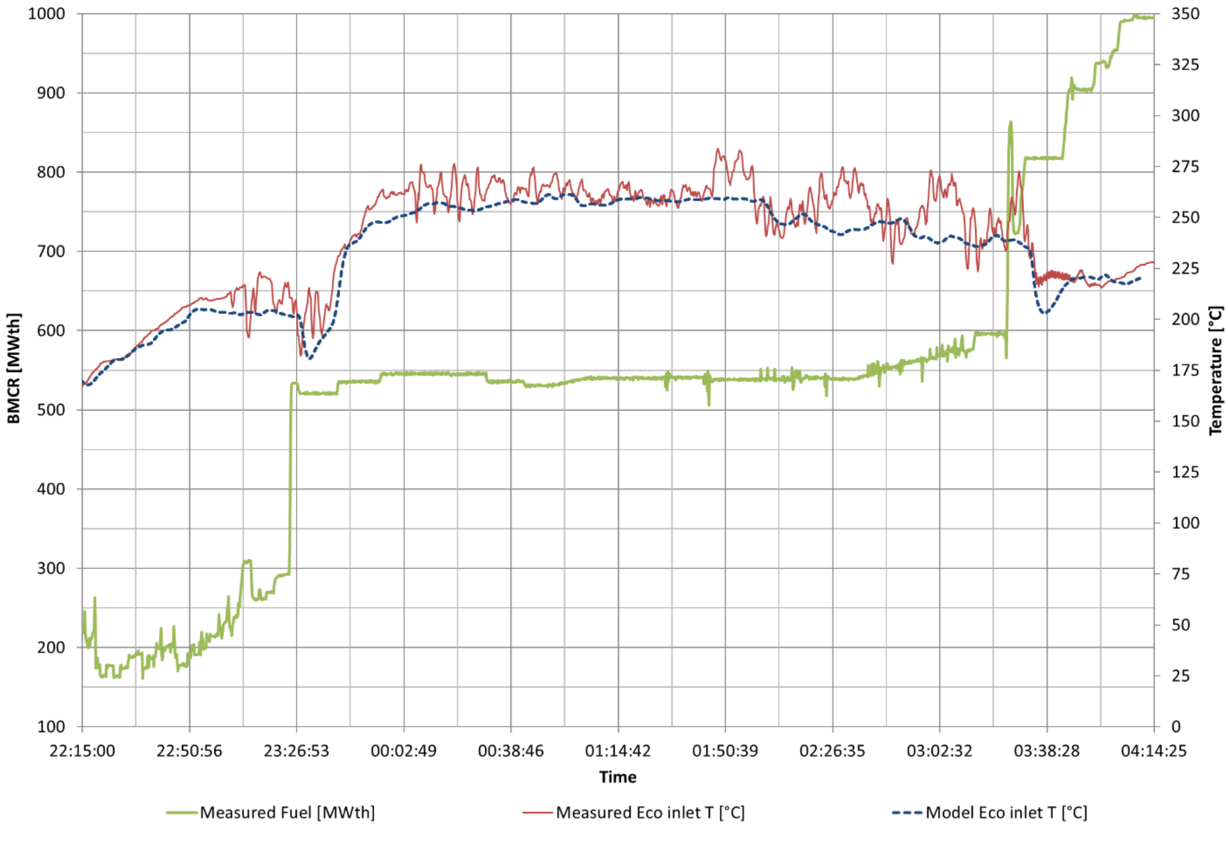


Figure 34: Complete cold start economiser inlet temperature model validation

Heat transfer plays an important role in the model validation. Using the real plant data as inputs has limitations, especially using the measured fuel input – which has its own limitations and inaccuracies. The measured fuel is based on the calorific value of coal used and the mill feeder speed, which can cause unmeasured fuel input to the furnace. As an example, when a mill is started it is usually purged with hot air before the coal feeder is started, the purging process can add a significant amount of unmeasured fuel and thus heat to the furnace. These effects can be seen in Figure 34 before the first mill is started before 23:26:53, mill purging is also suspected 01:50:39 (in which the mill probably failed to start), 02:26:35 as well as 03:02:32. However, even with some measurement inaccuracies, the model simulates the overall process behaviour well and the economiser inlet temperatures can be considered to be sufficiently accurate.

Figure 35 contains the results of the main steam mass flow of the simulation (dark blue dotted line) and the main steam measurements of the plant (red). The main steam mass flows obtained in the model are in good agreement with the measurement results, with the exception of the very low loads up to 75 [kg/s]. In this region the actual main steam mass flow measurement is notoriously inaccurate, hence the difference could be attributed to measurement uncertainty. Other exceptions of larger differences between the model and plant data are the suspected mill purging, as was the case for the economiser inlet temperatures. Smaller deviations are attributed to the heat transfer functions implemented in the model as well as the high frequency fluctuations which are discussed in more detail in section 5.3.4.

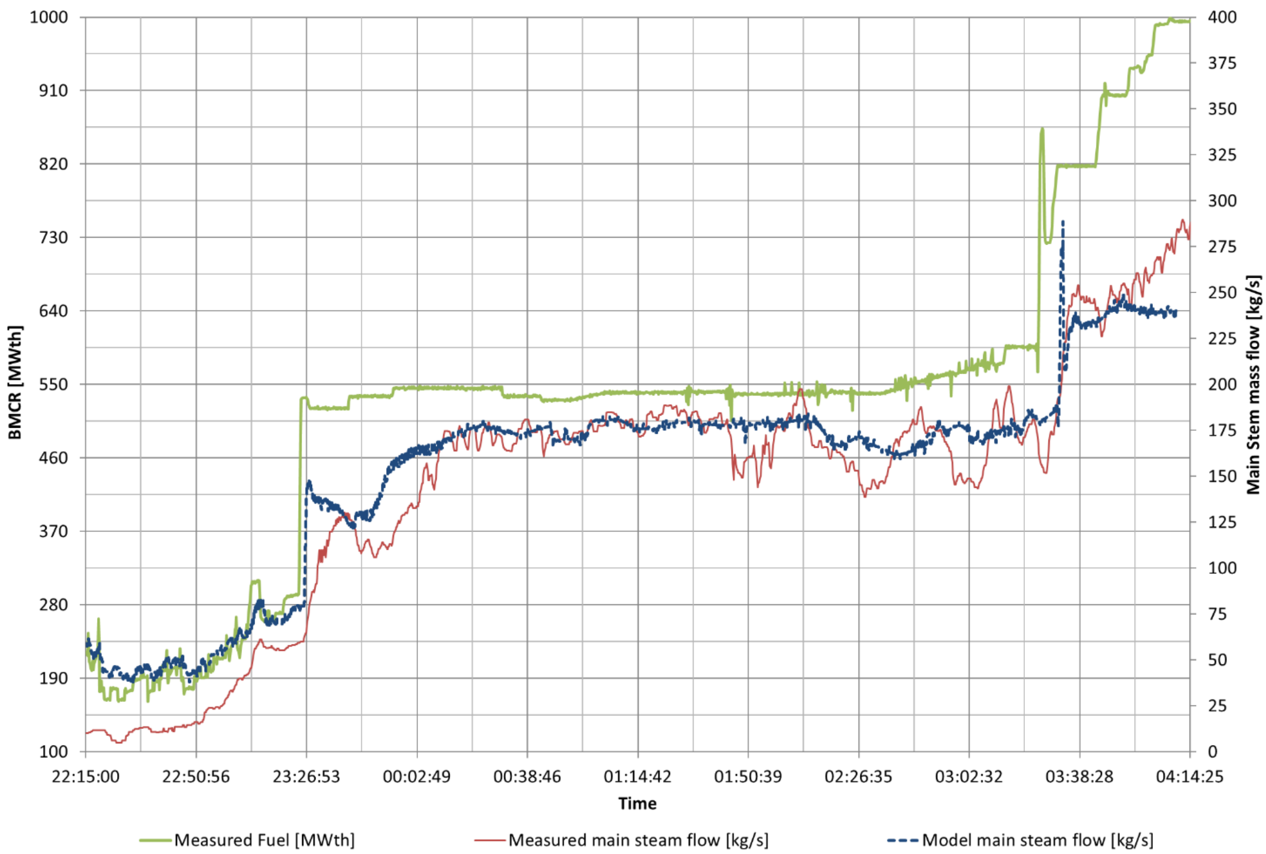


Figure 35: Complete cold start-up main steam mass flow validation results

Figure 36 compares the collecting vessel level measurement to the model simulation result.

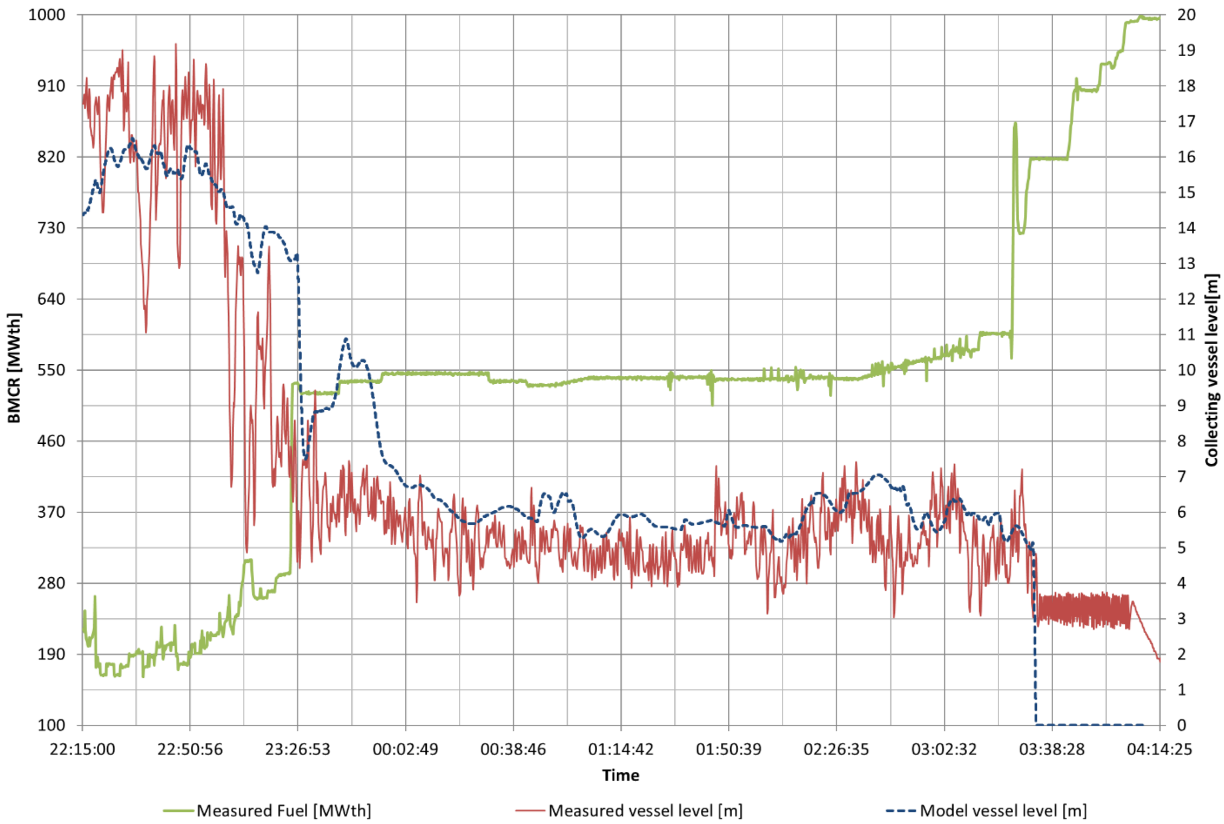


Figure 36: Complete cold start collecting vessel level validation results

In the first 60 minutes, the model predicts a slightly lower level in the collecting vessel but a slightly higher level after the mill is started. Just before the 03:38:28 time stamp, once-through operation is achieved, with the model showing the level decrease to zero [m], whereas the plant measurement still indicates a level. The level remaining is due to the boiler circulation pump that goes to leak-off – recirculating directly back to the collecting vessel. Hence the results are in good agreement regarding the one-through operational point. The measured level in the collecting vessel is subject to high frequency fluctuations that are not simulated in the model. A general discussion regarding the high frequency fluctuations can be found in section 5.3.4.

5. Model application

5.1 Steady state analysis

In continuation of the fundamental analysis presented in section 3.4, the model was used to study the steady state characteristics of the complete integrated start-up system. The "SS Controller" was used to calculate the circulation flow required to obtain thermodynamic equilibrium for various boiler loads, pressures, inlet temperatures as well as collecting vessel/tank quality boundary conditions. For a given main steam pressure, the relationship between the level in the vessel and the vessel quality is fixed as the quality boundary on the vessel dictates the mass fraction of vapour to total mixture mass which in turn dictated the volume fractions of saturated liquid and steam and thus the level inside the vessel. By changing the quality boundary condition and solving the circulation flow required for each of the different quality values, the steady state relationship between the level in the vessel and the circulation flow can be obtained. This relationship is presented in Figure 37.

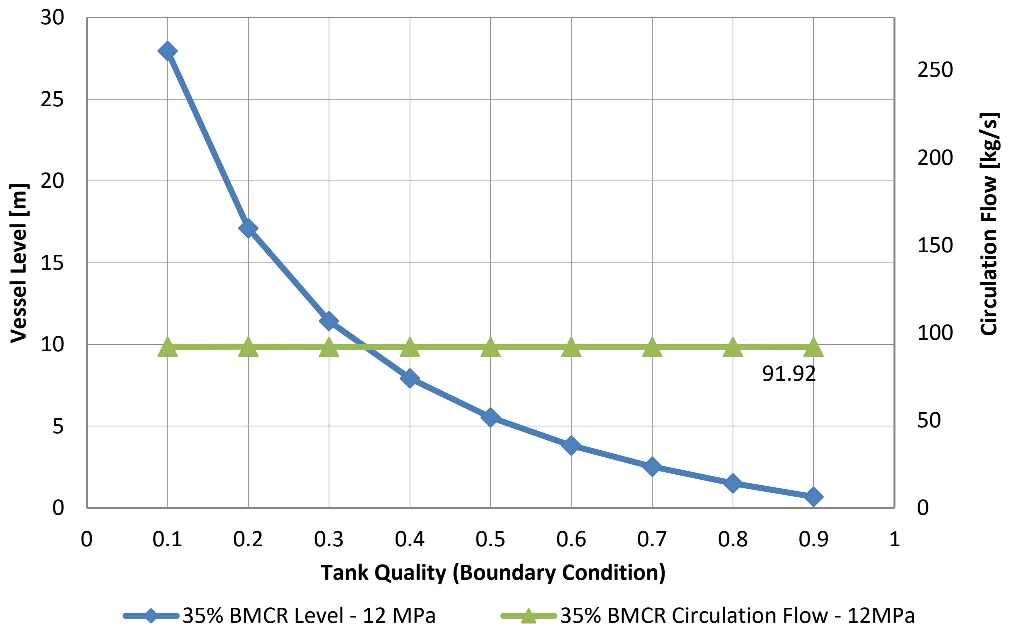


Figure 37: Fixed vessel level and quality relationship and circulation flow results

Analysing the results presented in Figure 37, it can be concluded that for a given boiler load, pressure and inlet temperature there exists only one circulating flow rate value that is independent of the level in the collecting vessel. This means that that thermodynamic equilibrium is also independent of the level in vessel. The finding is significant as it means that in steady state, to achieve energy balance in the collecting vessel, there can only be one circulating flow rate value, which is completely independent of the level inside the vessel.

This also remains true for any other boiler loads, as presented in Figure 38, which also supports the fundamental steady state analysis remarks presented in 3.4.

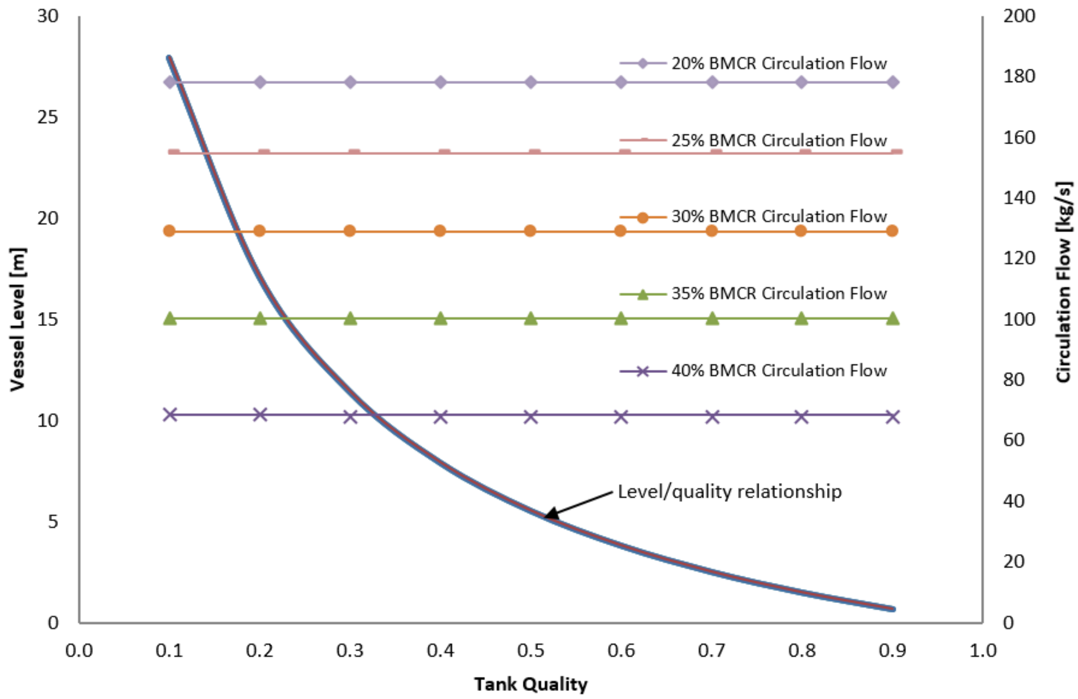


Figure 38: Fixed vessel level and quality relationship and circulation flow results for various boiler loads

In Figure 38, the thick purple line is the fixed level and quality relationships for the various boiler loads. As can be seen, there is a different circulation flow rate for each of the various boiler loads and for a constant boiler load again there is only one constant circulation flow for any level inside the vessel that ensures thermal equilibrium.

5.2 Transient parametric study without controls activated

To gain a deeper understanding of the level dynamics within the integrated system without any control system interference, simulations were conducted by changing individual parameters and analysing the effects on the collecting vessel level. The parameters chosen were based on plant knowledge and also the main parameters that are controlled in the system that are able to change during the start-up. The main parameters studied were changes in main steam pressure, changes in circulation flow rate which is directly related to a change in inlet feedwater flow, changes in feedwater inlet temperature and changes in boiler load. Some values in the descriptions are rounded, with decimal values omitted to provide better perspective.

5.2.1 Pressure step increase and decrease

First the effect of changing the main steam pressure was conducted at 39% BMCR. The main steam pressure was increased in 100 [kPa] steps, larger step changes results in model instabilities due to these large step changes being unrealistic. The results obtained are displayed in Figure 39.

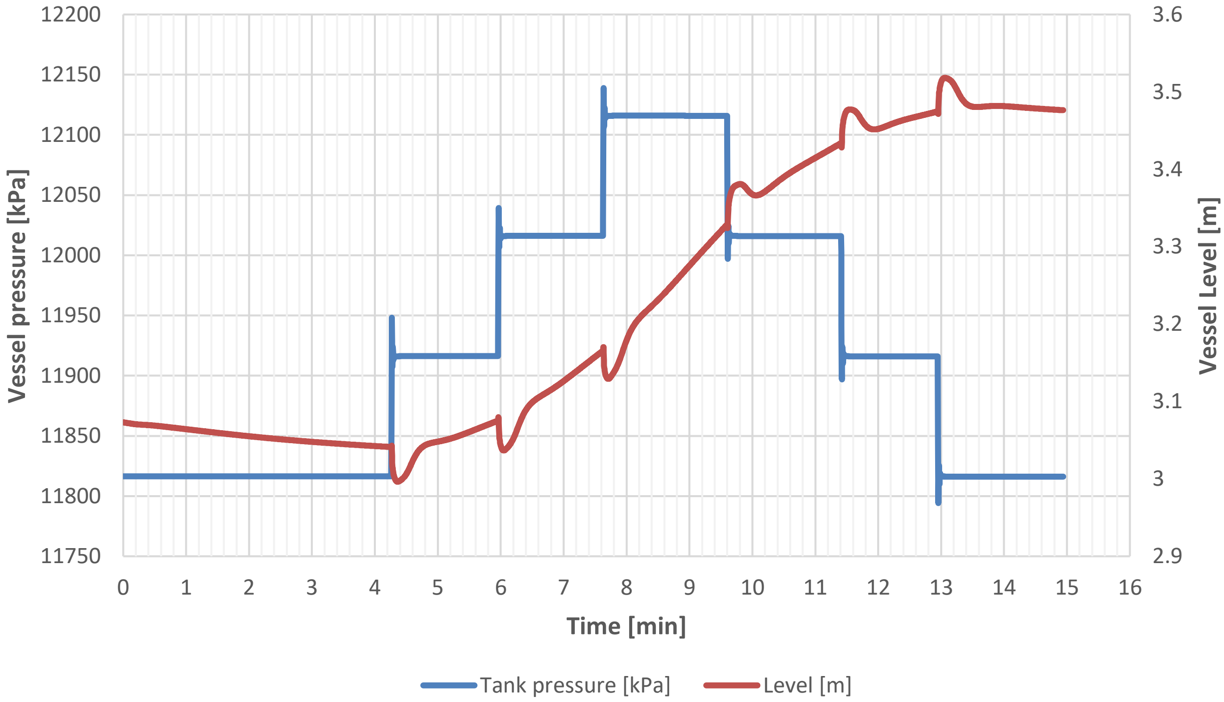


Figure 39: Pressure changes in 100 [kPa] steps and the effect on vessel level

With a sudden increase in main steam pressure the level in the vessel decreases steeply at first but starts to recover and reaches the pre-disturbed level with a steep rise within 35 [s]. The level thereafter increases, with each step of higher pressure, the rate of level increase is also increased.

When the pressure is decreased (9.6 [min]), a sudden rise in level is noted followed by a decrease and then a slower increase, which at lower pressures results in a level decrease, as can be seen with the step change at 13 [min].

The initial level anomaly might be attributed to the so-called "false water level" phenomenon whereby a change in pressure not only changes the density and saturation temperature of the working fluid, but also changes the specific heat capacity of water and volume void fraction that results in a level change. However, the initial anomaly might also be due to the non-minimum phase behaviour.

The increase and decrease of level after the anomaly are then due to the change in mass balance as higher steam flow rates are obtained at lower pressures and lower steam flow rates at higher pressures.

5.2.2 Linear changes in main steam pressure

Main steam pressure is linearly increased at a rate of 1 [kPa/s] from an initial pressure of 12000 [kPa] up to 12500 [kPa] (Figure 40). The main steam pressure is then also decreased at the same rate down to a pressure of 11500 [kPa] (Figure 42). The same observations that were made with the step increases can be seen with the gradual changes, changes in level are prolonged and undergo smooth changes. With the increase in pressure (blue), the level (red) in the vessel initially decreases and only then starts to increase due the mass flow imbalance created by the change in pressure.

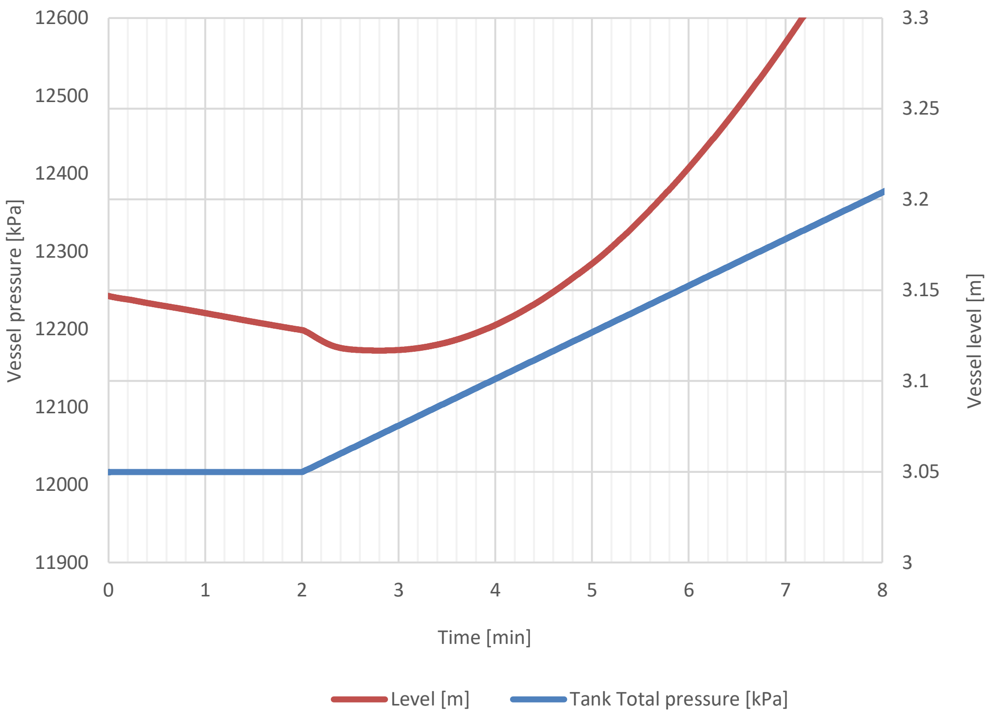


Figure 40: Linear pressure increase and the effect on the vessel level

For the increase in main steam pressure, the associated main steam mass flow change (reduction) is presented in Figure 41.

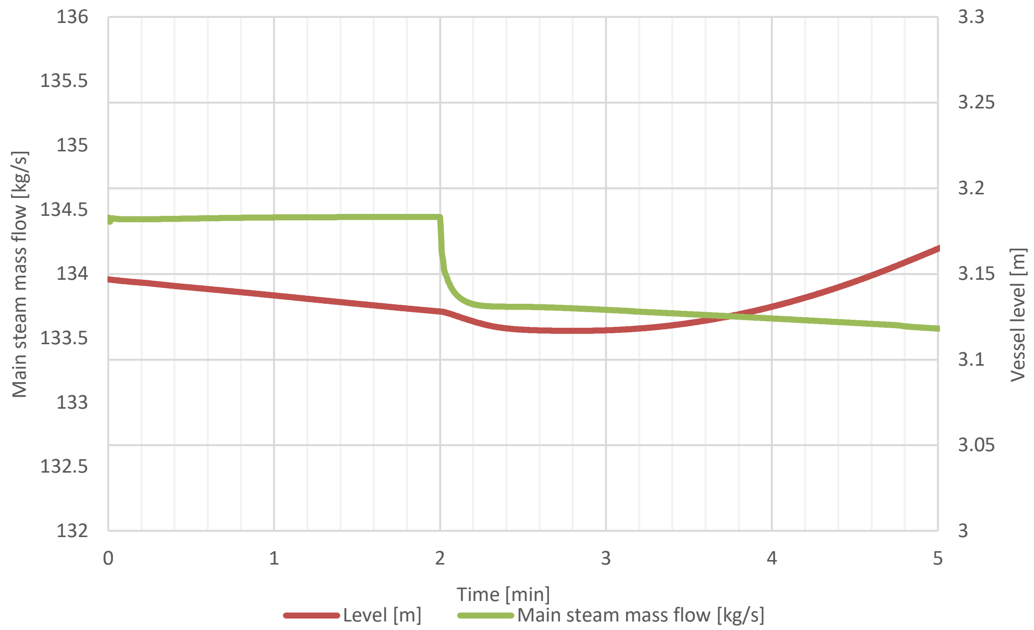


Figure 41: Main steam mass flow and level changes due to increased pressure

Figure 42 displays the change in level due the decrease in main steam pressure, whereby the level first increases and then only starts to decrease due to the mass flow imbalance caused by the reduction of the main steam mass flow.

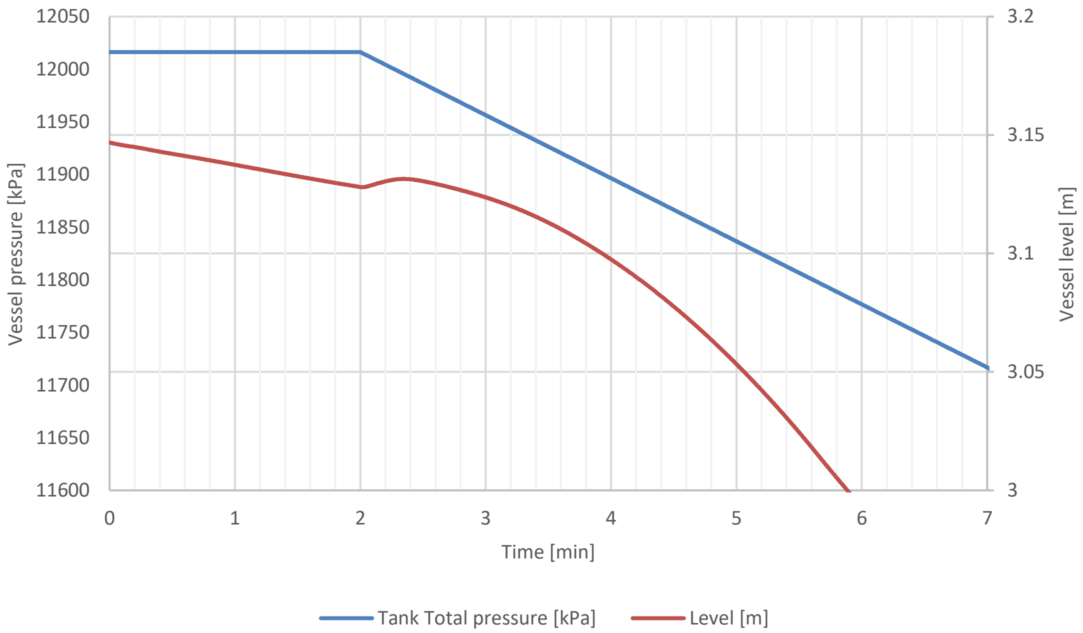


Figure 42: Linear pressure decrease and the effect on the vessel level

5.2.3 Changes in circulation flow rate

Changes in the circulation flow rate has multivariable effects and provides a good example of the complexity of the integrated system. Firstly, manipulation of the circulation flow rate instantaneously changes the inlet feedwater flow rate. Without the controls enabled the inlet feedwater is directly changed based on the mass balance equation *(9)*:

$$\dot{m}_{fwi} = \dot{m}_{fwt} - \dot{m}_{bcp} \tag{9}$$

With $\dot{m}_{fwi}$ the feedwater inlet flow, $\dot{m}_{fwt}$ is the total feedwater flow or minimum economiser/evaporator flow equal to 245 kg/s and $\dot{m}_{bcp}$ the circulation flow rate. Secondly, the flow rate change also causes the economiser inlet temperature to vary since the ratio of "cold" inlet feedwater and "hot" recirculation water changes. Thirdly, changes in the circulation flow rate also affects the pressure inside the collecting level. As the vessel pressure at the main steam outlet is fixed, the circulation flow rate change changes the pressure drop in the suction piping towards the BCP which disturbs the momentum balance around the collecting vessel.

Figure 43 displays the results obtained from the model for a change in the circulation flow rate at 30% BMCR and a 12 [MPa] main steam pressure. In this simulation the circulation flow rate is reduced, starting from the 2 [min] time stamp, circulation flow is reduced from 120 [kg/s] to 95 [kg/s] within 15 [s] and is then kept constant for about 6 [min]. The circulation flow rate is then increased from 95 [kg/s] to 130 [kg/s] within 15 [s] at the 8 [min] time stamp. The reduction in circulation flow rate (red colour) results in the total tank level (blue) to increase and the pressure (orange) to decrease. Even after no more adjustments are made to the flow rate the tank pressure continues to decline. The multivariable and delayed effects can clearly be seen – the reduction in tank/vessel pressure is not only due to lower circulation flow rates, but also due to the change in momentum balance around the tank and a reduction of economiser inlet temperature caused by the increased feedwater inlet flow, which also changes the pressure drop across the economiser/evaporator system. Boiler load units in the graphs are in [MWth].

The average transit time from the point where the circulation flow and feedwater flow meet at the entrance to the collecting vessel was calculated as 6.6 minutes for various boiler loads, while the circulation loop has an average transit time of around 8 minutes. The long transit time contributes to the delayed effect of mass and energy- storage and balance. Before the circulation flow disturbance occurs the steady state mass flow balance of Figure 31 is satisfied with the addition of the NPSH injection flow:

$$\dot{m}_{fwi} + \dot{m}_{NPSH} = \dot{m}_{ms} \tag{18}$$

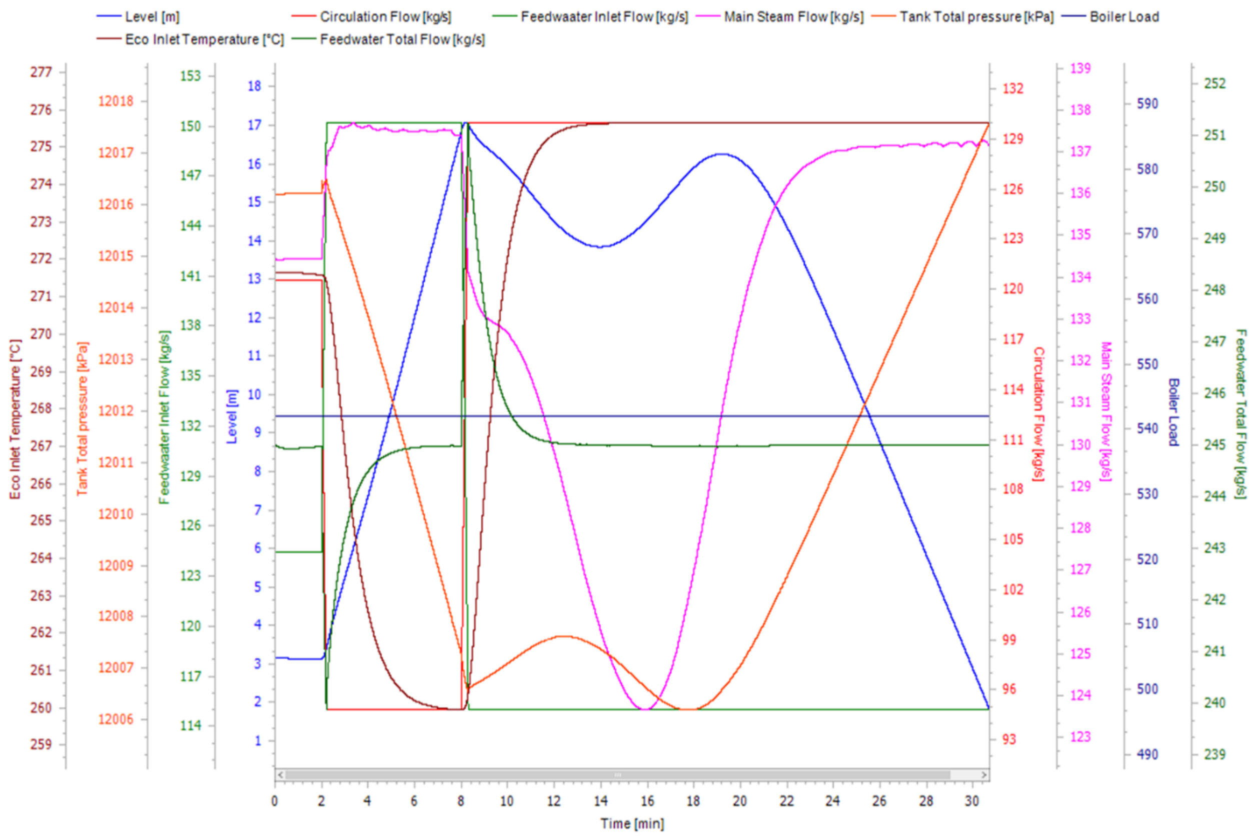


Figure 43: Circulation flow rate changes from 120 [kg/s] to 95 [kg/s] and increased again to 130 [kg/s]

After the disturbance (between 2 and 8 [min]) a significant mass flow imbalance is present. With the circulation flow reduced from 120 [kg/s] to 95 [kg/s], the feedwater inlet flow ($\dot{m}_{fwi}$) was increased from 125 [kg/s] to 150 [kg/s], with the NPSH injection flow ($\dot{m}_{NPSH}$) unchanged at 10 [kg/s], the total inlet flow to the control volume is 160 [kg/s] while the main steam exit flow ($\dot{m}_{ms}$) is only 137 [kg/s], leaving a deficit of 23 [kg/s]. Putting this deficit into perspective - this means that before the circulation flow is increased at time stamp 8 [min], an additional 8.2 [Tons] (approx.) of colder feedwater was added to the system without again leaving the system which takes time to circulate through the system.

Directly following the circulation flow increase (8 [min]) the level starts to decrease, the main steam flow reduces and the vessel pressure is increased. After the 12 [min] time stamp, a reduction in vessel pressure occurs and is followed by an additional increase in the vessel level after the 14 [min] time stamp. This increase in level is directly attributed to the lower economiser inlet temperature, reaching its lowest point around 6 minutes prior to the start of the level increase, due to the additional colder feedwater that was admitted to the system. This is also evidenced by the sudden decrease after 650 [s] in the evaporator exit quality as can be seen in Figure 44. When lower temperature feedwater enters the evaporator for a sufficient amount of time, the quality at the exit of the evaporator will reduce as the mass fraction of saturated steam reduces. The reduction in level seen at the 20 [min]/1200 [s] time stamp, is the effect of both the economiser inlet temperature increase 6 [min] prior (Figure 43) and associated exit quality increase (Figure 44), as well as mass imbalance created by changing the circulation flow rate to a higher value than the pre-disturbed value.

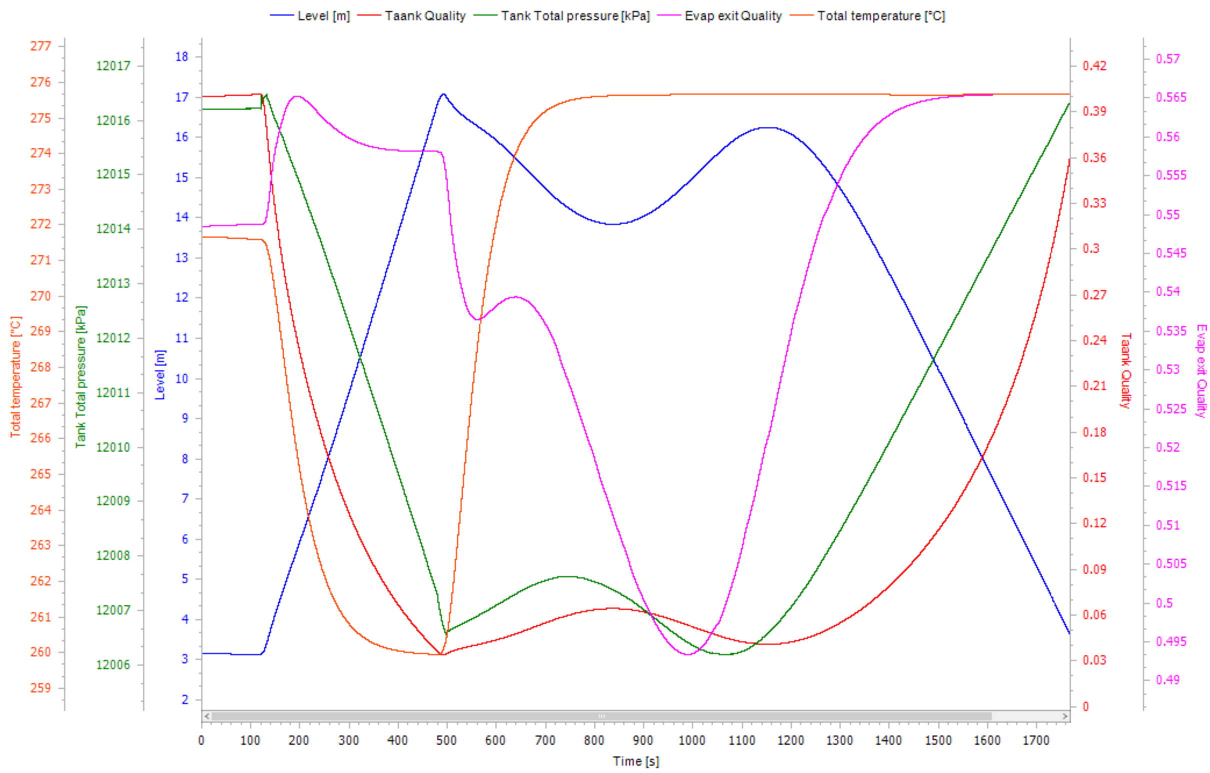


Figure 44: Evaporator exit quality during circulation flow disturbance

In the simulations above the circulation flow was increased above the original value of 120 [kg/s] to 130 [kg/s] at time stamp 8 [min]/480 [s]. When the circulation flow is returned to the pre-disturbed value of 120 [kg/s], the level increase continues until the maximum level is reached. The results are presented in Figure 45 where same simulation is conducted by lowering the circulation flow rate at 2 [min] time stamp but then returning the circulation to 120 [kg/s] at the 5.5 [min] time stamp. The notable difference is that the vessel level decreases only slightly after the increase in circulation flow rate at 8 [min] but thereafter continually increases with a much larger increasing level noticed after 12 [min] and the vessel overflowing shortly after. In essence, this simulation showcases the delayed effect of entering larger amounts of colder feedwater between 2 [min] and 8 [min], which causes the level to increase rapidly at some point much later.

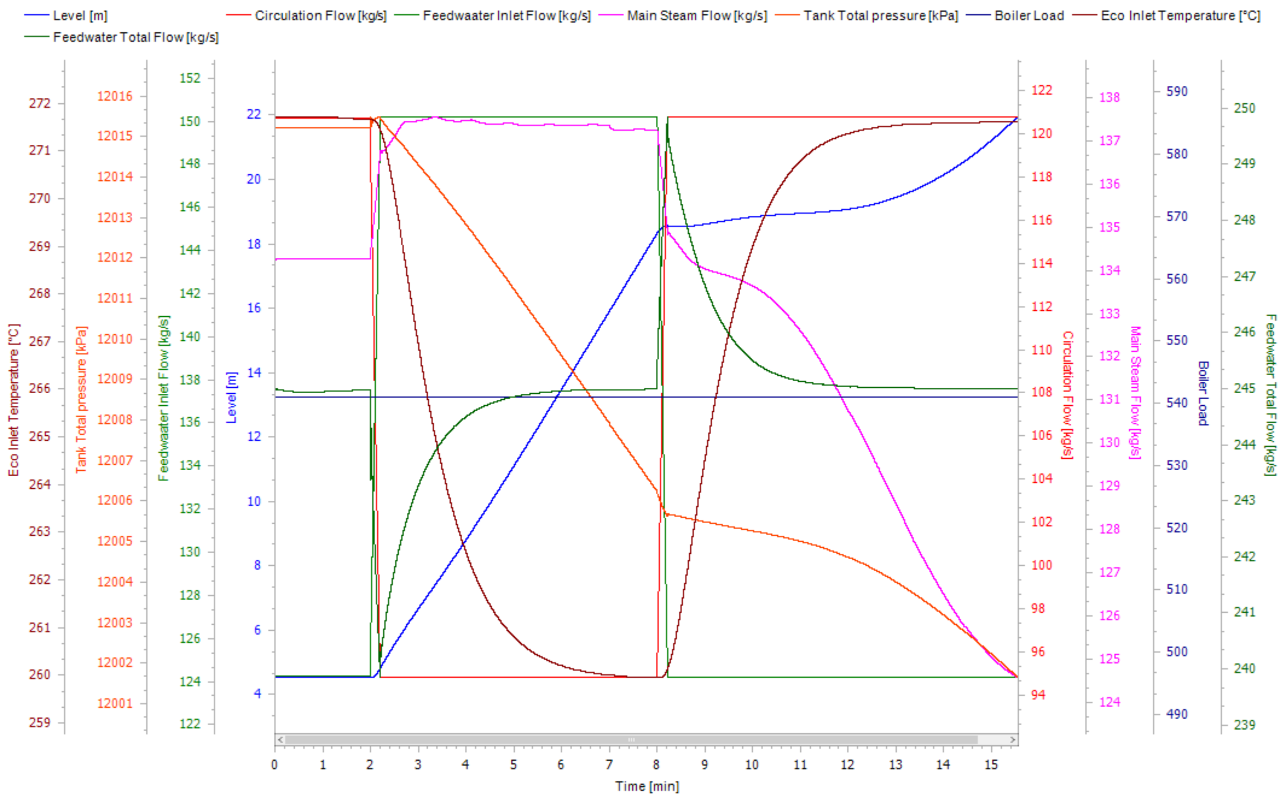


Figure 45: Circulation flow rate changes 120.7 [kg/s] to 95 [kg/s] and increased again to 120.7 [kg/s]

With the observations made, it is expected that the delayed effects on the level increases and decreases would be amplified with a lower feedwater inlet temperature. This is of course also a realistic possibility as the real plant situation could be that a HP heater bank is out of service, which causes a lower feedwater inlet temperature. Figure 46 displays the simulation results for changes in the circulation flow rate with lower feedwater inlet temperatures. The circulation flow is again reduced at the 2 [min] time stamp, from 133 [kg/s] to 95 [kg/s] within 15 [s]. The action to increase the circulation mass flow had to be done about 3 [min] earlier as the level increase is much faster and would result in the vessel carrying water over to the superheater inlet if not increased faster. Before the transient simulation is started a new steady state solution with different circulation flow value of 133 [kg/s] is obtained with the "SS Controller", as the energy balance is changed with a lower inlet feedwater temperature. Figure 46 contains the results, even with a shorter duration of decreased circulation flow and associated increased colder feedwater inlet flow, the lower inlet feedwater temperature results in the level increasing much faster as can be seen after 12 [min]. Figure 47 displays the same results with the evaporator quality shown. As can be seen the evaporator exit quality starts to reduce after 10 [min], with the level starting to increase rapidly after 13 [min] which again corresponds to the economiser inlet temperature reaching its lowest temperature around 6 [min] prior.

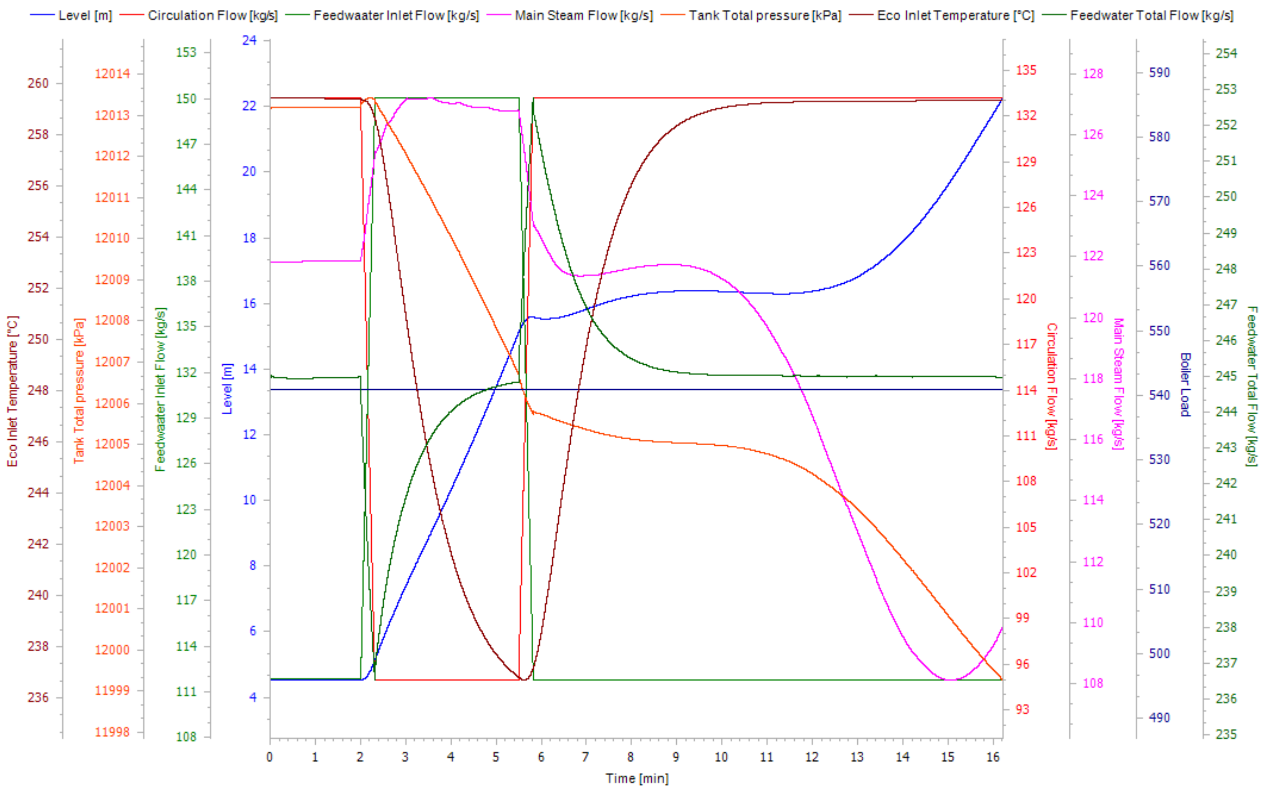


Figure 46: Circulation flow rate changes with decreased feedwater inlet temperature

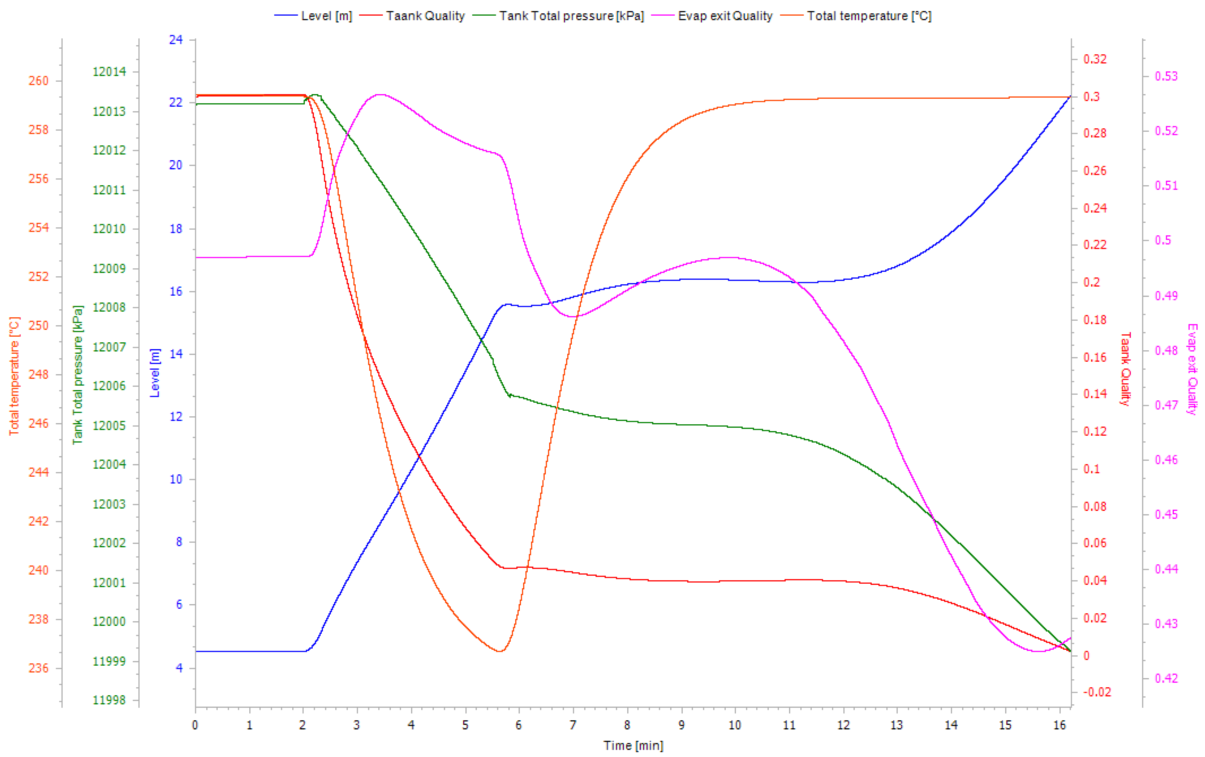


Figure 47: Circulation flow rate changes with decreased feedwater inlet temperature displaying exit quality

A common occurrence during the start-up of the once-through boiler is the collecting vessel level that suddenly diminishes for no apparent reason (circulation flow constant or decreasing). This is also commonly referred to as 'level collapse' within industry. As can be seen from the above results, an increase in feedwater inlet flow (due to reduction in circulation flow) results in a level increase around 6 minutes after the occurrence. The same phenomena in the opposite direction can thus be expected if the circulation flow is first increased (decreasing the colder feedwater inlet flow) and a large amount of warmer water is circulated in the system.

In the simulation presented in Figure 48 the feedwater temperature is kept lower as the effects are amplified, the boiler load remains unchanged at 30% BMCR. In Figure 48 a new steady state solution with lower tank quality boundary condition is obtained to enable the start of the simulation with a high level in the collecting vessel, as a large reduction in level is expected following an increased circulation flow rate. The circulation flow is increased from 133 [kg/s] to 160 [kg/s] between 1 [min and 2 [min] and then returned to 130 [kg/s] between 5 [min] and 6 [min]. The vessel level is initially decreased due to the higher circulation flow rate between 2 [min] and 6 [min]. Thereafter the circulation flow is decreased again to the pre-disturbed value and the level seems to stabilise. As expected the level suddenly starts to decrease at 12 [min], which is again around 6 [min] after the highest economiser inlet temperature is reached, and reduces further until the simulation is stopped. This sudden level decrease is termed 'level collapse' and is thus caused by a large amount of warmer feedwater (due to higher circulation) that was introduced into the economiser, causing the evaporator exit quality to increase followed by a rapid reduction in level seen much later, between six to eight minutes later in this case.

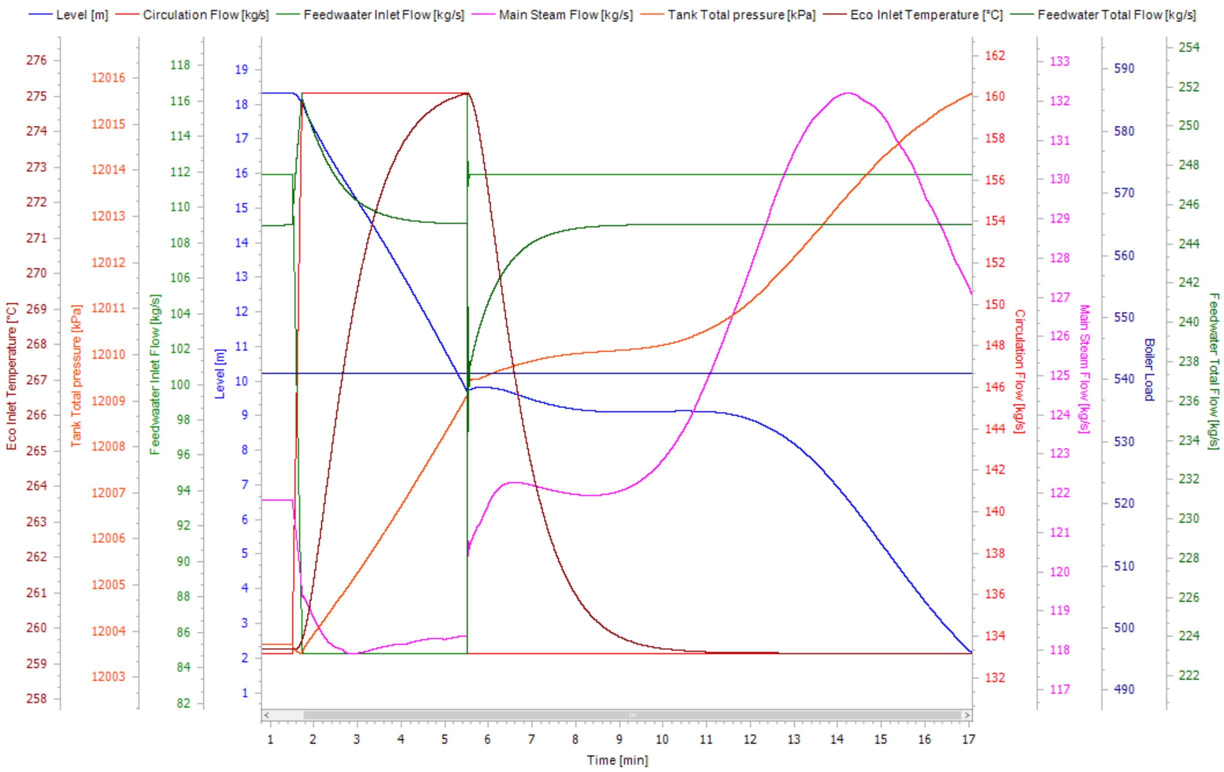


Figure 48: Increase in circulation flow followed by a reduction in circulation flow displaying level decrease

5.2.4 Changes in boiler load

With a reduction in boiler load, intuitively it can be expected that the collecting vessel level would rise due to a reduction in the evaporator exit quality as the heat transfer to the tubes is reduced. As such the level at the start of the simulation is fixed at a higher quality boundary condition for the vessel to start the simulation on a lower level. Figure 49 contains the results of the boiler load reduction. Boiler load units in the graphs are in [MWth].

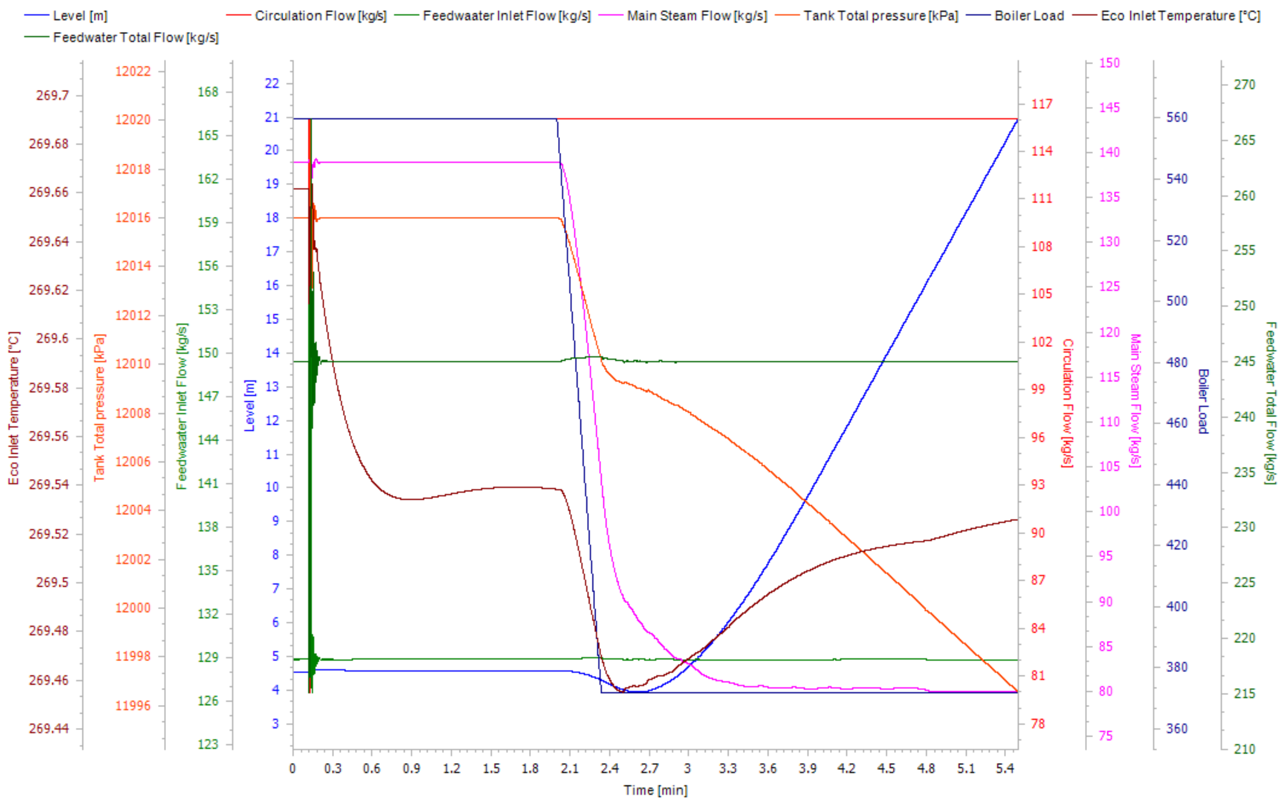


Figure 49: 10% reduction in boiler load and the effects on various parameters

A feedwater flow disturbance is created at 0.1 [s] for 0.1 [s], to change the graph axis values. Reduction of the boiler load starts at 2 [min], and is reduced from 30% BMCR to 20% BMCR at a rate of 0.5 [BMCR/s]. The main steam pressure boundary condition is kept constant at 12 [MPa]. The reduction in boiler load (dark blue) is immediately followed by a reduction in both pressure (orange) and main steam mass flow (pink). The reduction in main steam flow is significant as the flow decreases from 138 [kg/s] to 80 [kg/s] within 3 minutes. The tank pressure first reduces rapidly by about 6 [kPa] within 30 seconds, followed by a gradual reduction up to the point where the simulation is stopped. The vessel level initially decreases by a small amount due to the pressure reduction and the starts to rise at 2.6 [min] and continues to increase up to where the simulation is stopped. The rise in level is attributed to the reduction of steam flow caused by the lowering pressure and evaporator exit quality which results in a mass flow imbalance. Where the main steam mass flow starts to stabilise at around 80 [kg/s] after 3.3 [min], a large mass imbalance is present (calculated with equation *(15)*). With the deficit at 59 [kg/s], meaning that there is 59 [kg/s] feedwater being added to the system that is not exiting. Figure 50 displays the significant reduction in evaporator exit quality (pink) from 0.56 to 0.3

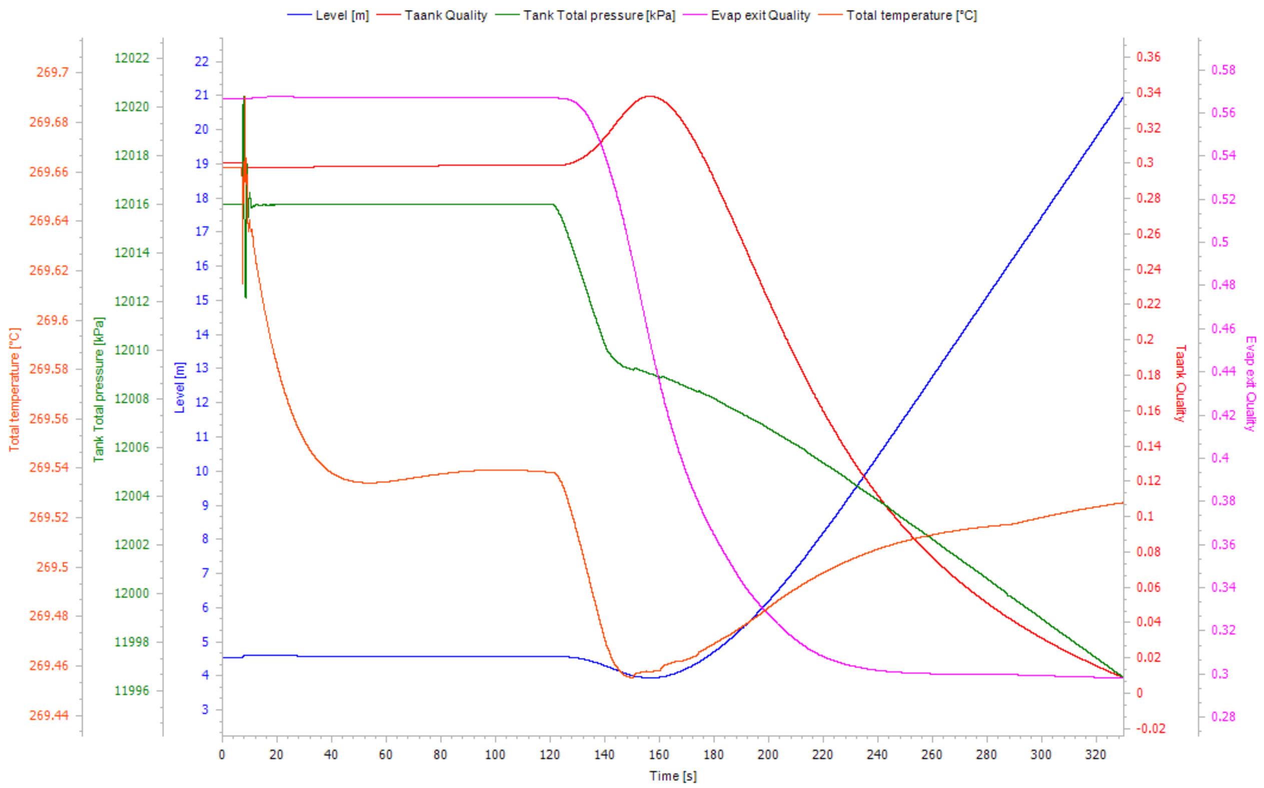


Figure 50: Boiler 10% load reduction and the effect on evaporator exit quality

For the boiler load increase simulation, it is expected that the vessel level would decrease and the simulation is started with a lower tank boundary condition to start the simulation at a higher level in the vessel. Figure 51 displays the results obtained when the boiler load is increased by 10% at 1 [min] time step. The main steam flow and tank pressure starts increasing immediately. The level initially increases due to the increased pressure, but then starts to decline rapidly due to the mass flow imbalance caused by the increasing evaporator exit quality. The evaporator exit quality increases from 0.56 to 0.83 within 3 minutes, this is displayed in Figure 52. Main steam flow increases from 138 [kg/s] to 197 [kg/s] thus creating a mass flow imbalance of 59 [kg/s].

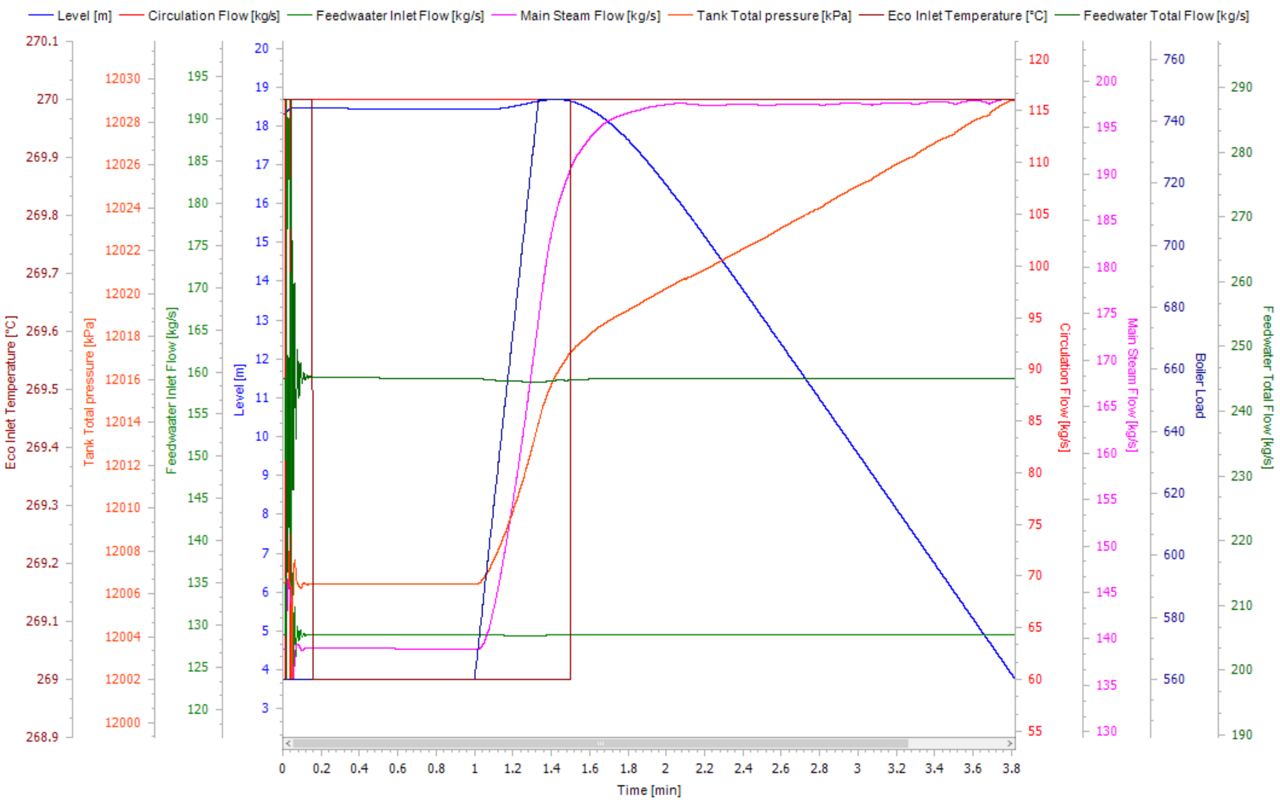


Figure 51: Boiler 10% load increase and the effects on various parameters

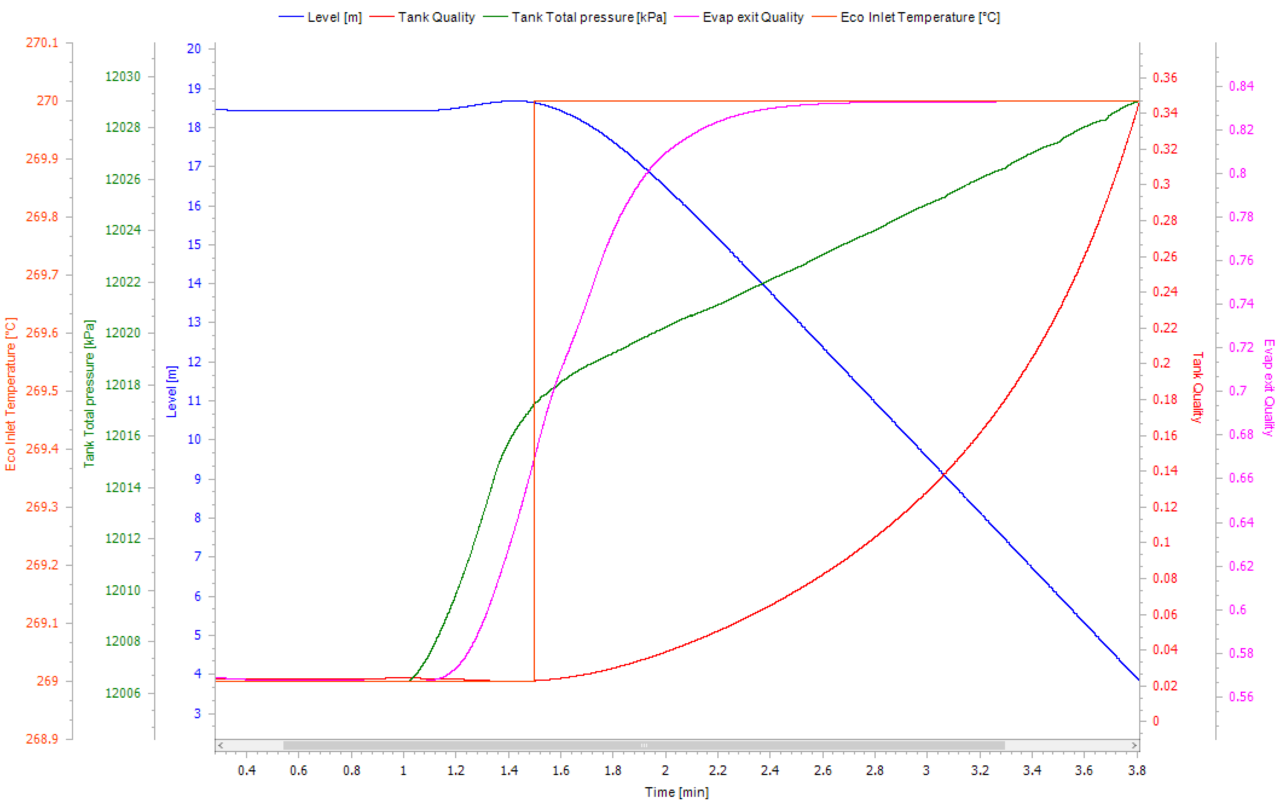


Figure 52: Boiler 10% load increase and the effect on evaporator exit quality

5.3 Changes in boiler load with the control system activated

From the results presented in section 5.1 and 5.2, it is clear that the level is changed based on three main parameters: Pressure inside the vessel, the mass flow balance (including circulation flow), and the quality of the mixture entering the vessel. Each parameter also has an effect on the other. In this section the results obtained for changing loads during start-up with the control system are presented.

5.3.1 Simulation results: Controller initialisation

Initialisation of the vessel level control already presents a challenge, as the circulation flow rate is determined by the level in the collecting vessel, whilst the simulation is started from a steady state with the circulation flow determined by a zero energy source balance on the vessel. Referring back to the steady state analysis presented in Figure 38: the circulation flow rate is independent of the vessel level, and varies only with load, pressure and inlet temperature. Thus when the controllers are initiated the circulation flow immediately changes to control the level. Figure 53 displays the results of starting/initialising the controllers. The circulation flow on the first time step (0.1 [s]) is 116 [kg/s], with the controller reducing the flow very quickly to 30 [kg/s], which causes the level to increase rapidly, with the controller following by increasing the circulation flow. After 15 [min] the circulation flow stabilises around the original 116 [kg/s] at a new level around 7.5 [m]. Re-calling the steady state solutions presented in section 5.1, since the boiler load did not change, the controller eventually reaches a new thermal equilibrium at the same circulation flow rate, but a different level.

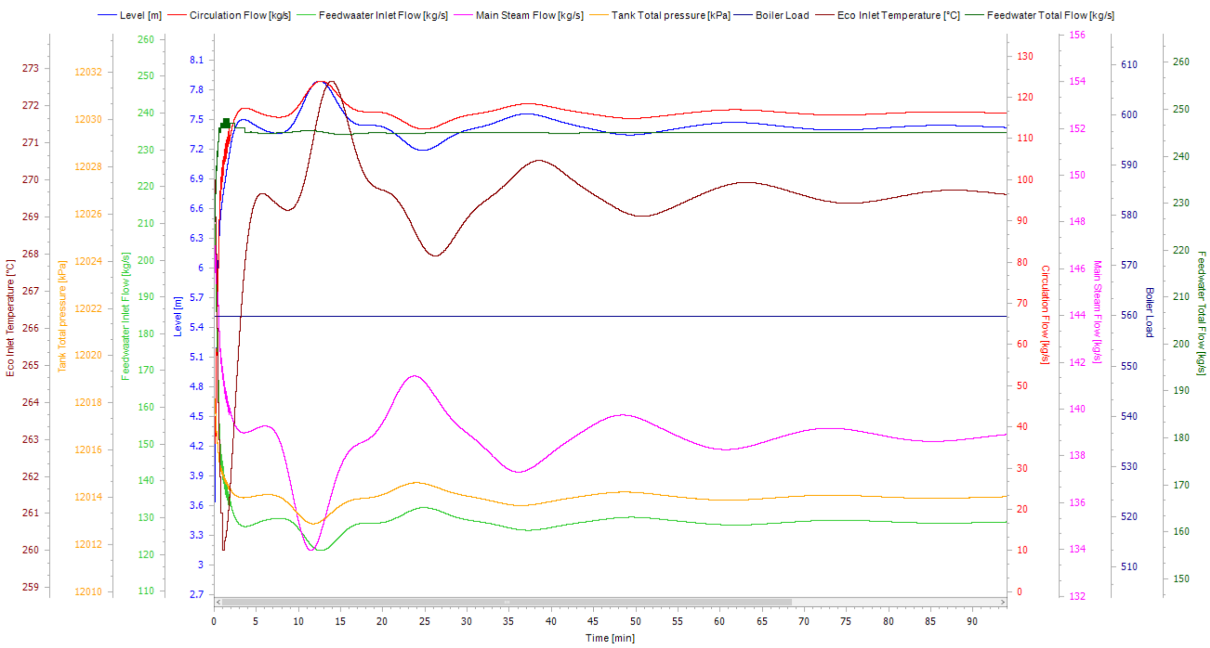


Figure 53: Start of controllers at 30 % BMCR

5.3.2 Simulation result: boiler load increase (30% to 40%)

Figure 54 displays the results obtained when the boiler load is increased at the 5 [min] time step from 30% to 40% BMCR. With the load increase the pressure and main steam flow is also increased as the evaporator exit quality is increased. The vessel level starts to decline, with the circulation flow also decreasing. The decrease in circulation flow lowers the economiser inlet temperature as the colder feedwater flow rate is increased by the feedwater controller. The main steam flow reaches a maximum value at around 191 [kg/s] around the 14 [min] min stamp and then starts to decrease as the increased colder feedwater starts to affect the evaporator exit quality and is followed by the level increasing.

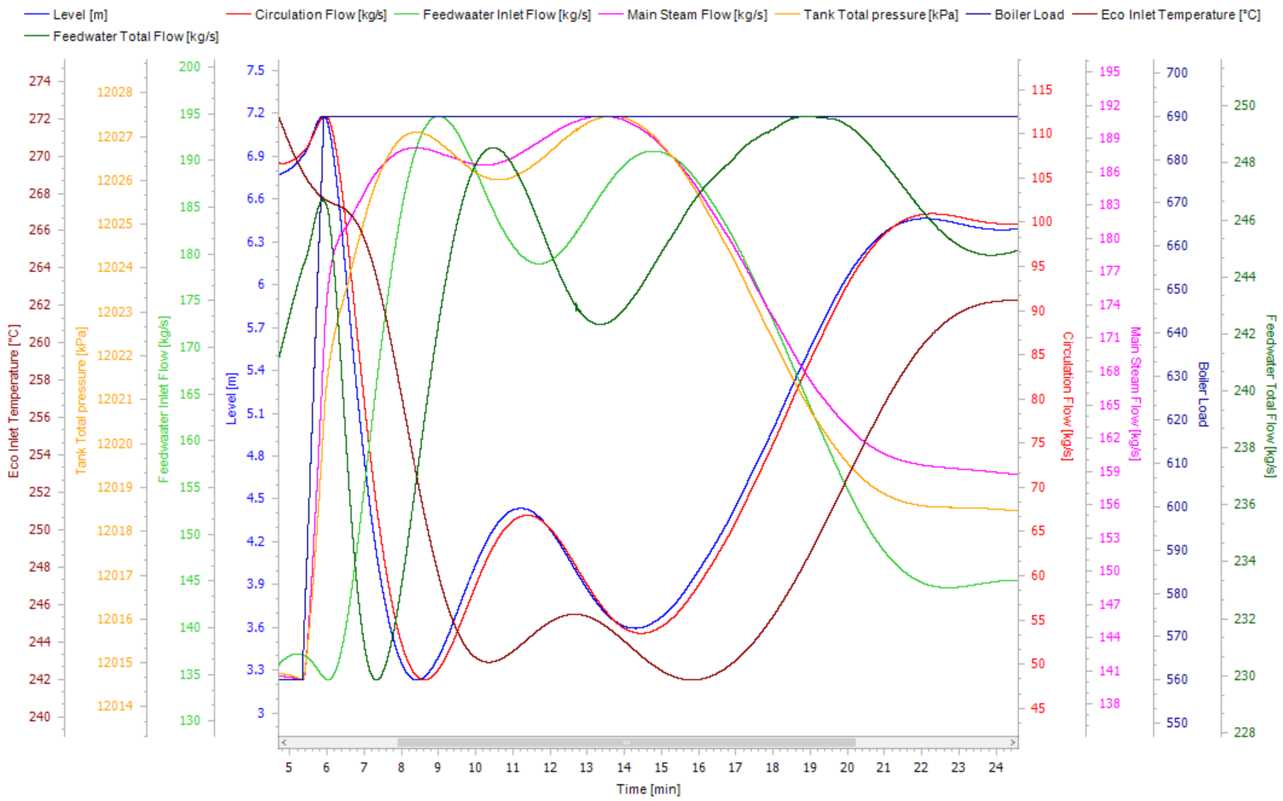


Figure 54: Simulation results for boiler load increase

It would seem that the system reaches some stabilisation after 24 [min], however this is not the case when considering the larger timeframe displayed in Figure 55. The system's unstable nature after the load disturbance can be seen continuing for very long periods. Usually at these loads the turbine is already synchronized, and large fluctuations in main steam mass flow is undesirable. The main steam mass flow fluctuation also has implications for superheater temperatures and superheater temperature control.

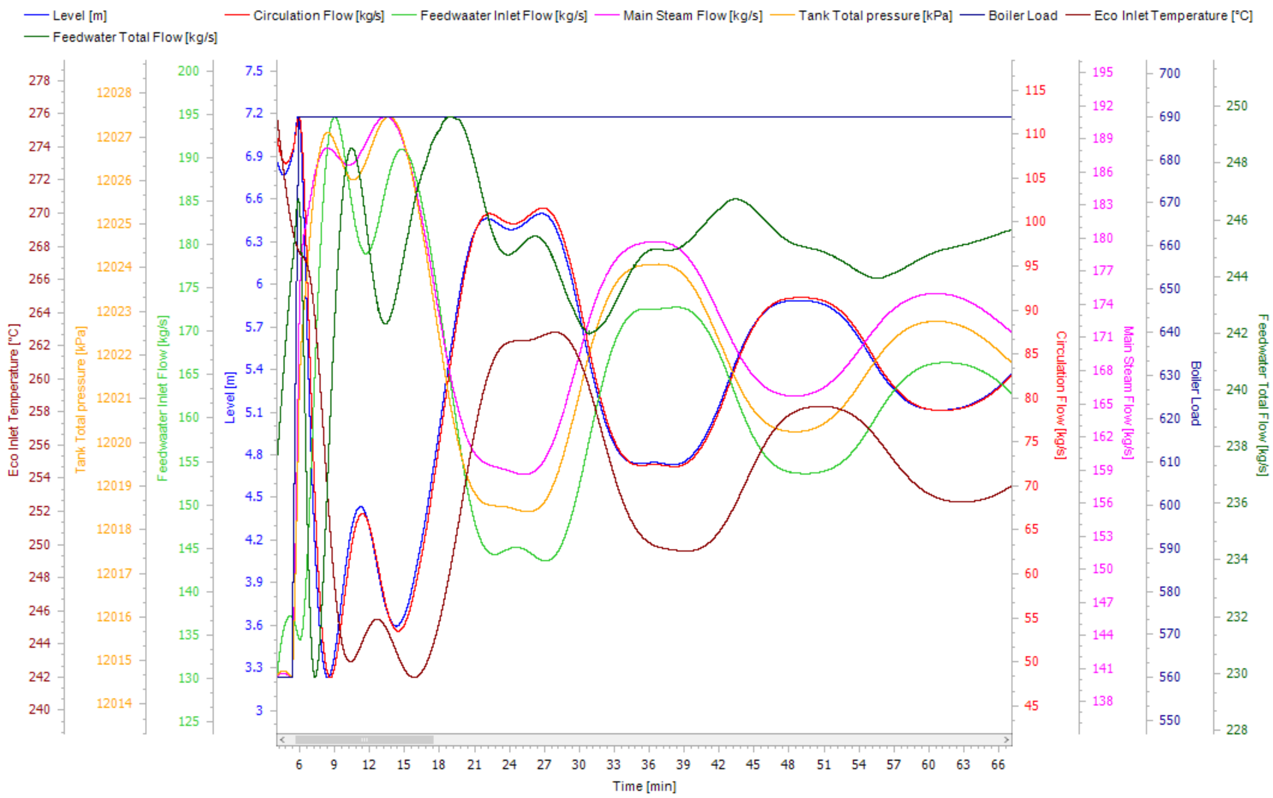


Figure 55: Simulation results for boiler load increase, larger time frame

5.3.3 Simulation result: boiler load decrease (39% to 30%)

In this simulation the loss of a mill is modelled by reducing the boiler load at the 14 [min] time stamp, refer to Figure 56. The level initially decreases rapidly due to lower pressure, but then starts to increase, with the circulation flow control also increasing the circulation flow causing an increase of the economiser inlet temperature from 211 [°C] to the first peak of 266 [°C]. Around 7 minutes after the first peak, the collecting vessel starts to rapidly decline due to the 'level collapse' (as explained previously) at about 29 [min]. The feedwater controller has some erratic behaviour at 32 [min], but soon after recovers. The initial 'level collapse' is then followed by a continuous large cycle of all main parameters, displayed in Figure 57, with the main steam flow cycling by as much as 65 [kg/s] long after the initial load reduction.

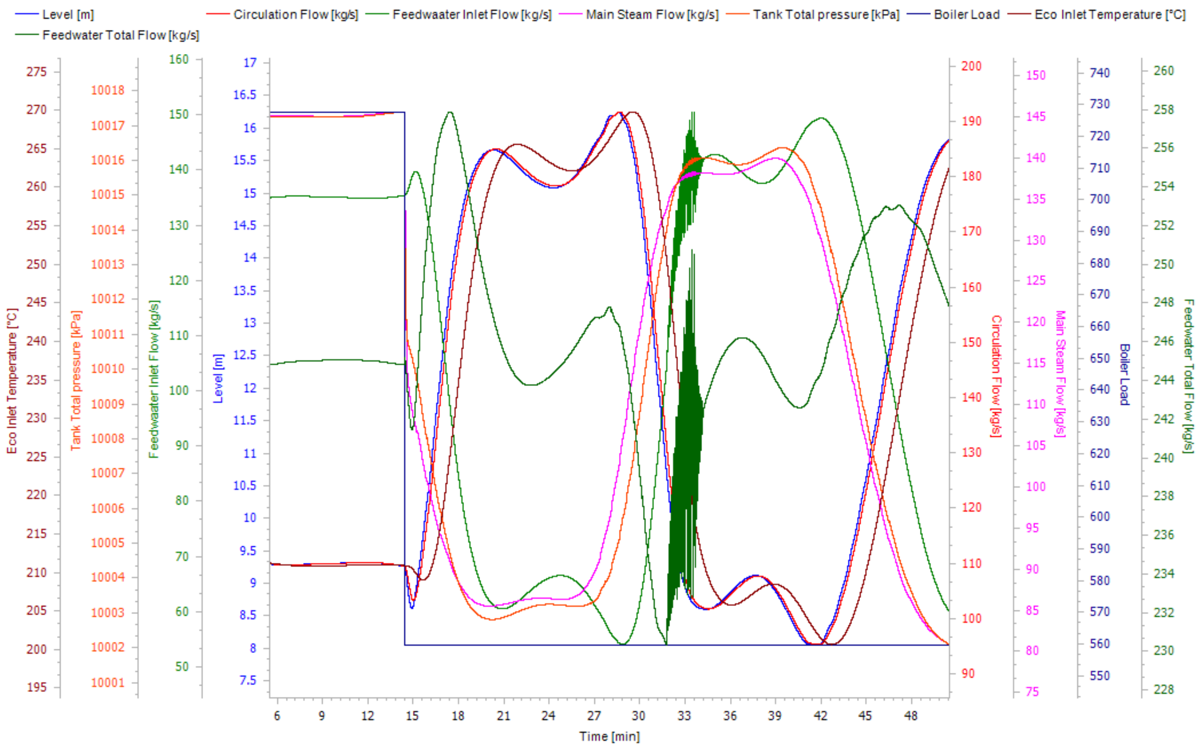


Figure 56: Boiler load reduction simulation

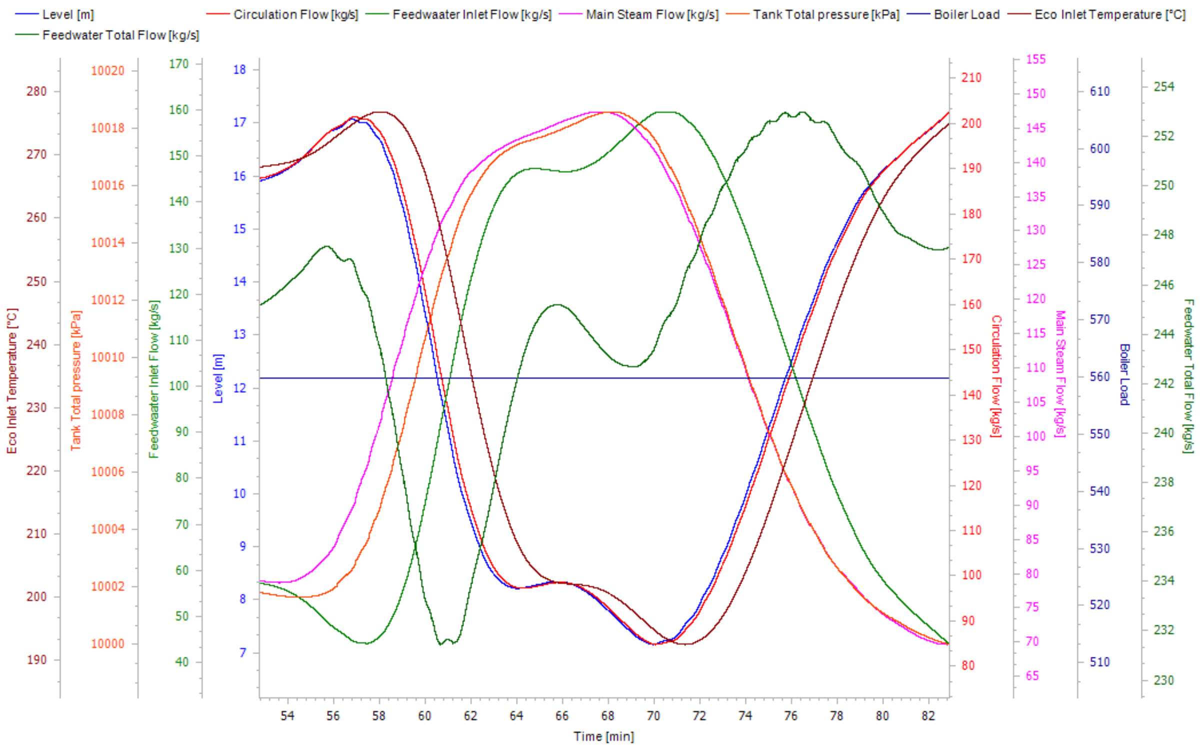


Figure 57: Large continuous cycle after load reduction

5.3.4 Simulation results using real time plant data – load reduction

With the controls activated in the model, a sudden load reduction is simulated by using actual measured plant data as boundary values. Plant data used is the feedwater inlet temperature, boiler measured fuel and main steam pressure (the first pressure measurement at the inlet of the superheater). The simulation is started at approximately 40% load, a mill is lost (tripped) around the 7 [min] time stamp and the boiler fuel reduces (after stabilisation) to an average of 29% BMCR.

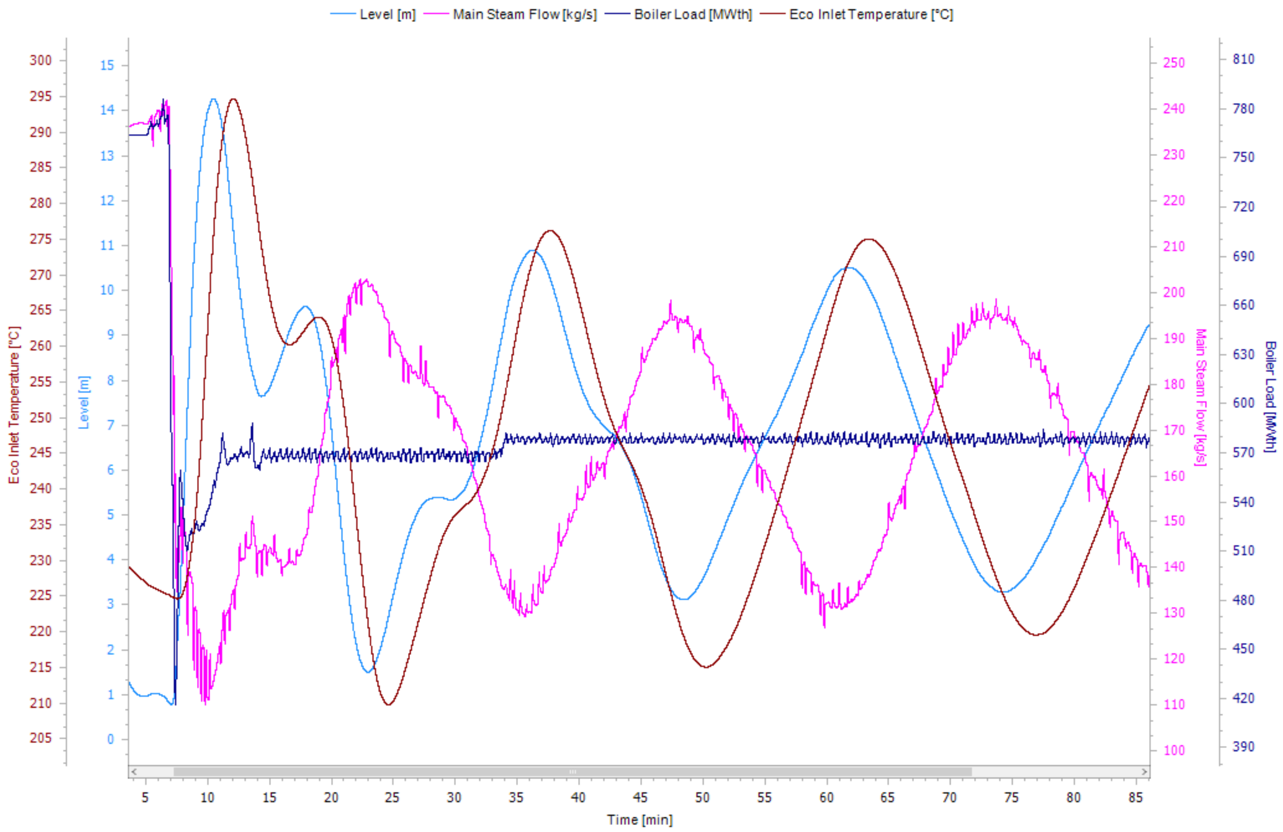


Figure 58: Simulation results after a mill trip indicating fluctuating level, main steam flow and economiser inlet temperature.

With the simulation started around the once-through point, the level in the vessel is very low (less than 1 [m]), after the mill trips, the vessel level increases rapidly to 14 [m], and thereafter cycles between 3 [m] and 11 [m] with an approximate period of 30 minutes. The main steam mass flow cycle has the same period of fluctuation between 200 [kg/s] and 130 [kg/s], with the same period but is almost 180° out of phase with the level and economiser inlet temperature cycling. Figure 59 contains the simulation results with all of the main parameters. From analysing the results, it is clear that the cycling is caused by the circulation flow that is changed based on the level in the vessel. With relatively fast changes in the vessel level the circulation flow is directly changed in proportion to the level change. The circulation flow change then changes the total feedwater flow, which the feedwater controller must then correct. This situation presents one of the challenges in tuning the feedwater control system. With quick changes in the circulation flow rates, the feedwater controller has to be fast enough to maintain the total feedwater flow set point, however the feedwater controller cannot be as fast as the changes in circulation flow, as this would entail extremely high gains on the feedwater controller which destabilises the system during any other operating condition. It is here where the most feedforwards and additions to the feedwater control system are found in industry, mostly to prevent reaching minimum flow trip values, without consideration of overflowing events. Initially, after the mill is tripped, the vessel level and circulation flow increases, with the circulation flow increasing from 10 [kg/s] to more than 180 [kg/s] within three minutes. The large amount of hotter recirculation flow then causes a 'level collapse' and cycle is repeated until the boiler load is changed significantly.

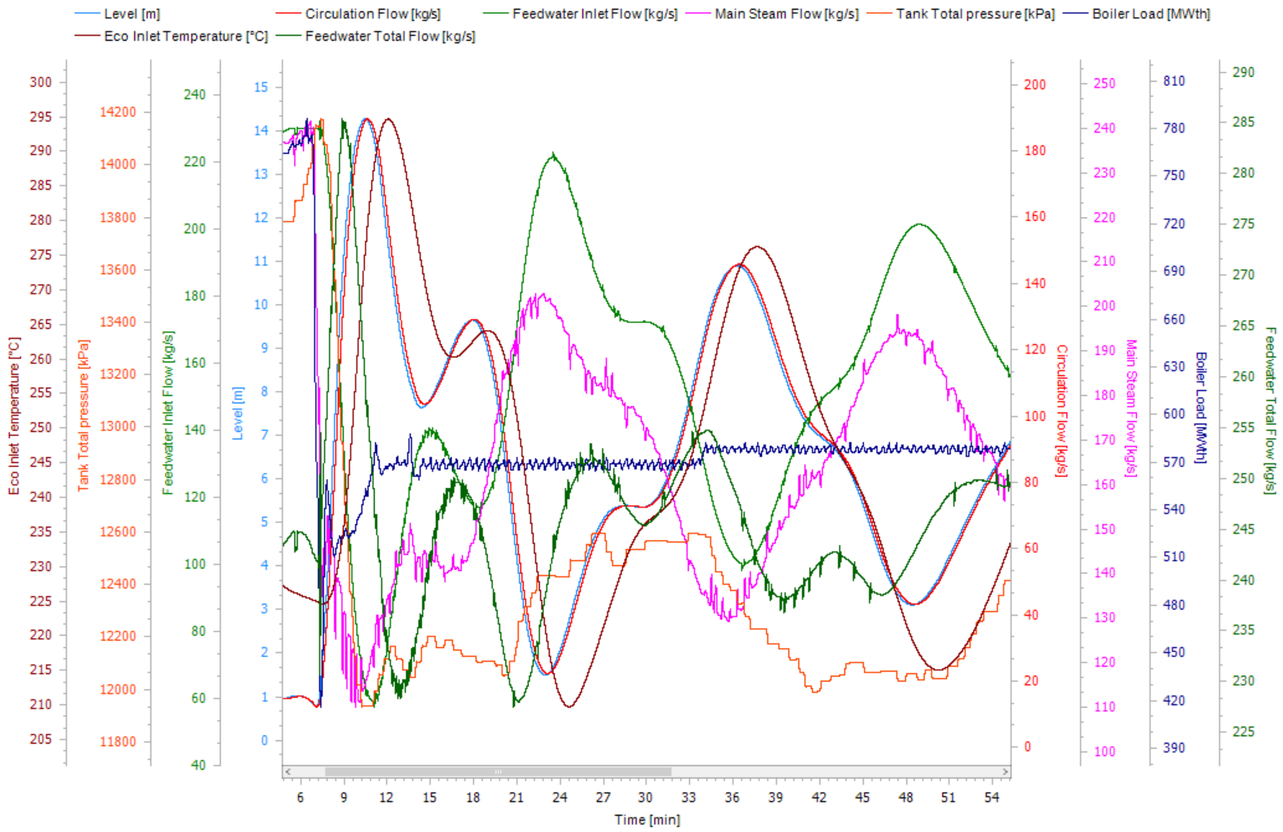


Figure 59: Simulation results after a mill trip with critical parameters

Reviewing plant data revealed that the plant also has a large cycle when a mill is tripped in the start-up process similar to the cyclic behaviour seen in the model after a load reduction presented in Figure 56 and Figure 57, as well as the results presented in Figure 58 and Figure 59. The actual measured economiser inlet temperature and boiler load during the mill trip is presented in Figure 60. A primary large fluctuation of the economiser inlet temperature is present with secondary spikes in temperature noticed within the larger cycle (which is explained later). The modelling results with only the economiser inlet temperature and boiler load is presented in Figure 61.

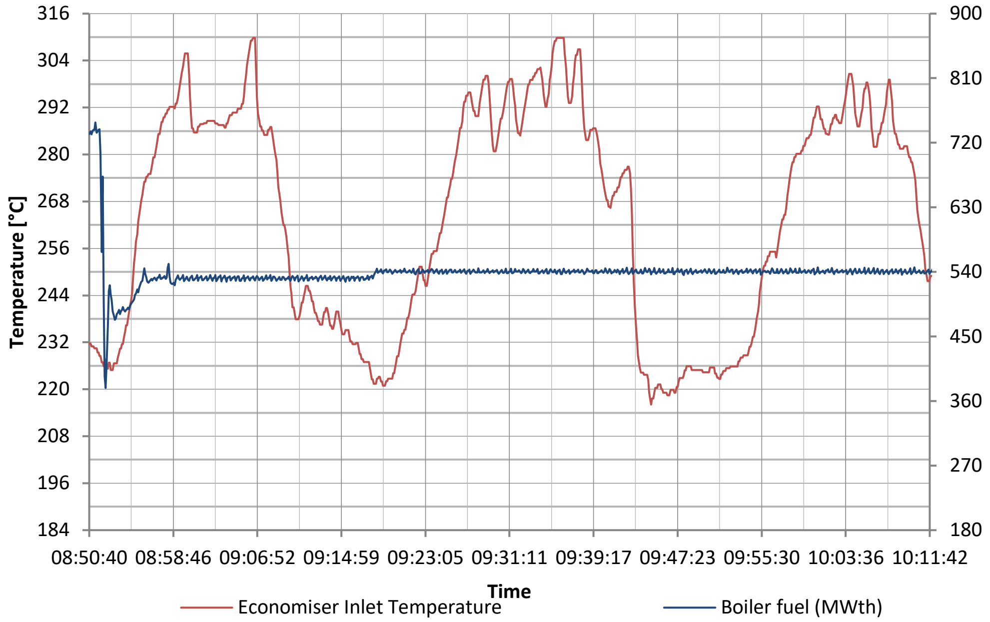


Figure 60: Measured plant data after loss of a mill

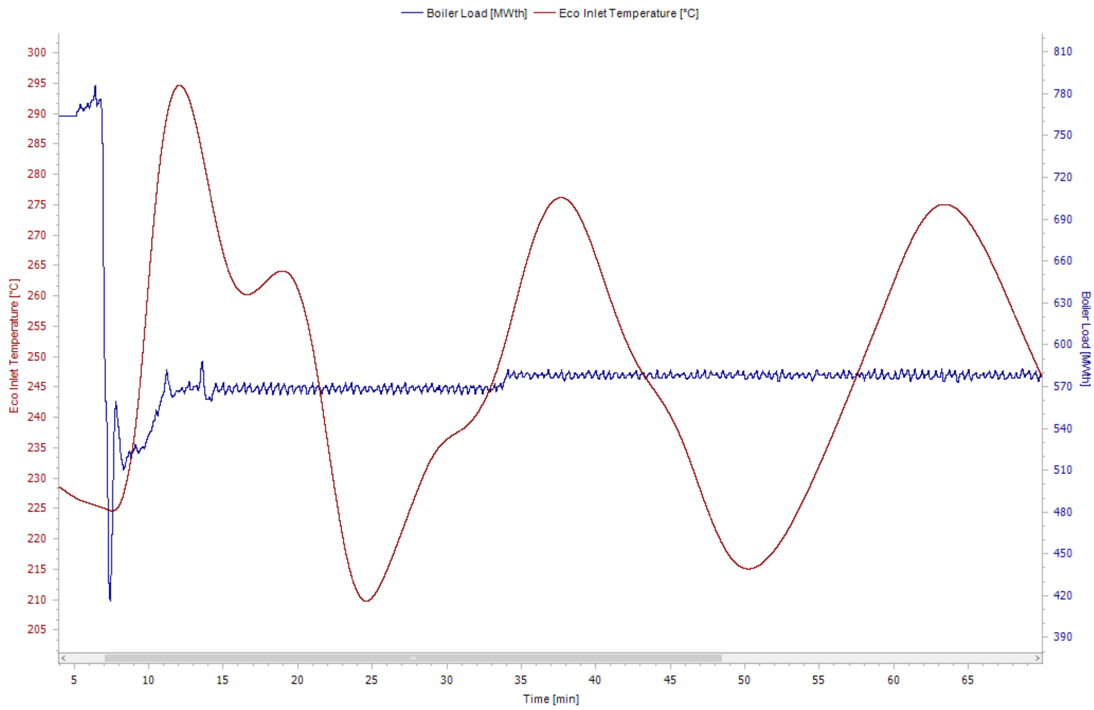


Figure 61: Modelling results for a mill trip showing economiser inlet temperature.

As can be seen in Figure 61, the cycling of economiser inlet temperature is very similar to the cycling presented for the linear load decrease (Figure 56 and Figure 57) as well as the measured data presented in Figure 60. As such, the results from the simulations were then compared to the measured data, and shows very good correlation in terms of the low frequency high amplitude oscillation at the economiser inlet after the load reduction. The results of the comparison are displayed in Figure 62. The absolute maximum temperatures obtained from the model (using plant data inputs) is around 20 [°C] lower than the measured value obtained from the measurement data. The plant data also shows that the higher and lower peak temperatures are maintained for longer periods of time. Further investigation revealed that this is due to the boiler circulation pump minimum flow protection that is activated. The boiler recirculation pump minimum flow control valve opens and the flow is re-circulated straight back to the collecting vessel when the circulation flow is lower than a certain value (+- 90 [kg/s]), hence the forward circulation flow is reduced, Opening and closing of the minimum flow valve can be seen in Figure 63 (purple line). The model does not include this phenomena but can be implemented without much effort as the level vs. flow circulation function can be modified to accommodate these states.

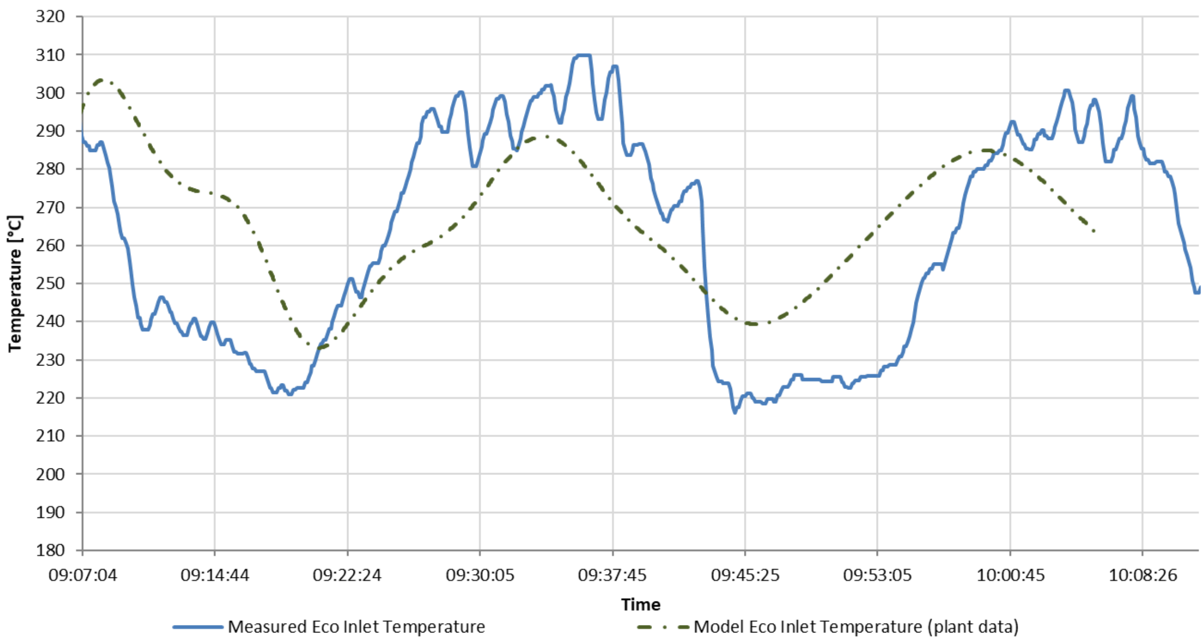


Figure 62: Comparison of economiser inlet temperature obtained from measurements and modelling after loss of a mill

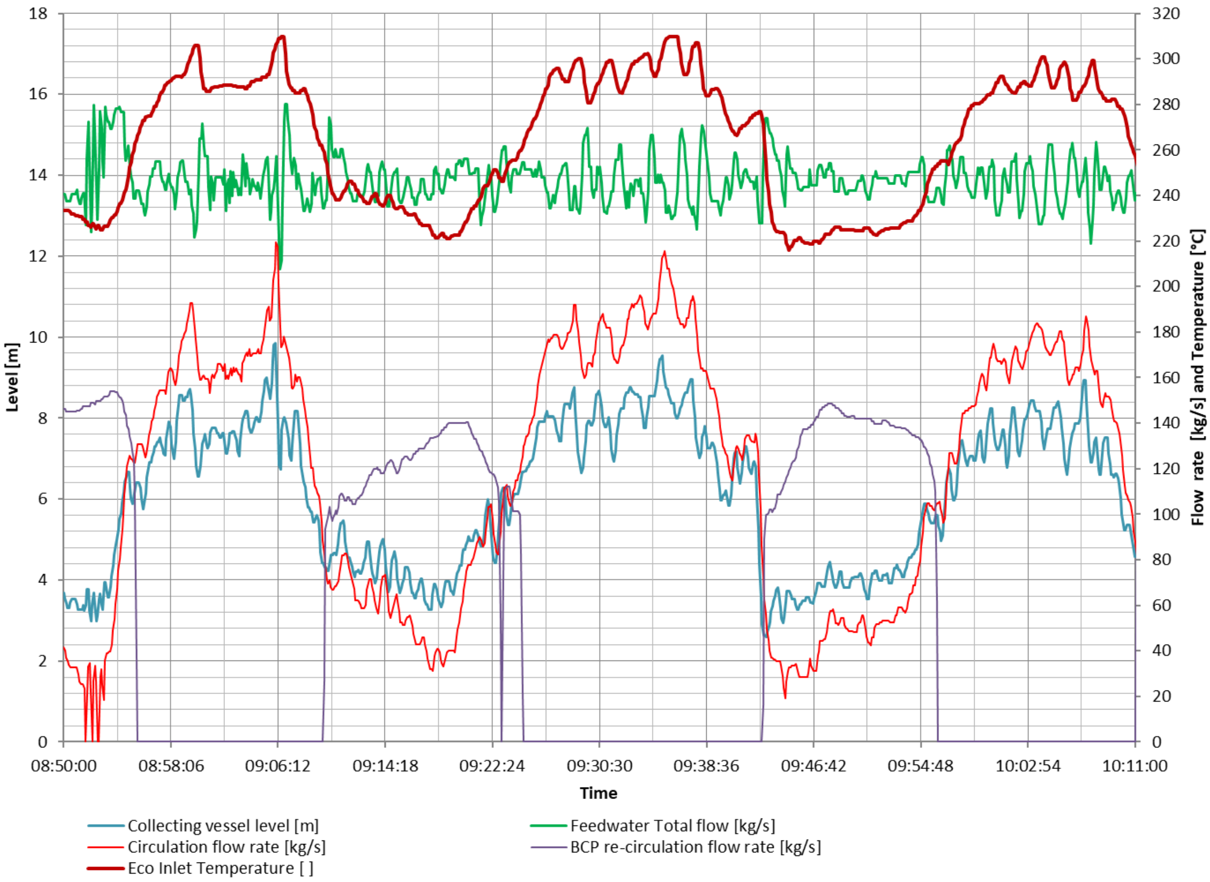


Figure 63: Various power plant measurements after transient loss of a mill

On the power plant the level in the vessel is also calculated based on differential pressure, which can become inaccurate during transients. As can be seen in Figure 59, the measured pressure close to the collecting vessel is not stable and has high frequency pulsations, which could influence the measurement of the vessel level. As the control valve position is based on this level, these errors can be propagated to the circulation flow rate, which disturbs the total economiser inlet flow which the feedwater controller then has to correct. As can be seen in Figure 63 where the feedwater total (economiser/evaporator) flow is not stable at any point after the initial disturbance. Contributions to these disturbances can also be amplified by the delayed reaction of the control valves and the over-and undershoot of set points. Another consideration for the level changes is the control valve characteristics, as the changes in flow rate in the model is linear in proportion to level, whereas the actual plant has control valves with nonlinear characteristics.

Without modelling the local disturbances from possible level measurement errors and circulation flow disturbances (nonlinearities), the model is accurate in simulating the overall process and control behaviour of the complete system. From studying Figure 63, it is concluded that the circulation control valve is also the source of the higher frequency temperature fluctuations, as small movements of the valve position leads to large flow disturbances, which has to be corrected by the feedwater controller which affects the total feedwater flow.

5.3.5 Simulation results for a complete cold start (load increase)

The model is also able to simulate a complete cold start of the once-through unit (this simulation was used in the transient validation). The description of the start-up is repeated here for ease of reference and with more detail. The simulation was started with the boiler pressure around 1 [MPa] and a firing rate of 10% BMCR. The firing rate is increased from 10% to 16% BMCR in the first 60 [min], where the dump valves are operating extensively due to the high level in the vessel. The first mill is then started (76 [min]) and the load is increased to 28% BMCR, with the mill in service the feedwater inlet temperature is gradually increased from 130 [°C] to 210 [°C] as the HP heater enters service, causing the economiser inlet temperature to also increase. As the mill is introduced, the collecting vessel level is reduced followed by a reduction in the circulation flow and associated decrease in economiser inlet temperature but is thereafter increased due to increased feedwater inlet temperature. With the pressure increasing to around 12 [MPa], the load is then kept constant until steam conditions are adequate for turbine synchronisation. The turbine is then synchronised at 224 [min], with no effect on the start-up system which indicates good main steam pressure control and coordination between the turbine inlet valves and HP bypass valves. Another mill is started after 322 [min] where load is first increased to 43% BMCR, and then to 53% at 341 [min], after which the once-through point is reached as there is no more level inside the collecting vessel. The model circulation flow has a low limit flow of 10 [kg/s] to still circulate the NPSH flow.

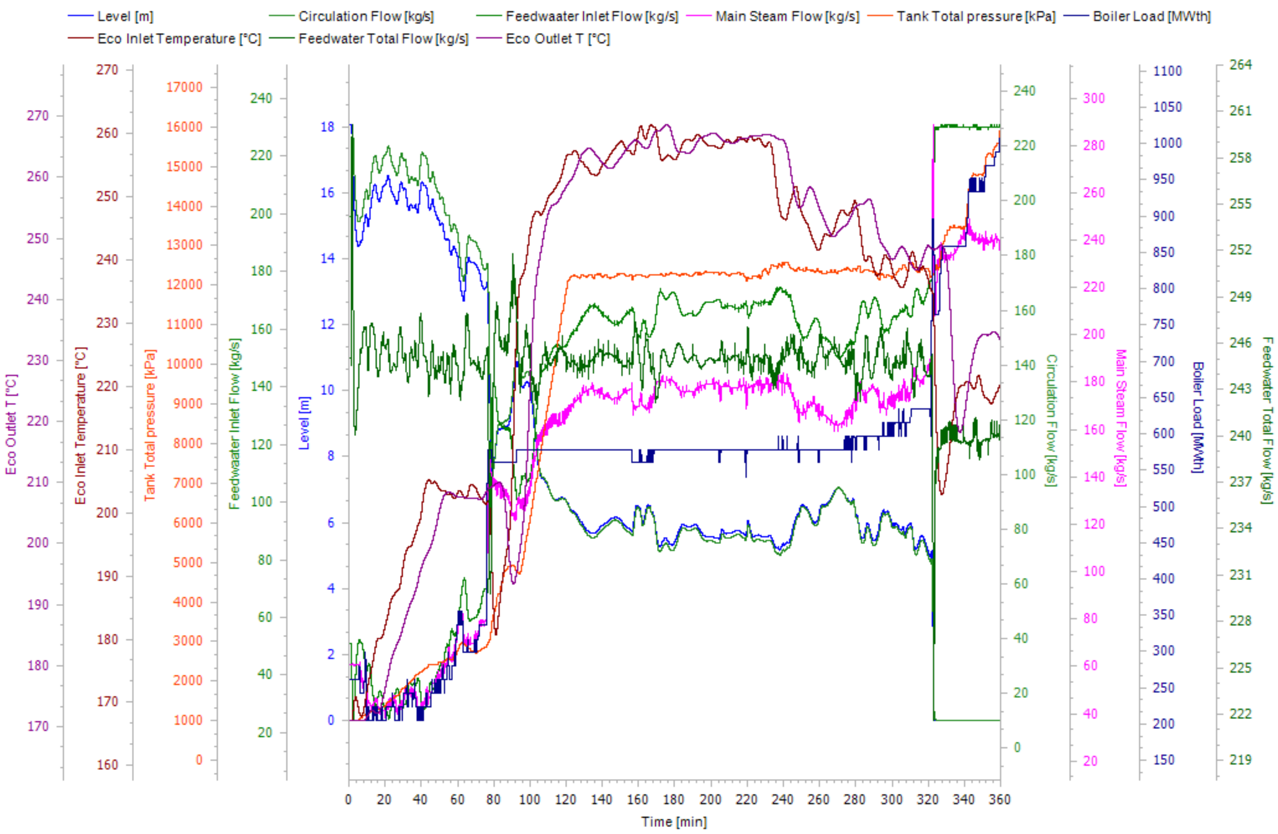


Figure 64: Cold start-up modelling results

5.4 Proposed new control strategy

In theory, controlling the vessel level does seem to make practical sense, and must be controlled to prevent too high level which could overflow the vessel and carry over water into the superheater and lead to quenching incidents. If the level is too low, the boiler circulation pump will trip to prevent cavitation and steam entering the suction side of the pump. Analysing the results from modelling and actual system behaviour reveals that the level control seems to be the source of unstable behaviour. The steady state fundamentals show that the level in the vessel is independent of the circulation flow, although in the dynamic/transient case the collecting vessel level immediately changes with a change in the circulation flow rate as the mass balance changes instantly. The circulation flow rate change, changes the energy balance and the consequences of changing the energy balance affects the level between 6 and 7 minutes after the flow rate change, causing a large period (low frequency) unstable condition. Additionally, the level is subject to high frequency unstable behaviour which is not displayed in some of the modelling results. This is attributed to both the control valve nonlinearities and associated delays as well as possible measurement inaccuracy from the deferential pressure measurement used to calculate the vessel level. With the circulation control valve acting directly on the changes in level, the feedwater controller then has to correct for the changes in circulation flow, causing flow disturbances on the economiser inlet and ultimately back to the collecting vessel. Changes in flow rates can also not happen in isolation as the pressure (momentum balance) also changes.

For these reasons it does not make sense to control the level in the vessel with circulation flow, as the system becomes unstable during transients and introduces 'slugs' of warmer and colder feedwater into the system which disturbs the level further. Not only is the level affected, but also the main steam flow rate and the differential temperature of thick walled components, which is not ideal and could be improved. From analysis of the results it is hypothesized that a more stable circulation flow rate should result in the whole system becoming more stable.

A new control strategy was proposed and tested. The strategy is based on the steady state mass balance of the larger control volume presented in section 3.4:

$$\dot{m}_{bcp} = \dot{m}_{fwt} + \dot{m}_{NPSH} - \dot{m}_{ms} \tag{19}$$

Equation *(19)* presents a very simple, first principle approach for what the circulation flow should be able to achieve in a steady state condition. Thus, equation *(19)* can be used to create a set-point for the circulation flow: the circulation flow $\dot{m}_{bcp}$ equals the measured feedwater total flow ($\dot{m}_{fwt}$) plus NPSH injection water flow ($\dot{m}_{NPSH}$) less the main steam mass flow ($\dot{m}_{ms}$). To prevent overflowing from high level and protection for low flow, an offset is added to equation *(19)* which will add 15 [kg/s] to the circulation flow set-point if the level is higher than 12 [m], and removes 15 [kg/s] if the level is lower than 6 [m], these are arbitrary values which can be optimized. The new proposed control was then implemented and tested in the model using the same real plant data inputs for a mill trip scenario and the results are presented in Figure 65. The level control previously discussed and presented is further referred to as legacy control, with the control based equation *(19)* simply as new control. At first glance the results seem unstable. However, the axis on the graphs are not in a fixed format, but is floating with the change in values, thus the changes in values need to be viewed in perspective relative to the range of the axis.

With the boiler fuel suddenly reduced, as before, a large reduction and main steam mass flow and increase of the vessel level are observed. The circulation flow is increased in proportion to the reducing main steam flow, with additional circulation when the level is above 12 [m]. The offset function removal of circulation flow rate can be seen between 28 [min] and 35 [min], and again at 55 [min] to 60 [min] and 83 [min] to 87 [min].

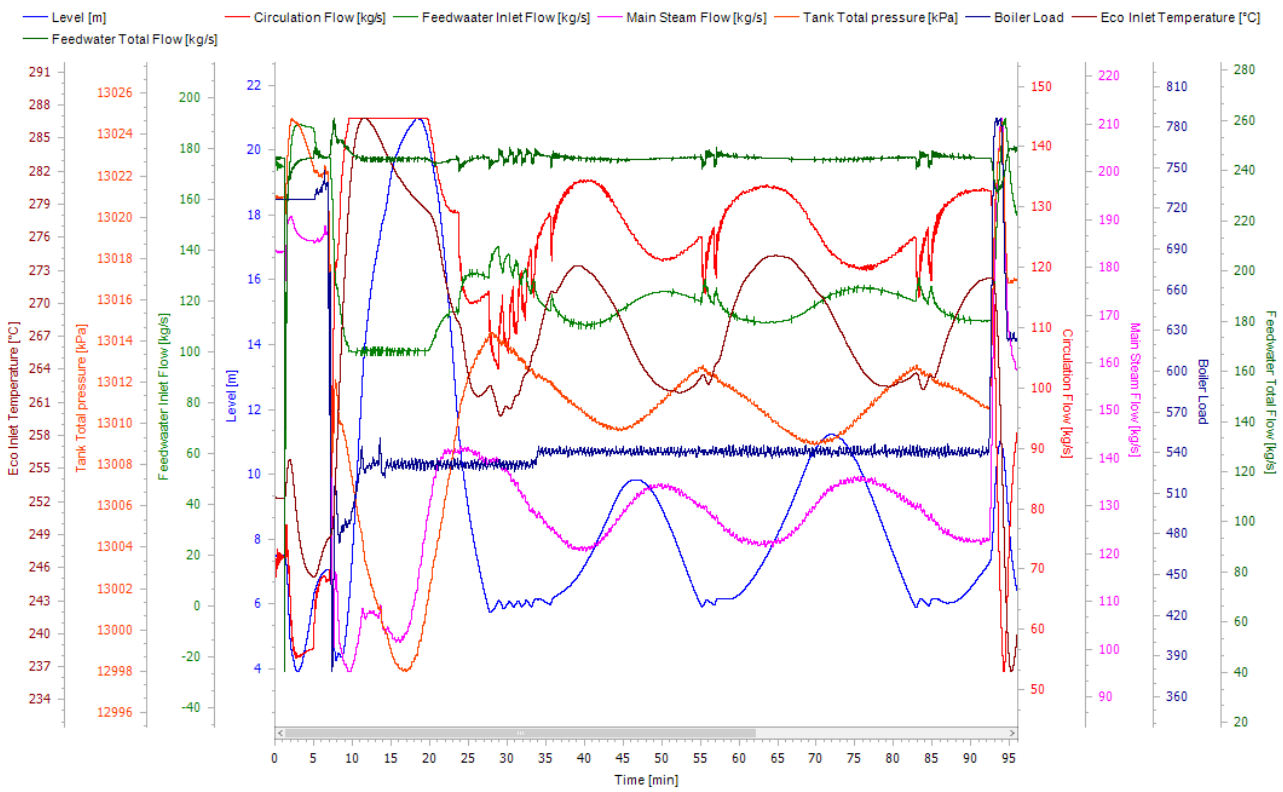


Figure 65: New control response, using actual plant data during mill trip

As the same data was used for the legacy level control, these results can be directly compared to the previous results of Figure 62 with regards to economiser inlet temperature. The economiser inlet temperature cycling is significantly reduced with the new control philosophy as displayed in Figure 66.

With the legacy control the absolute maximum differential temperature after the load disturbance is 70 [K], and cycled after the disturbance at a differential of 50 [K]. With the new control the maximum temperature differential after the disturbance is only 36 [K], and cycling with only 14 [K].

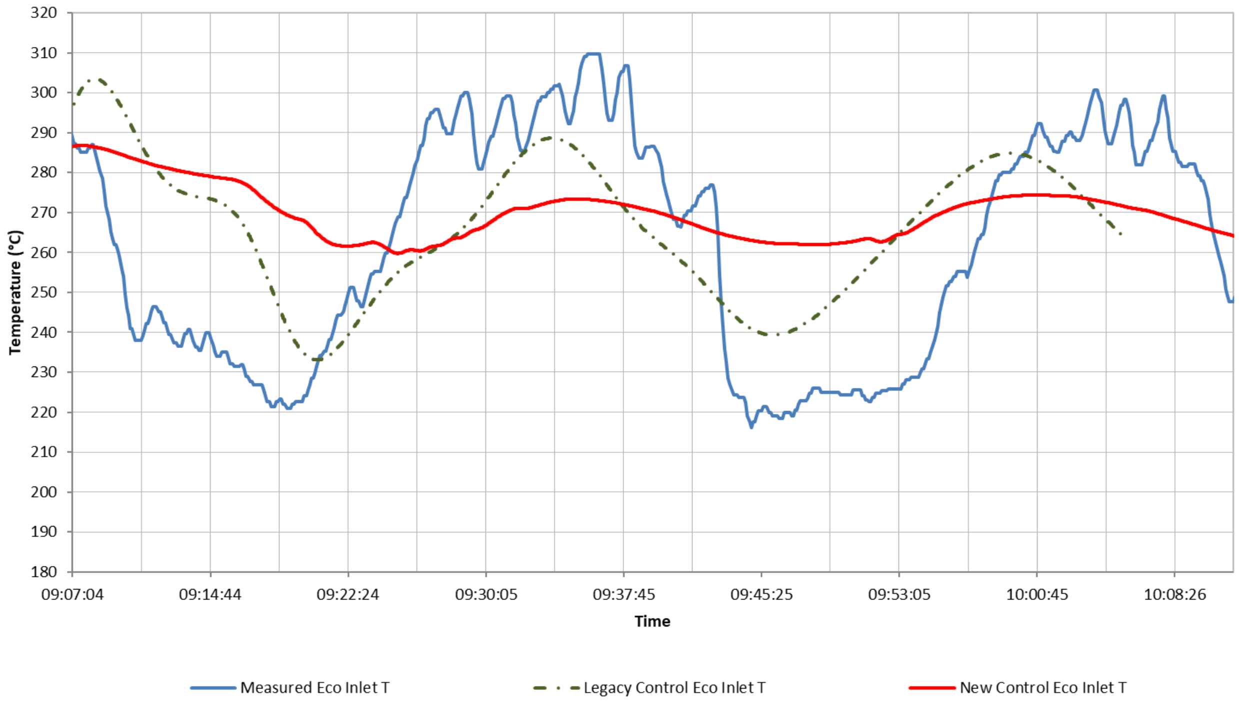


Figure 66: Economiser inlet temperatures comparison

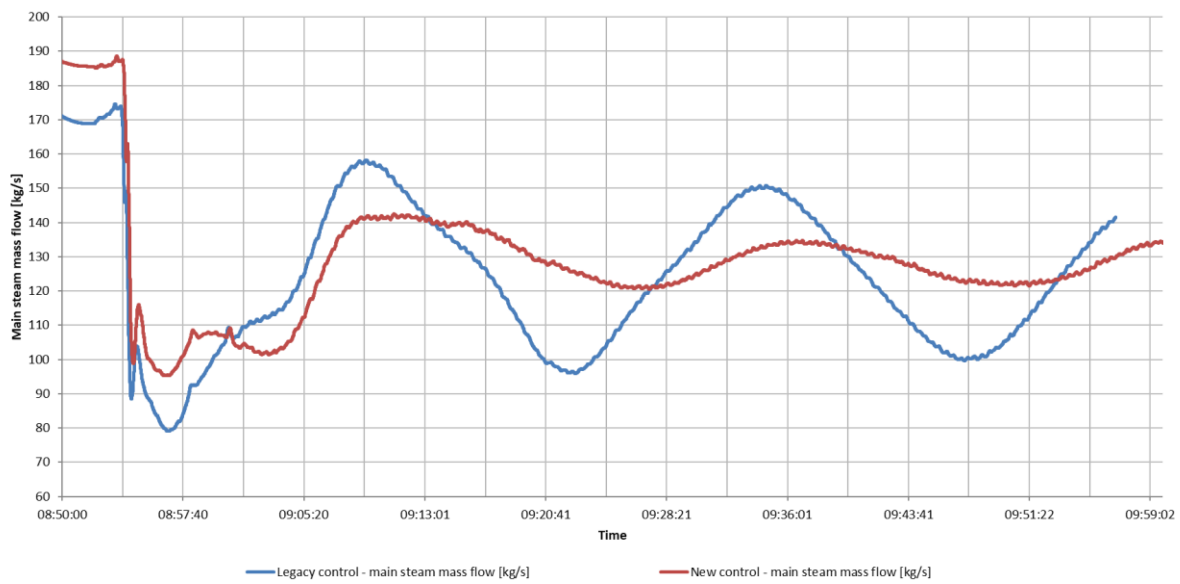


Figure 67: Main steam mass flow comparison

Figure 67 displays the results of the main steam mass flows produced by the models with the legacy control (blue) and new control (red). Due to the different control actions before the load disturbance, the main steam mass flows are not the same, however the initial process response is the same with the main steam flow reducing significantly, and increasing after the disturbance. With the legacy control, the main stem mass flow cycles with almost the same period as the economiser inlet temperature. The legacy control results in a cyclic main steam mass flow with an average amplitude of 56 [kg/s], with a maximum differential after the disturbance of 77 [kg/s]. The new control cycle average amplitude is only 17 [kg/s], with a maximum differential after the load disturbance of 47 [kg/s].

The new control was also simulated for a load increase similar to what was presented for the legacy control in Figure 54. As can be seen in Figure 68 the overall cycling is reduced, with the main steam mass flow also more stable after the initial increase due to the boiler load increase.

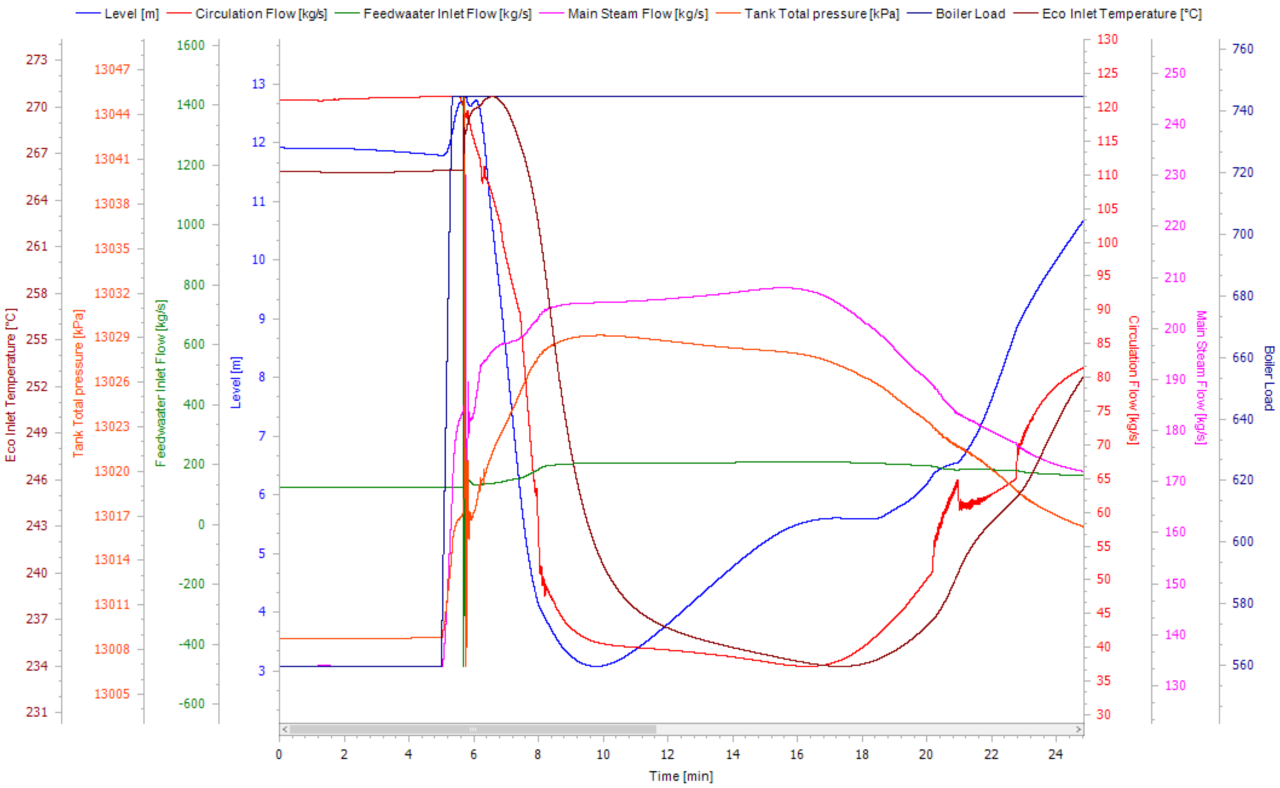


Figure 68: New control response for a load increase from 30% to 40% BMCR

6. Real plant implementation

During this study, there was an opportunity to test the proposed new control on a unit that is in the commissioning phase. The changes were implemented and tested, and the initial results showed significant improvement of the control performance and controllability of the system. Figure 69 and Figure 70 display the measurement results for an hour after a mill is tripped for the total inlet feedwater flow as well as the economiser inlet temperatures. Figure 69 is the system response before the control system change (legacy control) and Figure 70 of the new control. Both graphs have the scaling as well as time scale shortly after a mill is lost. With the new control the economiser inlet temperature cycling (red) is reduced and the feedwater total flow (blue) cycling is significantly reduced. One of the most challenging starts of a once-through boiler is a hot start, as large temperature differentials are experienced as colder feedwater is introduced into hot components. The new control strategy was also tested for hot start conditions and the results compared to the legacy control for a hot start. Figure 71 displays the economiser inlet temperature variations experienced with the legacy controls and recorded a maximum differential temperature of 162 [K]. Figure 72, with the new control, indicates less temperature variations with a maximum differential temperature of 108 [K].

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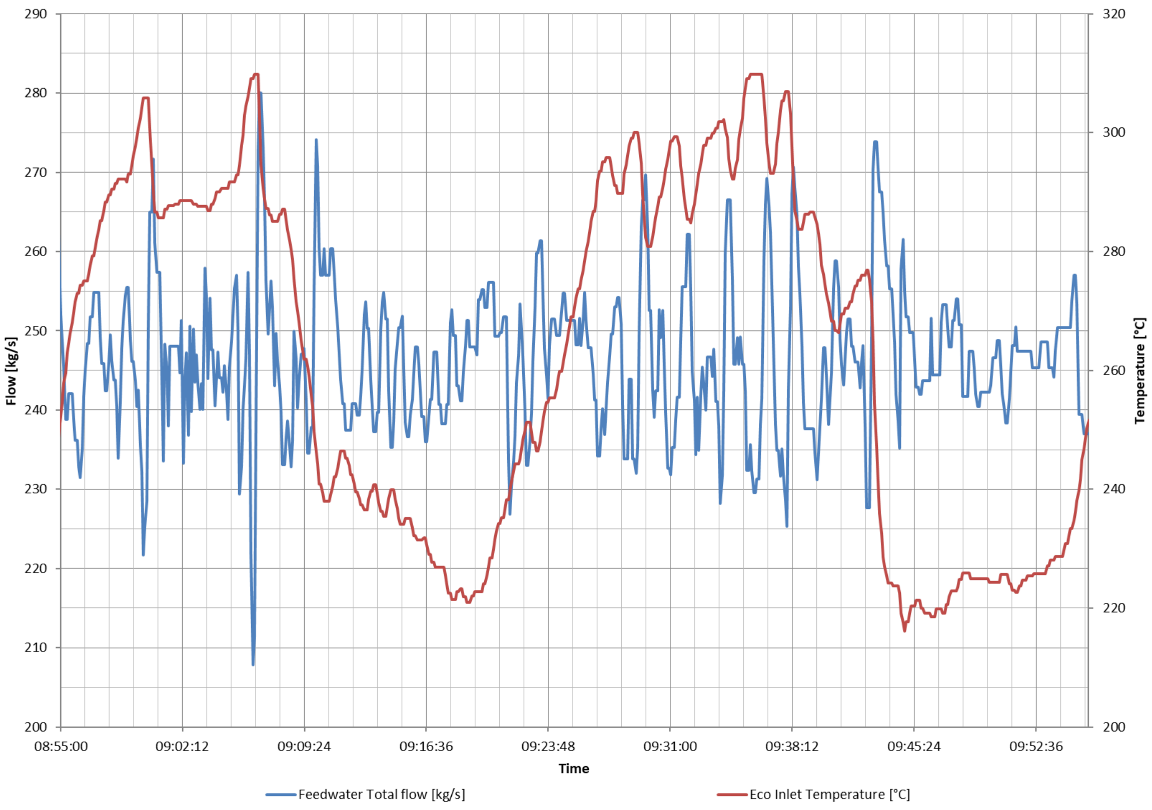

Figure 69: Legacy control feedwater flow and economiser inlet temperature after a mill trip

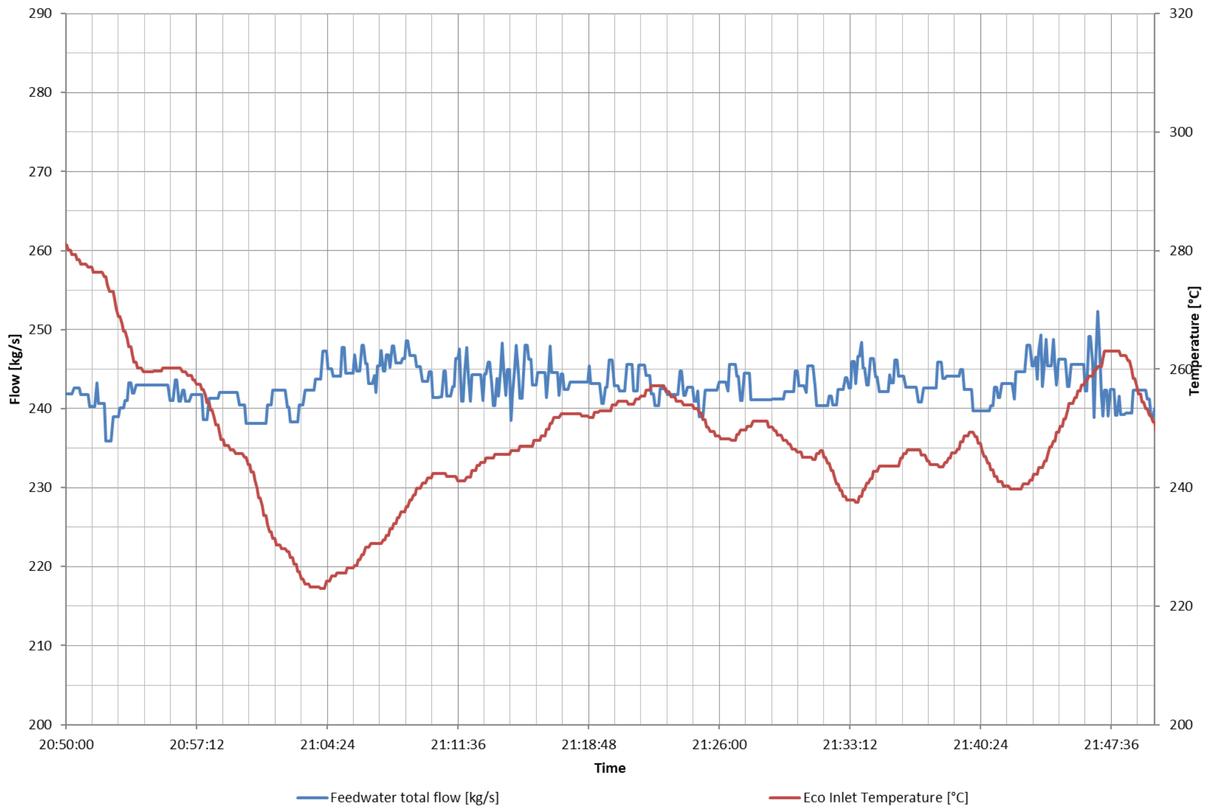

Figure 70: New control feedwater flow and economiser inlet temperature after a mill trip

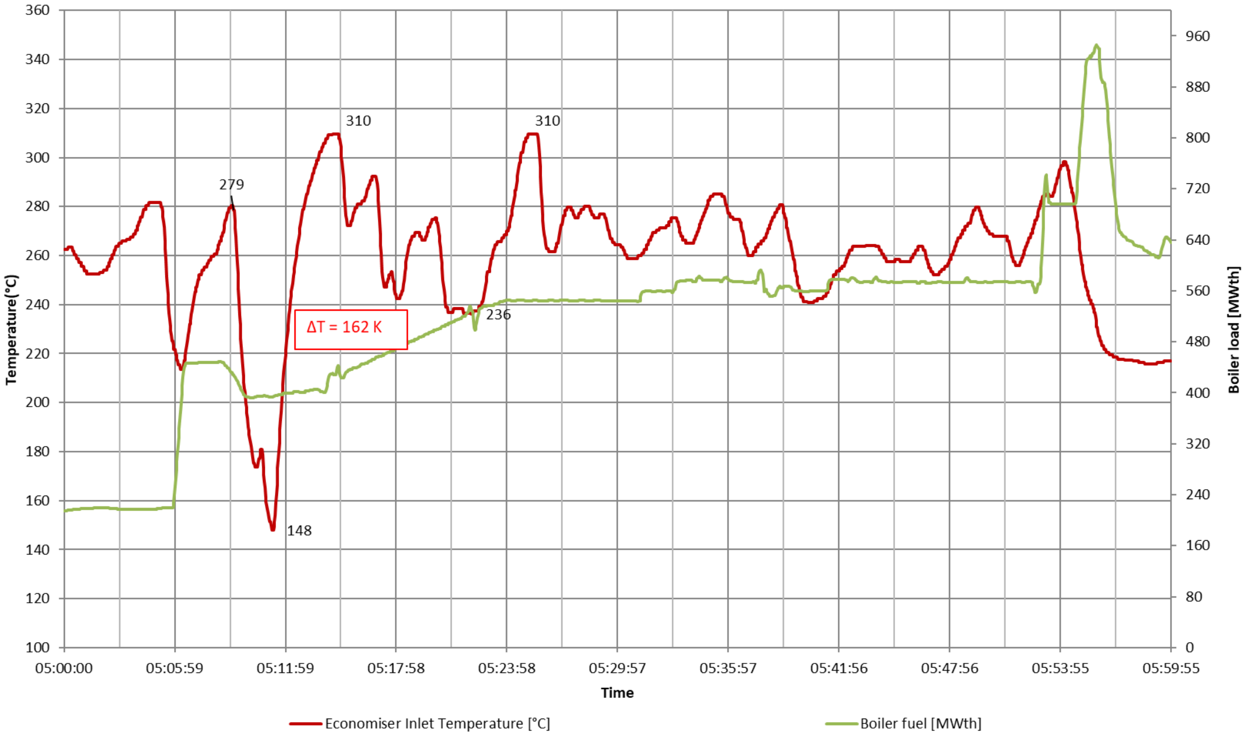

Figure 71: Legacy control, Economiser inlet temperature and boiler load during hot start

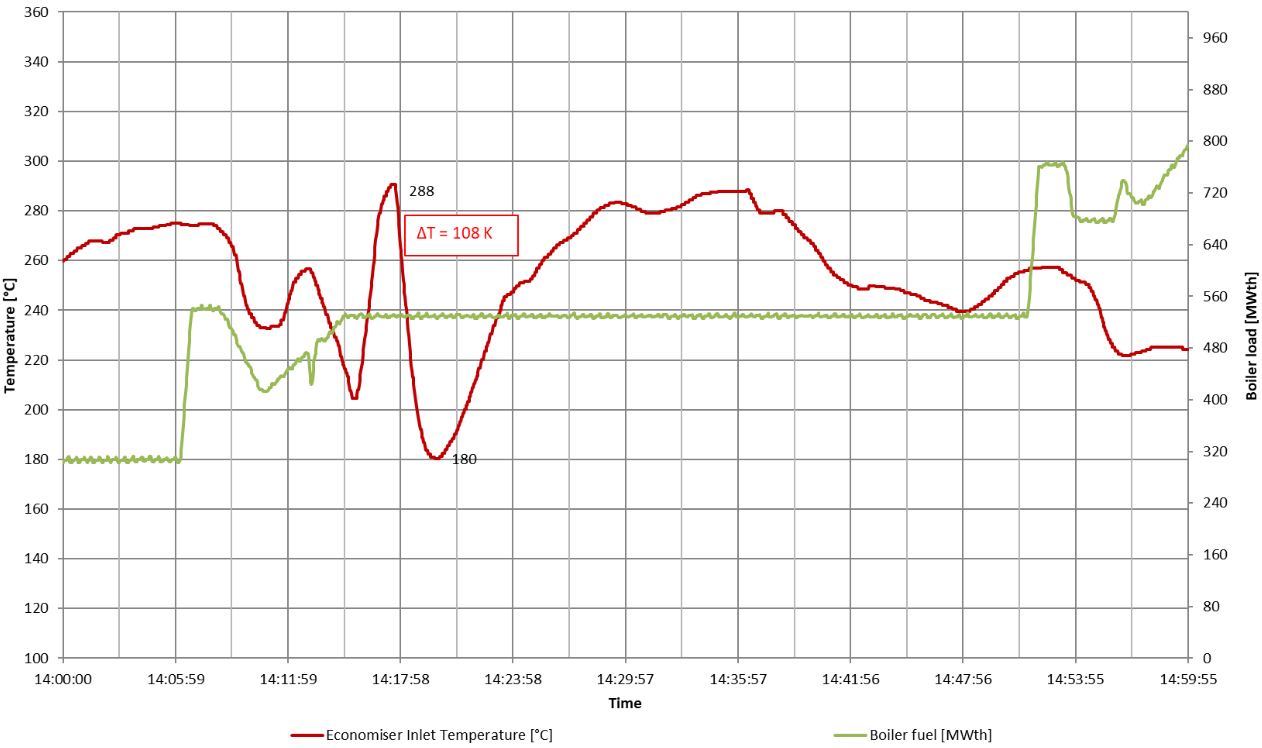

Figure 72: New control, Economiser inlet temperature and boiler load during hot start

7. Conclusions and Recommendations

7.1 Conclusions

In this study an integrated process model of a once-through boiler start-up forced recirculation system together with its controls was developed in Flownex. The model allows analysis of the operational procedures applied during low load operation, start-up, shut down and transients such as mill trips, load increases and loss of high pressure heaters. The model is robust and can simulate a vast array of power plant anomalies and the control behaviour during these events. The effects of single parameter changes can be simulated and analysed, helping to gain a deeper understanding of the process behaviour without interference from the control system. Real time plant data can be imported to the model which can greatly assist in root-cause identification of tripping or extreme events. The model is also able to simulate the control system with sufficient accuracy and changes to the control narrative can be implemented and tested with relative ease. The model can also be used to set up initial control variables.

Studying the simulation results, as well as the plant measurement data, indicated that the start-up system is never stable after the addition or removal of heat. Changes in the collecting vessel level causes (by control design) large changes to the economiser inlet temperature. This results in a delayed change to the evaporator exit quality, main steam mass flow and associated pressure in the vessel. In addition, controlling the vessel level with circulation flow results in large changes in the total feedwater flow. This must be corrected by the feedwater controller, which has undesirable cycling during stable loads, as well as underperformance during transients, which require complicated modifications to prevent tripping incidents.

It has been shown that changing the circulation flow control from a level based approach to a mass balance approach offers significant improvements in the control performance. It reduces potentially damaging thermal transients and also significantly reduces main steam mass flow and feedwater total flow excursions and cycling. The overall controllability of the start-up system is thus greatly improved, thereby reducing trip risks during the start-up of the once-through boiler. The model can thus provide utilities with potential savings through commissioning and operation of once-through units. The model and associated control narrative changes offer reduced trip risks during start-up, less thermal fatigue damage, less demineralised water wastage and reduced risks of turbine quenching. Perhaps one of the biggest benefits to utilities is that more stable operation is possible below the 'Benson point'. This creates the opportunity for better load following capabilities of once-through boilers as stable load conditions can be achieved below the 'Benson point' as shown with the revised control strategy.

In conclusion, gaining a deeper understanding of the process behaviour and translating the process understanding into a reliable process model has led to notable improvements in control performance.

7.2 Recommendations

The furnace heat transfer analysis of the evaporator that is used in the model for this study can be improved, whereby the heat transfer was linearly scaled from full load values. Recommended improvements include the different heat transfer characteristics for different mill configurations, which has the potential to improve the recommended mill configurations that can be used for optimal start-up.

The model can be expanded to include the superheater section, which can potentially offer new insights and control philosophies for enthalpy controllers which are activated after the Benson point. This offers potential improvements of main steam temperature control.

Accuracy of the model can also be improved by implementing the superheater section and downstream pressure control, as opposed to fixed pressure boundary condition at the inlet of the superheater section. This should improve model accuracy as the compressibility effects would be included as well as the transit time to the main steam mass flow measurement.

A study can be conducted into the accuracy of level measurement. As the measurement of level is calculated using the differential pressure, the 'wet leg'/reference leg can be modelled and the calculated level compared to the level calculated by the model. Further study can also be conducted into the level dynamics by using control valves in the simulations. It is believed that by introducing these nonlinearities, more accurate modelling results can also be obtained for the high frequency fluctuations of level and circulation flow observed on the plant.

Another potential improvement to the control narrative is perhaps controlling the circulation flow based on a function derived from the model. For the given conditions of pressure, inlet temperature and load level the function could predict the idealised circulation flow rate within the bounds of the calculated flow rate set point derived from equation (16). This potential improvement could unfortunately not be developed and tested as part of the current study due to time constraints.

A recommendation to Flownex developers: improvements to the linegraph tracking during transient simulations – a optional second cursor (vertical) could be a valuable tool to aid with analysis of graphical data.

Lastly, it is recommended that further studies be conducted at other once-through operating units to determine the feasibility of rolling out the new control strategy to the wider Eskom fleet.

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Appendix A. Functions used

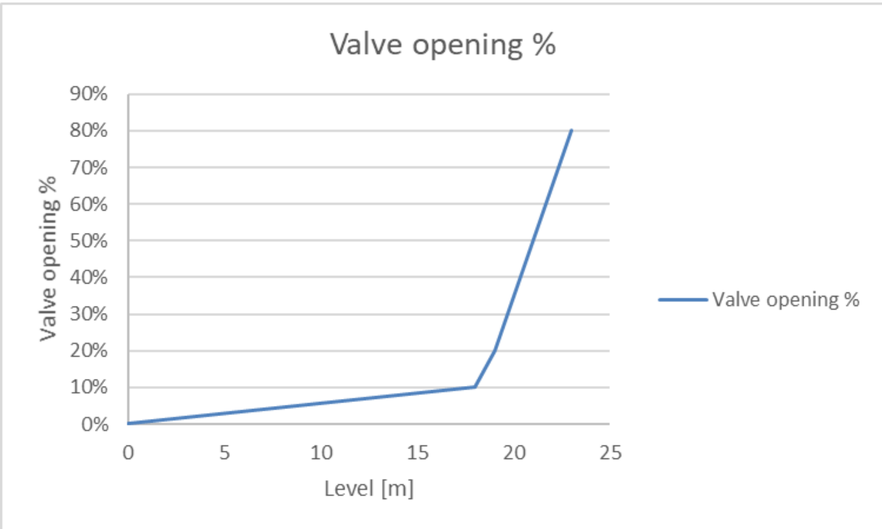
(i) Boiler Heat Addition

Boiler Heating Data		
Boiler section		% Heat Addition as a fraction of total
ECO	ECONOMISER	4.72%
EVAP	Evap 1- HOPPER	1.49%
	Evap 2- HELICAL WALL	29.59%
	Evap 3- HELICAL WALL	10.62%

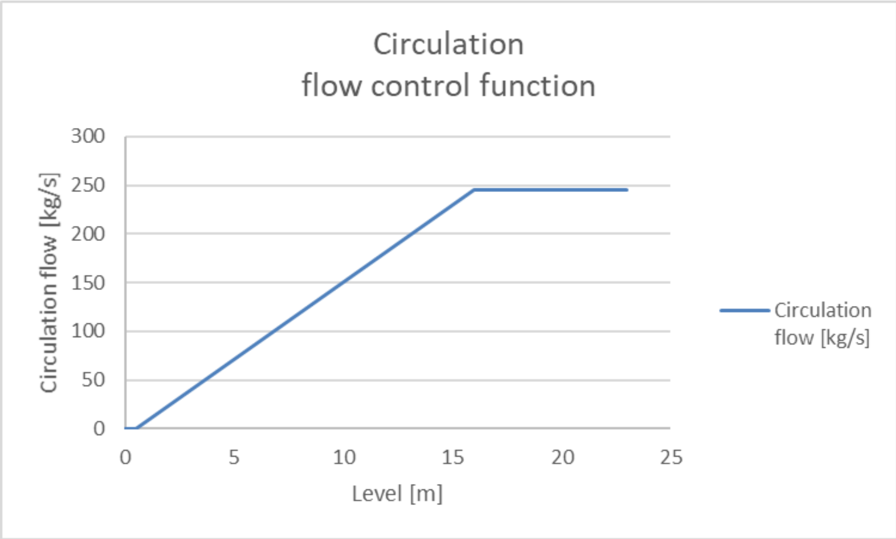
(ii) Tube Geometry

Boiler Part	HEAT	Number of Tubes	Length	Tube Diameter	thickn	Tube ID
ECONOMISER	Convection	1344	65.85	44.5	6.3	31.9
Evap 1- HOPPER	Radiation	436	35	38	6.3	25.4
Evap 2- HELICAL WALL	Radiation	436	110	38	5.6	26.8
Evap 3- HELICAL WALL	Radiation	436	55.4	44.5	7.1	30.3

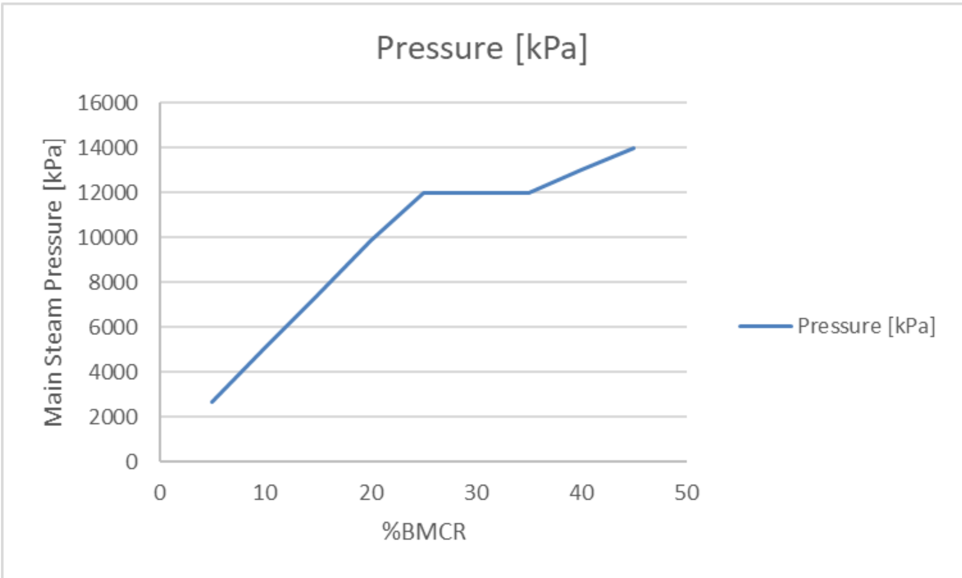
(iii) Blow down valve function generator



(iv) Collecting vessel level control function



(v) Idealised/design boiler pressure build up (cold start)



(vi) Idealised/design feedwater temperature increase

% BMCR	Feedwater Inlet Temperature [°C]
5	130
10	130
15	130
20	130
25	152.5
30	175
35	197.5
40	220
45	220
50	230
55	235