ISRN: LUTMDN/TMHP-19/5437-SE
ISSN: 0282-1990

LUNDS UNIVERSITET

Derivation of a correlation to determine a combined cycle's performance based on data from the gas turbine exhaust

Mattias Ingelström

Thesis for the degree of Master of Science in Engineering

Division of Thermal Power Engineering
Department of Energy Sciences
Faculty of Engineering | Lund University

Derivation of a correlation to determine a combined cycle's performance based on data from the gas turbine exhaust

Mattias Ingelström

June 2019, Lund

This degree project for the degree of Master of Science in Engineering has been conducted at the Division of Thermal Power Engineering, Department of Energy Sciences, Faculty of Engineering, Lund University and at Siemens Industrial Turbomachinery AB (SIT AB).

Supervisor at the Division of Thermal Power Engineering was Professor Magnus Genrup.
Supervisor at SIT AB was Dr. Klas Jonshagen.
Examiner at Lund University was Professor Jens Klingmann.

The project was carried out in cooperation with Siemens.

Thesis for the Degree of Master of Science in Engineering

ISRN LUTMDN/TMHP-19/5437-SE

ISSN 0282-1990

Division of Thermal Power Engineering

Department of Energy Sciences

Faculty of Engineering, Lund University

Box 118, 221 00 Lund

Sweden

www.energy.lth.se

"- We do have a lot in common. The same earth, the same air, the same sky. Maybe if we started looking at what's the same, instead of looking at what's different, well, who knows?"

Meowth

From the movie *Pokemon: The movie*

Abstract

Power production has a vital role in the functioning of society and will have a major part in the future of environmental and economical sustainability. Therefore, efficiencies of power plants are always a subject to improvements in order to utilise resources as efficient as possible. A Combined Cycle, combining a gas turbine topping cycle and a steam turbine bottoming cycle, is a widely used configuration for producing electricity at a high efficiency. It is not enough to have an understanding of the improvements on the efficiencies of both cycles individually, but also the interaction between them. When a gas turbine is to be chosen in a design process, for a new power plant or for an upgrade to an existing one, it is difficult to make an accurate and instantaneous estimation of the Combined Cycle efficiency. This report presents a method to estimate the efficiency down to a thousandth of a percent for a selection of constellations of Combined Cycles without any complex calculation processes. *The second law efficiency* proved to be the key to correlate changes of parameters resulting in a powerful method that will make accurate predictions of the efficiency for a wide set of power plants.

Keywords: Combined Cycle efficiency, second law efficiency, HRSG, exergy, irreversibilities

Sammanfattning

Elproduktion har en livsviktig roll i samhällets funktion och kommer ha en stor del i framtiden för den miljömässiga och ekonomiska hållbarheten. Därför kommer kraftverks verkningsgrader alltid att behöva förbättras för att använda resurser så effektivt som möjligt. Ett kombikraftverk, som kombinerar en gasturbin-toppcykel och en ångturbin-bottencykel, är en vida använd konfiguration för att producera el med en hög verkningsgrad. Det räcker inte att enbart ha en förståelse för förbättringarna av de båda individuella cyklerna, utan också interaktionen mellan dem. När en gasturbin ska väljas i en designprocess, för ett nytt kraftverk eller till en uppgradering av ett befintligt, är det svårt att noggrant och omedelbart bestämma kombikraftverkets verkningsgrad. Denna rapporten framför en metod för att, för ett urval av konstellationer av ett kombikraftverk, uppskatta verkningsgraden med en noggrannhet av en tusendels procent utan komplexa beräkningsprocesser. Nyckeln till att korrelera parameterförändringar visade sig vara *the second law efficiency* och resultatet blev en kraftfull metod som noggrant förutsäger verkningsgrader för en bred uppsättning av kraftverk.

Nyckelord: Kombikraftverks verkningsgrad, second law efficiency, rökgaspanna, exergi, irreversibiliteter

Acknowledgements

I would like to express my gratitude to the people that have made this thesis possible through guidance and support.

First of all I would like to thank my supervisor at SIT, Dr. Klas Jonshagen, for all the support and help that I have gotten throughout the thesis. You have been a fantastic resource with immense knowledge and has always made sure that I understand what I need to complete this study.

Also, a big thank you to the whole performance department at SIT as well as all of the other thesis people that has made this spring a fantastic and very enjoyable journey. A special mention to Teresa, thank you for all interesting discussions and support throughout the thesis.

I would also like to thank my supervisor at LTH, Prof. Magnus Genrup, for all the help throughout the thesis and for all the support during my studies. Thank you for contributing to the possibility of me getting summer jobs and this thesis helping me get a great end of my studies and a great start of my working career.

Last but not least, thank you to my family, friends and especially my Iris for the never-ending love and support throughout my studies.

Nomenclature

SI-units are used throughout this thesis except for the temperature where degrees Celsius is also used for clarity.

Roman Letters

c_p	Specific heat capacity	[kJ kg^{-1} K^{-1}]
h	Specific enthalpy	[kJ K^{-1}]
h_{exh}	Specific enthalpy of the exhaust	[kJ K^{-1}]
h_{mean}	Mean specific enthalpy	[kJ K^{-1}]
h_{stack}	Specific enthalpy of the stack	[kJ K^{-1}]
$\dot{m}_{\mathrm{exh}}$	Exhaust gas mass flow	[kg s^{-1}]
$\dot{P}_{\mathrm{BTC}}$	Power of BTC	[kJ s^{-1}]
$\dot{P}_{\mathrm{RBC}}$	Power of RBC	[kJ s^{-1}]
Q	Heat	[kJ]
$\mathrm{Q_H}$	Heat supply	[kJ]
$\dot{Q}_{\mathrm{in}}$	Rate of heat input	[kJ s^{-1}]
$\mathrm{Q_L}$	Heat sink	[kJ]
s	Specific entropy	[kJ kg^{-1} K^{-1}]
S	Entropy	[kJ K^{-1}]
T	Temperature	[K - °C]
$\bar{T}$	Mean temperature of heat addition	[K - °C]
T_{cond}	Condenser temperature	[K - °C]
T_{exh}	Exhaust gas temperature	[K - °C]
T_{L}	Low temperature	[K - °C]
T_{H}	High temperature	[K - °C]
W	Work	[kJ]

Greek Letters

η	Efficiency	[$-$]
η_{th}	Thermal efficiency	[$-$]
η_{2nd}	Second law efficiency	[$-$]
η_{BC}	Brayton Cycle efficiency	[$-$]
η_{BTC}	Brayton Topping Cycle efficiency	[$-$]

η_{carnot}	Carnot efficiency	$[-]$
η_{CC}	Combined Cycle efficiency	$[-]$
η_{HRSG}	HRSG efficiency	$[-]$
η_{RBC}	Rankine Bottoming Cycle efficiency	$[-]$
η_{RC}	Rankine Cycle efficiency	$[-]$

Abbreviations

BC	Brayton Cycle
BTC	Brayton Topping Cycle
CC	Combined Cycle
GT	Gas Turbine
HP	High Pressure
HRSG	Heat Recovery Steam Generator
IP	Intermediate Pressure
LP	Low Pressure
RBC	Rankine Bottoming Cycle
RC	Rankine Cycle
ST	Steam Turbine
TIT	Turbine Inlet Temperature

Contents

Abstract . . . i
Sammanfattning . . . ii
Acknowledgements . . . iii
Nomenclature . . . iv

1 Introduction **1**
1.1 Background . . . 1
1.2 Objective . . . 1
1.3 Delimitations . . . 2
1.4 Literature study and previous work . . . 2
1.5 Tools . . . 4
1.5.1 IPSEpro . . . 4
1.5.2 Matlab . . . 4

2 Theory **5**
2.1 Heat engine . . . 5
2.2 Thermodynamic cycles . . . 5
2.3 Brayton Cycle . . . 6
2.4 Rankine Cycle . . . 7
2.5 Exergy and irreversibilities . . . 8
2.6 Carnot cycle . . . 9
2.6.1 Step 1 - Isothermal expansion . . . 10
2.6.2 Step 2 - Isentropic expansion . . . 10
2.6.3 Step 3 - Isothermal compression . . . 10
2.6.4 Step 4 - Isentropic compression . . . 11
2.7 Cycle efficiencies . . . 11
2.7.1 Carnot efficiency . . . 11
2.7.2 First law efficiency . . . 11
2.7.3 Second law efficiency . . . 12
2.8 Combined cycle . . . 12
2.9 HRSG . . . 13
2.9.1 Exhaust temperature . . . 14
2.9.2 Live steam pressure . . . 15
2.9.3 Pressure levels . . . 16
2.9.4 Deaerator . . . 17
2.10 Interaction between top and bottom cycle . . . 18
2.11 Mean temperature of heat addition . . . 20
2.11.1 Distribution of mass flow . . . 20
2.11.2 Effect on the efficiency . . . 21
2.11.3 Estimation error . . . 22

3 Methodology **24**
3.1 Modelling . . . 24
3.1.1 Addition of deaerator . . . 25
3.1.2 Two pressure levels . . . 25
3.1.3 Applying constraint on steam turbine . . . 26
3.1.4 Three pressure levels . . . 26
3.2 Derivation of expression . . . 26

3.3 Correlation study: Two pressure levels with no constraint on the steam turbine . . . 28
3.3.1 Correction of T_H . . . 29
3.3.2 Optimising LP at a given HP . . . 31
3.3.3 Correction on thermodynamic state at stack . . . 34
3.4 Correlation study: Two pressure levels with constraint on the steam turbine . . . 35
3.4.1 Correction of T_H . . . 36
3.4.2 Correction of h_{stack} . . . 37
3.5 Correlation study: Two pressure levels with constraint on the steam turbine during off-design . . . 37
3.5.1 Correction of T_H with mass flow variation . . . 38
3.5.2 Correction of h_{stack} with mass flow variation . . . 39
3.5.3 Correction of T_H with exhaust temperature variation . . . 40
3.5.4 Correction of h_{stack} with exhaust temperature variation . . . 41
3.6 Correlation study: Three pressure levels with constraint on the steam turbine . . . 41
3.7 Correlation study: State-of-the-art power plant . . . 43

4 Results **45**
4.1 Two pressure levels with no constraint on the steam turbine . . . 45
4.2 Two pressure levels with constraint on the steam turbine . . . 48
4.3 Two pressure levels with constraint on the steam turbine during off-design . . . 50
4.3.1 Variation in mass flow of exhaust gas . . . 50
4.3.2 Variation in temperature of exhaust gas . . . 53
4.4 Three pressure levels with constraint on the steam turbine . . . 55
4.5 State-of-the-art power plant . . . 58
4.5.1 First order correction . . . 58
4.5.2 Second order correction . . . 60
4.6 Using corrections for different pressure levels . . . 63

5 Discussion **64**
5.1 Approach . . . 64
5.2 Methodology . . . 64
5.3 Results . . . 65

6 Conclusion **67**
6.1 Sources of errors . . . 67
6.2 Future work . . . 68

7 References **69**

1 Introduction

Ever since the cannon manufacturing in the 16^{th} century, the town Finspång has been a key location of the Swedish industry. The production of cannons and ammunition to the Swedish army made the town and region prosper. In early 20^{th} century the Swedish turbine manufacturer STAL moved their production to Finspång. After a number of takeovers and joint ventures the responsibility of manufacturing turbines now lays at Siemens Industrial Turbomachinery AB (SIT AB).

1.1 Background

In a world where focus lies on reaching a sustainable power production, the future role of gas turbines remains uncertain. As of today, having a world supplied with clean energy is not feasible since there is simply not enough installed capacity. Moreover, the reliability of power production from renewable energy sources is inadequate which means that alternative methods are required to maintain a stable grid. Even though the aim should be reaching the environmental goals, fossil fuel will most likely be a part of energy production to some extent, for many years to come.

One of the most efficient ways to produce power today from fossil fuel is to combine two different thermodynamic cycles; the Brayton Cycle (BC) and the Rankine Cycle (RC). This is called a Combined Cycle (CC). By exchanging heat in a Heat Recovery Steam Generator (HRSG) the otherwise wasted energy in the flue gas from the BC is utilised by producing steam in the RC. A steam turbine is then run on this steam to produce more power from energy that otherwise would be released into the surroundings.

The complexity of the combined cycle originates from the number of parameters that affects the heat energy exchange and that the two cycles can react in opposite ways to an alteration. That means that an improvement for the individual cycles could lead to an overall decrease in efficiency.

1.2 Objective

This project aims to reach an expression that will quickly and reliably determine the variation of the efficiency when a parameter has been altered. The basis for this is to give support in understanding how a change in the constellation of a power plant would affect the performance.

By continuing on a previous master thesis which resulted in a set of correction factors this project intends to enhance the usefulness by making the method more adaptable and versatile.

The method will be tried foremost for a scenario which will explore the design process of a power plant. In this scenario, the power plant has not yet been built or a bottoming cycle will be added to an already existing gas turbine. Thereafter, a pre-study will be conducted, using the same method, of a gas turbine being run off-design in an already existing combined cycle. The aim is to complete the study of the design process and lay forward a structure of how to finalise the off-design study.

1.3 Delimitations

The aforementioned complexity of parameters affecting the performance of a combined cycle makes it unrealistic to not restrain the study. Thereof, some limitations were made as follows:

- In real life, in the design stage where the correlations will be applicable, data from the gas cycle will be known. Thus it is appropriate to focus solely on factors that affect the steam cycle.
- The models representing the combined cycle were kept without auxiliary systems. The target was instead set on understanding the effect of the vital components of a power plant that is most likely to affect the efficiency.
- To not make a too wide study, key parameters were kept close to values of real life power plants.

1.4 Literature study and previous work

An extensive research of related studies and theoretical material was conducted in the prelude of the project. Since the topic of combined cycle is vast the search was foremost constricted to books and reports focusing on the HRSG. To reach an advanced understanding of the HRSG is vital for the project in order to make realistic models and scenarios.

As has been mentioned this report is a continued study on a topic that was conducted by Simon Frick [1]. Even if the approach will be different it will still be a valuable asset to review and analyse his work.

For purely theoretical material the course book in thermodynamics, *Thermodynamics: An engineering approach* written by Yunus A. Boles et al. [2], was consulted. Due to the wide use of the book its information is deemed trustworthy and is used when discussing basic principles. Literature concerning thermodynamics in general is abundant and after consultation with both supervisors the most appropriate works could be chosen. *Combined-Cycle Gas & Steam Turbine Power Plants* by Kehlhofer et al. [3] contains an in-depth review of the combined cycle and the book acted as a good stepping stone from the fundamental thermodynamics into the complexity which surrounds a combined cycle. This source also gave an understanding of the different types of systems and their respective benefits and draw-backs. The HRSG was also explained more in detail which made it easier to follow the text by El-Masri [4], where the HRSG is thoroughly analysed. By highlighting the responses of the HRSG when key parameters in a combined cycle are altered, it gave an indication on what parameters should be considered.

As well as books, articles about similar topics were acquired in order to revise previous studies and gather an understanding for how to make this project unique. A second law analysis of a combined cycle is discussed in a paper written by S. Can Gülen et al. [5]. This paper goes into detail of the concept of exergy and different irreversibilities that are present in a combined cycle. In order to get a grasp of the economic implications, a paper regarding the exergy-economic optimisation of a Combined Cycle was studied [6]. It was concluded that there is a dependency on the values of pinch points of the HRSG and the exergy-economic optimum. This originates from the principle of reducing the pinch point to zero which will have a positive impact on the overall efficiency but increase the heat exchanger area to infinity. Using the information gathered from these two papers it was possible to assume reasonable values of key parameters such as the pinch point.

From the literature study it is clear that other than the previous Master's thesis done in this field, no other similar correlation study has been conducted. This report will therefore continue on what has been done and focus on reaching a more general solution. Most references will be used to get an understanding on the study of Heat Recovery Steam Generators and second law analysis.

1.5 Tools

1.5.1 IPSEpro

The simulation tool IPSEpro by SimTech was used when constructing simulation models of the different types of combined cycles. The program solves heat equation balances iteratively. By using a model library of power plant components a system can be easily built. Known parameters are then set in order to solve unknowns. Thus, analysis can be done by varying one parameter at a time. From IPSEpro text-files were exported in order to be post-processed. The addition of PSScript made it possible to automate time-consuming and repetitive tasks.

1.5.2 Matlab

The calculation tool Matlab was used to process the text-files in order to generate results, make plots and tables. From IPSEpro, all parameters names and values are exported in a txt-file. Thus, Matlab was used to extract necessary values in order to determine the performance of each model. This required unique scripts for each model.

2 Theory

2.1 Heat engine

A heat engine is by definition an engine that transforms thermal and chemical energy to mechanical energy. Energy that can later on be used to produce work. A schematic of the process is shown in Figure 2.1.

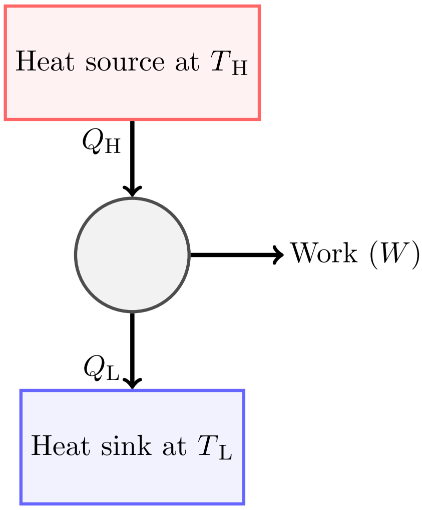

Figure 2.1: Diagram showing the interaction between thermal and mechanical energy.

Due to the temperature difference between the heat source and heat sink, thermal energy is being transferred to a lower thermal state and this transfer is used to drive a thermodynamic cycle. The circle in Figure 2.1 represents a part of the process which utilises the energy transfer in order to produce work. This process is called a thermodynamic cycle and will be discussed in the following section.

2.2 Thermodynamic cycles

In order to produce work in a heat engine a system able to convert thermal energy to mechanical energy is required. This process is called a thermodynamic cycle and can have many appearances. One classification which is used to differentiate between cycles is if they are open or closed. In an open cycle the working fluid passes through the cycle and is often characterised by having a heat addition process where the fluid is mixed with fuel, i.e. an internal combustion. In contrast, the closed cycle's working fluid never leaves the system and is externally heated.

The concept of the combined cycles will, in the coming segments, be discussed briefly. It is important to note that there can be many alterations to these cycles. Therefore, only the basic concepts which are needed are mentioned.

2.3 Brayton Cycle

The Brayton Cycle (BC) is a cycle that most often uses air as a working fluid in an open cycle and consists of a compressor, combustion chamber and a turbine. Ambient air enters the compressor where work is put on the gas which then leaves at a higher pressure and temperature. In the combustion chamber, fuel is added and combusted with the air supply. This raises the temperature before entering the turbine where mechanical energy is extracted. This can then be converted into electrical energy in a generator.

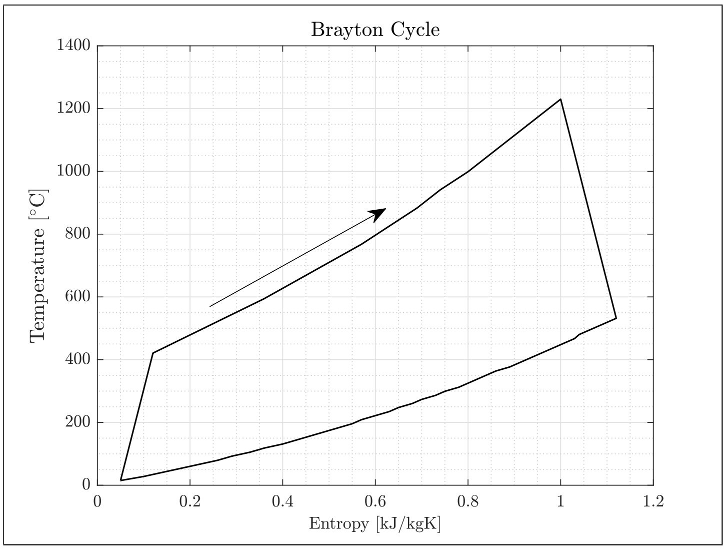

Figure 2.2: Temperature - entropy diagram for a typical Brayton cycle.

Figure 2.2 depicts a Brayton cycle and its four general processes in a temperature diagram. The bottom left corner represents the ambient state and from that point air is compressed and then combusted before entering the turbine. Thereafter, if the cycle is open, air is returned to ambient surroundings.

2.4 Rankine Cycle

The typical appearance of the Rankine Cycle (RC) is a closed cycle with water as a working fluid. Other than that, there are four typical processes; condensation where the water condenses and releases heat, pumping from the condensation pressure to the designated pressure, a constant pressure heat addition process and finally the turbine where work is produced.

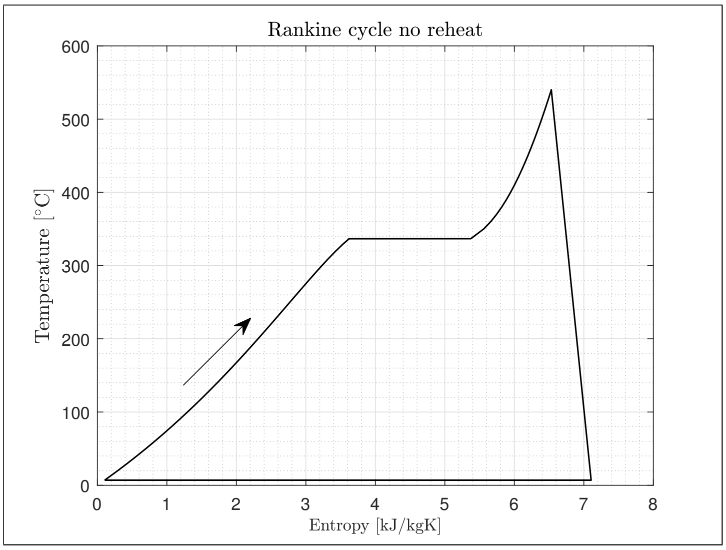

Figure 2.3: Temperature - entropy diagram for a typical Rankine cycle.

The phase change of the fluid yields a characteristic appearance as seen in Figure 2.3. In the bottom left corner water is saturated at the condensation pressure. The pumping process is not visible since this process is rather undemanding. After the fluid is raised to a higher pressure, the isobaric heat addition process takes place and it can take various forms. For instance heat can be added by a nuclear reactor or from combustion of natural gas in a boiler furnace.

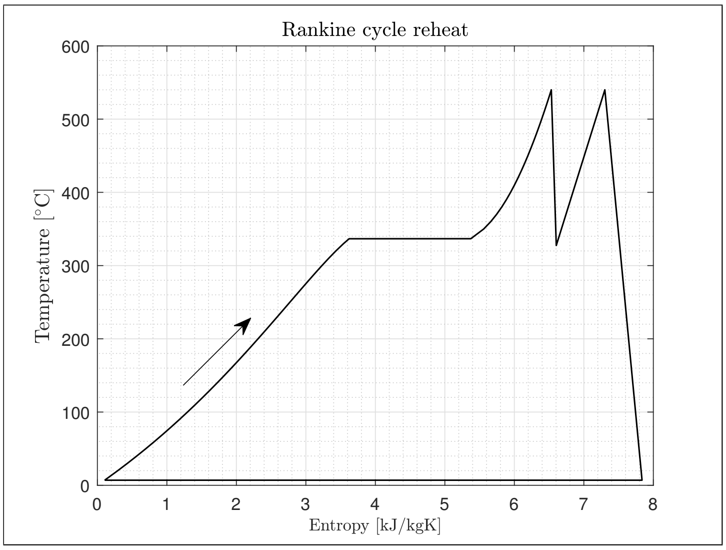

Figure 2.4: Temperature - entropy diagram for a Rankine cycle with a reheating process.

In Figure 2.4 the difference of adding a reheating process is visualised. This will increase the mean temperature of the heat addition process and will increase the efficiency of the cycle due to the steam having greater energy content per mass flow. In section 2.11 the mean temperature of heat addition and its impact on efficiency is discussed in detail.

2.5 Exergy and irreversibilities

The concept of exergy is important when differentiating between the first and second law of thermodynamics. It is a measure of the maximum available energy that can be transferred between two heat reservoirs. Referring to Figure 2.1, this would be the maximum possible energy that could be transferred between the heat source and heat sink. When a process is irreversible, i.e. the process cannot be reverted without adding energy to the system, exergy is destroyed. Thus, measuring irreversibilities is vital when locating components which have a significant contribution to the exergy losses. This information can then be used when optimising the power plant. Examples of irreversibilities are shown in Figure 2.5.

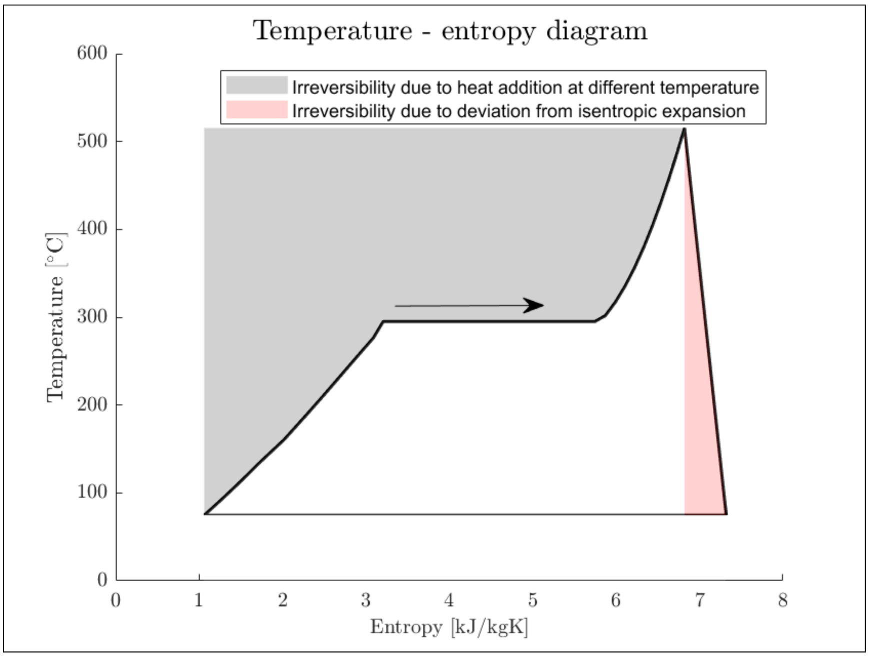

Figure 2.5: Highlighting two examples of irreversibilities in the heat addition and expansion phase in a Rankine Cycle.

From the basic principle of thermodynamics heat travels from a hot source to a cold which means it is not possible to reverse the heat addition coming from the steam cycle to the surrounding hot reservoir. To be able to reach a reversible heat addition it is required to occur at a constant temperature, i.e. at the top of the grey area. This is not a realistic process since the heat transfer area would have to be unreasonably large and would have to go on for a long period of time. The red area represents the deviation from an isentropic expansion that originates from losses. These losses emanate from various irreversible processes such as friction and leakage.

2.6 Carnot cycle

In 1824 the work of French physicist Sadi Carnot resulted in a definition of an ideal thermodynamic cycle that operates between a heat source with temperature T_H and a heat sink with temperature T_L. The Carnot cycle is constructed of four reversible processes. Referring to Figure 2.6 each process can be described.

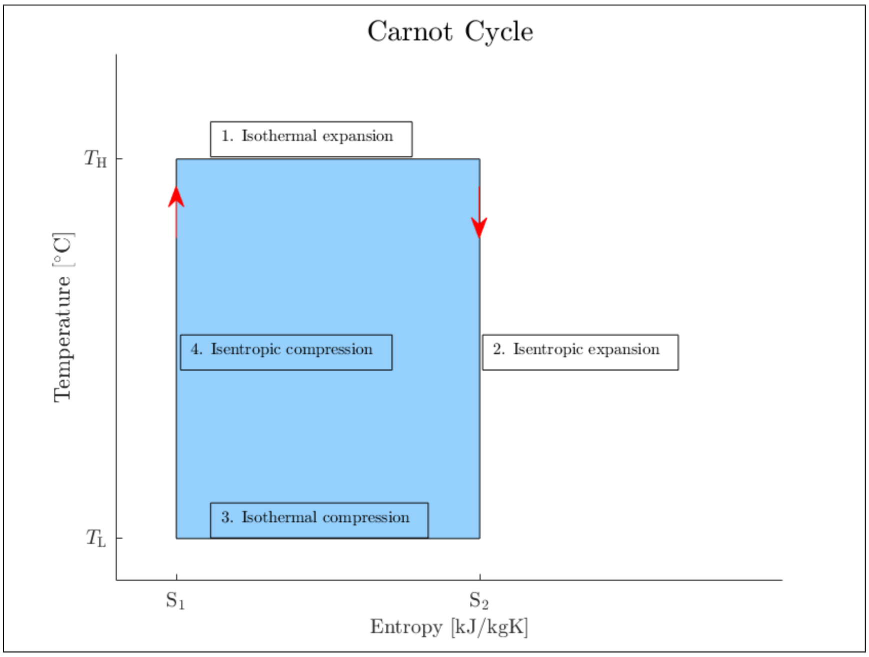

Figure 2.6: The Carnot cycle's four steps in a temperature-entropy diagram.

2.6.1 Step 1 - Isothermal expansion

While in thermal contact with the heat source which is at T_{H} the medium is expanded in a quasi-static process that does not change the internal energy of the system. Thus, the temperature is kept constant. The loss of temperature from the medium's expansion is countered by the addition of the heat from the source.

2.6.2 Step 2 - Isentropic expansion

After the isothermal heat addition the medium is thermally insulated from the heat source and heat sink. Therefore, there is neither heat gain nor loss resulting in an adiabatic process. The medium is expanded by a decrease in pressure until the heat sink temperature T_{L} is reached.

2.6.3 Step 3 - Isothermal compression

The medium is no longer insulated and comes into contact with the heat sink at temperature T_{L}. External work is done on the medium which results in a transfer of heat energy to the surroundings. This is also a quasi-static process, thus no increase in temperature.

2.6.4 Step 4 - Isentropic compression

The medium once again loses contact to the surroundings and is compressed adiabatically raising its temperature to that of the heat source, T_{H}. This step takes the medium to the initial state and the cycle can be repeated.

2.7 Cycle efficiencies

It is of utmost importance to differentiate between the efficiencies determined by the first and second law of thermodynamics. The first law efficiency yields a ratio of power output to the amount of energy put into the system whereas the second law relates the transfer of useful heat to an ideal case. This will be discussed more in detail in the following sections.

2.7.1 Carnot efficiency

This cycle is as stated before an ideal thermodynamic cycle since it is fully reversible. The efficiency, η, of a cycle is known as:

$$\eta = \frac{W}{Q_{\mathrm{H}}} = \frac{Q_{\mathrm{H}} - Q_{\mathrm{L}}}{Q_{\mathrm{H}}} \tag{2.1}$$

Where W is the work done by the system. Q_{H} and Q_{L} are the heat addition and rejection respectively. Furthermore,

$$Q_{\mathrm{H}} = \int_1^2 T dS = T_{\mathrm{H}}\big(S_2 - S_1\big) \tag{2.2}$$

$$Q_{\mathrm{L}} = \int_3^4 T dS = T_{\mathrm{L}}\big(S_2 - S_1\big) \tag{2.3}$$

Combining equations 2.2 and 2.3 with equation 2.1 yields the following expression known as the Carnot efficiency

$$\eta_{carnot} = 1 - \frac{T_{\mathrm{L}}}{T_{\mathrm{H}}} \tag{2.4}$$

2.7.2 First law efficiency

The first law of thermodynamics states that energy can neither be created nor destroyed, only converted to one form from another. Consequently, the first law efficiency is the proportion of converted net power output $\mathrm{W}_{\mathrm{net}}$

from a heat engine that has been supplied a specific amount of heat Q_{in}, i.e. the first law efficiency is defined as:

$$\eta_{th} = \frac{W_{\text{net}}}{Q_{\text{in}}} \tag{2.5}$$

2.7.3 Second law efficiency

The difference between the second law to the first is the definition of useful or available energy. It takes into account the irreversibilities of a system and relates the actual energy transfer process to the energy transfer of the same process but reversible. This efficiency is important when going into detail and analysing individual components of a cycle. It will give an indication of where in a cycle most useful energy is lost due to an irreversible process. This means that the second law efficiency is defined as in eq. (2.6).

$$\eta_{2nd} = \frac{\eta_{th}}{\eta_{carnot}} \tag{2.6}$$

2.8 Combined cycle

A combined cycle (CC) is constructed of a Brayton Topping cycle (BTC) and a Rankine Bottoming cycle (RBC) where the exhaust gas from the BTC acts as the heat addition to the RBC. By doing so heat that otherwise would be rejected is used to do work. This significantly increases the total thermal efficiency. However, designing a combined cycle is a difficult task. The combined efficiency depends not only on the efficiency of the individual cycles but also on the thermal interaction between them. The steam cycle is heavily dependent on the energy left after the gas turbine and an overdimensioned gas turbine would thus lower the combined efficiency. The expression for the combined cycle can be written as

$$\eta_{\text{CC}} = \eta_{\text{BTC}} - (1 - \eta_{\text{BTC}})\eta_{\text{RBC}} \tag{2.7}$$

It is important to note that the efficiency of the bottoming cycle is dependent on both the quality of the Rankine cycle as well as the efficiency of the steam generation.

$$\eta_{\text{RBC}} = \eta_{\text{RC}}\eta_{\text{HRSG}} \tag{2.8}$$

In other words η_{RBC} is the efficiency of transferring heat from the exhaust gas to the steam turbine output. As has been mentioned earlier the HRSG is an integral part of this project and the technology will be discussed in a later section.

The appearance of a combined cycle can vary in some ways such as an extra combustion in the HRSG or the use of a feedwater heating system (dearator). In this project, for simplicity, the surrounding auxiliary systems and extra applications are omitted from the analysis. Thus, leaving more depth in the structure of the HRSG and the number of parameters that will be studied.

2.9 HRSG

By means of heat exchange the exhaust gas from the gas turbine is used to generate steam in the Rankine cycle. This process is called Heat Recovery Steam Generator (HRSG), which has been mentioned before. In this section the general process and effects parameters have on the process will be discussed.

The efficiency of the process relates how much of the energy leaving the gas turbine is transferred to the steam. To explain this further a figure showing the energy transfer of the HRSG is shown below.

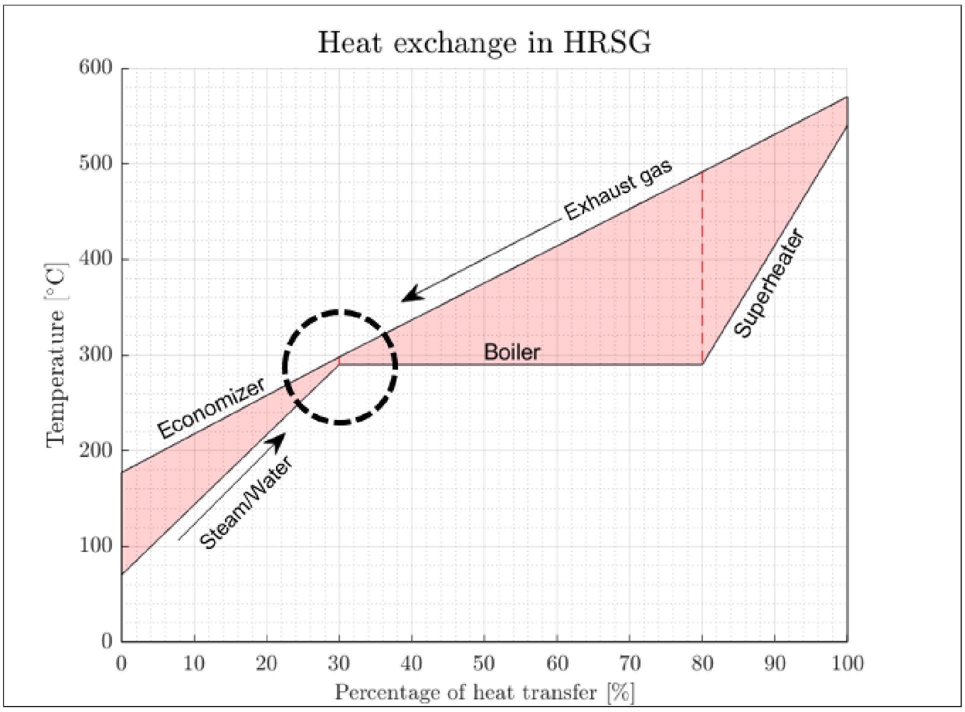

Figure 2.7: Visualisation of the heat transfer and flow direction of the HRSG.

Figure 2.7 shows two lines which enclose a red area. The red area illustrates the loss in potential work output due to temperature difference in the heat exchanger. The bottom line is the steam and the top is the exhaust gas flow. In the figure three sections are marked out. These are the three general sections in a HRSG: economizer, boiler and superheater. In the economizer, water is heated to the proximity of saturation temperature before entering the evaporation process. The evaporation process consists of two main parts; the steam drum and the boiler. The steam drum lets saturated water be circulated through the boiler. By controlling the level of water in the drum, the steam leaves the drum in a pipe higher than the water level. This step makes certain that no unwanted water is let through. By superheating, the water is heated beyond the saturation temperature to the designed turbine inlet temperature.

The most important parameter from a design perspective is the pinch point. This is where the temperatures of the flue gas and steam are the closest. The pinch point is highlighted by the circle in Figure 2.7. When designing a constraint on the pinch point it is important to understand the relation between a lower pinch point and an increase in efficiency [7]. However, the value needs to be non-zero since it otherwise would require an infinite heat transfer area.

One other constraint which is important when designing a HRSG is the stack temperature, i.e. the temperature leaving the HRSG to the environment. If it becomes too low the exhaust gas might condense which is not favoured. Also there are requirements on the maximum dispersion area of the exhaust when designing a power plant. Since the height of the chimney cannot be excessively high due to building restrictions the stack temperature is the most important parameter when controlling the dispersion. A lower stack temperature would lead to a dispersion at a lower altitude resulting in pollutants closer to the power plant. The constraint for the pinch point is kept throughout to maintain within the scope of this project.

In the next segments, some factors affecting the appearance and thus the efficiency of the HRSG will be discussed.

2.9.1 Exhaust temperature

Varying the exhaust temperature of the gas turbine keeping the pinch point constant will make the exhaust gas twist around the pinch point as illustrated in Figure 2.8.

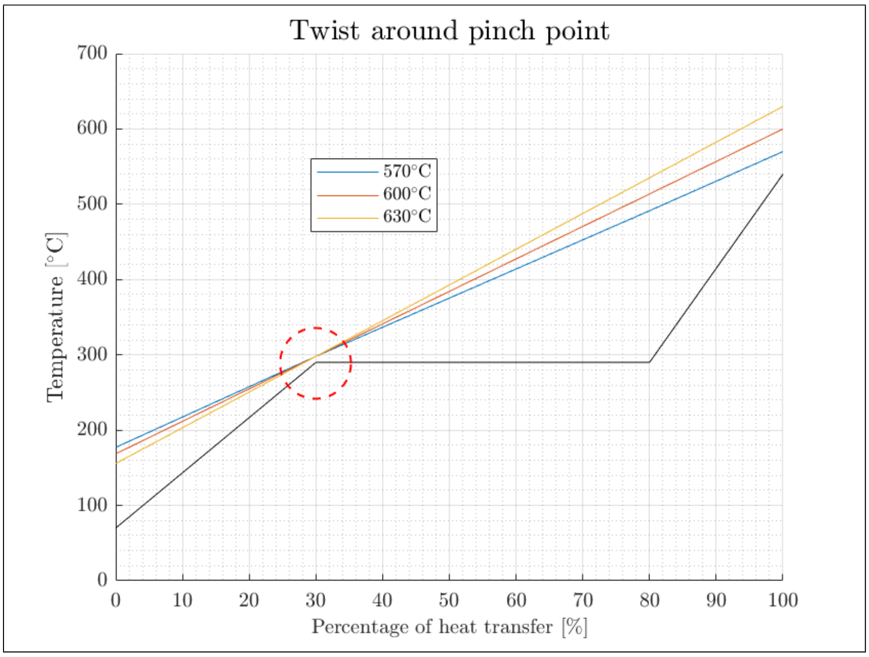

Figure 2.8: Shows the behaviour of the exhaust gas when altering its initial temperature.

In the Figure 2.8 the steam cycle is kept constant but depending on the configuration of the HRSG it will also be affected by changes in the exhaust temperature.

2.9.2 Live steam pressure

Choosing the live steam pressure in a steam cycle will have a great impact on the performance. Designing the pressure is a delicate matter since there will be a trade-off between the recovered heat in the HRSG and the power output. Increasing the pressure results in an increase in the evaporation temperature. Due to the pinch-constraint, the pinch shifts to the right and upwards as is shown in Figure 2.9.

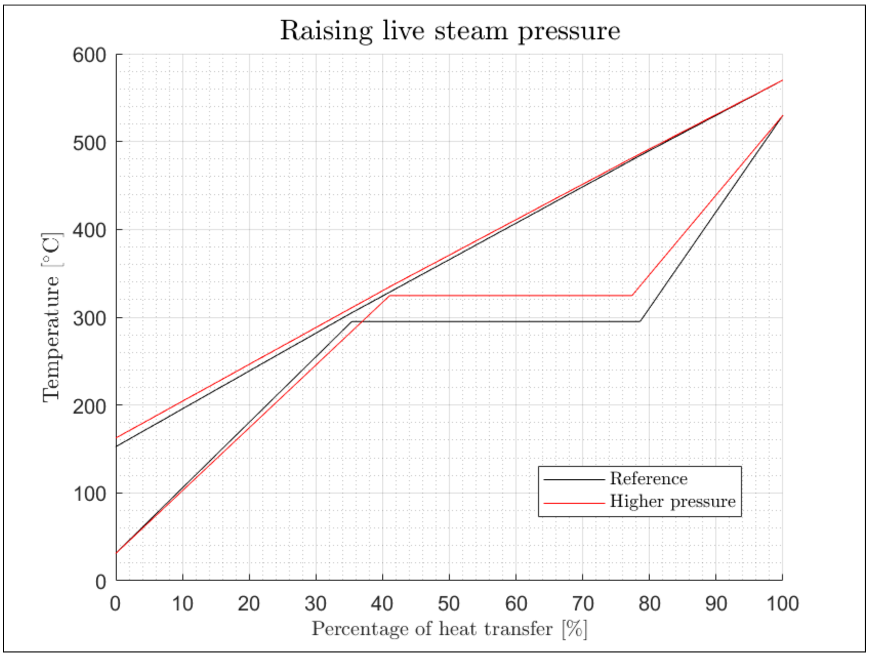

Figure 2.9: Alteration of heat transfer when raising the live steam pressure.

This results in a higher stack temperature, hence less heat has been transferred from the exhaust.

2.9.3 Pressure levels

From the previous section it is obvious having a high pressure improves power output whereas low pressure gives an more optimal stack temperature and thus more efficient steam generation. One logical way to benefit from both is to combine a higher and a lower pressure level. These will be referred to as HP and LP, respectively. By doing so the main steam flow is split and there will be high quality steam at the HP and the heat exchange at LP will lower the stack temperature increasing the efficiency of the HRSG [8]. By doing so the combined cycle becomes more complex since more components are needed. Of course, more pressure levels can be introduced but this will dramatically increase the cost of the power plant. The increase in performance is significant when adding a second pressure level and a bit smaller when adding a third [9] called intermediate pressure (IP). This in combination with the dramatic increase in costs of adding pressure levels makes it economically unreasonable to add a fourth.

The interaction between pressure levels is vital in understanding the behaviour of the HRSG. From what was discussed in the previous section, altering pressure will result in a change in boiling temperature. When more pressure levels are introduced the degree of freedom will increase making the design more complex. However, there exists an optimum in efficiency of the RBC for one unique combination of HP and LP. Thus, it is possible to remove one degree of freedom by stating that the LP is optimised after the given HP. When there exist three pressure levels one design approach is to assign the IP boiling temperature as the average of the two other boiling temperatures.

However, designing these optimum points are far from straightforward. Many factors such as the cost and available steam turbines affect the plausible locations of the pressure levels. One major design aspect is the change in volume flow when the pressure is increased. Higher pressure yields a lower volume flow [10], resulting in that the turbine requires a larger radius or higher rotational speed in order to compensate for the lower volume flow. Thus, it is concluded that the design of pressure levels in a combined cycle is an intricate process depending on many independent factors.

2.9.4 Deaerator

The deaerator is a common component in a steam cycle whose purpose is to separate any dissolved gases (mostly oxygen) from the feedwater. The structure of a deaerator may vary but the principle is the same; to use bleed steam from the steam turbine in order to deoxidise the feedwater. Otherwise the remaining oxygen could cause corrosion damage. This means that steam which would otherwise produce work is used to heat the feedwater close to saturation, resulting in a decrease in power output. However, by internally regenerating heat, irreversibilities that would arise from the exhaust gas heating the feedwater are diminished. Thus, the thermal efficiency of the system would increase. A visualisation of the gain in reversibility is shown below.

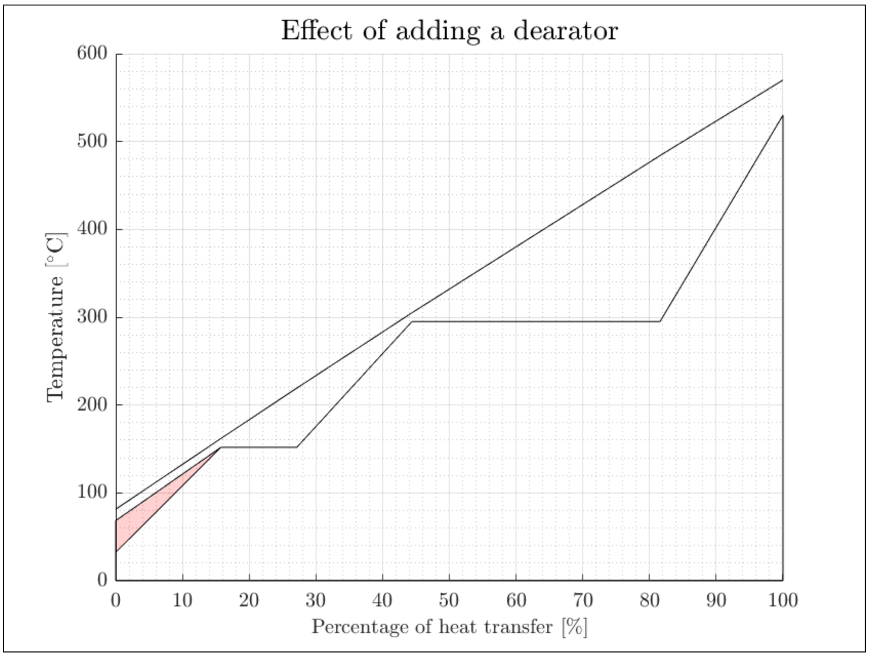

Figure 2.10: Shows the regenerated energy from the bleed steam.

The red shaded area in Figure 2.10 represents the gain in reversibility when a deaerator is introduced to the system.

2.10 Interaction between top and bottom cycle

To better understand the combined cycle it is appropriate to view it as two combined heat engines as shown below in Figure 2.11. The top cycle operates between the turbine inlet temperature (TIT) and the temperature of the exhaust gas (T_{exh}). The bottom cycle operates between the temperature which is reached after the HRSG and the condenser temperature T_{cond}. From the Carnot efficiency it is clear that changing the exhaust temperature will affect both cycles. By increasing the exhaust gas temperature the bottom cycle gets a greater operating temperature range whereas the top cycle gets a smaller.

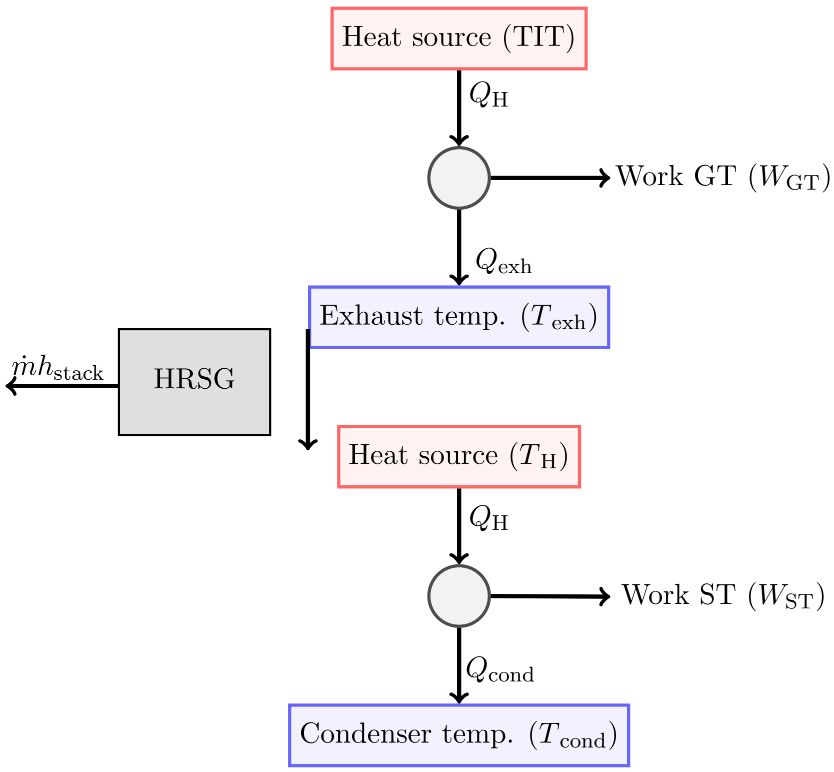

Figure 2.11: Visualisation of the combined cycle in terms of heat engines and the interaction between the top and bottom cycle.

The expression for the combined cycle efficiency can be written as:

$$\eta_{\mathrm{CC}} = \eta_{\mathrm{BTC}} + (1 - \eta_{\mathrm{BTC}})\eta_{\mathrm{RBC}} \tag{2.9}$$

Using the definition of the second law efficiency equation (2.9) can be rewritten:

$$\eta_{\mathrm{CC}} = \eta_{2nd,\mathrm{BTC}}(1 - \frac{T_{\mathrm{exh}}}{\mathrm{TIT}}) + \left(1 - \eta_{2nd,\mathrm{BTC}}(1 - \frac{T_{\mathrm{exh}}}{\mathrm{TIT}})\right)\eta_{2nd,\mathrm{RBC}}(1 - \frac{T_{\mathrm{cond}}}{T_{\mathrm{exh}}}) \tag{2.10}$$

To be able to examine the effect on the combined cycle efficiency when varying T_{exh} both second law efficiencies need to be assumed constant. They also need to be independent of the change in exhaust temperature. This is not true for many reasons. One example is that the amount of cooling air would be affected if changes are made to temperatures in the GT. In order to make a reasonably delimited report the scope was narrowed down to only analyze the effect on the bottoming cycle. From this stage it is therefore important to have in mind that there are many factors affecting the efficiency of a combined power plant that are neglected in order to be able to go into detail.

2.11 Mean temperature of heat addition

Recalling the aforementioned Carnot efficiency the two fundamental parameters influencing the possible efficiency of a thermodynamic cycle are the temperatures of heat addition and heat rejection. The temperature of heat rejection is most often not possible to lower since this is in most cases the temperature of the surroundings. Consequently, finding methods to increase the mean temperature of heat addition is vital to reach improved efficiencies. There are a number of variations to the original cycles that can be made in order to increase the efficiency. It is of note to understand that the addition of a process will increase the cost of the power plant significantly and the benefits achieved on the efficiency might affect the power output. Implying that all components of the cycle have to be overdimensioned to produce the needed power.

A method for estimating the mean temperature of heat addition is:

$$\bar{T} = \frac{\sum\limits_{n=1}^{n} \Delta h}{\Delta s} \tag{2.11}$$

Where n indicates the number of heat addition processes at constant pressure. Thus, the addition of pressure levels will add processes. The calculation considered the split of mass flow between each process. The effect of distributing the mass flow between pressure levels is a vital part of the change of mean temperature and will be discussed further.

2.11.1 Distribution of mass flow

If the total mass flow is constant but is increased in the HP and decreased in the LP, the mean temperature is increased. This will happen if the exhaust gas temperature from the gas turbine increases. Studying T-Q diagrams with different exhaust temperatures will clarify. Some percentage of heat transfer is moved from LP to the HP boiler which will yield a higher mean temperature. This effect is shown below in Figure 2.12.

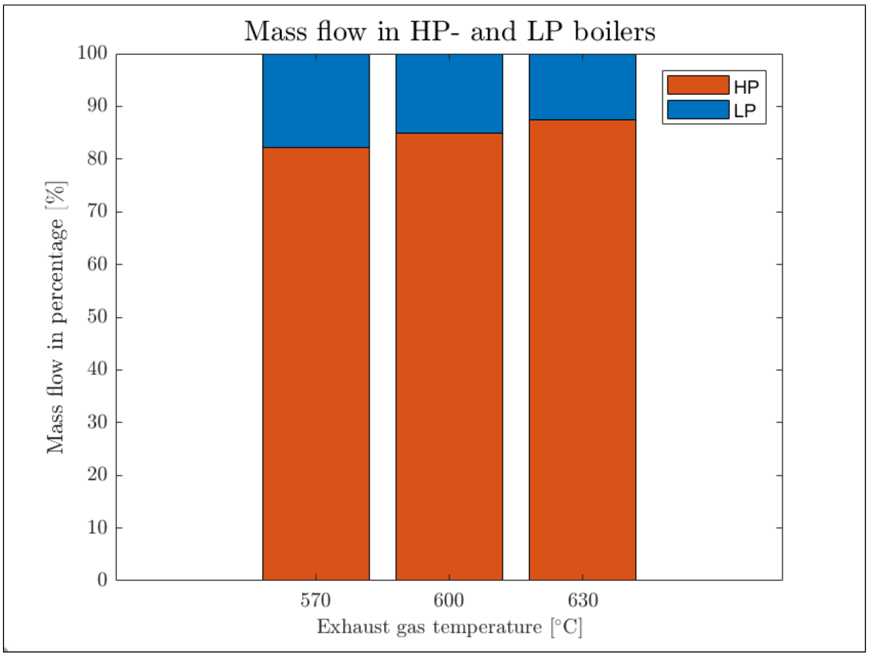

Figure 2.12: Shows the change in distribution of mass flow in the two boilers when changing the exhaust temperature.

As long as multiple pressure levels are present, there will always be a variation of mass flow distribution independent of the structure of the HRSG. The magnitude of the effect depends mostly on the pressure levels, i.e. between which temperatures the mass flow is distributed between.

2.11.2 Effect on the efficiency

The mean temperature of heat addition is directly related to the second law efficiency η_{2nd} which was mentioned in section 2.7.3 as the relation of the thermal efficiency against the Carnot efficiency. An increase in mean temperature yields an increase in the equivalent Carnot cycle which is described in Figure 2.13 which comes from the book *Gas Turbines for Electric Power Generation*[11] written by S. Can Gülen. If the apparent Carnot cycle is kept constant this will result in an increase in second law efficiency.

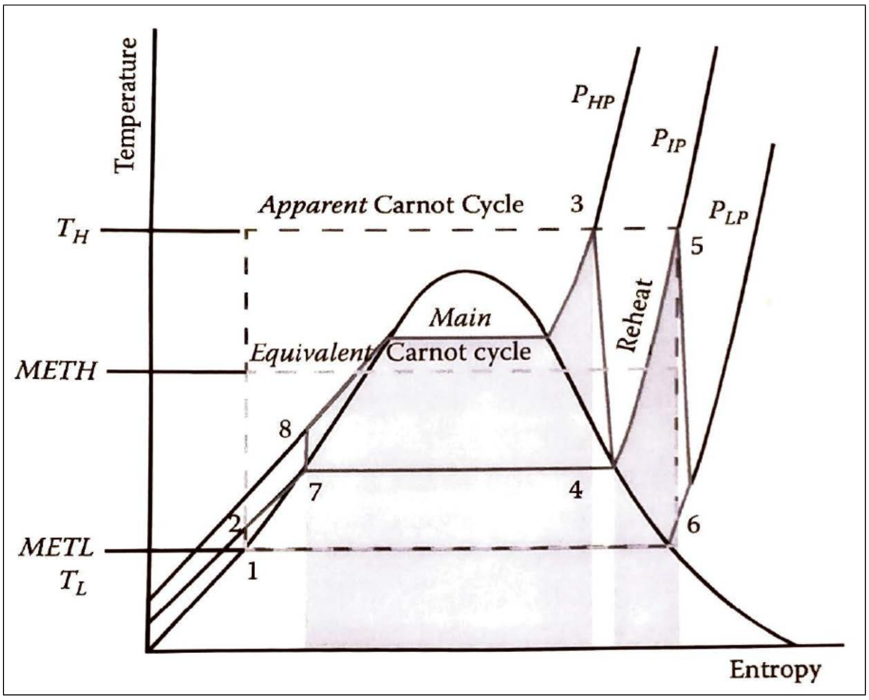

Figure 2.13: Temperature - entropy diagram highlighting the apparent and equivalent Carnot cycle.[11]

2.11.3 Estimation error

It has to be noted that the mean temperature of heat addition calculated from equation (2.11) does not result in the exact solution. The estimation is close and deemed acceptable since it is widely used, for instance in S. Can Gülen et al. [5]. The error comes from assuming a constant c_p in the heat addition occurring in the economizer and superheater. The error is shown in Figure 2.14.

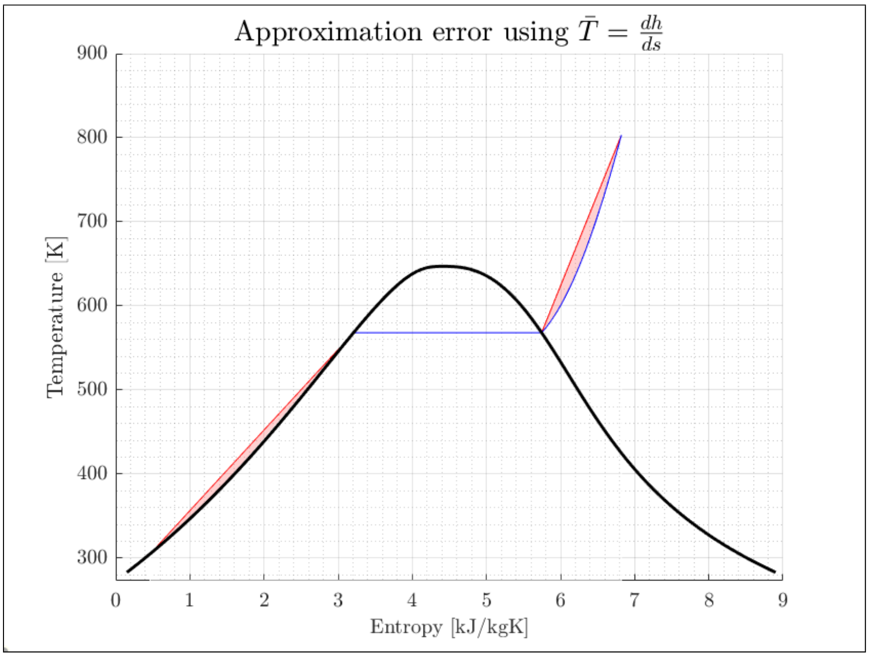

Figure 2.14: Visualisation of error using the estimation of mean temperature of heat addition.

The ensuing error is small and therefore neglected. Moreover, the analysis is done by studying variations in behaviour between a variety of cases. Thus the method was deemed acceptable.

3 Methodology

The approach of the project was straightforward, modelling a combined cycle which is capable of capturing a design point of a combined cycle and implement changes. Initially no unnecessary complexity was taken into account and after understanding one model, new components were added. This section will discuss the process of getting results and go into detail on the various models that were created. Additionally, the derivation of the expression which represents the combined cycle efficiency will be described.

The two scenarios, design and off-design, will be separated due to the fact that they will have completely different reactions to the same changes. The off-design scenario is only analysed on the two-pressure model since the method is substantially more time consuming and the aim is to see if the method for the design scenario is applicable to the off-design scenario. In the off-design scenario, to be able to capture the behaviour of varying parameters of a constant power plant, more complex models of steam turbines are needed. These turbines introduce the flow capacity which will result in a dependency between the pressure and volumetric flow. However, by introducing complexity to the system the calculation procedure becomes more unstable and the tolerance had to be increased in order to reach solutions. The second scenario will also have two possible parameter variations to analyse, both exhaust gas temperature and mass flow. For the design scenario, an alteration in mass flow only acts as a scaling factor and will not induce any change to the efficiency. However, with a set power plant, there will be an effect since heat transferring area of the heat exchangers are now fixed in the model and will not scale accordingly.

Furthermore, the correlations are assumed to be linear for all cases except for the most urging study which is the one for a state-of-the-art power plant in section 3.7. For this scenario, two corrections will be presented with differing advantages and disadvantages. A linear correlation will result in a more stable solution but will lead to, for a wider interval, a more substantial error than if the correlation is assumed to be of a second degree order. However, the error from a solution of a second degree order will increase more rapidly due to deviations from a reference scenario having a greater influence.

3.1 Modelling

The first model was a combined cycle with one pressure level in the HRSG. This is not a widely used configuration since it can be heavily improved by adding pressure levels as has been discussed in the theory section. However, due to its simplicity the behaviour of the HRSG could easily be analysed. The diagrams from the theory section is a result of the data from this model. Nonetheless, due to the irrelevance in general of this configuration no more attention was given to this data. The settings put into the components were as realistic as possible after referencing papers and text books concerning combined cycles values of e.g. pinch points, pressure ratios and exhaust temperatures.

3.1.1 Addition of deaerator

Understanding the trends when varying parameters made it possible to notice the changes implemented when adding components. Since it is vital to make sure no dissolved gases remain in the circulating fluid a deaerator is frequently used. Accordingly, a deaerator was added to all models except for the state-of-the-art case. In a Siemens state-of-the-art power plant a deaerator is not present during full load and deoxidation is executed in the condenser hot well.

How the deaerator is attached is a vital part in understanding the changes of the system when adding a third pressure level. The practice is to utilize excess energy in the LP stream. The mass flow through the economizer can be adjusted and the excess is recirculated to the feedwater heater to reach the designed temperature. In the model this is regulated in two separate ways, either by bypassing the deaerator with the recirculating LP stream or by steam injection drained from the steam turbine. The second alternative will decrease the efficiency since steam meant for expansion is irreversibly mixed with the feedwater. In a given design, the size of all heat exchangers are set which results in a given amount of heat transferred. Moreover, the deaerator has a required addition of energy and if the recirculated LP stream cannot fulfill this requirement the energy needs to be supplied by the steam injection. This concludes the following, at a certain LP the excess energy in the LP stream is not enough and this results in steam being drained from the steam turbine. Both cases are separate and opening the mixing of steam will close the bypass.

3.1.2 Two pressure levels

A natural modification to the model is to introduce another pressure level. This will generate another boiler and economizer. The addition of pressure levels will divide the main flow into two, one at a high pressure and one at a low pressure resulting in the possibility to utilise both an efficient steam generation and a high mean temperature of heat addition.

3.1.3 Applying constraint on steam turbine

Due to material restraints the temperature of the steam entering the steam turbine can not be too high. To control this there needs to be a cooling mechanism and in order to have as small irreversibilities as possible the mixing should happen as late as possible. In the model this is done by bypassing water from the HP boiler and mix it with the HP steam before entering the superheater. This would most likely not be the solution in real life applications and instead another superheater would be implemented where steam could be bypassed from the first to the next. However, the results from the model will be identical due to the fact that the mix is assumed to be fully superheated before entering the steam turbine.

3.1.4 Three pressure levels

The third pressure level IP was introduced with the condition that IP boiling temperature is the mean of the HP and the LP boiling temperatures. Designing a HRSG with three pressure levels is common and there is a significant effect on the combined cycle efficiency and the stack temperature. The IP level acts as a dampener between the shift in mass flow from LP to HP and will make changes in mean temperature less dramatic. One other alteration to the model is the addition of another superheater for the IP steam. This is common practice for a three-pressure level HRSG and will be affected when the pressure of the IP is subject to change.

As has been discussed in section 2.9.3 there needs to be a convincing incentive in terms increase in power production to meet the significant increase of the cost of the power plant. Hence, appropriate measures to the parameter settings in the model must be made. Temperatures and pressures are increased as would be the reasonable design choice if a three-pressure HRSG is chosen. The range of exhaust gas temperatures is now 590-650 °C and the range of HP is 100-150 bar.

3.2 Derivation of expression

Firstly, the expression which should be corrected needs to be determined. As mentioned in the section about this project's delimitations all factors affecting the gas cycle are neglected. Therefore, all data from the gas cycle can be assumed known. The expression is derived from the following equation representing the efficiency of the combined cycle.

$$\eta_{\mathrm{CC}} = \frac{\dot{P}_{\mathrm{BTC}} + \dot{P}_{\mathrm{RBC}}}{\dot{Q}\mathrm{in}} \tag{3.1}$$

Where $\dot{\mathrm{P}}_{\mathrm{BTC}}$ and $\dot{\mathrm{P}}_{\mathrm{RBC}}$ are the power output at the gas and steam generator respectively. $\dot{\mathrm{Q}}_{\mathrm{in}}$ is the total power input to the system which corresponds to the power in the combustion and pumps. The generated power in the bottom cycle can be written as the power difference between exhaust of the gas cycle and stack multiplied with the efficiency of the bottom cycle, thus eq. (3.1) can be rewritten as

$$\eta_{\mathrm{CC}} = \frac{\dot{P}_{\mathrm{BTC}} + \dot{m}_{\mathrm{exh}}\left(h_{\mathrm{exh}} - h_{\mathrm{stack}}\right)\eta_{\mathrm{RBC}}}{\dot{\mathrm{Q}}\mathrm{in}} \tag{3.2}$$

From eq. (3.2) all data concerning the BTC are assumed known. Thus, the remaining unknowns are the state at stack and the efficiency of the RBC. From what has been discussed in section 2.7.3 the thermal efficiency η_{RBC} can be reformulated in the following way:

$$\eta_{\mathrm{RBC}} = \eta_{2nd}\eta_{\mathrm{carnot}} = \eta_{2nd}\left(1 - \frac{T_{\mathrm{cond}}}{T_{\mathrm{H}}}\right) \tag{3.3}$$

Thus it is possible to combine eq. (3.2) and (3.3) to form an expression which will only have two unknowns; the thermodynamic state at stack h_{stack} and the second law efficiency η_{2nd} of the designated T_{H}.

$$\eta_{\mathrm{CC}} = \frac{\dot{P}_{\mathrm{BTC}} + \dot{m}_{\mathrm{exh}}\left(h_{\mathrm{exh}} - h_{\mathrm{stack}}\right)\eta_{2nd}\left(1 - \frac{T_{\mathrm{cond}}}{T_{\mathrm{H}}}\right)}{\dot{Q}_{\mathrm{in}}} \tag{3.4}$$

This is the final expression which will be examined. The unknowns are known for a reference case and the task will be to correct the changes that will occur them when altering the exhaust data of the GT. That means, eq. (3.4) is rewritten one final time introducing the correction factors x_{s} for the stack state and x_{H} for the high temperature T_{H};

$$\eta_{\mathrm{CC}} = \frac{\dot{P}_{\mathrm{BTC}} + \dot{m}_{\mathrm{exh}}\left(h_{\mathrm{exh}} - x_{\mathrm{s}}h_{\mathrm{stack,ref}}\right)\eta_{2nd,ref}\left(1 - \frac{T_{\mathrm{cond}}}{x_{\mathrm{H}}T_{\mathrm{H}}}\right)}{\dot{Q}_{\mathrm{in}}} \tag{3.5}$$

Thus, the natural next step is to analyse the behaviour of the two factors independently when alterations are made to the system. From the theory discussed in section 2.11.2 it is clear that the efficiency of a thermodynamic cycle is directly related to the mean temperature of heat addition. With the task in hand, knowing that in real life only the reference case will be known, it is sensible to put a correction on the choice of T_{H}, which is a unknown parameter connected to the steam cycle, and on the thermodynamic state of the stack. The correction factors in eq. (3.5) are dependent on a large amount of parameters. Therefore, it is vital to determine the behaviour of the unknowns when implementing changes to the system. In the next section, the trends will be shown and discussed with the goal to understand how a general method for obtaining the correction factors can be conducted. The method will be described for one scenario and then any deviations from the original method will be presented. Eq.(3.5) will be studied for two different representations of the high temperature T_{H}, the exhaust gas temperature (T_{exh}) and the mean enthalpy of the exhaust gas (h_{mean}).

3.3 Correlation study: Two pressure levels with no constraint on the steam turbine

To be able to conduct a correlation study, numerous variations of a reference case were done. By changing one parameter at a time its impact on the mean temperature and therefore the efficiency could be determined. The methodology will follow each scenario with the most simple first followed by scenarios of increasing complexity.

The first model subject to analysis is a two-pressure level HRSG including a deaerator designed with varying steam turbine inlet temperature. The design scenario this model is based on represents a case where the inlet temperature has not been determined. Instead the temperature difference between the steam turbine inlet and and gas turbine exhaust outlet is fixed. The fundamental aspect in understanding the appearance of the correlations is the behaviour of the response in mean temperature when implementing a change to the system. That means, how the mean temperature responds to a change in exhaust temperature or exhaust mass flow. By increasing the exhaust temperature for this model there will be two main parameters that will affect the mean temperature, i.e. the increase in inlet temperature and the shift in mass flow from the LP boiler to the HP boiler.

The design scenario predicts the CC performance if each point studied represents a unique design of the BC. Therefore, an increase of the exhaust gas mass flow will only scale the BC keeping the efficiency constant. The exhaust mass flow will be discussed more thoroughly in for the off-design study since changing mass flow in an already set power plant will have effects.

3.3.1 Correction of T_H

By varying the pressure of the HP steam the behaviour will differ since there will be a significant change of the energy available in the HP boiler. The boiling temperature will change as well as the amount steam going through. Following the discussion in section 2.11.1 the distribution of mass flow affects the mean temperature of heat addition. Thus, by varying the HP boiling temperature there will be a different rate of change in mean temperature. This is one dependency that needs to be considered. An analysis of different pressures was conducted in order to find the correlation.

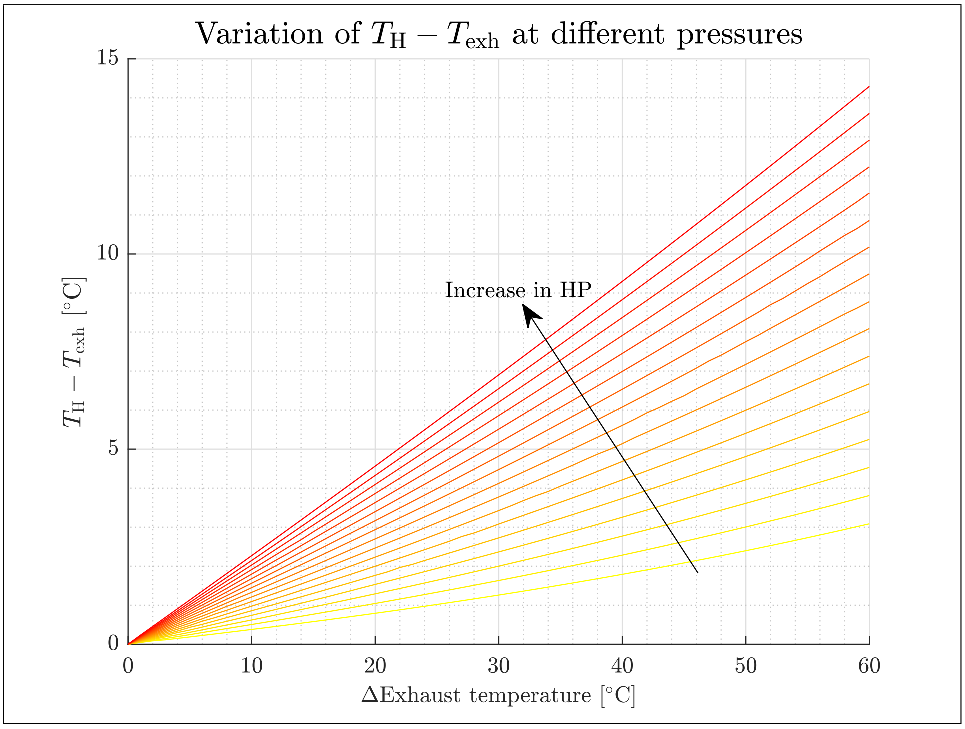

Figure 3.1: Shows effect of increasing the high pressure on the difference in exhaust temperature to T_H.

The gradient from yellow (lowest pressure) to red (highest pressure) in Figure 3.1 indicates the dependency of pressure on the correction factor. The y-axis represents the deviation of the correct T_H to the exhaust temperature T_{exh}. A higher pressure yields a higher gradient of the deviation which means that there is a more rapid change of mean temperature. Recalling the discussion about distribution of mass flow (section 2.11.1), the mean temperature increases due to a bigger part of the flow goes through the HP boiler. An increase in pressure implies an increase in boiling temperature and the mass flow is then distributed to a higher temperature. This will result in a faster rate of increase in mean temperature in a given interval of exhaust temperatures, proving that the correction factors will be dependent on the pressure. Therefore, the next step is to determine a way to correlate a variety of pressures to a parameter which will be known. The second law efficiency η_{2nd}, due to its relation to the mean temperature proved to be an appropriate parameter. Therefore the gradients of the deviations depicted in Figure 3.1 was correlated against the second law efficiency where the gradient is defined as

$$\frac{\partial(T_{\mathrm{H}} - T_{\mathrm{exh}})}{\partial(\Delta T_{\mathrm{exh}})} \tag{3.6}$$

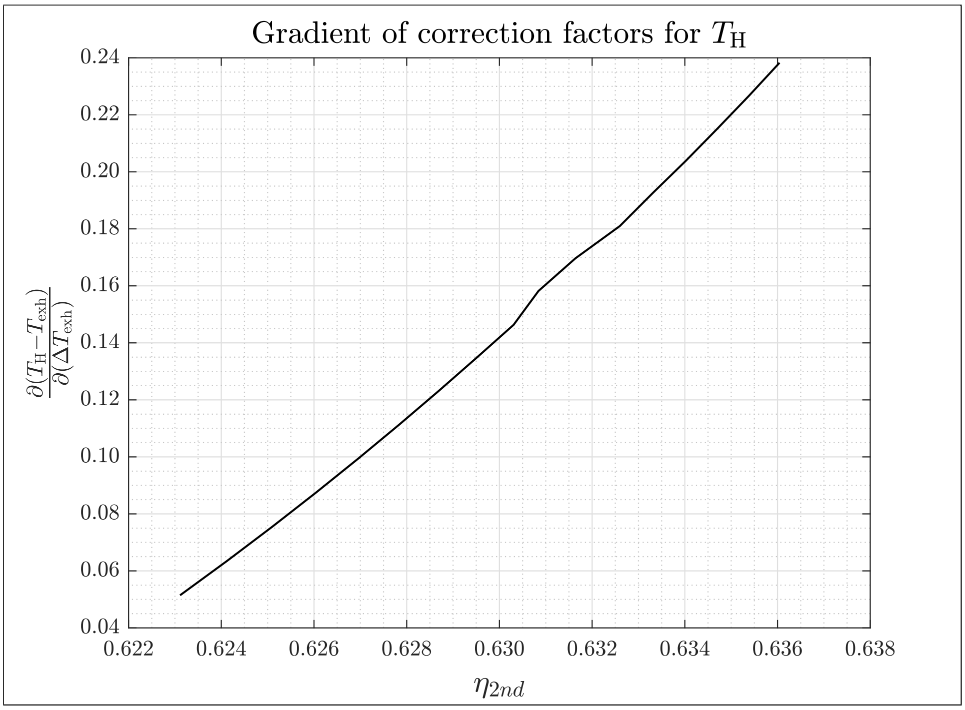

Figure 3.2: Shows the linear behaviour of the gradient of the correlation when varying η_{2nd}.

It is obvious from Figure 3.2 that there's a correlation between the value of the correction factors and the second law efficiency, η_{2nd}. However, this method consists of a substantial flaw. During this study the LP has been kept constant and by varying it, a dependency of the LP on the correction factors is also noticeable. Making correction factors for every imaginable combination would be too time-consuming to be viable. By recalling the discussion in section 2.9.3 regarding designing the LP after a given HP, it is possible to circumvent this obstacle by assuming an optimisation has been carried through in the design process.

3.3.2 Optimising LP at a given HP

By maintaining a constant HP and varying the LP it is possible to find the LP which will yield the maximum efficiency of the RBC. The efficiency of the RBC is given as eq. (2.8) and is the product of the efficiency of the HRSG and the one of the steam cycle. The reason why an optimal level of LP exists is due to the trade-off between increasing the mean temperature of heat addition and the efficiency of steam generation affecting the efficiencies of the steam cycle and the HRSG respectively.

Subsequently, the following procedure would be to correlate the optimised case's correction factors to their corresponding η_{2nd}. The optimisation is a rather complex process and there are a number of factors affecting the efficiencies and consequently there was no direct relation to the optimised LP when changing the HP. An optimisation process for a HP of 90 bar is visualised below in Figure 3.3 as an example.

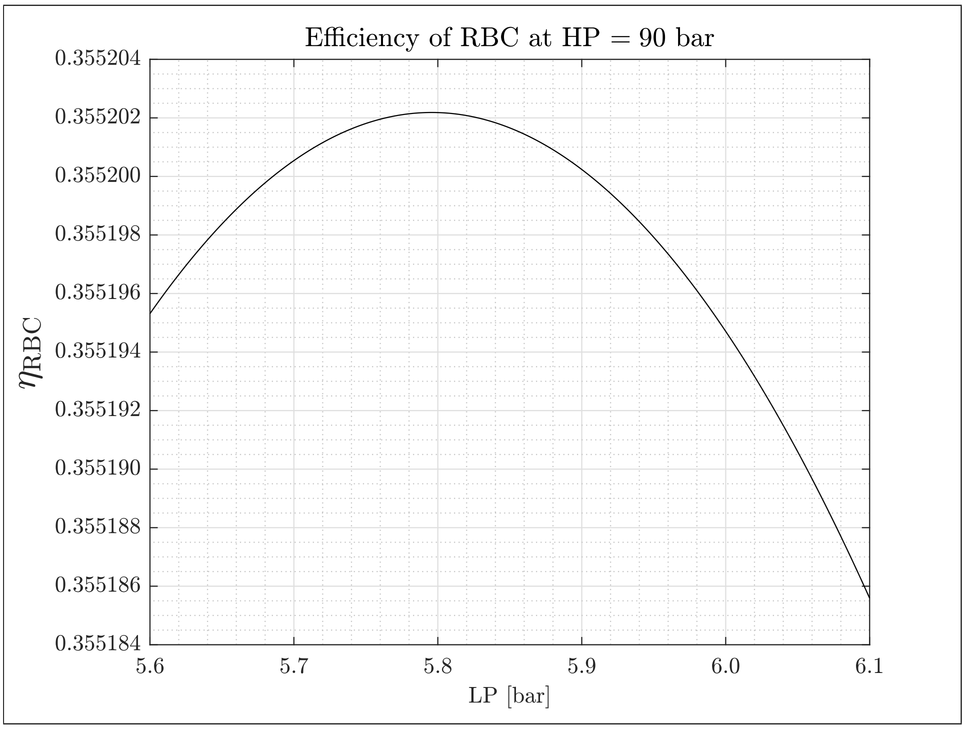

Figure 3.3: Shows the optimal LP for a HP of 90 bar.

In order to not do an unnecessarily extensive analysis, each data point was kept to one decimal point. When implementing this into a real steam cycle it is important to note that it is not possible to choose an exact LP since it will be limited to wherever there is space for steam injection. Hence, it is appropriate to not be too rigorous in this study. Instead, focus was shifted towards a sensitivity analysis which would identify the magnitude of error for a sub-optimal case. This optimisation process was conducted for all cases of HP within the range 80 bar to 120 bar.

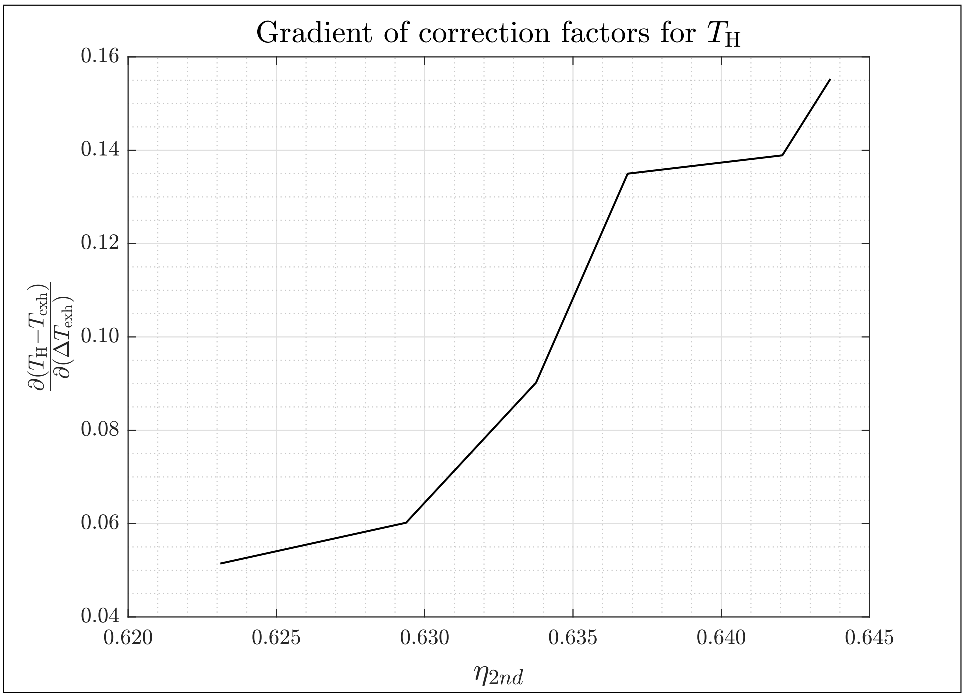

Figure 3.4: Variation of correction factor for the optimal cases.

The correlation between the gradients and η_{2nd} is, as is shown in Figure 3.4, not as linear as when the LP was kept constant due to the rather rough optimisation process rendering in the visual disturbance. This will induce an error when estimating the correlation as linear. This will be discussed further in the sensitivity analysis.

Up until this point the method does not capture the change occurring in the LP side which affects the thermodynamic state of the stack. A suggestion to complement the optimisation process in order to improve the correction factor would be to use the mean thermodynamic state of the exhaust instead of T_{exh} in eq. (3.4) with its corresponding η_{2nd}. The motivation behind this change is to make the correction factors less susceptible to big errors when the LP deviates from the optimised value.

Thus, eq. (3.5) is rewritten as follows

$$\eta_{\text{CC}} = \frac{\dot{P}_{\text{BTC}} + \dot{m}_{\text{exh}}\big(h_{\text{exh}} - x_{\text{s}} h_{\text{stack,ref}}\big)\eta_{2nd}\left(1 - \dfrac{T_{\text{cond}}}{x_{\text{H}} h_{\text{mean}}}\right)}{\dot{Q}_{\text{in}}} \tag{3.7}$$

Where h_{mean} is defined as

$$h_{\text{mean}} = \frac{h_{\text{exh}} + x_{\text{s}} h_{\text{stack,ref}}}{2} \tag{3.8}$$

Eq. (3.8) implies a dependency of the stack correction on the correction of T_{H} making it imperative to achieve an accurate x_{s}. It can be noted that dividing a temperature with an enthalpy is not physically possible and a more correct expression would include the mean exhaust gas temperature T_{mean} instead. However, this would require one more correction factor on the stack temperature and since the difference is negligible the correction was done for h_{mean}. To do a comparison between the two approaches, an error is introduced as the difference between the real value to the corrected value. In mathematical terms that means

$$\varepsilon = |\eta_{\text{CC,real}} - \eta_{\text{CC,calc}}| \tag{3.9}$$

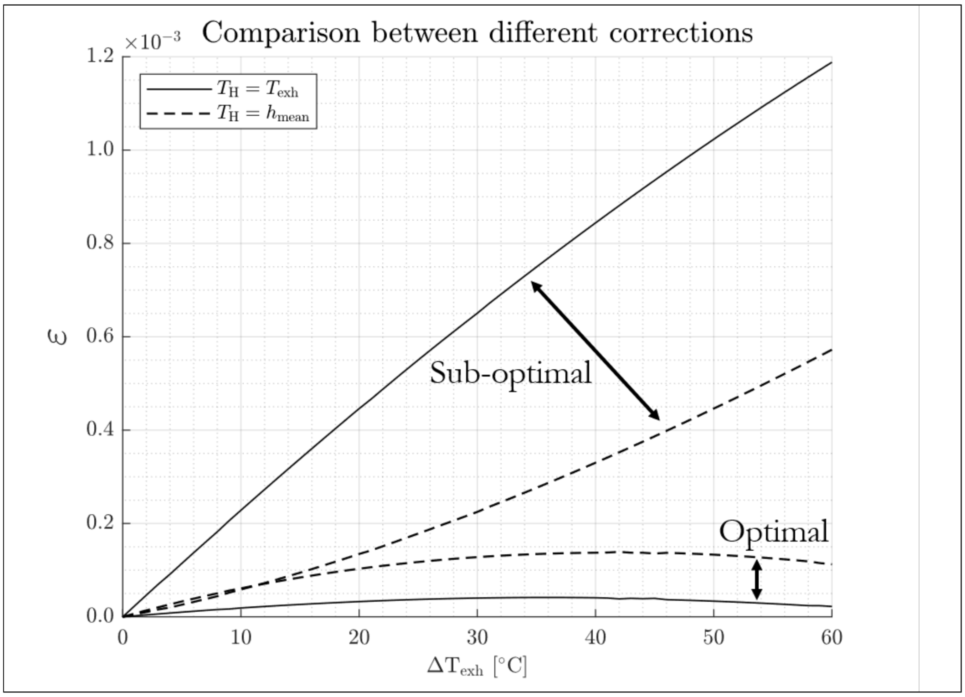

Figure 3.5: Comparison of the error for optimised and sub-optimal case on the combined cycle efficiency when using the different correction factors.

Figure 3.5 shows the decrease in error when the LP is far from the optimal. The two lower lines for both cases represent the error for the optimal LP whereas the other lines represent the error when the LP deviates from the optimal. Using the correction factor on h_{mean} will result in a smaller error when LP is changed. Even if there is a slight improvement on the optimal scenario when using T_{exh}, it is of more importance to have a method that is more stable in terms of LP variations. Therefore, when moving forward h_{mean} is used as the parameter that is corrected.

3.3.3 Correction on thermodynamic state at stack

After correcting the difference between the exhaust temperature and T_{H} the thermal state of the stack, h_{stack}, needs to be corrected as well. It was hypothesised that there would be a similar relation between the correction factor and η_{2nd} as has been discussed.

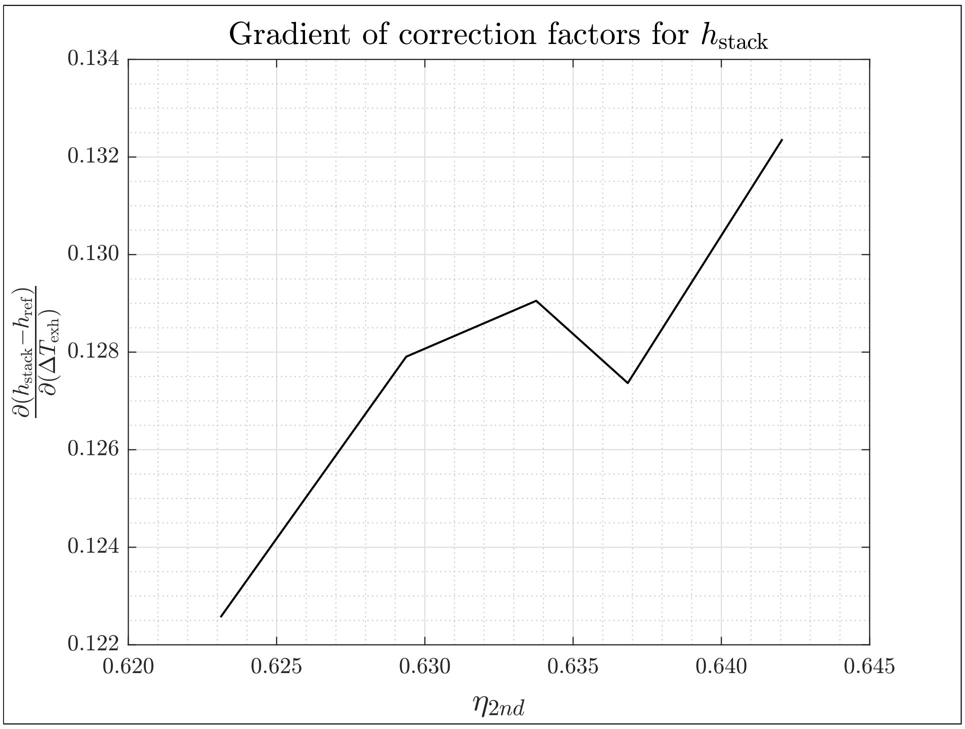

Figure 3.6: Variation of correction factor on the thermodynamic state of the stack h_{stack} for the optimal cases.

From Figure 3.6 it is less obvious that there is a correlation between the values of correction factors and the second law efficiency. One could imagine that assuming a linear correlation would impose a significant error. However, in consonance with a statement in the previous section, the focus will be on finding an efficient method followed by an analysis of the prevailing error. Thus, this linear correlation is assumed.

At this stage, methods for achieving correction factors for a range of different pressures have been implemented. By following these methods, correction factors can be estimated and then incorporated into eq. (3.5). The simulation program IPSEpro gives the true solution which can be compared to the corrected solution obtaining an error indicating the strength and accuracy of the method.

3.4 Correlation study: Two pressure levels with constraint on the steam turbine

By adding a constraint on the temperature of the steam turbine inlet the model becomes more realistically adapted. As has been discussed previously this is a necessary design parameter to keep constant in order to remain within the material capabilities of the steam turbine. Understanding the impact this model change has on the mean temperature of heat addition will be the key to understanding the resulting effect on the correlation.

When keeping the steam turbine inlet temperature constant the mean temperature will only be dependent on the exchange in mass flow between the boilers. The correlation will therefore be slower and its appearance needs to be studied in order to validate if the previous method still is applicable.

The optimisation process is done in the same manner and the correction will be done for h_{mean} since this has been proven as the method with most stability.

3.4.1 Correction of T_{H}

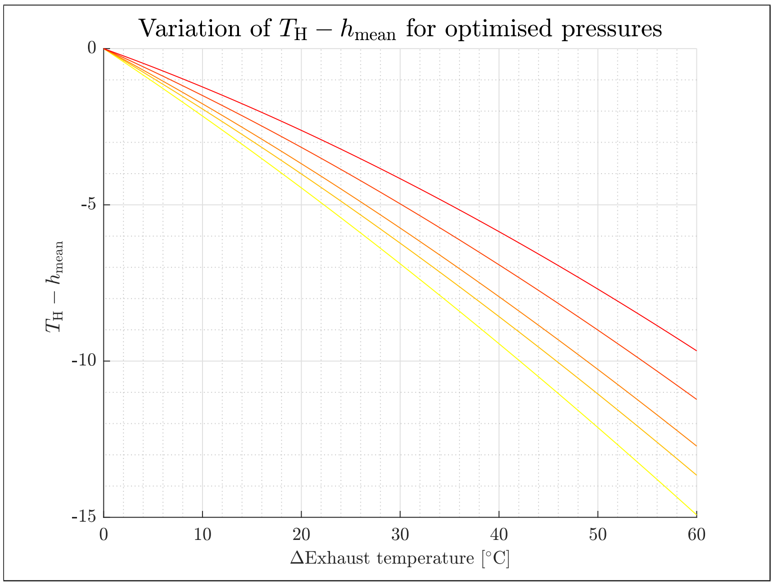

Figure 3.7: Shows the deviation of $T_{\mathrm{H}} - h_{\mathrm{mean}}$ for the optimised pressure combinations with a fixed steam turbine inlet temperature.

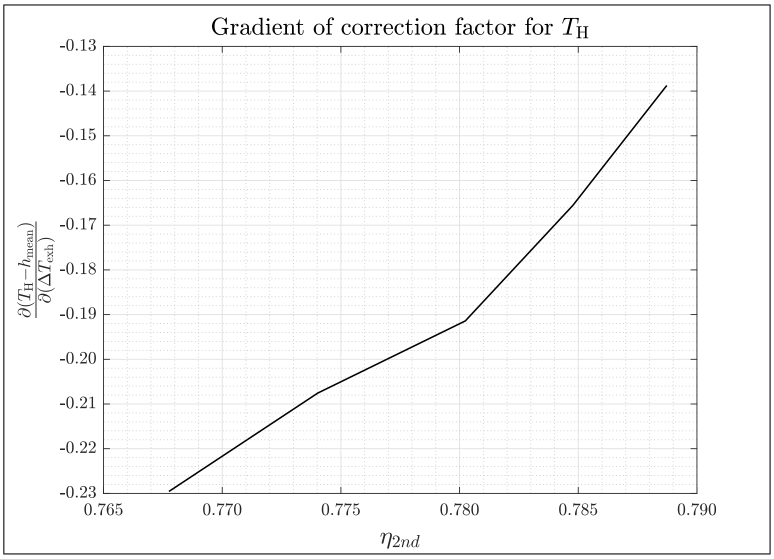

Figure 3.8: Gradients of the lines from Figure 3.7 in relation to the second law efficiency η_{2nd}.

It is obvious from Figures 3.7 and 3.8 that the relation between the gradients of the deviations are still linearly dependent and there is no unexpected behaviour other than $T_{\mathrm{H}} - h_{\mathrm{mean}}$ being negative which originates from that the variation in T_{H} is slower, as mentioned previously. However, when studying Figure 3.7 closer, the deviations are slightly curved meaning that by assuming linear correlations the error would be exponential.

3.4.2 Correction of h_{stack}

The same study was conducted for the correction factors on the thermodynamic state at the stack, h_{stack}.

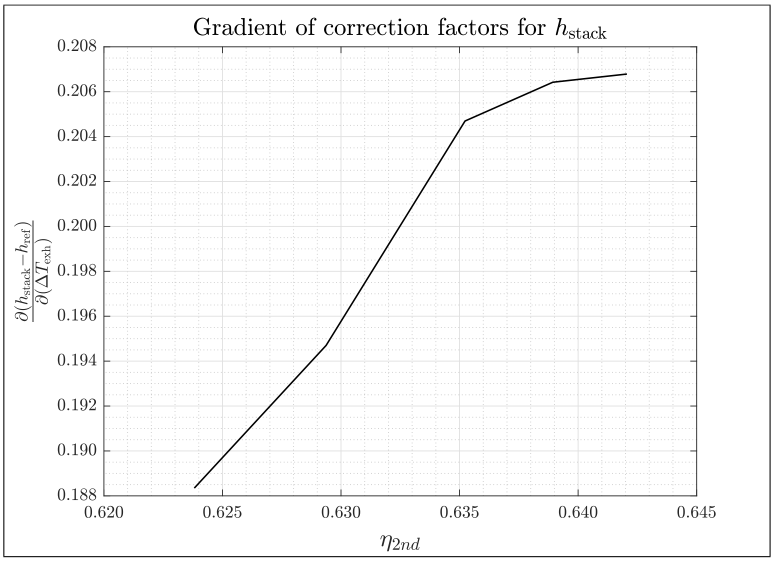

Figure 3.9: Gradients of the correlation factors of h_{stack} in relation to the second law efficiency when having a fixed steam turbine inlet temperature.

From Figure 3.9 it is evident that a correlation between the correction factors of the thermodynamic state of the stack and the second law efficiency still exists. Consequently, it can be concluded that the method is still valid for a fixed steam turbine inlet temperature.

3.5 Correlation study: Two pressure levels with constraint on the steam turbine during off-design

Even if the appearance of the model for the second scenario is almost identical to the first, the settings of parameters differ. Instead of having set pressure levels, design parameters in the steam turbines are kept constant. This will in turn lead to an alteration in pressure when exhaust temperature and mass flow are altered. Also, since there are two different parameter variations to consider, the models should be interchangeable. An increase in exhaust gas temperature will increase the amount of cooling water needed to maintain the steam turbine inlet temperature whereas an increase in exhaust gas mass flow decreased the water needed. Therefore, it was ensured that the spray was always active for both correlations.

Due to the increase in tolerance in order to maintain successful compilation of the iteration solver, the output data were of lesser quality and introduced disturbances to the continuity of the data. This lead to concern of the accuracy of the correlation study. Two separate studies were conducted to determine the correlation between the second law efficiency and the gradient of correction factors for exhaust gas temperature and mass flow.

3.5.1 Correction of T_{H} with mass flow variation

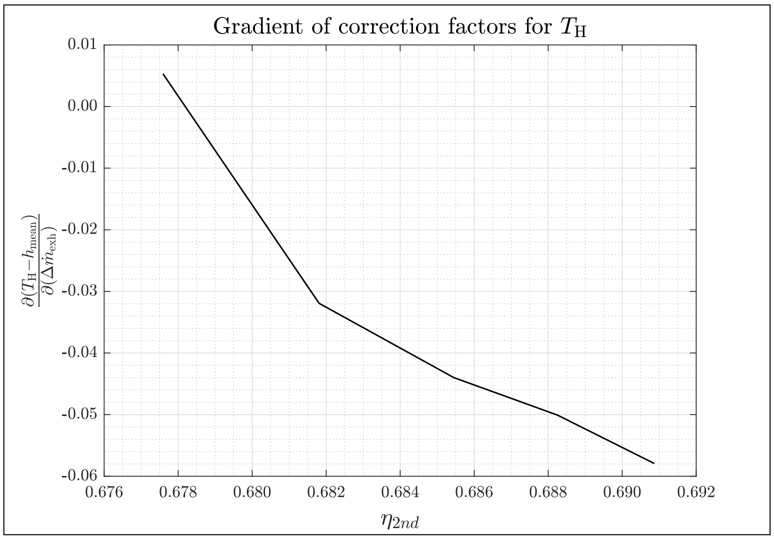

Figure 3.10: Gradients of the correlation factors of T_{H} in relation to the second law efficiency for the second scenario and varying the exhaust gas mass flow.

The lower quality of data affected the accuracy in getting the correction factors. However, there seem to be a trend between the gradient and the second law efficiency once again as seen in Figure 3.10. A linear behaviour was assumed and was used for the correction.

3.5.2 Correction of h_{stack} with mass flow variation

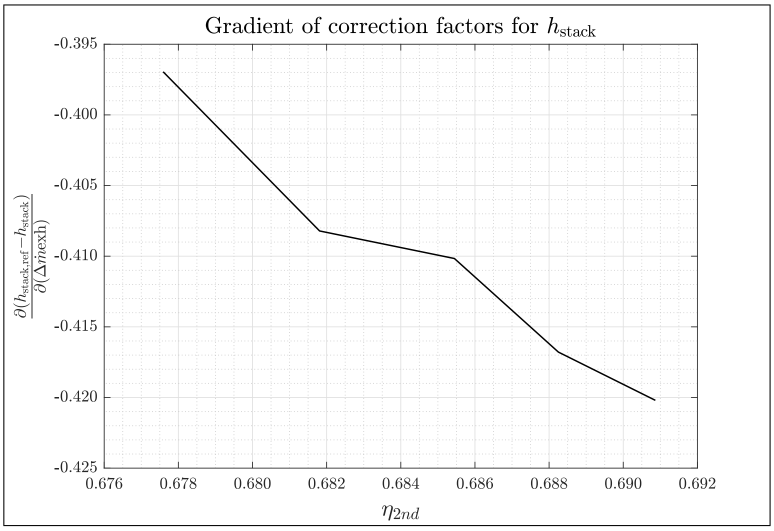

Figure 3.11: Gradients of the correlation factors of h_{stack} in relation to the second law efficiency for the second scenario and varying the exhaust gas mass flow.

The dependency of η_{2nd} is also clear when looking at the thermodynamic state of the stack. The gradient does not vary much but there is still a distinct negative trend, as shown in Figure 3.11.

3.5.3 Correction of T_{H} with exhaust temperature variation

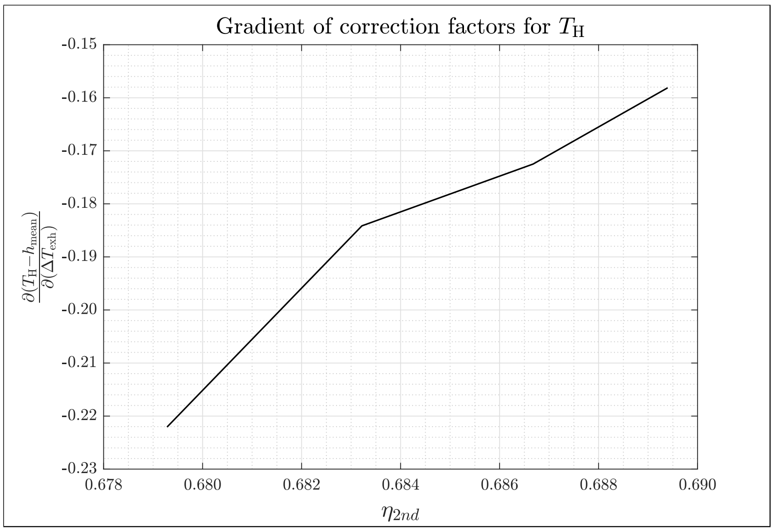

Figure 3.12: Gradients of the correlation factors of T_{H} in relation to the second law efficiency for the second scenario and varying the exhaust gas temperature.

From Figure 3.12 the correlation is once again visible. The quality of data was consistently faulting for this case and contributed to uncertainties for the results. Nevertheless, a linear correlation could be assumed.

3.5.4 Correction of h_{stack} with exhaust temperature variation

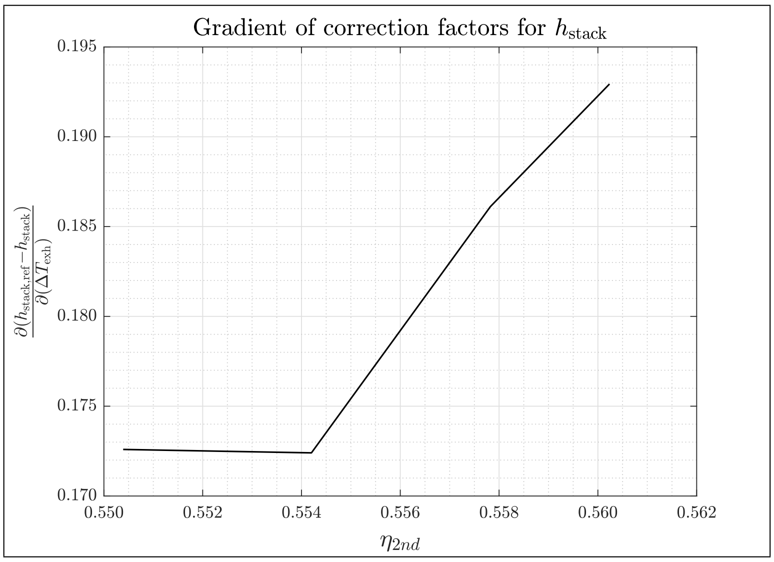

Figure 3.13: Gradients of the correlation factors of h_{stack} in relation to the second law efficiency for the second scenario and varying the exhaust gas temperature.

Figure 3.13 makes it clear that there is a linear correlation after a certain η_{2nd}. Due to the small change in gradients of the correction factors a linear behaviour is assumed without significant effects. The reason for the behaviour of the first data point originates from the many convergence issues of said data point.

3.6 Correlation study: Three pressure levels with constraint on the steam turbine

Recalling the discussion in section 3.1.4, it is to be expected that the behaviour of the three-pressure level HRSG should resemble the two-pressure level behaviour, although slower. The steam turbine inlet temperature is kept constant and will result in a change in mean temperature which will be mainly caused by the split in mass flow between the three boilers. By adding a boiler in between the other LP and HP the flow will be distributed with a lesser effect on the mean temperature than if there is only two boilers. Once again, the method has to be verified for this case to be able to conclude that the addition of an IP does not affect the behaviour of correction factors.

As aforementioned, parameters have been updated to reflect a realistic power plant using three pressure levels. This will not affect the approach and correction factors. An important issue that arises from adding a third pressure level is that during the optimisation process the optimal LP is rather low, around 2.6-2.7 bar. Having a LP this low requires steam injection in order to handle the energy demanded from the deaerator. The two different scenarios of heating the deaerator will affect the heat exchange in the economizer and the stack temperature will thus be affected differently when increasing the exhaust gas temperature. What this means in terms of obtaining the correction factors is that there will be two different correlations requiring separate correction factors. Incorporating factors for both in one method is irrelevant because the point at which the switch occurs will never be known. Thus, it is imperative to determine the most likely scenario for a certain designed power plant.

The optimal circumstances for the model done for this analysis appeared when the steam injection was active making it a proper assumption to study that correlation. From an article analysing a three-pressure level Combined Cycle power plant [12] the assumption of having steam injection for the optimised case is reinforced. Therefore, with the established reasoning it is possible to move forward with the general method.

During the study it was obvious that the added pressure level increased the efficiency of the HRSG resulting in a low stack temperature for all combinations of pressure levels, meaning that the dependency of η_{2nd} on the slope of the correction factors was almost flat. Therefore, a constant correction factor could be assumed.

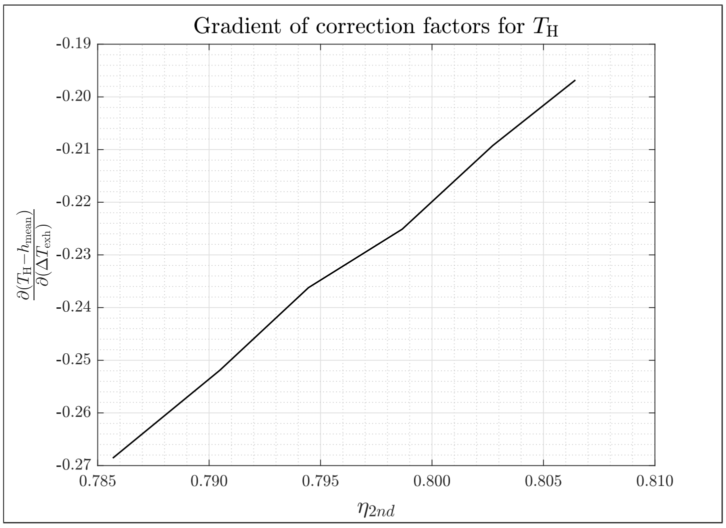

Figure 3.14: Gradients of the correlation factors of h_{stack} in relation to the second law efficiency for the first scenario with three pressure levels and varying the exhaust gas temperature.

However, as Figure 3.14 clearly shows, there is a larger variation for the correction factor for T_{H}.

3.7 Correlation study: State-of-the-art power plant

To be able to reach useful results, a model of a state-of-the-art power plant was constructed. This model was developed from a Siemens three-pressure reheat power plant. It differed mainly in that it has no deaerator active at full load and the deoxidisation takes place in the condenser with a technique called condensate polishing. Also, a so-called Benson boiler is used instead of a regular drum boiler. The Benson technique uses a once-through boiler without recirculating water. A more detailed description of the method and the Siemens power plant can be found here [13]. Understanding these complex components are not within the scope of this project and will therefore not be discussed further. However, a model that mimics the general behaviour of this power plant is possible to construct by implementing a fuel pre-heater and removing the deaerator from the previous three-pressure model.

The added reheat process will substantially affect the mean temperature of heat addition and the optimisation process is moved to the IP since this parameter is now left free to vary. In a power plant using a reheat stage the IP is the parameter to optimise in order to obtain the maximum efficiency. A report studying the optimisation of a reheat power plant [14] states that a usual optimum IP occurs at around 20-30 percent of the HP.

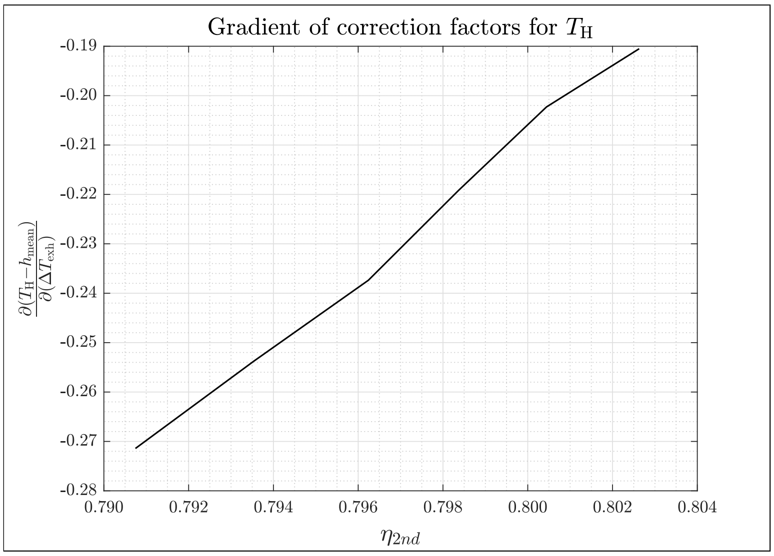

Figure 3.15: Gradients of the correlation factors of h_{stack} in relation to the second law efficiency for the first scenario with three pressure levels with reheat and varying the exhaust gas temperature.

When comparing Figure 3.15 to Figure 3.14 the correlation is very similar, implying that the correction factors could be interchangeable between the two different cases. Once more, the stack state is very resistant to changes in the system and can be assumed constant. Due to this being the most useful scenario for Siemens, a study of the exponential error was conducted. Therefore, another correction was done where the correlation is assumed to be of the second order.

4 Results

In this section the results obtained from the method will be presented in figures and tables. A variety of cases have been tried and the results will be shown independently. The structure of the results will be in three parts where the correction factors, x_{s} and x_{H} in eq.(3.7), will be presented first, followed by figures representing the sensitivity of the solution for a set of different pressures. Tables will act as a complement to the figures in order to highlight values of the accuracy of the solution for the optimised case. The error the correction factors induce, which is described in eq.(3.9), is used throughout the results section and depends on which type of correction is used. In order to show the importance of introducing the correction factors, the error when no correction factors are used is also presented.

4.1 Two pressure levels with no constraint on the steam turbine

The first results follows from the initial method of correcting an imagined combined cycle with no losses nor spray, therefore varying the temperature before the steam turbine inlet. The result of the correction factors are presented below.

$$x_{\mathrm{s}} = 1 - \frac{\big(0.3785\,\eta_{2nd} - 0.1668\big)\Delta T_{\mathrm{exh}}}{h_{\mathrm{stack,ref}}} \tag{4.1}$$

$$x_{\mathrm{H}} = 1 + \frac{\big(2.524\,\eta_{2nd} - 2.038\big)\Delta T_{\mathrm{exh}}}{h_{\mathrm{mean}}} \tag{4.2}$$

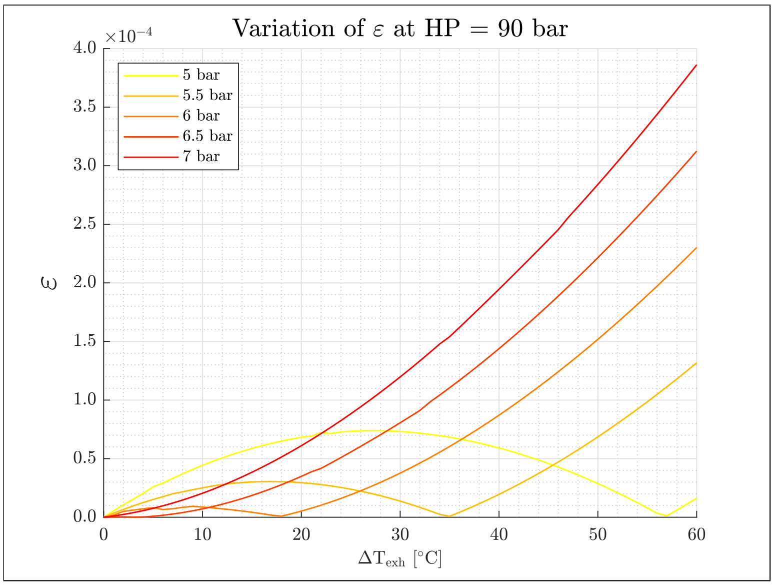

Figure 4.1: Variation of the error at different LPs with a constant HP of 90 bar.

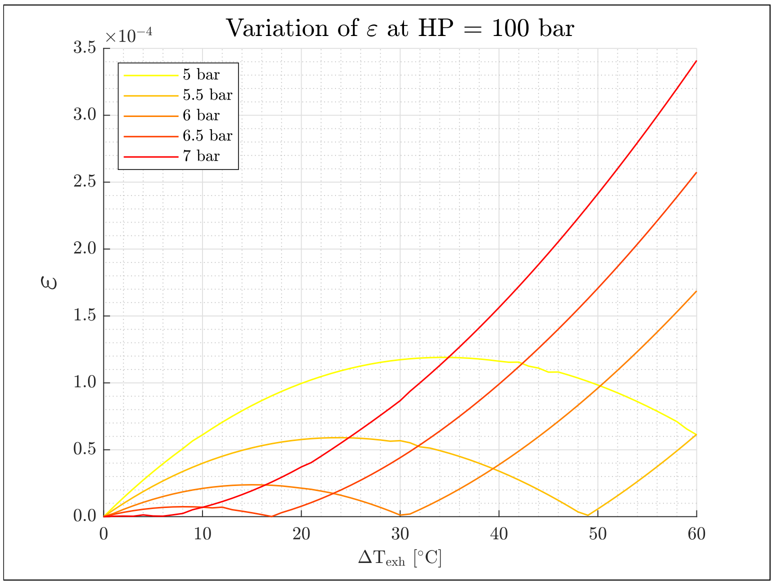

Figure 4.2: Variation of the error at different LPs with a constant HP of 100 bar.

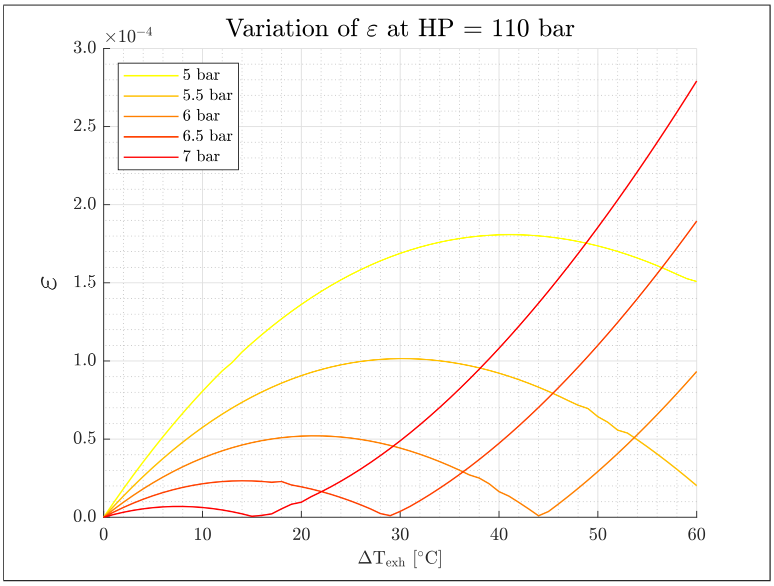

Figure 4.3: Variation of the error at different LPs with a constant HP of 110 bar.

Figures 4.1-4.3 are showing how the error varies with a change in exhaust temperature as well as the effect of changing the LP from the optimal preference. It can also be noted that the error increases exponentially, caused by the linear approximation of the deviation $T_{\text{H}} - h_{\text{mean}}$, as seen in Figure 3.7, which yields large errors at greater ΔT_{exh}.

	ε at HP = 90 bar		**ε at HP = 100 bar**		**ε at HP = 110 bar**	
Type of correction	**+10°C**	**+20°C**	**+10°C**	**+20°C**	**+10°C**	**+20°C**
No correction	$4.37 \cdot 10^{-4}$	$8.61 \cdot 10^{-4}$	$4.76 \cdot 10^{-4}$	$9.35 \cdot 10^{-4}$	$5.22 \cdot 10^{-4}$	$1.00 \cdot 10^{-3}$
Correction of T_{H}	$3.78 \cdot 10^{-4}$	$7.27 \cdot 10^{-4}$	$3.88 \cdot 10^{-4}$	$7.54 \cdot 10^{-4}$	$4.07 \cdot 10^{-4}$	$7.83 \cdot 10^{-4}$
Correction of h_{stack}	$1.97 \cdot 10^{-4}$	$4.14 \cdot 10^{-4}$	$1.65 \cdot 10^{-4}$	$3.51 \cdot 10^{-4}$	$1.28 \cdot 10^{-4}$	$2.75 \cdot 10^{-4}$
Both corrections	$7.17 \cdot 10^{-6}$	$5.25 \cdot 10^{-6}$	$6.99 \cdot 10^{-6}$	$7.79 \cdot 10^{-6}$	$6.24 \cdot 10^{-6}$	$9.62 \cdot 10^{-6}$

Table 1: Highlights values from Figures 4.1 - 4.3 and shows the error of the optimised case.

In order to show the effect of the correction factors, Table 1 presents the values of the error for the optimal case when there's first no correction and then adding only one of the corrections respectively and lastly combining both to reach the concluding error. The errors are as expected larger for when no correction is used and gets significantly smaller when both corrections are used. The quadratic behaviour makes it possible for errors at $\Delta T_{\text{exh}} = +20°\text{C}$ to be smaller than at $\Delta T_{\text{exh}} = +10°\text{C}$.

4.2 Two pressure levels with constraint on the steam turbine

These are the results from the correlation study of the case where the steam turbine inlet temperature was kept constant.

$$x_{\mathrm{s}} = 1 - \frac{\big(0.9487\,\eta_{2nd} - 0.5390\big)\Delta T_{\mathrm{exh}}}{h_{\mathrm{stack,ref}}} \tag{4.3}$$

$$x_{\mathrm{H}} = 1 + \frac{\big(4.012\,\eta_{2nd} - 3.333\big)\Delta T_{\mathrm{exh}}}{h_{\mathrm{mean}}} \tag{4.4}$$

As can be seen, the correction factors in eqs.(4.3-4.4) are of similar scale of magnitude as previously but with varying slope indicating a different correlation between the second law efficiency and the correction factors.

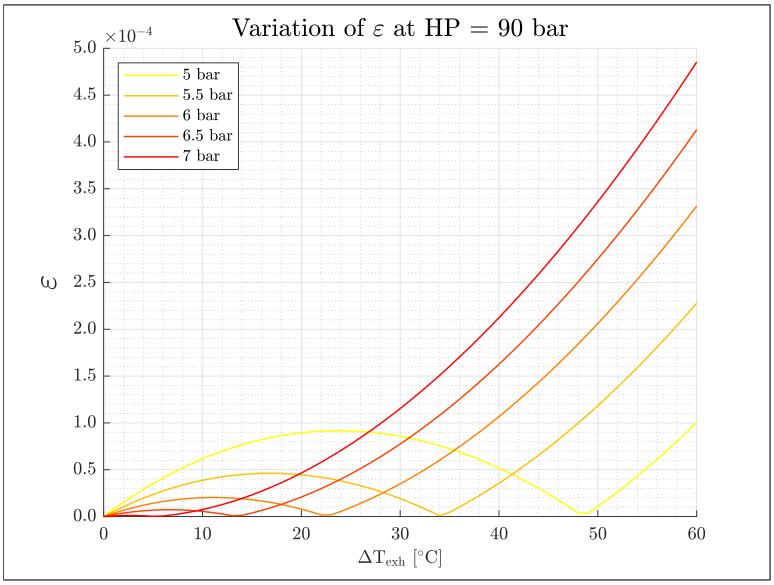

Figure 4.4: Variation of the error at different LPs with a constant HP of 90 bar.

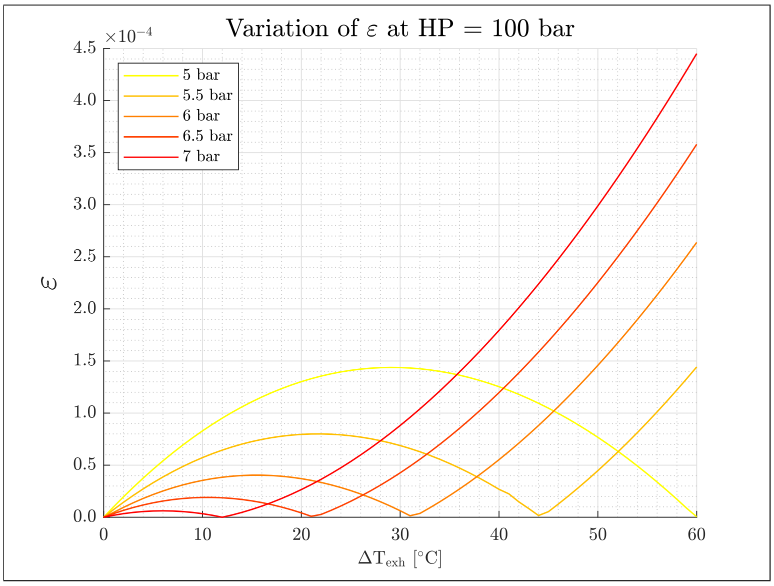

Figure 4.5: Variation of the error at different LPs with a constant HP of 100 bar.

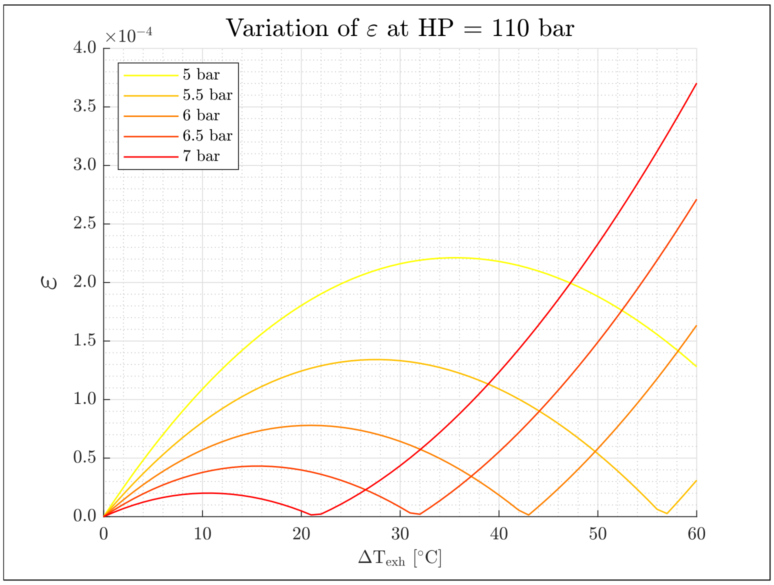

Figure 4.6: Variation of the error at different LPs with a constant HP of 110 bar.

Figures 4.4-4.6 shows the behaviour of the error for different combinations of pressure levels. Once again the exponential increase of the error is present.

	ε **at HP = 90 bar**		ε **at HP = 100 bar**		ε **at HP = 110 bar**	
Type of correction	**+10°C**	**+20°C**	**+10°C**	**+20°C**	**+10°C**	**+20°C**
No correction	$6.18 \cdot 10^{-4}$	$1.22 \cdot 10^{-3}$	$6.39 \cdot 10^{-4}$	$1.26 \cdot 10^{-3}$	$6.40 \cdot 10^{-4}$	$1.27 \cdot 10^{-3}$
Correction of T_{H}	$6.03 \cdot 10^{-4}$	$1.16 \cdot 10^{-3}$	$6.22 \cdot 10^{-4}$	$1.19 \cdot 10^{-3}$	$6.52 \cdot 10^{-4}$	$1.25 \cdot 10^{-3}$
Correction of h_{stack}	$3.21 \cdot 10^{-4}$	$4.14 \cdot 10^{-4}$	$2.77 \cdot 10^{-4}$	$3.51 \cdot 10^{-4}$	$2.17 \cdot 10^{-4}$	$2.75 \cdot 10^{-4}$
Both corrections	$5.53 \cdot 10^{-6}$	$7.97 \cdot 10^{-6}$	$3.40 \cdot 10^{-6}$	$4.18 \cdot 10^{-6}$	$2.01 \cdot 10^{-6}$	$4.88 \cdot 10^{-6}$

Table 2: Highlights values from Figures 4.4 - 4.6 and shows the error of the optimised case.

By studying Tables 1 and 2 it is evident that the magnitude of the errors are comparable and the improvement of adding the correction factors is clearly visible.

4.3 Two pressure levels with constraint on the steam turbine during off-design

As previously mentioned, there's a dependency on the correction factors when having a fixed power plant with a varying exhaust gas mass flow. Thereof, four correction factors are required, two for each parameter. The correction factors will have subscripts m and T indicating if they are dependent on the exhaust mass flow or temperature. The resulting correction factor would thus take both mass flow and temperature variation into account assuming that these are two completely independent changes.

4.3.1 Variation in mass flow of exhaust gas

$$x_{\mathrm{s,m}} = 1 - \frac{\big(-1.680\,\eta_{2nd} + 0.7399\big)\Delta\dot{m}_{\mathrm{exh}}}{h_{\mathrm{stack,ref}}} \tag{4.5}$$

$$x_{\mathrm{H,m}} = 1 + \frac{\big(-4.513\,\eta_{2nd} + 3.055\big)\Delta\dot{m}_{\mathrm{exh}}}{h_{\mathrm{mean}}} \tag{4.6}$$

The coefficients in eqs.(4.5) before η_{2nd} for both correction factors are greater than before indicating that there is a bigger influence of changing the mass flow for this scenario.

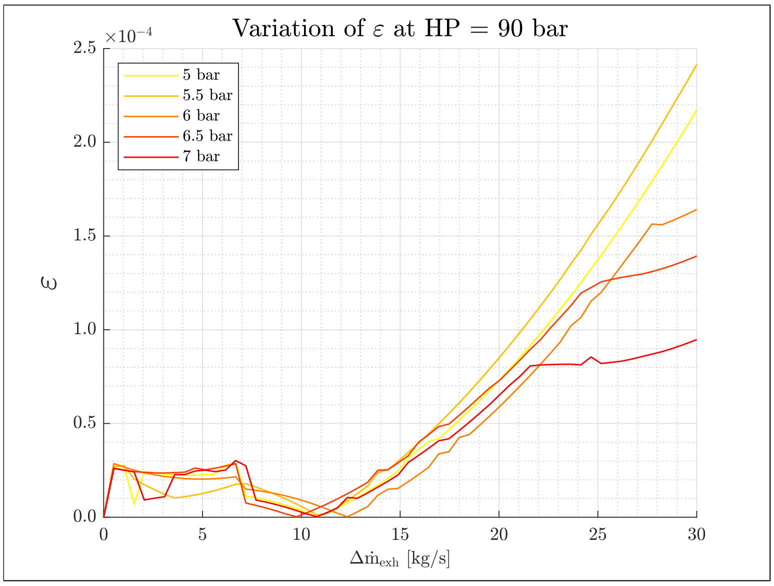

Figure 4.7: Variation of the error at different LPs with a constant HP of 90 bar.

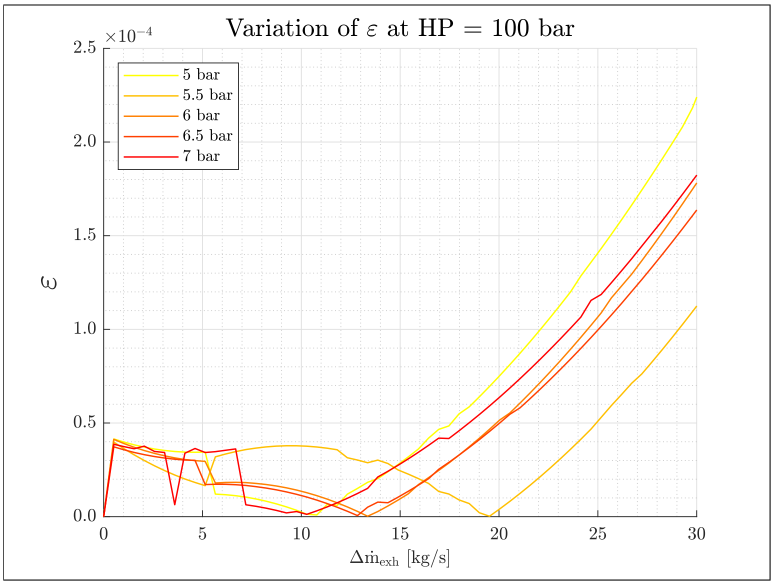

Figure 4.8: Variation of the error at different LPs with a constant HP of 100 bar.

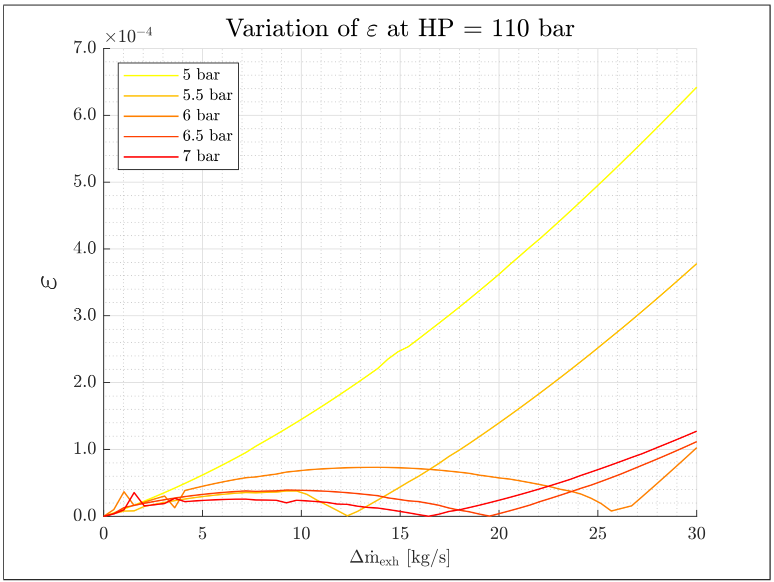

Figure 4.9: Variation of the error at different LPs with a constant HP of 110 bar.

The lower quality of the collected data is obvious from Figures 4.7-4.9. The disturbance is visible and creates uncertainties in the accuracy but does not affect the magnitude of the error dramatically since the values of the error are still relatively small.

	ε at HP = 90 bar		**ε at HP = 100 bar**		**ε at HP = 110 bar**	
Type of correction	**+5kg/s**	**+10kg/s**	**+5kg/s**	**+10kg/s**	**+5kg/s**	**+10kg/s**
No correction	$1.48 \cdot 10^{-3}$	$2.90 \cdot 10^{-3}$	$1.50 \cdot 10^{-3}$	$2.90 \cdot 10^{-3}$	$1.49 \cdot 10^{-3}$	$2.99 \cdot 10^{-3}$
Correction of T_{H}	$5.67 \cdot 10^{-4}$	$1.10 \cdot 10^{-3}$	$5.77 \cdot 10^{-4}$	$1.08 \cdot 10^{-3}$	$5.51 \cdot 10^{-4}$	$1.13 \cdot 10^{-3}$
Correction of h_{stack}	$9.19 \cdot 10^{-4}$	$1.78 \cdot 10^{-3}$	$9.33 \cdot 10^{-4}$	$1.77 \cdot 10^{-3}$	$9.12 \cdot 10^{-4}$	$1.82 \cdot 10^{-3}$
Both corrections	$1.30 \cdot 10^{-5}$	$1.73 \cdot 10^{-6}$	$1.66 \cdot 10^{-5}$	$1.24 \cdot 10^{-6}$	$2.40 \cdot 10^{-5}$	$2.32 \cdot 10^{-5}$

Table 3: Highlights values from Figures 4.7 - 4.9 and shows the error of the optimised case.

The errors, as shown in Table 3, for this scenario are larger which is to be expected from the uncertainty of the data coming from the lower tolerance in the solver. For 90 and 100 bar there is a large uncertainty in the beginning of the datasets resulting in more significant errors.

4.3.2 Variation in temperature of exhaust gas

$$x_{\mathrm{s,T}} = 1 - \frac{\left(2.164\,\eta_{2nd} - 1.301\right)\Delta T_{\mathrm{exh}}}{h_{\mathrm{stack,ref}}} \tag{4.7}$$

$$x_{\mathrm{H,T}} = 1 + \frac{\left(6.099\,\eta_{2nd} - 4.360\right)\Delta T_{\mathrm{exh}}}{h_{\mathrm{mean}}} \tag{4.8}$$

Once again the coefficients, eqs.(4.7-4.8), are larger for this scenario indicating greater sensitivity to changes of the system.

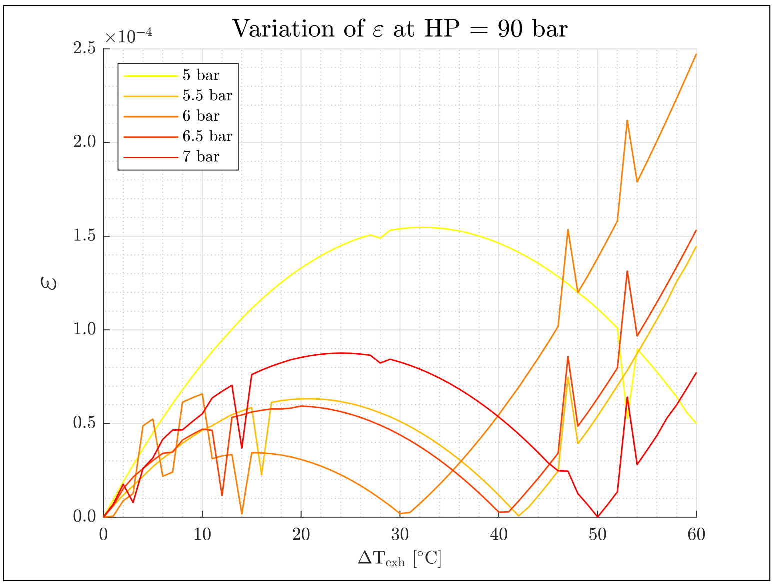

Figure 4.10: Variation of the error at different LPs with a constant HP of 90 bar.

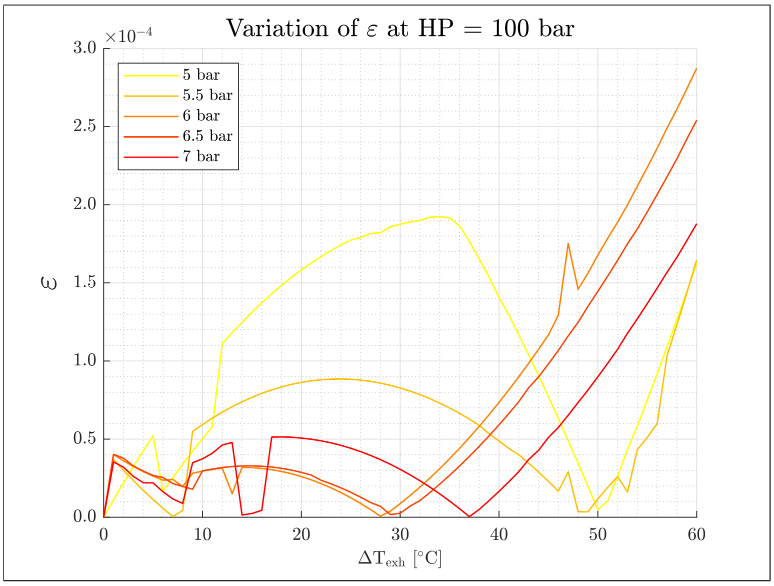

Figure 4.11: Variation of the error at different LPs with a constant HP of 100 bar.

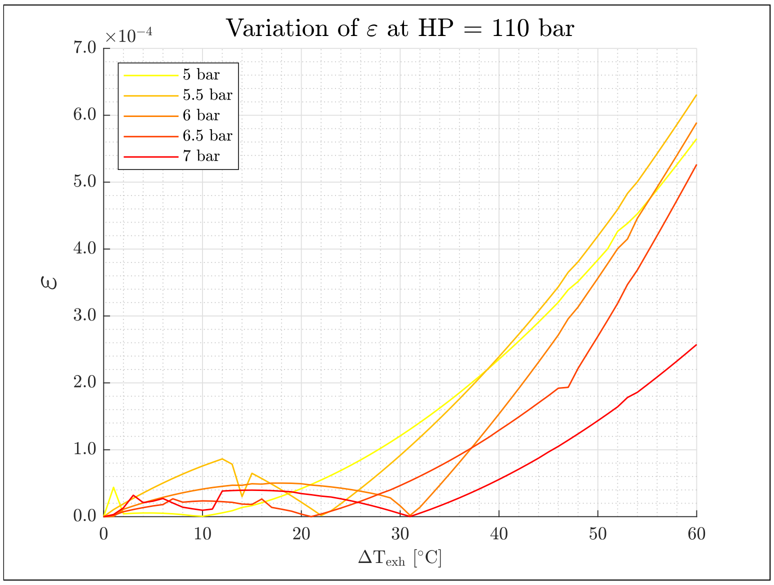

Figure 4.12: Variation of the error at different LPs with a constant HP of 110 bar.

The tolerance level is clearly affecting the data for the temperature variation case as well, as shown in Figures 4.10-4.12. Trends are still consistent to earlier cases even if there are large uncertainties.

	ε **at HP = 90 bar**		ε **at HP = 100 bar**		ε **at HP = 110 bar**	
Type of correction	**+10°C**	**+20°C**	**+10°C**	**+20°C**	**+10°C**	**+20°C**
No correction	$1.93 \cdot 10^{-4}$	$3.60 \cdot 10^{-4}$	$2.39 \cdot 10^{-4}$	$4.45 \cdot 10^{-4}$	$1.69 \cdot 10^{-4}$	$3.00 \cdot 10^{-4}$
Correction of T_{H}	$5.15 \cdot 10^{-4}$	$9.86 \cdot 10^{-4}$	$5.23 \cdot 10^{-4}$	$9.98 \cdot 10^{-4}$	$4.76 \cdot 10^{-4}$	$8.97 \cdot 10^{-4}$
Correction of h_{stack}	$2.76 \cdot 10^{-4}$	$5.67 \cdot 10^{-4}$	$2.56 \cdot 10^{-4}$	$5.31 \cdot 10^{-4}$	$3.08 \cdot 10^{-4}$	$6.43 \cdot 10^{-4}$
Both corrections	$4.63 \cdot 10^{-5}$	$3.08 \cdot 10^{-5}$	$2.495 \cdot 10^{-5}$	$2.60 \cdot 10^{-5}$	$5.23 \cdot 10^{-7}$	$3.72 \cdot 10^{-6}$

Table 4: Highlights values from Figures 4.10 - 4.12 and shows the error of the optimised case.

The values shown in Table 4 indicates an unpredictability due to the degree of tolerance. The errors for an uncorrected expression is better than introducing the individual correction factors. However, when combining them the error becomes smaller but not significantly.

4.4 Three pressure levels with constraint on the steam turbine

$$x_{\mathrm{s}} = 1 - \frac{0.07819\,\Delta T_{\mathrm{exh}}}{h_{\mathrm{stack,ref}}} \tag{4.9}$$

$$x_{\mathrm{H}} = 1 + \frac{\big(3.441\,\eta_{2nd} - 2.972\big)\Delta T_{\mathrm{exh}}}{h_{\mathrm{mean}}} \tag{4.10}$$

The greatest difference between the added pressure level is the constant correction factor, eq.(4.9), of the thermodynamic state of the stack whereas the correction of T_{H} is of similar appearance as for the one for two pressure levels.

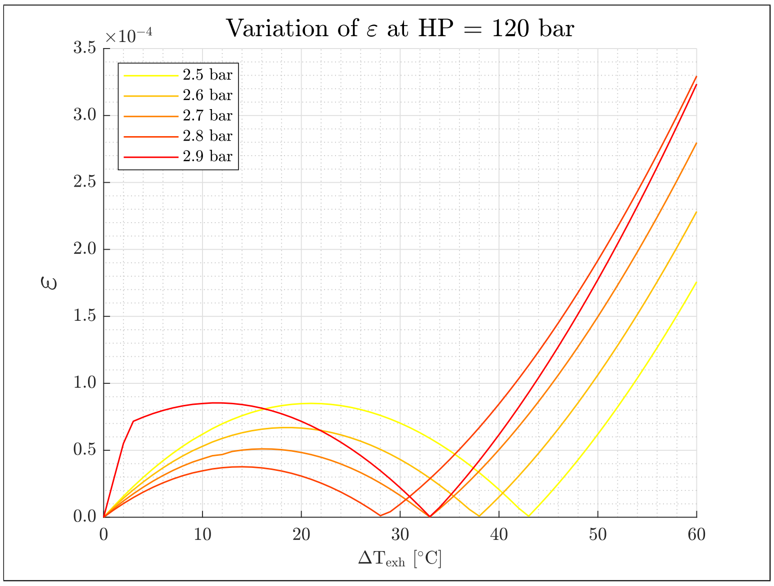

Figure 4.13: Variation of the error at different LPs with a constant HP of 120 bar.

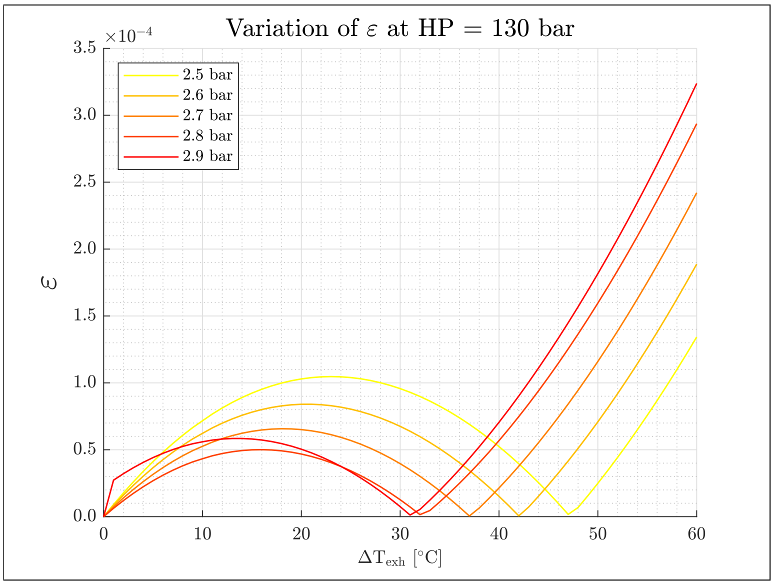

Figure 4.14: Variation of the error at different LPs with a constant HP of 130 bar.

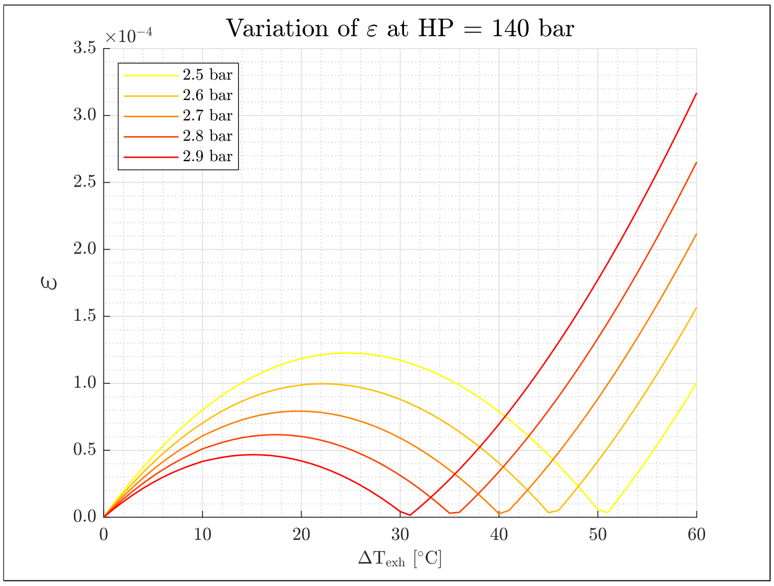

Figure 4.15: Variation of the error at different LPs with a constant HP of 140 bar.

The trends for Figures 4.13-4.15 are consistent except for the discontinuity in Figures 4.13 and 4.14 for the LP 2.9 bar. This comes from the switch in method of heating the deaerator where at 2.9 bar there is no longer steam injection resulting in another correlation for T_{H}.

	ε at HP = 120 bar		**ε at HP = 100 bar**		**ε at HP = 110 bar**	
Type of correction	**+10°C**	**+20°C**	**+10°C**	**+20°C**	**+10°C**	**+20°C**
No correction	$4.32 \cdot 10^{-5}$	$6.50 \cdot 10^{-5}$	$9.17 \cdot 10^{-5}$	$1.60 \cdot 10^{-4}$	$1.43 \cdot 10^{-4}$	$2.60 \cdot 10^{-4}$
Correction of T_{H}	$3.22 \cdot 10^{-4}$	$5.97 \cdot 10^{-4}$	$3.32 \cdot 10^{-4}$	$6.16 \cdot 10^{-4}$	$3.41 \cdot 10^{-4}$	$6.33 \cdot 10^{-4}$
Correction of h_{stack}	$2.45 \cdot 10^{-4}$	$5.03 \cdot 10^{-4}$	$1.98 \cdot 10^{-4}$	$4.10 \cdot 10^{-4}$	$1.48 \cdot 10^{-4}$	$3.13 \cdot 10^{-4}$
Both corrections	$3.47 \cdot 10^{-5}$	$3.07 \cdot 10^{-5}$	$4.33 \cdot 10^{-5}$	$4.68 \cdot 10^{-5}$	$4.17 \cdot 10^{-5}$	$4.21 \cdot 10^{-5}$

Table 5: Highlights values from Figures 4.13 - 4.15 and shows the error of the optimised case.

The errors shown in Table 5 are quite large compared to the ones for the cases with two pressure levels. Also, the errors for the uncorrected state is considerably lower than before. The reason for this is that the two uncorrected terms induce errors of opposite sign, one over-predicting and one under-predicting the efficiency. For small ΔT_{exh} these errors almost cancel each other out. When using both corrections, the error is still smaller than for the uncorrected state but much less significant.

4.5 State-of-the-art power plant

4.5.1 First order correction

$$x_s = 1 - \frac{0.1303\,\Delta T_{\text{exh}}}{h_{\text{stack,ref}}} \tag{4.11}$$

$$x_H = 1 + \frac{\left(6.998\,\eta_{2nd} - 5.807\right)\Delta T_{\text{exh}}}{h_{\text{mean}}} \tag{4.12}$$

The correction factors, eqs.(4.11-4.12), are quite different from the previous case where the coefficients have doubled in size, a result of the added reheat stage. Also, this study is done for a different set of IP levels instead of varying the LP which is kept constant.

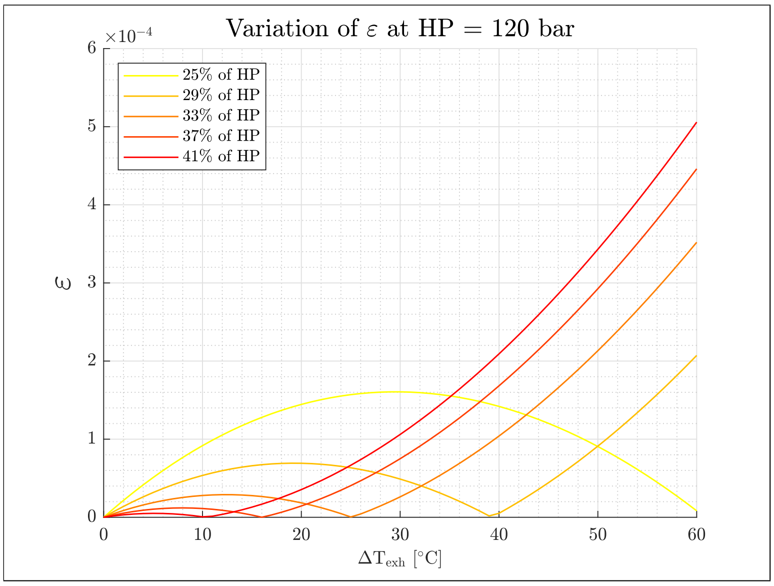

Figure 4.16: Variation of the error at different IPs with a constant HP of 120 bar.

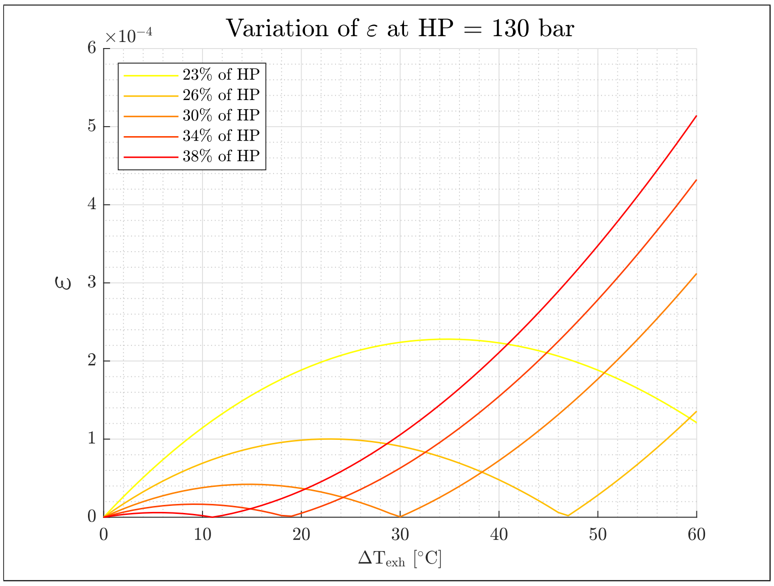

Figure 4.17: Variation of the error at different IPs with a constant HP of 130 bar.

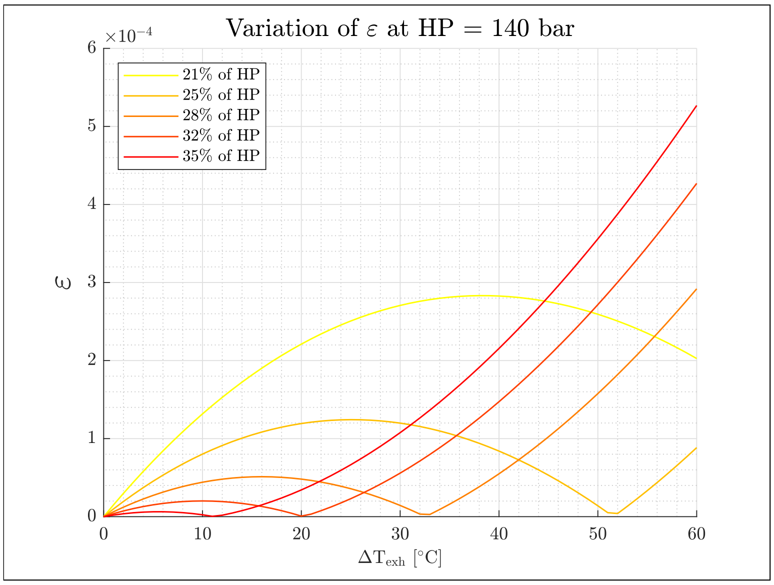

Figure 4.18: Variation of the error at different IPs with a constant HP of 140 bar.

Figures 4.16-4.18 show that the error is, once again, increasing exponentially. Also, the negative effect of changing the IP from the optimal is substantial.

	ε at HP = 120 bar		ε at HP = 130 bar		ε at HP = 140 bar	
Type of correction	**+10°C**	**+20°C**	**+10°C**	**+20°C**	**+10°C**	**+20°C**
No correction	$1.41 \cdot 10^{-4}$	$2.56 \cdot 10^{-4}$	$1.97 \cdot 10^{-4}$	$3.67 \cdot 10^{-4}$	$2.49 \cdot 10^{-4}$	$4.68 \cdot 10^{-4}$
Correction of T_{H}	$5.17 \cdot 10^{-4}$	$9.81 \cdot 10^{-4}$	$5.29 \cdot 10^{-4}$	$1.00 \cdot 10^{-3}$	$5.37 \cdot 10^{-4}$	$1.02 \cdot 10^{-3}$
Correction of h_{stack}	$3.48 \cdot 10^{-4}$	$7.09 \cdot 10^{-4}$	$2.93 \cdot 10^{-4}$	$6.01 \cdot 10^{-4}$	$2.43 \cdot 10^{-4}$	$5.01 \cdot 10^{-4}$
Both corrections	$4.24 \cdot 10^{-7}$	$1.92 \cdot 10^{-5}$	$5.46 \cdot 10^{-6}$	$2.81 \cdot 10^{-5}$	$9.91 \cdot 10^{-6}$	$2.05 \cdot 10^{-5}$

Table 6: Highlights values from Figures 4.16 - 4.18 and shows the error of the optimised IP.

Table 6 shows that the correction is working as desired. Moreover, it is once again clear that the errors induced by keeping the unknowns constant counter each other resulting in a relatively small error for the uncorrected scenario.

4.5.2 Second order correction

$$x_{\mathrm{s}} = 1 - \frac{0.1303\,\Delta T_{\mathrm{exh}}}{h_{\mathrm{stack,ref}}} \tag{4.13}$$

$$x_{\mathrm{H}} = 1 + \frac{\left(-0.0076499\,\eta_{2nd} + 0.0054793\right)\left(\Delta T_{\mathrm{exh}}\right)^2 + \left(7.228\,\eta_{2nd} - 5.971\right)\Delta T_{\mathrm{exh}}}{h_{\mathrm{mean}}} \tag{4.14}$$

The correction of the stack, eq.(4.13), remains the same but the correction for the mean temperature of heat addition is altered to capture the quadratic behaviour to get more exact solutions.

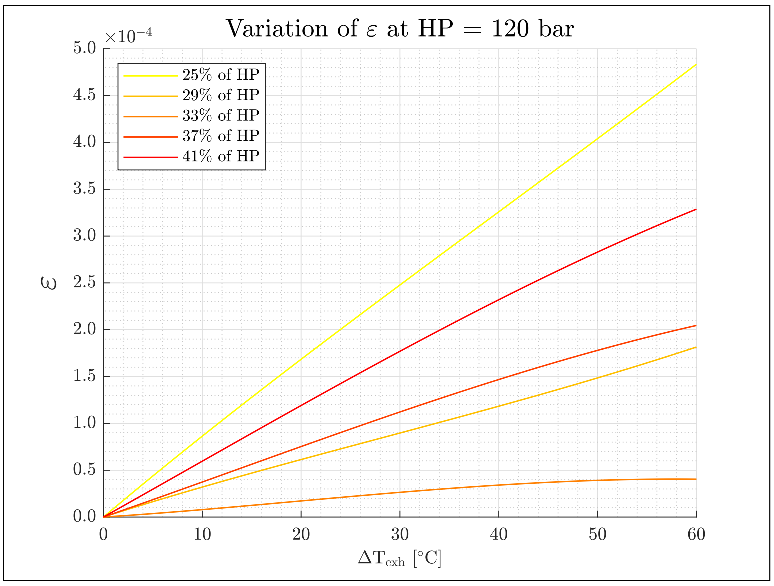

Figure 4.19: Variation of the error at different IPs with a constant HP of 120 bar.

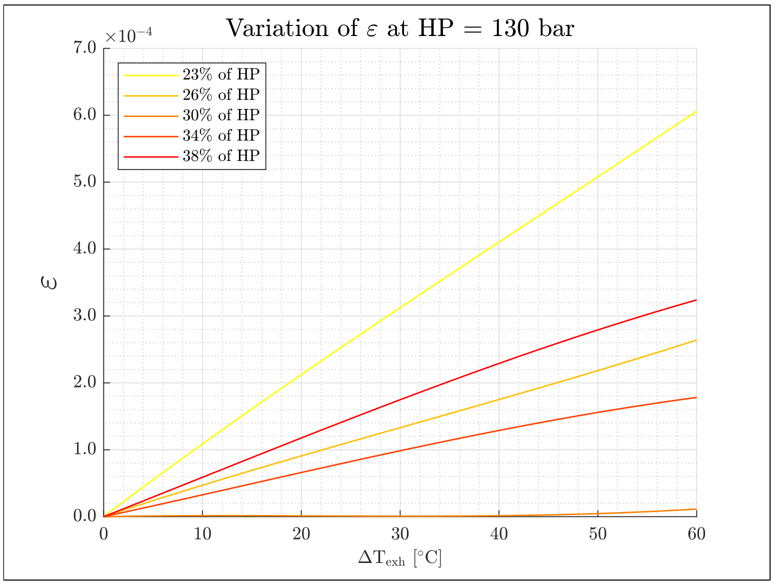

Figure 4.20: Variation of the error at different IPs with a constant HP of 130 bar.

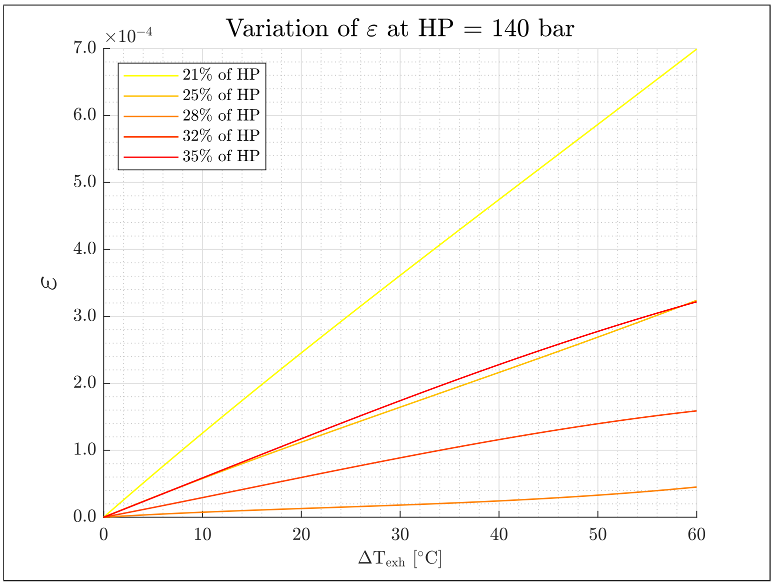

Figure 4.21: Variation of the error at different IPs with a constant HP of 140 bar.

Figures 4.19-4.21 show that the error is now behaving linearly due to the correction taking the non-linearity of $T_{\mathrm{H}} - h_{\mathrm{mean}}$ into account. When deviating from the designated optimum IP the error increases rapidly.

	ε at HP = 120 bar		ε at HP = 130 bar		ε at HP = 140 bar	
Type of correction	**+10°C**	**+20°C**	**+10°C**	**+20°C**	**+10°C**	**+20°C**
No correction	$1.41 \cdot 10^{-4}$	$2.56 \cdot 10^{-4}$	$1.97 \cdot 10^{-4}$	$3.67 \cdot 10^{-4}$	$2.49 \cdot 10^{-4}$	$4.68 \cdot 10^{-4}$
Correction of T_{H}	$4.81 \cdot 10^{-4}$	$9.45 \cdot 10^{-4}$	$4.91 \cdot 10^{-4}$	$9.66 \cdot 10^{-4}$	$4.98 \cdot 10^{-4}$	$9.80 \cdot 10^{-4}$
Correction of h_{stack}	$3.48 \cdot 10^{-4}$	$7.09 \cdot 10^{-4}$	$2.93 \cdot 10^{-4}$	$6.01 \cdot 10^{-4}$	$2.43 \cdot 10^{-4}$	$5.01 \cdot 10^{-4}$
Both corrections	$7.92 \cdot 10^{-6}$	$1.72 \cdot 10^{-5}$	$1.47 \cdot 10^{-6}$	$1.20 \cdot 10^{-5}$	$7.48 \cdot 10^{-6}$	$1.30 \cdot 10^{-5}$

Table 7: Highlights values from Figures 4.19 - 4.21 and shows the error of the optimised IP.

It is of interest to compare the results from Table 6 and 7 to get an understanding of the apparent effect of choosing a quadratic correction instead of a linear. The magnitudes of the errors are quite similar but by checking the values of the error when only the correction of T_{H} is present it is noticeable that all errors are better for the quadratic correction. The stability across a wider range of ΔT_{exh} is also preferable. It is imperative, however, that the power plant doesn't deviate from the optimum due to the rapidly increasing errors.

4.6 Using corrections for different pressure levels

In order to determine if it would be possible to use the correction factors for a two-pressure on a three-pressure CC with a similar scenario, the correction factors as seen in eqs.(4.5-4.6) are used for the scenario with three pressure levels with no reheat. These two models both have a steam turbine with constraint in inlet temperature and the difference in behaviour of the mean temperature of heat addition will therefore come from the addition of a pressure level. The results of this is presented in table 8 as well as the resulting error without any correction.

	ε **at HP = 120 bar**		ε **at HP = 130 bar**		ε **at HP = 140 bar**	
Type of correction	**+10°C**	**+20°C**	**+10°C**	**+20°C**	**+10°C**	**+20°C**
No correction	$4.32 \cdot 10^{-5}$	$6.50 \cdot 10^{-5}$	$9.17 \cdot 10^{-5}$	$1.60 \cdot 10^{-4}$	$1.43 \cdot 10^{-5}$	$2.60 \cdot 10^{-4}$
Both corrections	$2.69 \cdot 10^{-4}$	$6.10 \cdot 10^{-4}$	$2.89 \cdot 10^{-4}$	$6.12 \cdot 10^{-4}$	$2.92 \cdot 10^{-4}$	$6.17 \cdot 10^{-4}$

Table 8: Shows the values of the error for the uncorrected estimation as well as when the correction factors for the two-pressure case are used for estimating the efficiency of the three-pressure.

It is evident from Table 8 that using the correction factors for more than one scenario is not recommended since leaving it uncorrected results in a more accurate estimation.

5 Discussion

This section will address the validity of the choices made in the methodology section, as well as providing a discussion of the appearances and accuracy of the results. The methodology has to some extent already been reviewed but an overall discussion will be presented below. First, however, the project's approach will be evaluated.

5.1 Approach

The aim of the project was to conduct studies in order to be able to construct a method to determine the efficiency of a combined cycle from only gas turbine exhaust data. Due to the material at hand, IPSEpro and the accompanying macro scripts, many various power plants were modelled and studied. The trouble was not to do a correlation study of a single power plant but instead to determine how the structure of a power plant would alter the correction factors. This means that a vast selection of structures could be analysed which would be an interesting study but too time-consuming for this thesis. The decision that was made was to focus more on the method and its validity which ended up feeling like the correct decision.

5.2 Methodology

Measures were taken to validate the method as thoroughly as possible where the behaviour of the mean temperature of heat addition was closely studied. After understanding how parameters affected the mean temperature it became clear how models should be created in order to reach desired results. The general structure of the models and the range of values for key parameters were validated against drawings of existing power plants with two and three pressure levels. As was expected, the correlations were different for all models which originates from that depending on the structure of the power plant, the mean temperature will change in a unique way. Therefore, a correlation study was needed for each case.

Furthermore, the reasoning behind using the second law efficiency to relate variations in pressure was first theoretically hypothesised and later on shown with numerous figures. Due to the impact on the mean temperature by a change in HP it proved to be the key to get a considerably general solution. This is also the main difference between this thesis project and the one that was done previously [1]. The relation was continuous for all cases, however with a varying degree of correlation which is caused by the optimisation process that takes more aspects into consideration. Another improvement was the alteration of the correction of the change in mean temperature. By choosing the mean enthalpy state of the exhaust gas the effect of the changing enthalpy state of the stack was to some extent taken into account. This resulted in a more accurate result for cases with a sub-optimal LP.

The model for the off-design scenario provided difficulties where a choice had to be made between the accuracy of model parameter and solution convergence. By choosing a greater accuracy of the model the consistency could be kept throughout the parameter variation which led to the possibility of getting a correlation. This, however, resulted in faulting accuracy which raises questions about the credibility of the results. Nevertheless, if the other choice had been made, no correlation could have been found and it would not be possible to obtain any results.

5.3 Results

The magnitude of the errors that are obtained are, in a relatively wide range of deviation of exhaust gas temperature, very satisfying. To be able to determine the combined cycle efficiency down to a thousandth of a percent (error of $1 \cdot 10^{-5}$) is deemed very accurate. There are of course errors much larger than these values but it needs to be pointed out that some combinations of HP, IP and LP are completely irrelevant and was only shown to illustrate the precision of the method. Because, when designing pressure levels they will be in the vicinity of the optimal values. For instance, the IP in the state-of-the-art power plant can be designed to the exact optimum since there will be another steam turbine entrance. Therefore, the most important values are the ones for the optimised case which are all acceptable for most ΔT_{exh}. The decision to change the correlation for h_{mean} instead of T_{exh} improved the errors when altering the LP.

Moreover, even if the values of exhaust gas temperature and mass flow are all realistic, the range of the variation is not. In practice, only smaller alterations will be possible since a too high increase in parameters will lead to an upgrade in other components of the power plant. This thesis is meant to study the role of upgrading the steam turbine and does not take other aspects into account.

When taking into account the quadratic behaviour of $T_{\mathrm{H}} - h_{\mathrm{mean}}$, the errors became much more stable for the optimum case which is favourable. Nevertheless, the quadratic dependency makes it unstable in terms of changes from the optimum which could result in large errors. It is of great importance to explore this further in terms of validation of the results.

The errors for the off-design scenario are of substantially larger magnitude which does raise question about the accuracy and the usability of the method for this case. The difficulties originate from the bigger tolerance in the solver as well as not setting the pressure levels, but keeping the turbine geometries constant. Because of the variability of the pressure levels, it is difficult to capture just one behaviour since, due to the bigger tolerance, different solutions can be found. This is seen in Figures 4.7 - 4.12 as the discontinuities between data points. If more precise data could be extracted from IPSEpro, the method would most likely be more successful. The trends, however, are still comparable to the other figures, meaning that the method is working, but the gathered data is erroneous.

6 Conclusion

This project aimed to present a method to determine the CC efficiency from GT exhaust data mainly for a design scenario. This method has been presented and the following results proves that the method is stable for an optimal power plant. It is important to note that the method has not been validated against any real power plant and has only been tested on representational models made in IPSEpro. The extensive use and continuous success in using the method for estimating the efficiency for different constellations implies its applicability in being used for real power plants as well. The discovery of the dependency of HP on the second law efficiency was the key to obtain this degree of applicability.

The error is behaving in a quadratic manner and it is thereof important to weigh in the possibility of large errors if the power plant proves to be sub-optimal. If that can accounted for, the best method is to use the quadratic correction. However, if a rougher estimation is sufficient and the range of parameters does not vary too widely, the linear correction factors are proved to be accurate and quite resistant to deviations from the optimal scenario.

When using correction factors from similar but not identical scenarios, the estimated error becomes large and it is therefore concluded that it is not favourable to use other correction factors than the ones representing the exact scenario.

6.1 Sources of errors

Due to the many scenarios and models that have been subject to analysis many errors have been eliminated such as inconsistent models when analysing correlations. The macro scripts in IPSEpro made it possible to eliminate human errors when applying values to key parameters due to the script doing so for each dataset.

The main error that remains is the convergence issue when analysing the off-design scenario. It makes the results questionable. The source of the error is possible to remove by constructing a more stable model.

6.2 Future work

This section will discuss what ideas and thoughts are left after finishing this project and studies that will improve on what has been conducted.

- As has been discussed, the method needs to be more extensively validated in order to determine the applicability in using it for real power plants. By using the presented method it is possible to study a power plant of choice and make correction factors for it. For every added component it is recommended to study the new behaviour in order to not get a faulting estimation.
- Due to the success of the quadratic correction this study should be conducted for the remaining cases in order to present a better estimation for these scenarios.
- As has been discussed, there is no favour in using a correction factor that is not meant for one scenario to estimate the efficiency. It is therefore critical to make sure that correction factors for a power plant have been presented before using them.

7 References

[1] Simon Frick. Relating gas turbine performance to combined cycle efficiency. Master's thesis, Lund University, 2018.

[2] Y. A. Çengel and M. A. Boles. *Thermodynamics - An Engineering Approach in SI Units.* McGraw-Hill Education, New York, NY, USA, 8th edition, 2015.

[3] Rolf Kehlhofer, Bert Rukes, Frank Hannemann, and Franz Stirnimann. *Combined-Cycle Gas Steam turbine Power Plants.* PennWell Corporation, Tulsa, OK, USA, 3rd edition, 2009.

[4] M. A. El-Masri. Design of gas turbine combined cycle and cogeneration systems - theory practice and optimization. *Seminar notes, Thermoflow, Inc., Sadbury*, 2007.

[5] S Can Gülen and Raub W. Smith. Second law efficiency of the rankine bottoming cycle of a combined cycle power plant. *Journal of Engineering for Gas Turbines and Power-transactions of The Asme - J ENG GAS TURB POWER-T ASME*, 132, 01 2010.

[6] Alessandro Franco and Alessandro Russo. Combined cycle plant efficiency increase based on the optimization of the heat recovery steam generator operating parameters. *International Journal of Thermal Sciences*, 41(9):843 – 859, 2002.

[7] Bassam G. Jabboury and Mohamed A. Darwish. The effect of the operating parameters of heat recovery steam generators on combined cycle/sea-water desalination plant performance. *Heat Recovery Systems and CHP*, 10(3):255 – 267, 1990.

[8] Mohammad Tajik Mansouri, Pouria Ahmadi, Abdolsaeid Ganjeh Kaviri, and Mohammad Nazri Mohd Jaafar. Exergetic and economic evaluation of the effect of hrsg configurations on the performance of combined cycle power plants. *Energy Conversion and Management*, 58:47 – 58, 2012.

[9] Tangellapalli Srinivas, Shivani Gupta, and B V Reddy. Thermodynamic modeling and optimization of multi-pressure heat recovery steam generator in combined power cycle. *Journal of Scientific and Industrial Research*, 67, 10 2008.

[10] H. I. H. Saravanamuttoo, G. F. C. Rogers, H. Cohen, and P. V. Straznicky. *Gas Turbine Theory.* Pearson Education Ltd, Seventh Edition, 2017.

[11] S Can Gülen. *Gas Turbines for Electric Power Generation*. Cambridge Univeristy Press, First edition, 2019.

[12] Tangellapalli Srinivas, B V. Reddy, and Shivani Gupta. Parametric simulation of combined cycle power pant: A case study. *International Journal of Thermodynamics*, 14, 03 2011.

[13] Benson® technology. https://new.siemens.com/global/en/products/energy/power-generation/power-plants/benson-technology.html. Accessed: 2019-05-17.

[14] M.A. Habib, S.A.M. Said, and I. Al-Zaharna. Thermodynamic optimization of reheat regenerative thermal-power plants. *Applied Energy*, 63(1):17 – 34, 1999.