

Evaluation of sub- and supercritical Rankine cycle optimisation criteria

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Declaration

I Petrus Johannes Pieters (8704285127089) hereby declare that the work done in this dissertation is a presentation of my own work (research and programming). Wherever I made use of the work of others, I made every effort to indicate this clearly. Some of the information contained in this dissertation has been gained from various journal articles; text books etc., and has been referenced accordingly.

The work was done under the guidance of Professor Chris Storm, at the North-West University (Potchefstroom Campus).



PJ Pieters

29 April 2016

Date

In my capacity as supervisor of this dissertation, I certify that the above statements are true to the best of my knowledge.



Prof CP Storm

29 April 2016

Date

Abstract

The main purpose for this study has been to model an advanced real Rankine cycle for sub and super-critical boilers with all the components as encountered in the industry mathematically and to optimise each cycle.

First the Development of the Rankine cycle is illustrated from the most effective theoretical Carnot cycle through to an advanced ideal Rankine cycle with feed water heating and compared to each other by means of the results obtained from EES. After the Ideal Rankine cycle with all the relevant components had been programmed and discussed the Cycle was further developed into an advanced real Rankine cycle.

The Advanced real Rankine cycle consists of Superheat, Reheat, two high-pressure feed heaters, a de-aerator, three (super-critical) or four (sub-critical) low-pressure feed heaters, a condenser, a condensate extraction pump and one main feed water pump. The real cycle made provision for pressure losses, efficiencies, steam attemperation and temperature losses.

The following were optimised to get the maximum efficiency and net mechanical work for each cycle:

- Feed pump maximum pressure
- High pressure turbine expansion
- Two high pressure feed water heaters
- The de-aerator
- Three or four low pressure feed water heaters

The study touches on low pressure turbine outlet steam quality, but keeps it constant through the optimisation stages.

To finish off, a comparison between sub- and super-critical Rankine cycles was done before and after optimisation.

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Table of Symbols:

LPT - Low pressure turbine

IPT - Intermediate pressure turbine

HPT - High pressure turbine

η - Efficiency

kPa - Kilopascal

MPa - Mega pascal

T - Temperature

S - Entropy

h - Enthalpy

% - Percentage

CO₂ - Carbon dioxide

W - Work

Q - Heat

MW - Megawatt

P - Pressure

EXP - Extraction pump

FDP - Feed pump

EES - Engineering Equation Solver

°C - Degrees Celsius

Kg/s - Kilogram per second

kJ - Kilo Joule

1. Introduction

1.1. Background

The Rankine cycle (named after William John Macquorn Rankine) is a two stage working fluid cycle that is mostly used with water as working fluid for steam turbine power generating systems. The cycle is a thermodynamic cycle of a heat engine that converts heat into mechanical work.

The Rankine cycle is a model that is used to predict the performance of steam turbine systems. The Rankine cycle can be improved by adding superheating and reheating to increase the thermal efficiency and net mechanical work. Regenerative feed water heating is also a way to increase the efficiency of the overall cycle.

Regarding the maximum cycle pressure, Rankine cycles are further categorised in sub- and super-critical cycles. This study focuses on both these Rankine cycles including the optimisation and comparison of both. For validation the layout, design and operating parameters of Kriel Power Station (sub-critical) and Medupi power station (super-critical) are used.

The optimisation focuses on boiler feed pump discharge pressure, high pressure turbine expansion, high pressure heater tap off pressures, de-aerator tap off pressure and low pressure heaters tap off pressure. Optimisation allows us to get an optimised point where the cycle reaches a maximum for both efficiency and net mechanical work.

1.2. Problem statement

- With better materials it is now possible to reach higher temperatures on the T-s diagram. If the maximum temperature is higher the required optimum boiler pressure also increases for the limit of the steam quality at the low pressure turbine steam quality. Eventually the required pressure increases above the critical point of water. Modelling and cycle optimisation are thus required.
- No history was found on a full scale optimisation study.

1.3. Objectives

- Compile models of each cycle to optimise the following
 - Maximum boiler pressure;
 - High pressure turbine expansion;
 - High pressure feed heater steam tap off pressure;
 - Low pressure feed heater steam tap off pressure; and
 - De-aerator steam tap off point.
- Find the maximum optimal point between efficiency and net mechanical work by using a factor line.
- Increase cycle efficiency and net mechanical work through optimisation.
- Analyse the data and make changes if necessary.
- Compare the sub-critical cycle against the super-critical cycle before and after optimisation.

1.4. Experimental procedure and research methodology

The following experimental procedures were used in support of the general research methodology adopted.

- A literature survey;
- Software research;
- A request was made for plant data from Kriel power station and Medupi power station;
- Models for sub-critical and super-critical cycles using computer software were compiled;
- Input parameters were verified against data obtained from Kriel power station and Medupi power station;
- Validation of results against data obtained from Kriel power station and Medupi power station was done;
- Each cycle was optimised with two different approaches;
- Data validation was done after optimisation and changes made; and
- A conclusion was reached.

1.5. Assumptions and limitations

The following assumptions and limitations were made during the optimisation of the sub-and super-critical cycle optimisation:

Boiler feed pump (FDP)

- The boiler feed pumps have an efficiency of 90%.
- No steam feed pump were used, only electric feed pumps.

Condenser extraction pump

- The condenser extraction pumps have an efficiency of 90%.

Condenser

- Atmospheric temperature is held constant
- The temperature in the condenser is 50°C for the super-critical cycle and 40°C for the sub critical cycle.

High pressure turbine (HPT)

- Steam quality at the low pressure turbine outlet is held constant as current design.
- The High pressure turbine has an efficiency of 93%

Intermediate pressure turbine (IPT)

- The Intermediate pressure turbine has an efficiency of 91%.

Low pressure turbine (LPT)

- The low pressure turbine has an efficiency of 85%.

High pressure heater 7 (HPH7)

- The steam tap off point is located just after the high pressure turbine (HPT) outlet on the cold reheat line, thus the high pressure turbine outlet pressure is taken as the tap off pressure for high pressure heater 7.
- The heat transfer in high pressure heater 7 is assumed to be 100% effective.

High pressure heater 6 (HPH6)

- The heat transfer in high pressure heater 6 is assumed to be 100% effective.
- The tap off point from the intermediate pressure turbine is assumed to be a maximum of 67% of the inlet pressure for intermediate pressure turbine, thus a minimum of 33% pressure loss through the turbine.
- The tap off point minimum is assumed to be the pressure at the intermediate pressure turbine outlet.

Low pressure Heaters (LP1, LP2, LP3 & LP4) *(LP4 only apply for sub-critical cycle)*

- The heat transfer in the low pressure heaters is assumed to be 100% effective.
- Tap off points from the intermediate pressure turbine is assumed to be a maximum of 67% of the inlet pressure for intermediate pressure turbine, thus a minimum of 33% pressure loss through the turbine.
- The tap off point minimum is assumed to be the pressure at the intermediate pressure turbine outlet.
- Tap off points from the low pressure turbine is assumed to be a maximum of 67% of the low pressure turbine inlet pressure, thus a minimum of 33% pressure loss in through the turbine.
- The tap off point minimum is assumed to be the pressure at the low pressure turbine outlet.
- The assumption is made that low pressure heater 4 outlet flows into low pressure heater 3.*(Only for sub-critical cycle)*
- The assumption is made that low pressure heater 3 outlet flows into low pressure heater 2.
- The assumption is made that low pressure heater 2 outlet flows into low pressure heater 1.
- The assumption is made that low pressure heater 1 outlet flows into the condenser.

1.6. Dissertation summary

- Chapter 2: The literature survey can be found in this chapter. This was done according to Kriel and Medupi power stations and also focuses on the Rankine cycle itself and previous optimisation studies.
- Chapter 3: Development of the Rankine cycle can be found in this chapter. This chapter focuses on the development of the Rankine cycle from the Carnot cycle up to the current Rankine cycle with superheat, reheat and feed water heating.
- Chapter 4: Rankine cycle programming methodology to enable optimisation can be found in this chapter. This chapter focuses on the different ways to increase the cycle efficiency and net mechanical work and the effect it will have on the rest of the cycles.
- Chapter 5: Programming of the sub- and super-critical Rankine cycle can be found in this chapter. This chapter explains how the cycles were programmed for each component and what calculations were used to obtain the results.
- Chapter 6: Results of sub- and super-critical Rankine cycles can be found here. The results are presented for each method, cycle and run.
- Chapter 7: The chapter presents the conclusion after the results were obtained and analysed.
- Chapter 8: This chapter contains references used.
- Chapter 9: This is the Appendix chapter where more background research regarding the study can be found.

2. Literature survey and existing technology

2.1. Rankine cycle

The Rankine cycle is a mathematical model of a cycle that uses mostly water as a working fluid. The water is constantly evaporated and condensed. The cycle is used to predict the performance, temperatures, pressure and quality of steam in power generating machines. Kinetic energy in the form of coal is transferred to mechanical energy to create electricity. The four stages in a Rankine cycle can be seen below on a component flow diagram (Figure 1), where water is pumped into a heat source (boiler) that transfers it into steam. The steam is fed to the turbine where steam is transferred into mechanical work. The steam is condensed into water again and fed into the pump. The water is pumped to the boiler and the cycle repeats itself.

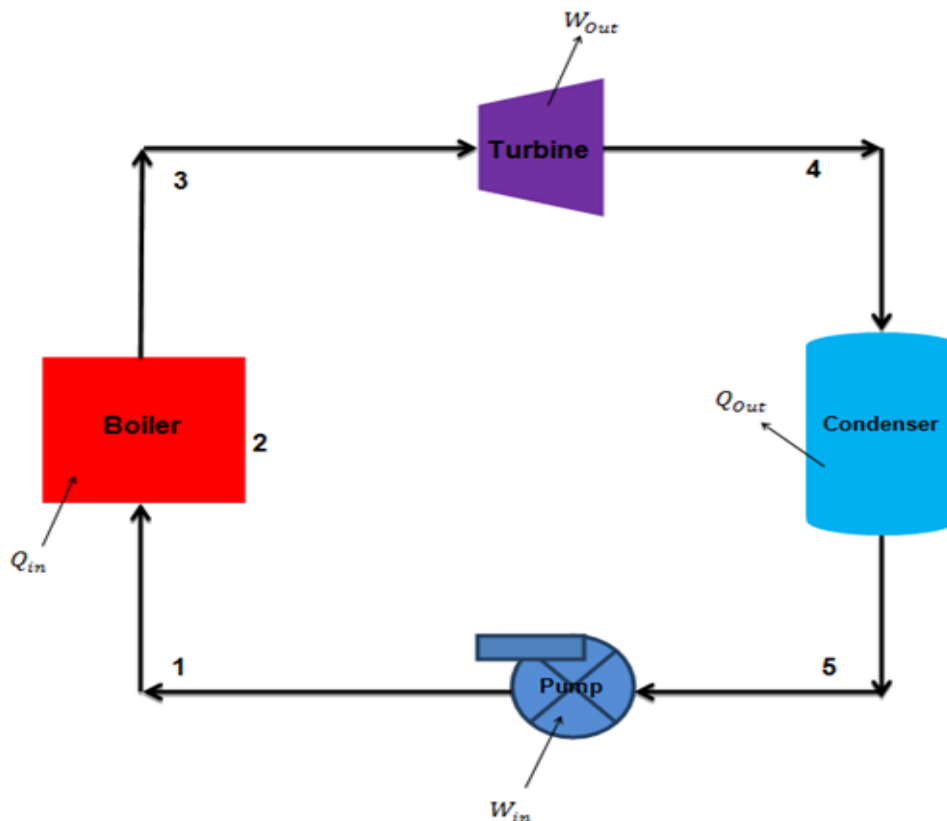


Figure 1: Rankine cycle process flow diagram

2.1.1. Stages

2.1.1.1. Isobaric (Heat Gain) (1-2-3)

This stage is where water enters the boiler as a compressed liquid (1). The water is pumped through thousands of boiler tubes where heat is transferred from the burning coal to the water. The water is heated to saturated temperature (2). More energy is transferred and the liquid evaporates into fully saturated steam (3)

2.1.1.2. Isentropic expansion (work out) (3-4)

This stage is where energy is transformed from kinetic to mechanical. The saturated steam (3) expands in the turbine. The steam is forced through the turbine blades which results in the turbine rotating. This rotating turbine is connected to a generator where mechanical energy is transferred to electrical energy. The steam loses a lot of energy before entering the condenser (4).

2.1.1.3. Isobaric (heat rejection) (4-5)

This stage is where the two phase mixture leaves the turbine and enters the condenser (4). Heat is extracted from the mixture through a heat transfer process using tubes and cold water in the condenser. The saturated water leaves the condenser and is now ready for the feed pump (5).

2.1.1.4. Isentropic compression (work in) (5-1)

This stage is where the condensed water (5) is compressed by the feed pump before entering the boiler (1). The temperature increases somewhat during the compression stage. The work input through the feed pump will be much less than the work output. The reason for this is because the volume of water is much less than the volume of steam. The compressed water has a much higher saturation point and this allows us to add a lot more energy to the water before it turns into steam.

2.2. Sub-critical Rankine cycle plant layout (Kriel)

Demineralised water is pumped into a make-up tank which feeds the de-aerator. The water is fed to the boiler feed pump which pumps the water through the economiser. After the economiser the water makes its way down to the bottom of the boiler to enter the division wall inlet. The water is pumped through the division wall to the top. The water makes its way down for a second time and runs through the spiral wall tubes to halfway up in the boiler where it enters the vertical wall all the way to the top. At the top the steam collects in the separating vessels before entering superheater 1 (during the start-up phase the water and steam mix accumulate in four separating vessels where the steam and water separate). After separation the water accumulates in the collecting vessel and is fed to the economiser again). The steam enters superheater 1 then superheater 2 and then superheater 3. After superheater 3 the steam quality is ready for the high pressure turbine. After the high pressure turbine expansion the steam loses a lot of energy and returns to the boiler where it enters Reheater 1 and then Reheater 2, then returns as superheated steam to the intermediate and low pressure turbines.

After the low pressure turbine the water and steam mixture enters the condenser where heat transfer takes place and energy is extracted from the mixture to turn it into saturated water. The water is pumped (using an extraction pump) through the low pressure heaters to raise the feed water temperature. The water then enters the de-aerator where more heat is added before it enters the feed pump. The feed pump raises the water pressure and is pumped through the high pressure heaters where more heat is added before entering the economiser.

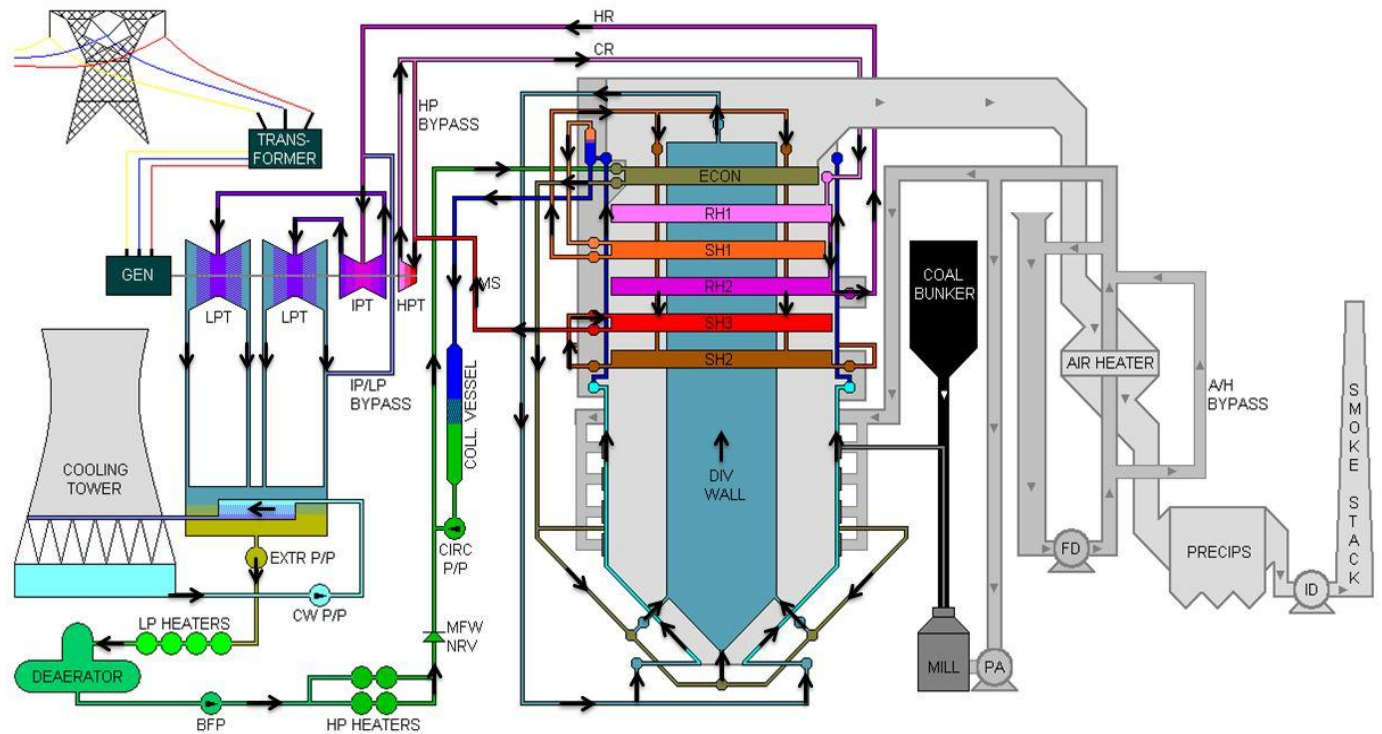


Figure 2: Illustration of Kriel Power Station flow diagram

The low pressure heaters use tap off steam from the low and intermediate pressure turbines. The de-aerator and high pressure heater 6 uses tap off steam from the intermediate turbine. High pressure heater 7 uses tap off steam from the high pressure outlet line (cold reheat line).

2.3. Super-critical Rankine cycle plant layout (Medupi)

The plant layout for Medupi power station is the same as at the layout Kriel power station is using, except the super-critical cycle does not have a low pressure heater 4 and therefore the low pressure heater 3 uses tap off steam from the intermediate turbine. The super-critical plant also makes use of air cooled condensers instead of water cooled condensers.

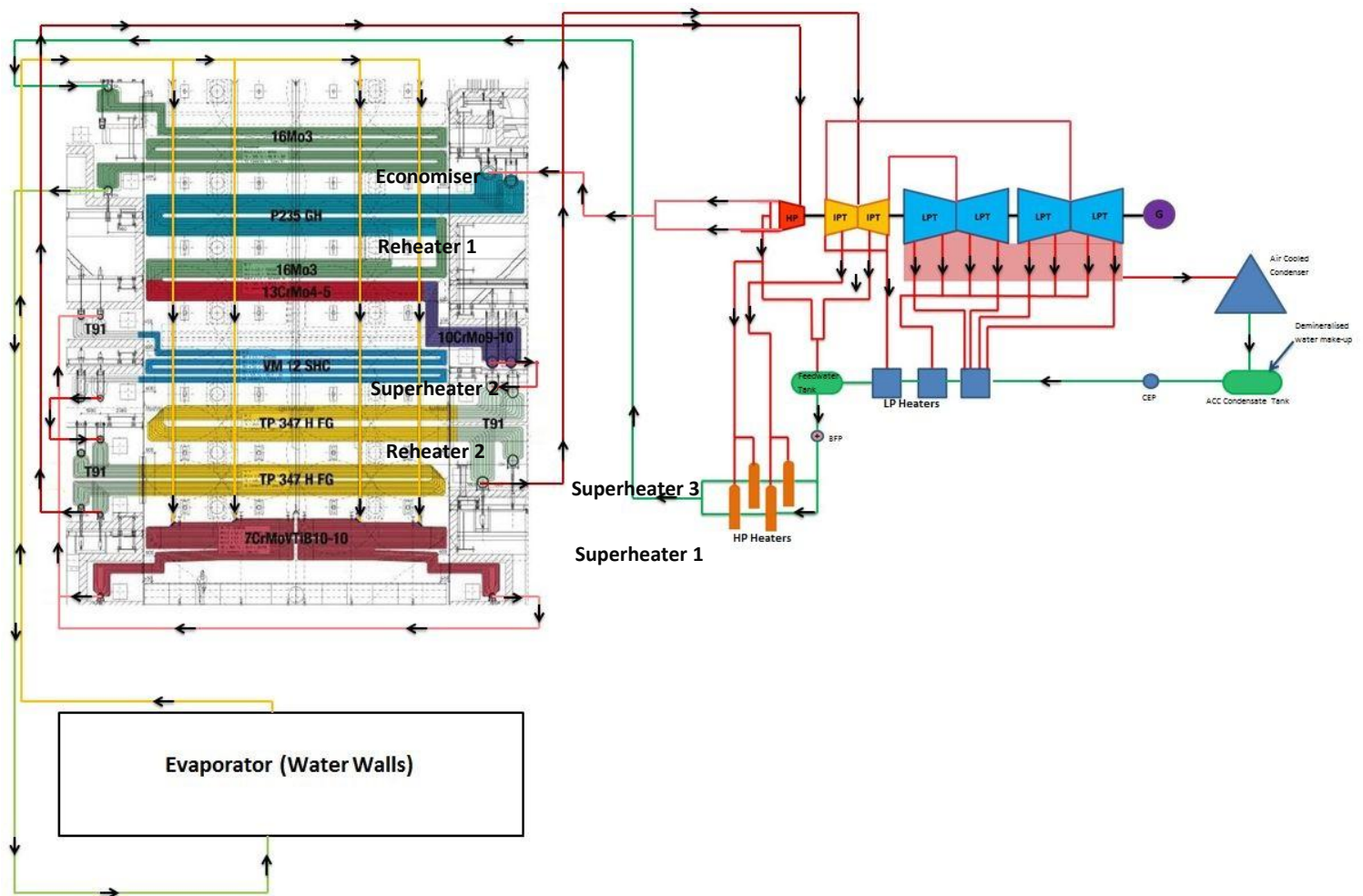


Figure 3: Illustration of Medupi Power Station flow diagram

2.4. Optimisation of Rankine cycle

2.4.1. Efficiency optimisation

To optimise the tap off pressure for the regenerative feed water heaters the pressure must be decremented from the maximum tap off pressure to the minimum tap off pressure in a certain number of intervals. Cycle efficiency is then plotted against the tap off pressure to get a trend for the different points.

The picture below illustrates that the tap off steam at maximum is not allowed to expand sufficiently in the turbine, therefore the efficiency is lower at maximum tap off pressure than at lower tap off pressures (Harshal D Akolekar, 2014).

In the picture below it can also be concluded that steam lower than 1100 kPa is less efficient to tap off. Because the energy in the steam is much lower at lower tap off pressures as it has already lost a lot of energy in the turbine expansion, therefore much less energy can be transferred to the boiler feed water.

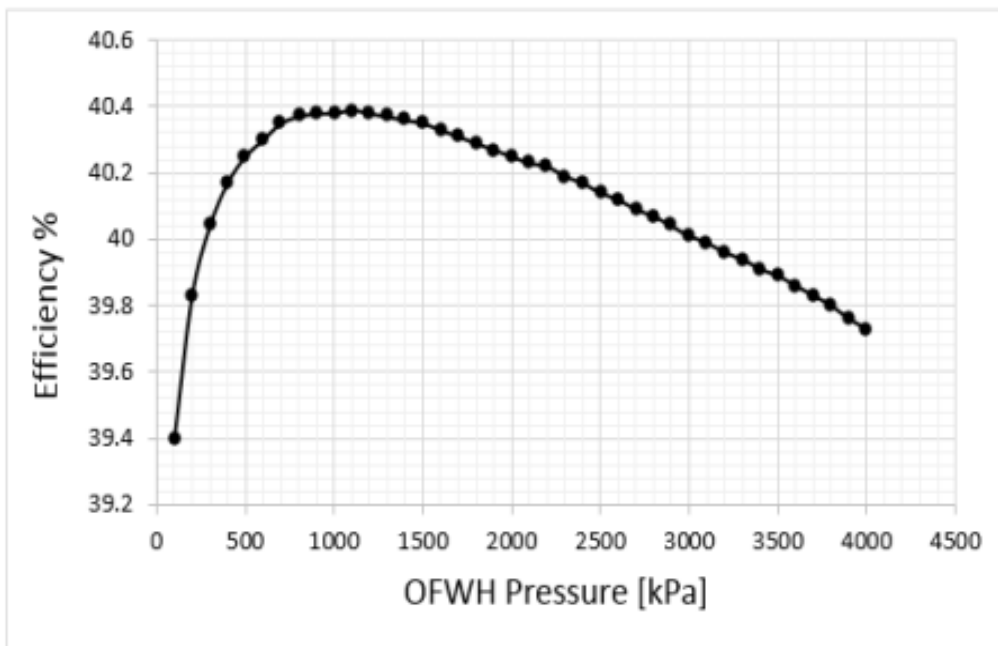


Figure 4: Cycle efficiency plotted against feed heater tap off pressure (Harshal Akolekar, 2014)

2.4.2. Multi-objective optimisation

When optimising the Rankine cycle it is important not only to look at efficiency but at mechanical work as well because mechanical work is also important as it will play a major role in generating power. In figure 5 the extraction mass flow for the feed heaters is taken from maximum to minimum mass flow. The efficiency is plotted on an x-y diagram (red line). The net mechanical work is also plotted on an x-y2 scale (blue line). Because these two variables have different scales it's important to plot a local maximum/minimum line. The local maximum/minimum line will indicate what the optimum point of these two variables is. In figure 5 below it can be seen that the efficiency keeps improving as more mass flow is taken, but this has a negative impact on the mechanical work as more energy is taken away from the turbine and less turbine expansion takes place. That is why the net mechanical work line reduces as more mass flow is tapped off.

This highlights the importance to use optimisation in the Rankine cycle

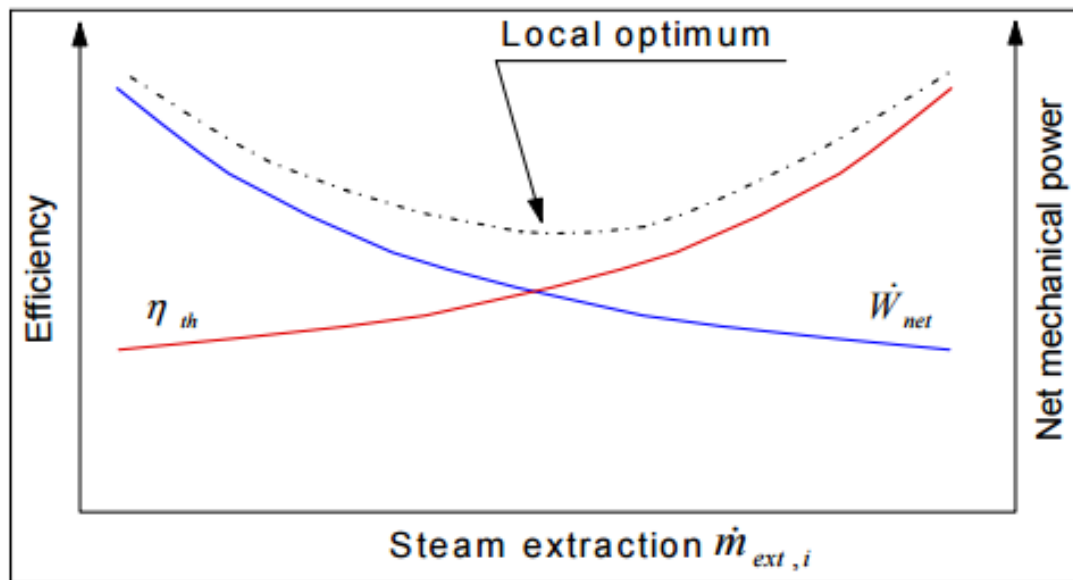


Figure 5: Cycle efficiency, net mechanical work and local minimum/maximum line plotted against steam extraction mass flow (Laure, 2011)

3. Development of the Rankine cycle and optimisation

3.1. Development of Rankine cycle

For illustration purposes the following cycles were programed on EES using Kelvin for temperature and kPa for pressure. Maximum pressure was taken as 16 000 kPa and minimum as 5 kPa.

3.1.1. Carnot cycle

To illustrate the development of the Rankine cycle we will start at the Carnot cycle. Sadi Carnot was the developer of the Carnot cycle and is applicable to any cycle such as steam and gas.

Heat is added and rejected at constant temperatures. Compression and expansion are also carried out at constant entropies. The Carnot is known as the ideal cycle as it offers maximum efficiency between temperatures of source and sink.

Cycle efficiency can be represented by the net mechanical work output divided by the total heat input.

$$\text{Cycle } \eta = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}$$

Where T_1 and T_2 are the absolute temperatures of the source and sink (Rayaprolu, 2009).

In the illustration below (figure 6) on the T-s diagram the boiler produce steam at 16MPa and condenses at 5kPa.

The Carnot cycle consists of four stages that are totally reversible.

The stages consist of:

Stage 1 – 2: Isothermal heat addition – Heat is supplied at constant temperature

Stage 2 – 3: Isentropic expansion – steam or gas expands from a high pressure and temperature to a low pressure and temperature.

Stage 3 – 4: Isothermal heat rejection – Heat is rejected at constant temperature.

Stage 4 – 1: Isentropic compression – steam or gas is compressed from a low pressure and temperature to a high pressure and temperature.

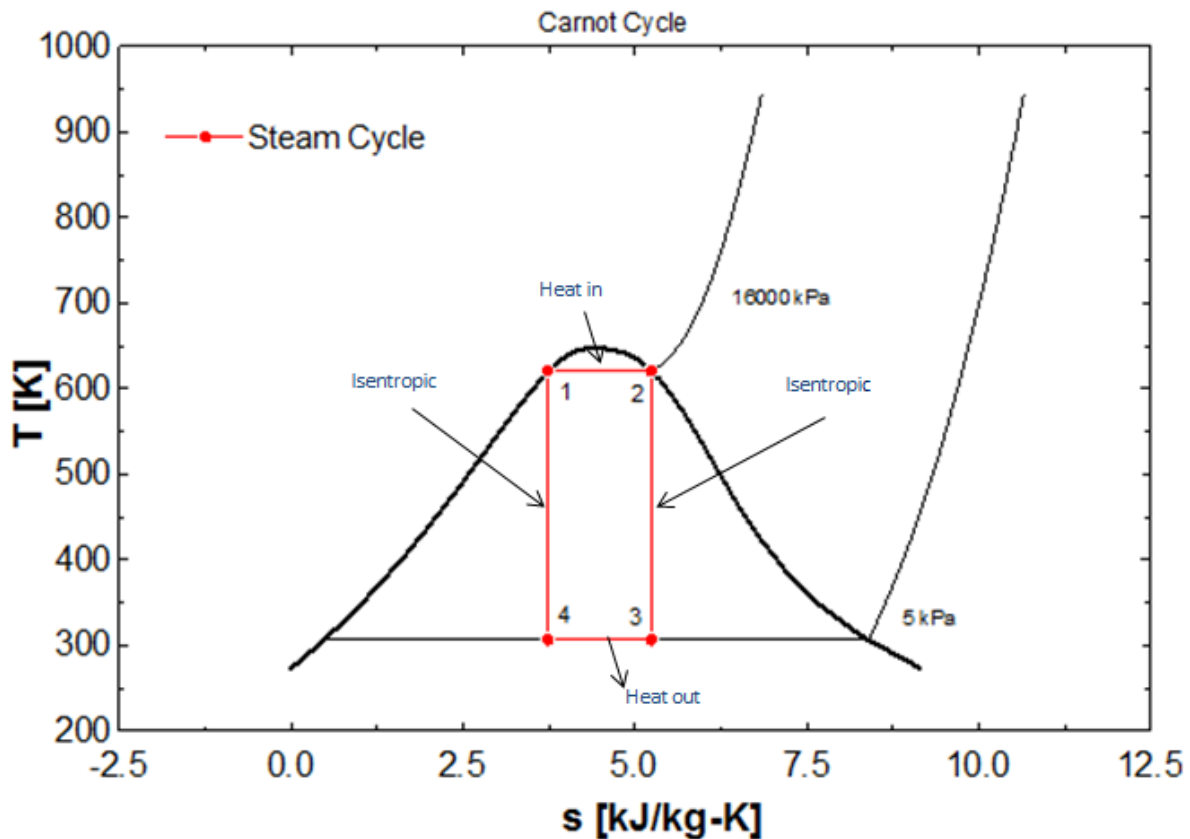


Figure 6: T-s diagram to illustrate the Carnot cycle

The Carnot cycle is not a practical cycle because of the following reasons:

- Steam quality at point 3 would corrode the low pressure turbine blades and blade replacement would become necessary regularly, and this would have a massive impact on cost and time.
- A compressor is needed that can compress steam with a low quality to a high pressure, thus a high work input is needed.
- Very large heat exchangers are needed.
- Point 4 cannot be controlled.

The real value of the Carnot cycle is the standard against which actual or ideal cycles can be compared.

By letting the steam condense fully to water (quality = 0) at point 4 the cycle evolves into 'n basic Rankine cycle that will be discussed next.

3.1.2. Basic Rankine cycle

The Rankine cycle is an idealised cycle for steam power plants and consists of the following stages.

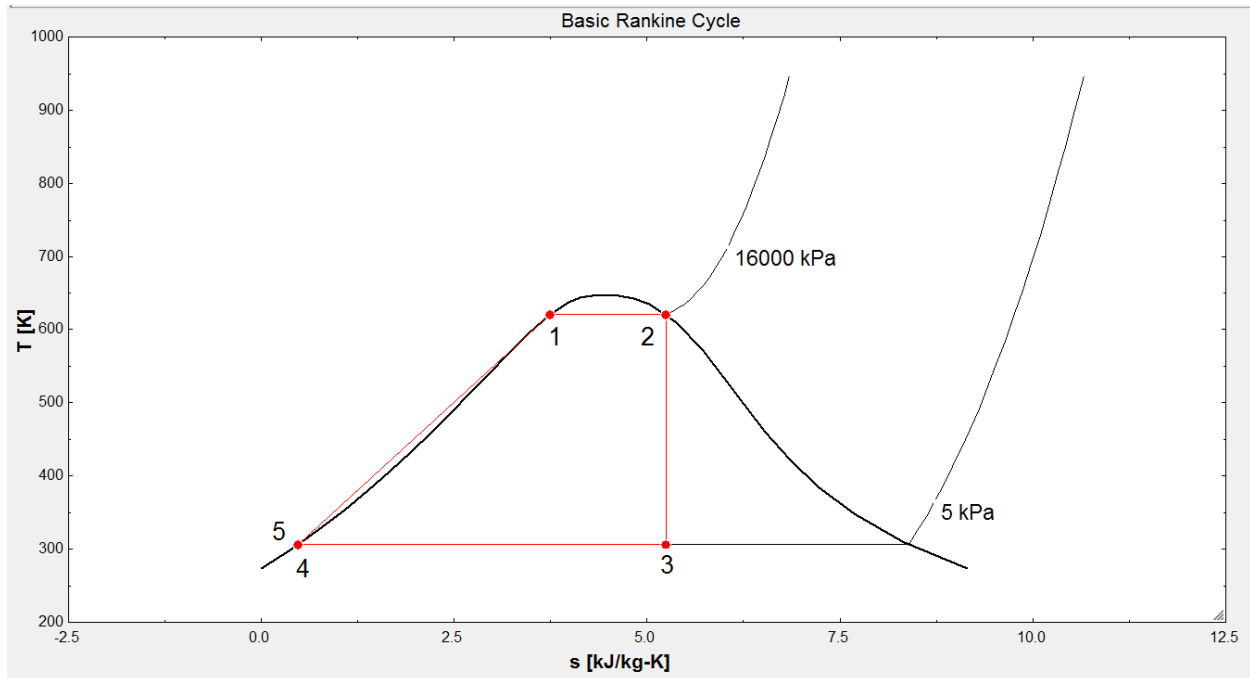


Figure 7: T-s diagram to illustrate the Basic Rankine cycle

Stages 5 - 1: Isobaric heat gain – Water enters the boiler under high pressure from the boiler feed pump. Heat is added until the water reaches saturation point (1).

Stages 1 - 2: Isobaric heat gain – Further heat is added until the steam is fully saturated.

Stages 2 - 3: Isentropic expansion – The fully saturated steam expands in a turbine, producing mechanical energy. Because of the expansion over the turbine blades the turbine rotates.

Stages 3 - 4: Isobaric heat rejection– The steam/water mixture enters the condenser where the pressure is well below atmospheric pressure. The pressure and heat transfer forces the mixture to reach the saturation point and become saturated water.

Stages 4 - 5: Isentropic compression – The boiler feed pump raises the pressure of the water. Because the volume of saturated water is much smaller than saturated steam the work input to raise the pressure is relatively small (THERMOPEDIA, 2011).

The steam quality at point 3 contains a lot of water content and is not ideal for the turbine blades, as super heat is needed.

3.1.3. Rankine cycle with superheat

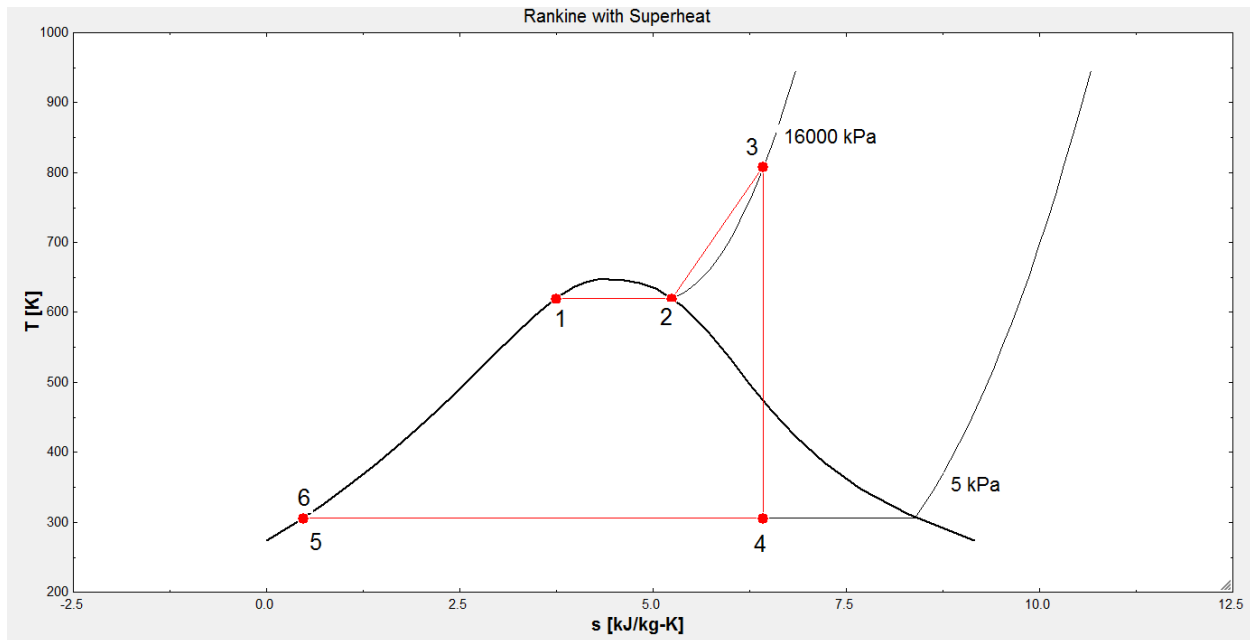


Figure 8: T-s diagram to illustrate the Rankine cycle with super heat

Stages 6 - 1: Isobaric heat gain – Water enters the boiler under high pressure from the boiler feed pump. Heat is added until the water reaches saturation point (1).

Stages 1 - 2: Isobaric heat gain – Further heat is added until the steam is fully saturated.

Stages 2 - 3: Isobaric heat gain – Further heat is added until the steam reaches super steam. This creates a higher steam temperature at constant pressure and more turbine expansion is possible that creates more work output.

Stages 3 - 4: Isentropic expansion – The fully saturated steam expands in a turbine, producing mechanical energy. Because of the expansion over the turbine blades the turbine rotates.

Stages 4 - 5: Isobaric heat rejection– The steam/water mixture enters the condenser where the pressure is well below atmospheric pressure. The pressure and heat transfer force the mixture to reach the saturation point and become saturated water.

Stages 5 - 6: Isentropic compression – The boiler feed pump raises the pressure of the water. Because the volume of saturated water is much smaller than saturated steam and thus the work input to raise the pressure is relatively small.

The steam quality at point 4 still contains a lot of water content and is not ideal for the turbine blades, thus a reheat of the steam/water mixture is needed.

3.1.4. Rankine cycle with superheat and reheat

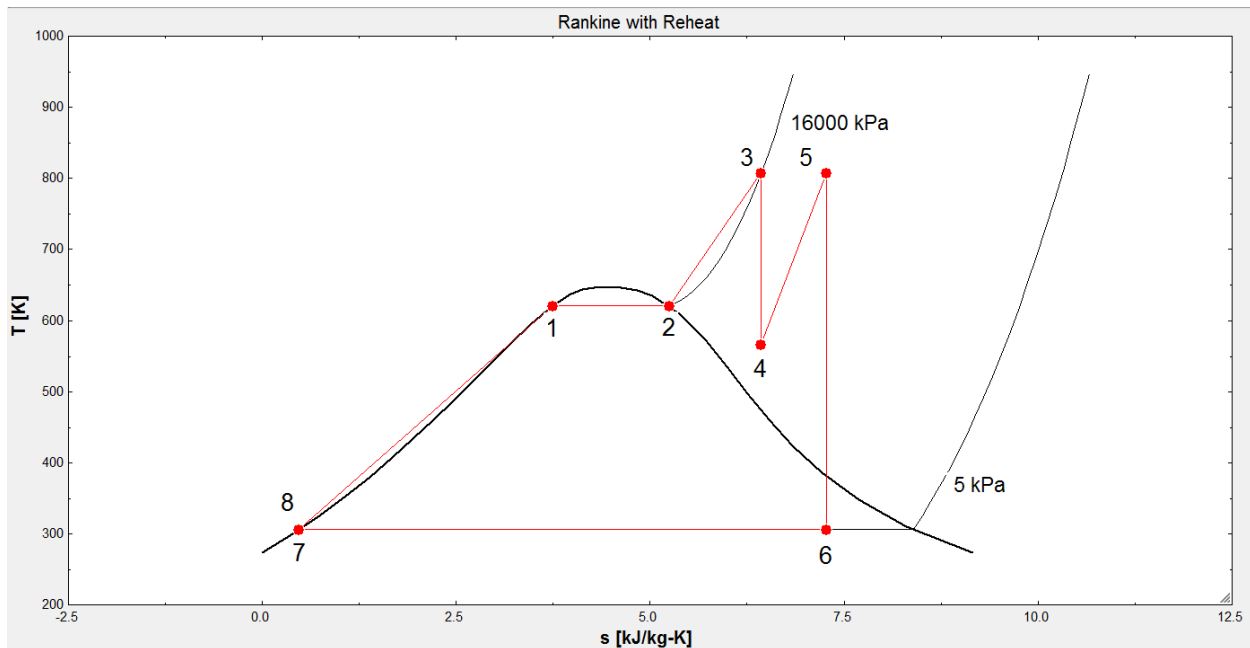


Figure 9: T-s diagram for the Rankine cycle with superheat and reheat

Stages 8 - 1: Isobaric heat gain – Water enters the boiler under high pressure from the boiler feed pump. Heat is added until the water reaches saturation point (1).

Stages 1 - 2: Isobaric heat gain – Further heat is added until the steam is fully saturated.

Stages 2 - 3: Isobaric heat gain – Further heat is added until the steam reaches super-heated steam. This creates a higher steam temperature at constant pressure and more turbine expansion is possible that creates more work output.

Stages 3 - 4: Isentropic expansion – The fully saturated steam expands in a turbine, producing mechanical energy. Because of the expansion over the turbine blades the turbine rotates.

Stages 4 - 5: Isobaric heat gain – The steam returns to the boiler and heat is added again to reach super-heated steam. The steam is then returned to a second and third turbine for further mechanical work.

Stages 5 - 6: Isentropic expansion – The steam expands in the second turbine before it fully expands in the third turbine.

Stages 6 - 7: Isobaric heat rejection– The steam/water mixture enters the condenser where the pressure is well below atmospheric pressure. The pressure and heat transfer force the mixture to reach the saturation point to become saturated water.

Stages 6 - 7: Isentropic compression – The boiler feed pump raises the pressure of the water. Because the volume of saturated water is much smaller than saturated steam the work input to raise the pressure is relatively small.

The steam quality at point 6 contains very high steam content and can be used without damaging the turbine blades.

3.1.5. Rankine cycle with superheat, reheat and feed water heating

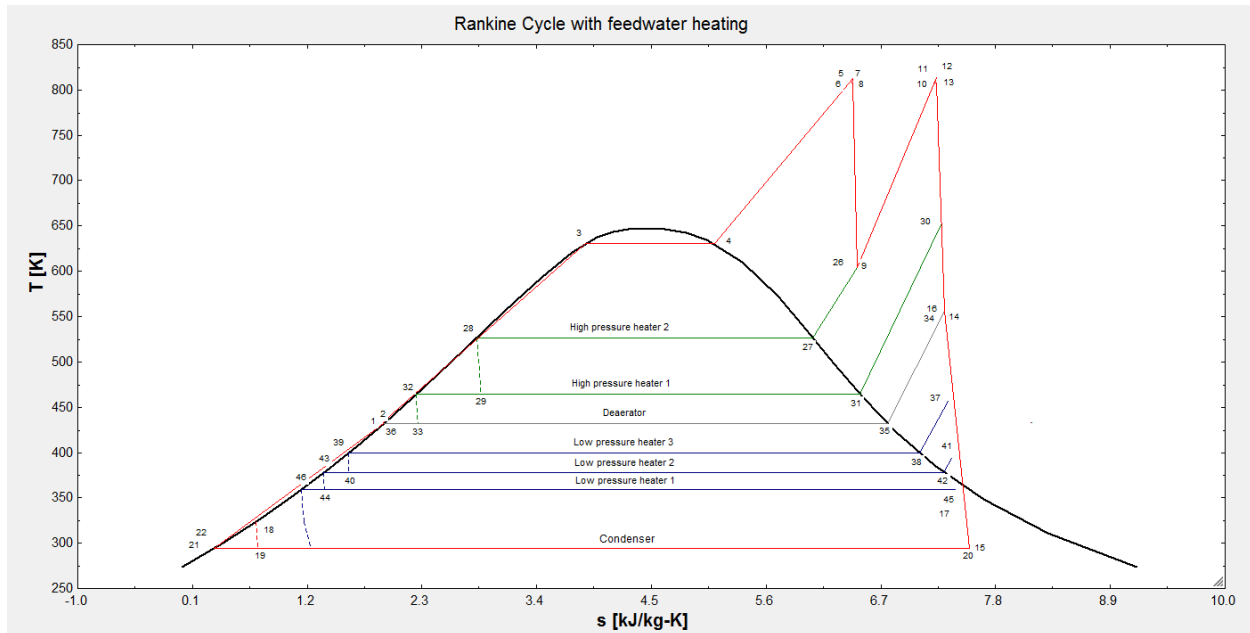


Figure 10: T-s diagram for the Rankine cycle with superheat, reheat and feed water heating

Feed water heating takes place to get a better efficiency. Looking at the Carnot cycle the highest efficiency takes place if heat transfer takes place isothermally. Bled off steam from various points on the turbines is used in this process (THERMOPEDIA, 2011).

Feed water heating takes place after the saturated water leaves the condenser. Low pressure non-contact heaters use tap off steam from the second and third stage turbines to transfer heat to the water before entering the de-aerator. The de-aerator is a contact heater which uses tap off steam from the second stage turbine. The water is fed to the boiler feed pump and then enters the high pressure non-contact heaters. Heat transfer takes place further in the high pressure heaters before entering the boiler. High pressure heater one uses tap off steam from the second stage turbine and high pressure heater two uses tap off steam from the cold reheat line.

The stages remain the same as the previous illustration for the Rankine cycle with reheat. The feed water heating transfers energy from the steam to the water, thus less energy is needed from the coal, which results in a more effective overall Rankine cycle.

3.2. Steam quality at turbine outlet

3.2.1. Influence of boiler pressure

For this illustration the temperature and mass flow are held constant.

Figure 11 below illustrates a few different pressure lines on the T-s diagram and the effect of it on the steam quality at the turbine exit.

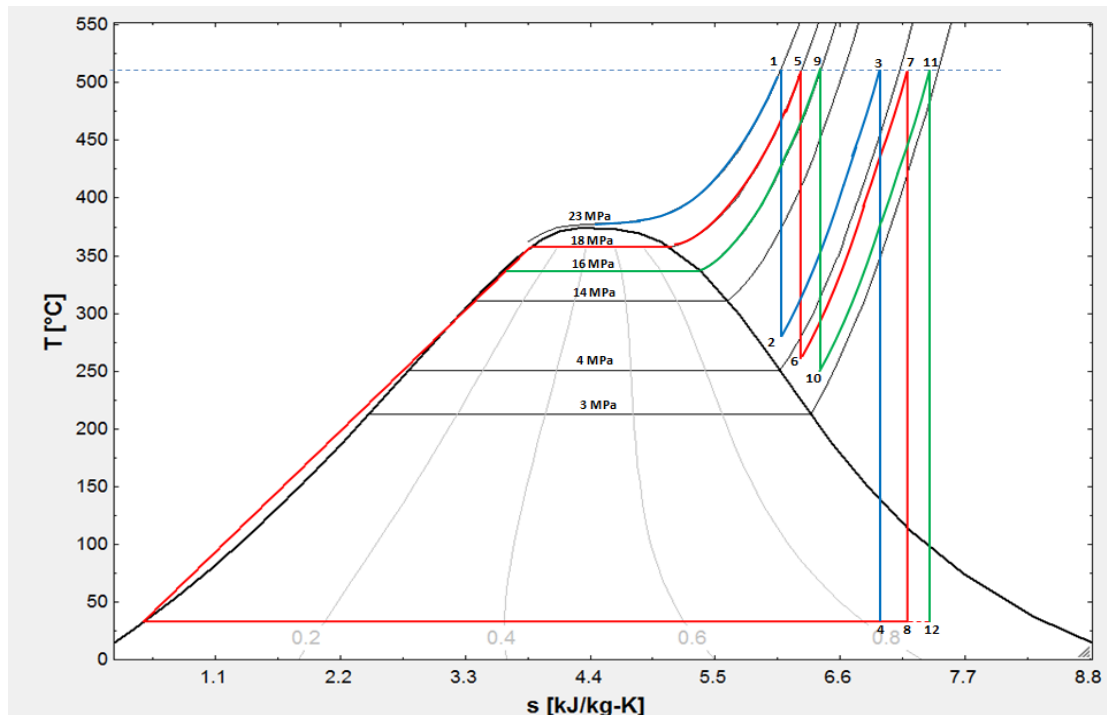


Figure 11: T-s diagram with different pressure lines and the effect of it on the low pressure turbine outlet steam quality.

First we analyse the 16 MPa line (green line). The steam is heated to maximum temperature (9), and then expands in the high pressure turbine (10) before it is reheated to maximum temperature (11) again. The steam expands in the intermediate pressure turbine and then fully in the low pressure turbine (12).

If the pressure is increased to 18 MPa (red line) a new pressure curve is visible. The same process follows as above.

If the pressure is increased to 23 MPa (blue line) the pressure curve is above the two phase stage, and this means that the super-critical stage is reached.

The steam quality at the low pressure turbine outlet decreases as the pressure increases (12 – 8 – 4), but the area under the graph increases, which means more net mechanical work.

With better metallurgical conditions the boiler pressure can be raised to increase the mechanical work output.

3.2.2. Influence by maximum temperature

For this illustration the pressure and mass flow is held constant as well as the turbine expansion pressure.

Figure 12 below illustrates a few different temperature lines on the T-s diagram and the effect of it on the steam quality at the turbine exit

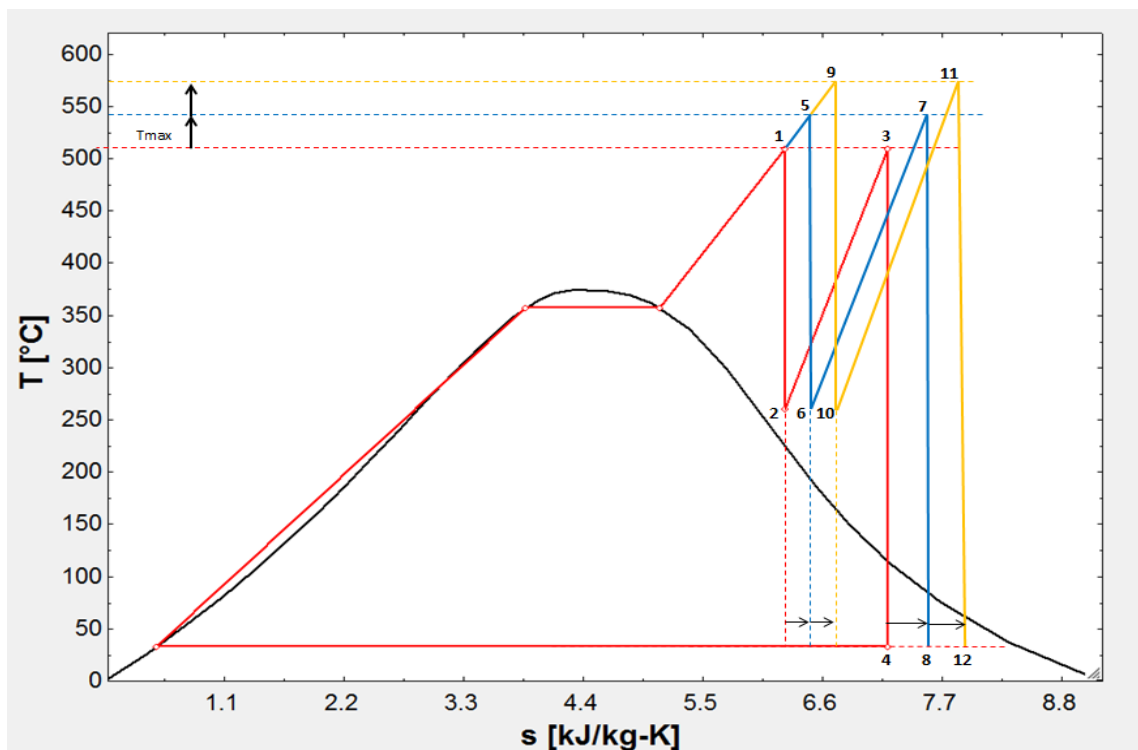


Figure 12: T-s diagram with different temperature lines and the effect of it on the low pressure turbine outlet steam quality.

First the lowest temperature is analysed (red line). The steam is heated to the maximum temperature (1) before it expands in the high pressure turbine (2). The steam is heated again to maximum temperature and then expands in the intermediate and low pressure turbines (4).

Figure 12 above illustrates the rising of the maximum temperature to point 5 (blue line) and also point 9 (yellow line). The same process as above follows.

The illustration shows that if the temperature increases the steam quality at the low pressure turbine outlet increases. This is ideal for the low pressure turbine blades.

3.2.3. Influence by high pressure turbine expansion

For this illustration the temperature, mass flow and boiler pressure are held constant.

Figure 13 below illustrates a few different pressure lines on the T-s diagram and the effect of the high pressure turbine expansion on the steam quality at the turbine exit.

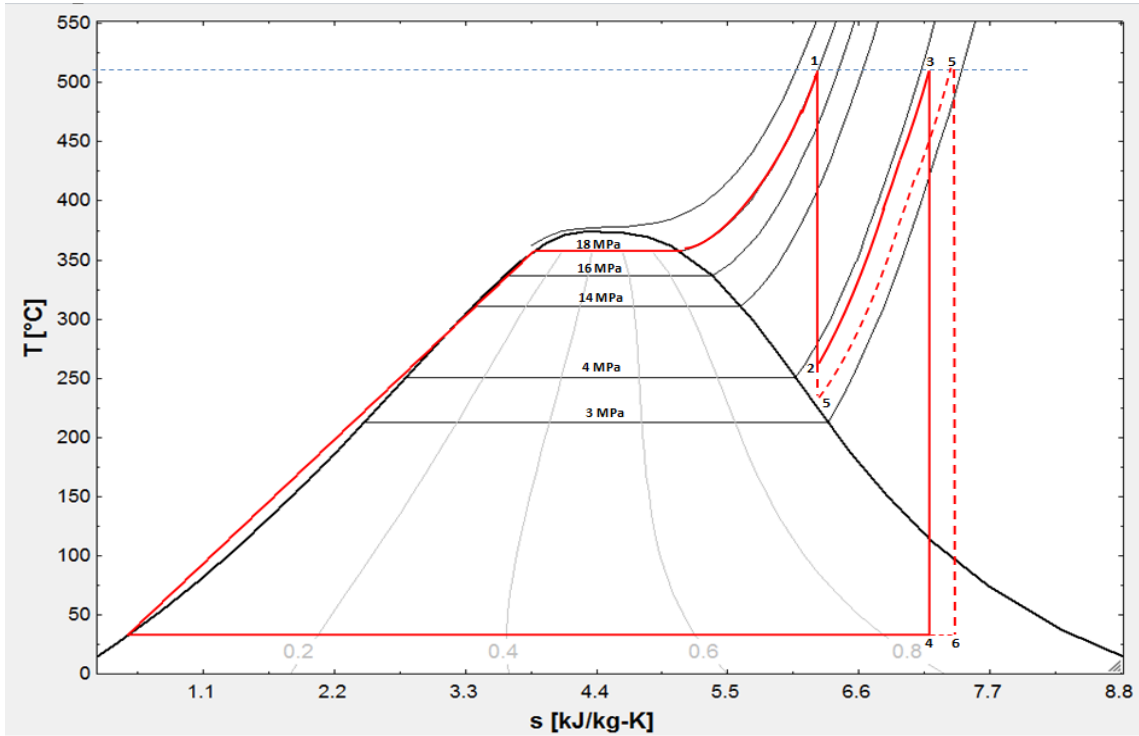


Figure 13: T-s diagram to illustrate the high pressure turbine expansion and the effect of it on the low pressure turbine outlet steam quality

If the high pressure turbine expands further to point 5, then the pressure line changes because the turbine expands further and more work is done, therefore more energy is used. Thus the steam should gain more heat in the reheater than previously. With the lower pressure line because of further expansion the steam quality improves at the low pressure turbine outlet (6).

The area under the graph increases, thus net mechanical work increases. The high pressure turbine expansion is restricted by the capability of the heat gain in the reheater as well as the low pressure turbine outlet steam quality.

3.3. Natural vs forced circulation boilers

3.3.1. Natural circulation (steam drum boilers)

Natural circulation boilers use a steam drum. The steam drum level is maintained by the boiler feed pump. The water is fed into the steam drum and travels down to the bottom mud drum where it is fed into the boiler tubes. Heat is applied and the hot water rises to the top where it accumulates and enters the drum again. The drum separates the steam and the water where the water goes down to the bottom mud drum again. The steam travels to the superheater for further heating. The steam drum has a fixed separation point, which means a molecule of water can make many passes through the evaporation tubes before turning into steam for further heating.

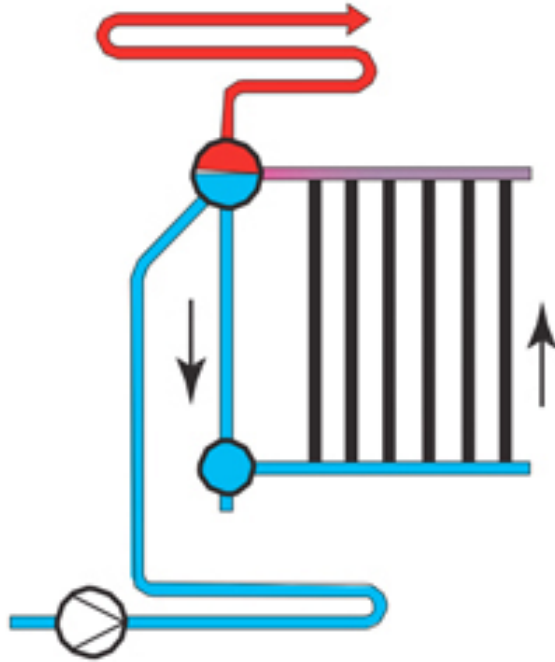


Figure 14: Illustration of the flow for drum boilers (Power)

Advantages

- Easier construction and cheaper to build, no spiral wall required.
- Less water consumption.
- More tolerant of feed water impurities.
- High reliability.
- Constant heat transfer areas.
- High partial load range.

Disadvantages

- The drum is part of the high pressure components and limits the operating flexibility due to high thermal stresses.
- One evaporation end point, the drum.
- High circulation ratio, which leads to a big evaporator area.
- More tube failures because of larger diameter tubes.
- Sensitive to load variations.

3.3.2. Forced circulation (once through boilers)

Forced circulation or once through boilers do not make use of a steam drum. Water enters the boiler from the boiler feed pump. The water level is controlled by the firing rate through the evaporation and circulation rate. The water travels through the boiler tubes and evaporates fully, but this only applies after start-up. During start up the boiler uses separating vessels to separate the steam and water mixture. The fully evaporated steam travels to the superheaters for further heating.

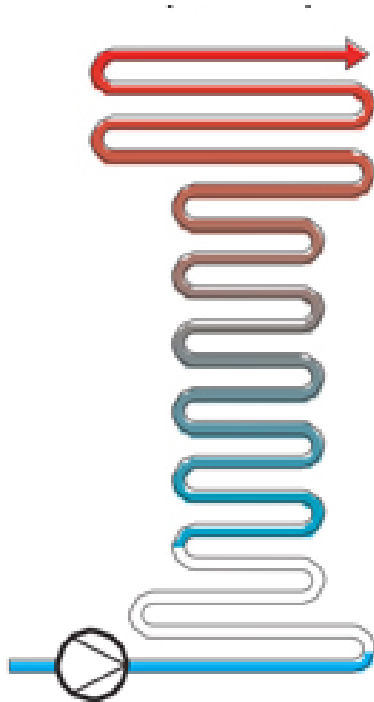


Figure 15: Illustration of the flow for once through boilers (Power)

Advantages

- Does not have a high pressure drum, thus more operating flexibility and lower stress operation.
- High overall efficiency, even at part loads.
- Shorter start-up time.
- Suitable for all coal grades.
- More equal distribution of the water in the tubes.
- Ideal for sliding pressure operation, thus more control over load changes.
- Produces less CO₂, because it is more efficient.

Disadvantages

- It's necessary for higher grade material as the evaporator forms part of the first superheater.
- Difficult construction because of spiral tubes.
- Feed pump needed for forced circulation.
- Higher water consumption.

3.4. Difference between sub- and super-critical Rankine cycles

In this study both cycles use a once through boiler with forced circulation.

3.4.1. Sub-critical Rankine cycle

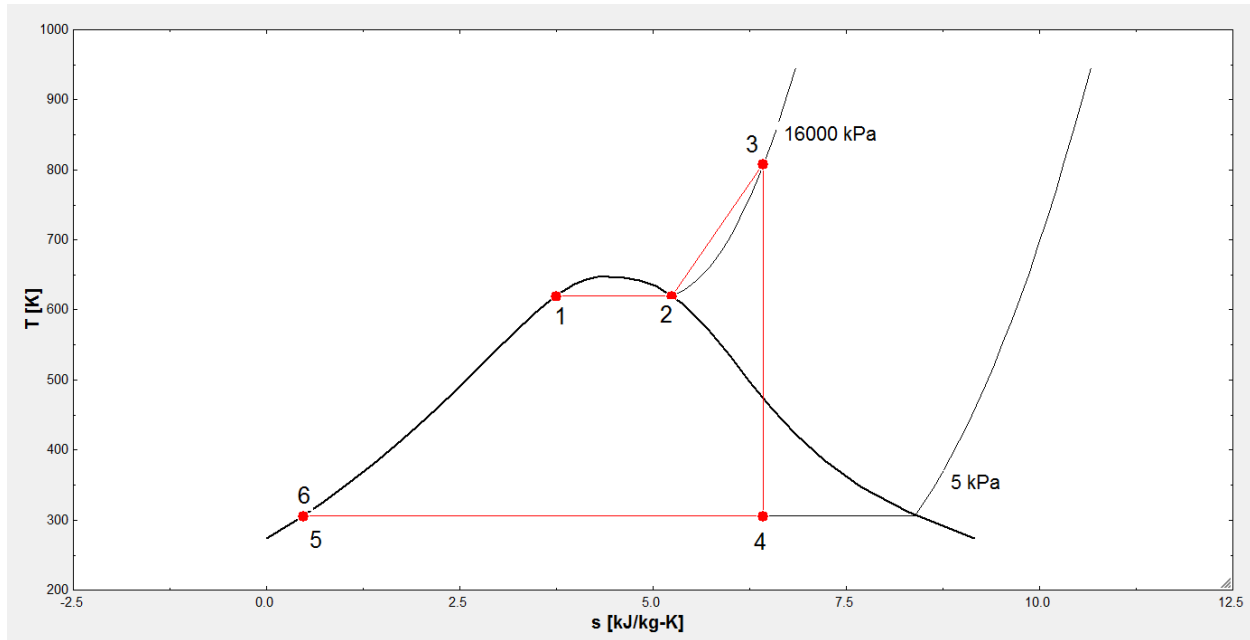


Figure 16: Illustration of a sub-critical Rankine T-s diagram

A sub-critical Rankine cycle operates under the critical pressure of water (22.06 MPa). This mean a two phase stage must exist.

Figure 16 above illustrates a T-s diagram for sub-critical Rankine cycles. At point 1 the saturated water turns into a saturated mixture and at point 2 the mixture turns into saturated steam.

3.4.2. Super-critical Rankine cycle

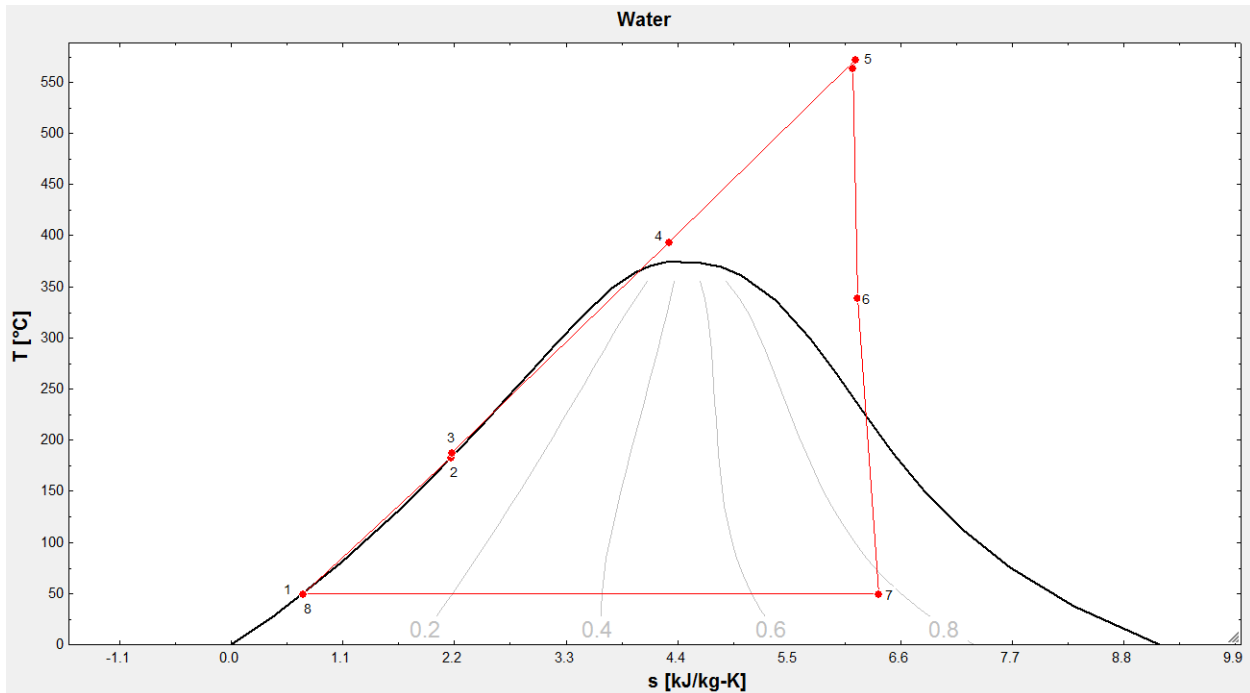


Figure 17: Illustration of a super-critical Rankine cycle T-s diagram

Super-critical Rankine cycle operates above the critical pressure of water (22.06 MPa). This means no two phase stage exists.

Figure 17 illustrates the T-s diagram for super-critical cycles. Point 4 is the critical point for this cycle where water transfers into steam without a two stage phase.

With better metallurgical conditions higher pressure and temperature can be achieved, therefore super-critical cycles are possible. With the possibility to increase the pressure and temperature a higher efficiency and net mechanical work can be reached.

4. Rankine cycle programming methodology to enable optimisation

4.1. Rankine cycle efficiency

4.1.1. Increasing boiler pressure

One way to increase the Rankine cycle efficiency is to increase the boiler pressure at constant temperature. This will increase the pressure where evaporation takes places.

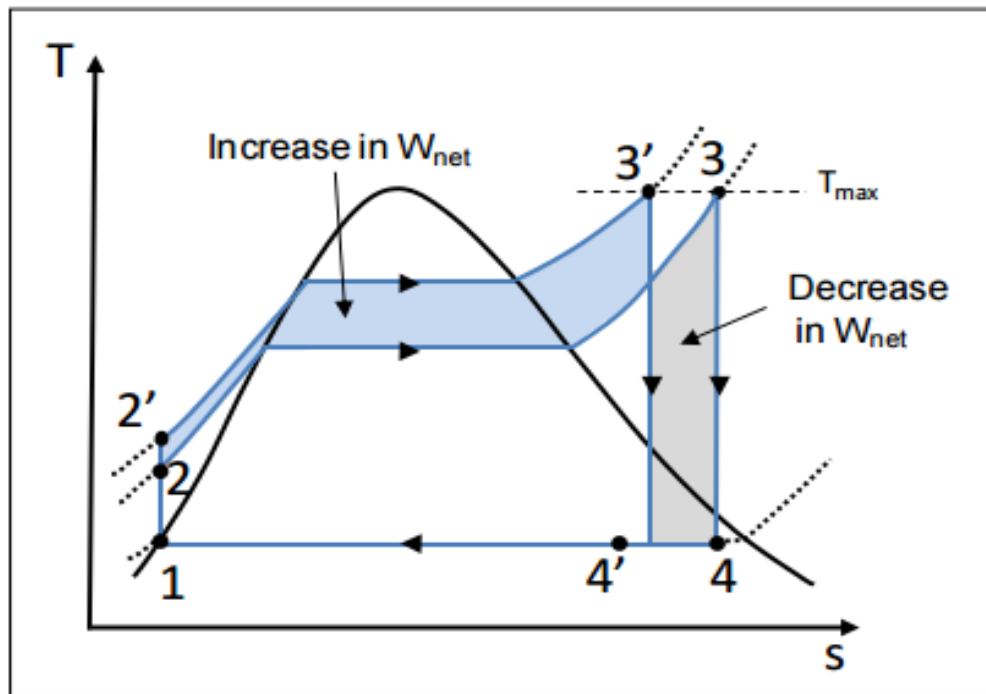


Figure 18: Effect of boiler pressure on net mechanical work (Laure, 2011)

By using the Rankine cycle efficiency formula the following can be concluded

$$\text{Cycle } \eta = \frac{W_{net}}{Q_{in}} = \frac{W_{out} - W_{in}}{Q_{in}}$$

From figure 18 a slight increase in heat in (Q_{in}) and work in (W_{in}), with a significantly increase in work out (W_{out}) can be observed, therefore **Cycle efficiency** (η) increases.

The area represented the increase in net mechanical work is greater than the area that represents a decrease in net mechanical work (B-Cubed, 2014)

The limit is determined by optimisation of the net mechanical work and cycle efficiency, but metallurgical conditions also limit the pressure that can be added.

4.1.2. Increasing boiler temperature

Another way to increase the Rankine cycle efficiency is to reach higher temperatures with constant pressure.

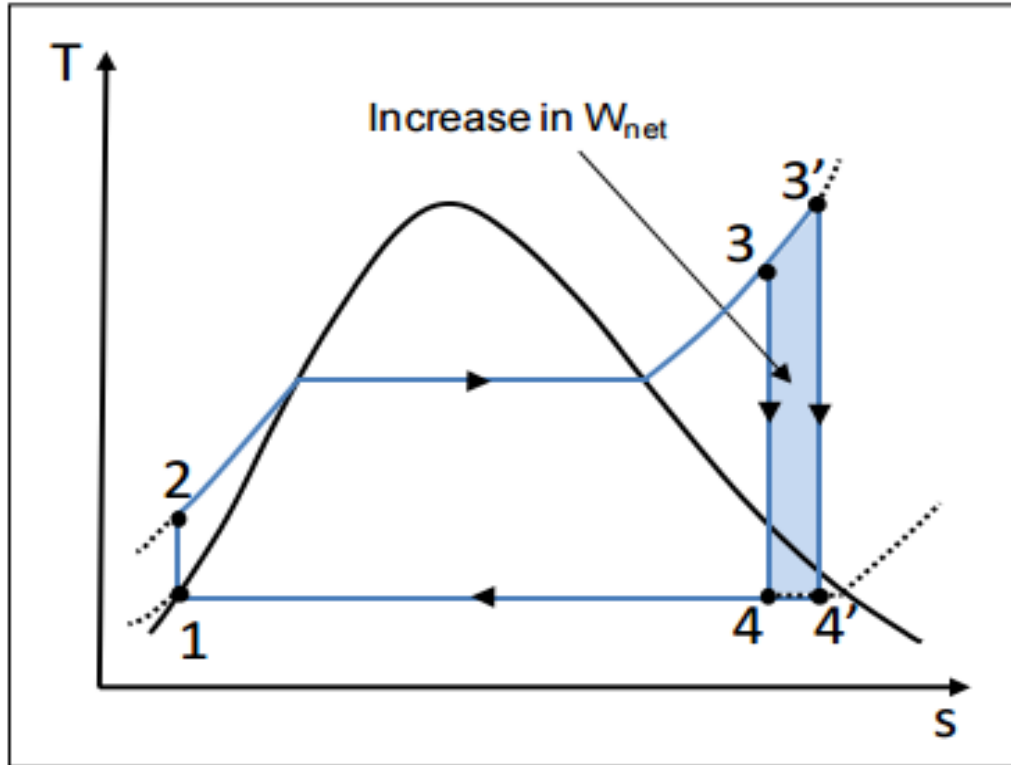


Figure 19: Effect of temperature on net mechanical work (Laure, 2011)

By using the cycle efficiency formula the following can be concluded

$$\text{Cycle } \eta = \frac{W_{net}}{Q_{in}} = \frac{W_{out} - W_{in}}{Q_{in}}$$

From the above figure a large increase in work out (W_{out}) is visible with no change in work in (W_{in}), therefore **Cycle efficiency** (η) increases.

By using temperature to increase the Rankine cycle the steam quality at the turbine exit increases as well, and this is ideal for the turbine blades.

The incensement of temperature is limited by metallurgical conditions and economical infrastructure as the heat added will also be increased (Electrical4u.com, 2015).

4.1.3. Lowering the condenser pressure

By lowering the condenser pressure an increase in the Rankine cycle efficiency is possible. This way more turbine expansion is possible.

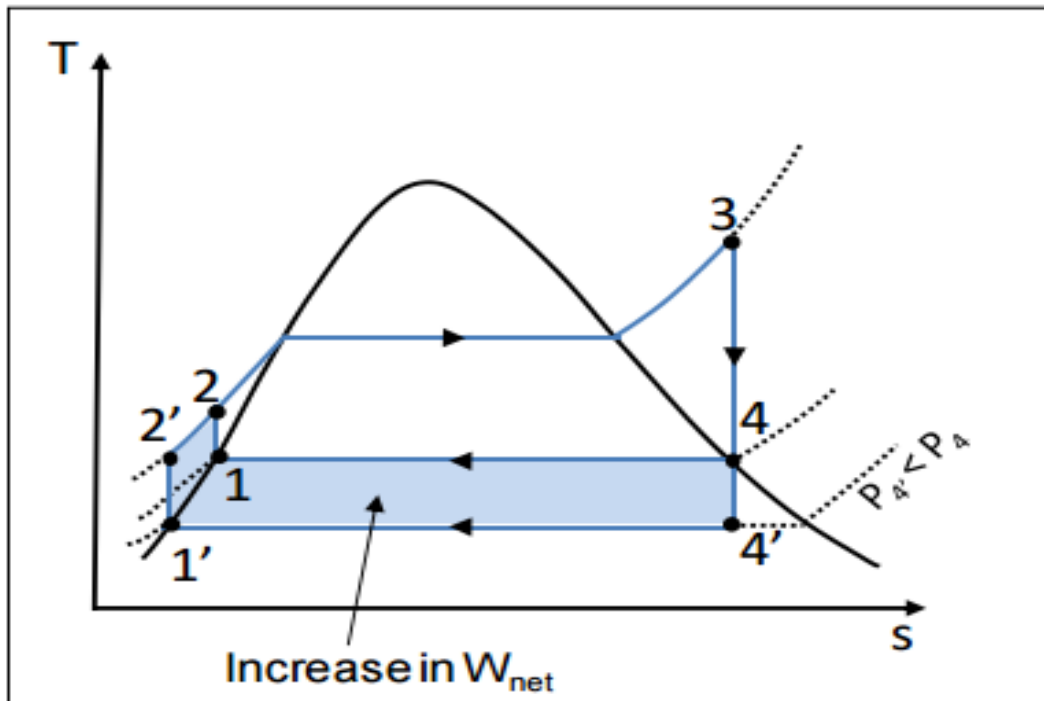


Figure 20: Effect of lowering the condenser pressure on net mechanical work (Laure, 2011)

By using the cycle efficiency formula the following can be concluded

$$\text{Cycle } \eta = \frac{W_{net}}{Q_{in}} = \frac{W_{out} - W_{in}}{Q_{in}}$$

From the above figure 20 work out (W_{out}) increases significantly while work in (W_{in}) increases slightly, therefore **Cycle efficiency (η)** will also increase.

The limit for condenser pressure is limited by cooling water temperature, which will influence the saturation pressure of the mixture entering the condenser. Steam quality at the turbine exit will increase as the condenser pressure decrease. This is not ideal for the turbine blades (Electrical4u.com, 2015).

5. Programming of sub- and supercritical Rankine cycles

Both cycles are based on once through boilers and drum boilers were not taken in account.

5.1. Sub-critical Rankine cycle

This cycle was programmed in EES by using the heat balance diagram of Kriel power station at maximum output (500 MW).

5.1.1. Rankine cycle without optimisation

5.1.1.1. Condenser outlet

The condenser outlet temperature is held constant by ignoring the ambient temperature, and this serves as the first variable. The condenser is a heat exchanger and extracts heat from the steam/water mixture, therefore saturated water can be expected at the condenser outlet. With two known variables the rest can be calculated.

5.1.1.2. Extraction pump discharge

The extraction pump has a constant pressure because the de-aerator has a regulating valve between the extraction pump and de-aerator inlet. With the pressure constant the first variable is known. Because the extraction pump uses saturated water coming from the condenser outlet the second variable is known.

But because the extraction pump has efficiency the following equation should apply.



$h_{4,}$ = with pressure and $S_{4,}$ as input

With this equation h_4 can be calculated for the extraction pump.

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5.1.1.3. Low pressure heater 1

The pressure is taken as a percentage of the low pressure turbine inlet pressure. The correct pressure is taken from the heat balance diagram, thus the first variable is known.

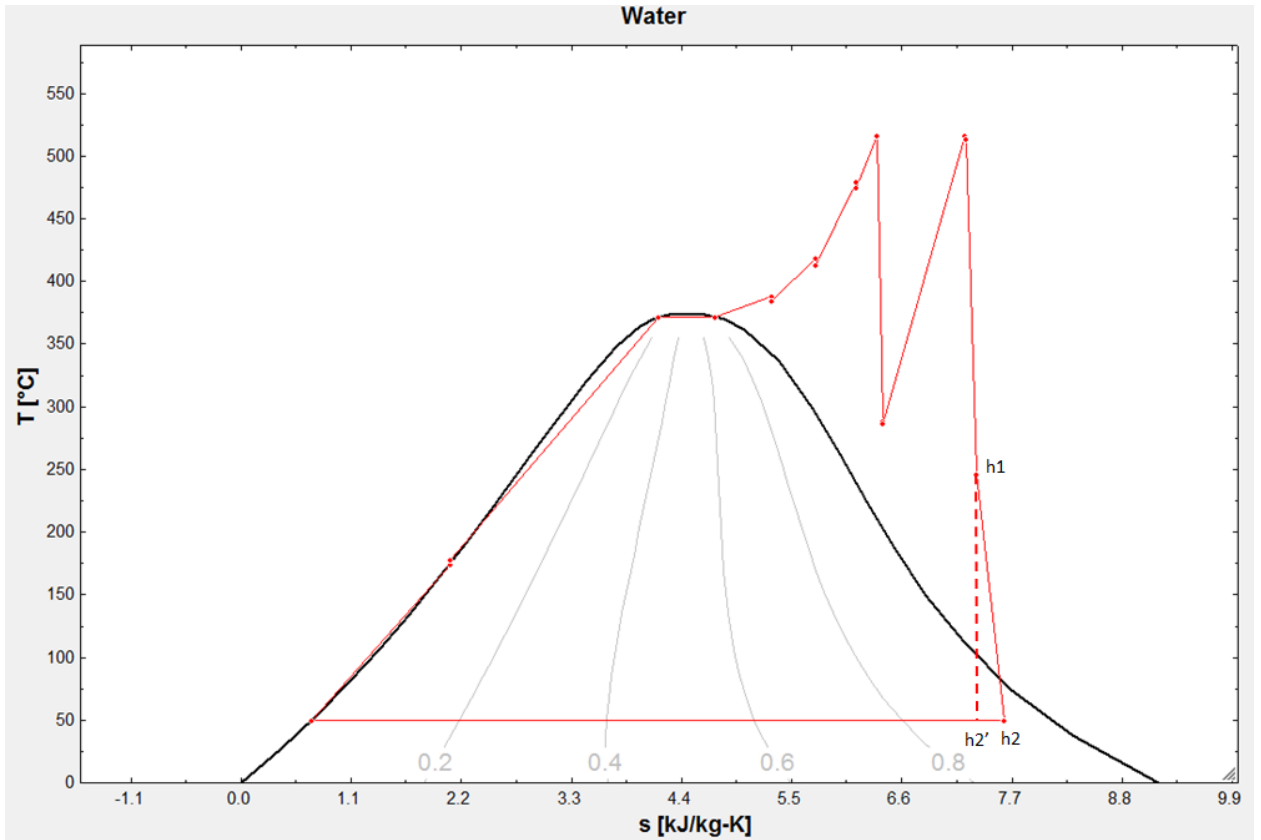


Figure 22: Illustration of the efficiency effect of the low pressure turbine on the Rankine cycle

Because the water is still saturated liquid the second variable is known.

But because the steam taps off at the low pressure turbine the efficiency of the low pressure turbine must be taken into account. The following calculation must be implemented.

$$s_1 = s_2,$$

$$h_{2'} = \text{with pressure and } s_2, \text{ as input}$$

$$LPT \eta = \frac{h_1 - h_2}{h_1 - h_{2'}}$$

With this equation h_2 can be calculated for the pressure of low pressure heater 1. With these variables the rest can be calculated.

5.1.1.4. Low pressure heater 2

The pressure is taken as a percentage of the low pressure turbine inlet pressure. The correct pressure is taken from the heat balance diagram, thus the first variable is known. Because the water is still saturated liquid the second variable is known.

The same calculation for low pressure heater 1 (5.1.1.3) applies for low pressure heater 2 to compensate for the efficiency effect. With these variables the rest can be calculated.

5.1.1.5. Low pressure heater 3

The pressure is taken as a percentage of the low pressure turbine inlet pressure. The correct pressure is taken from the heat balance diagram, thus the first variable is known. Because the water is still saturated liquid the second variable is known.

The same calculation for low pressure heater 1 (5.1.1.3) applies for low pressure heater 3 to compensate for the efficiency effect. With these variables the rest can be calculated.

5.1.1.6. Low pressure heater 4

The pressure is taken as a percentage of the intermediate pressure turbine inlet pressure. The correct pressure is taken from the heat balance diagram, thus the first variable is known. Because the water is still saturated liquid the second variable is known.

But because the steam taps off at the intermediate pressure turbine the efficiency of the intermediate pressure turbine must be taken into account. The following calculation must be implemented.

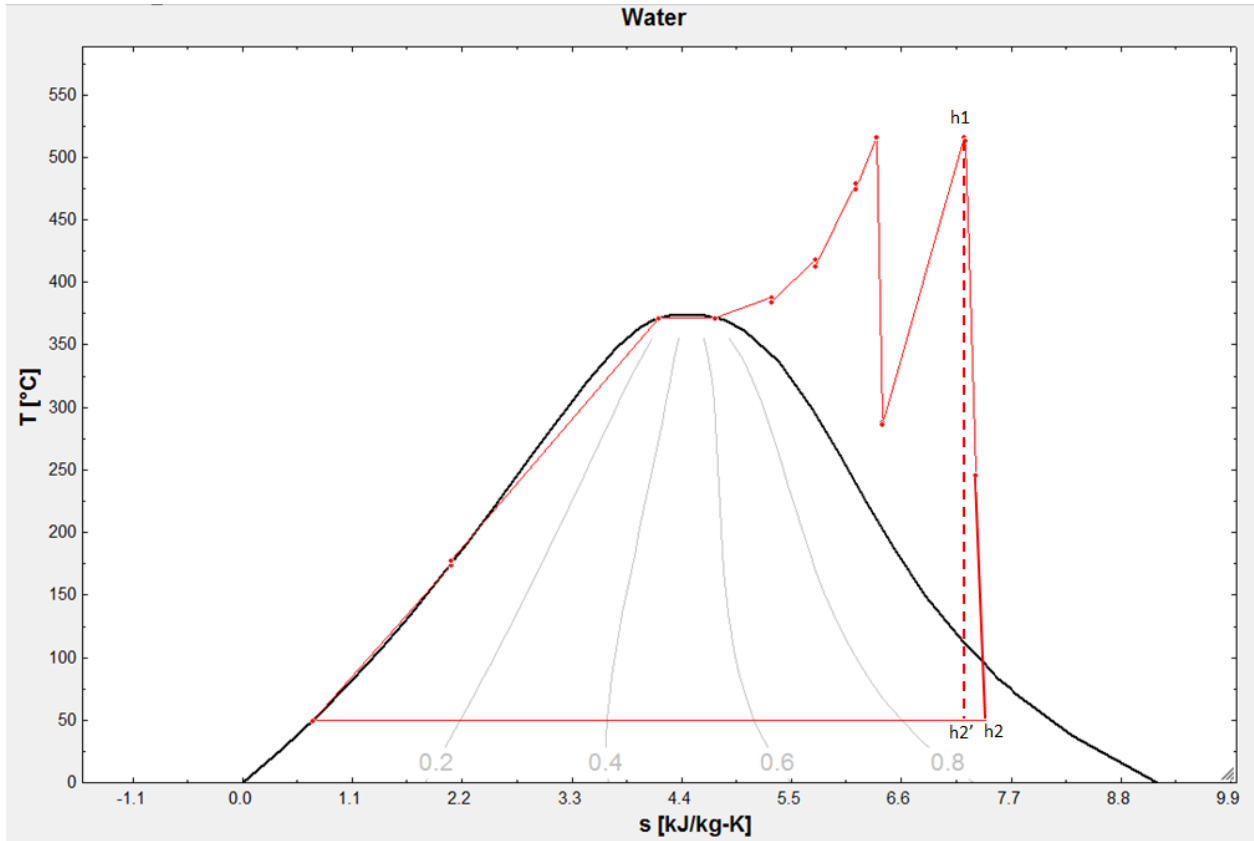


Figure 23: Illustration of the efficiency effect of the intermediate pressure turbine on the Rankine cycle

$$S_1 = S_2,$$

$$h_{2'} = \text{with pressure and } S_2, \text{ as input}$$

$$IPT \eta = \frac{h_1 - h_2}{h_1 - h_{2'}}$$

With this equation h_2 can be calculated for the pressure of low pressure heater 4.

5.1.1.7. De-aerator

The pressure is taken as a percentage of the intermediate pressure turbine inlet pressure. The correct pressure is taken from the heat balance diagram, thus the first variable is known. Because the water is still saturated liquid the second variable is known.

The same calculation for low pressure heater 4 (5.1.1.6) applies for the de-aerator to compensate for the efficiency effect. With these variables the rest can be calculated.

5.1.1.8. High pressure heater 6

The pressure is taken as a percentage of the intermediate pressure turbine inlet pressure. The correct pressure is taken from the heat balance diagram, thus the first variable is known. Because the water is still saturated liquid the second variable is known.

The same calculation for low pressure heater 4 (5.1.1.6) applies for high pressure heater 6 to compensate for the efficiency effect. With these variables known the rest can be calculated.

5.1.1.9. High pressure heater 7

Because high pressure heater 7 taps steam off from the high pressure turbine outlet line (cold reheat line) the pressure cannot differ too much. A percentage of 0.1 is taken for the pressure loss in the cold reheat line. With the pressure known and the water is still liquid two variables are known.

The same calculation for low pressure heater 4 (5.1.1.6) applies for high pressure heater 7 to compensate for the efficiency effect. With these variables the rest can be calculated.

5.1.1.10. Boiler feed pump

The maximum pressure is known and the water is still saturated liquid, thus two variables are known. With two known variables the rest of the variables can be calculated.

But because the boiler feed pump has an efficiency the following equation should apply.

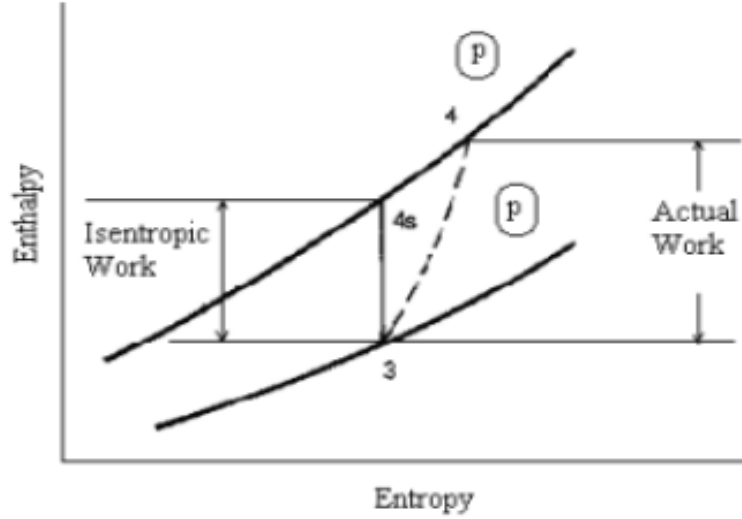


Figure 24: Enthalpy plotted against Entropy to illustrate the effect of efficiency on a pump (An analysis of a thermal power plant working on a Rankine cycle: A theoretical investigation, 2008)

$$S_3 = S_{4'}$$

$$h_{4'} = \text{with pressure and } S_{4'} \text{ as input}$$

$$FDP \eta = \frac{h_{4'} - h_3}{h_4 - h_3}$$

With this equation h_4 can be calculated for the pressure output of the boiler feed pump pressure. With these variables known the rest can be calculated.

5.1.1.11. Evaporator

The pressure is taken as a percentage of the maximum pressure to compensate for the pressure loss in the lines and evaporator tubes.

A procedure was programmed to determine whether the pressure is sub-critical or super-critical. If the pressure is sub-critical a two-phase mixture should exist and the water turns from saturated liquid to saturated steam. Thus two variables are known and the rest can be calculated.

If the pressure is super-critical the pressure line missed the two phase stage and the critical point for water is reached. Thus two variables are known and the rest can be calculated.

5.1.1.12. Superheater attemperator 1

5.1.1.12.1. Inlet

The pressure is taken as a percentage of the evaporator inlet pressure to compensate for the pressure loss in the main steam pipes.

The temperature is taken as a percentage loss in in the main steam pipelines.

5.1.1.12.2. Outlet

The temperature that is tempered off is taken from the heat balance diagram and as a percentage of the maximum temperature. The entropy is kept the same as the inlet.

With two variables known the rest can be calculated.

5.1.1.13. Superheaters

5.1.1.13.1. Superheater 1

The temperature and pressure for superheater 1 inlet are the same as attemperator 1 outlet, because attemperator 1 is situated at the superheater 1 inlet. The outlet pressure and temperature are taken from the heat balance diagram and are a percentage of the maximum temperature.

5.1.1.14. Superheater attemperator 2

5.1.1.14.1. Inlet

The pressure is taken as a percentage of the superheater 1 inlet pressure to compensate for the pressure loss in superheater 1 tubes.

The temperature is taken as a percentage of the maximum temperature to compensate for the temperature gain in superheater 1.

5.1.1.14.2. Outlet

The temperature that is tempered off is taken from the heat balance diagram and is taken as a percentage of the maximum temperature. The entropy is kept the same as the inlet.

With two variables known the rest can be calculated.

5.1.1.15. Superheater 2

The temperature and pressure for superheater 2 are the same as the attemperator 2 outlet, because attemperator 2 is situated at the superheater 2 inlet. The outlet pressure and temperature are taken from the heat balance diagram and are a percentage of the maximum temperature.

5.1.1.16. Superheater attemperator 3

5.1.1.16.1. Inlet

The pressure is taken as a percentage of the superheater 2 inlet pressure to compensate for the pressure loss in superheater 2 tubes.

The temperature is taken as a percentage of the maximum temperature to compensate for the temperature gain in superheater 2.

5.1.1.16.2. Outlet

The temperature that is tempered off is taken from the heat balance diagram and is taken as a percentage of the maximum temperature. The entropy is kept the same as the inlet.

With two variables known the rest can be calculated.

5.1.1.17. Superheater 3

The temperature and pressure for superheater 3 is the same as attemperator 3 outlet, because attemperator 3 is situated at superheater 3 inlet. The outlet pressure and

$$S_1 = S_{2'}$$

$h_{2'}$ = *with pressure and $S_{2'}$ as input*

$$HPT \eta = \frac{h_1 - h_2}{h_1 - h_{2'}}$$

With this equation h_2 can be calculated for the pressure output of the high pressure turbine.

With two variables known the rest can be calculated.

5.1.1.19. Reheater attemperators

5.1.1.19.1. Inlet

The pressure is taken as a percentage of high pressure turbine outlet pressure and the pressure loss to high pressure heater 7. The temperature is taken to be the same as the high pressure turbine outlet.

With two variables known the rest can be calculated.

5.1.1.19.2. Outlet

The temperature that is tempered off is taken from the heat balance diagram and is taken as a percentage of the maximum temperature. The entropy is kept the same as the inlet.

With two variables known the rest can be calculated.

5.1.1.20. Reheaters

The pressure is taken as a percentage of the attemperators outlet pressure. The temperature is taken as the maximum temperature.

With two variables known the rest can be calculated.

5.1.1.21. Intermediate pressure turbine

The pressure is taken as a percentage of the Reheater outlet pressure to compensate for the pressure loss in hot reheat line.

Because the intermediate pressure turbine has an efficiency the following equation should apply.

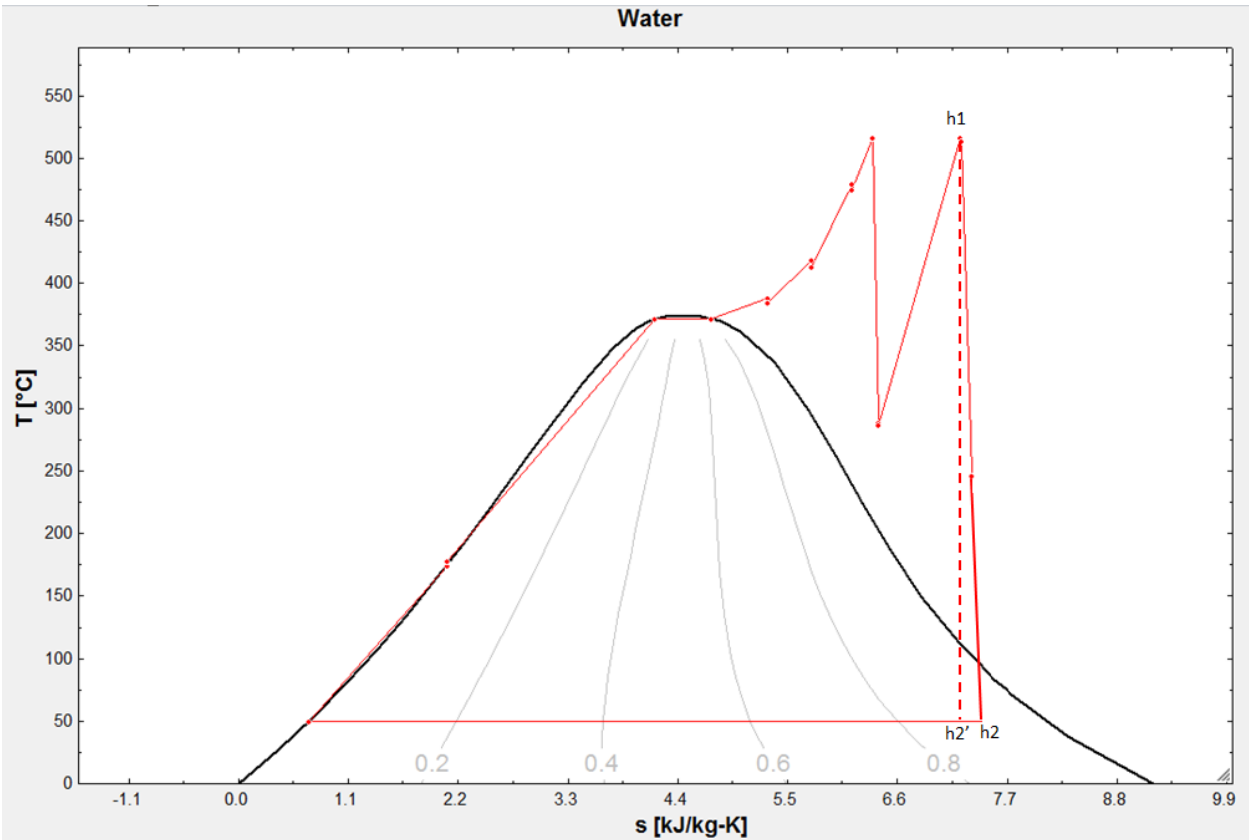


Figure 26: Illustration of the efficiency effect of the intermediate pressure turbine on the Rankine cycle

$$S_1 = S_2,$$

$$h_{2'} = \text{with pressure and } S_2, \text{ as input}$$

$$IPT \eta = \frac{h_1 - h_2}{h_1 - h_{2'}}$$

With this equation h_2 can be calculated for the pressure output of the intermediate pressure turbine.

With two variables known the rest can be calculated.

5.1.1.22. Low pressure turbine

The pressure is taken as a percentage of the low pressure turbine to compensate for the pressure loss through the intermediate turbine.

Because the low pressure turbine has an efficiency the following equation should apply.

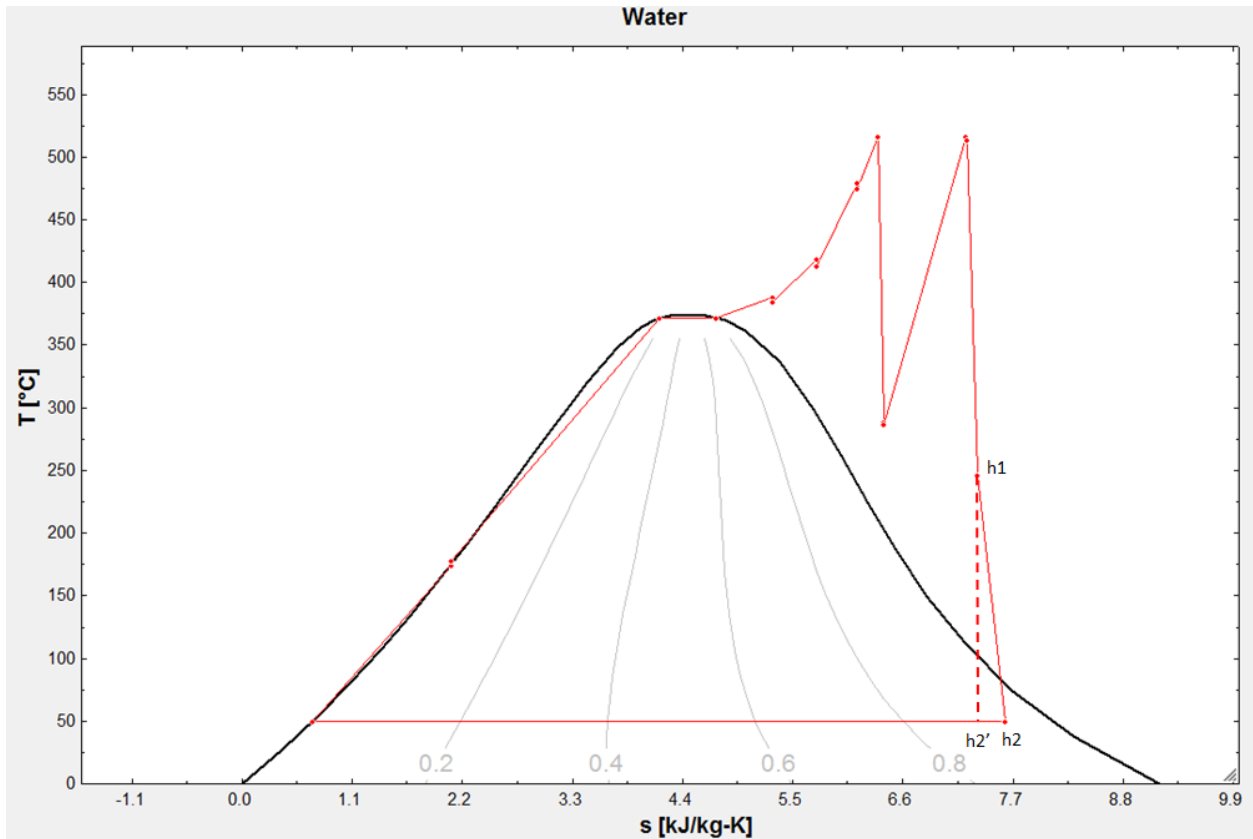


Figure 27: Illustration of the efficiency effect of the low pressure turbine on the Rankine cycle

$$S_1 = S_2,$$

$$h_{2'} = \text{with pressure and } S_{2'} \text{ as input}$$

$$LPT \eta = \frac{h_1 - h_2}{h_1 - h_{2'}}$$

With this equation h_2 can be calculated for the pressure output of the low pressure turbine.

With two variables known the rest can be calculated.

5.1.1.23. Superheater attemperators tap off point

The pressure for the tap off point for the superheaters is taken as the maximum pressure. The enthalpy is taken as the feed pump outlet enthalpy.

With two variables known the rest can be calculated.

5.1.1.24. Reheater attemperators tap off point

The pressure for the tap off point for the Reheaters is taken as a percentage of the maximum pressure because the tap off point is on the interconnecting stage of the feed pump. The enthalpy is taken on the same line as the feed pump with the corresponding pressure.

5.1.2. Optimisation of the sub-critical Rankine cycle

5.1.2.1. Boiler feed pump

A factor line was programmed to get the optimal for both efficiency and net mechanical work. The following equation was used.

$$FACT = (W_{nett} * Eta_{cycle})/1000$$

The factor line is divided by a number to fit in on the scale with the cycle efficiency.

A new parametric table was created with 500 runs. The boiler feed pump was run from 5 MPa to 48 MPa in even distributing increments. The net mechanical work, cycle efficiency and the factor line are used as variables of the pressure.

The results is plotted on a graph to get the optimal point

5.1.2.2. High pressure turbine expansion pressure

A new parametric table was created with 500 runs. The high pressure turbine expansion pressure was run from 10% to 90% of the expansion pressure. The pressure, net mechanical work, cycle efficiency and the factor line were added as variables.

The results is plotted on a graph to get the optimal point

The optimal point is limited by the steam quality at the low pressure turbine outlet. In this study we did not want to compromise the low pressure turbine blades and kept the steam quality as designed.

5.1.2.3. High pressure heater 6 tap off pressure

A new parametric table was created with 500 runs. The tap off pressure was taken as a percentage of the intermediate turbine inlet pressure. The tap off pressure percentage was run from 0.01% to 99.99%. The pressure, net mechanical work, cycle efficiency and the factor line were added as variables.

The results are plotted on a graph to get the optimal point

An assumption is made that a minimum pressure loss of 33% through the Intermediate pressure turbine was possible.

5.1.2.4. De-aerator tap off pressure

A new parametric table was created with 500 runs. The tap off pressure was taken as a percentage of the intermediate turbine inlet pressure. The tap off pressure percentage was run from 0.01% to 99.99%. The pressure, net mechanical work, cycle efficiency and the factor line were added as variables.

The results are plotted on a graph to get the optimal point. An assumption is made that a minimum pressure loss of 33% through the Intermediate pressure turbine was possible.

5.1.2.5. Low pressure heater 4 tap off pressure

A new parametric table was created with 500 runs. The tap off pressure was taken as a percentage of the intermediate turbine inlet pressure. The tap off pressure percentage was run from 0.01% to 99.99%. The pressure, net mechanical work, cycle efficiency and the factor line were added as variables.

The results are plotted on a graph to get the optimal point

The minimum pressure that can be achieved for the tap off point is equal to the intermediate pressure turbine outlet pressure.

5.1.2.6. Low pressure heater 3 tap off pressure

A new parametric table was created with 500 runs. The tap off pressure was taken as a percentage of the low pressure turbine inlet pressure. The tap off pressure percentage was run from 0.01% to 99.99%. The pressure, net mechanical work, cycle efficiency and the factor line were added as variables.

The results are plotted on a graph to get the optimal point.

An assumption is made that a minimum pressure loss of 33% through the low pressure turbine is possible.

5.1.2.7. Low pressure heater 2 tap off pressure

A new parametric table was created with 500 runs. The tap off pressure was taken as a percentage of the low pressure turbine inlet pressure. The tap off pressure percentage was run from 0.01% to 99.99%. The pressure, net mechanical work, cycle efficiency and the factor line were added as variables.

The results are plotted on a graph to get the optimal point.

No change was visible to the net mechanical work or efficiency for different pressures.

5.1.2.8. Low pressure heater 1 tap off pressure

A new parametric table was created with 500 runs. The tap off pressure was taken as a percentage of the low pressure turbine inlet pressure. The tap off pressure percentage was run from 0.01% to 99.99%. The pressure, net mechanical work, cycle efficiency and the factor line were added as variables.

The results are plotted on a graph to get the optimal point

No change visible to the net mechanical work or efficiency for different pressures.

5.2. Super-critical Rankine cycle

This cycle was programmed in EES by using the heat balance diagram of Medupi power station at maximum output (794 MW).

5.2.1. Rankine cycle without optimisation

The same programming method used in 5.1 applies for the super-critical cycle except the cycle does not have a low pressure heater 4, therefore the only difference is the low pressure heater 3 steam tap off point is situated on the Intermediate pressure turbine instead of the low pressure turbine as in the above sub-critical cycle. Only the new tap off point will be discussed for the super-critical cycle.

The pressure is taken as a percentage of the intermediate pressure turbine inlet pressure. The correct pressure is taken from the heat balance diagram, thus the first variable is known. Because the water is still saturated liquid the second variable is known and the rest of the variables can be calculated.

But because the steam taps off at the intermediate pressure turbine the efficiency of the intermediate pressure turbine must be taken into account. The following calculation must be implemented.

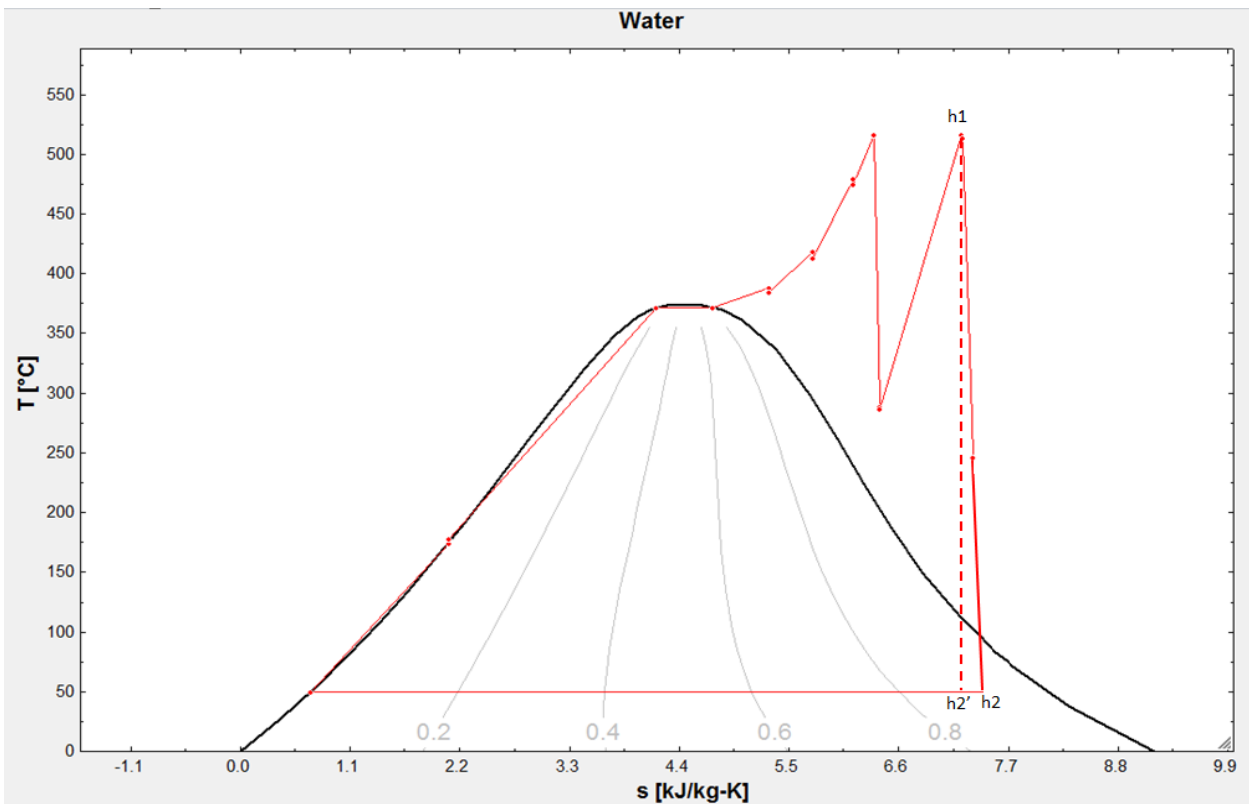


Figure 28: Illustration of the efficiency effect of the intermediate pressure turbine on the Rankine cycle

$$S_1 = S_{2'}$$

$$h_{2'} = \text{with pressure and } S_{2'} \text{ as input}$$

$$IPT \eta = \frac{h_1 - h_2}{h_1 - h_{2'}}$$

With this equation h_2 can be calculated for the pressure of low pressure heater 3.

5.3. Sub- and super-critical Rankine cycle optimisation

Two optimisation methods are used to determine whether the order of optimisation makes a difference.

5.3.1. Optimisation method 1

This method was based on the following steam tap off order.

Feed pump – High pressure turbine expansion pressure – High pressure heater 6 – De-aerator – Low pressure heater 4 (*only applies for sub-critical cycle*) – Low pressure heater 3 - Low pressure heater 2 - Low pressure heater 1.

Then the cycle repeats itself for the second and third runs. After the third run the boiler feed pump pressure and high pressure expansion pressure were optimised to see whether any values differed.

After the third run the values became stable and there was no need for any further runs.

The optimisation method is the same for super-critical cycle except low pressure heater 4 does not exist.

5.3.2. Optimisation method 2

This method was based on the following steam tap off order.

Feed pump – High pressure heater 6 – High pressure turbine expansion pressure – Low pressure heater 1 – Low pressure heater 2 - Low pressure heater 3 - Low pressure heater 4 (*only applies for sub-critical cycle*).

Then the cycle repeats for the second and third run. After the third run the boiler feed pump pressure and high pressure expansion pressure was optimised to see if any values differ.

After the third run the values become stable and there was no need for any further runs.

The optimisation method is the same for super-critical cycle except low pressure heater 4 doesn't exist.

6. Results of sub- and super-critical Rankine cycles

6.1. Sub-critical optimisation results

The sub critical cycle was based on the heat balance diagrams for Kriel power station at 500MW (full load) output.

6.1.1. Sub-critical cycle without optimisation

6.1.1.1. Input parameters

The input parameters are taken from the original heat balance diagram for Kriel power station.

Feed pump	22.15 MPa
Maximum Temperature	516°C
Minimum Temperature	40°C
Maximum mass flow	415 kg/s
Extraction pump discharge pressure	2.388 MPa
High pressure heater 6 bled steam tap off pressure	1.738 MPa
De-aerator bled steam tap off pressure	1.032 MPa
Low pressure heater 4 bled steam tap off pressure	0.4106 MPa
Low pressure heater 3 bled steam tap off pressure	0.1743 MPa
Low pressure heater 2 bled steam tap off pressure	0.06046 MPa
Low pressure heater 1 bled steam tap off pressure	0.02991 MPa

Table 1: Sub-critical boiler input parameters as on the heat balance diagram

6.1.1.2. Results

Cycle efficiency	42.69 %
Mega Watt out	538.4 MW
Total Work in	28.68 kJ
Total Work Out	1326 kJ
Net mechanical work	1297 kJ
Heat added	3039 kJ
Heat rejected	1553 kJ

Table 2: Results for sub-critical cycle from input parameters before optimisation

Water

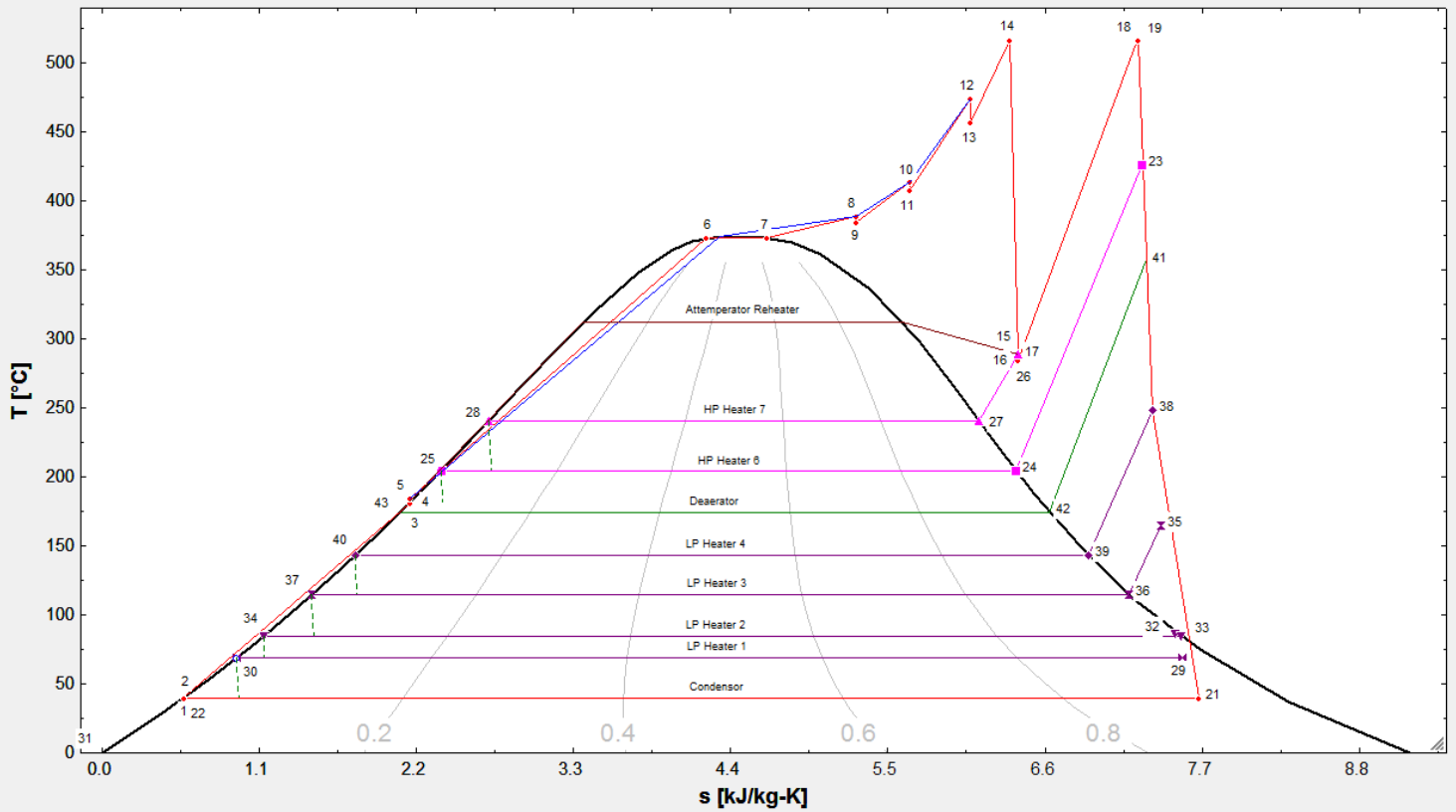


Figure 29: T-s diagram after programmed on EES for sub-critical cycle before any optimisation

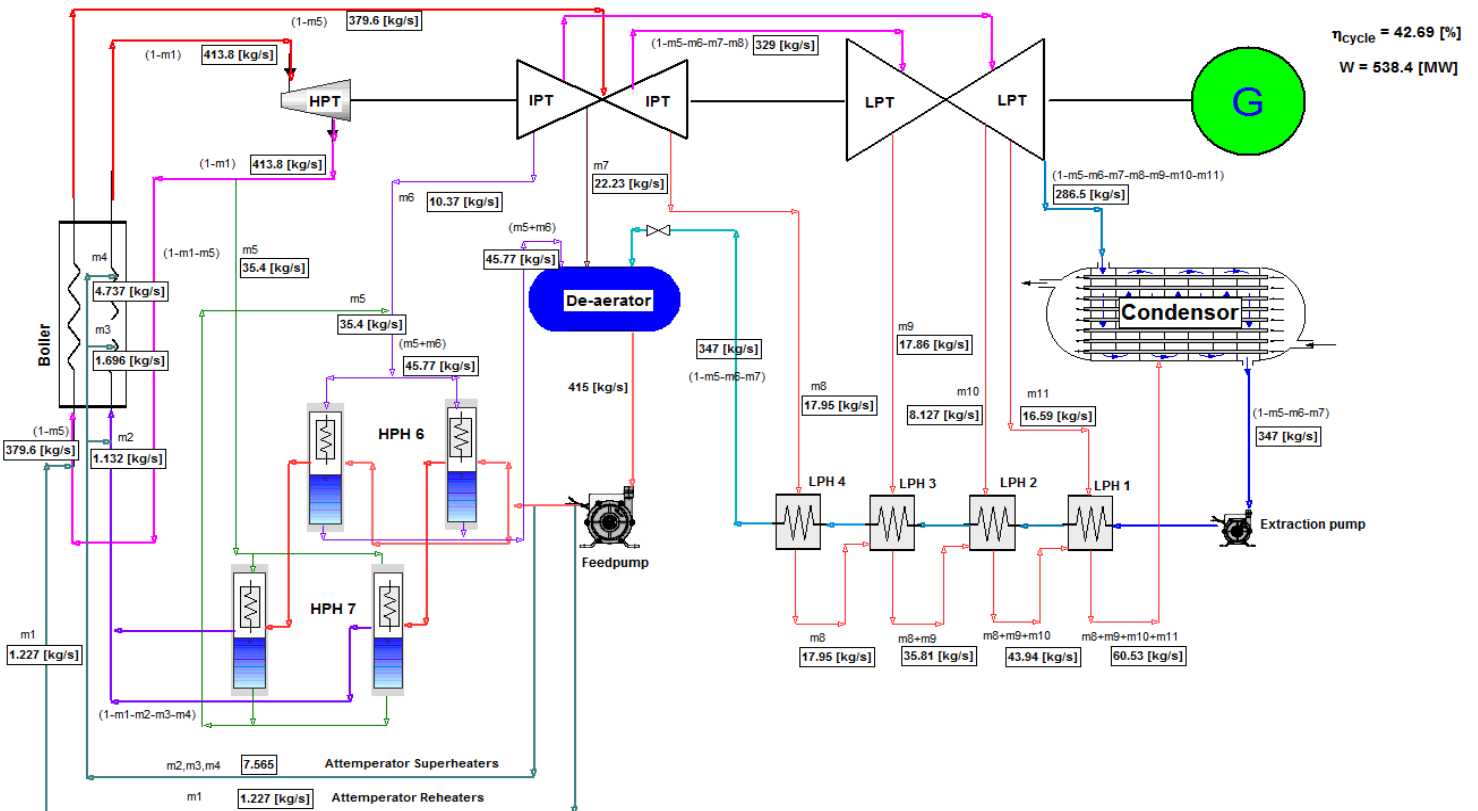


Figure 30: Illustration of Kriel power station plant layout

6.1.2. Sub-critical cycle optimisation – first method

6.1.2.1. First run

6.1.2.1.1. Boiler feed pump pressure

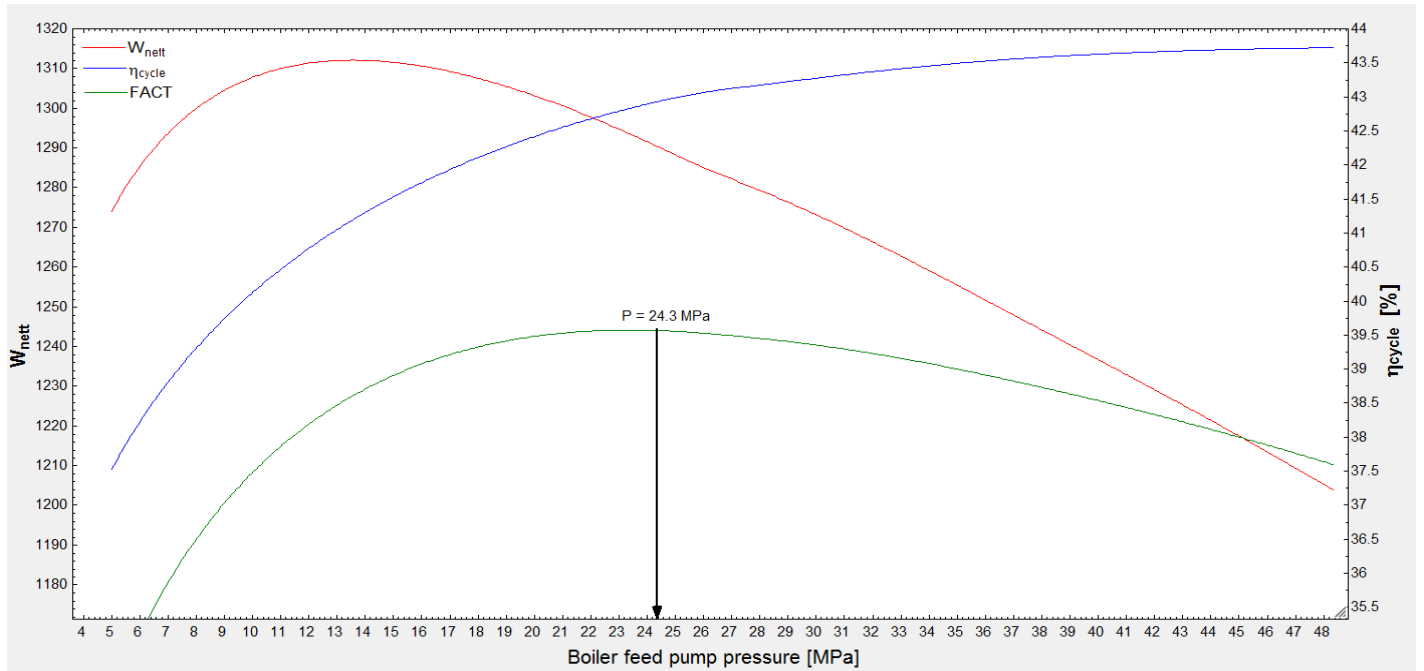


Figure 32: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (1st method, 1st run)

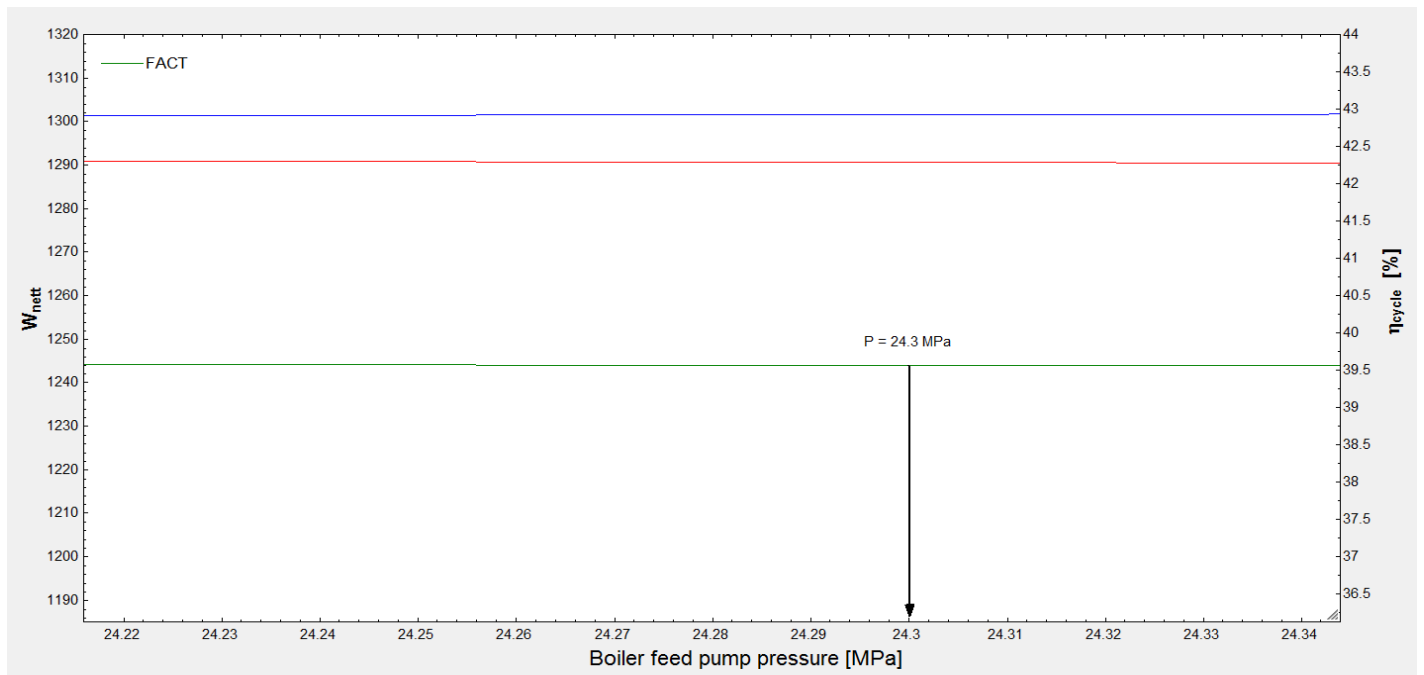


Figure 31: Cycle efficiency, net-work and the factor line plotted against boiler feed pump discharge pressure (Zoomed in) (1st method, 1st run)

Before optimisation		After optimisation	
Boiler feed pump pressure	22.14 MPa	Boiler feed pump pressure	24.3 MPa
Cycle efficiency	42.69%	Cycle efficiency	42.92%
Mega Watt output	538.4 MW	Mega Watt output	535.6 MW
Net mechanical work	1397 kJ	Net mechanical work	1291 kJ
Heat added	3039 kJ	Heat added	3007 kJ
Heat rejected	1553 kJ	Heat rejected	1525 kJ

Table 3: Before and after optimisation results for the boiler feed pump pressure (1st method, 1st run)

6.1.2.1.2. High pressure turbine expansion

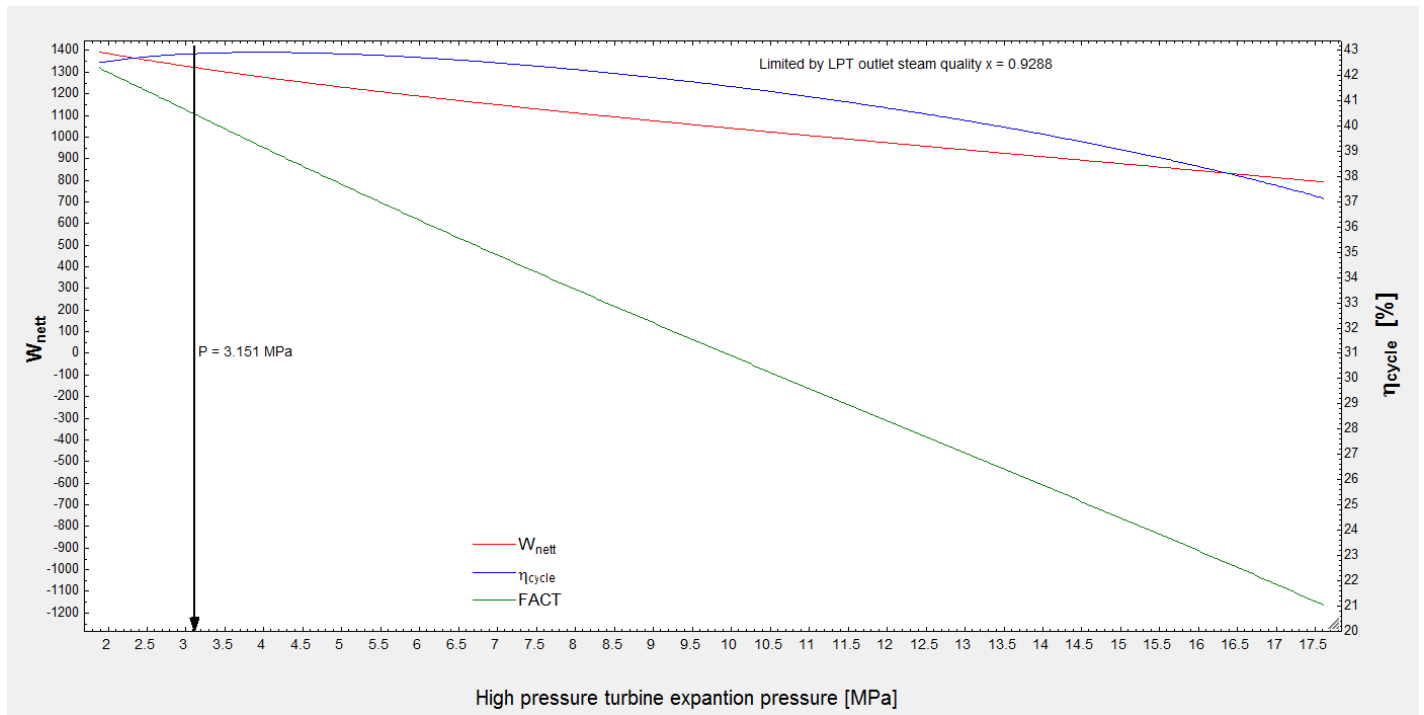


Figure 33: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (1st method, 1st run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.715 MPa	High pressure turbine expansion pressure	3.151 MPa
Cycle efficiency	42.92%	Cycle efficiency	42.88%
Mega Watt output	535.6 MW	Mega Watt output	547.5 MW
Net mechanical work	1291 kJ	Net mechanical work	1319 kJ
Heat added	3007 kJ	Heat added	3077 kJ
Heat rejected	1525 kJ	Heat rejected	1575 kJ
LPT outlet steam quality	0.9196	LPT outlet steam quality	0.9288

Table 4: Before and after optimisation results for the high pressure turbine expansion pressure (1st method, 1st run)

6.1.2.1.3. High pressure heater 6

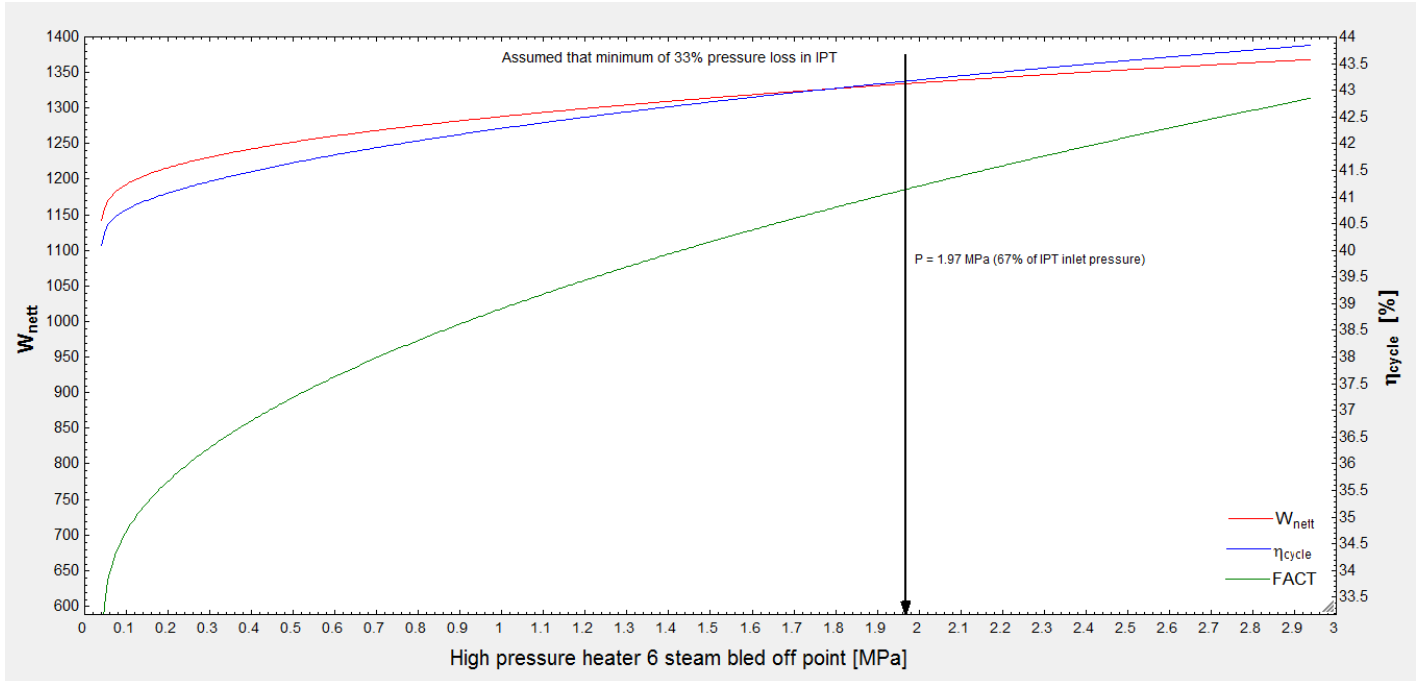


Figure 34: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (1st method, 1st run)

Before optimisation		After optimisation	
High pressure heater 6 steam tap off pressure	1.614 MPa	High pressure heater 6 steam tap off pressure	1.97 MPa
Cycle efficiency	42.88%	Cycle efficiency	43.17%
Mega Watt output	547.5 MW	Mega Watt output	553.7 MW
Net mechanical work	1319 kJ	Net mechanical work	1334 kJ
Heat added	3077 kJ	Heat added	3091 kJ
Heat rejected	1575 kJ	Heat rejected	1584 kJ

Table 5: Before and after optimisation results for high pressure heater 6 steam tap off pressure (1st method, 1st run)

6.1.2.1.4. De-aerator

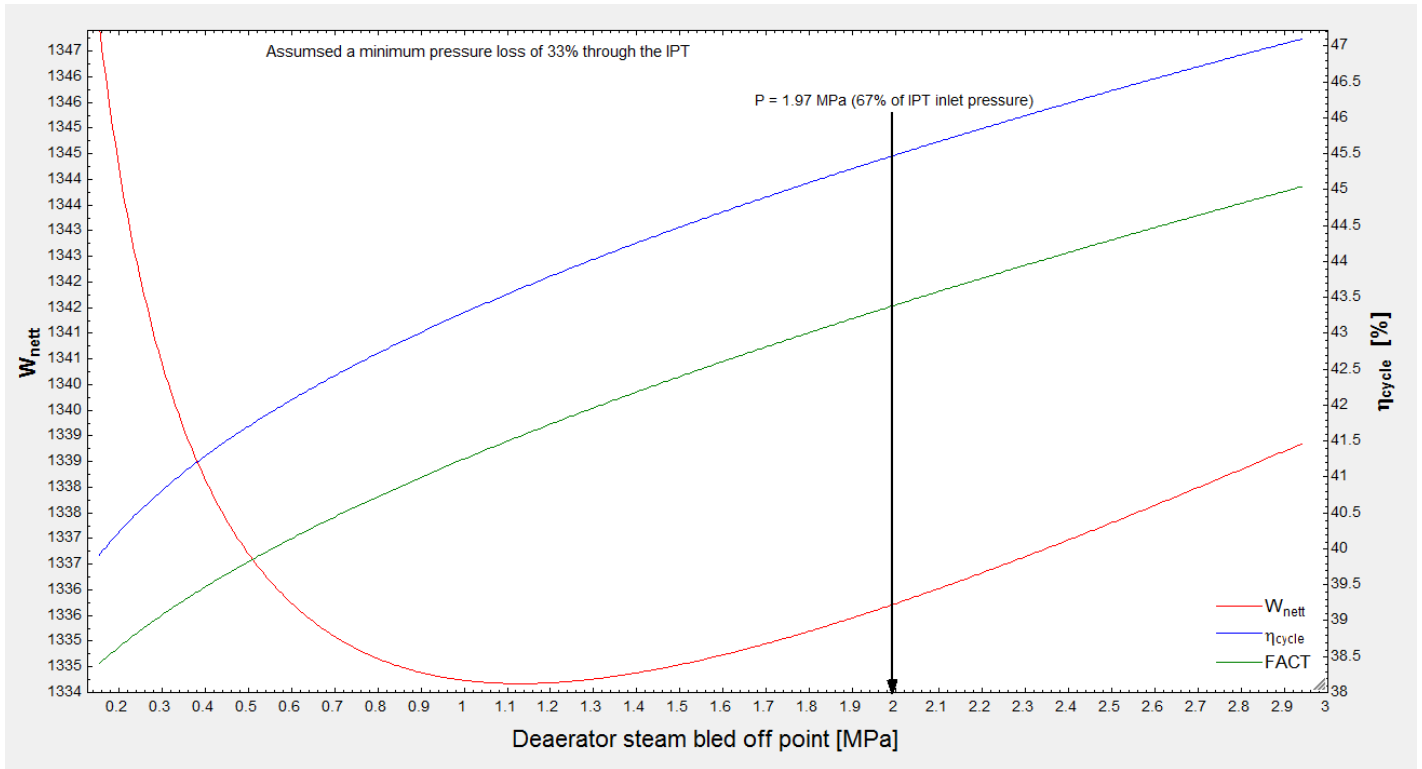


Figure 35: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
High pressure heater 6 steam tap off pressure	0.9581 MPa	High pressure heater 6 steam tap off pressure	1.97 MPa
Cycle efficiency	43.17%	Cycle efficiency	45.43%
Mega Watt output	553.7 MW	Mega Watt output	554.3 MW
Net mechanical work	1334 kJ	Net mechanical work	1336 kJ
Heat added	3091 kJ	Heat added	2940 kJ
Heat rejected	1584 kJ	Heat rejected	1589 kJ

Table 6: Before and after optimisation results for the De-aerator steam tap off pressure (1st method, 1st run)

6.1.2.1.5. Low pressure heater 4

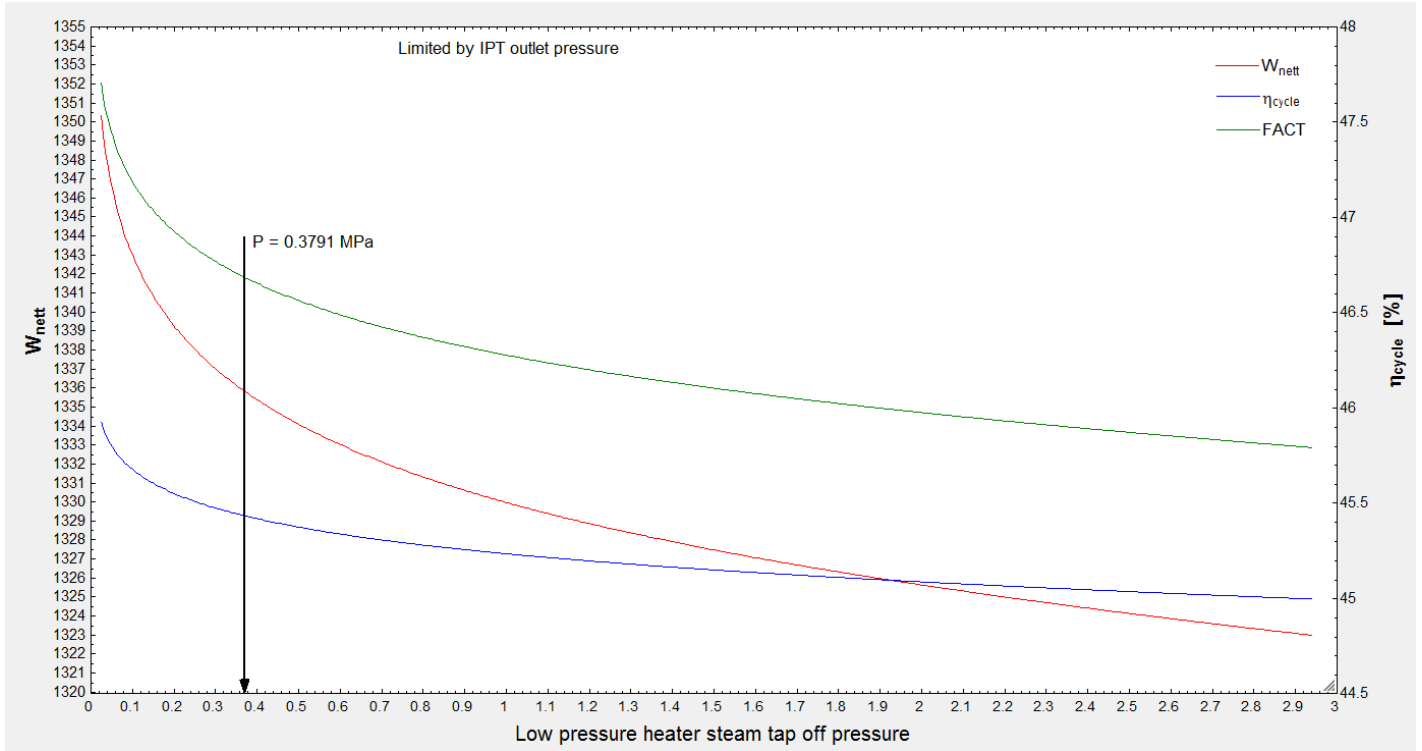


Figure 36: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 4 steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 4 steam tap off pressure	0.3812 MPa	Low pressure heater 4 steam tap off pressure	0.3791 MPa
Cycle efficiency	45.43%	Cycle efficiency	45.43%
Mega Watt output	554.3 MW	Mega Watt output	554.3 MW
Net mechanical work	1336 kJ	Net mechanical work	1336 kJ
Heat added	2940 kJ	Heat added	2940 kJ
Heat rejected	1589 kJ	Heat rejected	1589 kJ

Table 7: Before and after optimisation results for low pressure heater 4 steam tap off pressure (1st method, 1st run)

6.1.2.1.6. Low pressure heater 3

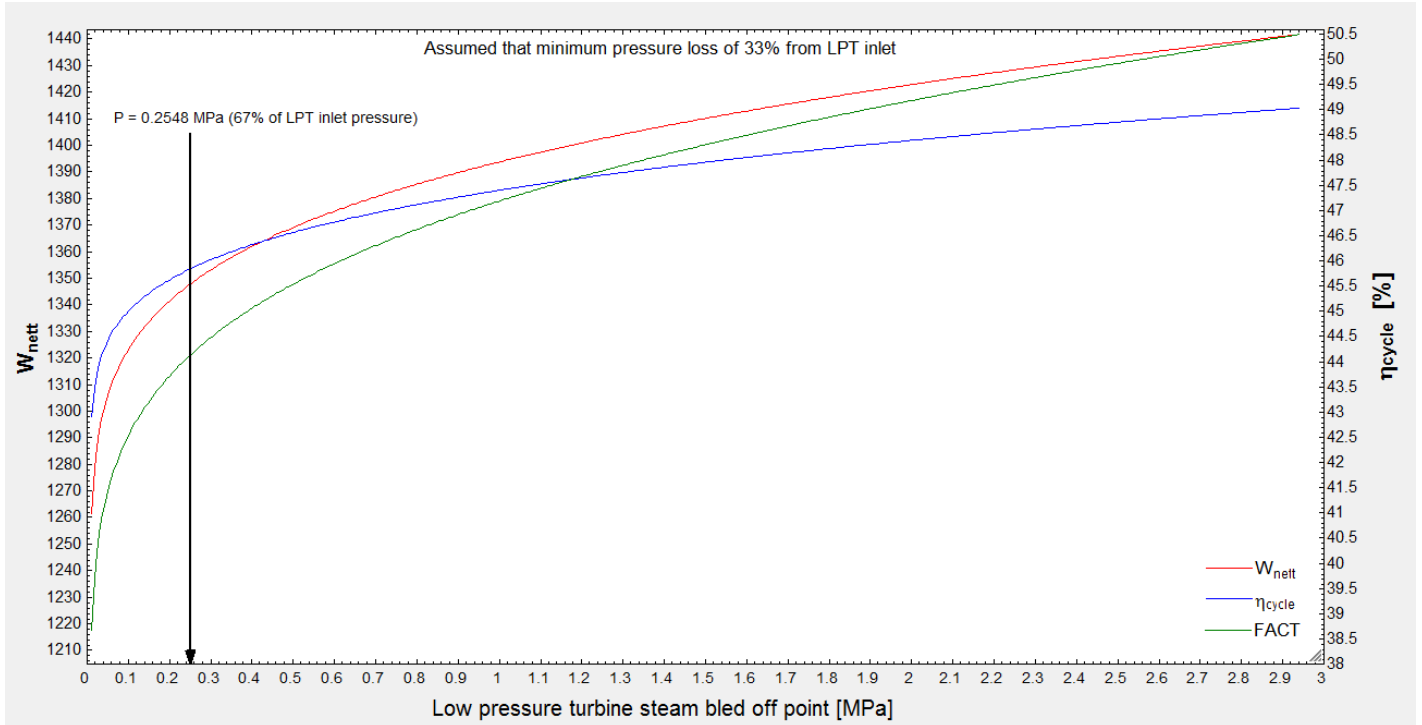


Figure 37: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.1618 MPa	Low pressure heater 3 steam tap off pressure	0.2548 MPa
Cycle efficiency	45.43%	Cycle efficiency	45.86%
Mega Watt output	554.3 MW	Mega Watt output	559.6 MW
Net mechanical work	1336 kJ	Net mechanical work	1348 kJ
Heat added	2940 kJ	Heat added	2940 kJ
Heat rejected	1589 kJ	Heat rejected	1591 kJ

Table 8: Before and after optimisation results for low pressure heater 3 steam tap off pressure (1st method, 1st run)

6.1.2.1.7. Low pressure heater 2

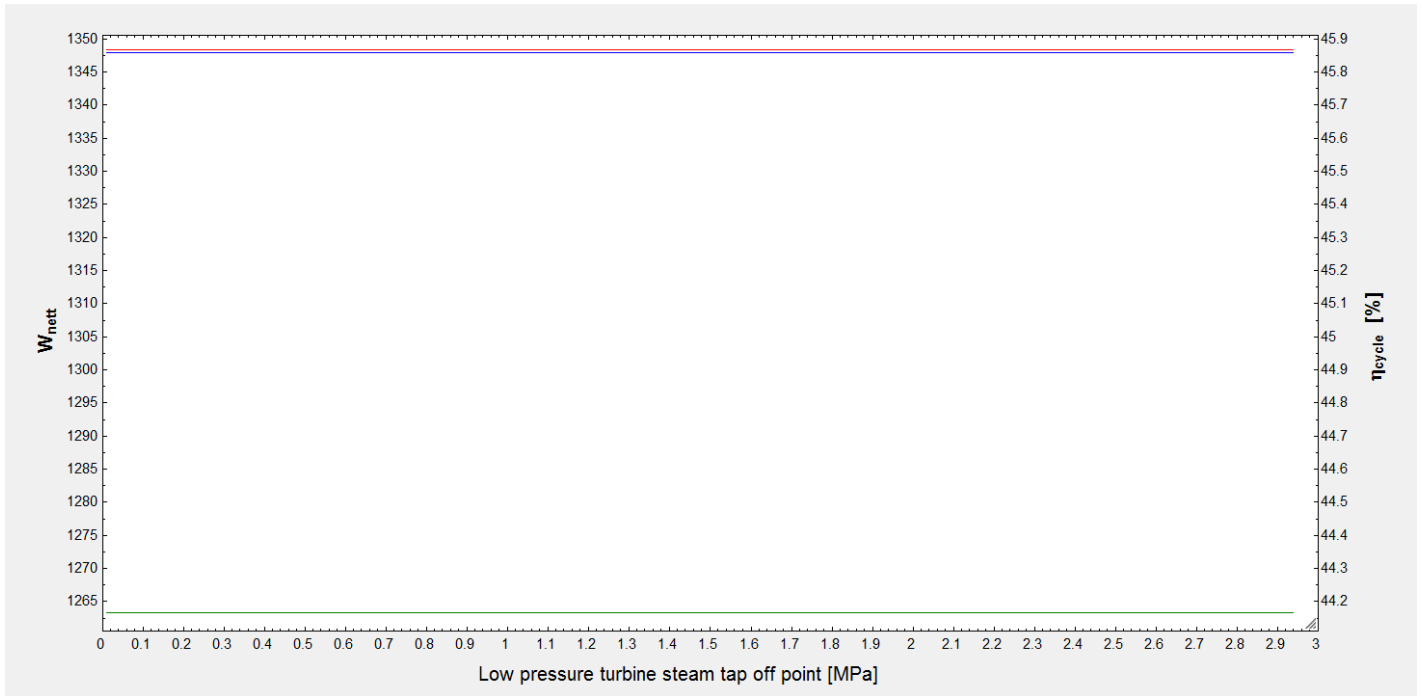


Figure 38: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.05614 MPa	Low pressure heater 2 steam tap off pressure	0.05614 MPa
Cycle efficiency	45.86%	Cycle efficiency	45.86%
Mega Watt output	559.6 MW	Mega Watt output	559.6 MW
Net mechanical work	1348 kJ	Net mechanical work	1348 kJ
Heat added	2940 kJ	Heat added	2940 kJ
Heat rejected	1591 kJ	Heat rejected	1591 kJ

Table 9: Before and after optimisation results for low pressure heater 2 steam tap off pressure (1st method, 1st run)

6.1.2.1.8. Low pressure heater 1

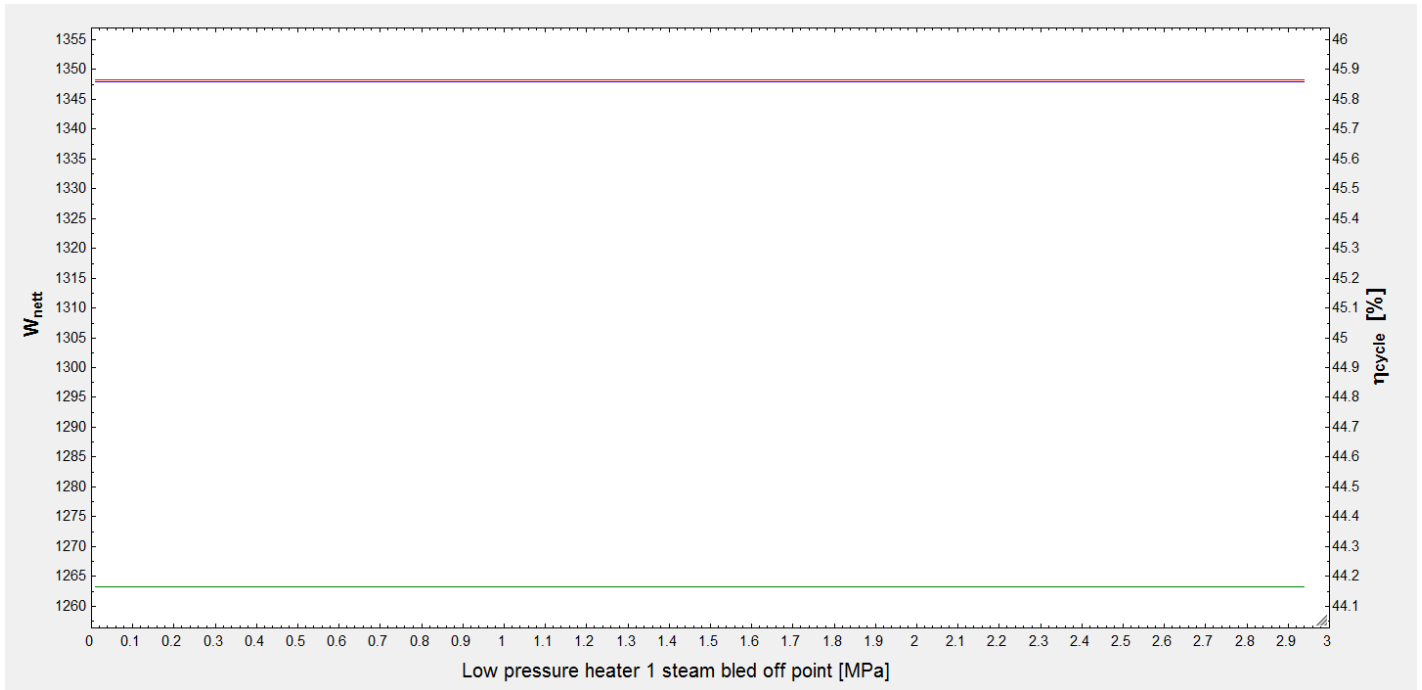


Figure 39: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.05614 MPa	Low pressure heater 1 steam tap off pressure	0.05614 MPa
Cycle efficiency	45.86%	Cycle efficiency	45.86%
Mega Watt output	559.6 MW	Mega Watt output	559.6 MW
Net mechanical work	1348 kJ	Net mechanical work	1348 kJ
Heat added	2940 kJ	Heat added	2940 kJ
Heat rejected	1591 kJ	Heat rejected	1591 kJ

Table 10: Before and after optimisation results for low pressure heater 4 steam tap off pressure (1st method, 1st run)

6.1.2.2. Second run

6.1.2.2.1. Boiler feed pump pressure

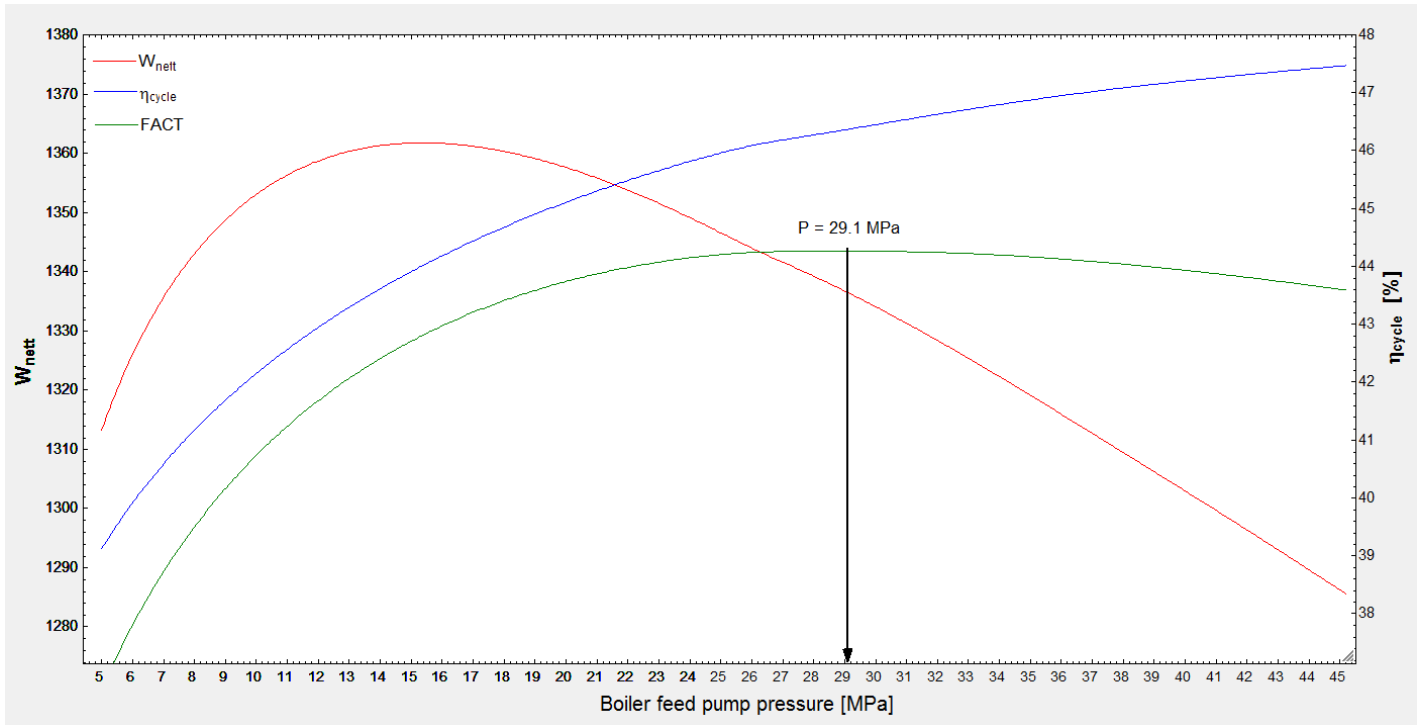


Figure 40: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (1st method, 2nd run)

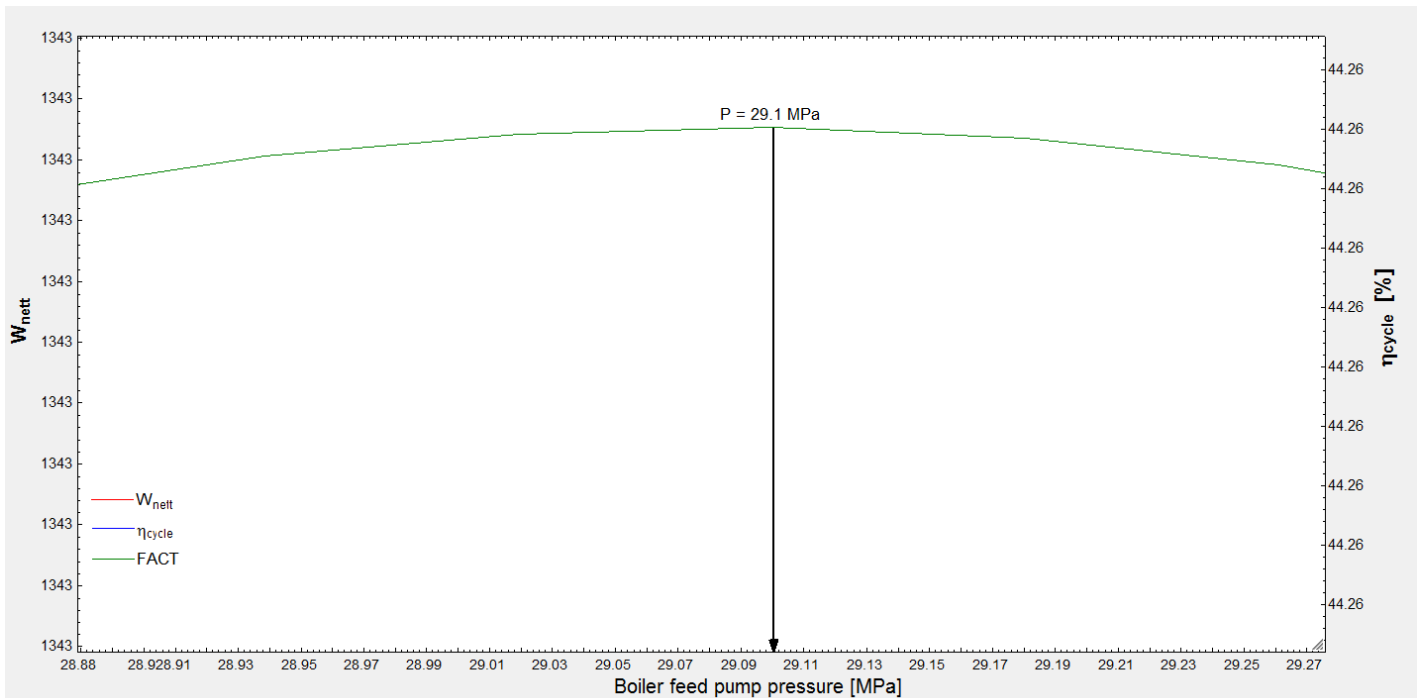


Figure 41: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (1st method, 2nd run)

Before optimisation		After optimisation	
Boiler feed pump pressure	24.3 MPa	Boiler feed pump pressure	29.1 MPa
Cycle efficiency	45.86%	Cycle efficiency	46.37%
Mega Watt output	559.6 MW	Mega Watt output	554.6 MW
Net mechanical work	1348 kJ	Net mechanical work	1336 kJ
Heat added	2940 kJ	Heat added	2882 kJ
Heat rejected	1591 kJ	Heat rejected	1550 kJ

Table 11: Before and after optimisation results for the boiler feed pump discharge pressure (1st method, 2nd run)

6.1.2.2.2. High pressure turbine expansion

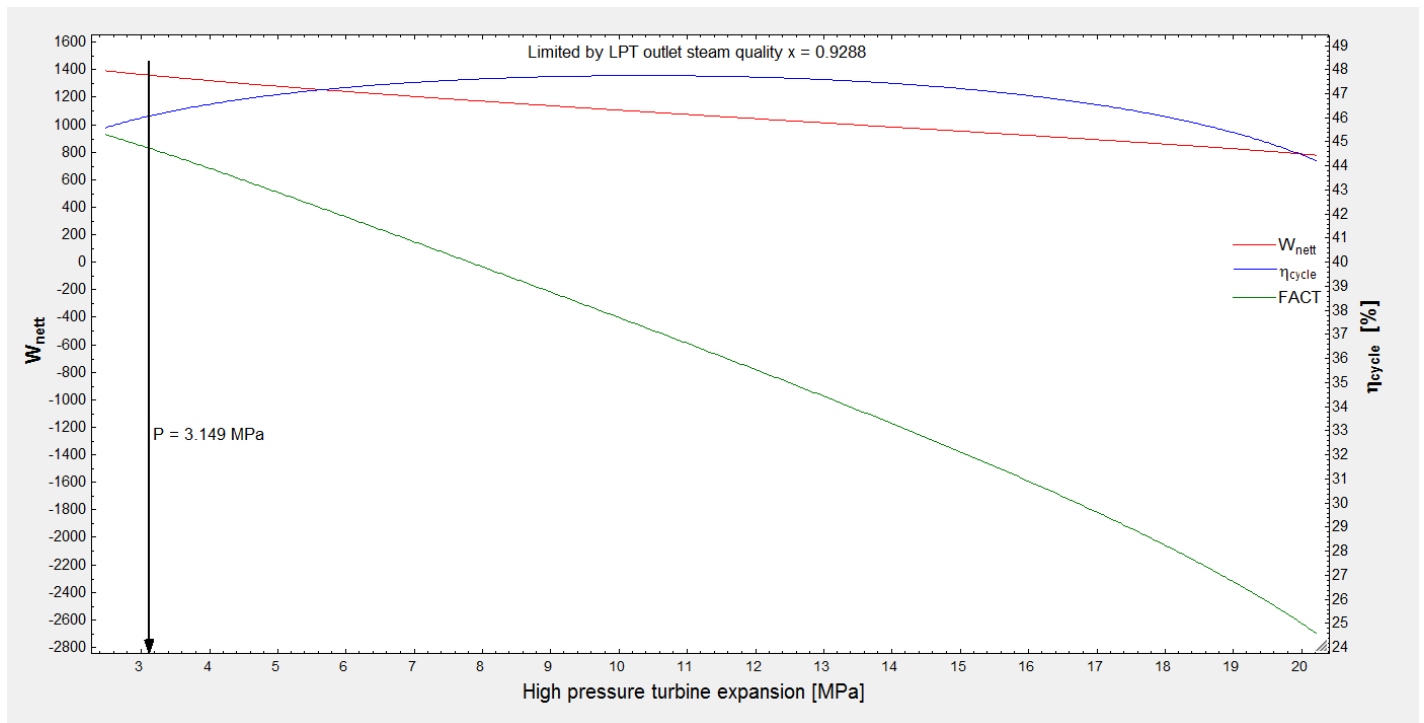


Figure 42: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (1st method, 2nd run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.618 MPa	High pressure turbine expansion pressure	3.149 MPa
Cycle efficiency	46.37%	Cycle efficiency	46.09%
Mega Watt output	554.6 MW	Mega Watt output	563.5 MW
Net mechanical work	1336 kJ	Net mechanical work	1358 kJ
Heat added	2882 kJ	Heat added	2946 kJ
Heat rejected	1550 kJ	Heat rejected	1592 kJ
LPT outlet steam quality	0.9288	LPT outlet steam quality	0.9288

Table 12: Before and after optimisation results for the high pressure turbine expansion pressure (1st method, 2nd run)

6.1.2.2.3. High pressure heater 6

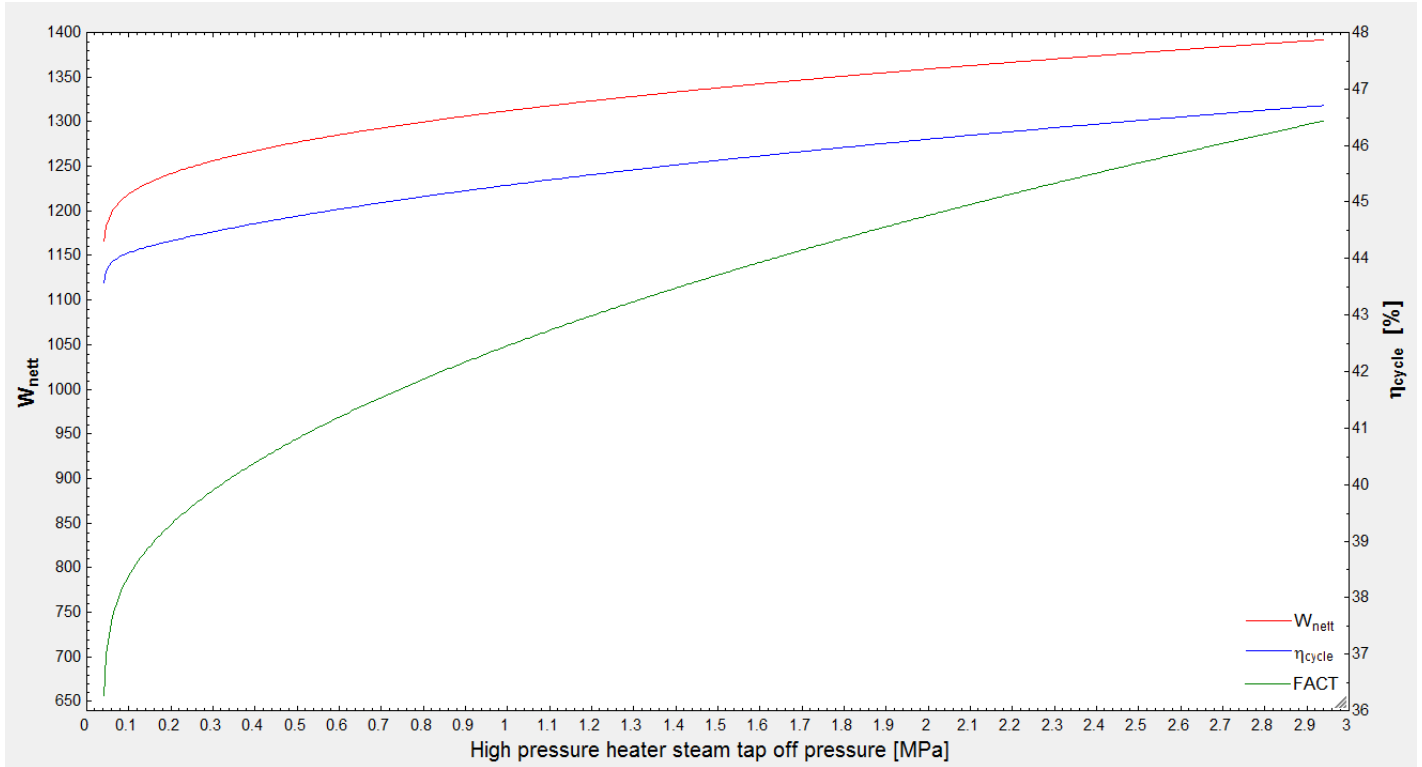


Figure 43: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
High pressure heater 6 steam tap off pressure	1.97 MPa	High pressure heater 6 steam tap off pressure	1.97 MPa
Cycle efficiency	46.09%	Cycle efficiency	46.09%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 13: Before and after optimisation results for high pressure heater 6 steam tap off pressure (1st method, 2nd run)

6.1.2.2.4. De-aerator

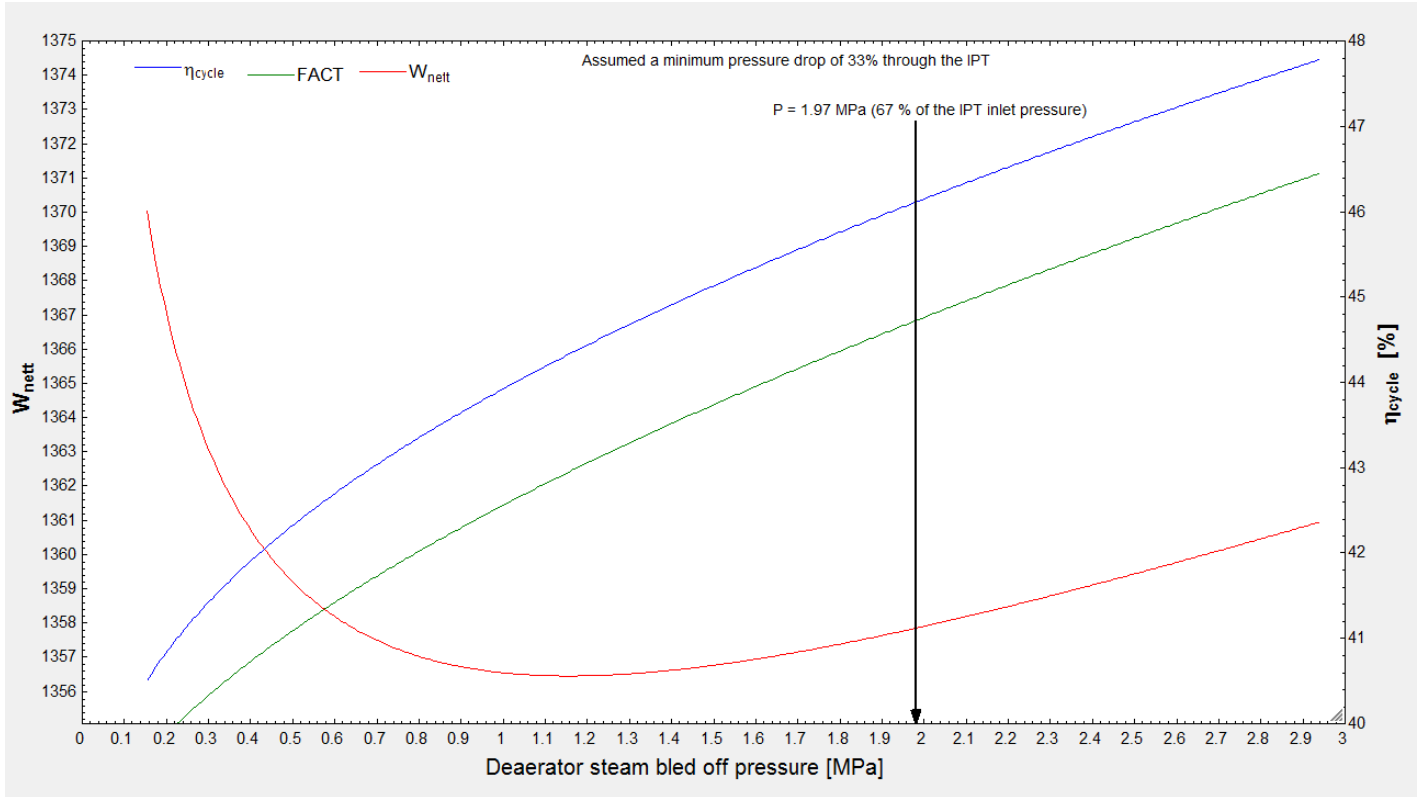


Figure 44: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
De-aerator steam tap off pressure	1.97 MPa	De-aerator steam tap off pressure	1.97 MPa
Cycle efficiency	46.09%	Cycle efficiency	46.09%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 14: Before and after optimisation results for the de-aerator steam tap off pressure (1st method, 2nd run)

6.1.2.2.5. Low pressure heater 4

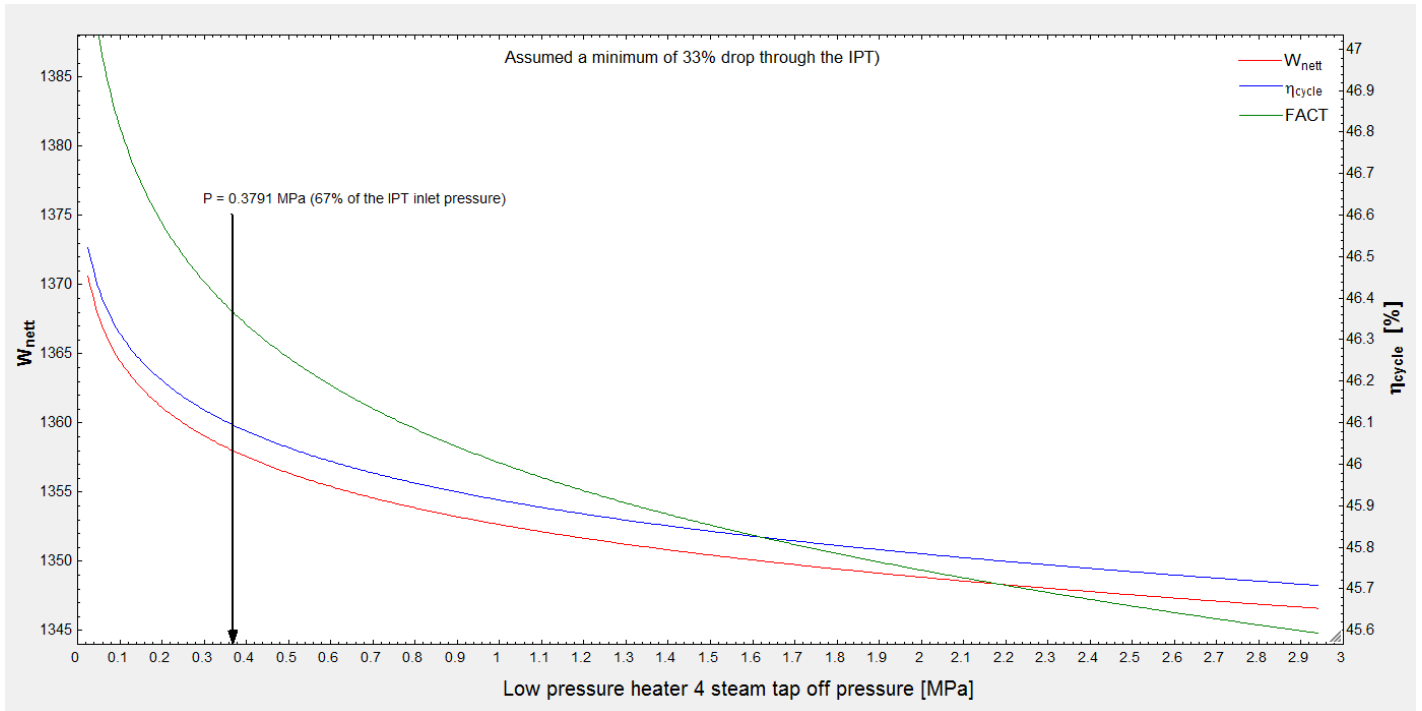


Figure 45: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 4 steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 4 steam tap off pressure	0.3791 MPa	Low pressure heater 4 steam tap off pressure	0.3791 MPa
Cycle efficiency	46.09%	Cycle efficiency	46.09%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 15: Before and after optimisation results for low pressure heater 4 steam tap off pressure (1st method, 2nd run)

6.1.2.2.6. Low pressure heater 3

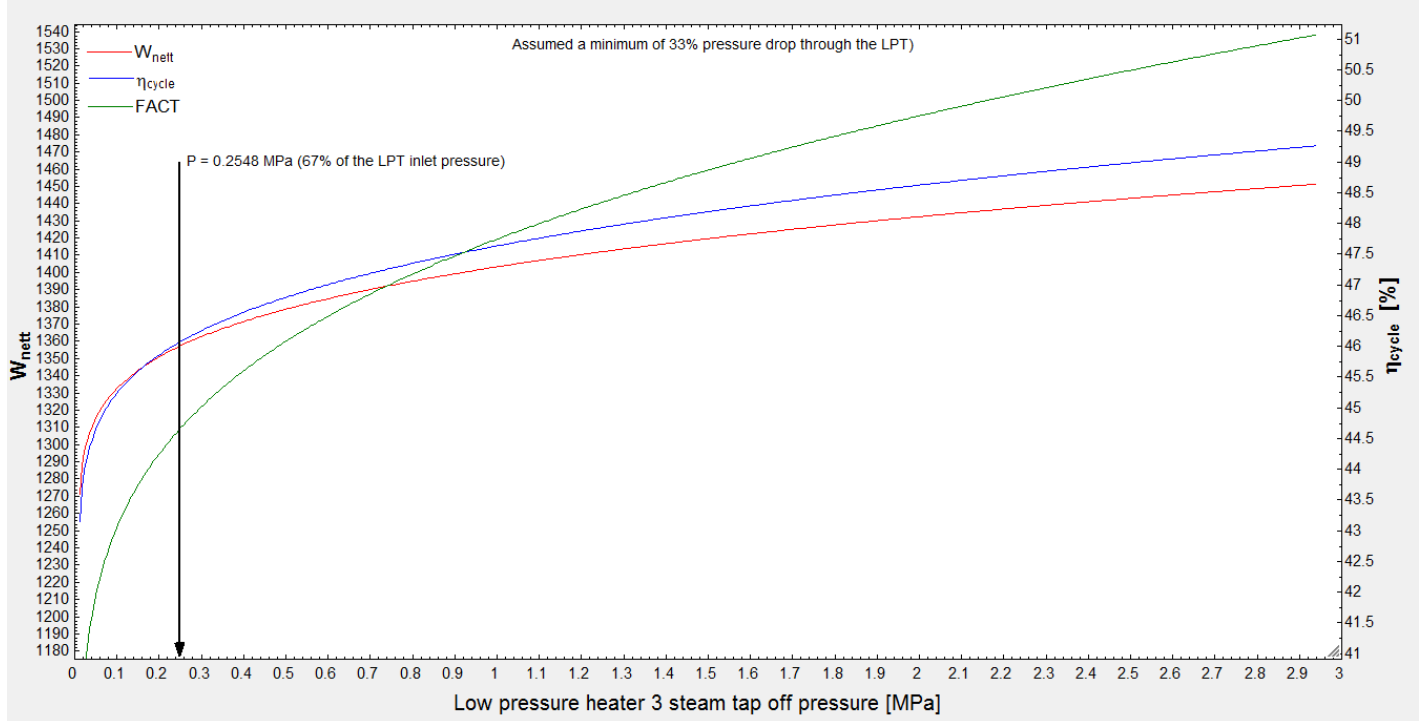


Figure 46: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.2548 MPa	Low pressure heater 3 steam tap off pressure	0.2548 MPa
Cycle efficiency	46.09%	Cycle efficiency	46.09%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 16: Before and after optimisation results for low pressure heater 3 steam tap off pressure (1st method, 2nd run)

6.1.2.2.7. Low pressure heater 2

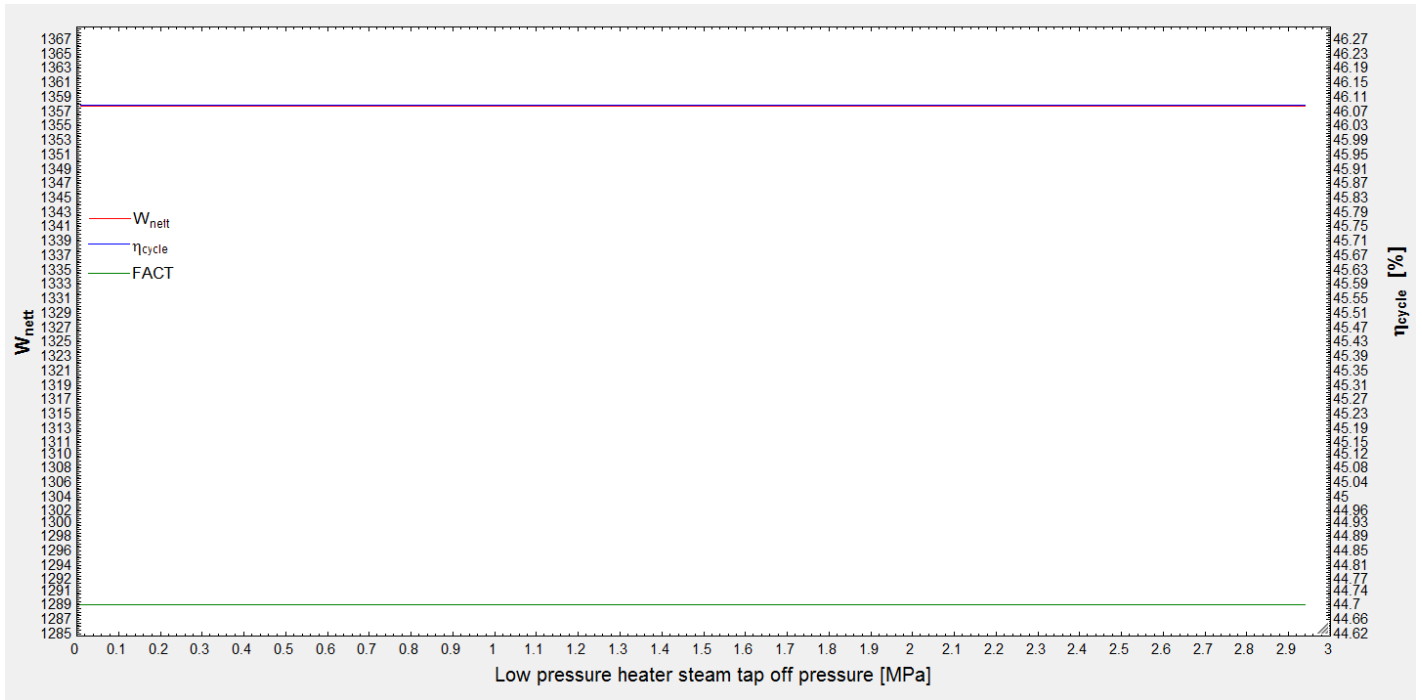


Figure 47: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.05613 MPa	Low pressure heater 2 steam tap off pressure	0.05613 MPa
Cycle efficiency	46.09%	Cycle efficiency	46.09%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 17: Before and after optimisation results for low pressure heater 2 steam tap off pressure (1st method, 2nd run)

6.1.2.2.8. Low pressure heater 1

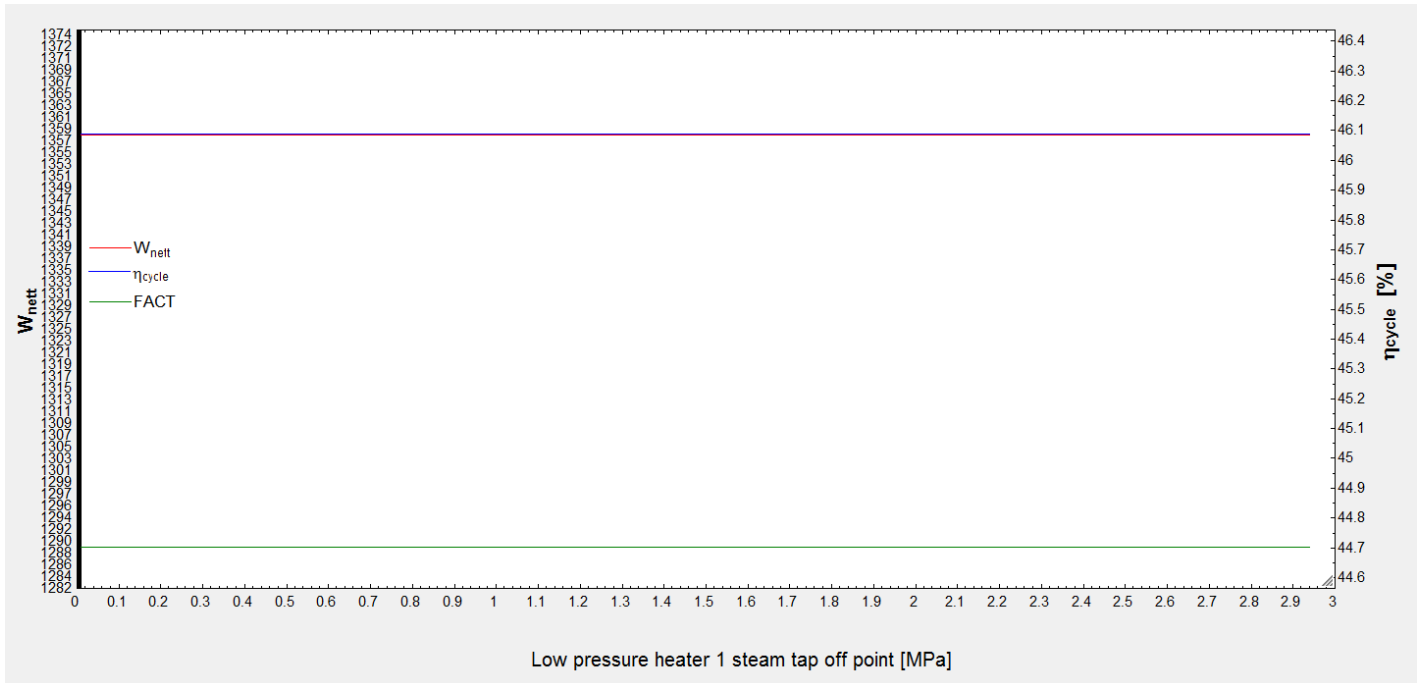


Figure 48: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.02777 MPa	Low pressure heater 1 steam tap off pressure	0.02777 MPa
Cycle efficiency	46.09%	Cycle efficiency	46.09%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 18: Before and after optimisation results for low pressure heater 1 steam tap off pressure (1st method, 2nd run)

6.1.2.3. Third run

6.1.2.3.1. Boiler feed pump pressure

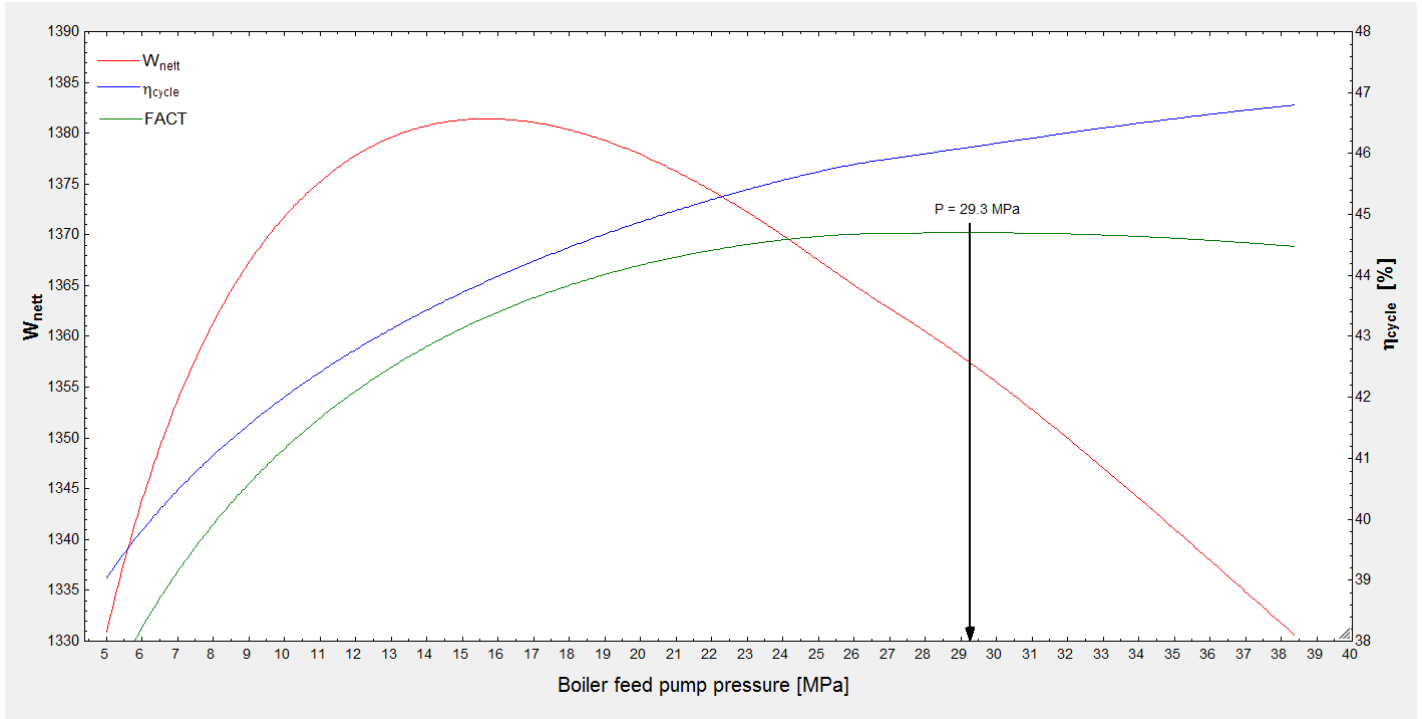


Figure 49: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (1st method, 3rd run)

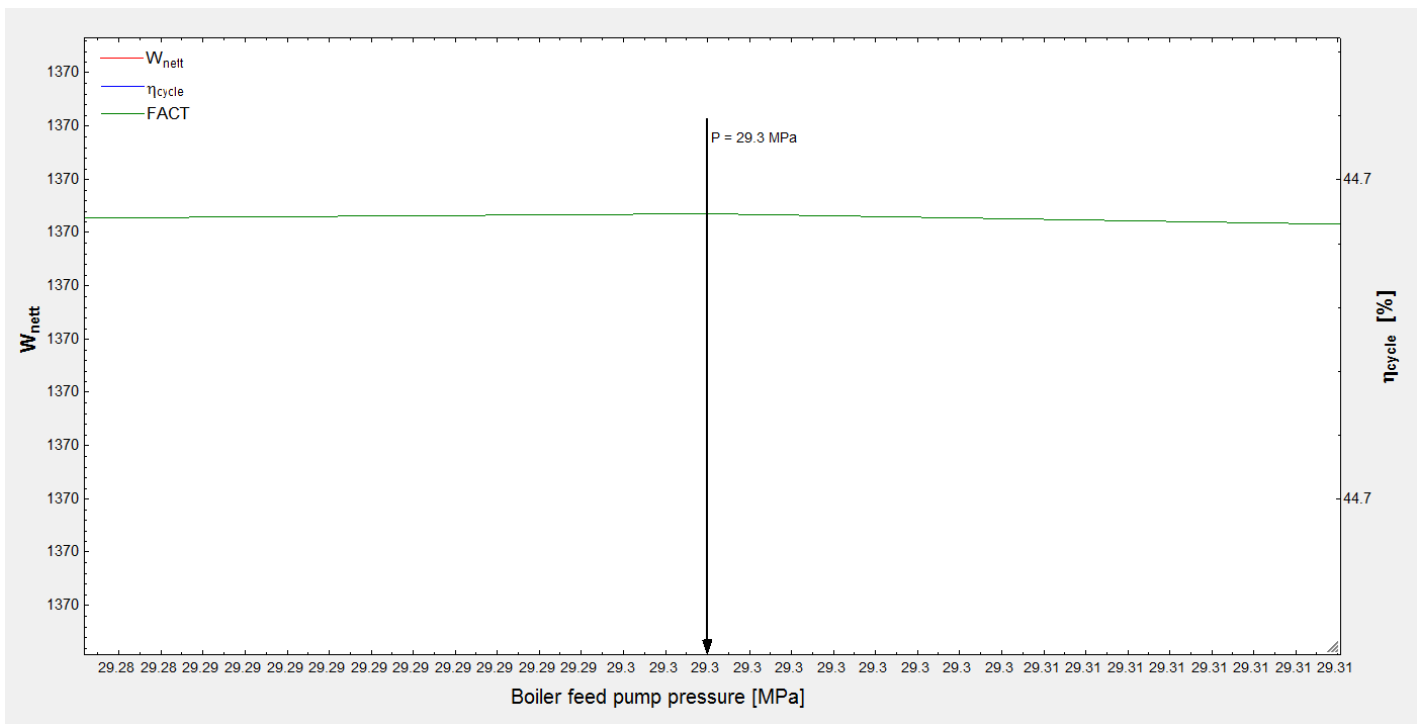


Figure 50: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (1st method, 3rd run)

Before optimisation		After optimisation	
Boiler feed pump pressure	29.1 MPa	Boiler feed pump pressure	29.3 MPa
Cycle efficiency	46.09%	Cycle efficiency	46.11%
Mega Watt output	563.5 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1357 kJ
Heat added	2946 kJ	Heat added	2944 kJ
Heat rejected	1592 kJ	Heat rejected	1591 kJ

Table 19: Before and after optimisation results for the boiler feed pump discharge pressure (1st method, 3rd run)

6.1.2.3.2. High pressure turbine expansion

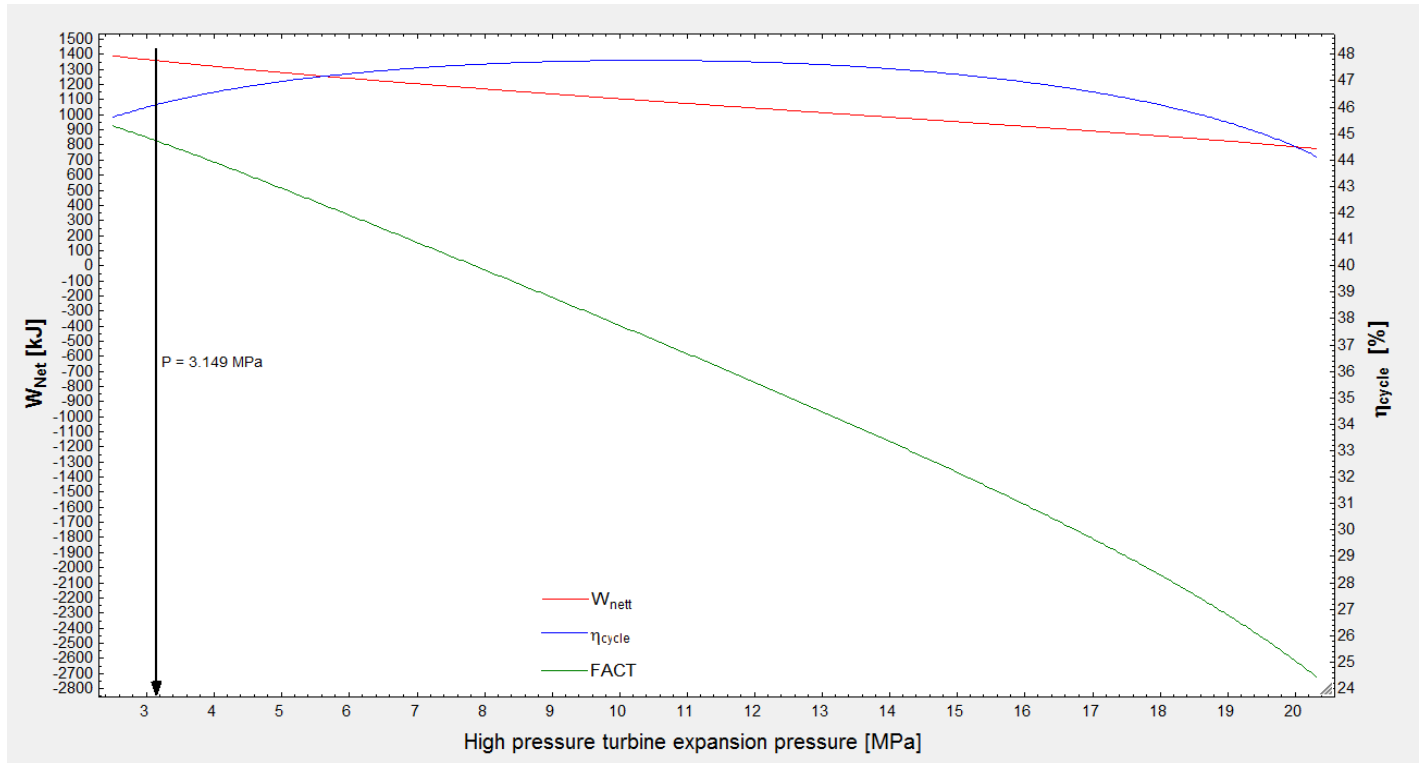


Figure 51: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (1st method, 3rd run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.165 MPa	High pressure turbine expansion pressure	3.149 MPa
Cycle efficiency	46.11%	Cycle efficiency	46.1%
Mega Watt output	563.3.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1357 kJ	Net mechanical work	1358 kJ
Heat added	2944 kJ	Heat added	2946 kJ
Heat rejected	1591 kJ	Heat rejected	1592 kJ
LPT outlet steam quality	0.9285	LPT outlet steam quality	0.9288

Table 20: Before and after optimisation results for the high pressure turbine expansion pressure (1st method, 3rd run)

6.1.2.3.3. High pressure heater 6

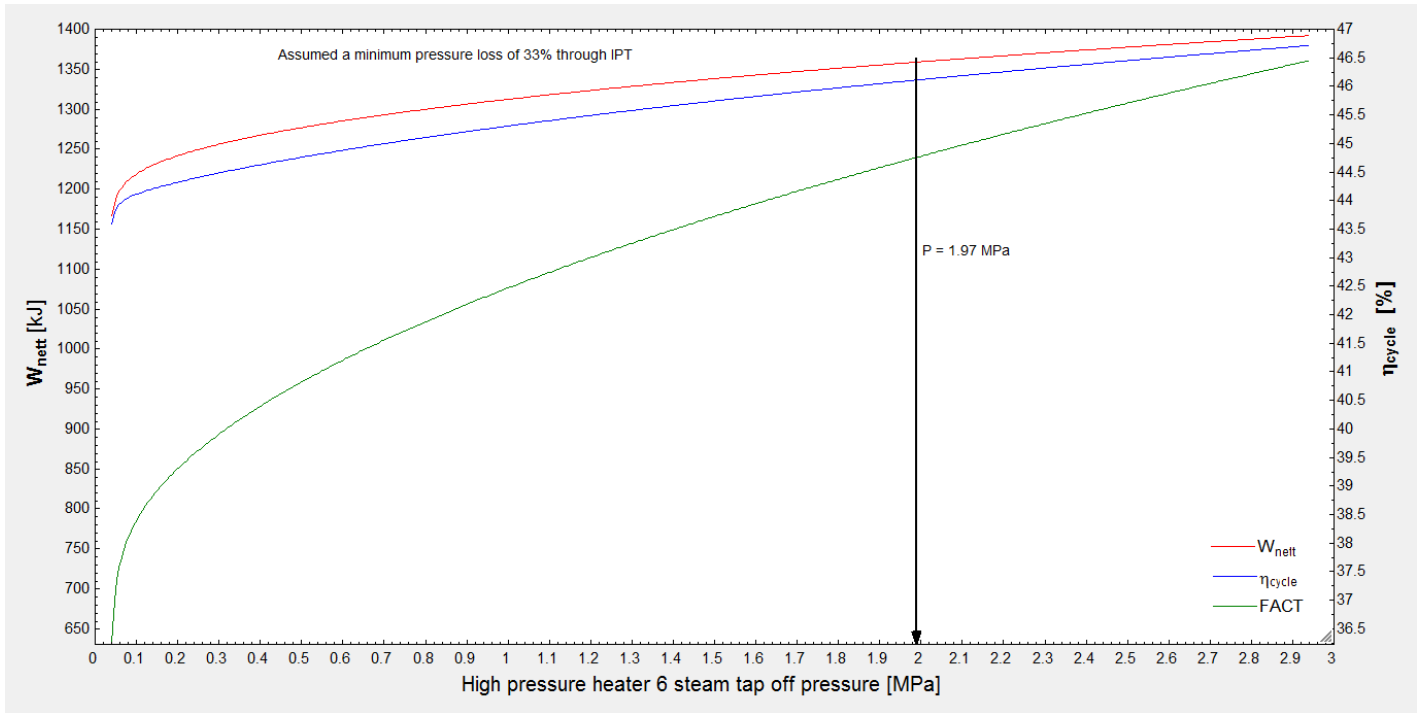


Figure 52: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
High pressure heater 6 steam tap off pressure	1.97 MPa	High pressure heater 6 steam tap off pressure	1.97 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 21: Before and after optimisation results for high pressure heater 6 steam tap off pressure (1st method, 3rd run)

6.1.2.3.4. De-aerator

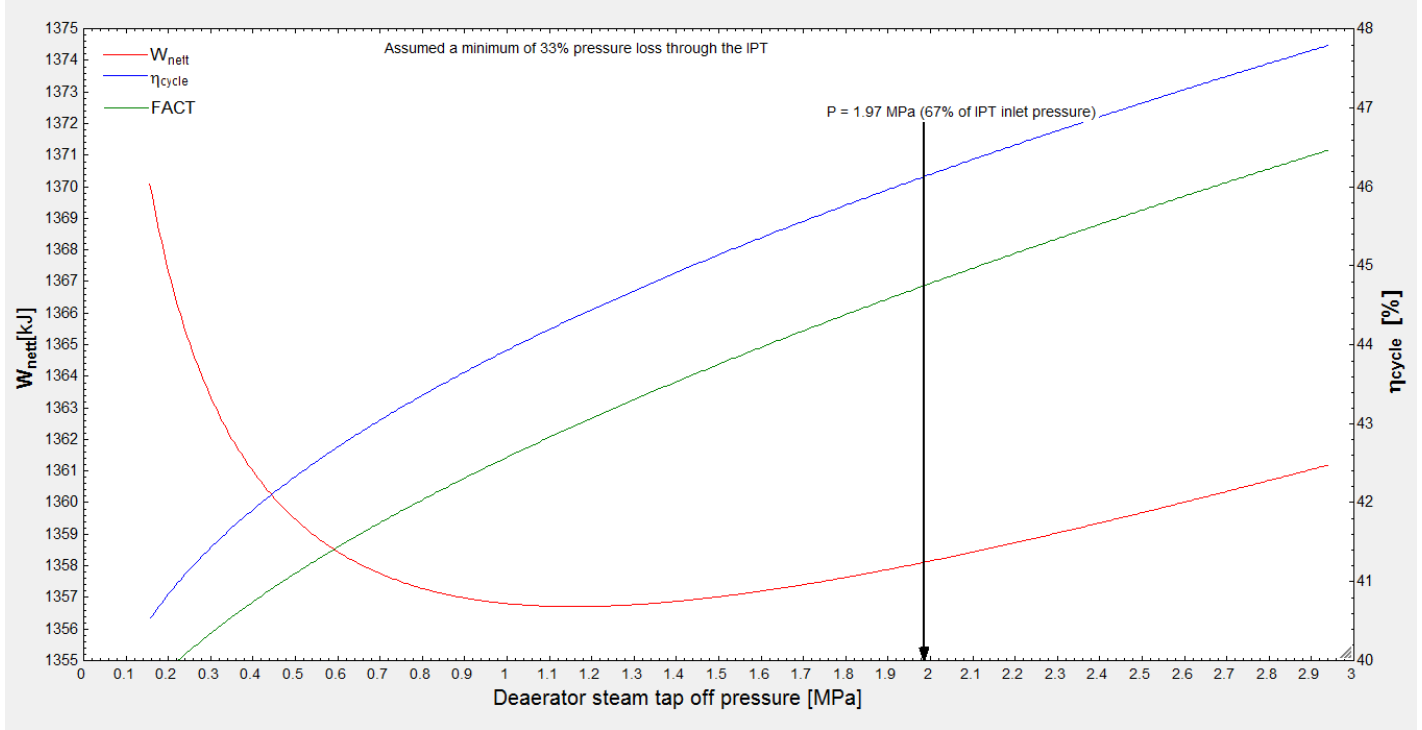


Figure 53: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
De-aerator steam tap off pressure	1.97 MPa	De-aerator steam tap off pressure	1.97 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 22: Before and after optimisation results for the de-aerator steam tap off pressure (1st method, 3rd run)

6.1.2.3.5. Low pressure heater 4

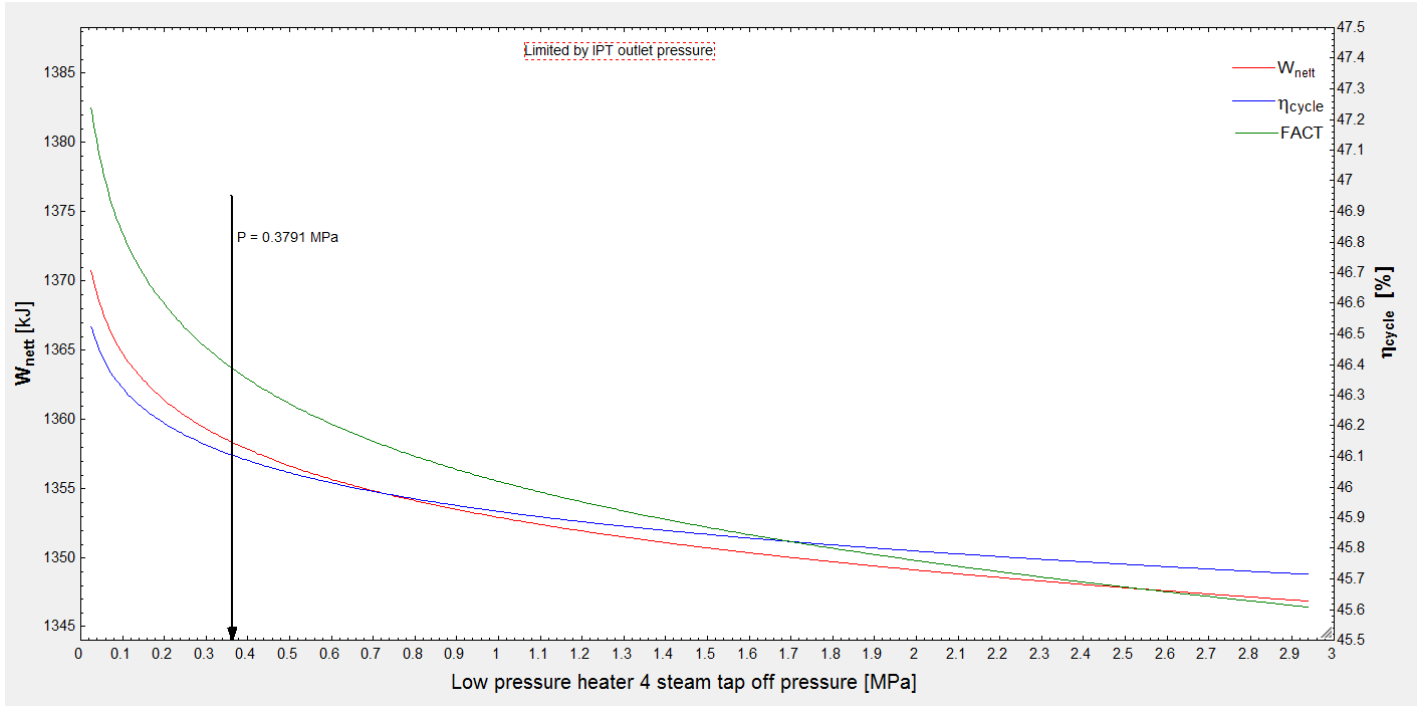


Figure 54: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 4 steam bled off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 4 steam tap off pressure	0.3791 MPa	Low pressure heater 4 steam tap off pressure	0.3791 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 23: Before and after optimisation results for low pressure heater 4 steam tap off pressure (1st method, 3rd run)

6.1.2.3.6. Low pressure heater 3

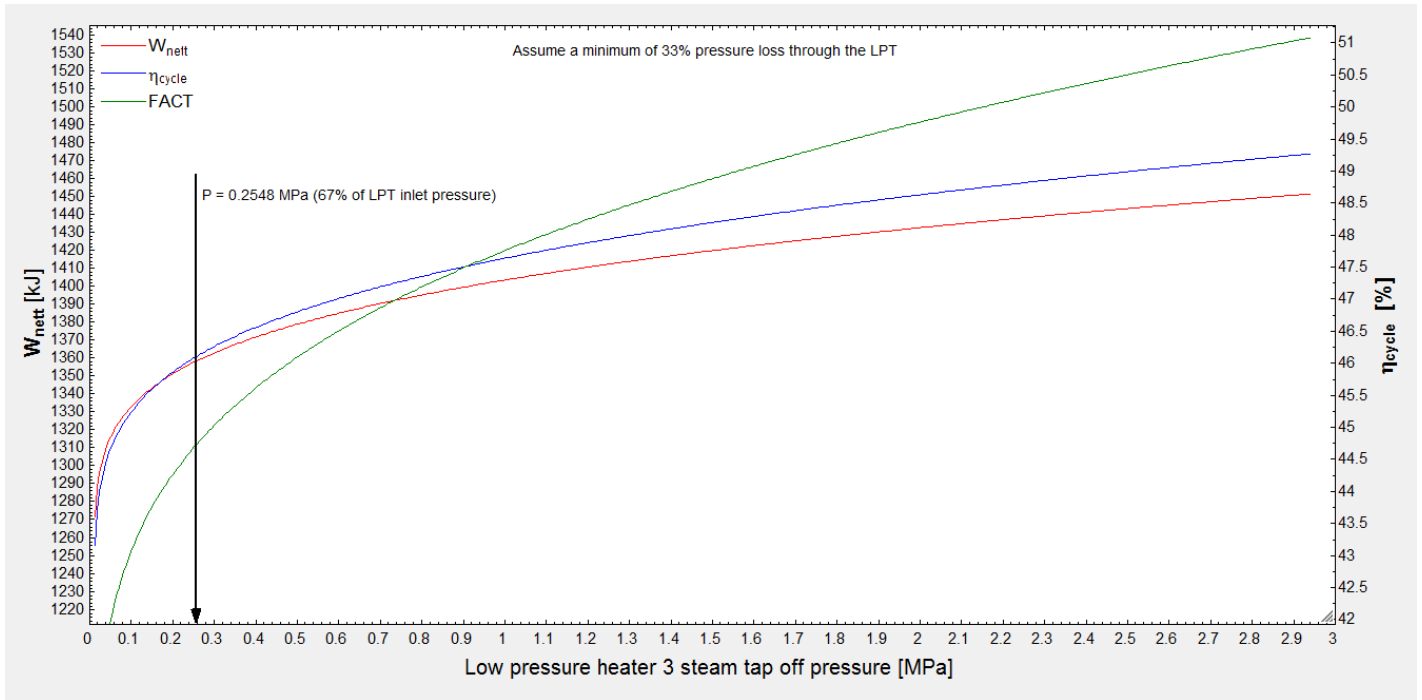


Figure 55: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.2548 MPa	Low pressure heater 3 steam tap off pressure	0.2548 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 24: Before and after optimisation results for low pressure heater 3 steam tap off pressure (1st method, 3rd run)

6.1.2.3.7. Low pressure heater 2

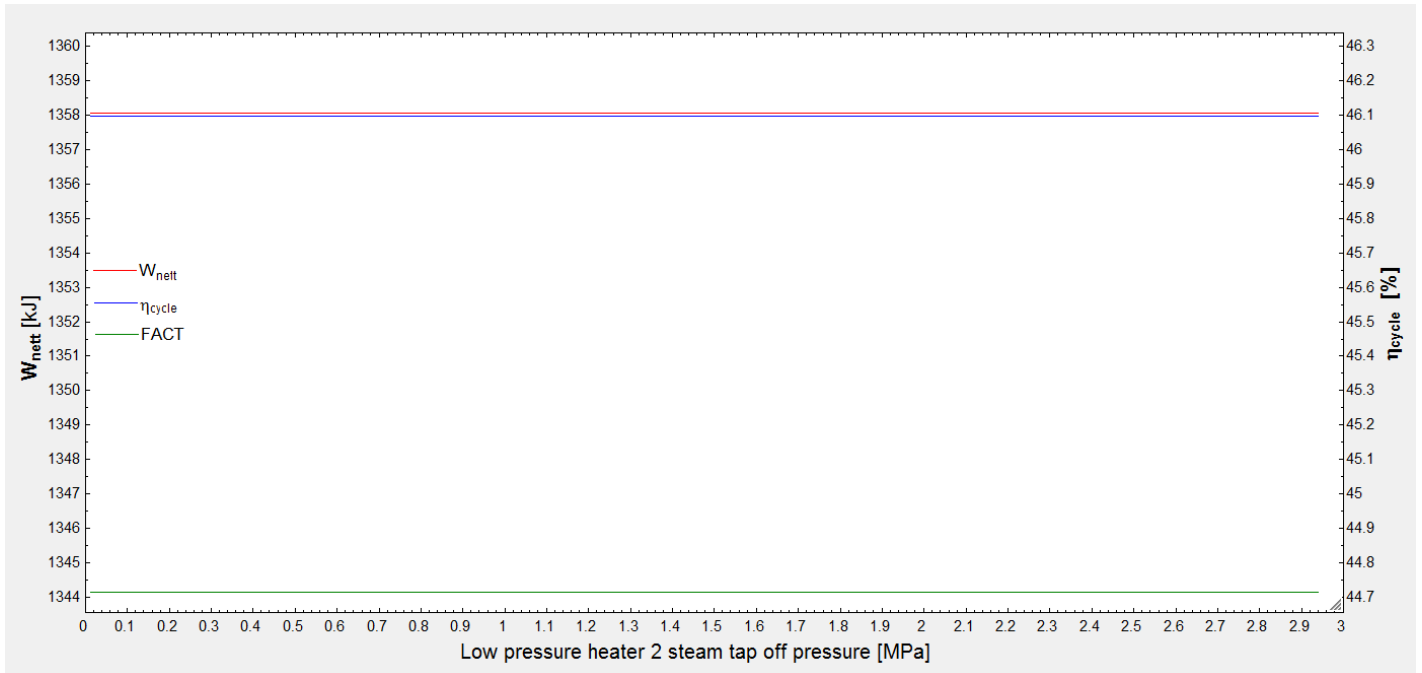


Figure 56: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.05613 MPa	Low pressure heater 2 steam tap off pressure	0.05613 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 25: Before and after optimisation results for low pressure heater 2 steam tap off pressure (1st method, 3rd run)

6.1.2.3.8. Low pressure heater 1

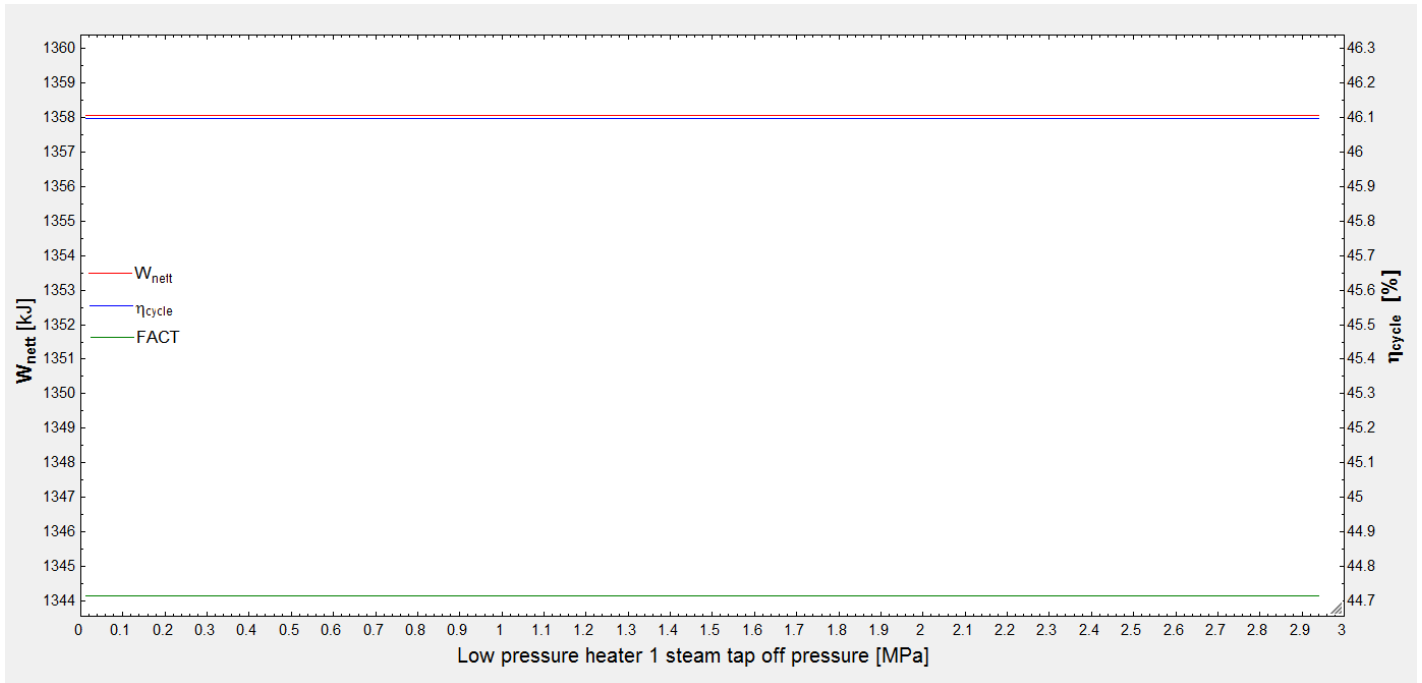


Figure 57: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.02777 MPa	Low pressure heater 1 steam tap off pressure	0.02777 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 26: Before and after optimisation results for low pressure heater 1 steam tap off pressure (1st method, 3rd run)

6.1.2.3.9. Boiler feed pump pressure

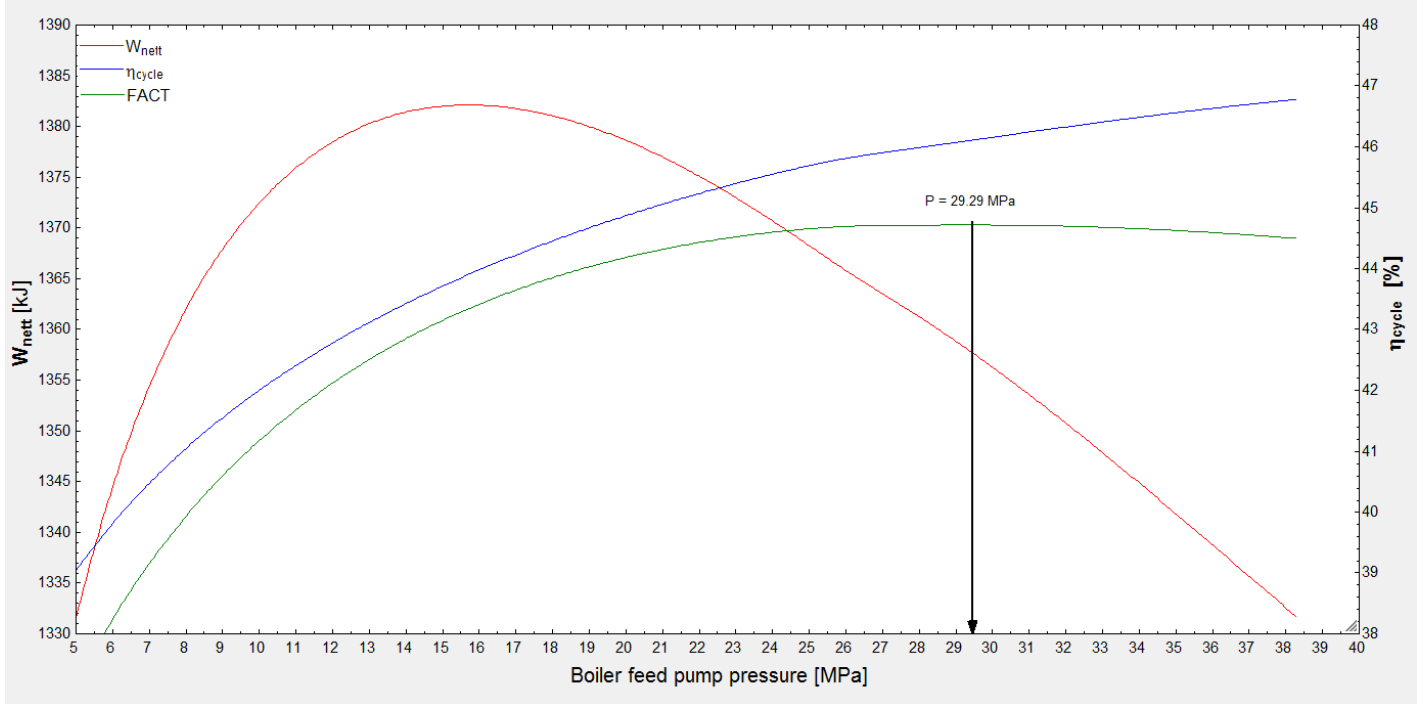


Figure 58: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (1st method, 4th run)

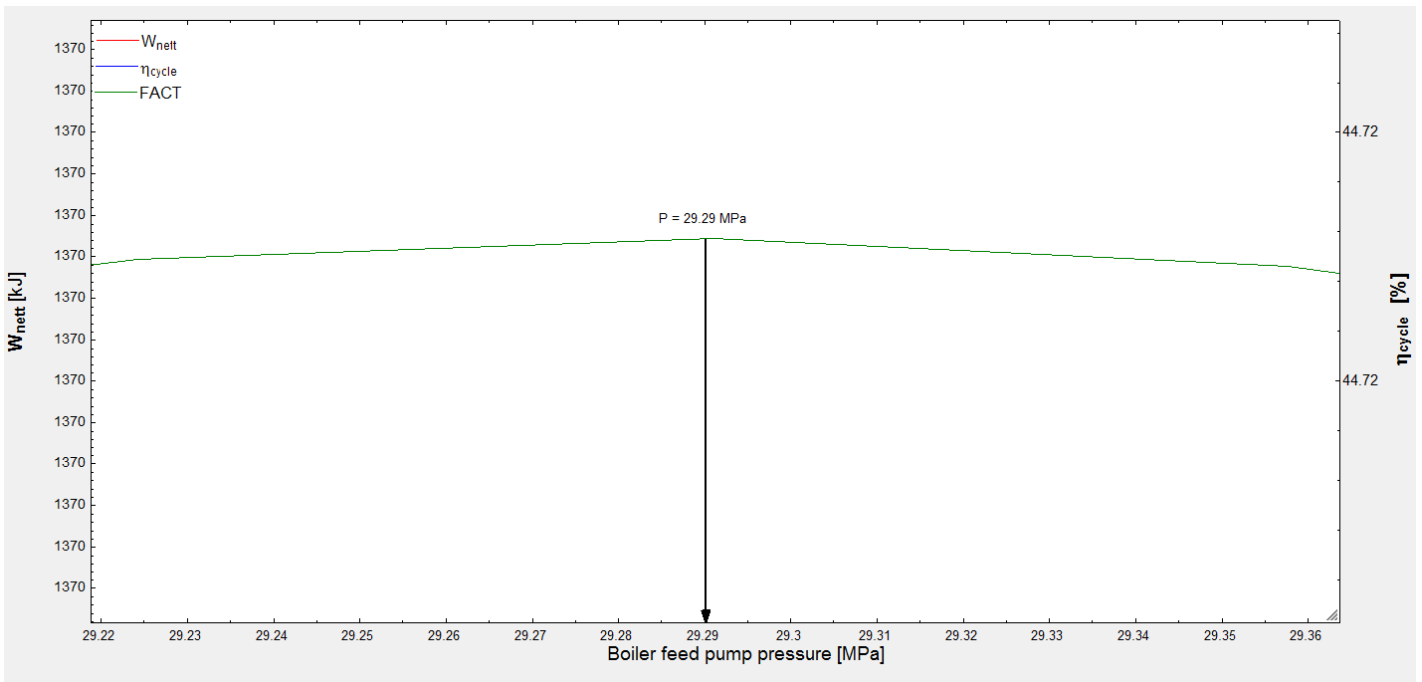


Figure 59: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (1st method, 4th run)

Before optimisation		After optimisation	
Boiler feed pump pressure	29.3 MPa	Boiler feed pump pressure	29.29 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 27: Before and after optimisation results for the boiler feed pump discharge pressure (1st method, 4th run)

6.1.2.3.10. High pressure turbine expansion pressure

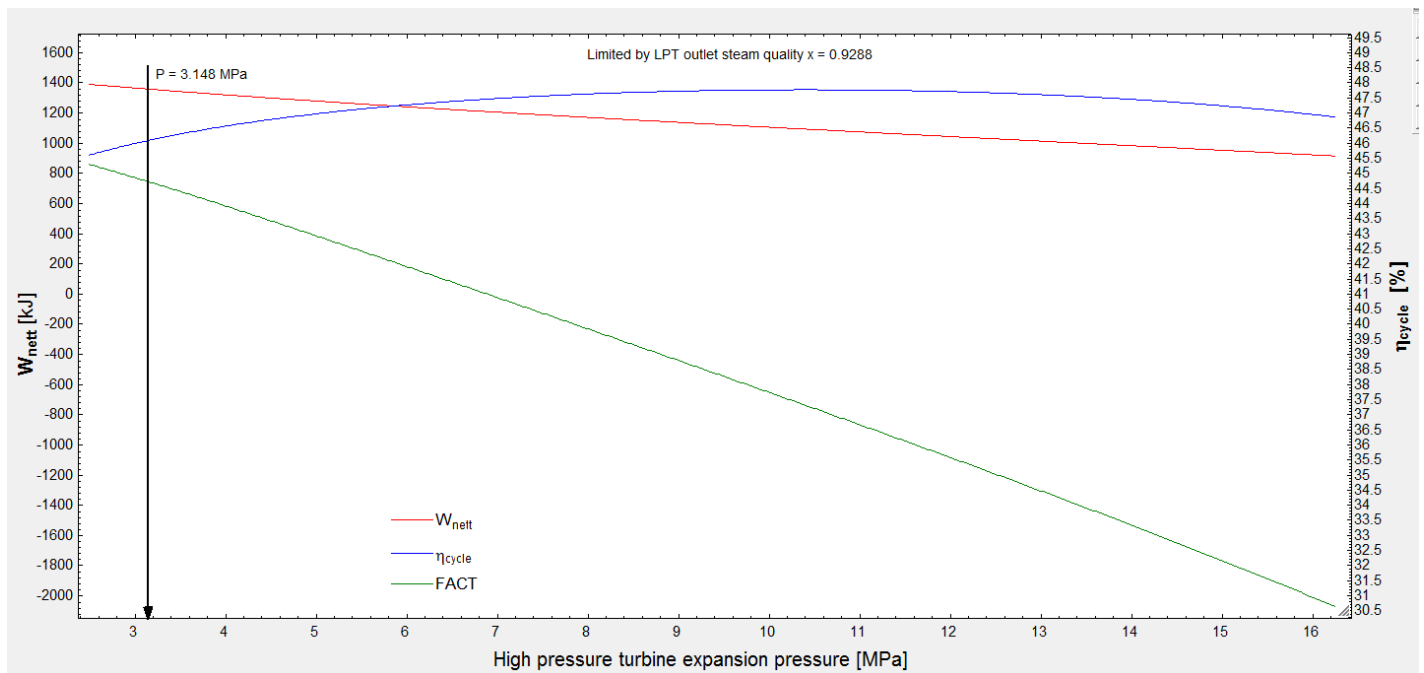


Figure 60: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (1st method, 4th run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.148 MPa	High pressure turbine expansion pressure	3.148 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.6 MW	Mega Watt output	563.6 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ
LPT outlet steam quality	0.9288	LPT outlet steam quality	0.9288

Table 28: Before and after optimisation results for the high pressure turbine expansion pressure (1st method, 4th run)

6.1.3. Sub-critical cycle optimisation – second method

6.1.3.1. First run

6.1.3.1.1. Boiler feed pump pressure

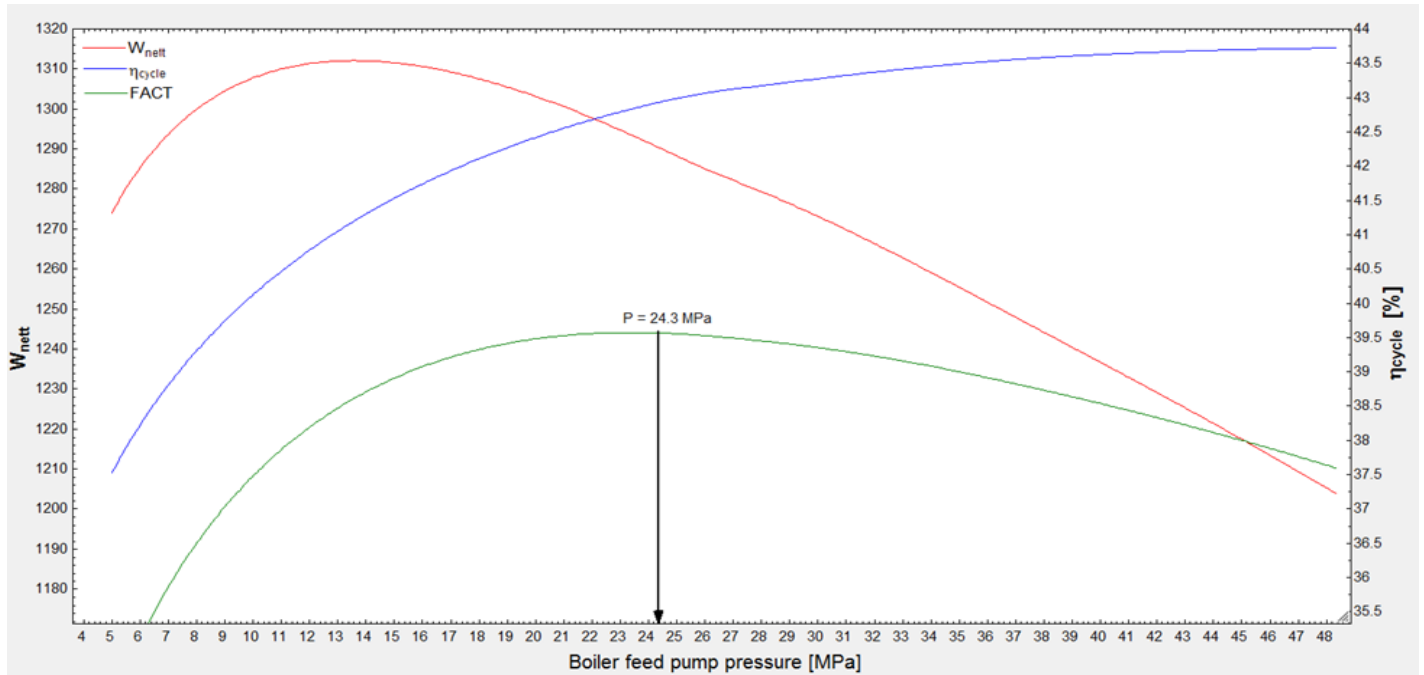


Figure 61: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (2nd method, 1st run)

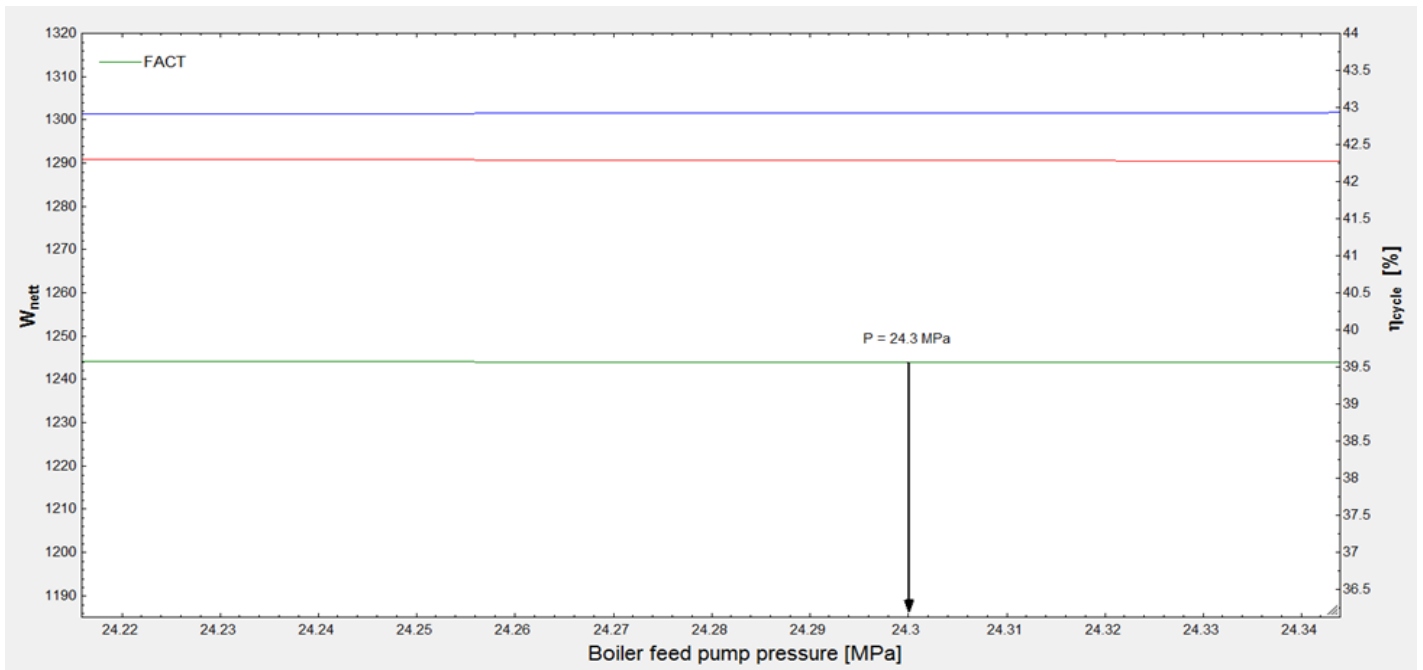


Figure 62: Cycle efficiency, net-work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (2nd method, 1st run)

Before optimisation	
Boiler feed pump pressure	22.14 MPa
Cycle efficiency	42.69%
Mega Watt output	538.4 MW
Net mechanical work	1397 kJ
Heat added	3039 kJ
Heat rejected	1553 kJ

After optimisation	
Boiler feed pump pressure	24.3 MPa
Cycle efficiency	42.92%
Mega Watt output	535.6 MW
Net mechanical work	1291 kJ
Heat added	3007 kJ
Heat rejected	1525 kJ

Table 29: Before and after optimisation results for the boiler feed pump discharge pressure (2nd method, 1st run)

6.1.3.1.2. High pressure heater 6

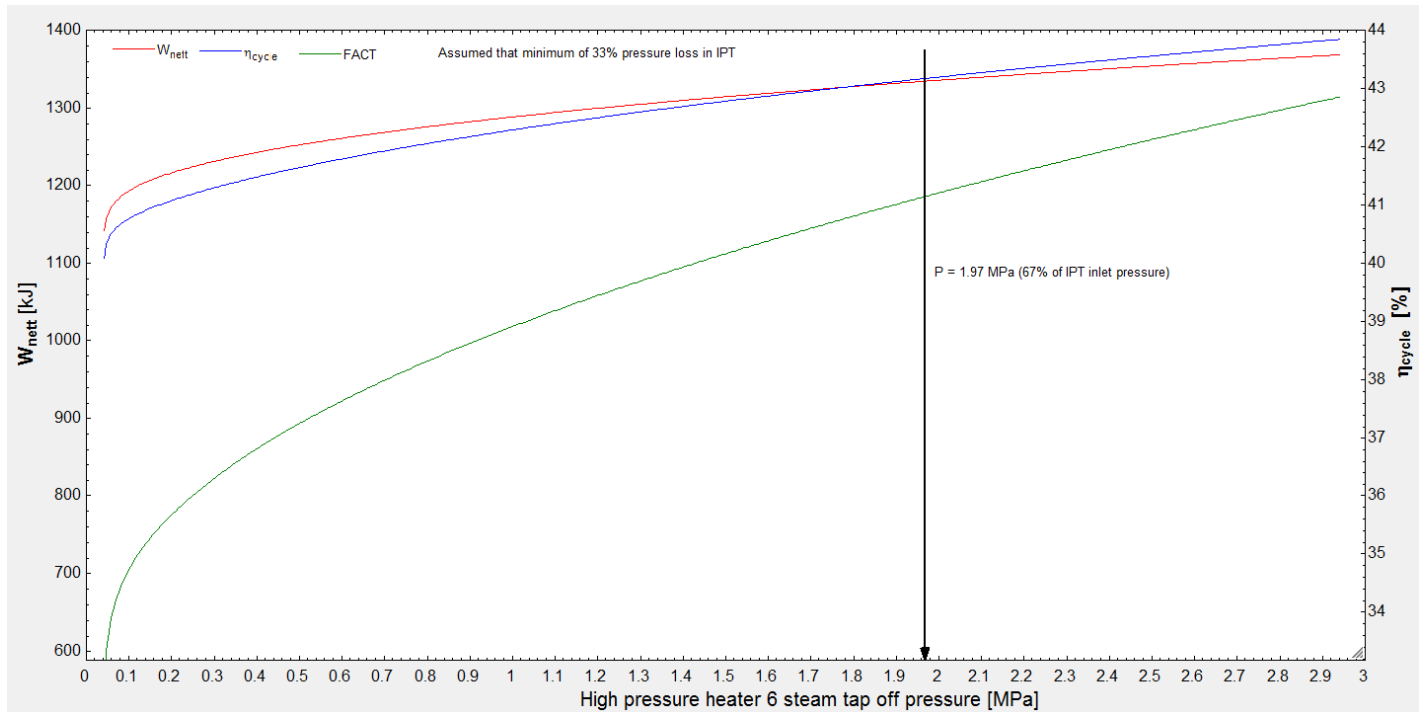


Figure 63: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (2nd method, 1st run)

Before optimisation	
High pressure heater 6 steam tap off pressure	1.9 MPa
Cycle efficiency	42.92%
Mega Watt output	535.6 MW
Net mechanical work	1291 kJ
Heat added	3007 kJ
Heat rejected	1525 kJ

After optimisation	
High pressure heater 6 steam tap off pressure	1.97 MPa
Cycle efficiency	43.17%
Mega Watt output	553.7 MW
Net mechanical work	1334 kJ
Heat added	3091 kJ
Heat rejected	1584 kJ

Table 30: Before and after optimisation results for high pressure heater 6 steam tap off pressure (2nd method, 1st run)

6.1.3.1.3. High pressure turbine expansion

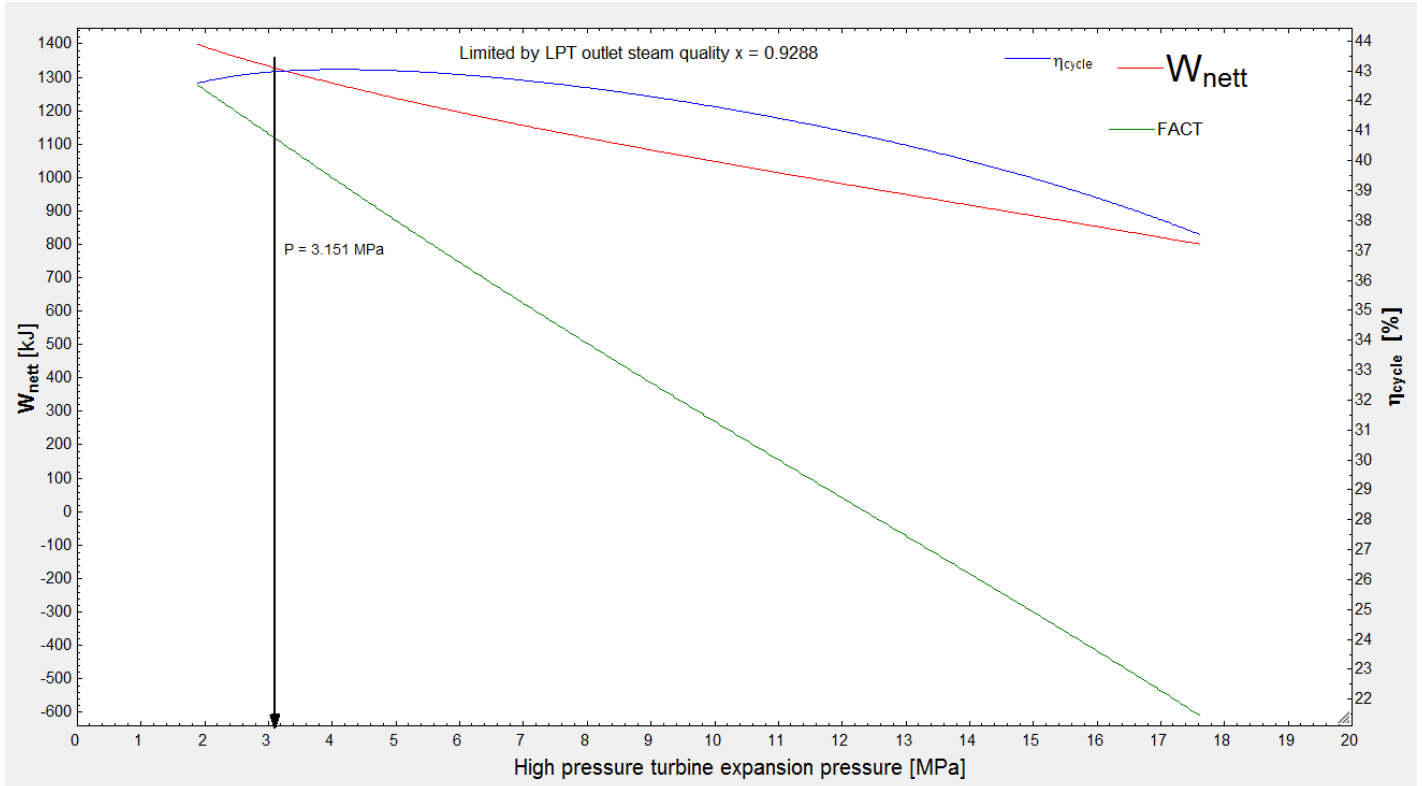


Figure 64: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (2nd method, 1st run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.715 MPa	High pressure turbine expansion pressure	3.151 MPa
Cycle efficiency	43.17%	Cycle efficiency	43.17%
Mega Watt output	553.7 MW	Mega Watt output	553.7 MW
Net mechanical work	1334 kJ	Net mechanical work	1334 kJ
Heat added	3091 kJ	Heat added	3091 kJ
Heat rejected	1584 kJ	Heat rejected	1584 kJ

Table 31: Before and after optimisation results for the high pressure turbine expansion pressure (2nd method, 1st run)

6.1.3.1.4. Low pressure heater 1

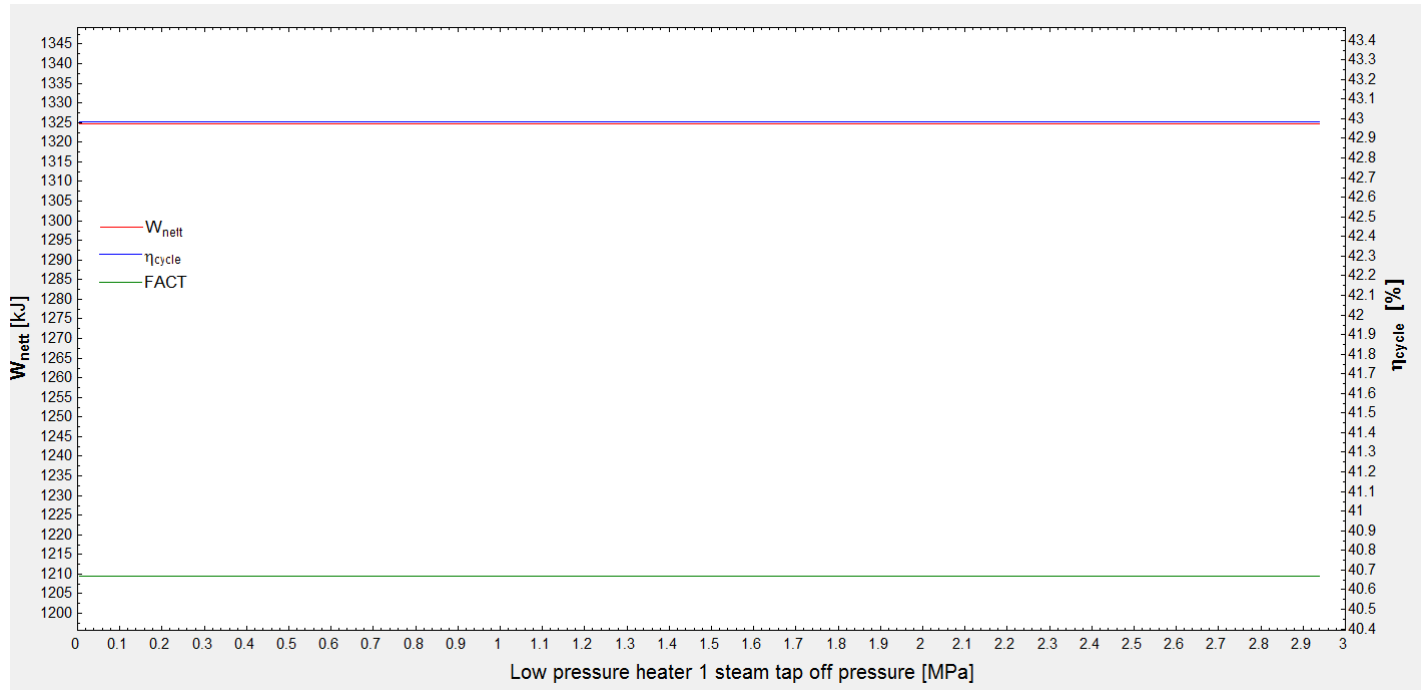


Figure 65: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (2nd method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.02779 MPa	Low pressure heater 1 steam tap off pressure	0.02779 MPa
Cycle efficiency	45.66 %	Cycle efficiency	45.66 %
Mega Watt output	557.8 MW	Mega Watt output	557.8 MW
Net mechanical work	1344 kJ	Net mechanical work	1344 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1606 kJ	Heat rejected	1606 kJ

Table 32: Before and after optimisation results for low pressure heater 1 steam tap off pressure (2nd method, 1st run)

6.1.3.1.5. Low pressure heater 2

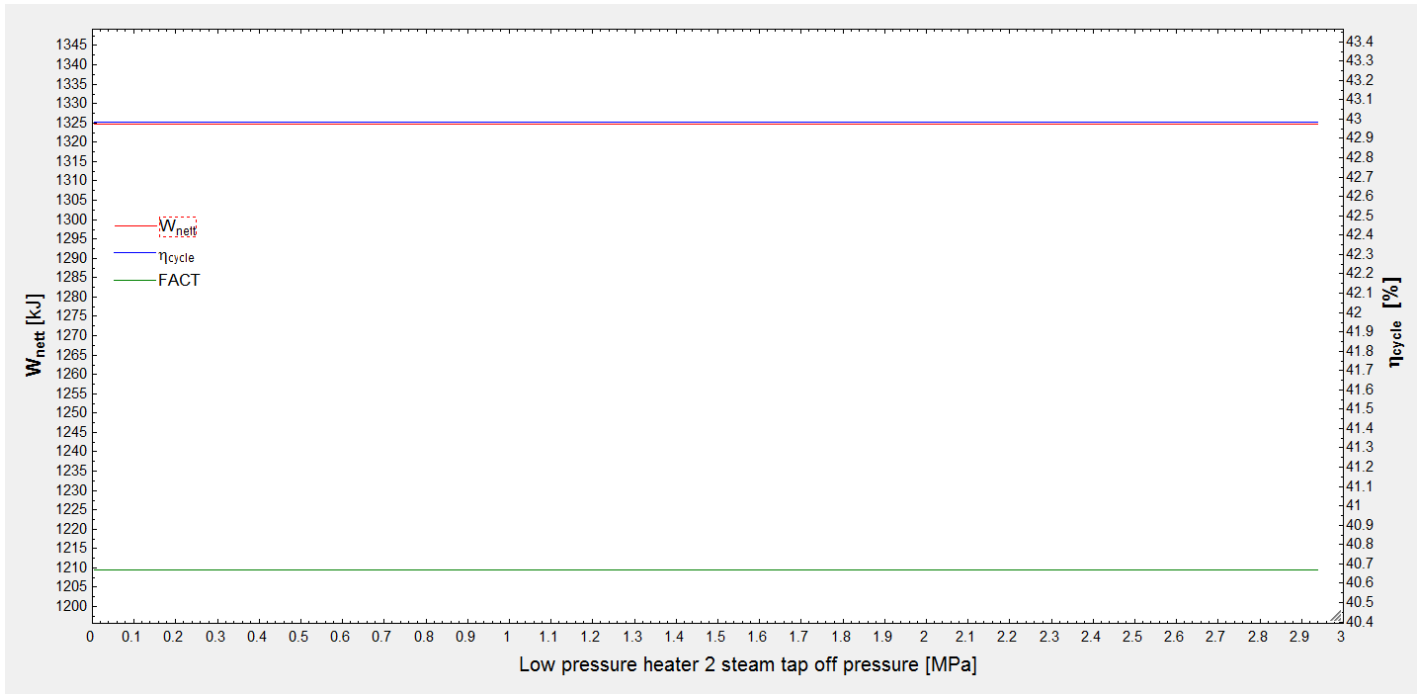


Figure 66: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (2nd method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.05617 MPa	Low pressure heater 2 steam tap off pressure	0.05617 MPa
Cycle efficiency	45.66 %	Cycle efficiency	45.66 %
Mega Watt output	557.8 MW	Mega Watt output	557.8 MW
Net mechanical work	1344 kJ	Net mechanical work	1344 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1606 kJ	Heat rejected	1606 kJ

Table 33: Before and after optimisation results for low pressure heater 2 steam tap off pressure (2nd method, 1st run)

6.1.3.1.6. Low pressure heater 3

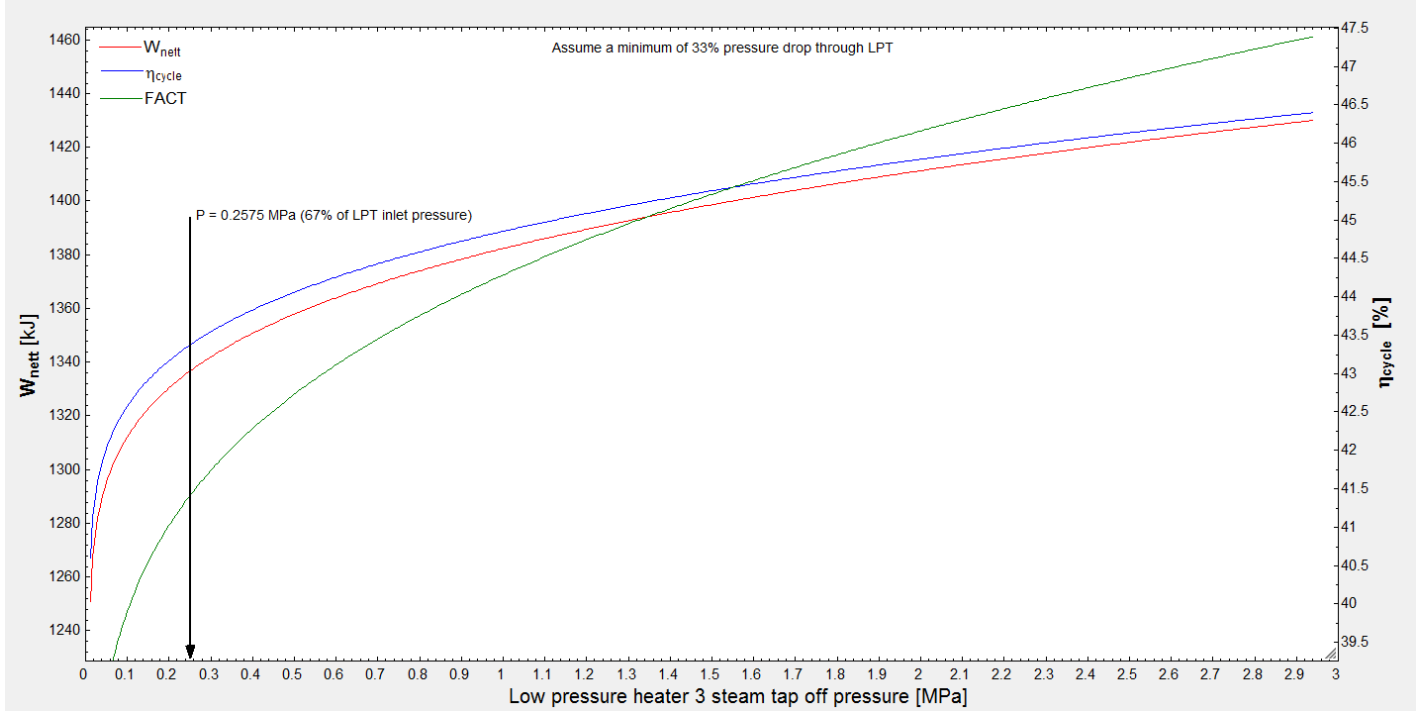


Figure 67: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (2nd method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.2542 MPa	Low pressure heater 3 steam tap off pressure	0.2575 MPa
Cycle efficiency	45.66 %	Cycle efficiency	45.66 %
Mega Watt output	557.8 MW	Mega Watt output	557.8 MW
Net mechanical work	1344 kJ	Net mechanical work	1344 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1606 kJ	Heat rejected	1606 kJ

Table 34: Before and after optimisation results for low pressure heater 3 steam tap off pressure (2nd method, 1st run)

6.1.3.1.7. Low pressure heater 4

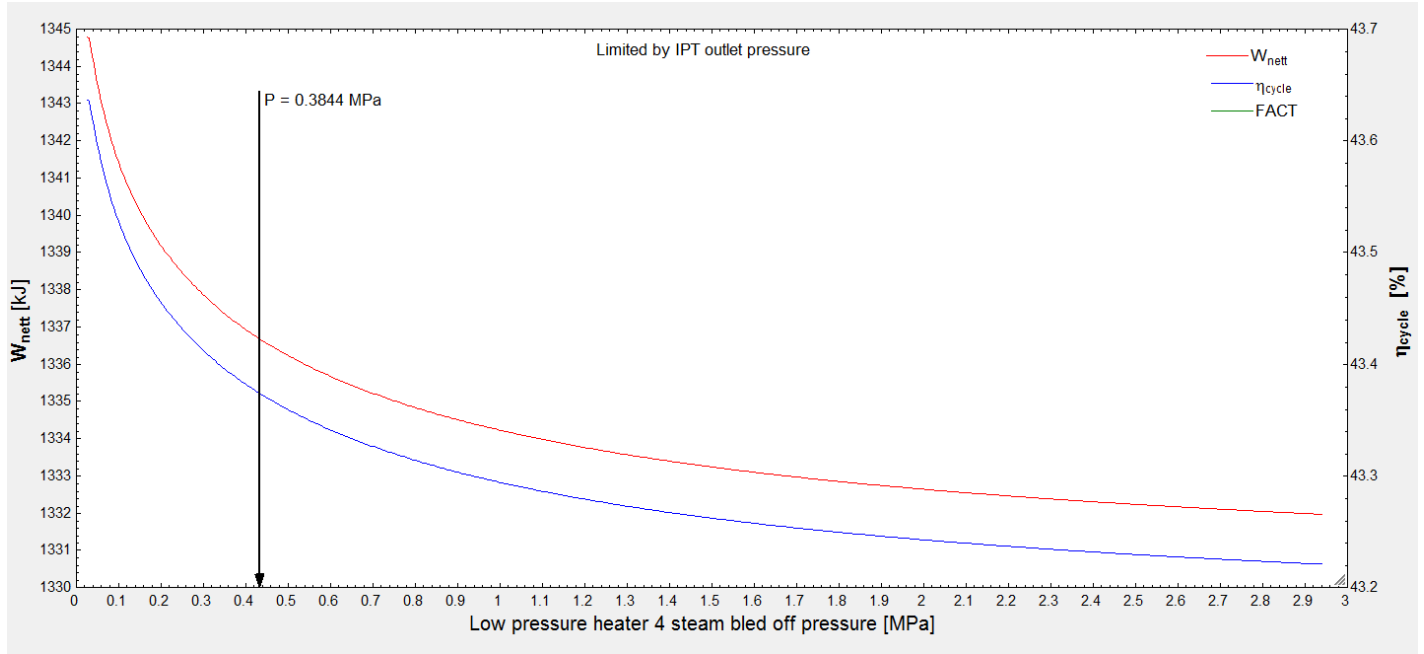


Figure 68: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 4 steam bled off pressure (2nd method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 4 steam tap off pressure	1.971 MPa	Low pressure heater 4 steam tap off pressure	0.3794 MPa
Cycle efficiency	45.66 %	Cycle efficiency	45.97 %
Mega Watt output	557.8 MW	Mega Watt output	561.5 MW
Net mechanical work	1344 kJ	Net mechanical work	1353 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1606 kJ	Heat rejected	1591 kJ

Table 35: Before and after optimisation results for low pressure heater 4 steam tap off pressure (2nd method, 1st run)

6.1.3.1.8. De-aerator

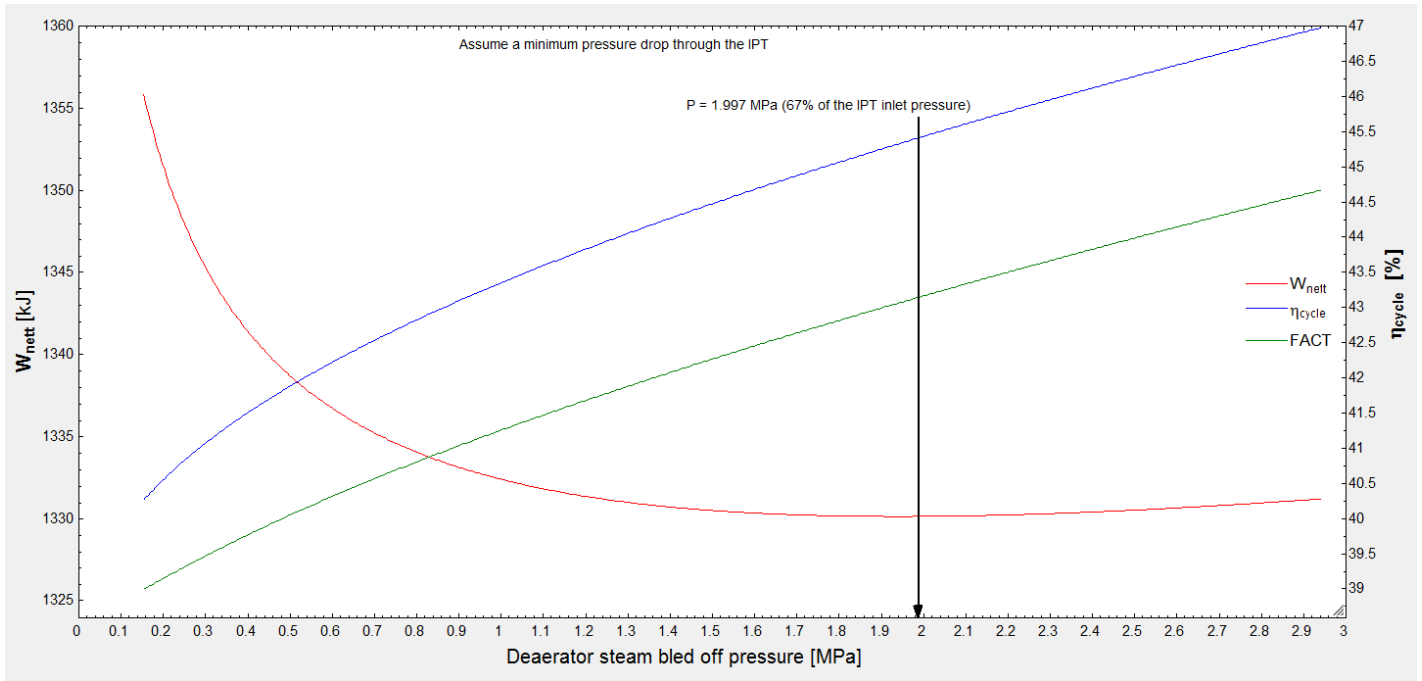


Figure 69: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (2nd method, 1st run)

Before optimisation		After optimisation	
De-aerator steam tap off pressure	1.971 MPa	De-aerator steam tap off pressure	1.97 MPa
Cycle efficiency	45.97 %	Cycle efficiency	45.96 %
Mega Watt output	561.5 MW	Mega Watt output	561.5 MW
Net mechanical work	1353 kJ	Net mechanical work	1353 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1591 kJ	Heat rejected	1591 kJ

Table 36: Before and after optimisation results for the de-aerator steam tap off pressure (2nd method, 1st run)

6.1.3.2. Second run

6.1.3.2.1. Boiler feed pump pressure

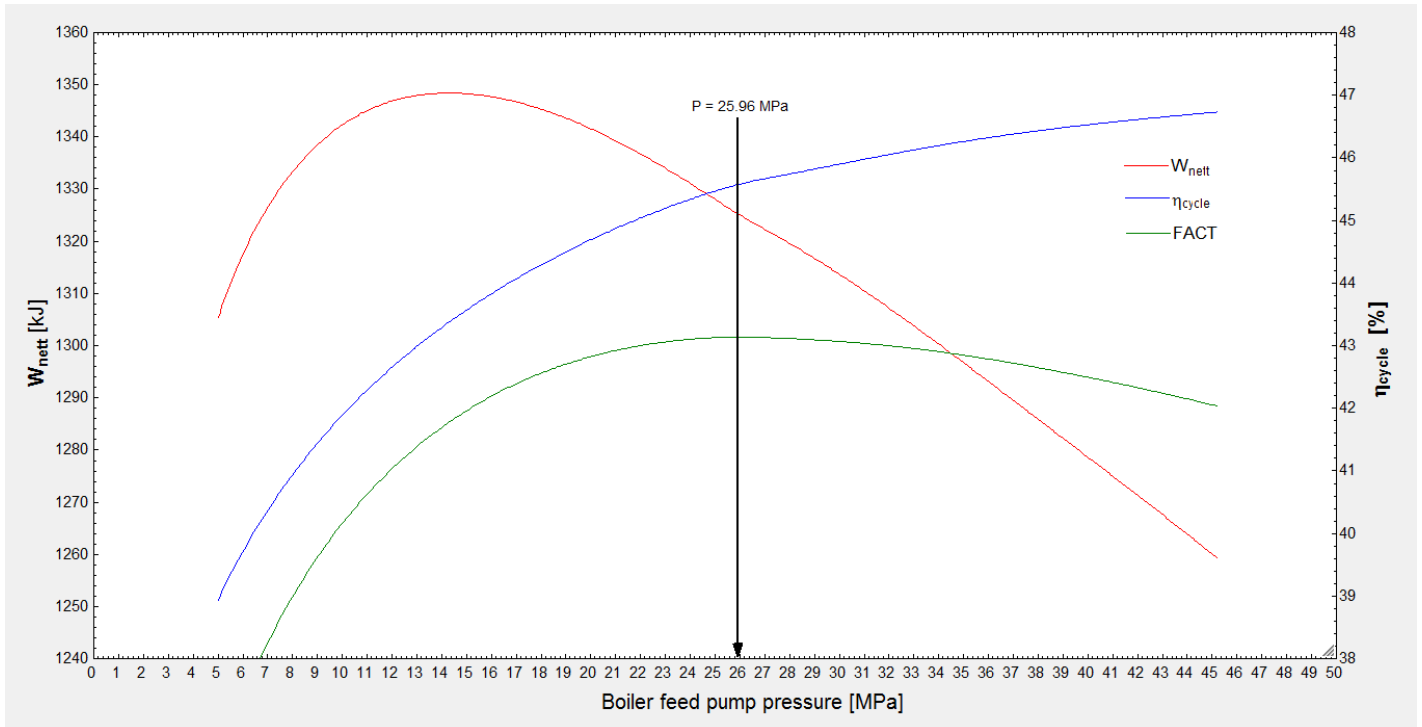


Figure 70: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (2nd method, 2nd run)

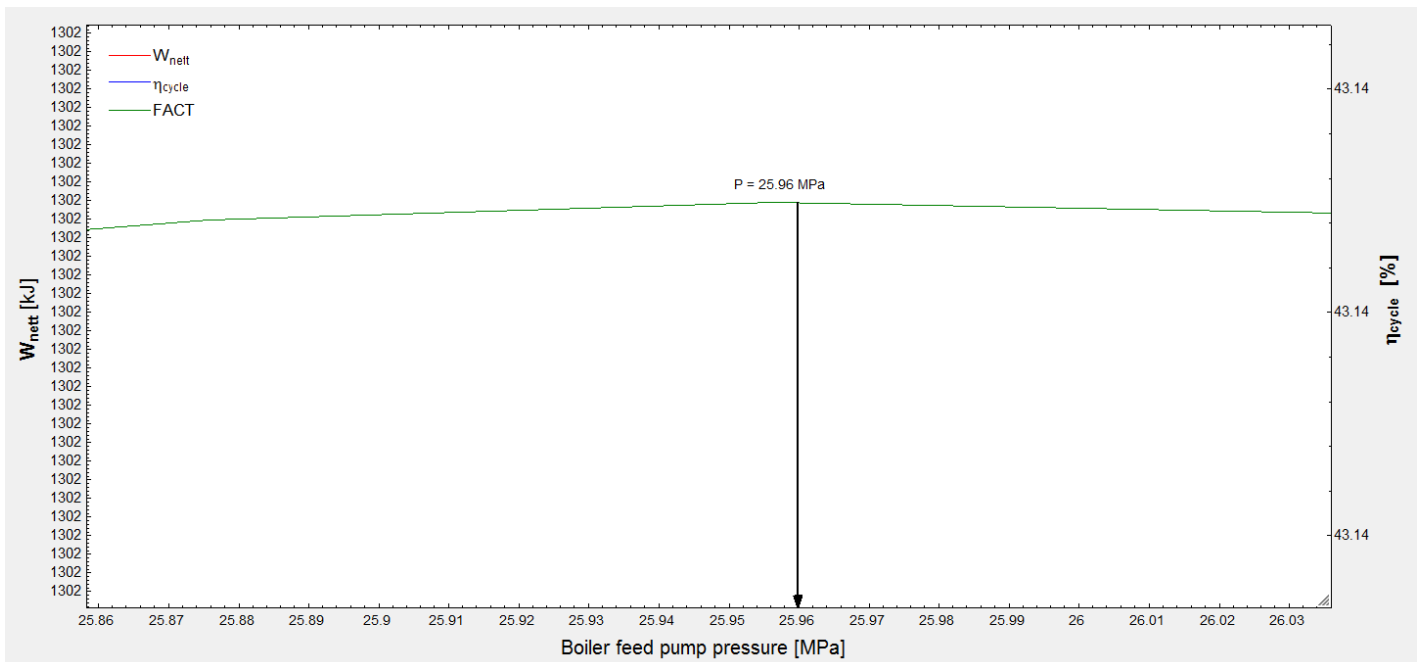


Figure 71: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (Zoomed in) (2nd method, 2nd run)

Before optimisation		After optimisation	
Boiler feed pump pressure	24.3 MPa	Boiler feed pump pressure	25.96 MPa
Cycle efficiency	45.96 %	Cycle efficiency	45.58%
Mega Watt output	561.5 MW	Mega Watt output	549.9 MW
Net mechanical work	1353 kJ	Net mechanical work	1325 kJ
Heat added	2943 kJ	Heat added	2907 kJ
Heat rejected	1591 kJ	Heat rejected	1585 kJ

Table 37: Before and after optimisation results for the boiler feed pump discharge (2nd method, 2nd run)

6.1.3.2.2. High pressure heater 6

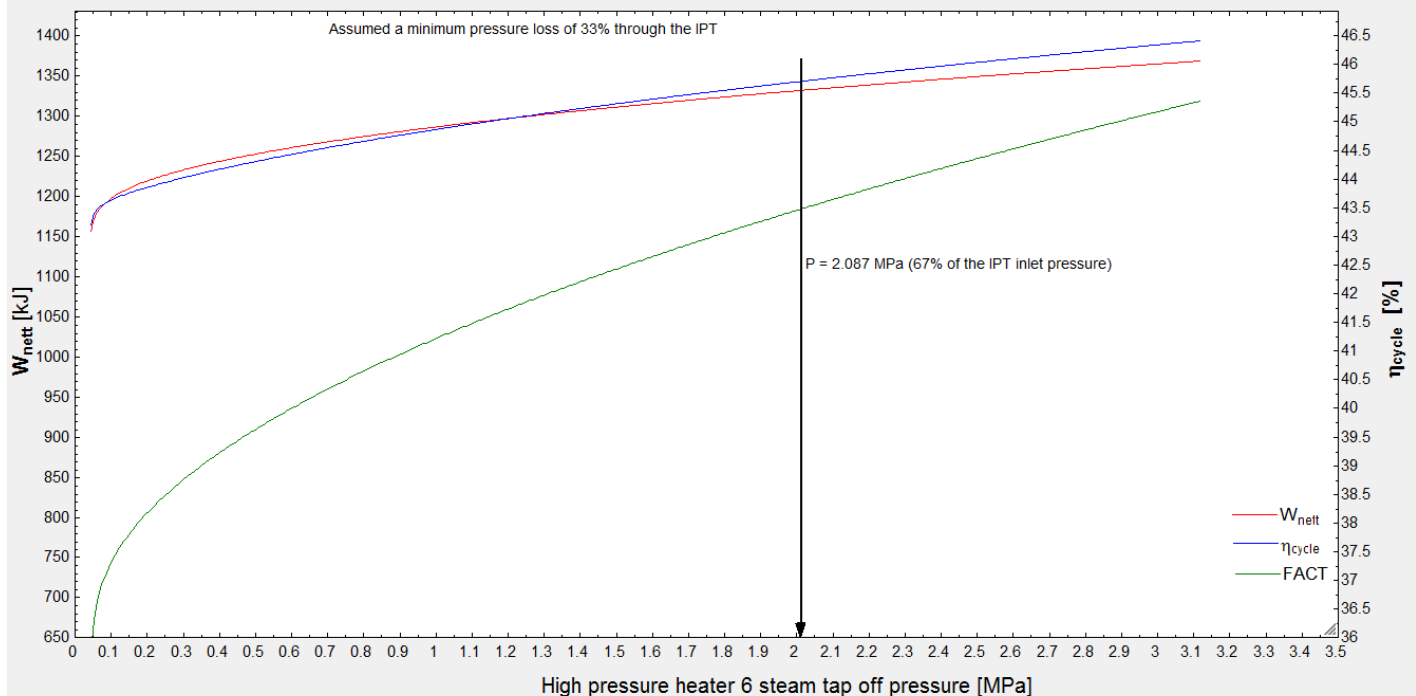


Figure 72: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
High pressure heater 6 steam bled off pressure	1.838 MPa	High pressure heater 6 steam bled off pressure	2.087 MPa
Cycle efficiency	45.58%	Cycle efficiency	45.76%
Mega Watt output	549.9 MW	Mega Watt output	553.8 MW
Net mechanical work	1325 kJ	Net mechanical work	1335 kJ
Heat added	2907 kJ	Heat added	2917 kJ
Heat rejected	1585 kJ	Heat rejected	1589 kJ

Table 38: Before and after optimisation results for high pressure heater 6 steam tap off pressure (2nd method, 2nd run)

6.1.3.2.3. High pressure turbine expansion

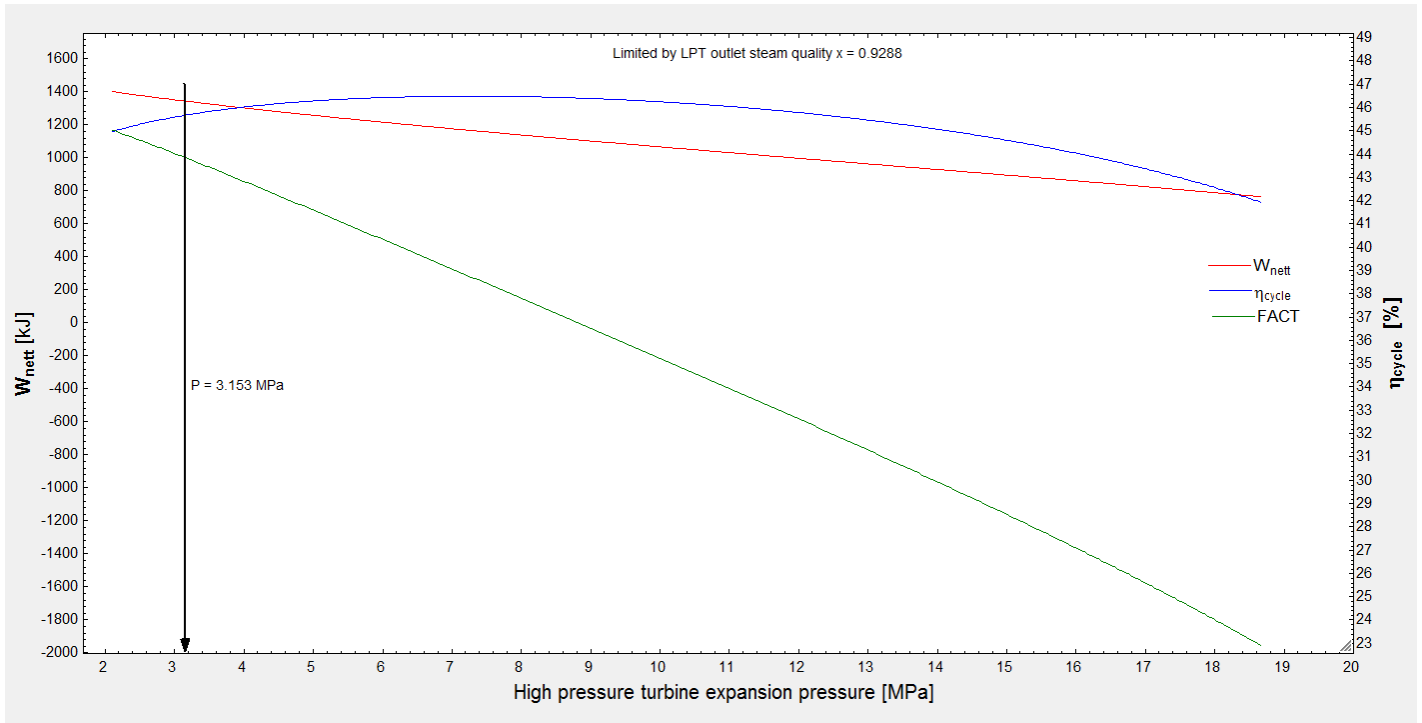


Figure 73: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.341 MPa	High pressure turbine expansion pressure	3.153 MPa
Cycle efficiency	45.58%	Cycle efficiency	45.66%
Mega Watt output	549.9 MW	Mega Watt output	557.8 MW
Net mechanical work	1325 kJ	Net mechanical work	1344 kJ
Heat added	2907 kJ	Heat added	2943 kJ
Heat rejected	1585 kJ	Heat rejected	1606 kJ
LPT outlet steam quality	X = 0.9255	LPT outlet steam quality	X = 0.9288

Table 39: Before and after optimisation results for the high pressure turbine expansion pressure (2nd method, 2nd run)

6.1.3.2.4. Low pressure heater 1

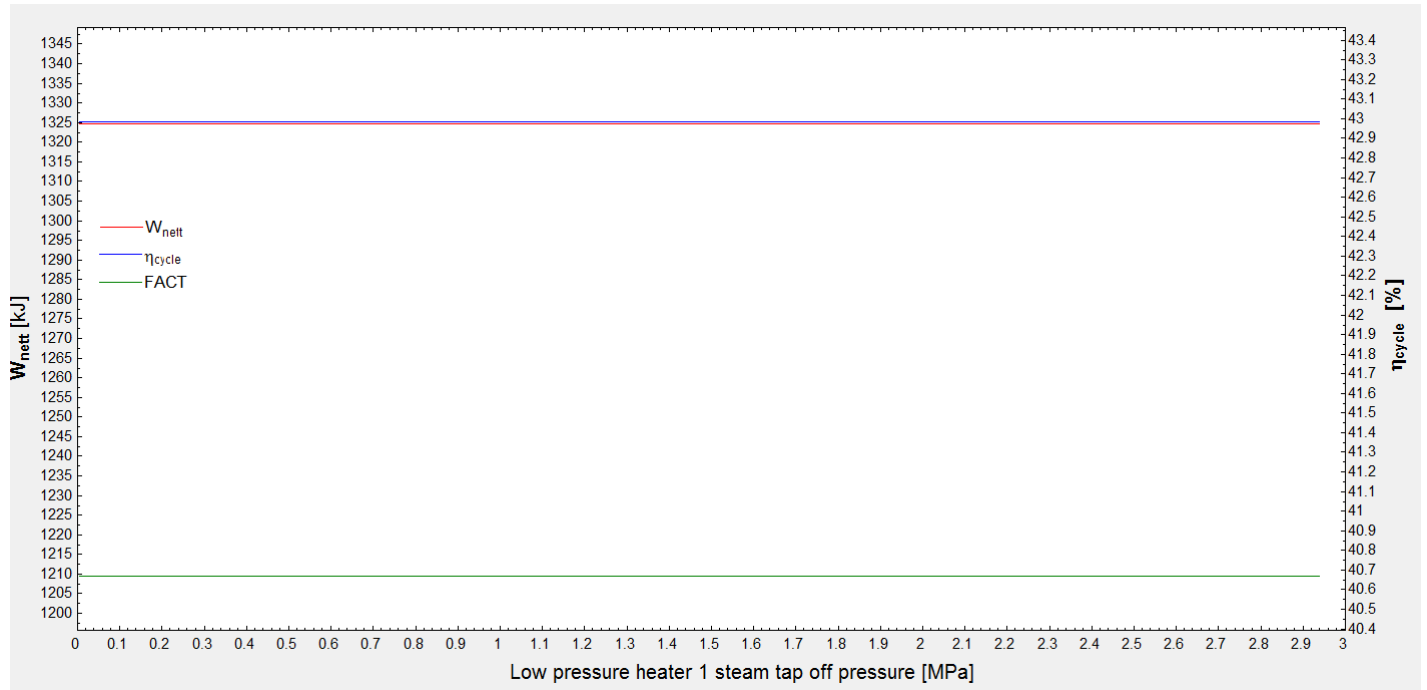


Figure 74: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 1 steam bled off pressure	0.02779 MPa	Low pressure heater 1 steam bled off pressure	0.02779 MPa
Cycle efficiency	45.66%	Cycle efficiency	45.66%
Mega Watt output	557.8 MW	Mega Watt output	557.8 MW
Net mechanical work	1344 kJ	Net mechanical work	1344 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1606 kJ	Heat rejected	1606 kJ

Table 40: Before and after optimisation results for low pressure heater 1 steam tap off pressure (2nd method, 2nd run)

6.1.3.2.5. Low pressure heater 2

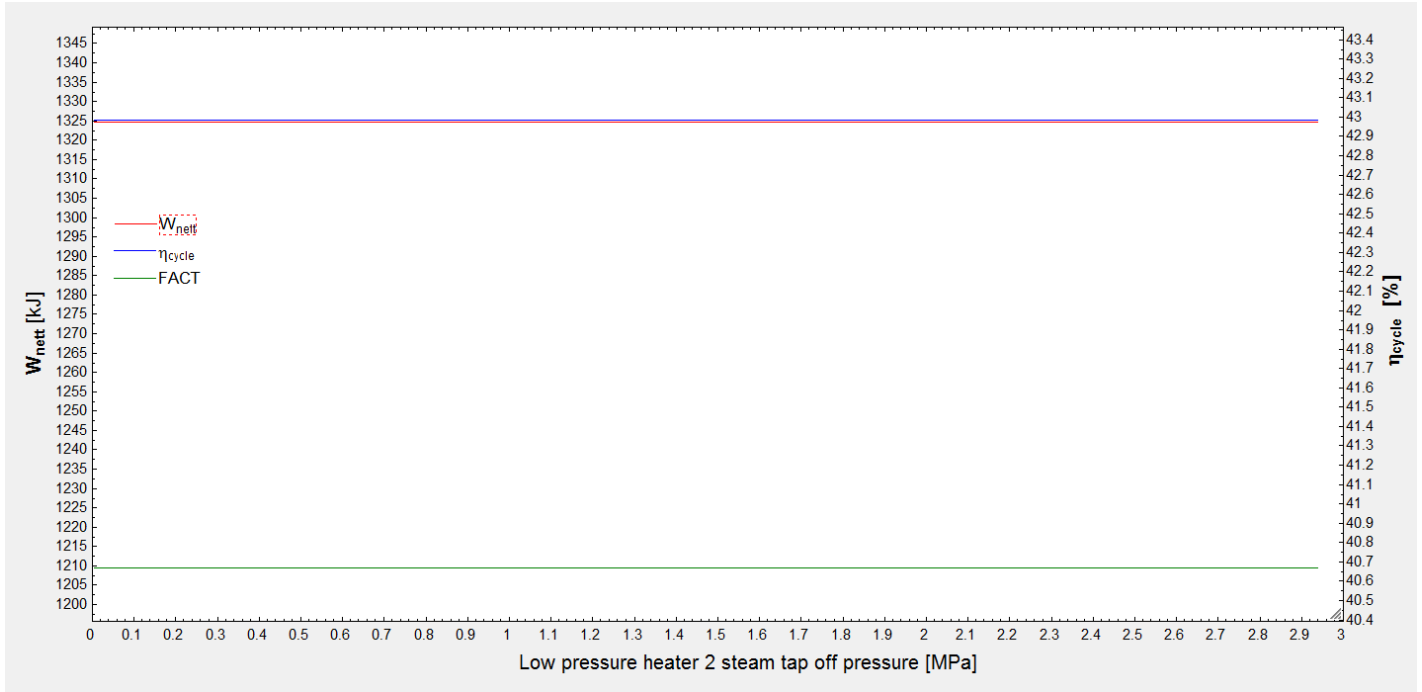


Figure 75: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 2 steam bled off pressure	0.05617 MPa	Low pressure heater 2 steam bled off pressure	0.05617 MPa
Cycle efficiency	45.66%	Cycle efficiency	45.66%
Mega Watt output	557.8 MW	Mega Watt output	557.8 MW
Net mechanical work	1344 kJ	Net mechanical work	1344 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1606 kJ	Heat rejected	1606 kJ

Table 41: Before and after optimisation results for low pressure heater 2 steam tap off pressure (2nd method, 2nd run)

6.1.3.2.6. Low pressure heater 3

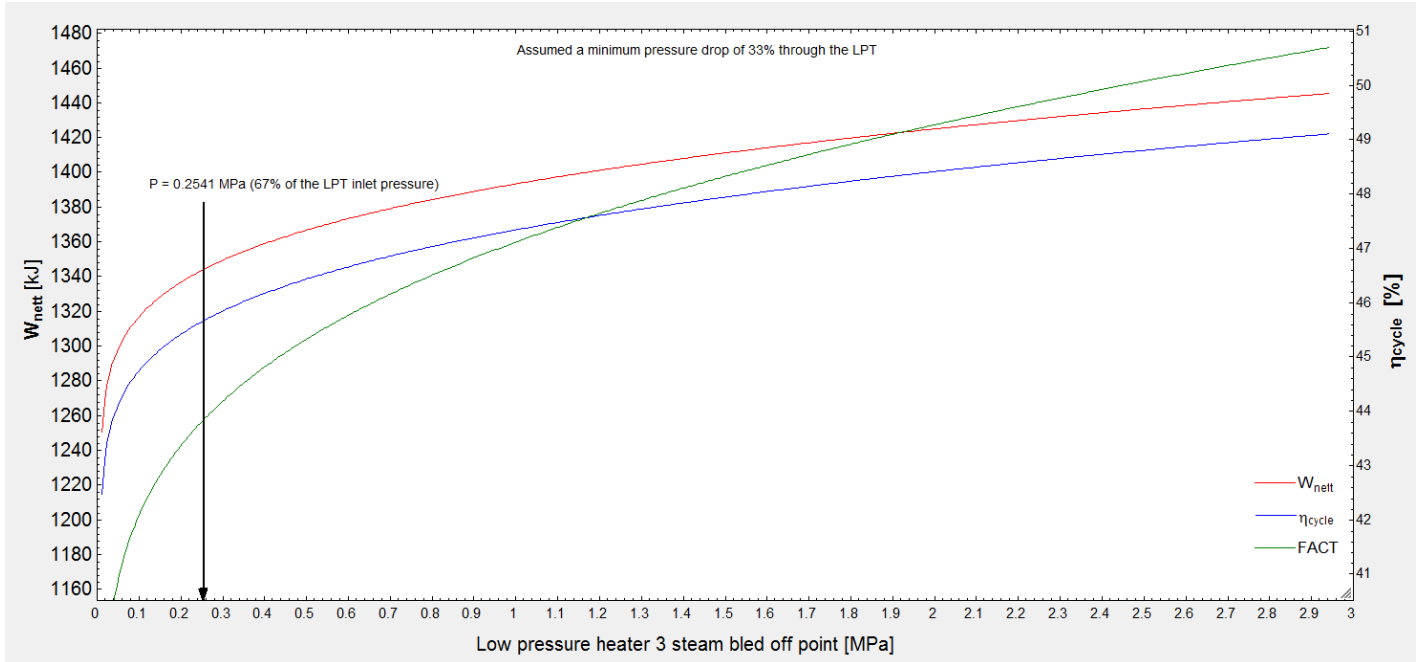


Figure 76: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 3 steam bled off pressure	0.2542 MPa	Low pressure heater 3 steam bled off pressure	0.2541 MPa
Cycle efficiency	45.66%	Cycle efficiency	45.66%
Mega Watt output	557.8 MW	Mega Watt output	557.8 MW
Net mechanical work	1344 kJ	Net mechanical work	1344 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1606 kJ	Heat rejected	1606 kJ

Table 42: Before and after optimisation results for low pressure heater 3 steam tap off pressure (2nd method, 2nd run)

6.1.3.2.7. Low pressure heater 4

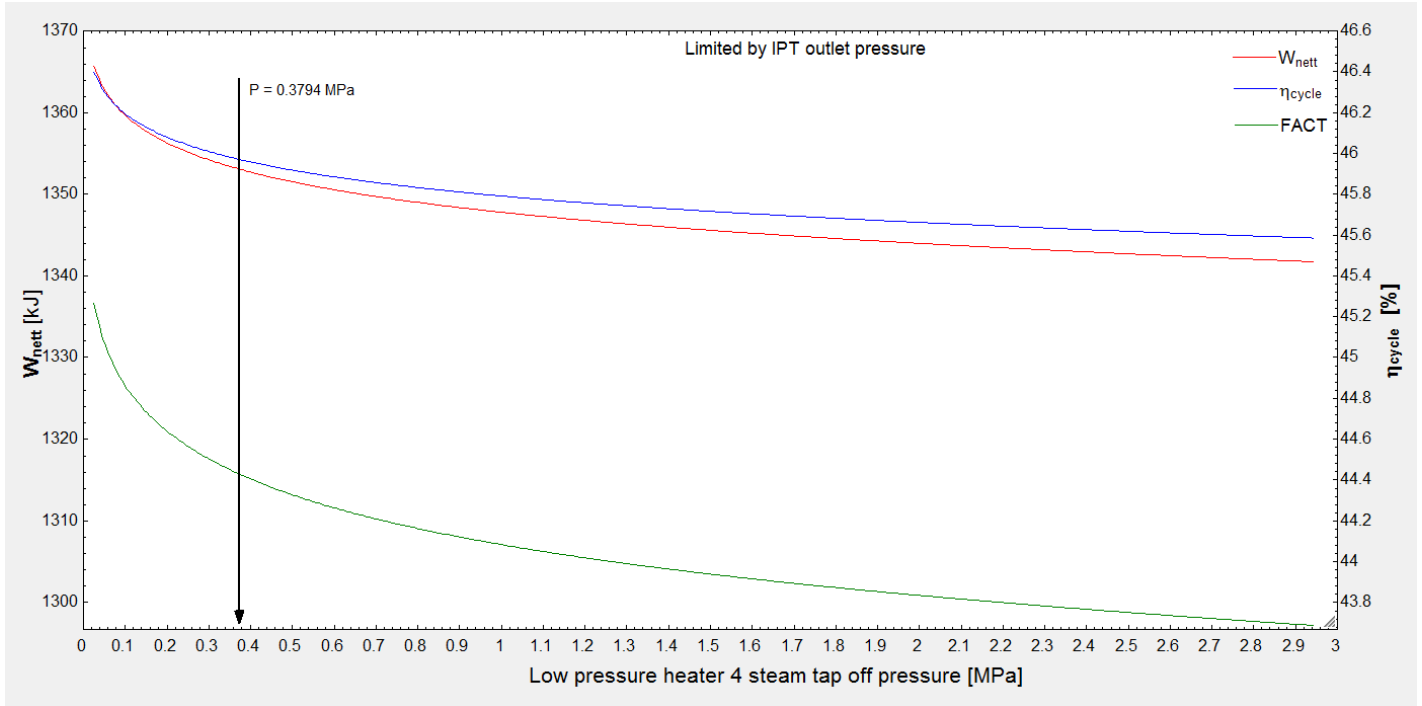


Figure 77: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 4 steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 4 steam bled off pressure	1.971 MPa	Low pressure heater 4 steam bled off pressure	0.3794 MPa
Cycle efficiency	45.66%	Cycle efficiency	45.97%
Mega Watt output	557.8 MW	Mega Watt output	561.5 MW
Net mechanical work	1344 kJ	Net mechanical work	1353 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1606 kJ	Heat rejected	1591 kJ

Table 43: Before and after optimisation results for low pressure heater 4 steam tap off pressure (2nd method, 2nd run)

6.1.3.2.8. De-aerator

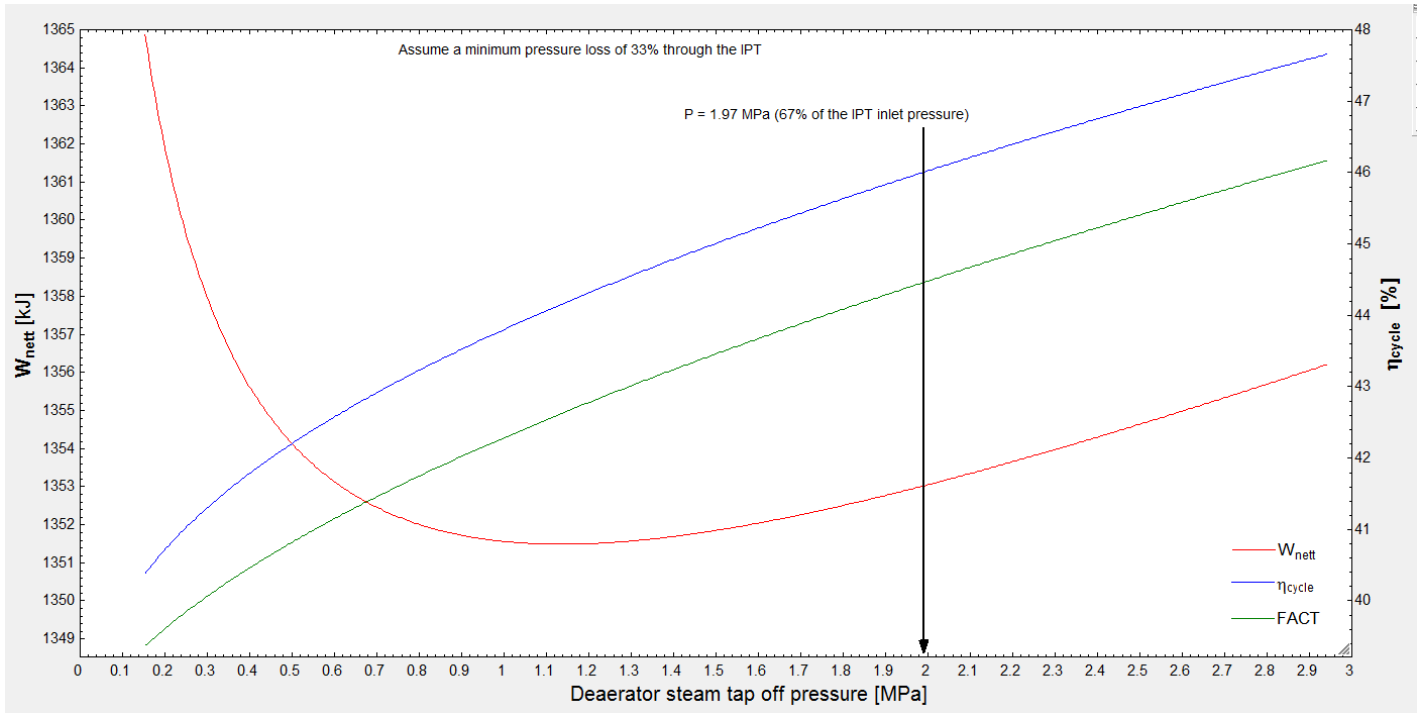


Figure 78: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
De-aerator steam bled off pressure	1.971 MPa	De-aerator steam bled off pressure	1.97 MPa
Cycle efficiency	45.97%	Cycle efficiency	45.96%
Mega Watt output	561.5 MW	Mega Watt output	561.5 MW
Net mechanical work	1353 kJ	Net mechanical work	1353 kJ
Heat added	2943 kJ	Heat added	2943 kJ
Heat rejected	1591 kJ	Heat rejected	1591 kJ

Table 44: Before and after optimisation results for the de-aerator steam tap off pressure (2nd method, 2nd run)

6.1.3.3. Third run

6.1.3.3.1. Boiler feed pump pressure

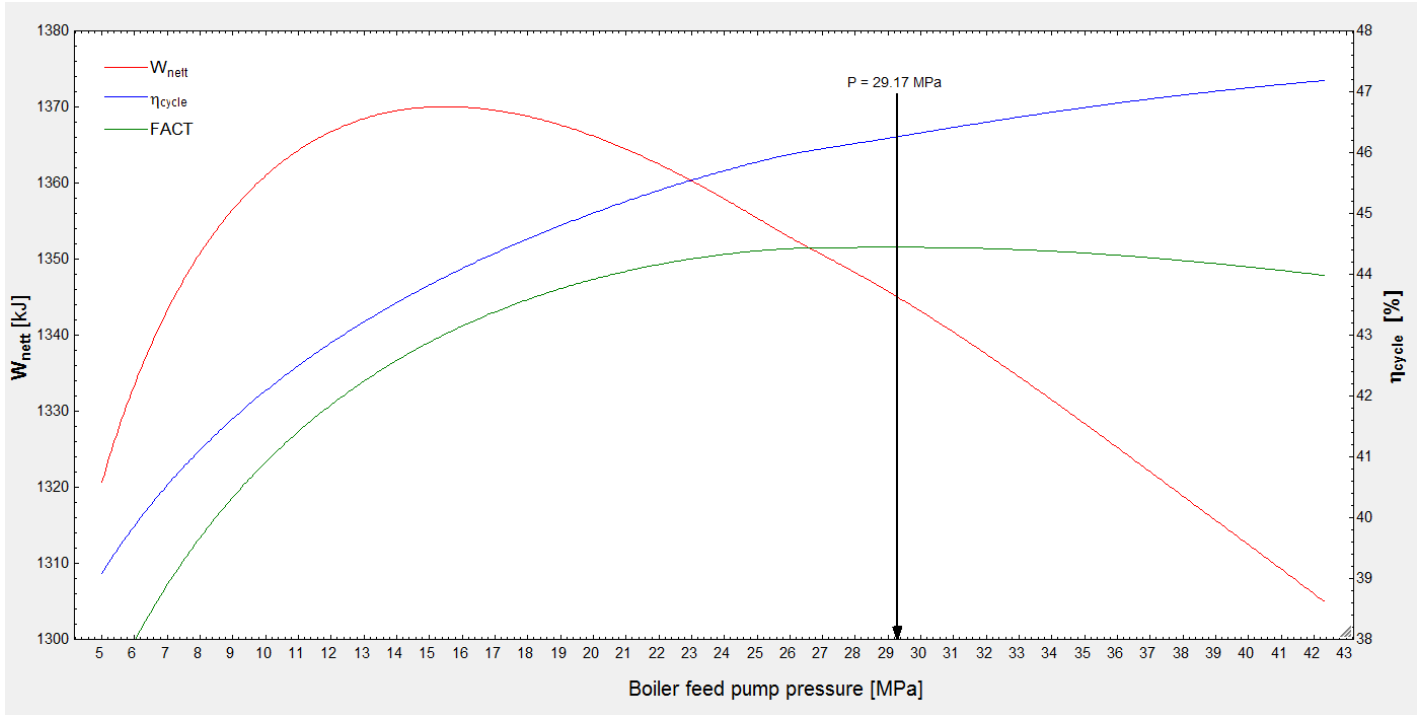


Figure 79: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (2nd method, 3rd run)

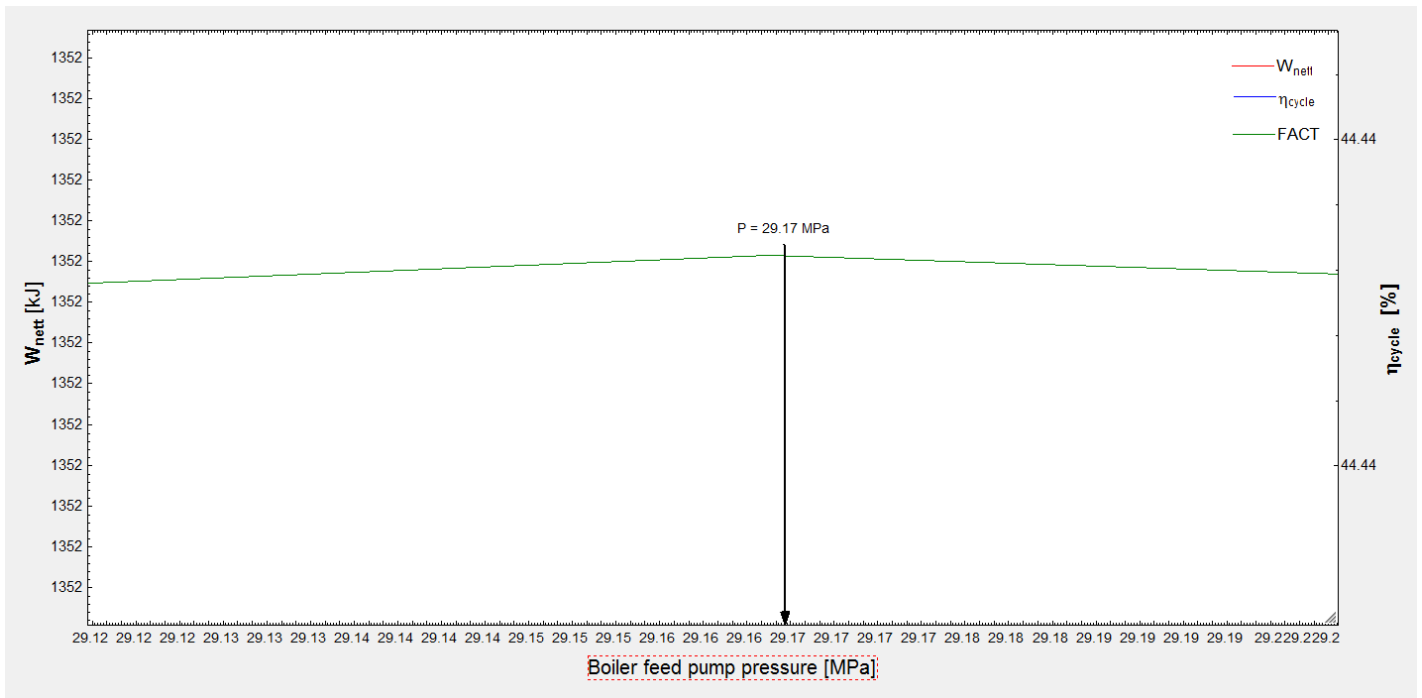


Figure 80: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (Zoomed in) (2nd method, 3rd run)

Before optimisation		After optimisation	
Boiler feed pump pressure	25.96 MPa	Boiler feed pump pressure	29.17 MPa
Cycle efficiency	45.96%	Cycle efficiency	46.25%
Mega Watt output	561.5 MW	Mega Watt output	558.3 MW
Net mechanical work	1353 kJ	Net mechanical work	1345 kJ
Heat added	2943 kJ	Heat added	2909 kJ
Heat rejected	1591 kJ	Heat rejected	1567 kJ

Table 45: Before and after optimisation results for the boiler feed pump discharge (2nd method, 3rd run)

6.1.3.3.2. High pressure heater 6

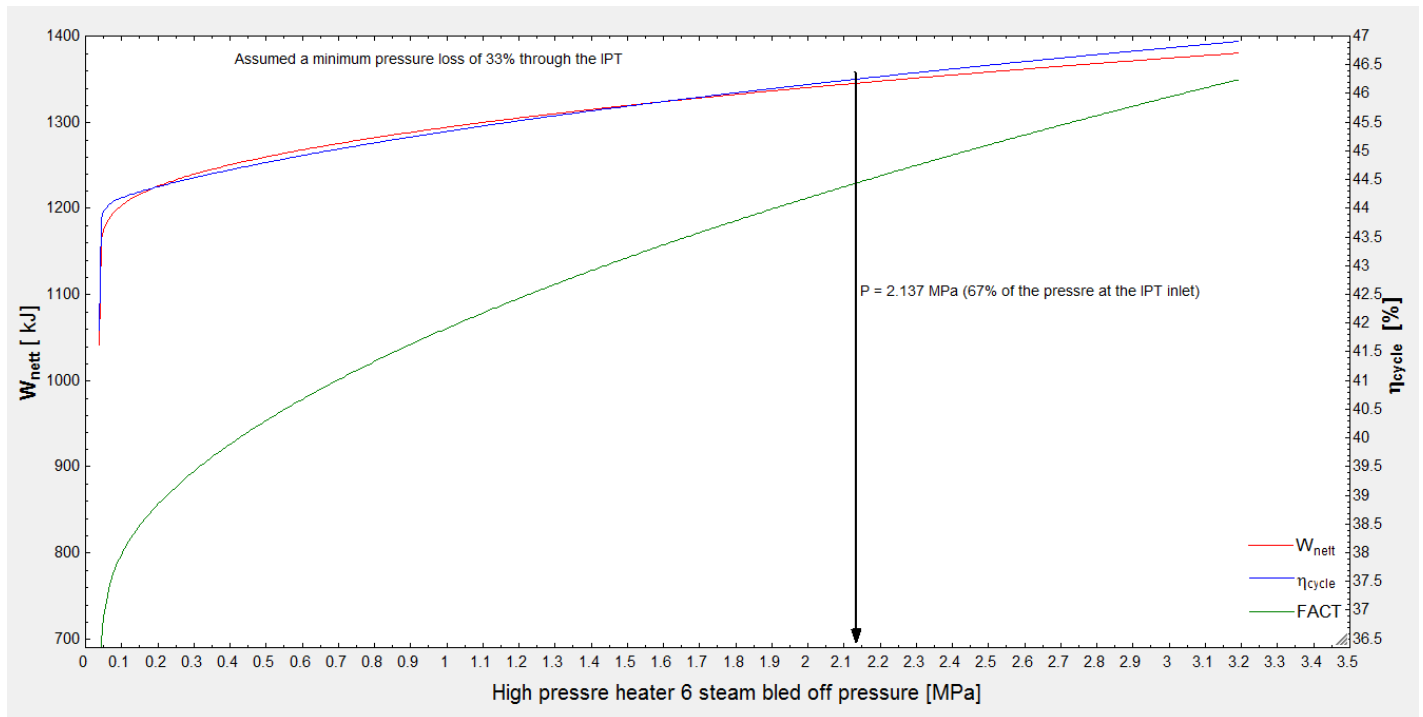


Figure 81: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
High pressure heater 6 steam tap off pressure	2.137 MPa	High pressure heater 6 steam tap off pressure	2.137 MPa
Cycle efficiency	46.25%	Cycle efficiency	46.25%
Mega Watt output	558.3 MW	Mega Watt output	558.3 MW
Net mechanical work	1345 kJ	Net mechanical work	1345 kJ
Heat added	2909 kJ	Heat added	2909 kJ
Heat rejected	1567 kJ	Heat rejected	1567 kJ

Table 46: Before and after optimisation results for high pressure heater 6 steam tap off pressure (2nd method, 3rd run)

6.1.3.3.3. High pressure turbine expansion

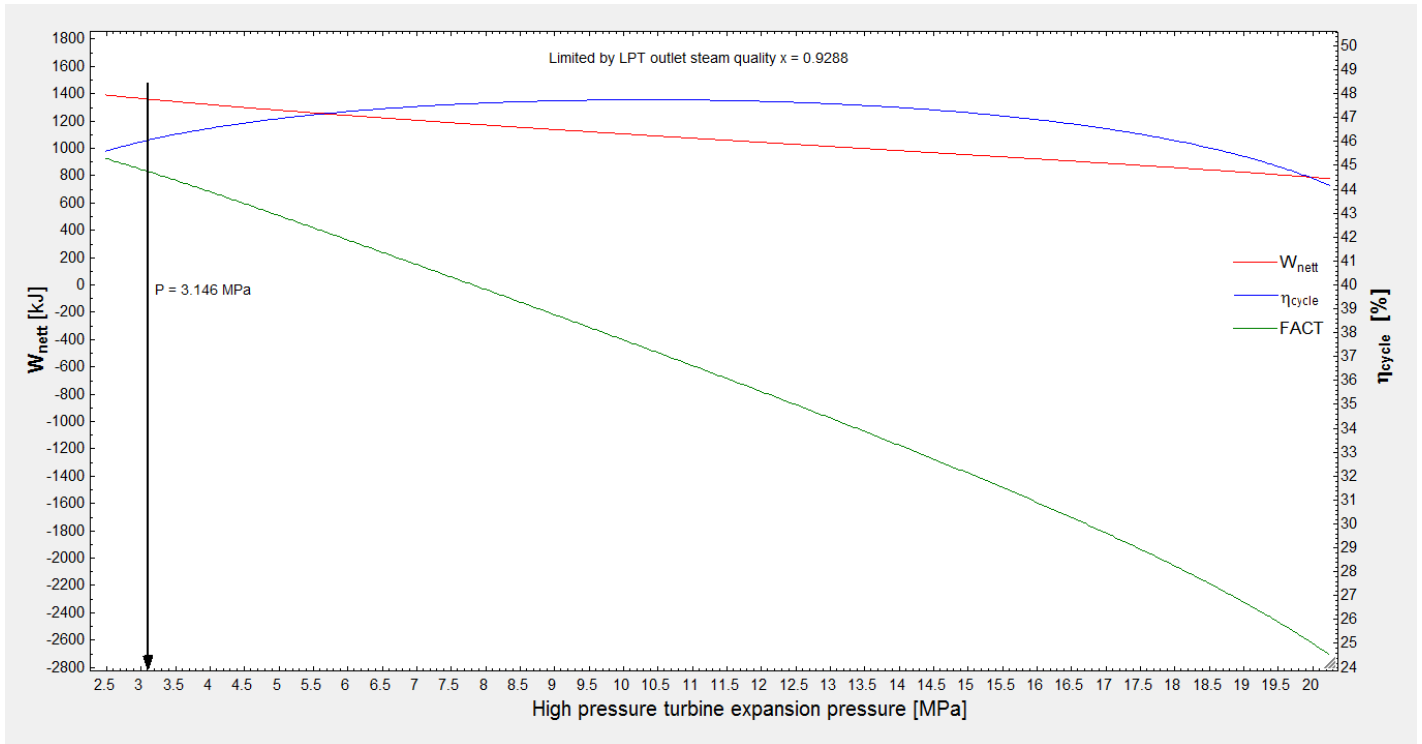


Figure 82: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.42 MPa	High pressure turbine expansion pressure	3.146 MPa
Cycle efficiency	46.25%	Cycle efficiency	46.08%
Mega Watt output	558.3 MW	Mega Watt output	563.5 MW
Net mechanical work	1345 kJ	Net mechanical work	1358 kJ
Heat added	2909 kJ	Heat added	2947 kJ
Heat rejected	1567 kJ	Heat rejected	1592 kJ
LPT outlet steam quality	X = 0.9242	LPT outlet steam quality	X = 0.9288

Table 47: Before and after optimisation results for the high pressure turbine expansion pressure (2nd method, 3rd run)

6.1.3.3.4. Low pressure heater 1

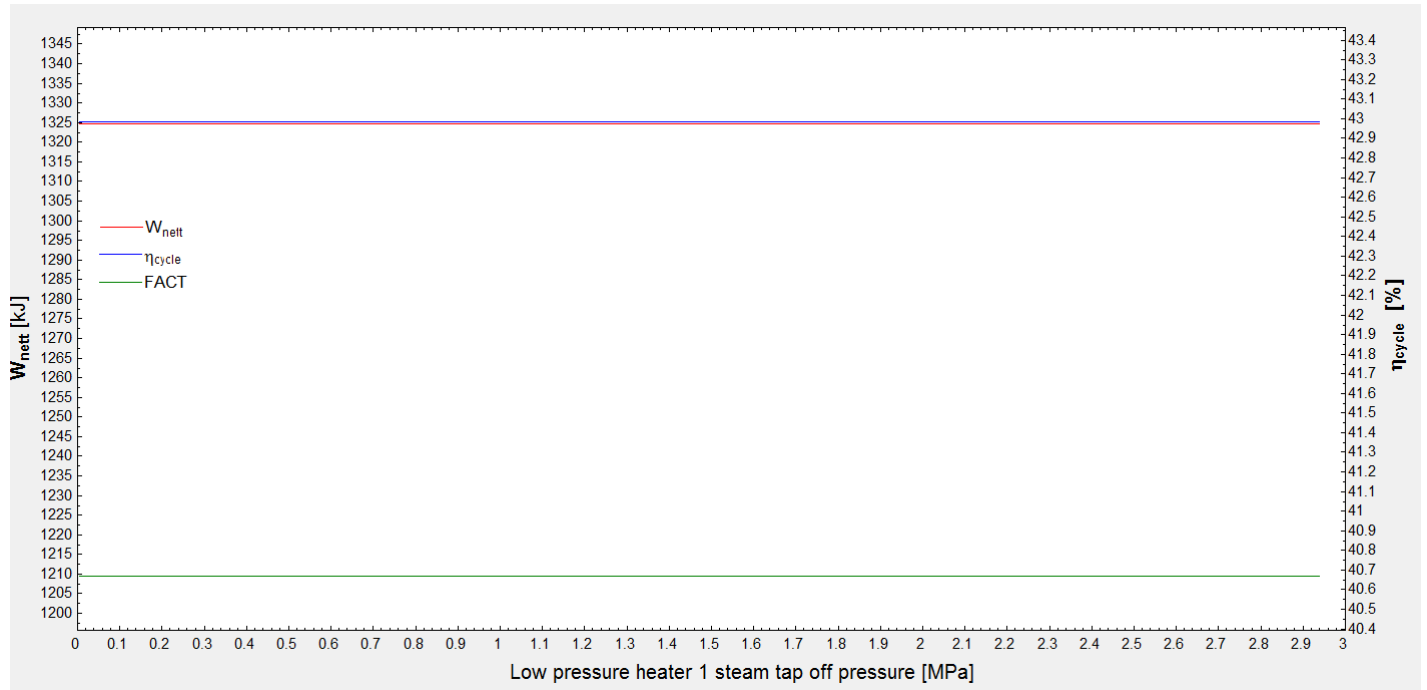


Figure 83: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.02775 MPa	Low pressure heater 1 steam tap off pressure	0.02775 MPa
Cycle efficiency	46.08%	Cycle efficiency	46.08%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2947 kJ	Heat added	2947 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 48: Before and after optimisation results for low pressure heater 1 steam tap off pressure (2nd method, 3rd run)

6.1.3.3.5. Low pressure heater 2

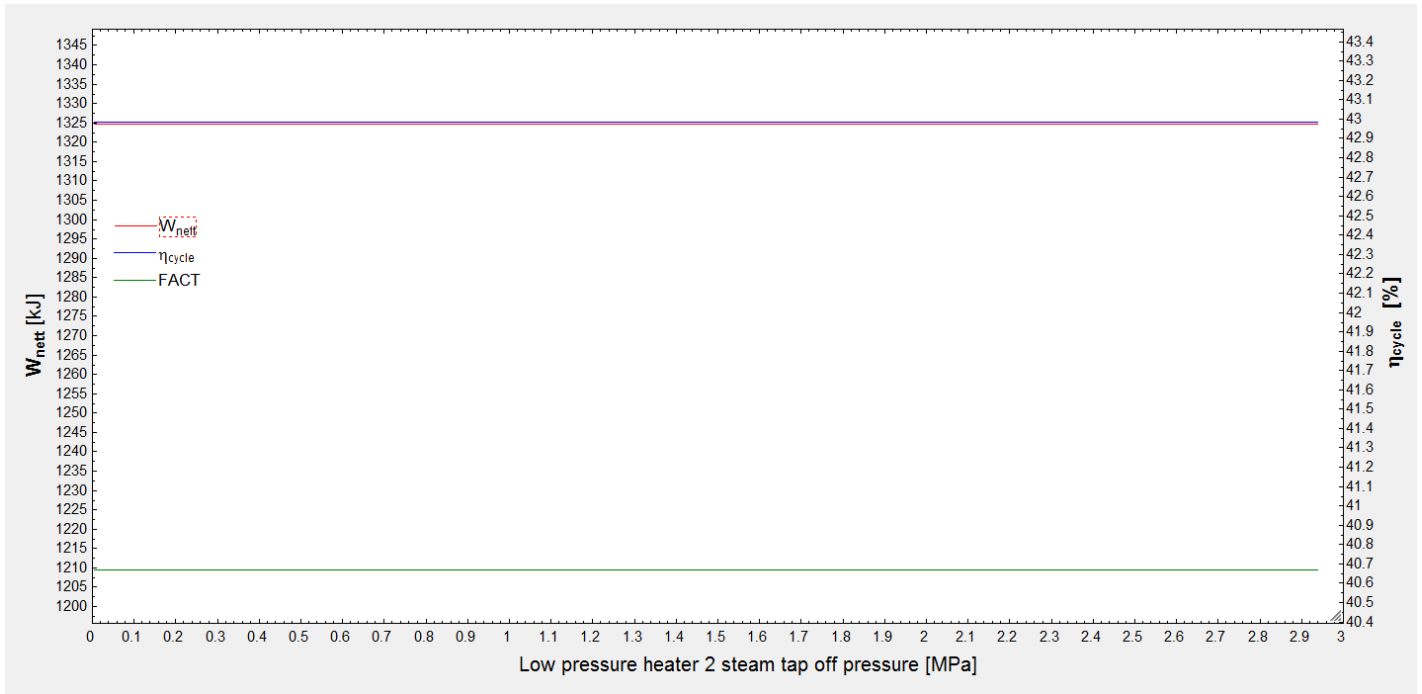


Figure 84: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.05609 MPa	Low pressure heater 2 steam tap off pressure	0.05609 MPa
Cycle efficiency	46.08%	Cycle efficiency	46.08%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2947 kJ	Heat added	2947 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 49: Before and after optimisation results for low pressure heater 2 steam tap off pressure (2nd method, 3rd run)

6.1.3.3.6. Low pressure heater 3

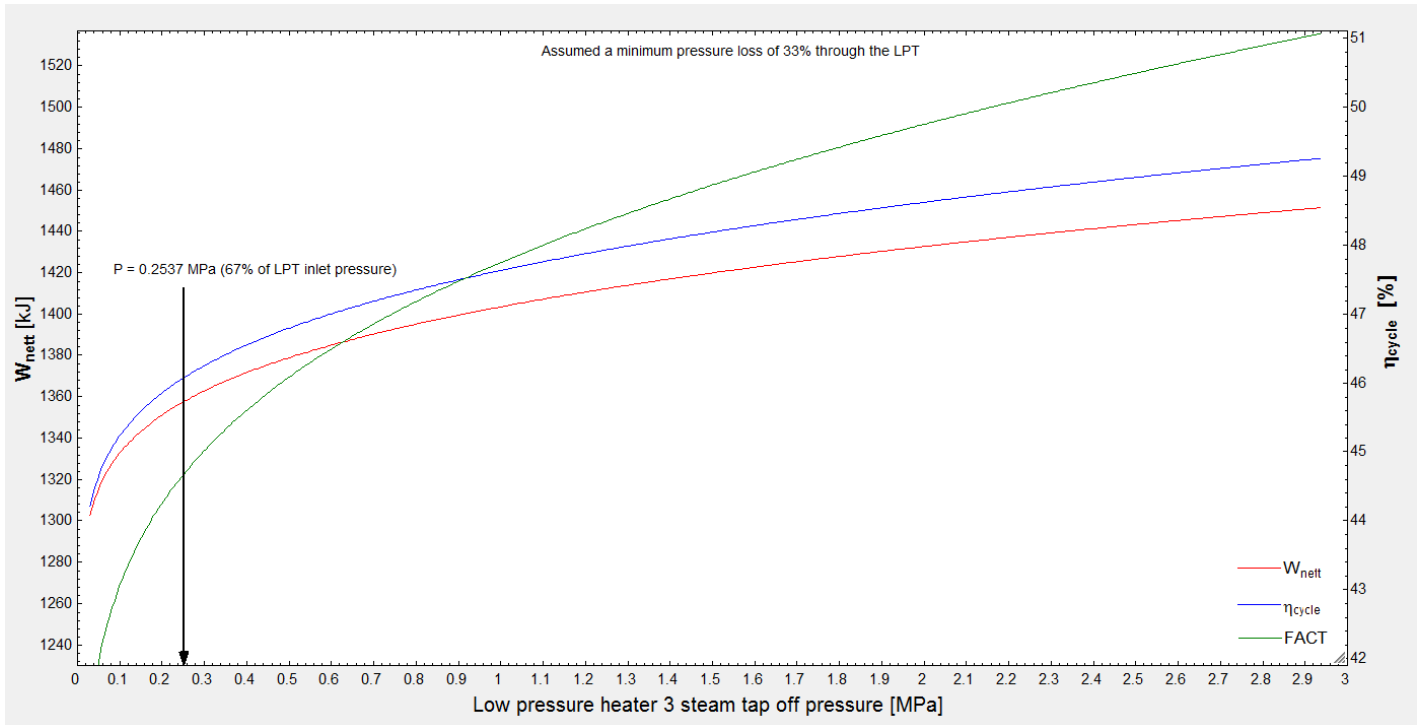


Figure 85: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steams bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.2537 MPa	Low pressure heater 3 steam tap off pressure	0.2537 MPa
Cycle efficiency	46.08%	Cycle efficiency	46.08%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2947 kJ	Heat added	2947 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 50: Before and after optimisation results for low pressure heater 3 steam tap off pressure (2nd method, 3rd run)

6.1.3.3.7. Low pressure heater 4

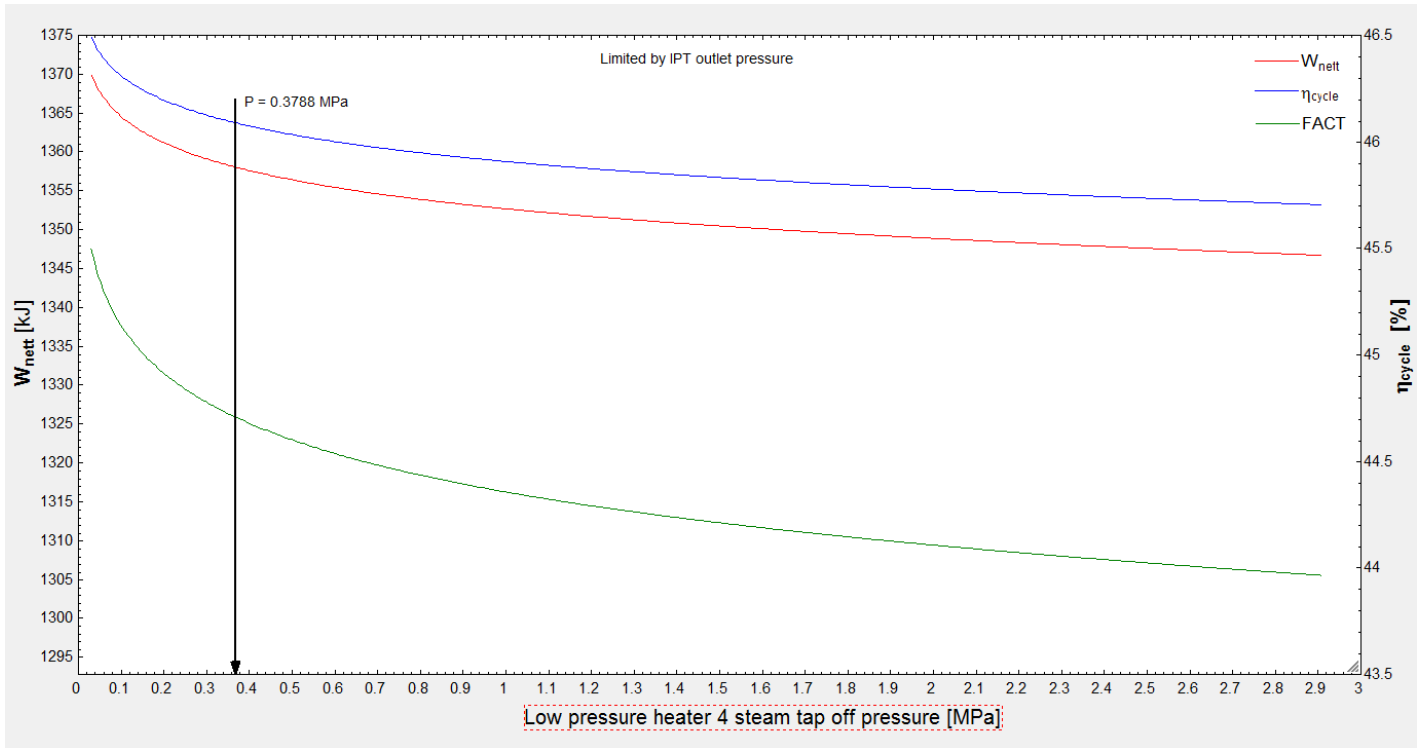


Figure 86: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 4 steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 4 steam tap off pressure	0.3788 MPa	Low pressure heater 4 steam tap off pressure	0.3788 MPa
Cycle efficiency	46.08%	Cycle efficiency	46.08%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2947 kJ	Heat added	2947 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 51: Before and after optimisation results for low pressure heater 4 steam tap off pressure (2nd method, 3rd run)

6.1.3.3.8. De-aerator

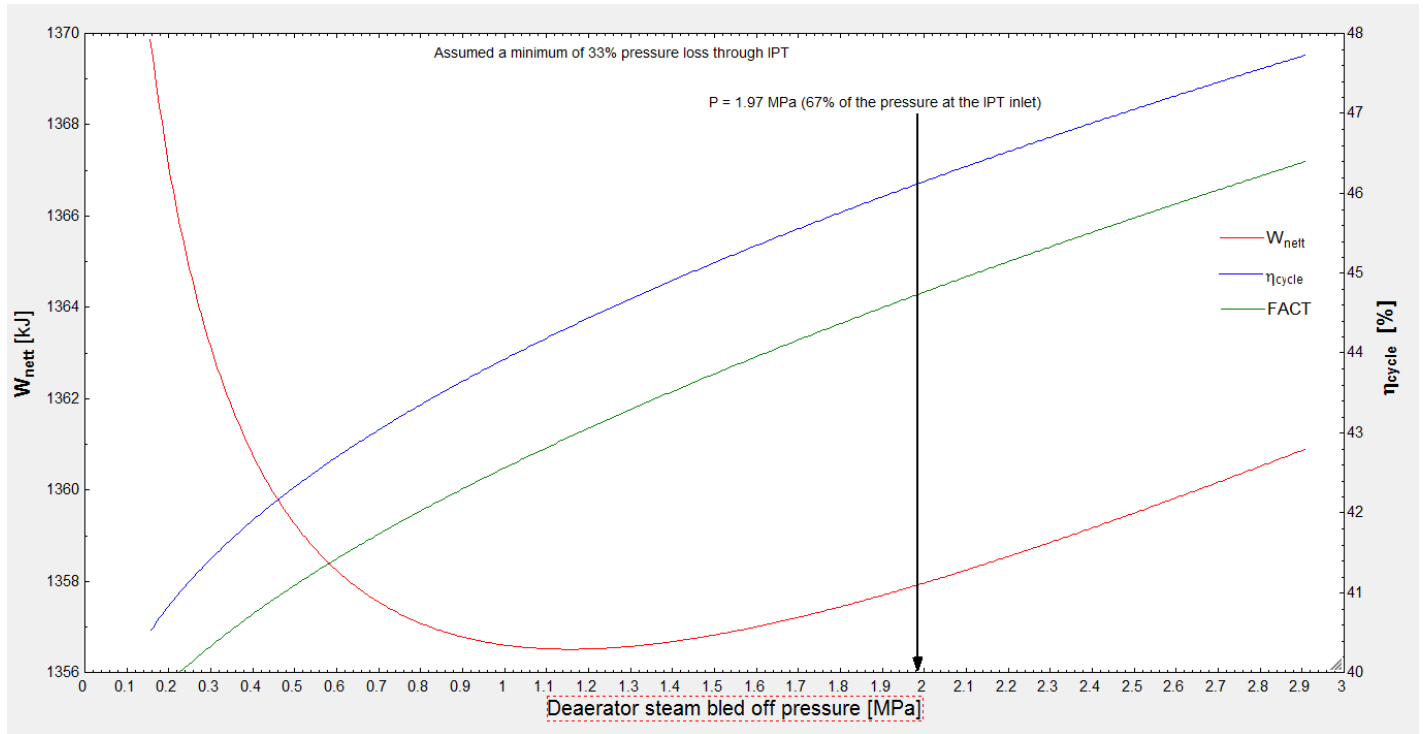


Figure 87: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
De-aerator steam tap off pressure	1.967 MPa	De-aerator steam tap off pressure	1.97 MPa
Cycle efficiency	46.08%	Cycle efficiency	46.09%
Mega Watt output	563.5 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2947 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 52: Before and after optimisation results for the de-aerator steam tap off pressure (2nd method, 3rd run)

6.1.3.3.9. Boiler feed pump

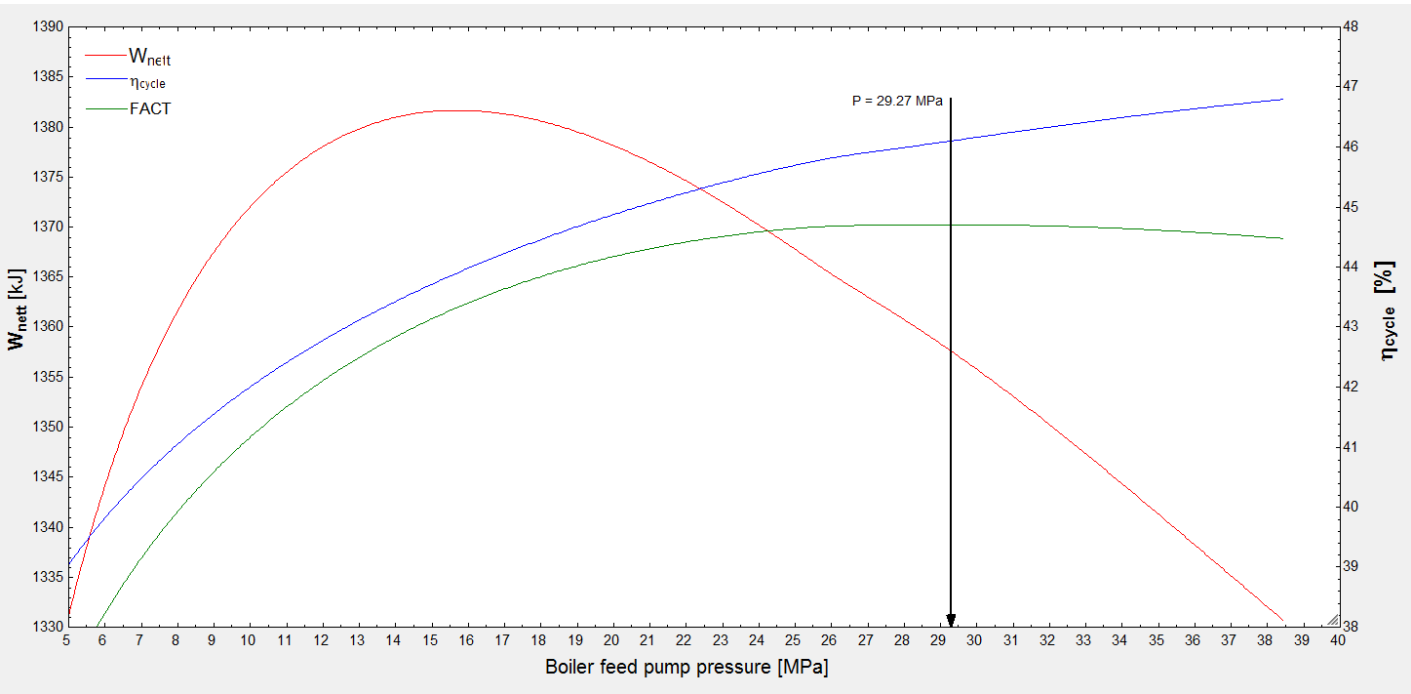


Figure 88: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (2nd method, 4th run)

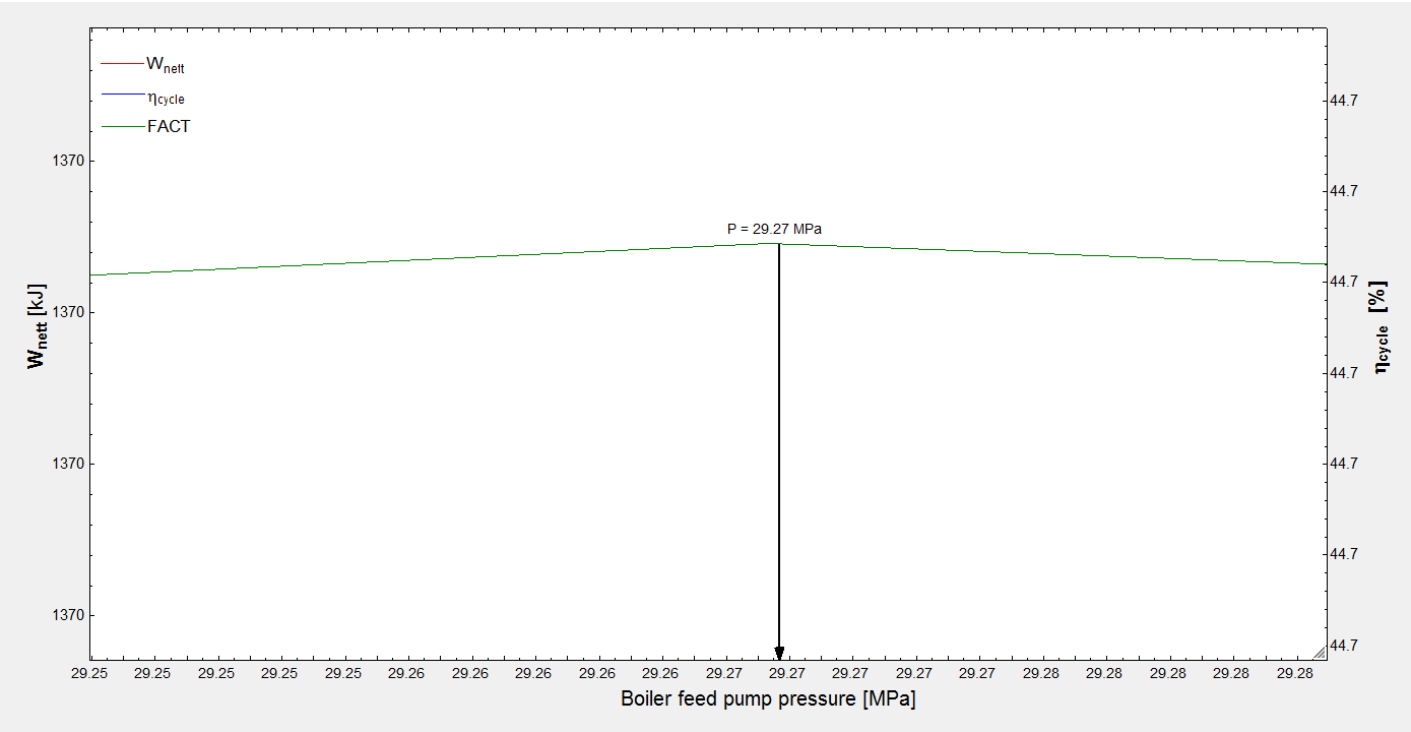


Figure 89: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (2nd method, 4th run)

Before optimisation		After optimisation	
Boiler feed pump pressure	29.17 MPa	Boiler feed pump pressure	29.27 MPa
Cycle efficiency	46.09%	Cycle efficiency	46.1%
Mega Watt output	563.5 MW	Mega Watt output	563.4 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2946 kJ	Heat added	2945 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ

Table 53: Before and after optimisation results for the boiler feed pump discharge (2nd method, 4th run)

6.1.3.3.10. High pressure turbine expansion

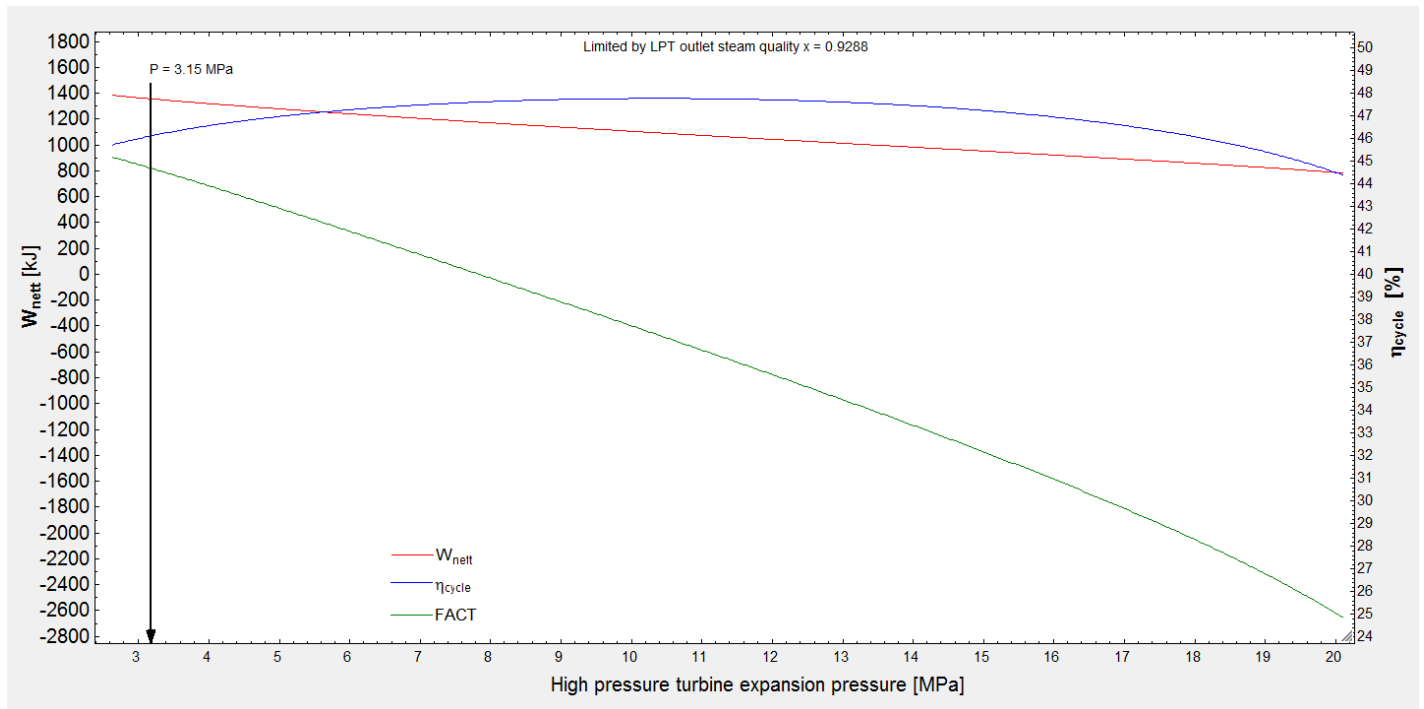


Figure 90: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (2nd method, 4th run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.154 MPa	High pressure turbine expansion pressure	3.15 MPa
Cycle efficiency	46.1%	Cycle efficiency	46.1%
Mega Watt output	563.4 MW	Mega Watt output	563.5 MW
Net mechanical work	1358 kJ	Net mechanical work	1358 kJ
Heat added	2945 kJ	Heat added	2946 kJ
Heat rejected	1592 kJ	Heat rejected	1592 kJ
LPT outlet steam quality	X = 0.9287	LPT outlet steam quality	X = 0.9288

Table 54: Before and after optimisation results for the high pressure turbine expansion pressure (2nd method, 4th run)

6.1.4. Adjustments after sub-critical cycle was optimised

After optimisation the first and second methods produced the same results.

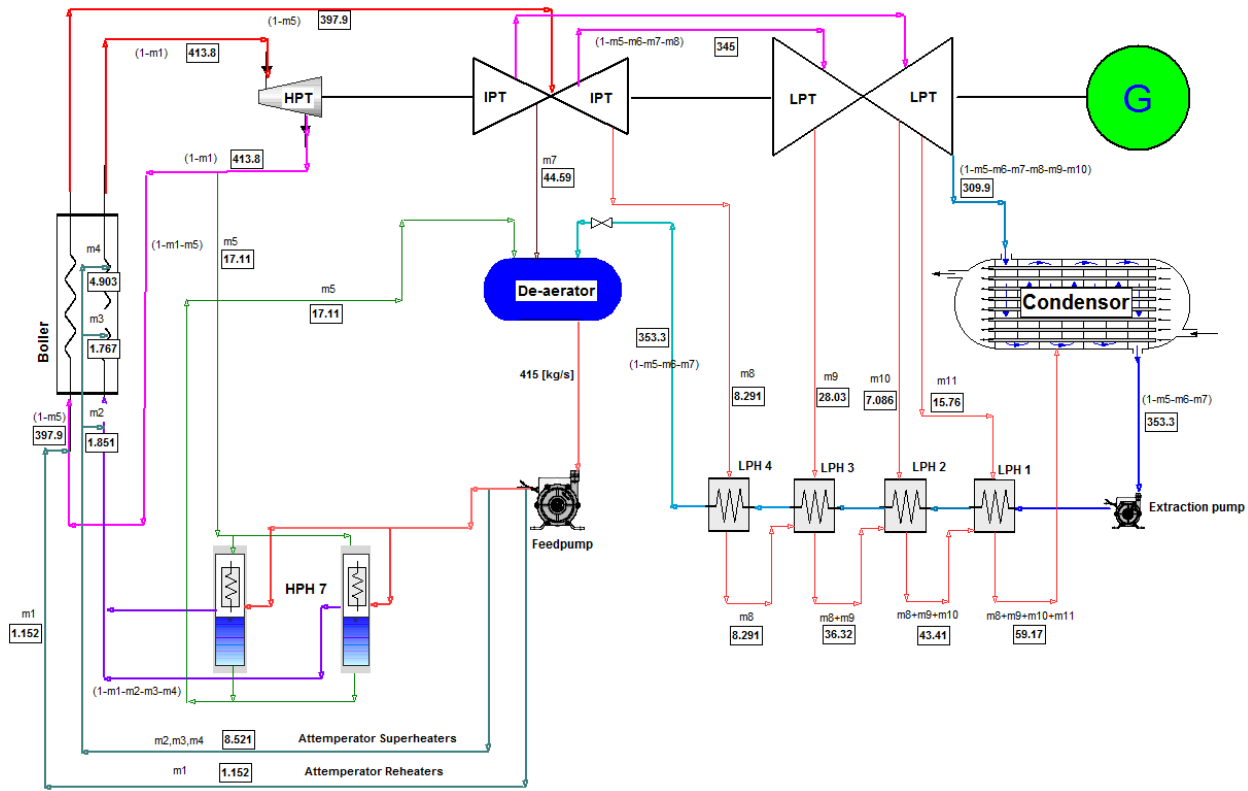


Figure 91: Sub-critical plant layout with steam flows, after high pressure heater 6 was taken out

The result of the tap off steam flow for high pressure heater 6 was negative, which means that steam was flowing back into the intermediate pressure turbine. Thus high pressure heater 6 was no longer necessary and was removed. The cycle was calculated again and the following results were obtained. For this section the steam tap off flow m_6 is equal to 0.

After high pressure heater 6 was taken out	
Maximum boiler pressure	3.154 MPa
Cycle efficiency	46.34%
Mega Watt output	569.2 MW
Net mechanical work	1372 kJ
Heat added	2960 kJ
Heat rejected	1600 kJ

Table 55: Results for sub-critical cycle after optimisation and after high pressure heater 6 was taken out

6.1.5. Summary of sub-critical results

First method			
	Boiler feed pump pressure	Efficiency	Mega Watt output
Current operating parameters	22.14 MPa	42.69%	538.4 MW
After first run	24.3 MPa	45.86%	559.6 MW
After second run	29.1 MPa	46.09%	563.5 MW
After third run	29.3 MPa	46.10%	563.6 MW
After fourth run	29.29 MPa	46.10%	563.6 MW
After HPH6 was taken out	29.29 MPa	46.34%	569.2 MW

Second method			
	Boiler feed pump pressure	Efficiency	Mega Watt output
Current operating parameters	22.14 MPa	42.69%	538.4 MW
After first run	24.3 MPa	45.96%	561.5 MW
After second run	25.96 MPa	45.96%	561.5 MW
After third run	29.17 MPa	46.10%	563.6 MW
After fourth run	29.27 MPa	46.10%	563.4 MW

Table 56: Optimisation results for each run and method of the sub-critical Rankine cycle

6.2. Super-critical optimisation results

The super-critical cycle was based on the heat balance diagrams from Medupi power station at 794MW output

6.2.1. Super-critical cycle without optimisation

6.2.1.1. Input parameters

The input parameters were taken from the original heat balance diagram for Medupi power station.

Feed pump	30.13 MPa
Maximum Temperature	570°C
Minimum Temperature	50°C
Maximum mass flow	635 kg/s
Extraction pump discharge pressure	2.5 MPa
High pressure heater 6 bled steam tap off pressure	2.745 MPa
De-aerator bled steam tap off pressure	1.12 MPa
Low pressure heater 3 bled steam tap off pressure	0.591 MPa
Low pressure heater 2 bled steam tap off pressure	0.2671 MPa
Low pressure heater 1 bled steam tap off pressure	0.09048 MPa

Table 57: Super-critical Rankine cycle input parameters

6.2.1.2. Results

This is the current Rankine cycle as it is on the heat balance diagram of Medupi Power station

Cycle efficiency	42.31 %
Mega Watt out	831.6 MW
Total Work in	38.65 kJ
Total Work out	1348 kJ
Net mechanical work	1310 kJ
Heat added	3096 kJ
Heat rejected	1495 kJ

Table 58: Results for super-critical cycle with input parameters before optimisation

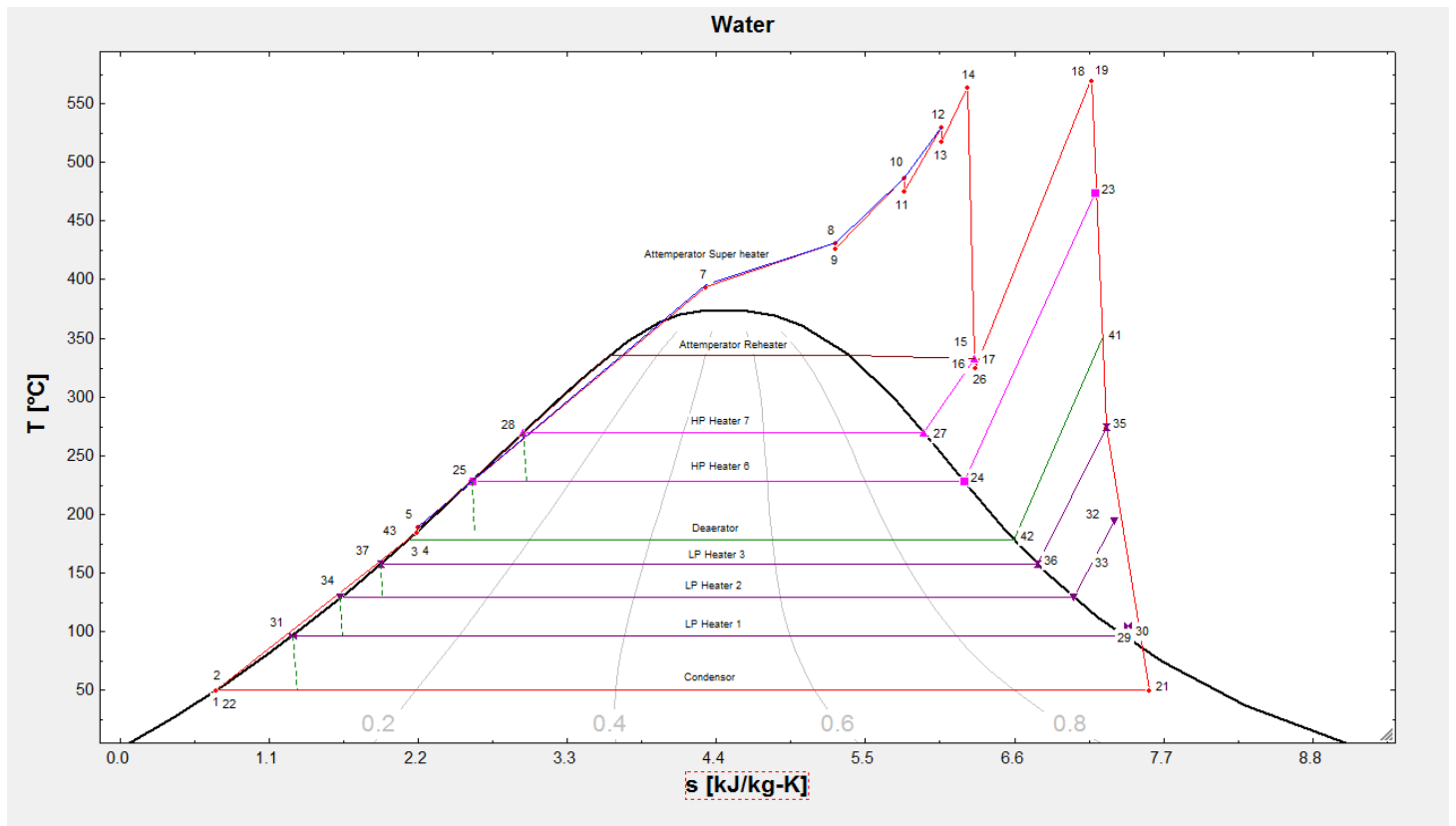


Figure 92: Current super-critical Rankine cycle before any optimisation

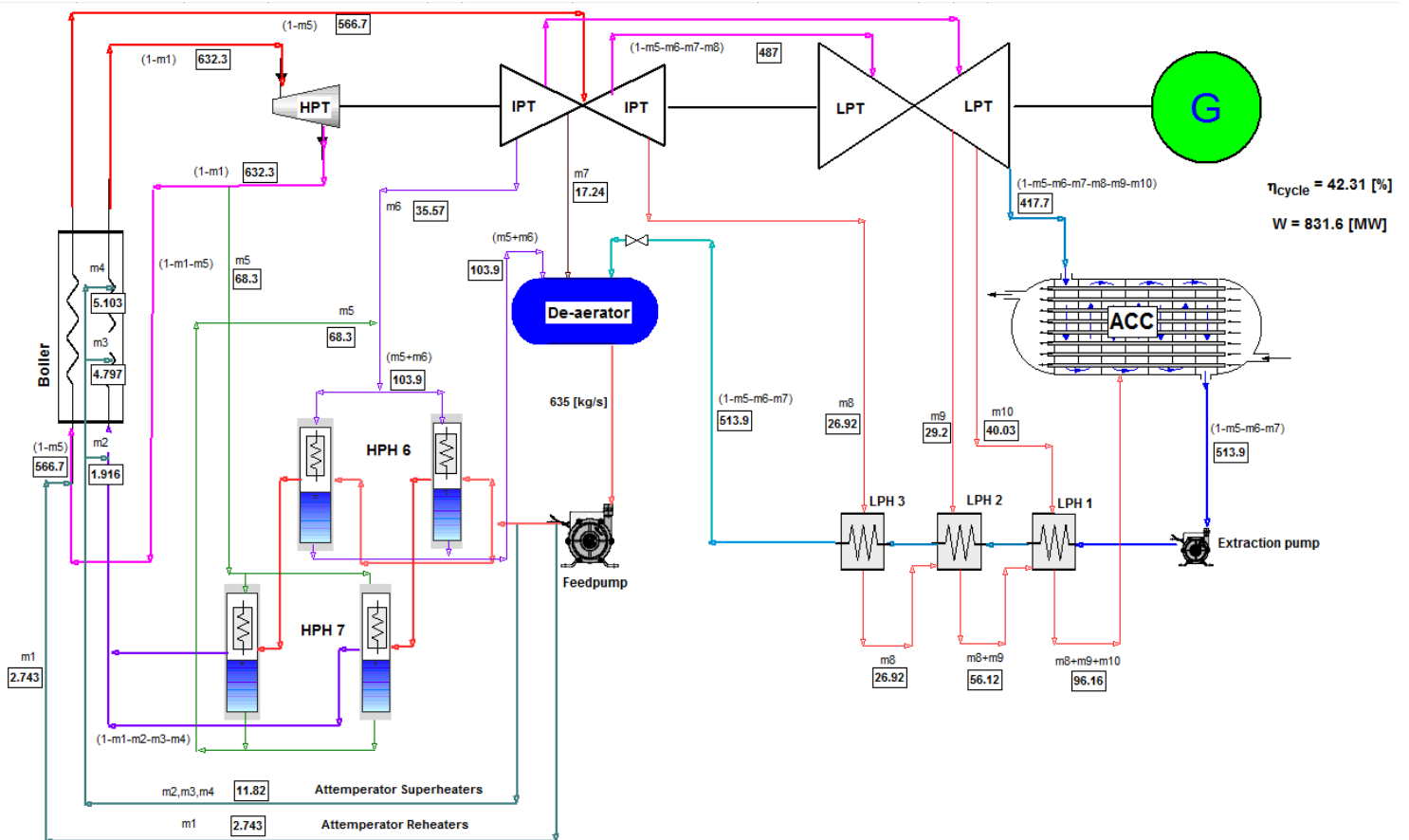


Figure 93: Plant layout at Medupi power station (super-critical)

6.2.2. Super-critical cycle optimisation – first method

6.2.2.1. First run

6.2.2.1.1. Boiler feed pump pressure

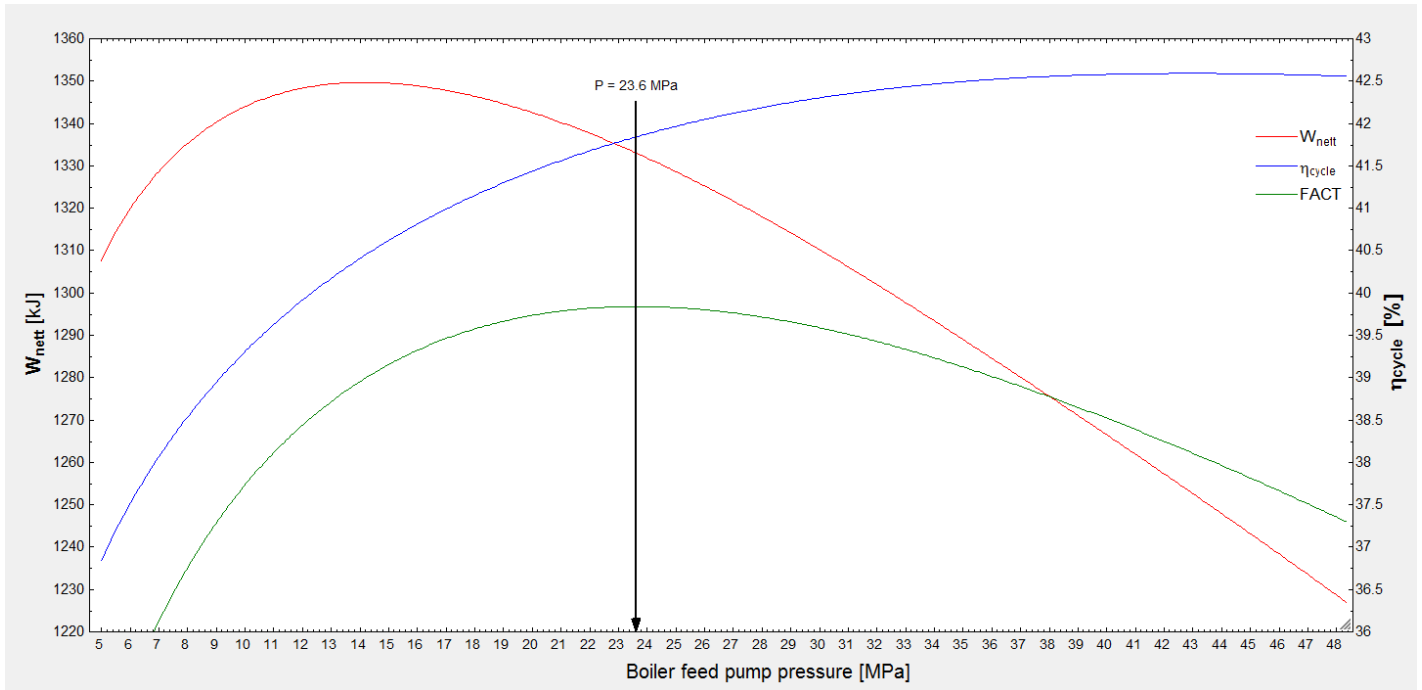


Figure 94: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (1st method, 1st run)

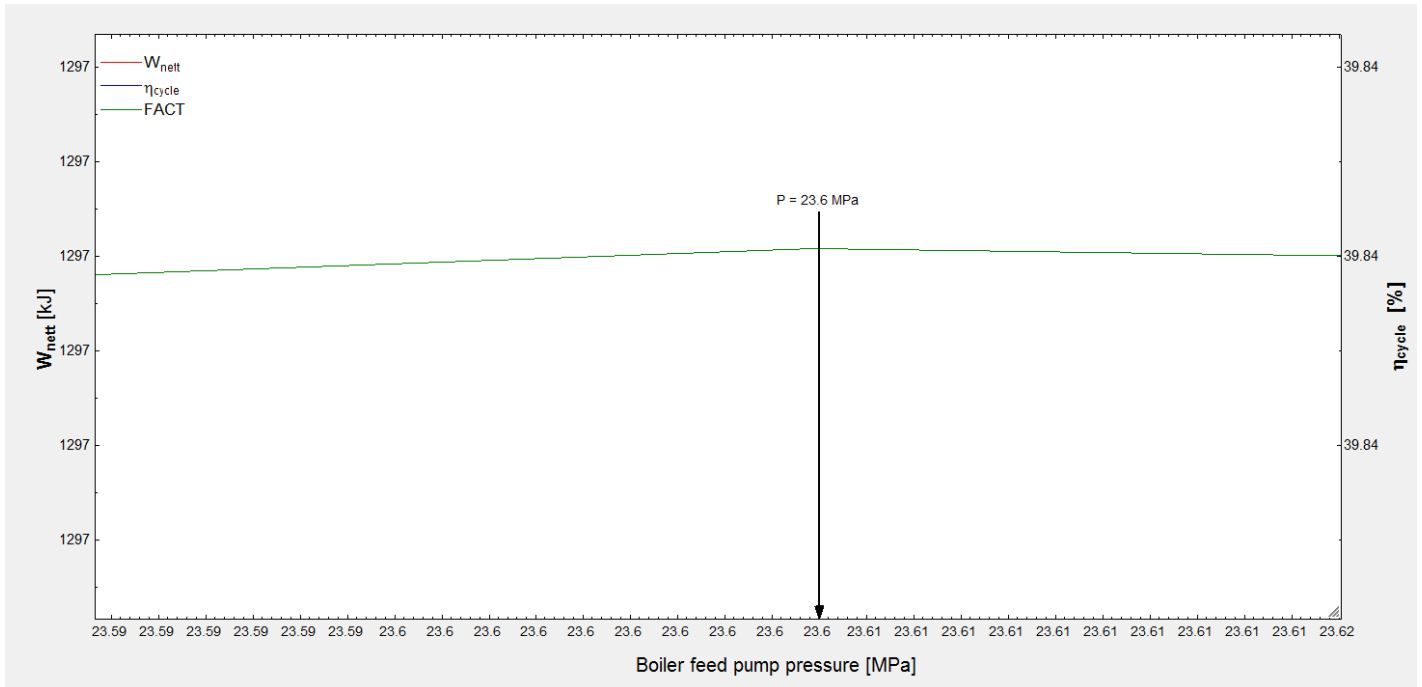


Figure 95: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (Zoomed in) (1st method, 1st run)

Before optimisation		After optimisation	
Boiler feed pump pressure	30.13 MPa	Boiler feed pump pressure	23.6 MPa
Cycle efficiency	42.31%	Cycle efficiency	41.84%
Mega Watt output	831.6 MW	Mega Watt output	846.5 MW
Net mechanical work	1310 kJ	Net mechanical work	1333 kJ
Heat added	3096 kJ	Heat added	3186 kJ
Heat rejected	1495 kJ	Heat rejected	1575 kJ

Table 59: Before and after optimisation results for the boiler feed pump discharge (1st method, 1st run)

6.2.2.1.2. High pressure turbine expansion

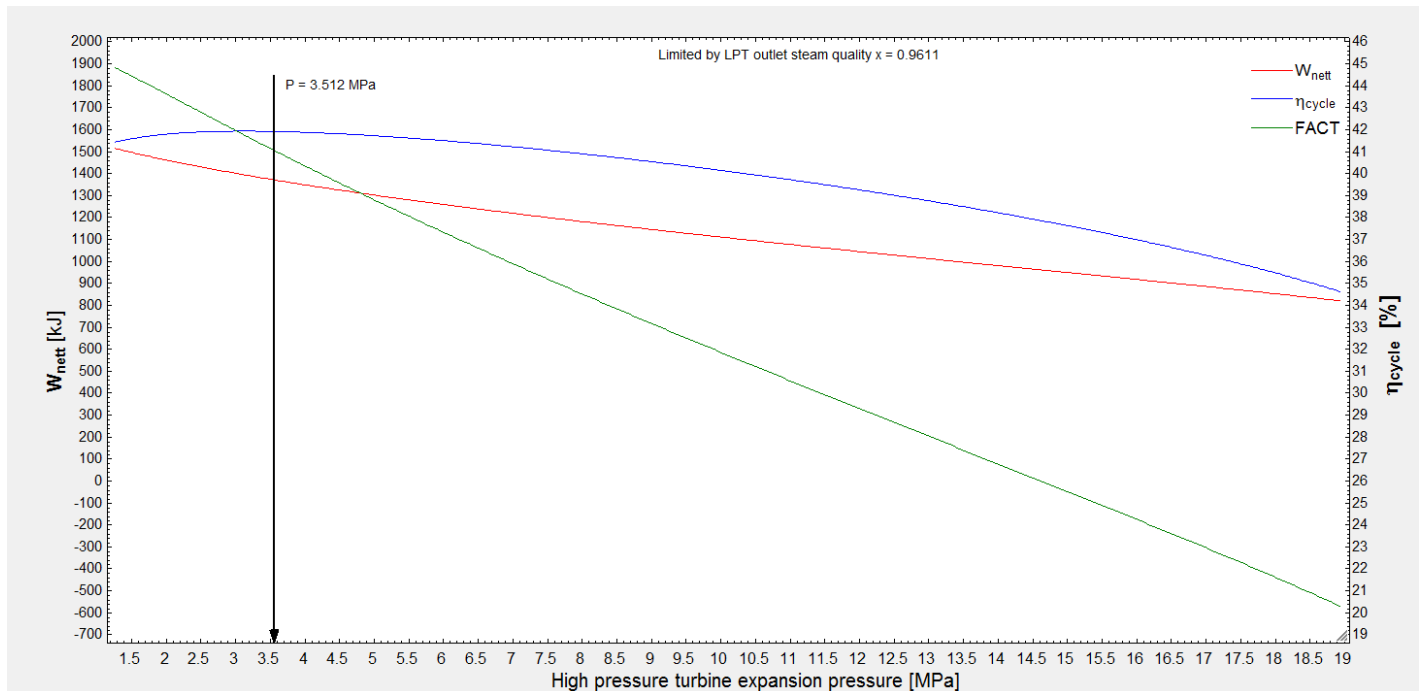


Figure 96: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (1st method, 1st run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	4.323 MPa	High pressure turbine expansion pressure	3.512 MPa
Cycle efficiency	41.84%	Cycle efficiency	41.92 %
Mega Watt output	846.5 MW	Mega Watt output	872 MW
Net mechanical work	1333 kJ	Net mechanical work	1373 kJ
Heat added	3186 kJ	Heat added	3276 kJ
Heat rejected	1575 kJ	Heat rejected	1640 kJ
LPT outlet steam quality	X = 0.9491	LPT outlet steam quality	X = 0.9611

Table 60: Before and after optimisation results for the high pressure turbine expansion pressure (1st method, 1st run)

6.2.2.1.3. High pressure heater 6

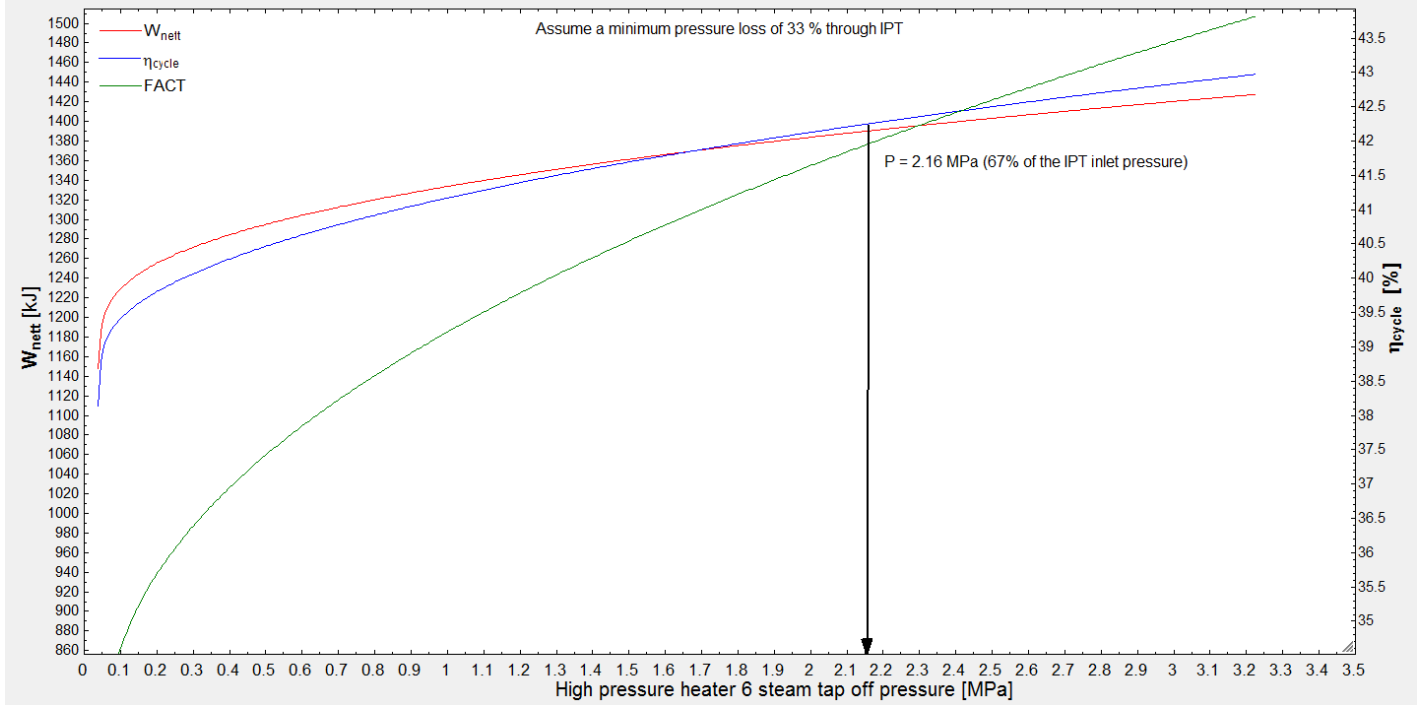


Figure 97: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (1st method, 1st run)

Before optimisation		After optimisation	
High pressure heater 6 steam bled off pressure	1.755 MPa	High pressure heater 6 steam bled off pressure	2.16 MPa
Cycle efficiency	41.92 %	Cycle efficiency	42.25 %
Mega Watt output	872 MW	Mega Watt output	882.8 MW
Net mechanical work	1373 kJ	Net mechanical work	1390 kJ
Heat added	3276 kJ	Heat added	3291 kJ
Heat rejected	1640 kJ	Heat rejected	1651 kJ

Table 61: Before and after optimisation results for high pressure heater 6 steam tap off pressure (1st method, 1st run)

6.2.2.1.4. De-aerator

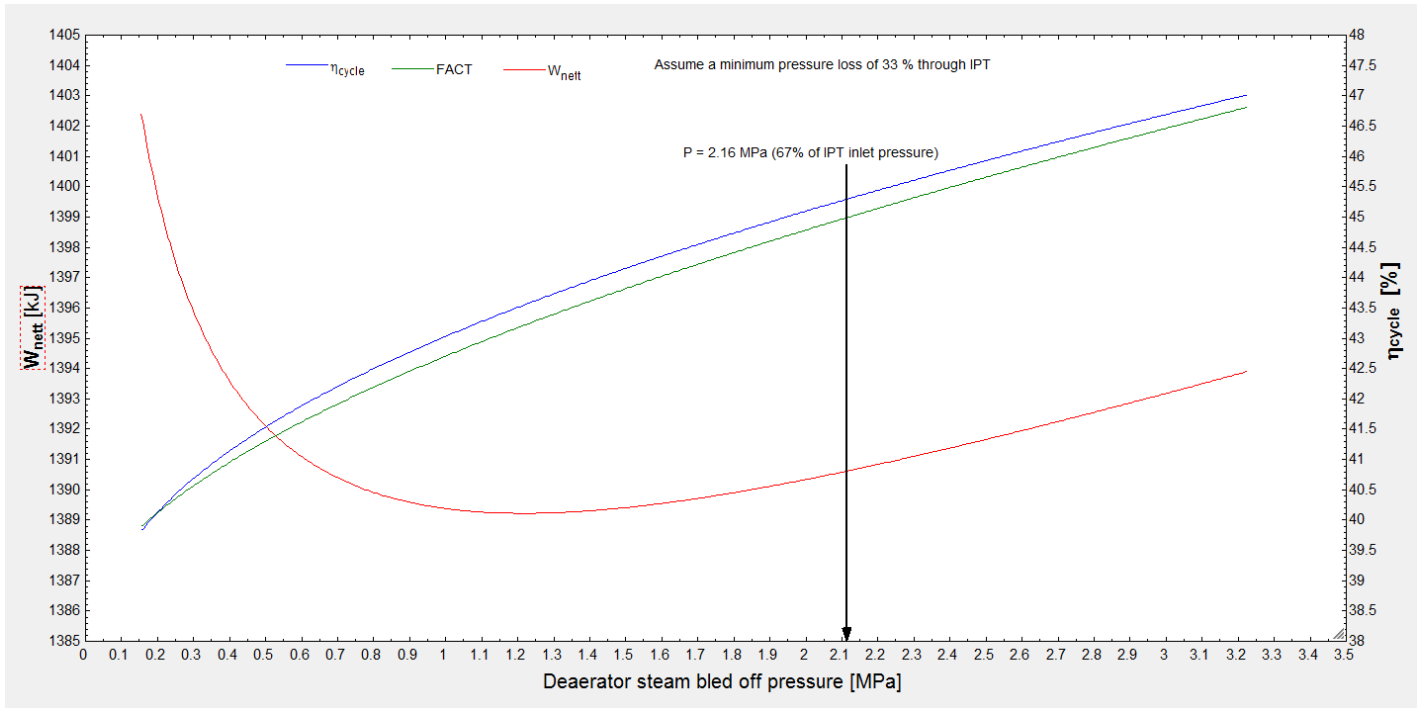


Figure 98: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
De-aerator steam bled off pressure	0.7161 MPa	De-aerator steam bled off pressure	2.16 MPa
Cycle efficiency	42.25 %	Cycle efficiency	45.37 %
Mega Watt output	882.8 MW	Mega Watt output	883.1 MW
Net mechanical work	1390 kJ	Net mechanical work	1391 kJ
Heat added	3291 kJ	Heat added	3065 kJ
Heat rejected	1651 kJ	Heat rejected	1653 kJ

Table 62: Before and after optimisation results for the de-aerator steam tap off pressure (1st method, 1st run)

6.2.2.1.5. Low pressure heater 3

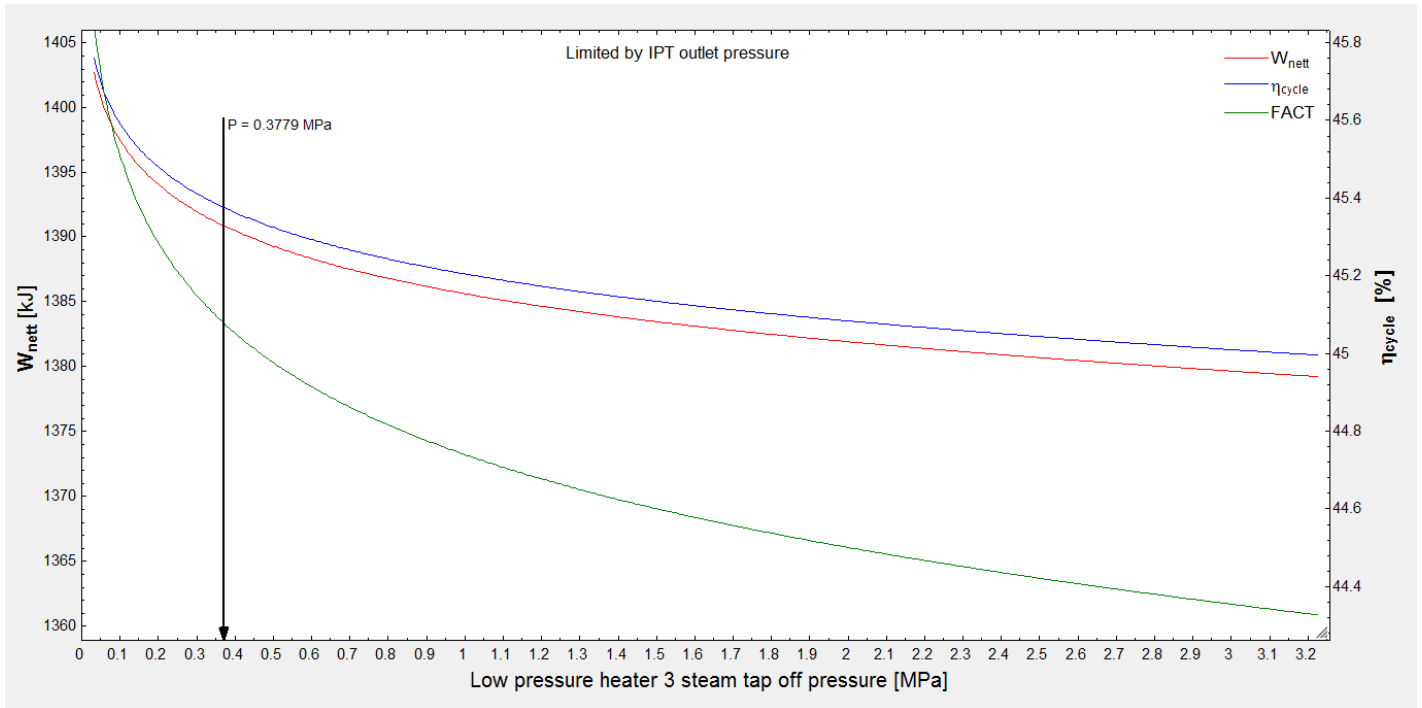


Figure 99: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.378 MPa	Low pressure heater 3 steam tap off pressure	0.3779 MPa
Cycle efficiency	45.37 %	Cycle efficiency	45.37 %
Mega Watt output	883.1 MW	Mega Watt output	883.1 MW
Net mechanical work	1391 kJ	Net mechanical work	1391 kJ
Heat added	3065 kJ	Heat added	3065 kJ
Heat rejected	1653 kJ	Heat rejected	1653 kJ

Table 63: Before and after optimisation results for low pressure heater 3 steam tap off pressure (1st method, 1st run)

6.2.2.1.6. Low pressure heater 2

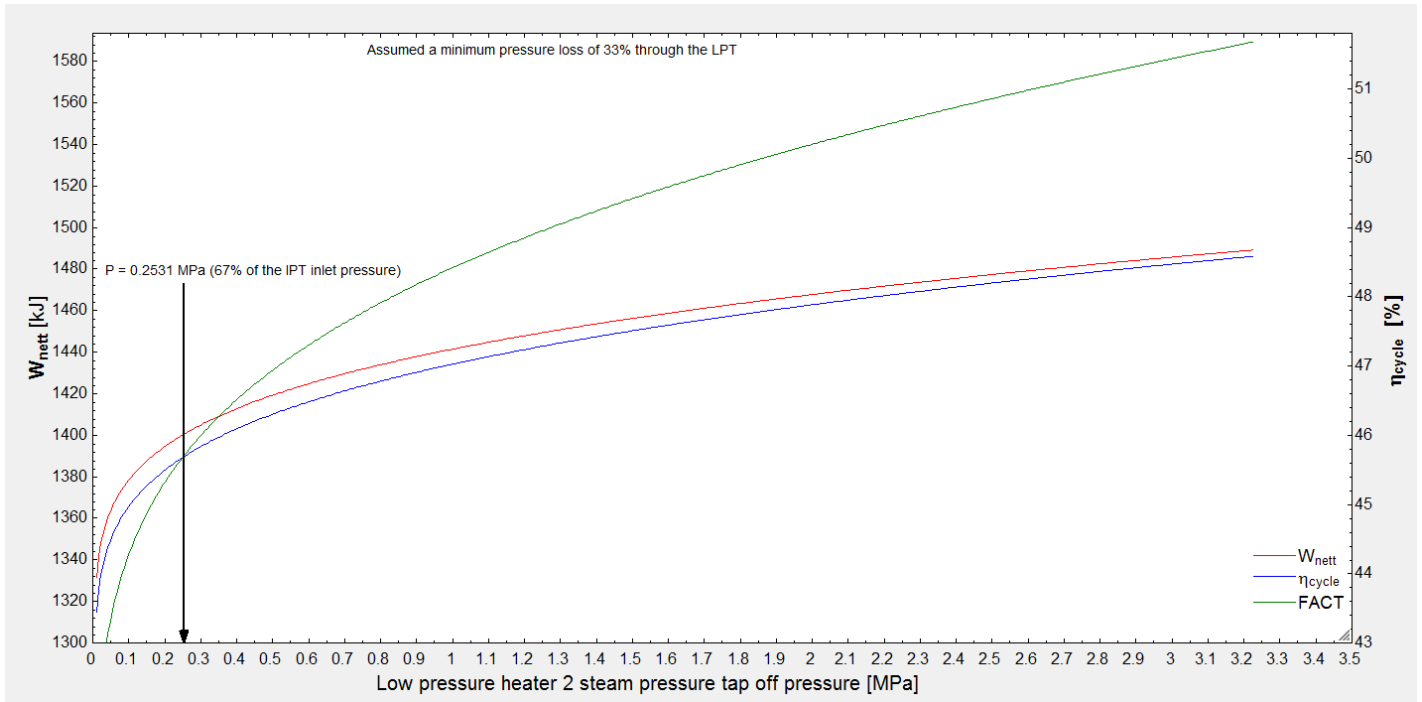


Figure 100: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.1709 MPa	Low pressure heater 2 steam tap off pressure	0.2531 MPa
Cycle efficiency	45.37 %	Cycle efficiency	45.69 %
Mega Watt output	883.1 MW	Mega Watt output	889.3 MW
Net mechanical work	1391 kJ	Net mechanical work	1400 kJ
Heat added	3065 kJ	Heat added	3065 kJ
Heat rejected	1653 kJ	Heat rejected	1655 kJ

Table 64: Before and after optimisation results for low pressure heater 2 steam tap off pressure (1st method, 1st run)

6.2.2.1.7. Low pressure heater 1

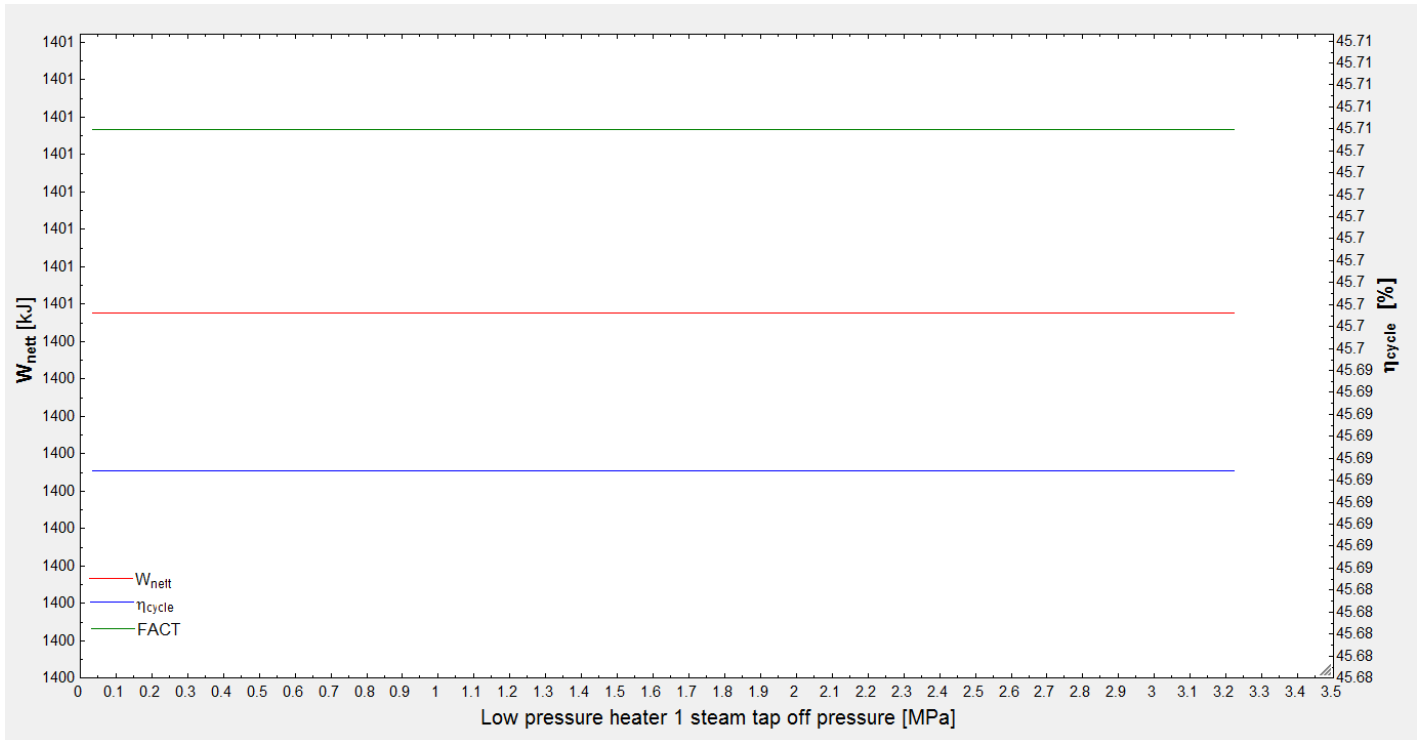


Figure 101: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.05787 MPa	Low pressure heater 1 steam tap off pressure	0.05787 MPa
Cycle efficiency	45.69 %	Cycle efficiency	45.69 %
Mega Watt output	889.3 MW	Mega Watt output	889.3 MW
Net mechanical work	1400 kJ	Net mechanical work	1400 kJ
Heat added	3065 kJ	Heat added	3065 kJ
Heat rejected	1655 kJ	Heat rejected	1655 kJ

Table 65: Before and after optimisation results for low pressure heater 1 steam tap off pressure (1st method, 1st run)

6.2.2.2. Second run

6.2.2.2.1. Boiler feed pump pressure

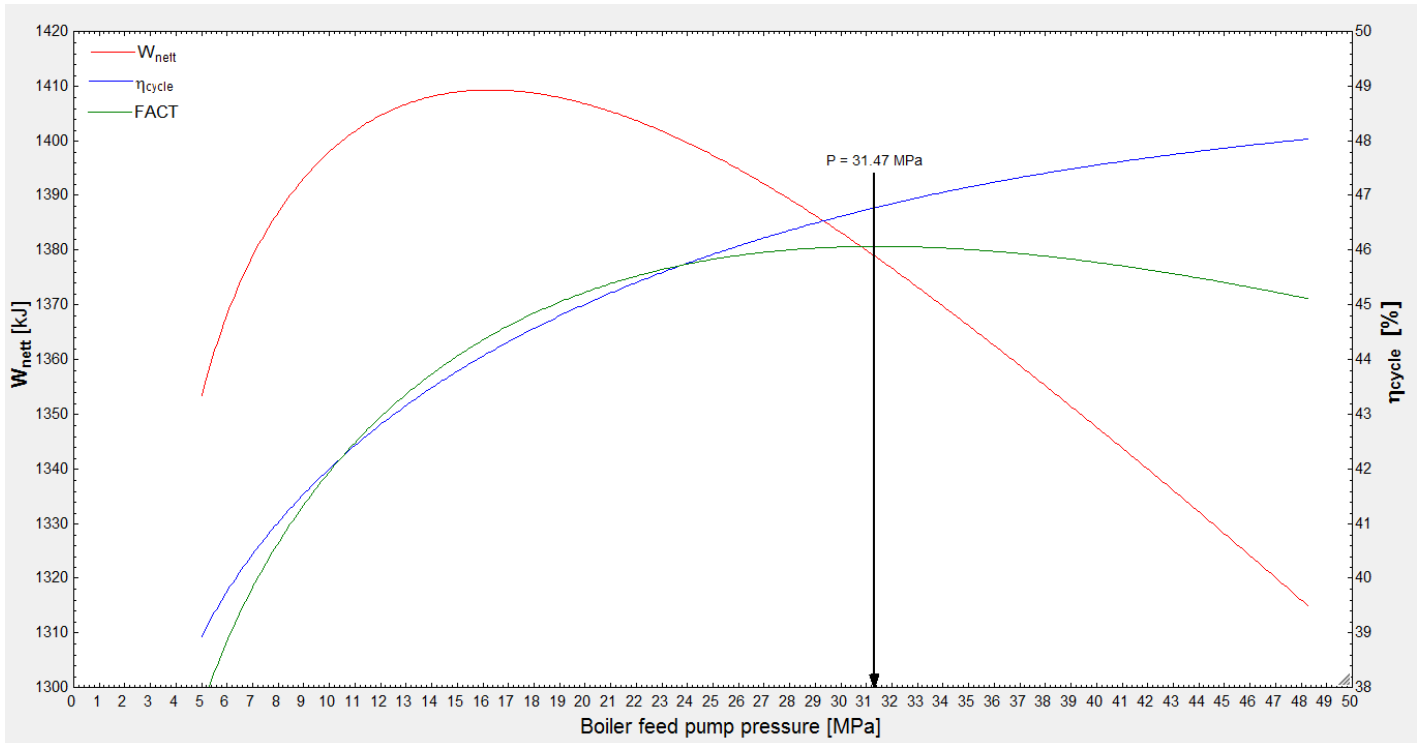


Figure 102: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (1st method, 2nd run)

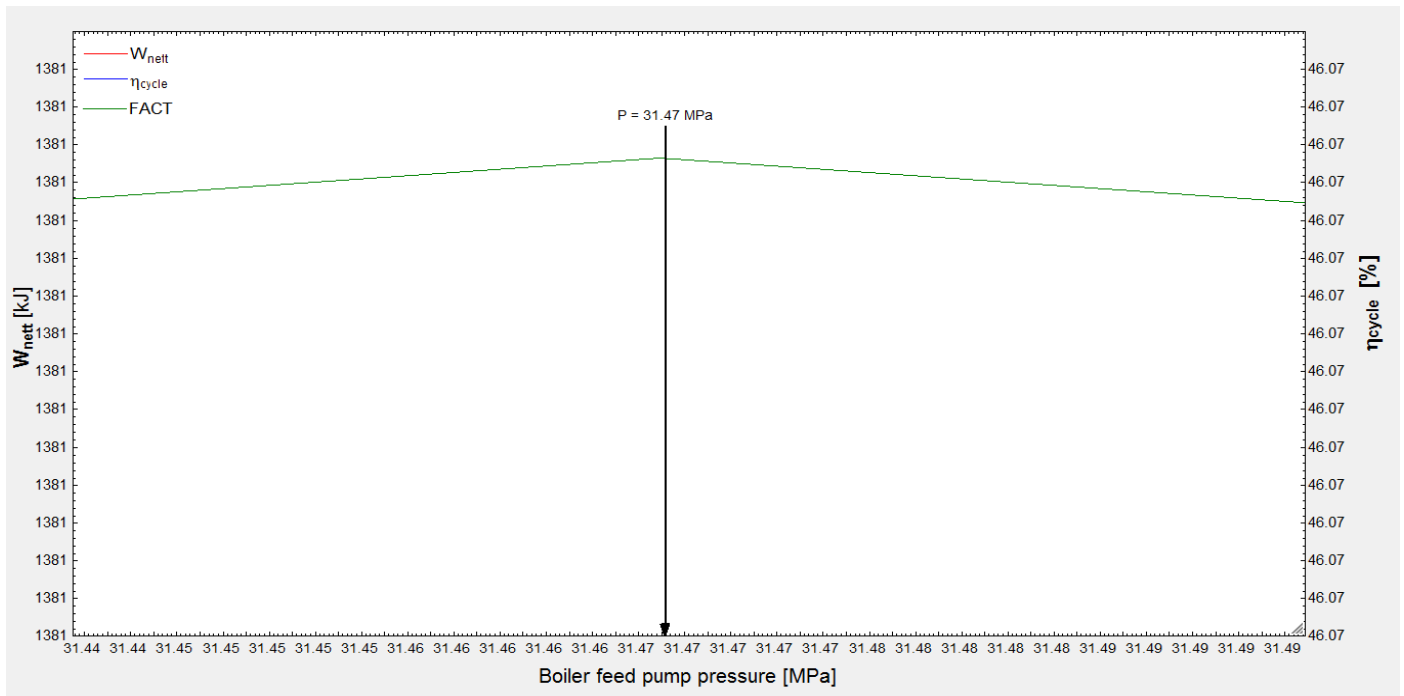


Figure 103: Cycle efficiency, net-work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (1st method, 2nd run)

Before optimisation		After optimisation	
Boiler feed pump pressure	23.6 MPa	Boiler feed pump pressure	31.47 MPa
Cycle efficiency	45.69 %	Cycle efficiency	46.79 %
Mega Watt output	889.3 MW	Mega Watt output	875.3 MW
Net mechanical work	1400 kJ	Net mechanical work	1378 kJ
Heat added	3065 kJ	Heat added	2946 kJ
Heat rejected	1655 kJ	Heat rejected	1565 kJ

Table 66: Before and after optimisation results for the boiler feed pump discharge (1st method, 2nd run)

6.2.2.2.2. High pressure turbine expansion

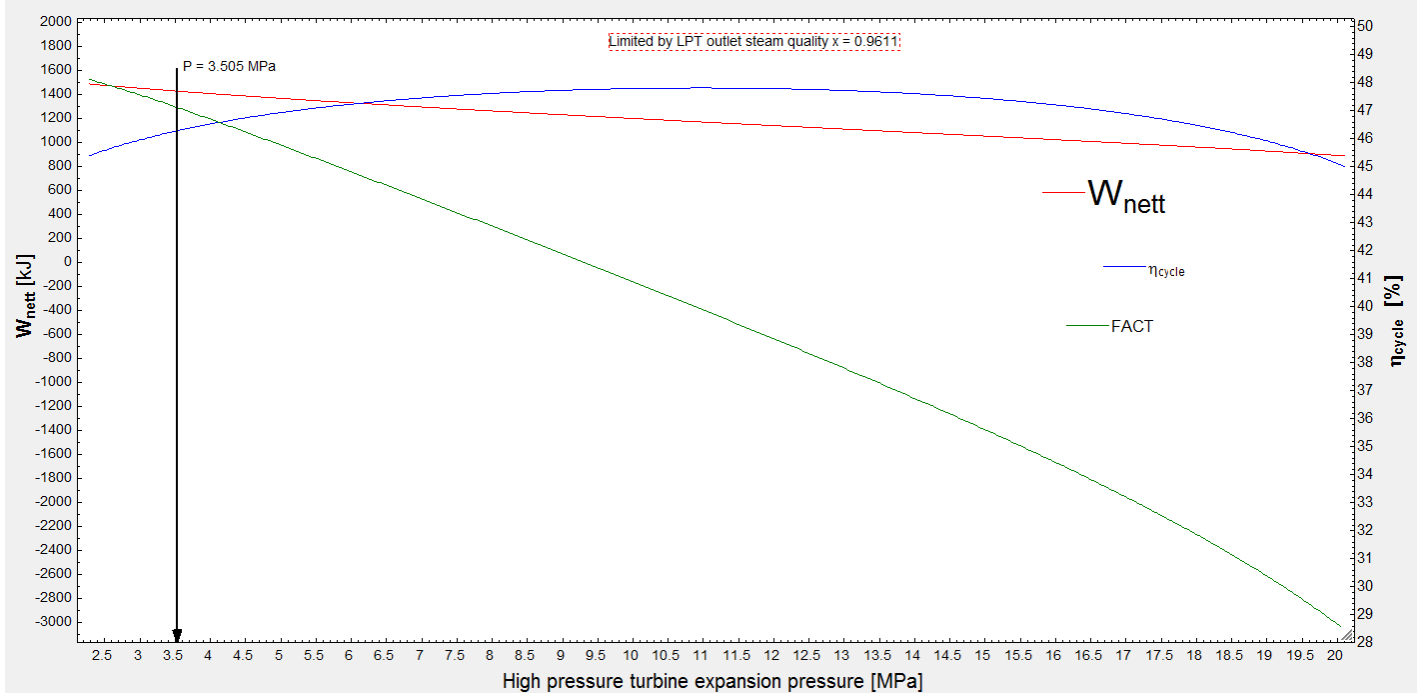


Figure 104: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (1st method, 2nd run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	4.66 MPa	High pressure turbine expansion pressure	3.505 MPa
Cycle efficiency	46.79 %	Cycle efficiency	46.25 %
Mega Watt output	875.3 MW	Mega Watt output	905.6 MW
Net mechanical work	1378 kJ	Net mechanical work	1426 kJ
Heat added	2946 kJ	Heat added	3083 kJ
Heat rejected	1565 kJ	Heat rejected	1656 kJ
LPT outlet steam quality	$X = 0.9446$	LPT outlet steam quality	$X = 0.9611$

Table 67: Before and after optimisation results for the high pressure turbine expansion pressure (1st method, 2nd run)

6.2.2.2.3. High pressure heater 6

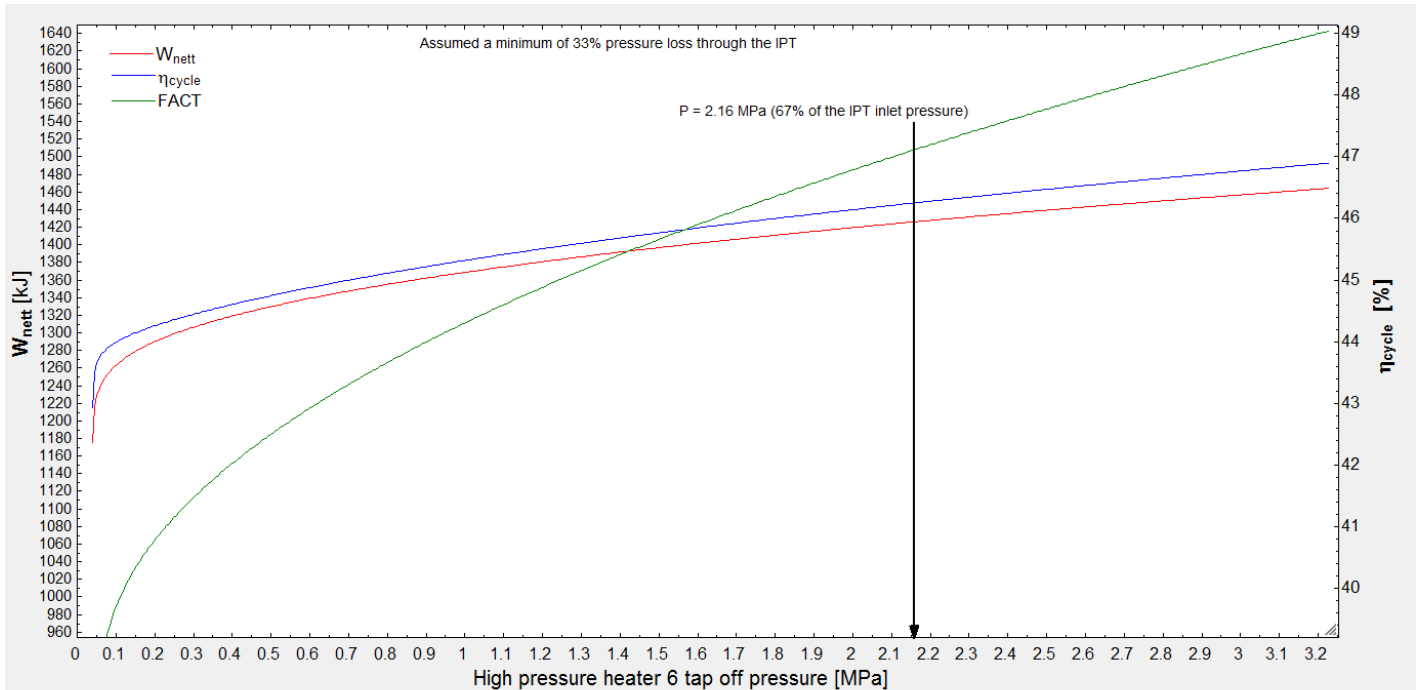


Figure 105: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (1st method, 1st run)

Before optimisation		After optimisation	
High pressure heater 6 steam bled off pressure	2.161 MPa	High pressure heater 6 steam bled off pressure	2.16 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3083 kJ	Heat added	3083 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 68: Before and after optimisation results for high pressure heater 6 steam tap off pressure (1st method, 2nd run)

6.2.2.2.4. De-aerator

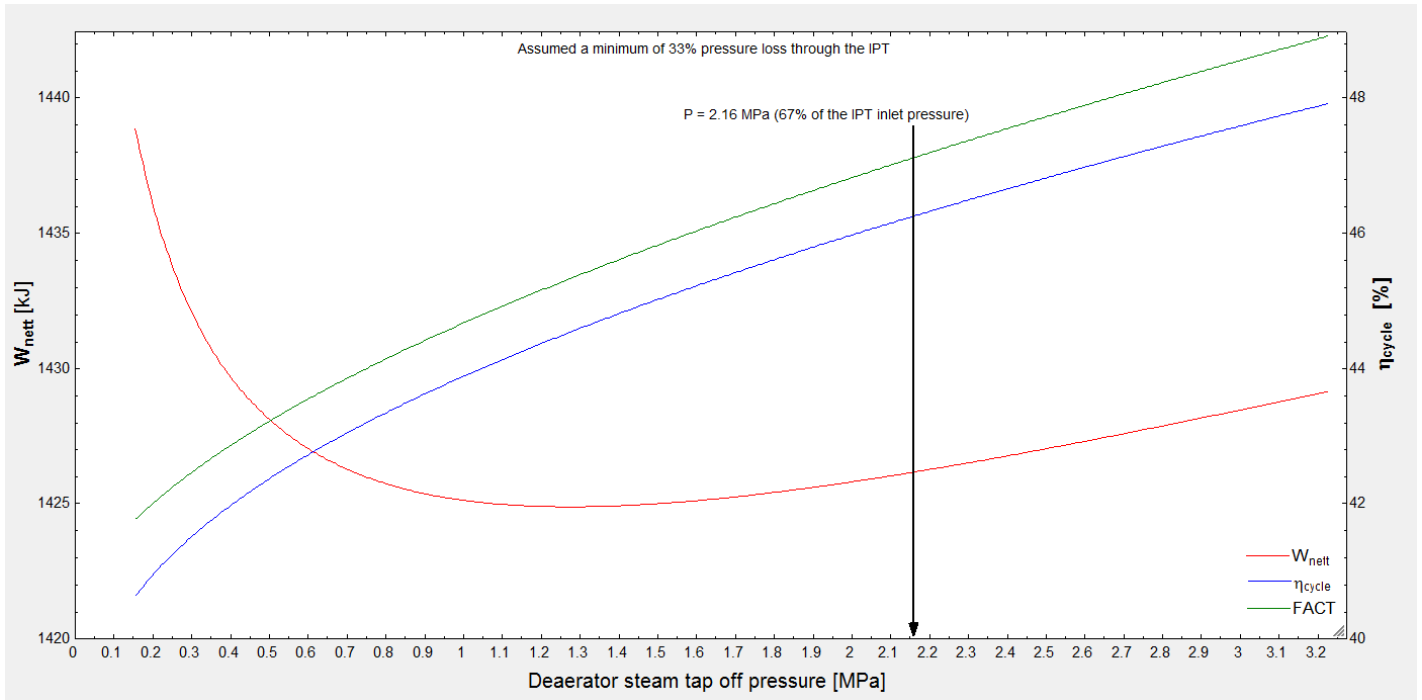


Figure 106: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
De-aerator steam bled off pressure	2.161 MPa	De-aerator steam bled off pressure	2.161 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3083 kJ	Heat added	3084 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 69: Before and after optimisation results for the de-aerator steam tap off pressure (1st method, 2nd run)

6.2.2.2.5. Low pressure heater 3

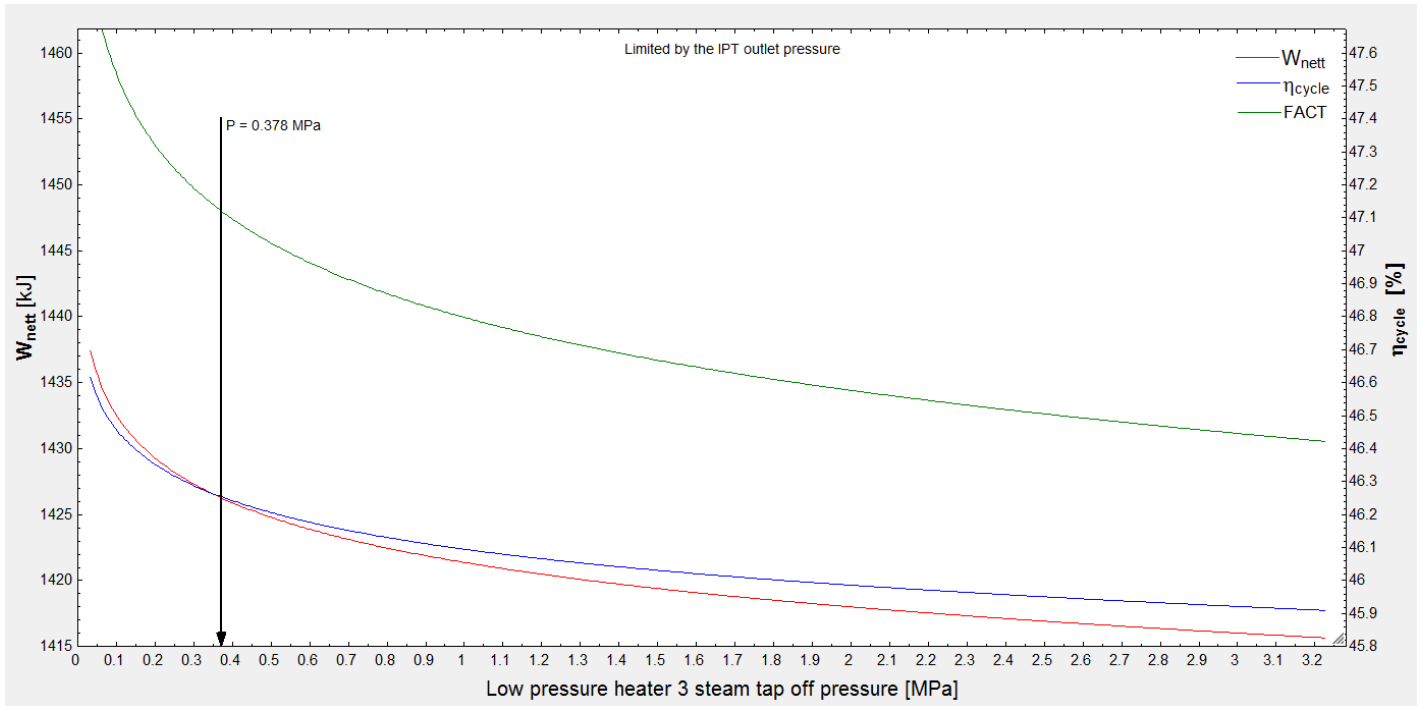


Figure 107: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.378 MPa	Low pressure heater 3 steam tap off pressure	0.378 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3084 kJ	Heat added	3084 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 70: Before and after optimisation results for low pressure heater 3 steam tap off pressure (1st method, 2nd run)

6.2.2.2.6. Low pressure heater 2

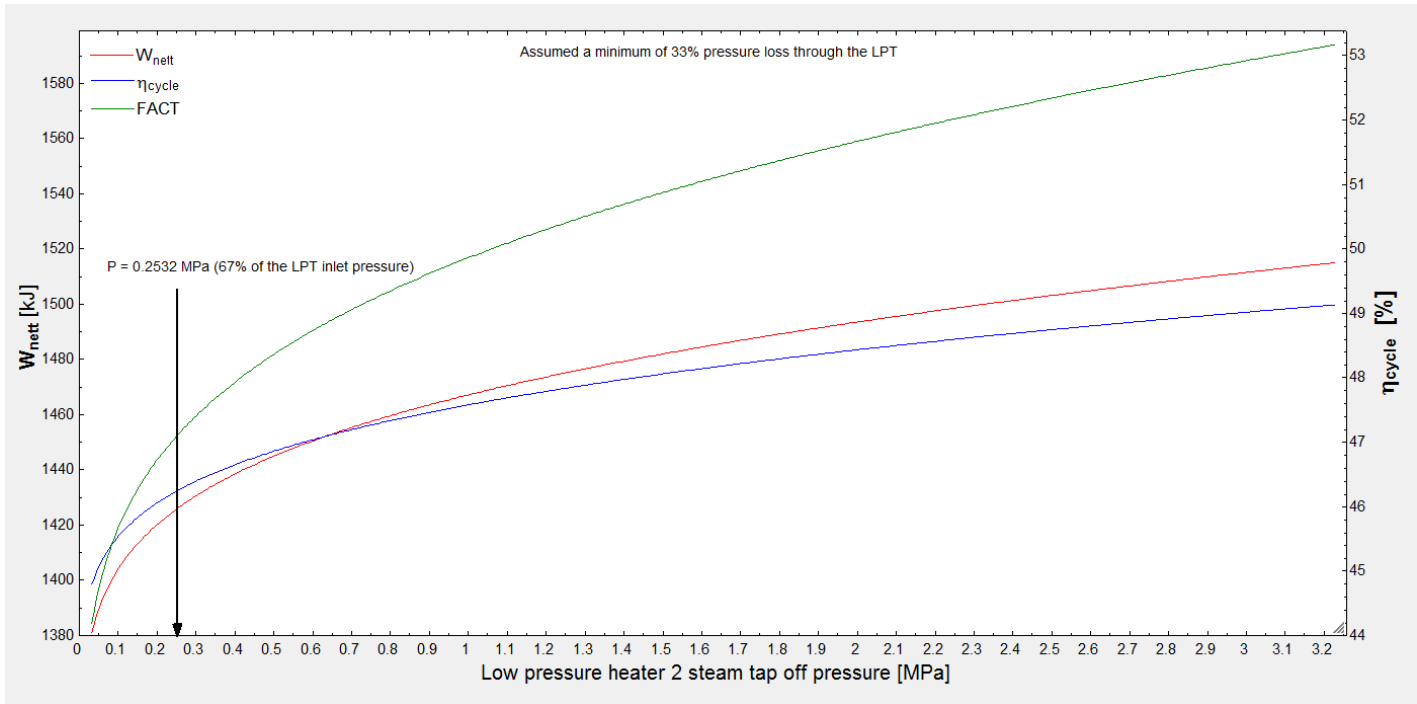


Figure 108: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.2532 MPa	Low pressure heater 2 steam tap off pressure	0.378 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3084 kJ	Heat added	3084 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 71: Before and after optimisation results for low pressure heater 2 steam tap off pressure (1st method, 2nd run)

6.2.2.2.7. Low pressure heater 1

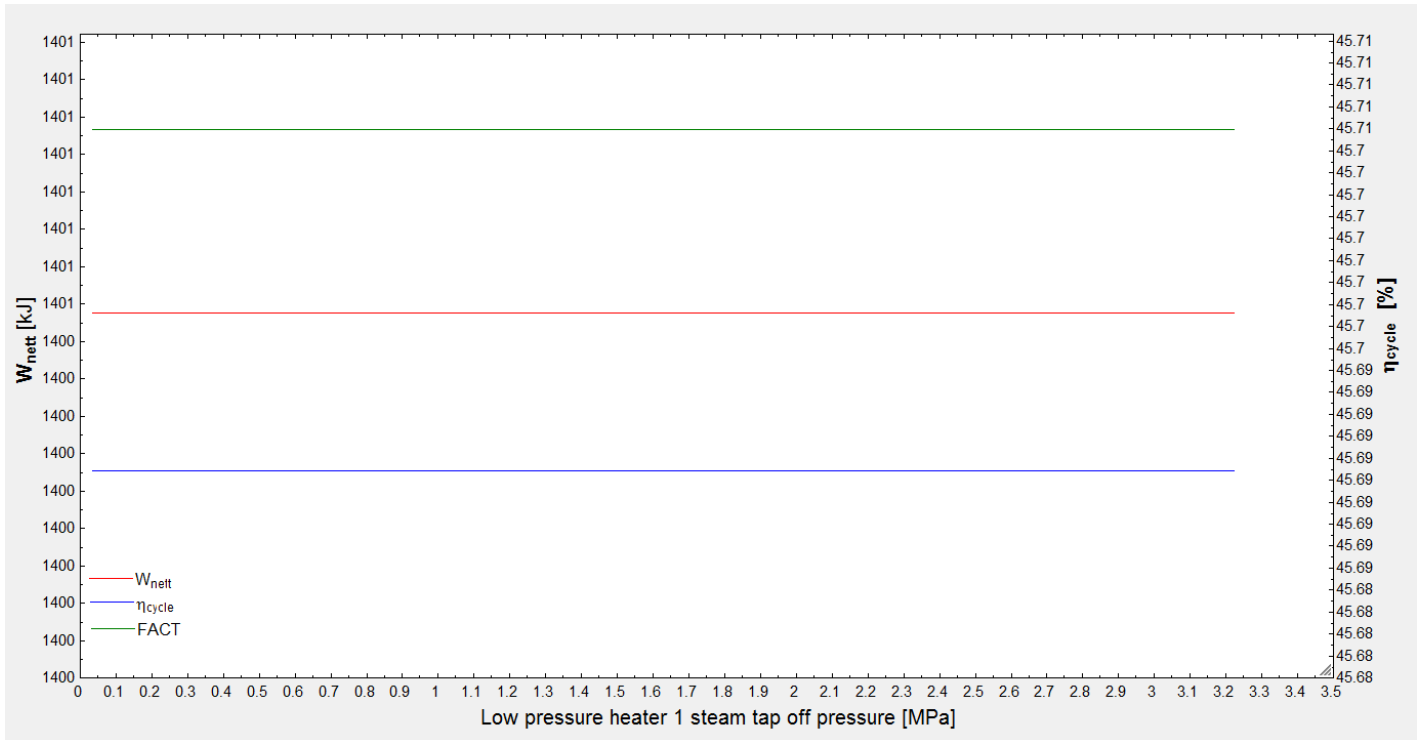


Figure 109: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.0579 MPa	Low pressure heater 1 steam tap off pressure	0.0579 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3084 kJ	Heat added	3084 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 72: Before and after optimisation results for low pressure heater 4 steam tap off pressure (1st method, 2nd run)

6.2.2.3. Third run

6.2.2.3.1. Boiler feed pump pressure

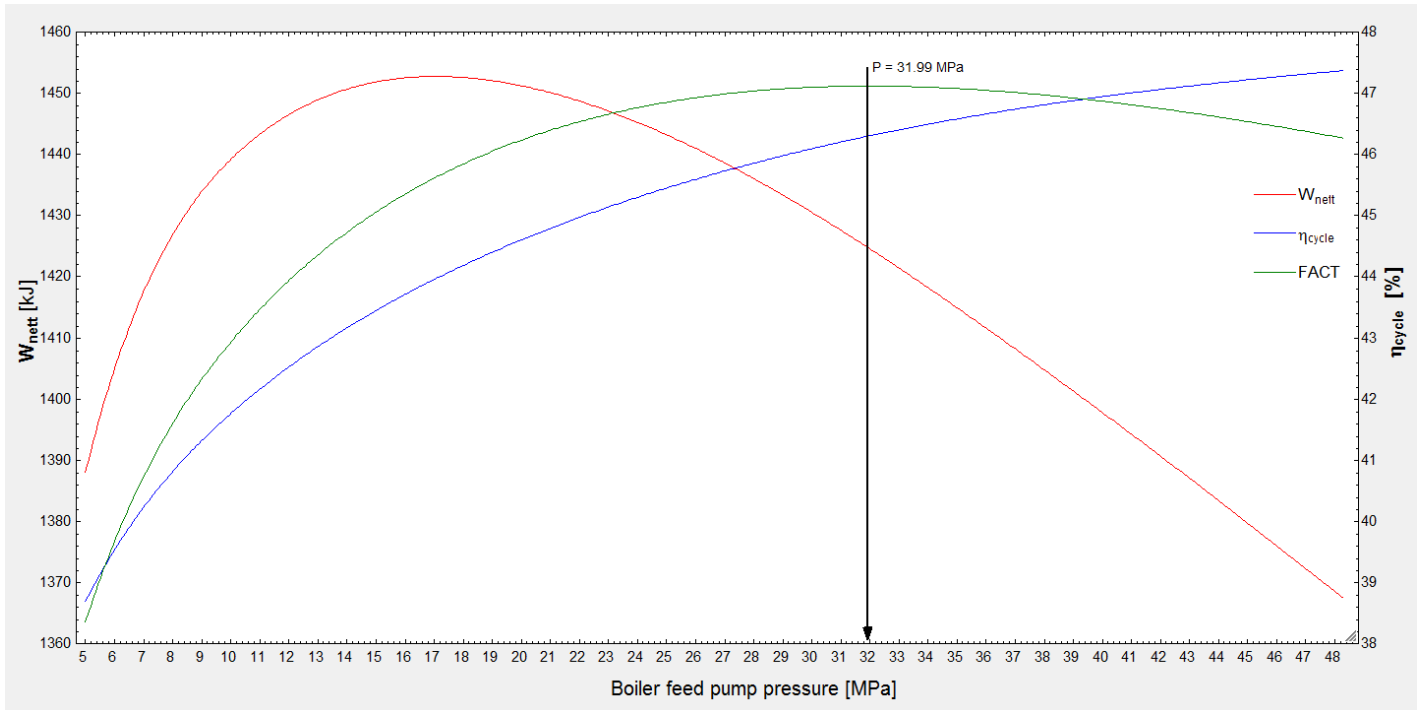


Figure 110: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (1st method, 3rd run)

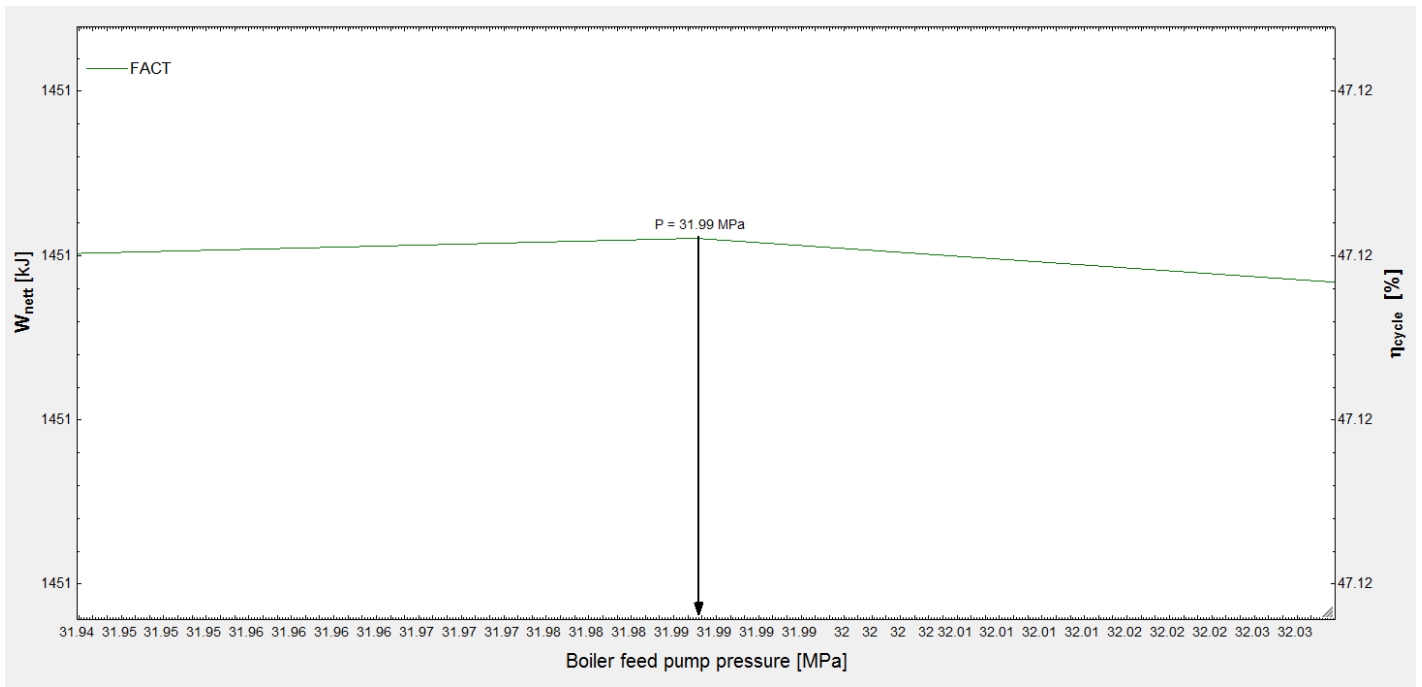


Figure 111: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (1st method, 3rd run)

Before optimisation	
Boiler feed pump pressure	31.47 MPa
Cycle efficiency	46.25 %
Mega Watt output	905.6 MW
Net mechanical work	1426 kJ
Heat added	3084 kJ
Heat rejected	1656 kJ

After optimisation	
Boiler feed pump pressure	31.99 MPa
Cycle efficiency	46.3 %
Mega Watt output	904.6 MW
Net mechanical work	1425 kJ
Heat added	3077 kJ
Heat rejected	1651 kJ

Table 73: Before and after optimisation results for the boiler feed pump discharge (1st method, 3rd run)

6.2.2.3.2. High pressure turbine expansion

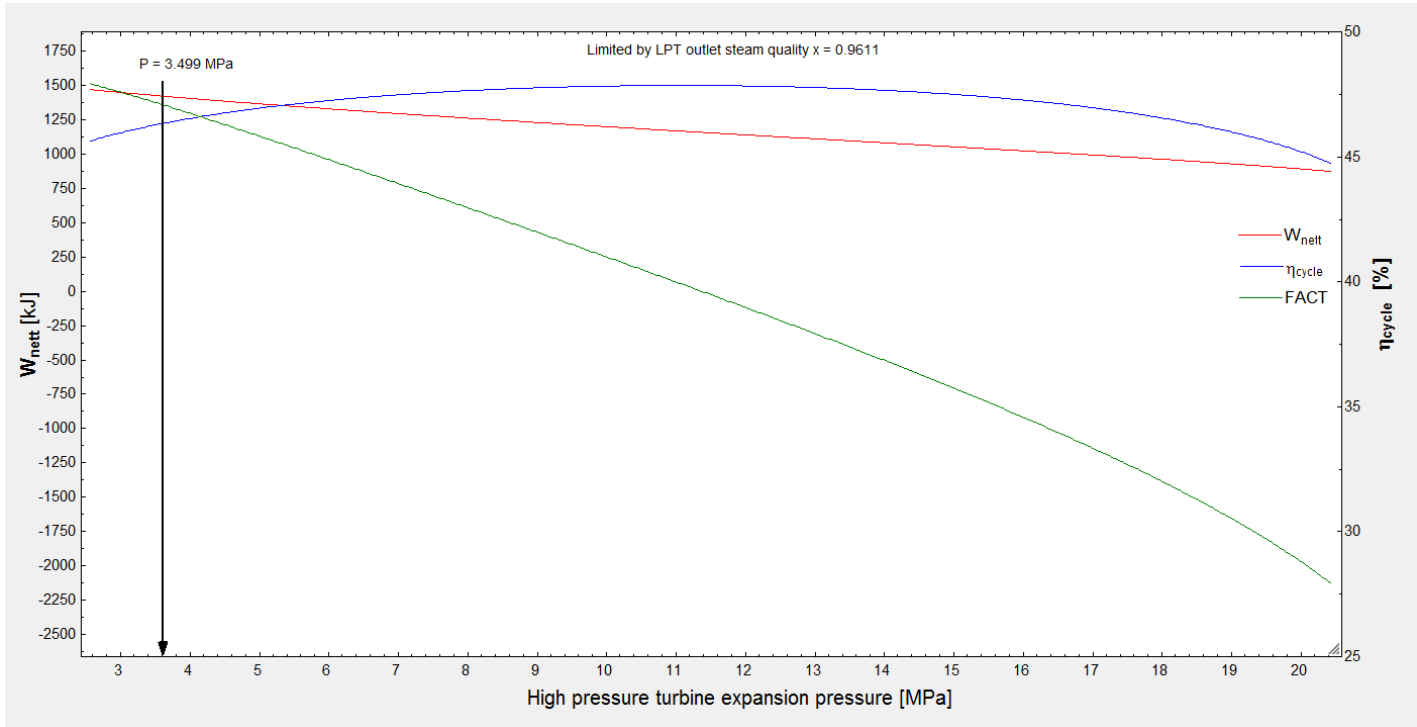


Figure 112: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (1st method, 3rd run)

Before optimisation	
High pressure turbine expansion pressure	3.56 MPa
Cycle efficiency	46.3 %
Mega Watt output	904.6 MW
Net mechanical work	1425 kJ
Heat added	3077 kJ
Heat rejected	1651 kJ
LPT outlet steam quality	X = 0.9601

After optimisation	
High pressure turbine expansion pressure	3.499 MPa
Cycle efficiency	46.27 %
Mega Watt output	906.3 MW
Net mechanical work	1425 kJ
Heat added	3077 kJ
Heat rejected	1651 kJ
LPT outlet steam quality	X = 0.9611

Table 74: Before and after optimisation results for the high pressure turbine expansion pressure (1st method, 3rd run)

6.2.2.3.3. High pressure heater 6

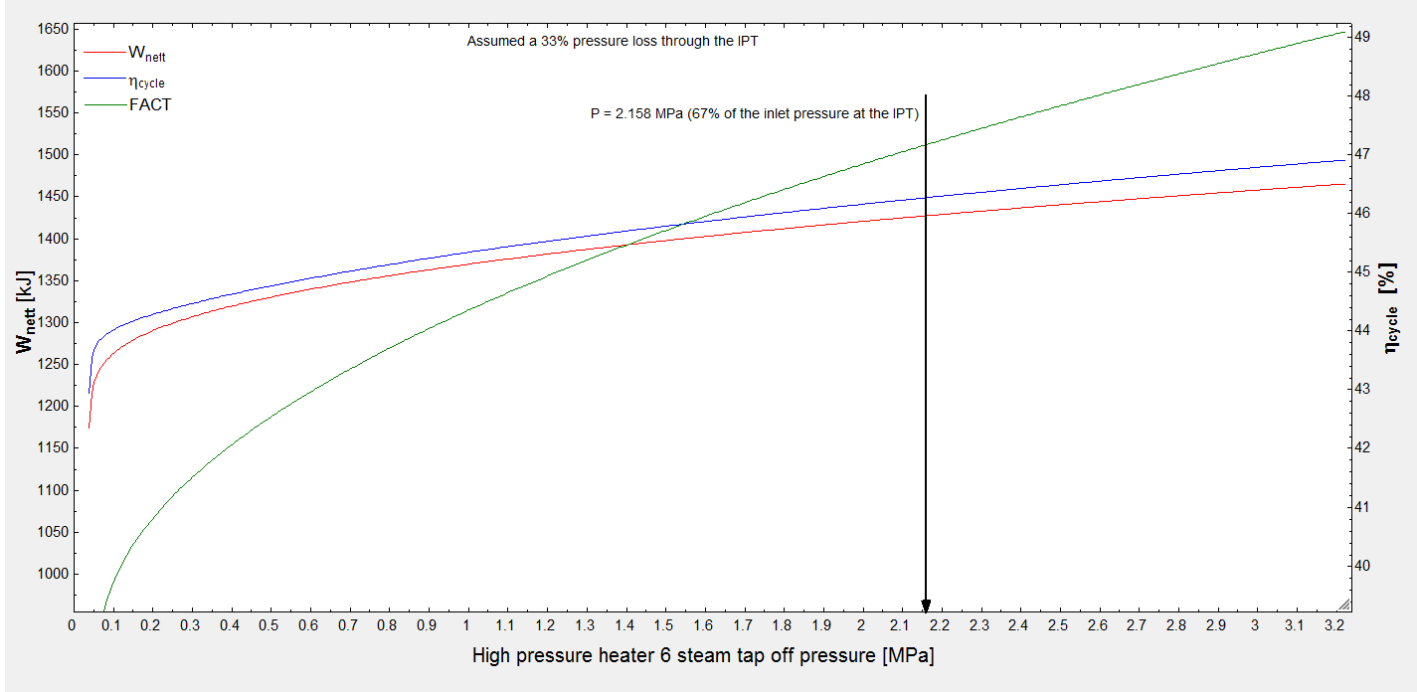


Figure 113: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
High pressure heater 6 steam tap off pressure	2.158 MPa	High pressure heater 6 steam tap off pressure	2.158 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.3 MW	Mega Watt output	906.3 MW
Net mechanical work	1425 kJ	Net mechanical work	1427 kJ
Heat added	3077 kJ	Heat added	3085 kJ
Heat rejected	1651 kJ	Heat rejected	1657 kJ

Table 75: Before and after optimisation results for high pressure heater 6 steam tap off pressure (1st method, 3rd run)

6.2.2.3.4. De-aerator

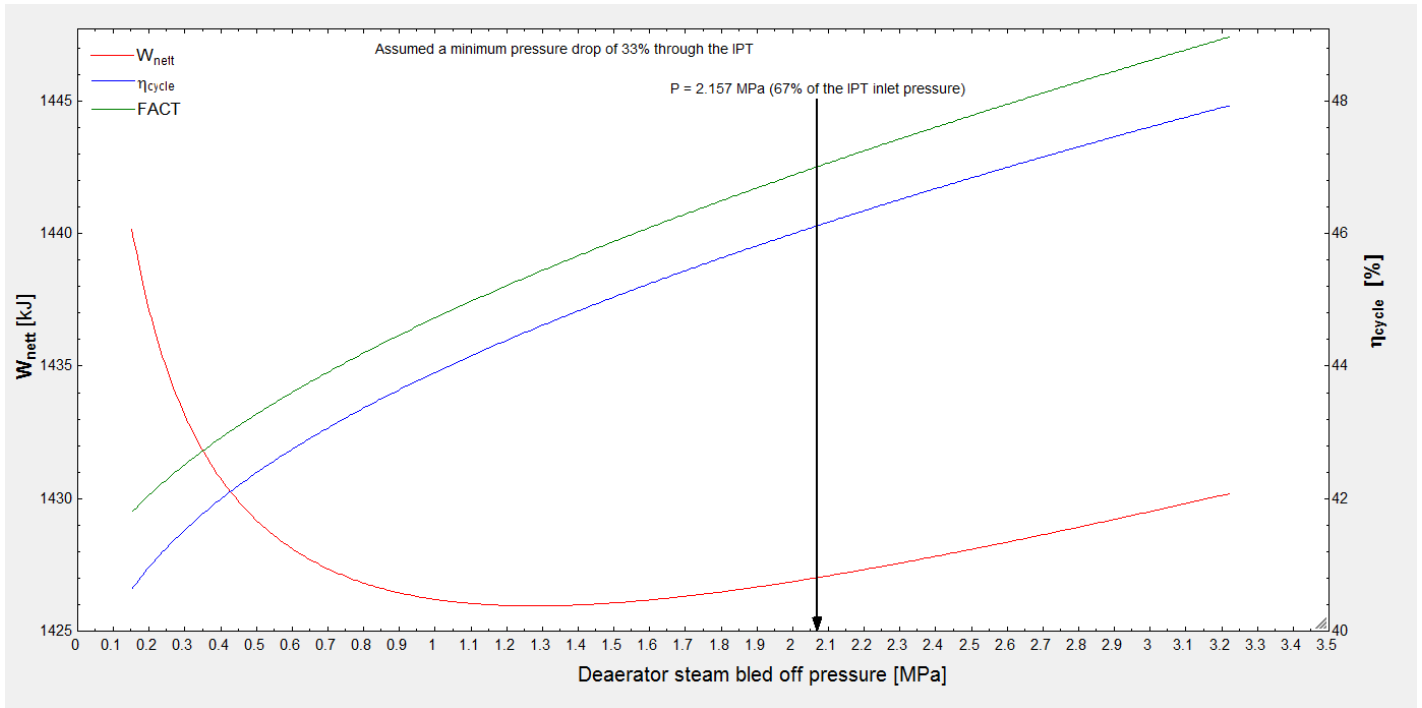


Figure 114: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
De-aerator steam tap off pressure	2.157 MPa	De-aerator steam tap off pressure	2.157 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.3 MW	Mega Watt output	906.3 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1657 kJ	Heat rejected	1657 kJ

Table 76: Before and after optimisation results for the de-aerator steam tap off pressure (1st method, 3rd run)

6.2.2.3.5. Low pressure heater 3

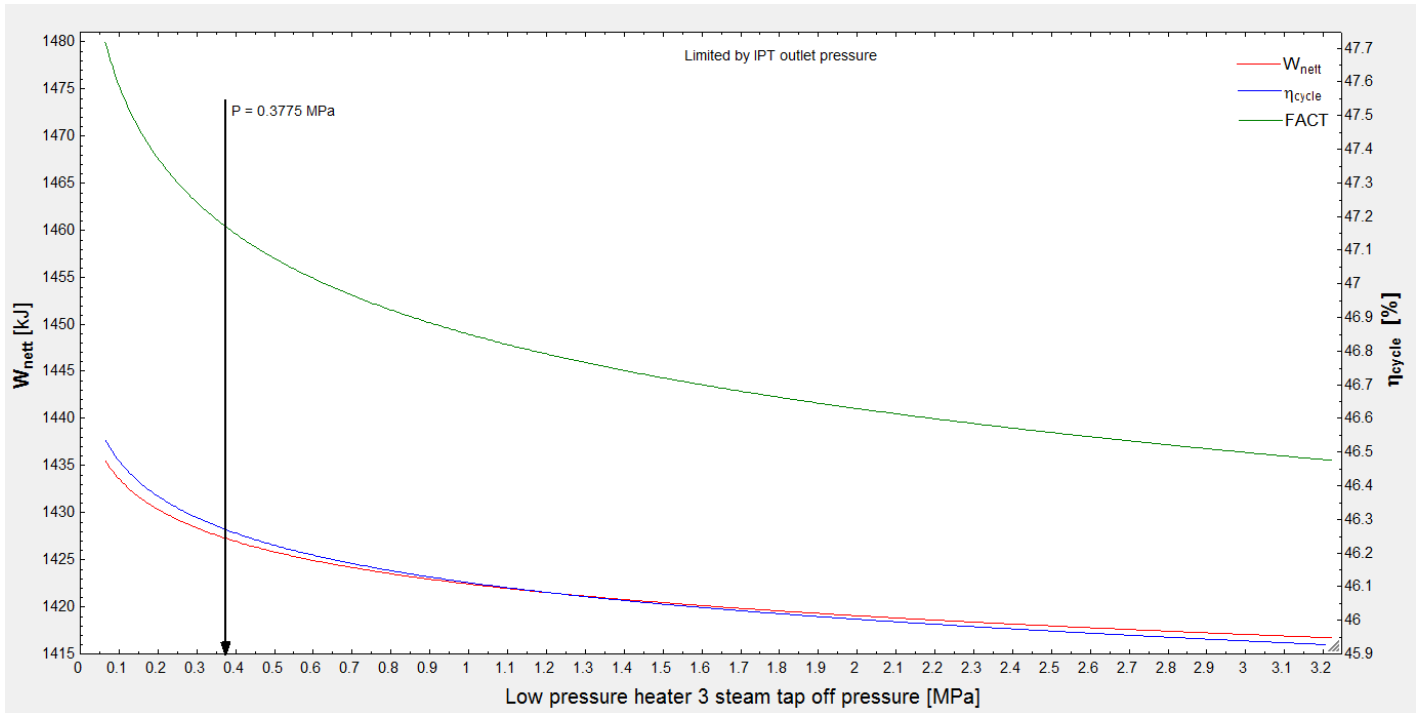


Figure 115: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.3775 MPa	Low pressure heater 3 steam tap off pressure	0.3775 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.3 MW	Mega Watt output	906.3 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1657 kJ	Heat rejected	1657 kJ

Table 77: Before and after optimisation results for low pressure heater 3 steam tap off pressure (1st method, 3rd run)

6.2.2.3.6. Low pressure heater 2

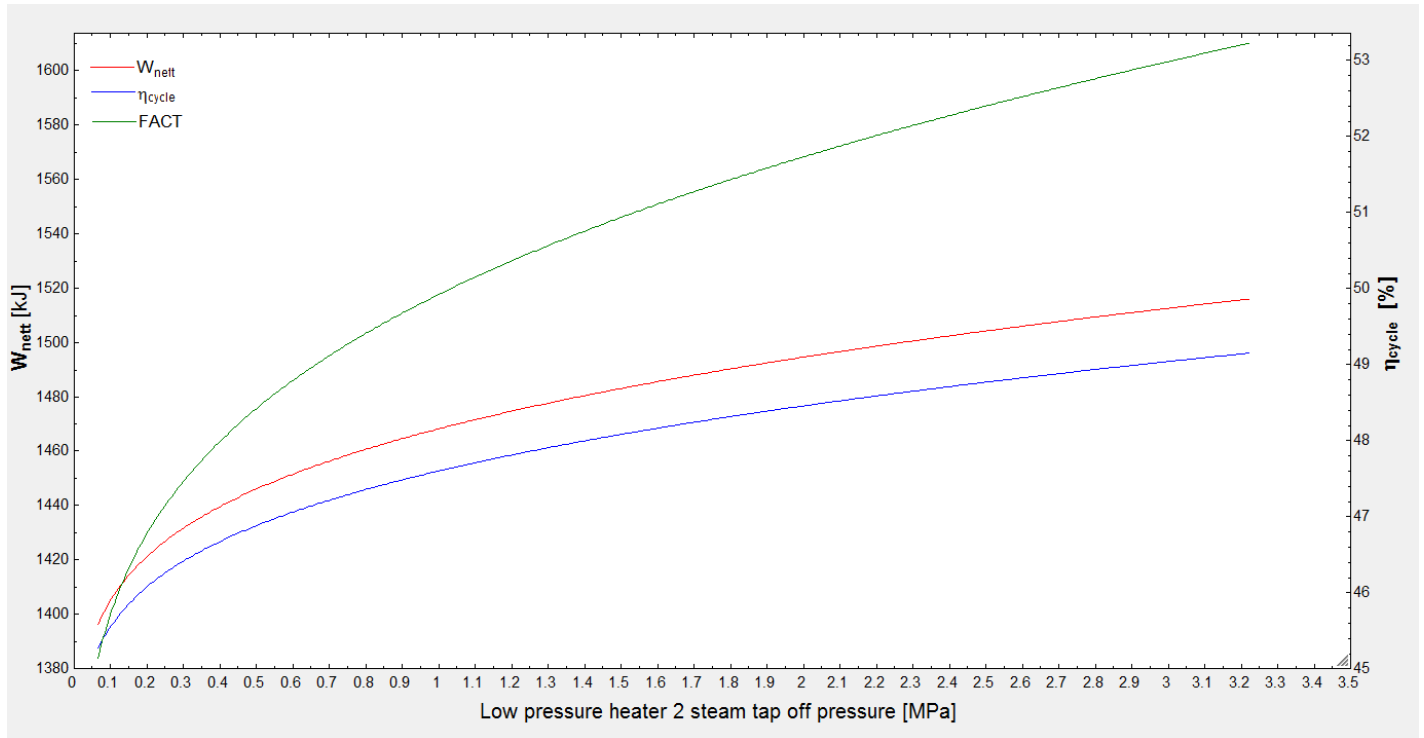


Figure 116: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.2528 MPa	Low pressure heater 2 steam tap off pressure	0.2529 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.3 MW	Mega Watt output	906.3 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1657 kJ	Heat rejected	1657 kJ

Table 78: Before and after optimisation results for low pressure heater 2 steam tap off pressure (1st method, 3rd run)

6.2.2.3.7. Low pressure heater 1

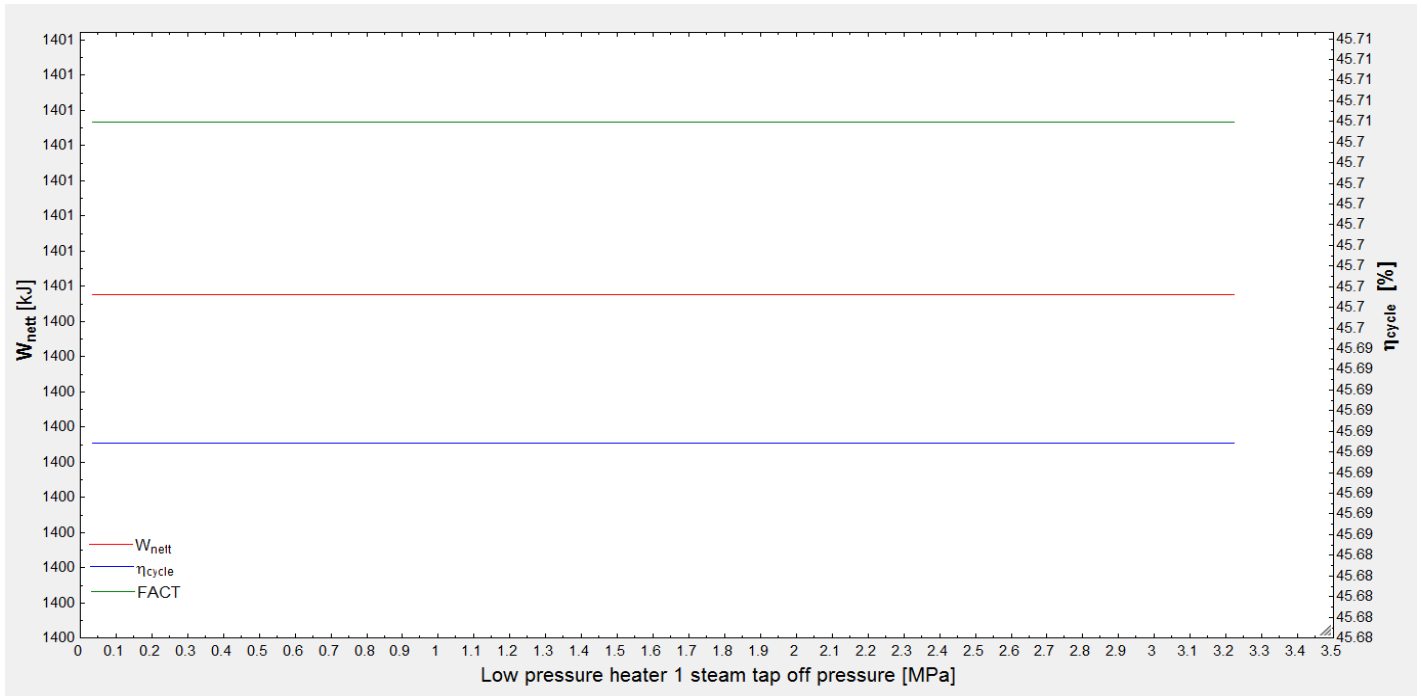


Figure 117: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (1st method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.05781 MPa	Low pressure heater 1 steam tap off pressure	0.05781 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.3 MW	Mega Watt output	906.3 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1657 kJ	Heat rejected	1657 kJ

Table 79: Before and after optimisation results for low pressure heater 1 steam tap off pressure (1st method, 3rd run)

6.2.2.3.8. Boiler feed pump

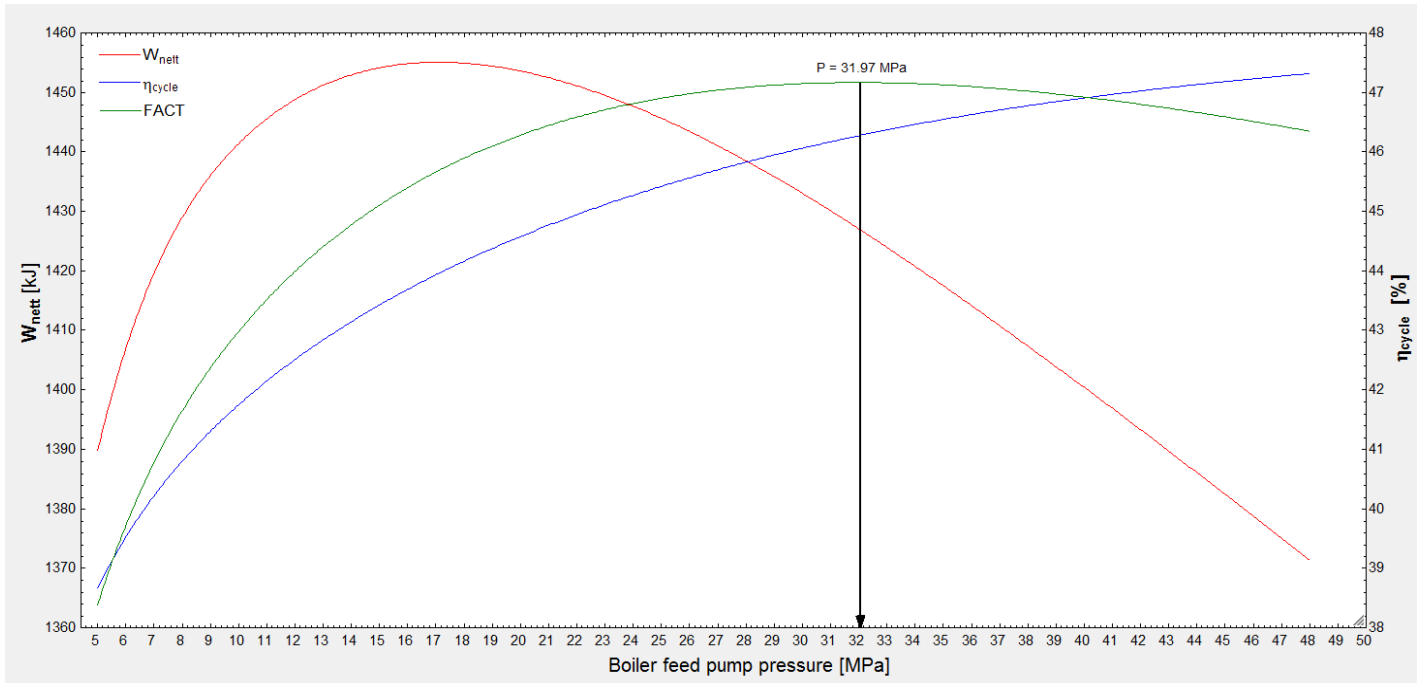


Figure 118: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (1st method, 4th run)

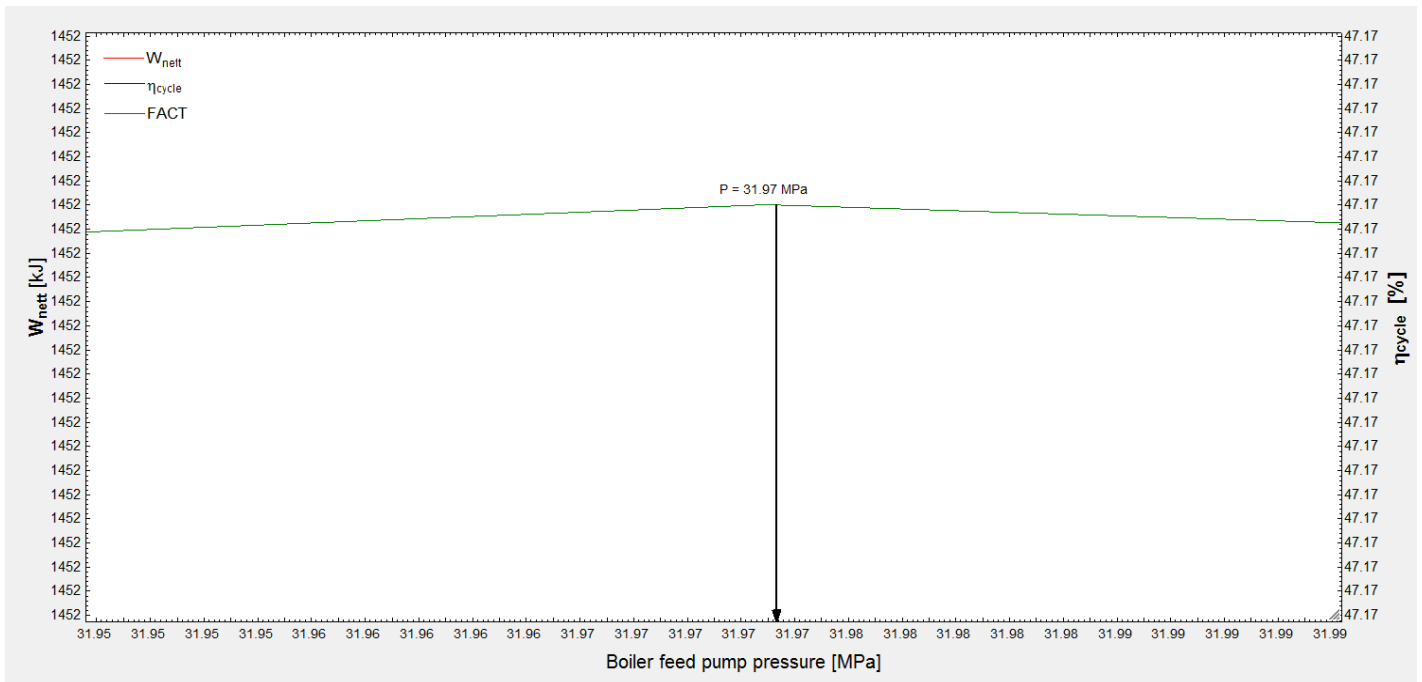


Figure 119: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (Zoomed in) (1st method, 4th run)

Before optimisation		After optimisation	
Boiler feed pump pressure	31.99 MPa	Boiler feed pump pressure	31.97 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.3 MW	Mega Watt output	906.3 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1657 kJ	Heat rejected	1657 kJ

Table 80: Before and after optimisation results for the boiler feed pump discharge (1st method, 4th run)

6.2.2.3.9. High pressure turbine expansion

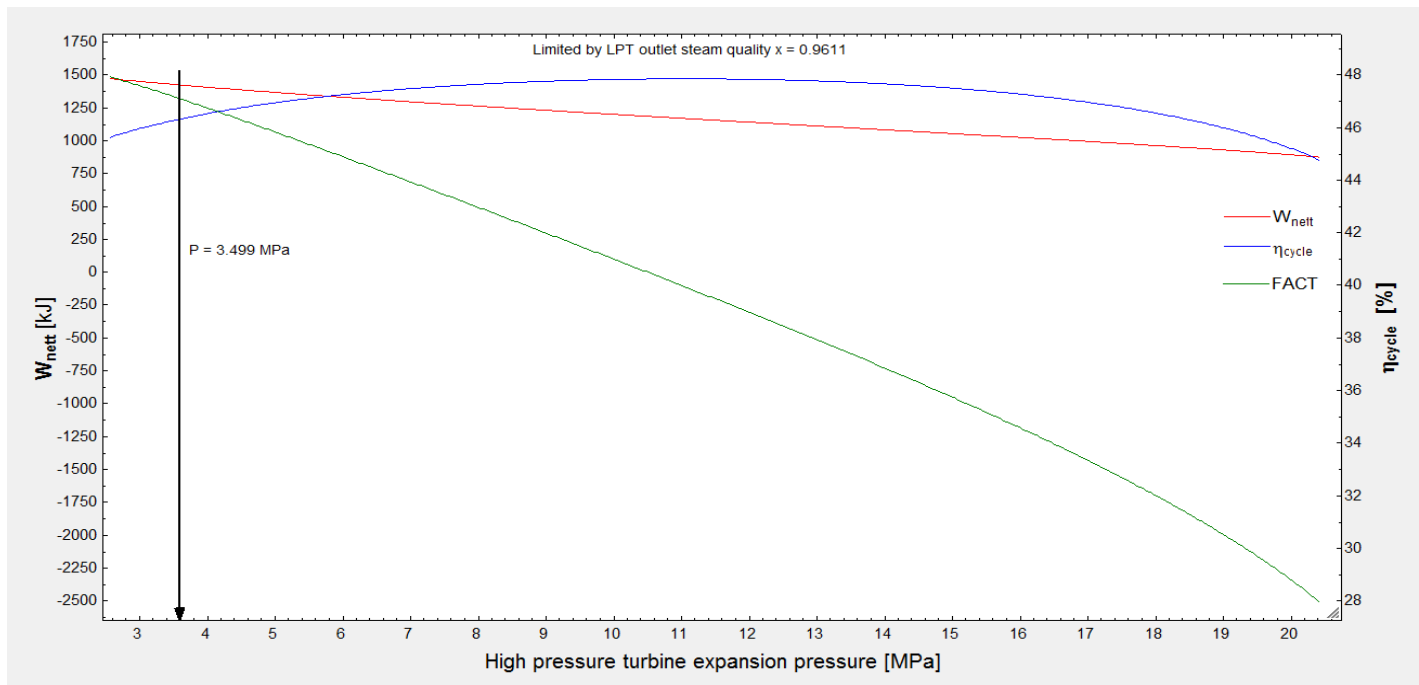


Figure 120: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (1st method, 4th run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.497 MPa	High pressure turbine expansion pressure	3.499 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.3 MW	Mega Watt output	906.3 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1657 kJ	Heat rejected	1656 kJ
LPT outlet steam quality	X = 0.9612	LPT outlet steam quality	X = 0.9611

Table 81: Before and after optimisation results for the high pressure turbine expansion pressure (1st method, 4th run)

6.2.3. Super-critical cycle optimisation – second method

6.2.3.1. First run

6.2.3.1.1. Boiler feed pump pressure

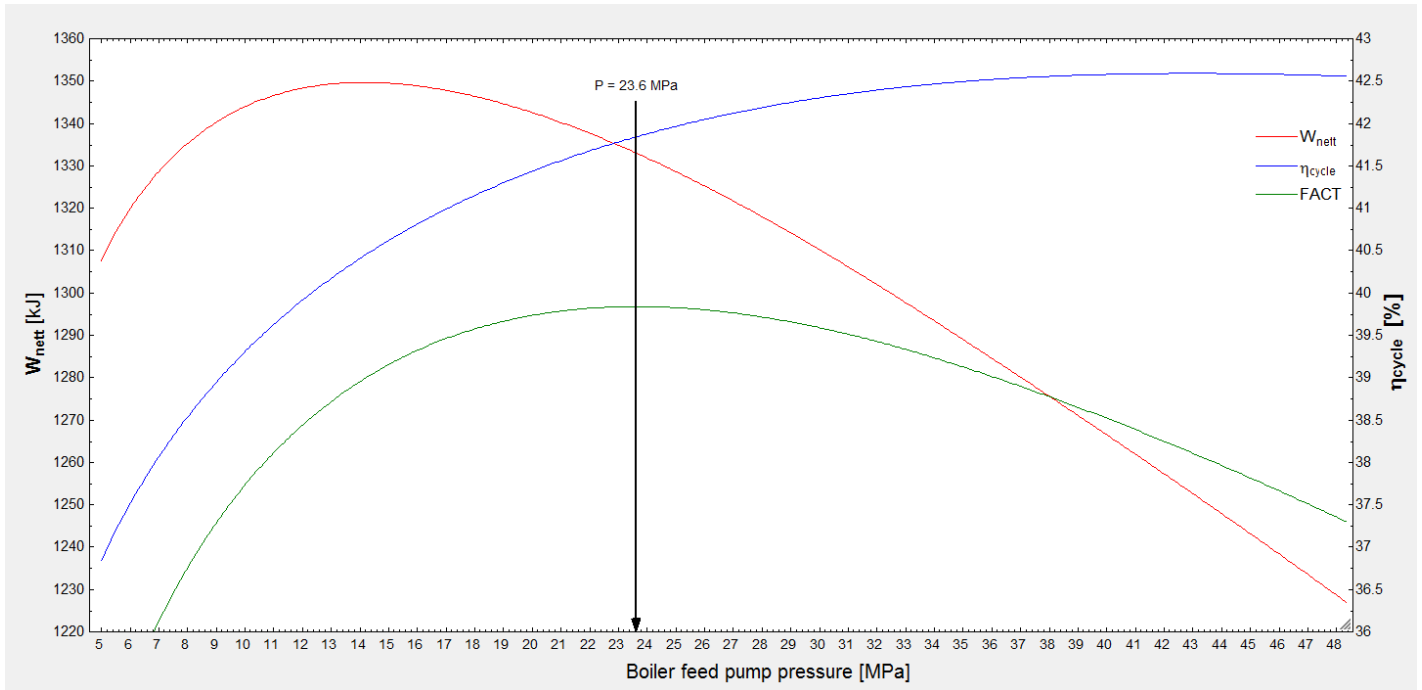


Figure 121: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (2nd method, 1st run)

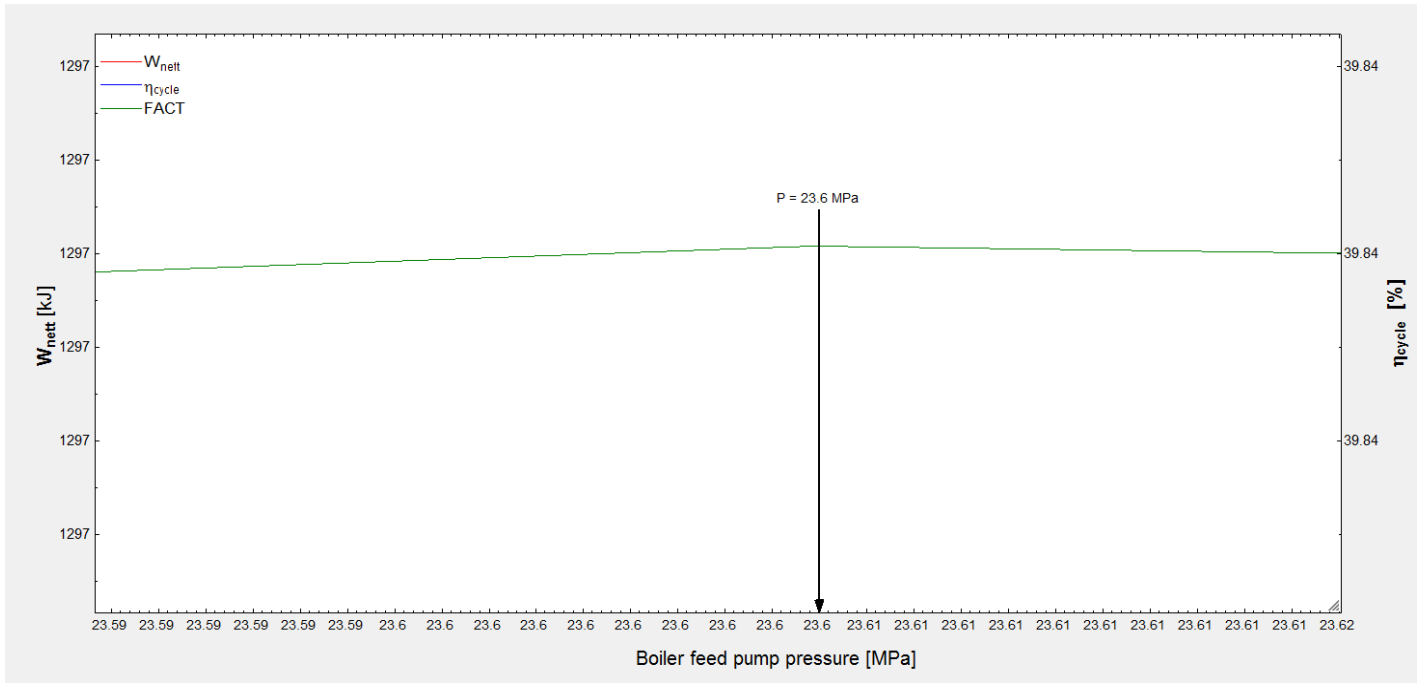


Figure 122: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (2nd method, 1st run)

Before optimisation		After optimisation	
Boiler feed pump pressure	30.13 MPa	Boiler feed pump pressure	23.6 MPa
Cycle efficiency	42.31%	Cycle efficiency	41.84 %
Mega Watt output	831.6 MW	Mega Watt output	846.5 MW
Net mechanical work	1310 kJ	Net mechanical work	1333 kJ
Heat added	3096 kJ	Heat added	3186 kJ
Heat rejected	1495 kJ	Heat rejected	1575 kJ

Table 82: Before and after optimisation results for the boiler feed pump discharge pressure (2nd method, 1st run)

6.2.3.1.2. High pressure heater 6

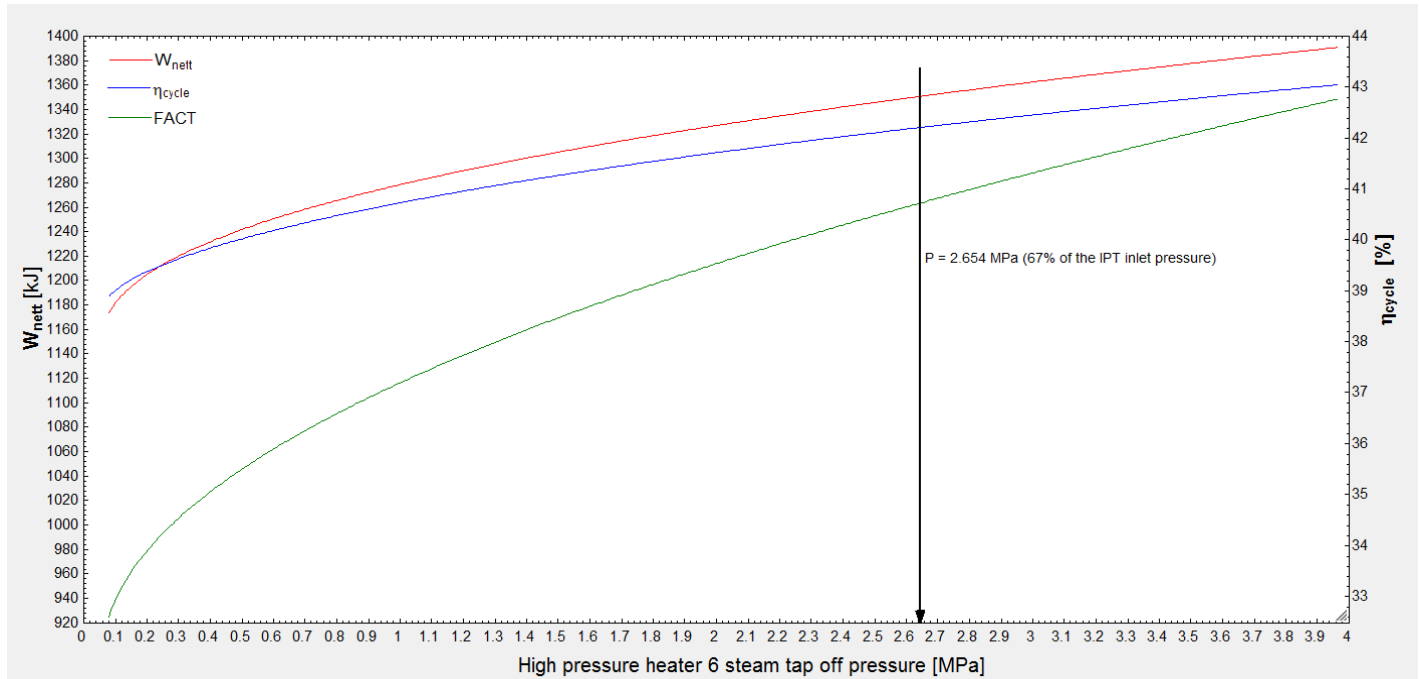


Figure 123: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (1st method, 1st run)

Before optimisation		After optimisation	
High pressure heater 6 steam tap off pressure	2.157 MPa	High pressure heater 6 steam tap off pressure	2.654 MPa
Cycle efficiency	41.84 %	Cycle efficiency	42.21 %
Mega Watt output	846.5 MW	Mega Watt output	857.9 MW
Net mechanical work	1333 kJ	Net mechanical work	1351 kJ
Heat added	3186 kJ	Heat added	3200 kJ
Heat rejected	1575 kJ	Heat rejected	1586 kJ

Table 83: Before and after optimisation results for high pressure heater 6 steam tap off pressure (2nd method, 1st run)

6.2.3.1.3. High pressure turbine expansion

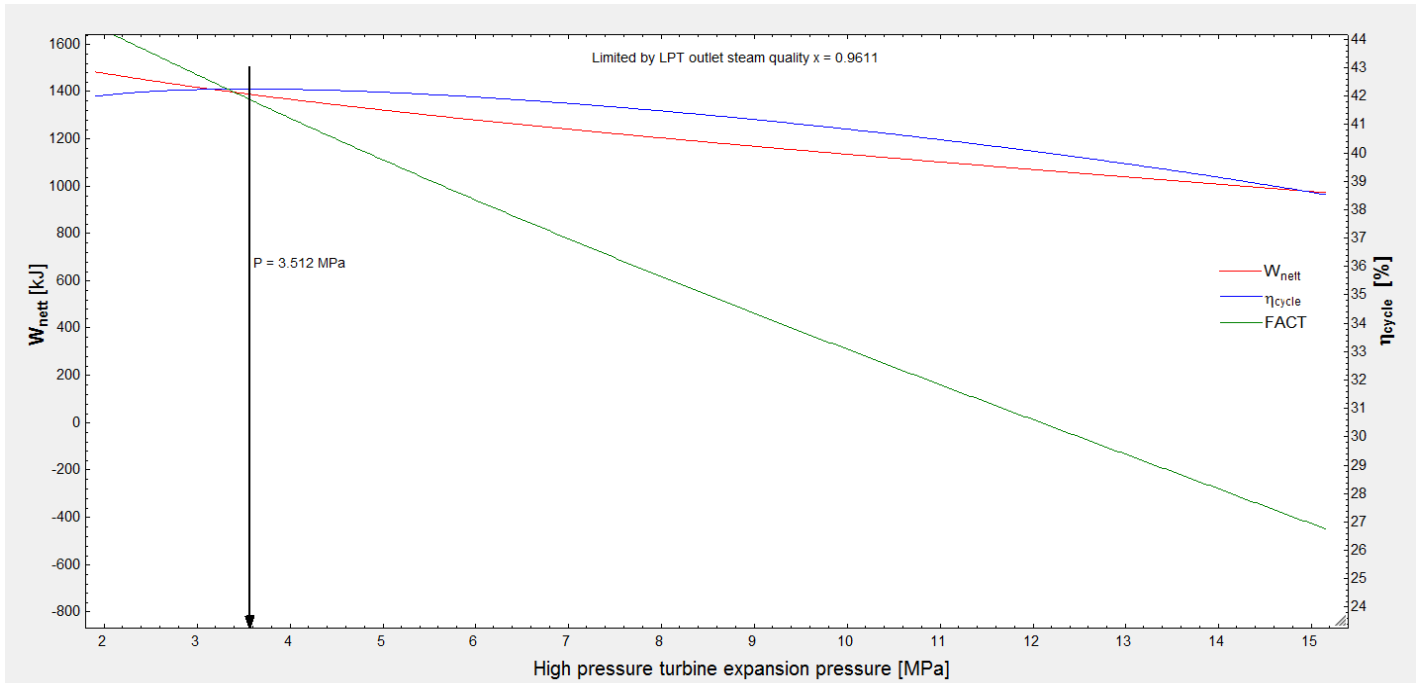


Figure 124: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (2nd method, 1st run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	4.323 MPa	High pressure turbine expansion pressure	3.512 MPa
Cycle efficiency	42.21 %	Cycle efficiency	42.25 %
Mega Watt output	857.9 MW	Mega Watt output	882.8 MW
Net mechanical work	1351 kJ	Net mechanical work	1390 kJ
Heat added	3200 kJ	Heat added	3291 kJ
Heat rejected	1586 kJ	Heat rejected	1651 kJ
LPT outlet steam quality	$X = 0.9491$	LPT outlet steam quality	$X = 0.9611$

Table 84: Before and after optimisation results for the high pressure turbine expansion pressure (2nd method, 1st run)

6.2.3.1.4. Low pressure heater 1

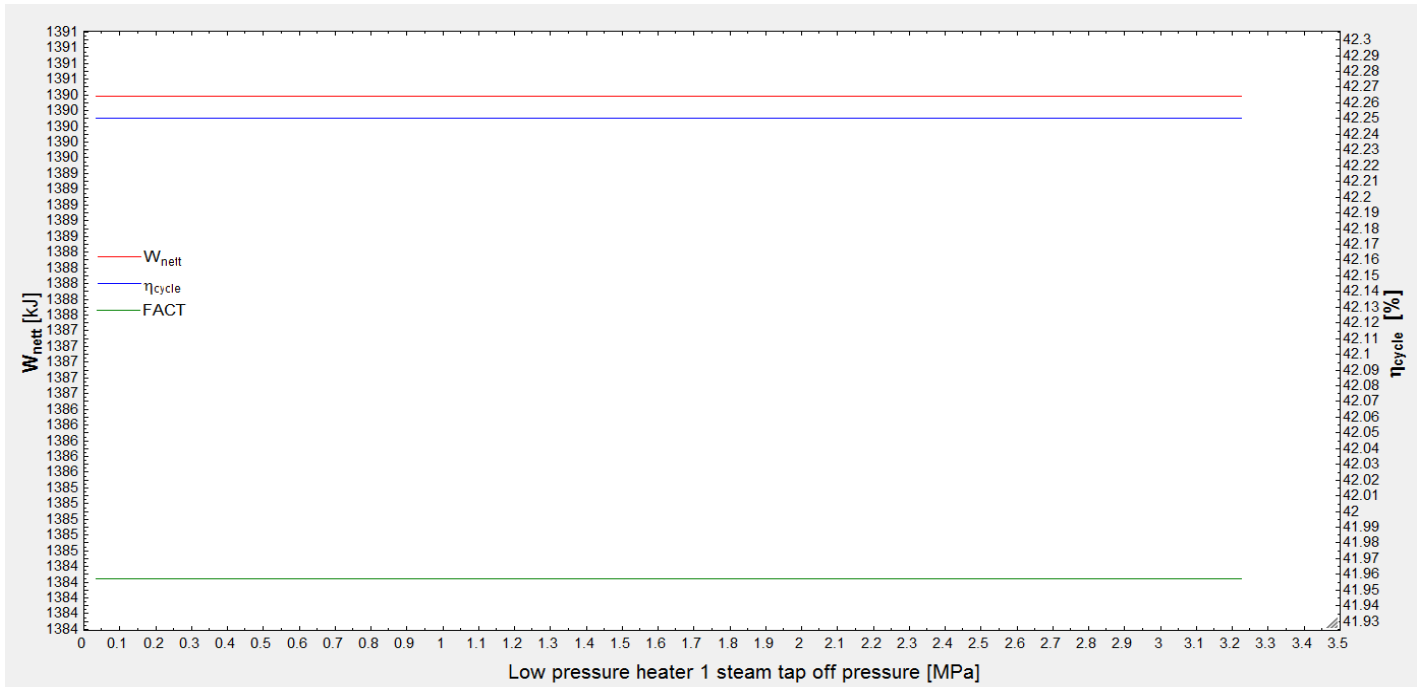


Figure 125: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (2nd method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.05787 MPa	Low pressure heater 1 steam tap off pressure	0.05787 MPa
Cycle efficiency	42.25 %	Cycle efficiency	42.25 %
Mega Watt output	882.8 MW	Mega Watt output	882.8 MW
Net mechanical work	1390 kJ	Net mechanical work	1390 kJ
Heat added	3291 kJ	Heat added	3291 kJ
Heat rejected	1651 kJ	Heat rejected	1651 kJ

Table 85: Before and after optimisation results for low pressure heater 1 steam tap off pressure (2nd method, 1st run)

6.2.3.1.5. Low pressure heater 2

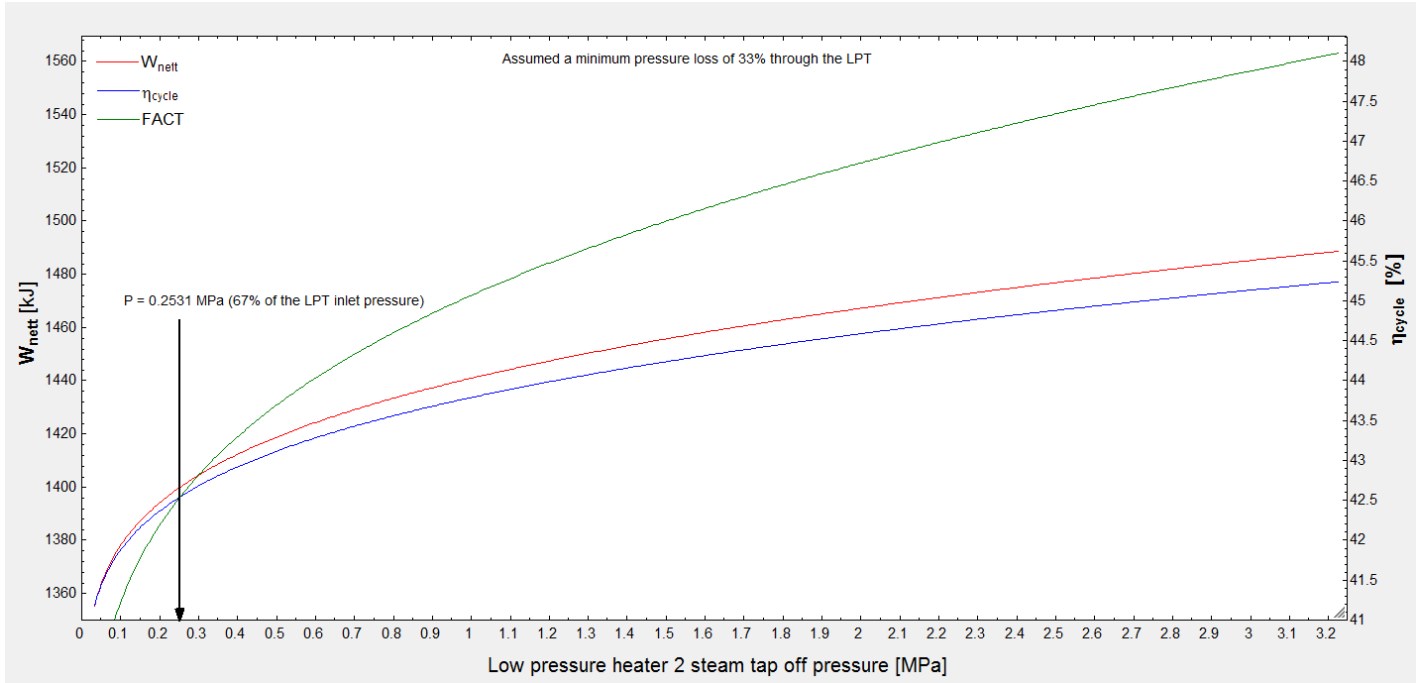


Figure 126: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (2nd method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.1709 MPa	Low pressure heater 2 steam tap off pressure	0.2531 MPa
Cycle efficiency	42.25 %	Cycle efficiency	42.55 %
Mega Watt output	882.8 MW	Mega Watt output	889 MW
Net mechanical work	1390 kJ	Net mechanical work	1390 kJ
Heat added	3291 kJ	Heat added	3291 kJ
Heat rejected	1651 kJ	Heat rejected	1651 kJ

Table 86: Before and after optimisation results for low pressure heater 2 steam tap off pressure (2nd method, 1st run)

6.2.3.1.6. Low pressure heater 3

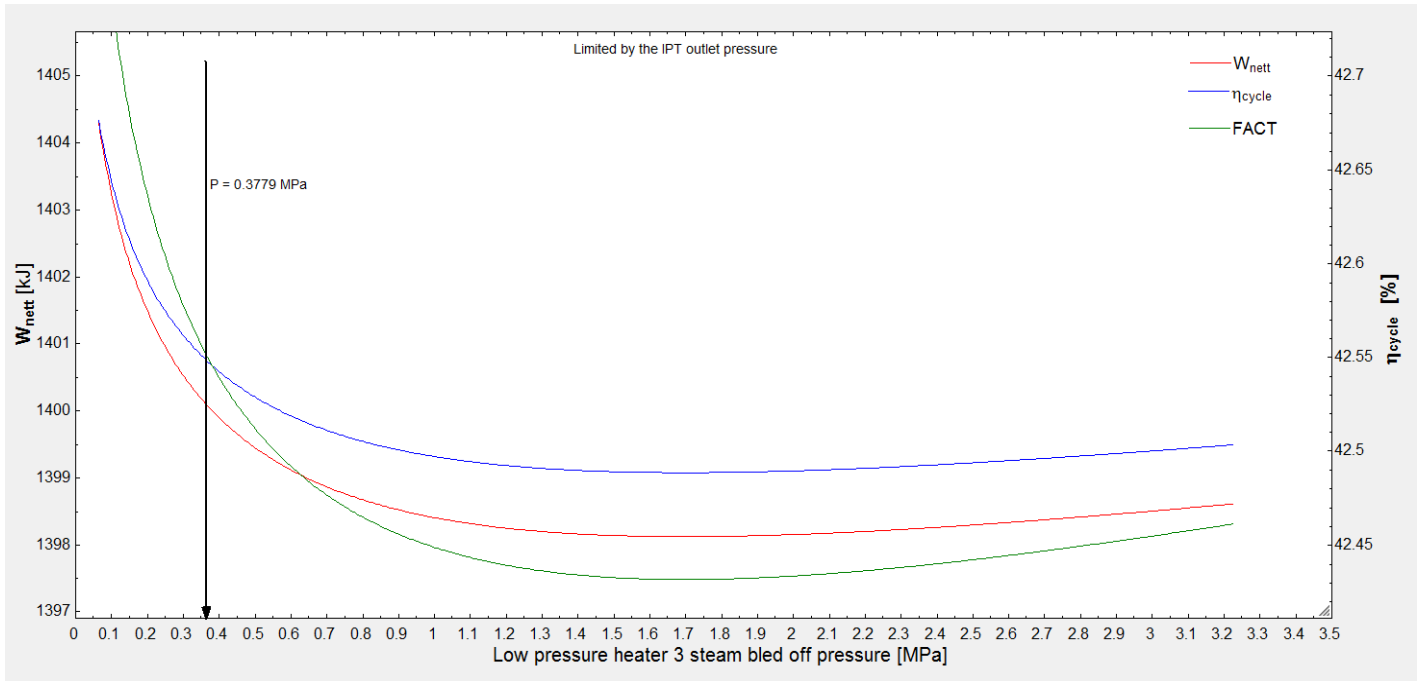


Figure 127: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (1st method, 1st run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.378 MPa	Low pressure heater 3 steam tap off pressure	0.3779 MPa
Cycle efficiency	42.55 %	Cycle efficiency	42.55 %
Mega Watt output	889 MW	Mega Watt output	889 MW
Net mechanical work	1390 kJ	Net mechanical work	1400 kJ
Heat added	3291 kJ	Heat added	3291 kJ
Heat rejected	1651 kJ	Heat rejected	1653 kJ

Table 87: Before and after optimisation results for low pressure heater 3 steam tap off pressure (2nd method, 1st run)

6.2.3.1.7. De-aerator

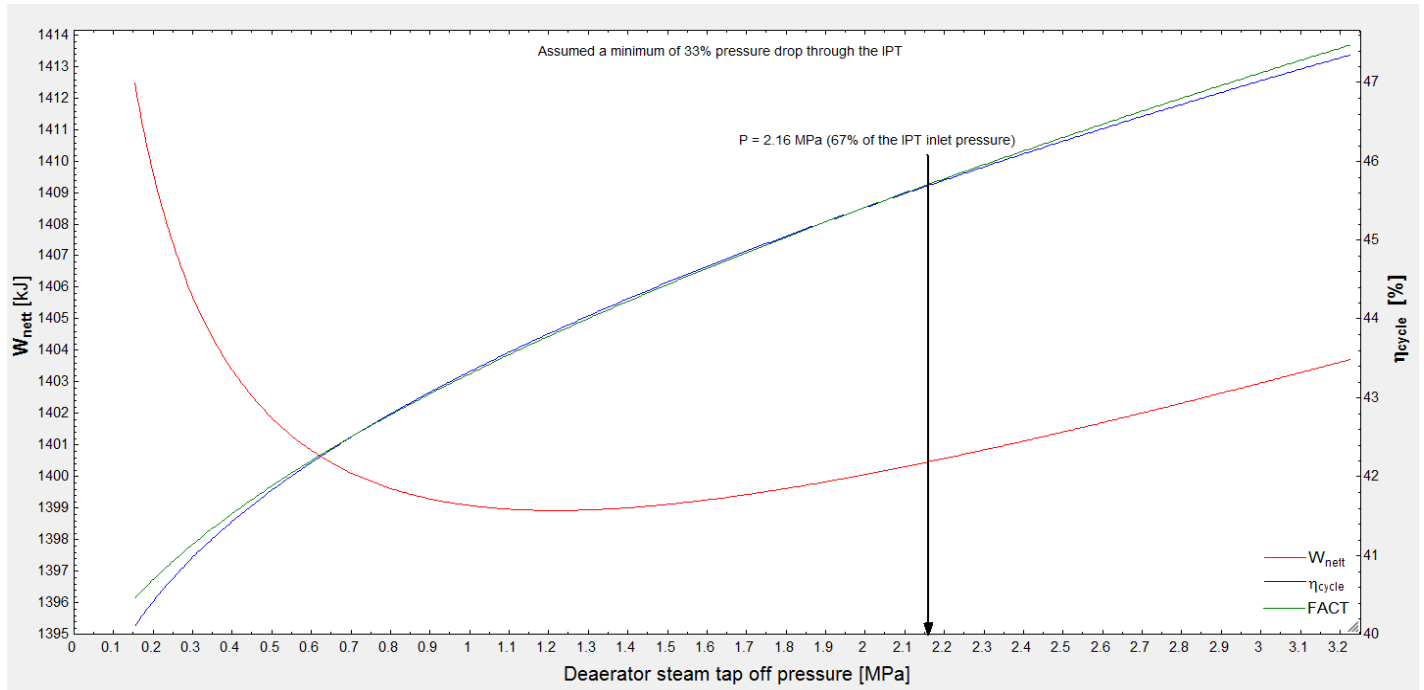


Figure 128: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (2nd method, 1st run)

Before optimisation		After optimisation	
De-aerator steam bled off pressure	0.7161 MPa	De-aerator steam bled off pressure	2.16 MPa
Cycle efficiency	42.55 %	Cycle efficiency	45.69 %
Mega Watt output	889 MW	Mega Watt output	889.3 MW
Net mechanical work	1400 kJ	Net mechanical work	1400 kJ
Heat added	3291 kJ	Heat added	3065 kJ
Heat rejected	1653 kJ	Heat rejected	1655 kJ

Table 88: Before and after optimisation results for the de-aerator steam tap off pressure (2nd method, 1st run)

6.2.3.2. Second run

6.2.3.2.1. Boiler feed pump pressure

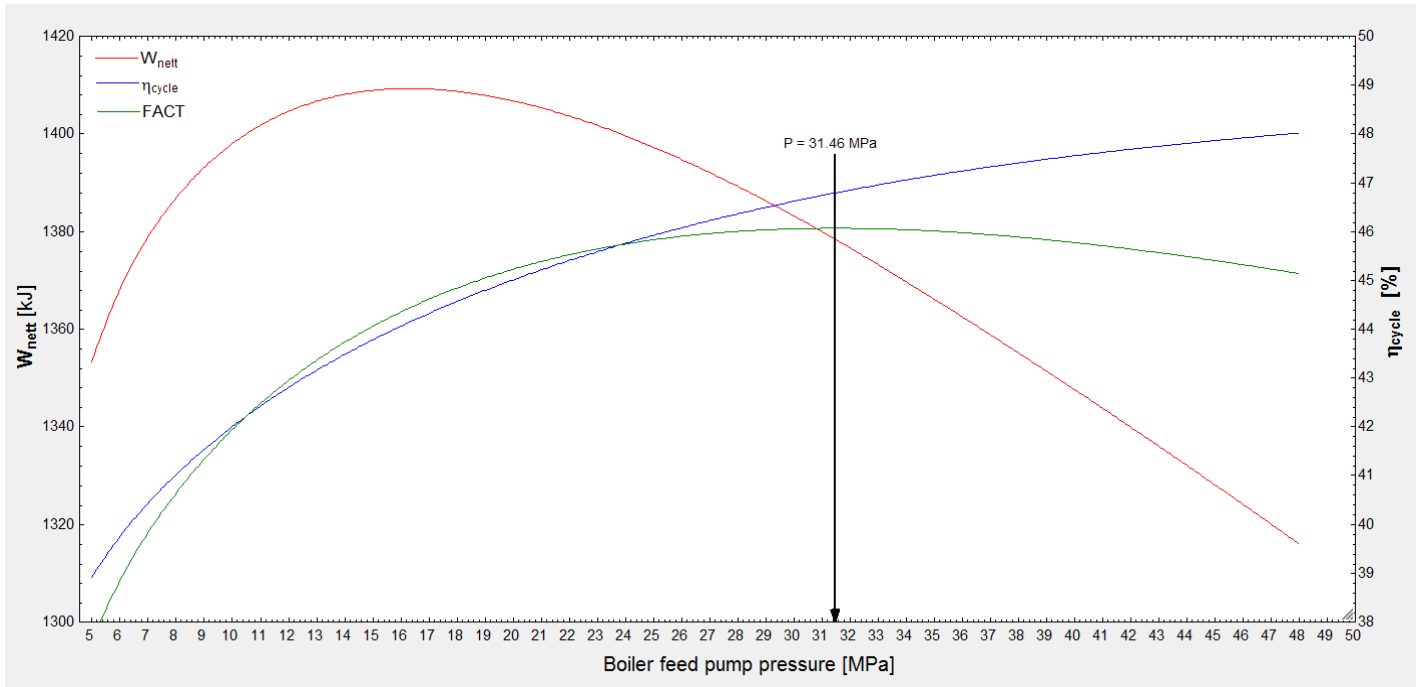


Figure 129: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (2nd method, 2nd run)

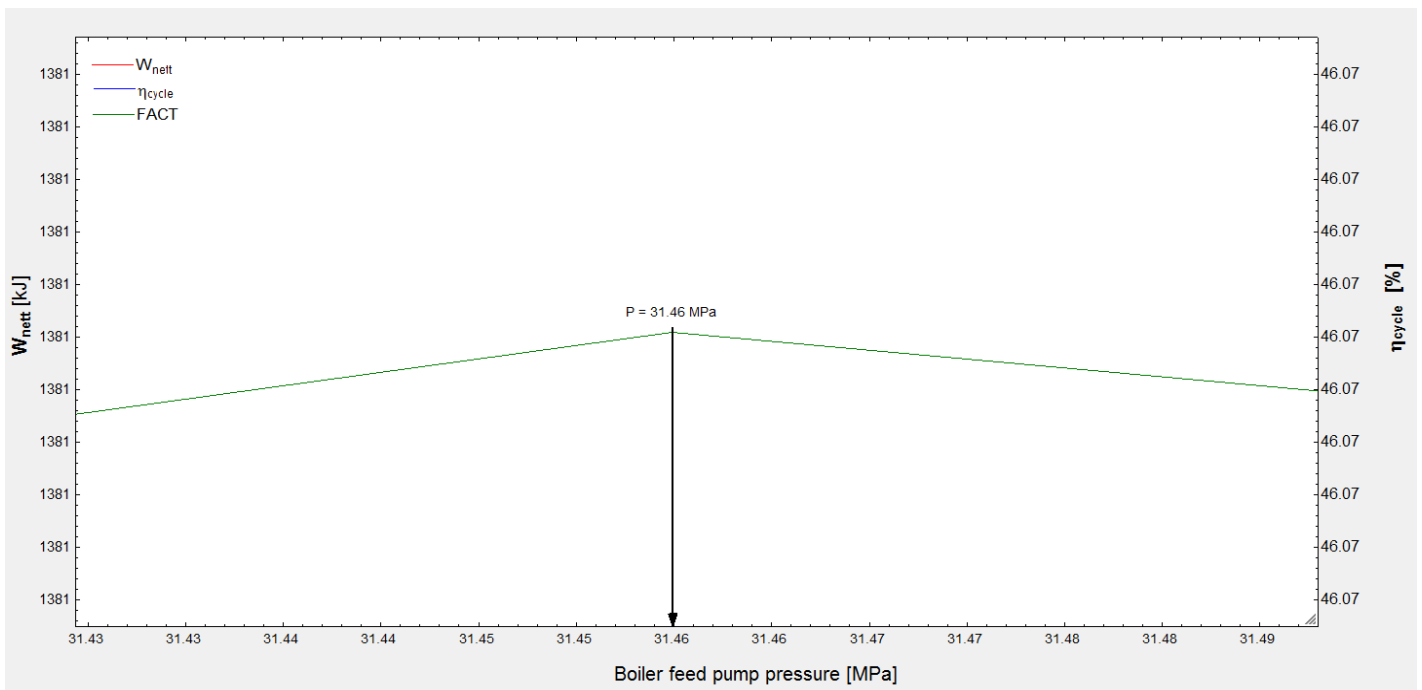


Figure 130: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (2nd method, 2nd run)

Before optimisation		After optimisation	
Boiler feed pump pressure	23.6 MPa	Boiler feed pump pressure	31.46 MPa
Cycle efficiency	45.69 %	Cycle efficiency	46.79 %
Mega Watt output	889.3 MW	Mega Watt output	875.3.3 MW
Net mechanical work	1400 kJ	Net mechanical work	1378 kJ
Heat added	3065 kJ	Heat added	2946 kJ
Heat rejected	1655 kJ	Heat rejected	1565 kJ

Table 89: Before and after optimisation results for the boiler feed pump discharge (2nd method, 2nd run)

6.2.3.2.2. High pressure heater 6

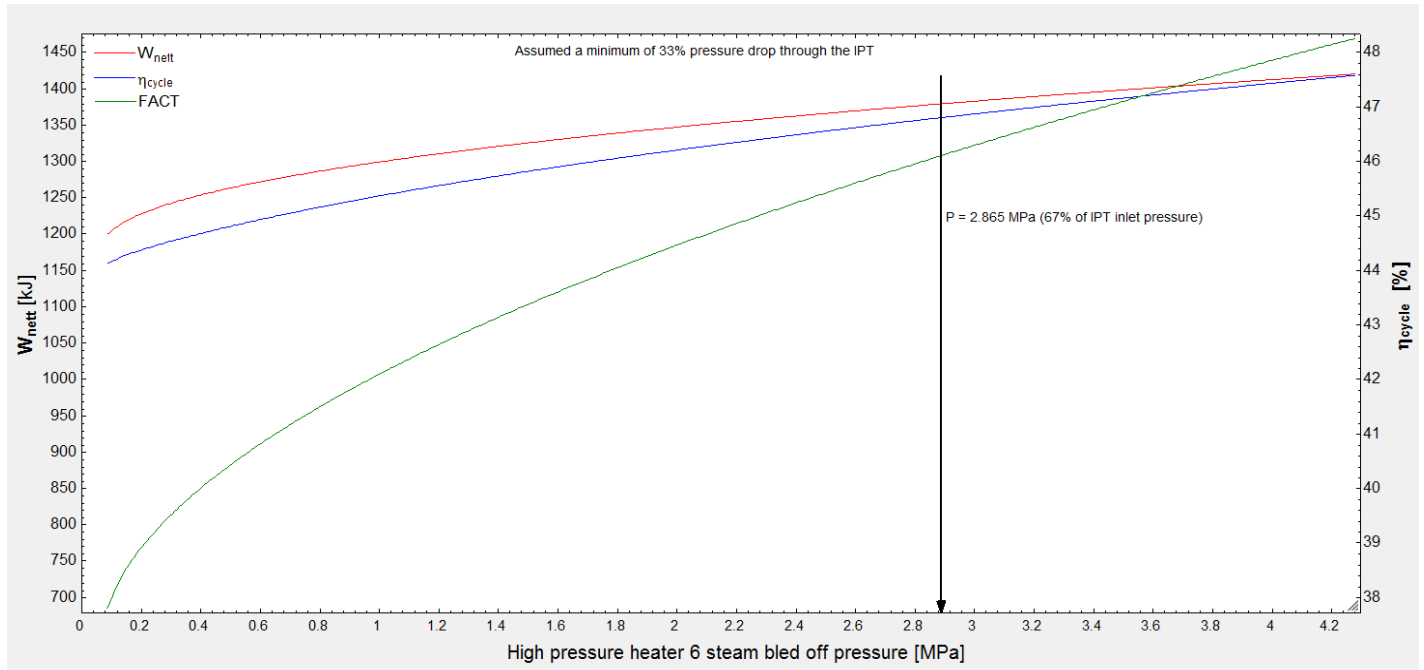


Figure 131: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
High pressure heater 6 steam bled off pressure	2.865 MPa	High pressure heater 6 steam bled off pressure	2.865 MPa
Cycle efficiency	46.79 %	Cycle efficiency	46.79 %
Mega Watt output	875.3 MW	Mega Watt output	875.3 MW
Net mechanical work	1378 kJ	Net mechanical work	1378 kJ
Heat added	2946 kJ	Heat added	2946 kJ
Heat rejected	1565 kJ	Heat rejected	1565 kJ

Table 90: Before and after optimisation results for high pressure heater 6 steam tap off pressure (2nd method, 2nd run)

6.2.3.2.3. High pressure turbine expansion

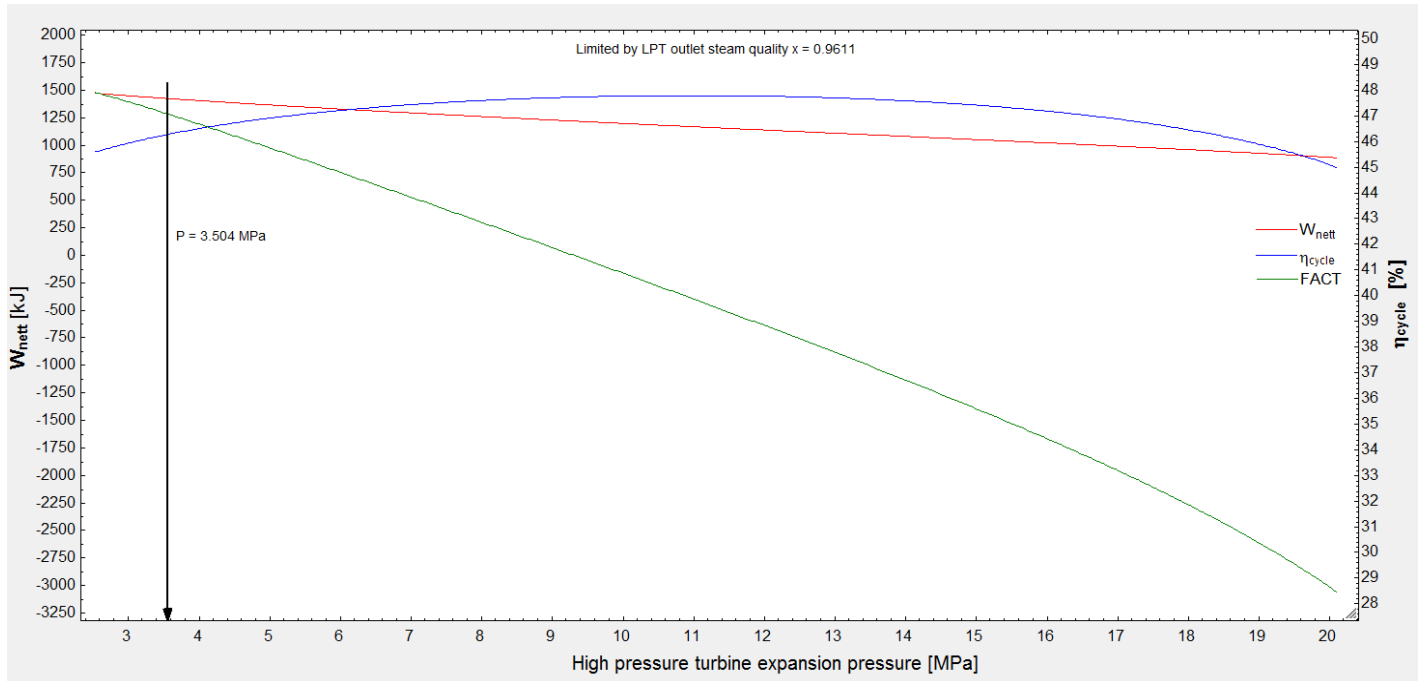


Figure 132: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	4.659 MPa	High pressure turbine expansion pressure	3.504 MPa
Cycle efficiency	46.79 %	Cycle efficiency	46.25 %
Mega Watt output	875.3 MW	Mega Watt output	905.6 MW
Net mechanical work	1378 kJ	Net mechanical work	1426 kJ
Heat added	2946 kJ	Heat added	3084 kJ
Heat rejected	1565 kJ	Heat rejected	1656 kJ

Table 91: Before and after optimisation results for the high pressure turbine expansion pressure (2nd method, 2nd run)

6.2.3.2.4. Low pressure heater 1

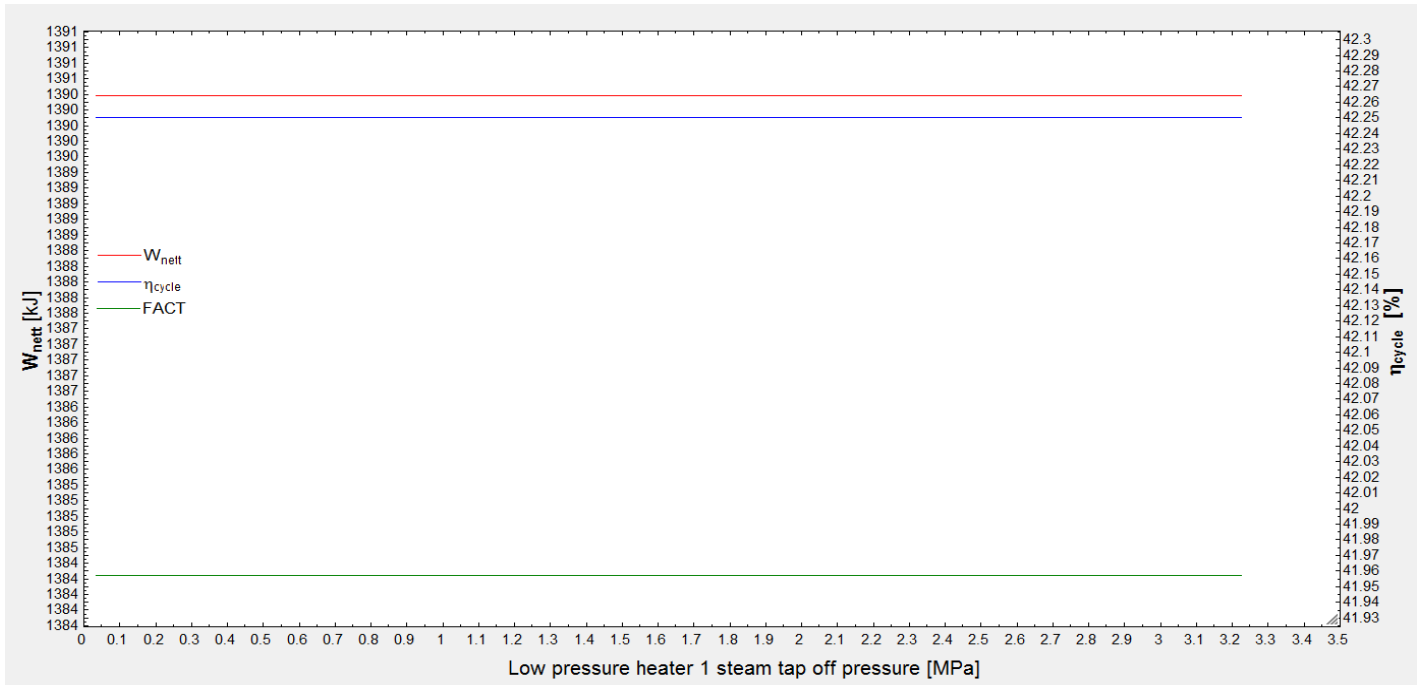


Figure 133: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.05788 MPa	Low pressure heater 1 steam tap off pressure	0.05788 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3084 kJ	Heat added	3084 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 92: Before and after optimisation results for low pressure heater 1 steam tap off pressure (2nd method, 2nd run)

6.2.3.2.5. Low pressure heater 2

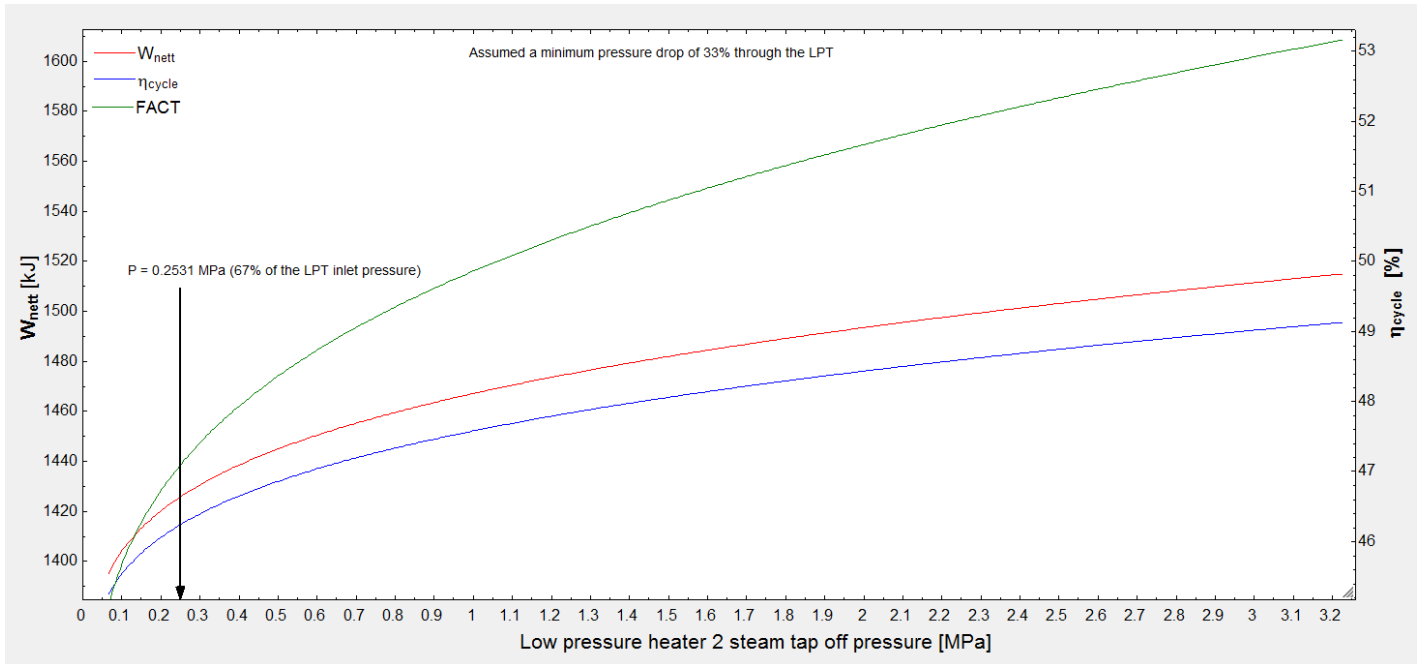


Figure 134: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.2531 MPa	Low pressure heater 2 steam tap off pressure	0.2531 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3084 kJ	Heat added	3084 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 93: Before and after optimisation results for low pressure heater 2 steam tap off pressure (2nd method, 2nd run)

6.2.3.2.6. Low pressure heater 3

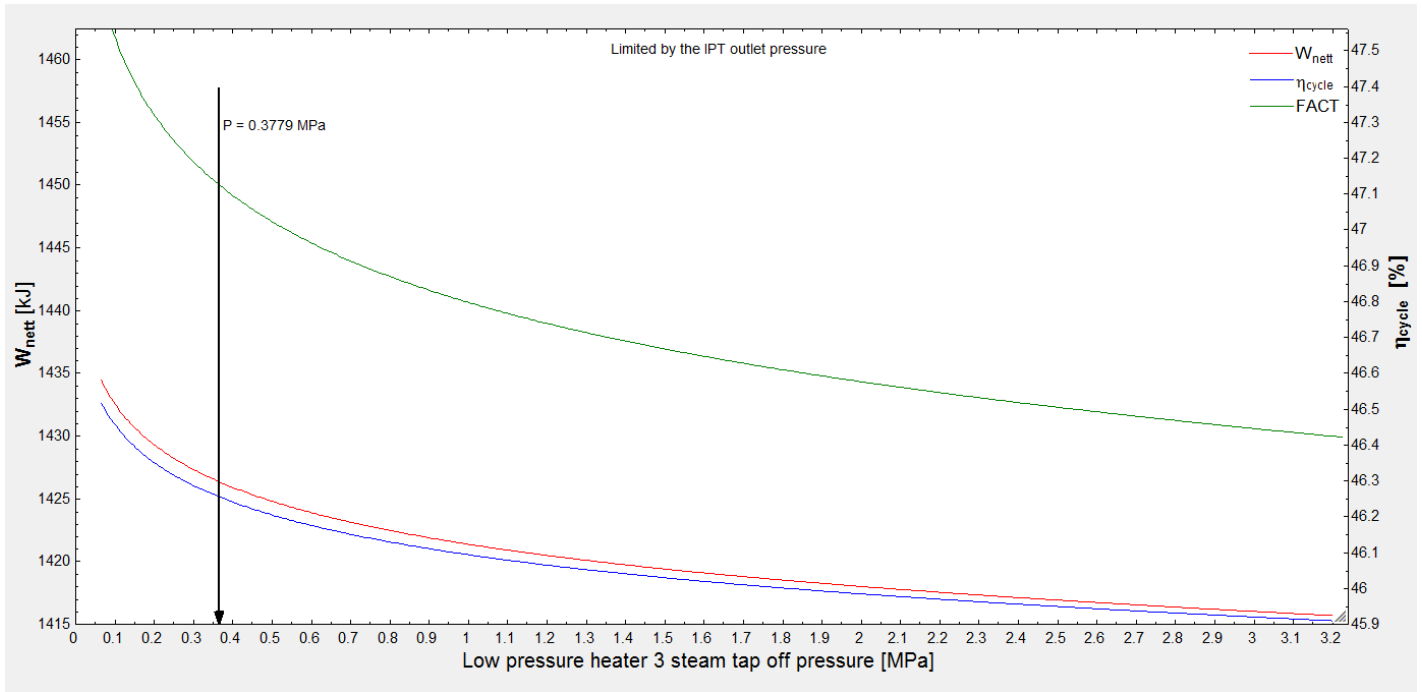


Figure 135: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (1st method, 2nd run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.3779 MPa	Low pressure heater 3 steam tap off pressure	0.3779 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3084 kJ	Heat added	3084 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 94: Before and after optimisation results for low pressure heater 3 steam tap off pressure (2nd method, 2nd run)

6.2.3.2.7. De-aerator

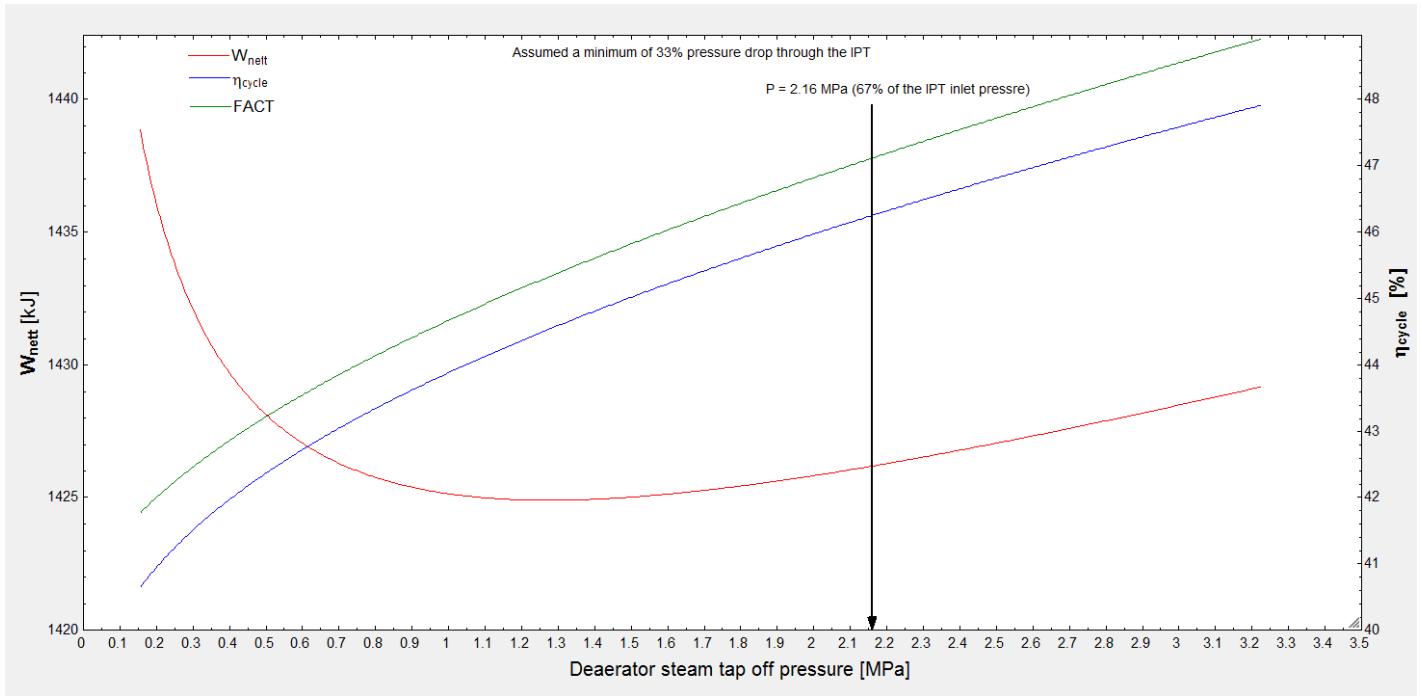


Figure 136: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (2nd method, 2nd run)

Before optimisation		After optimisation	
De-aerator steam tap off pressure	2.16 MPa	De-aerator steam tap off pressure	2.16 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.25 %
Mega Watt output	905.6 MW	Mega Watt output	905.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1426 kJ
Heat added	3084 kJ	Heat added	3084 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 95: Before and after optimisation results for the de-aerator steam tap off pressure (2nd method, 2nd run)

6.2.3.3. Third run

6.2.3.3.1. Boiler feed pump pressure

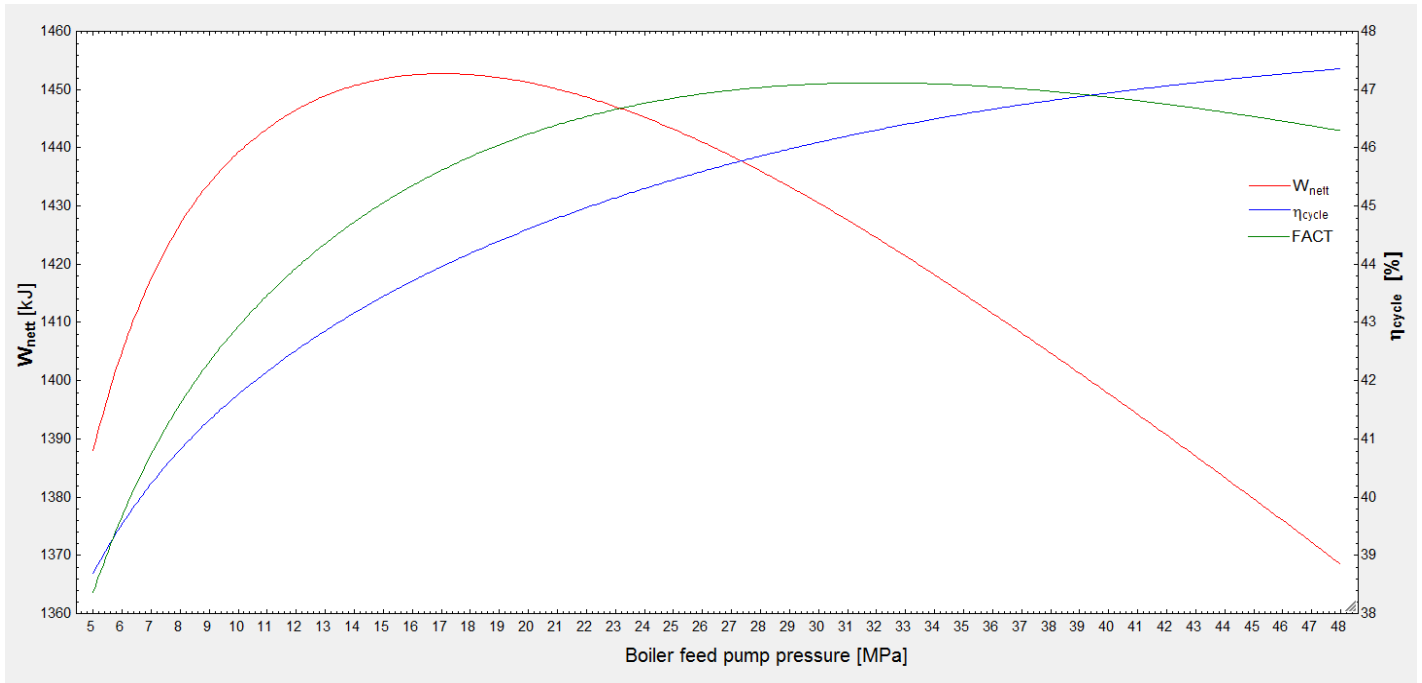


Figure 137: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (2nd method, 3rd run)

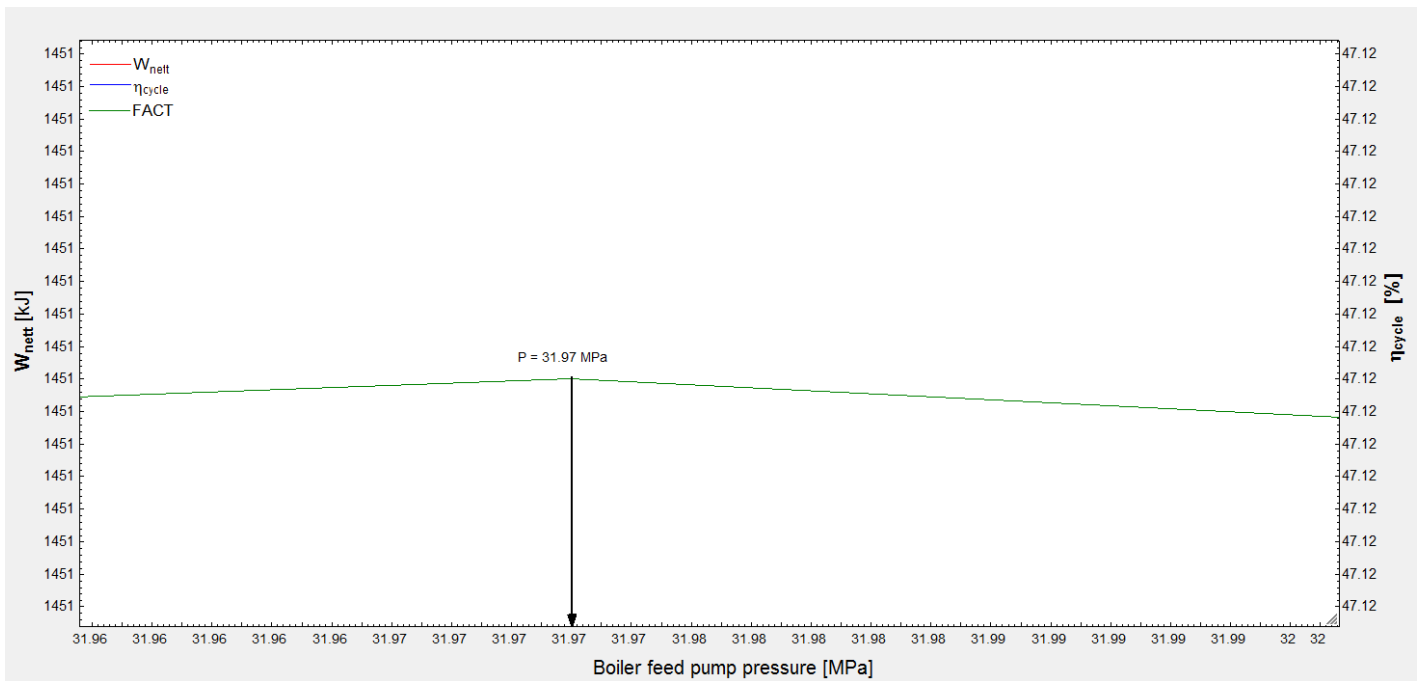


Figure 138: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (2nd method, 3rd run)

Before optimisation		After optimisation	
Boiler feed pump pressure	31.46 MPa	Boiler feed pump pressure	31.97 MPa
Cycle efficiency	46.25 %	Cycle efficiency	46.3 %
Mega Watt output	905.6 MW	Mega Watt output	904.6 MW
Net mechanical work	1426 kJ	Net mechanical work	1466 kJ
Heat added	3084 kJ	Heat added	3077 kJ
Heat rejected	1656 kJ	Heat rejected	1651 kJ

Table 96: Before and after optimisation results for the boiler feed pump discharge (2nd method, 3rd run)

6.2.3.3.2. High pressure heater 6

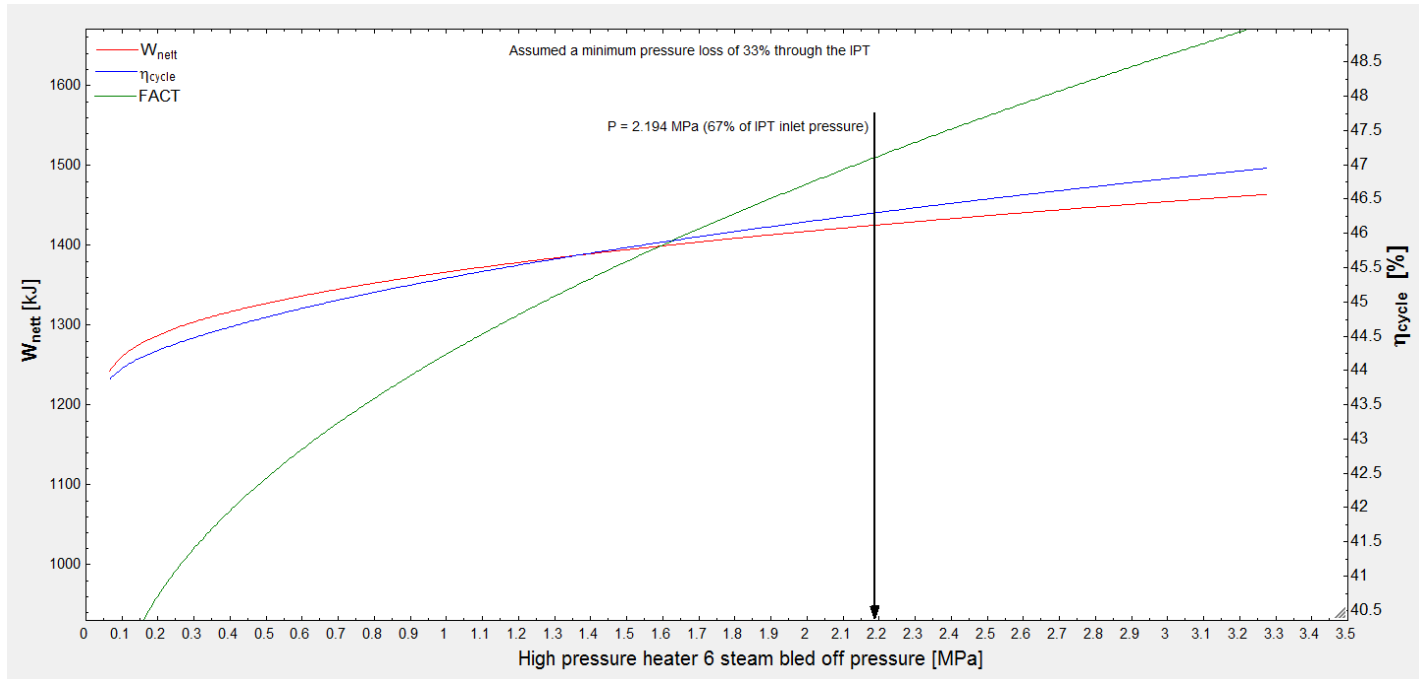


Figure 139: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure heater 6 steam tap off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
High pressure heater 6 steam tap off pressure	2.194 MPa	High pressure heater 6 steam tap off pressure	2.194 MPa
Cycle efficiency	46.3 %	Cycle efficiency	46.3 %
Mega Watt output	904.6 MW	Mega Watt output	904.6 MW
Net mechanical work	1425 kJ	Net mechanical work	1425 kJ
Heat added	3077 kJ	Heat added	3077 kJ
Heat rejected	1651 kJ	Heat rejected	1651 kJ

Table 97: Before and after optimisation results for high pressure heater 6 steam tap off pressure (2nd method, 3rd run)

6.2.3.3.3. High pressure turbine expansion

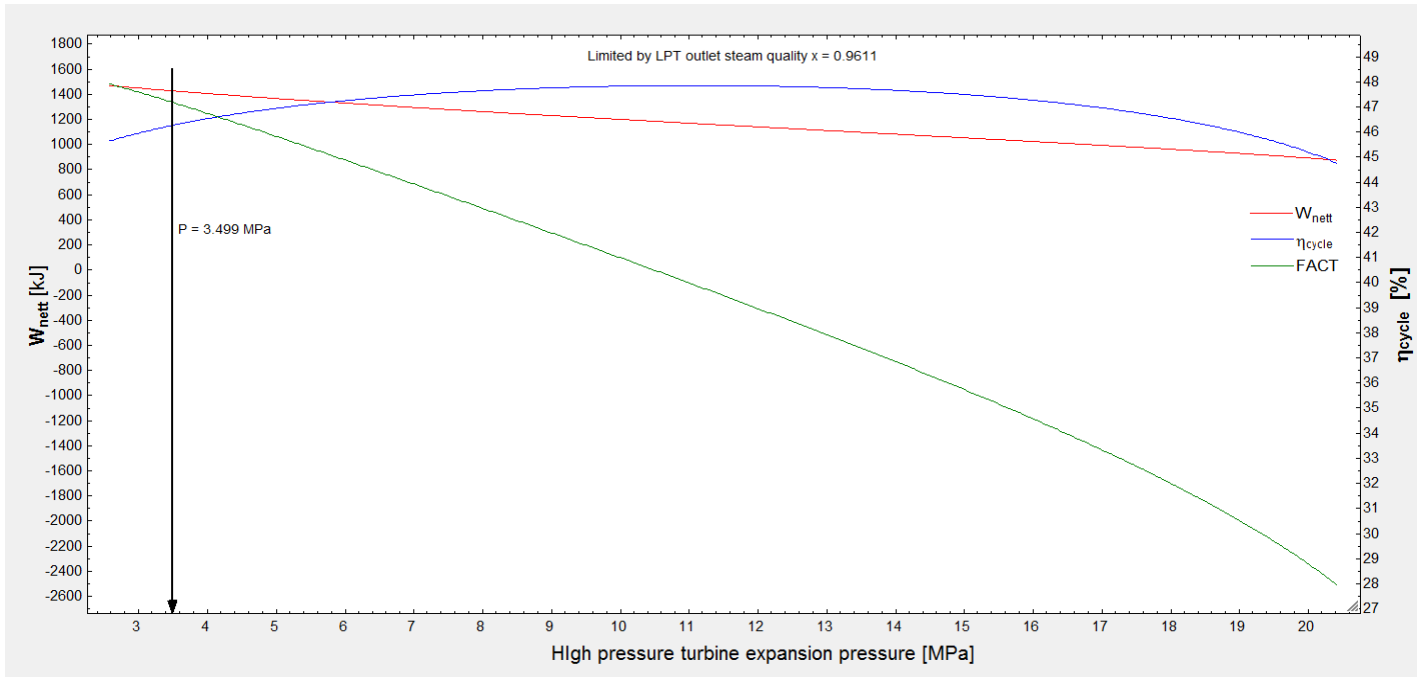


Figure 140: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.558 MPa	High pressure turbine expansion pressure	3.499 MPa
Cycle efficiency	46.3 %	Cycle efficiency	46.27 %
Mega Watt output	904.6 MW	Mega Watt output	906.2 MW
Net mechanical work	1425 kJ	Net mechanical work	1427 kJ
Heat added	3077 kJ	Heat added	3085 kJ
Heat rejected	1651 kJ	Heat rejected	1656 kJ
LPT outlet steam quality	X = 0.9602	LPT outlet steam quality	X = 0.9611

Table 98: Before and after optimisation results for the high pressure turbine expansion pressure (2nd method, 3rd run)

6.2.3.3.4. Low pressure heater 1

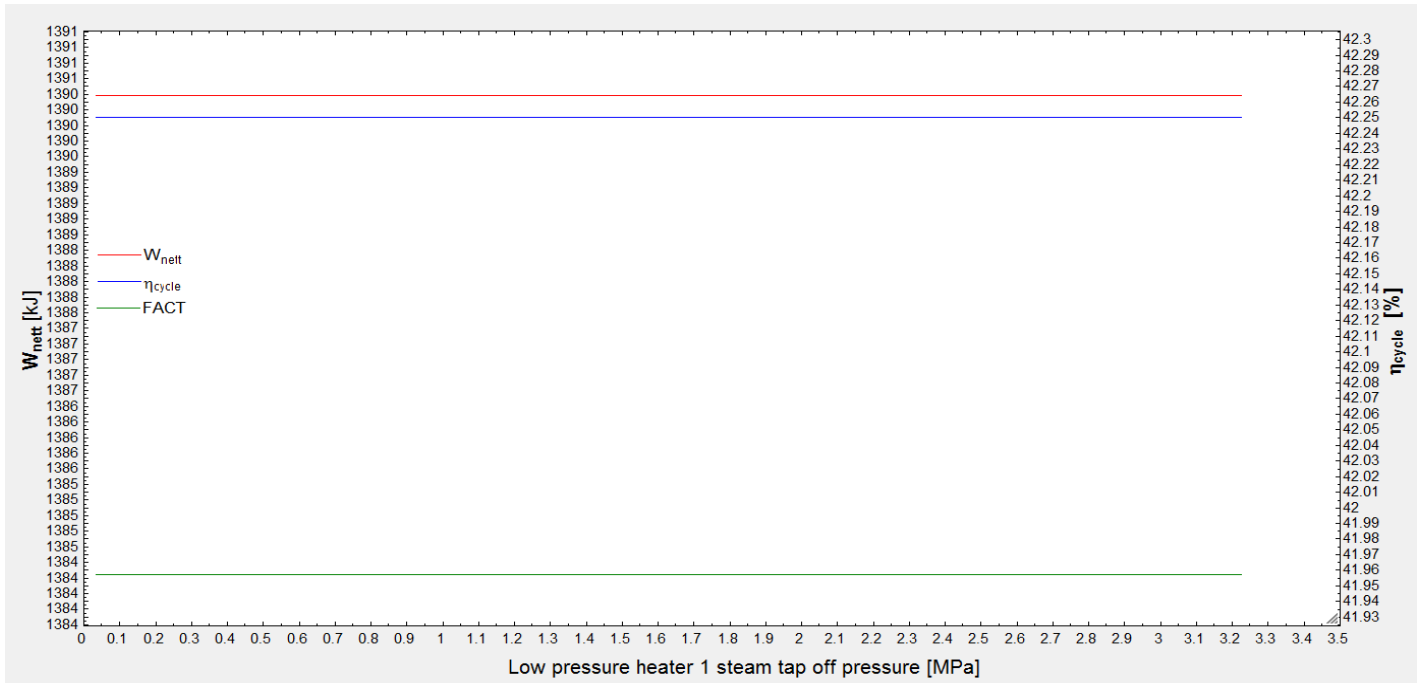


Figure 141: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 1 steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 1 steam tap off pressure	0.05782 MPa	Low pressure heater 1 steam tap off pressure	0.05782 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.2 MW	Mega Watt output	906.2 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 99: Before and after optimisation results for low pressure heater 1 steam tap off pressure (2nd method, 3rd run)

6.2.3.3.5. Low pressure heater 2

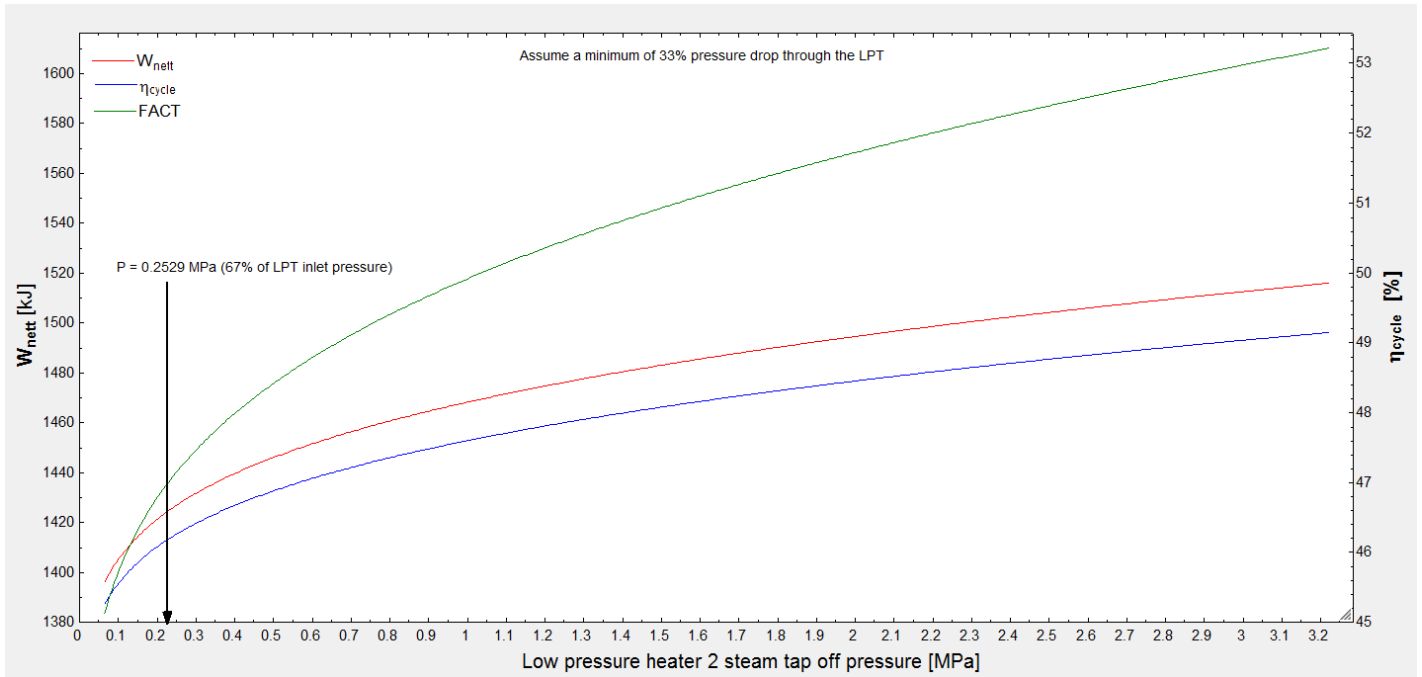


Figure 142: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 2 steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 2 steam tap off pressure	0.2529 MPa	Low pressure heater 2 steam tap off pressure	0.2529 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.2 MW	Mega Watt output	906.2 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 100: Before and after optimisation results for low pressure heater 2 steam tap off pressure (2nd method, 3rd run)

6.2.3.3.6. Low pressure heater 3

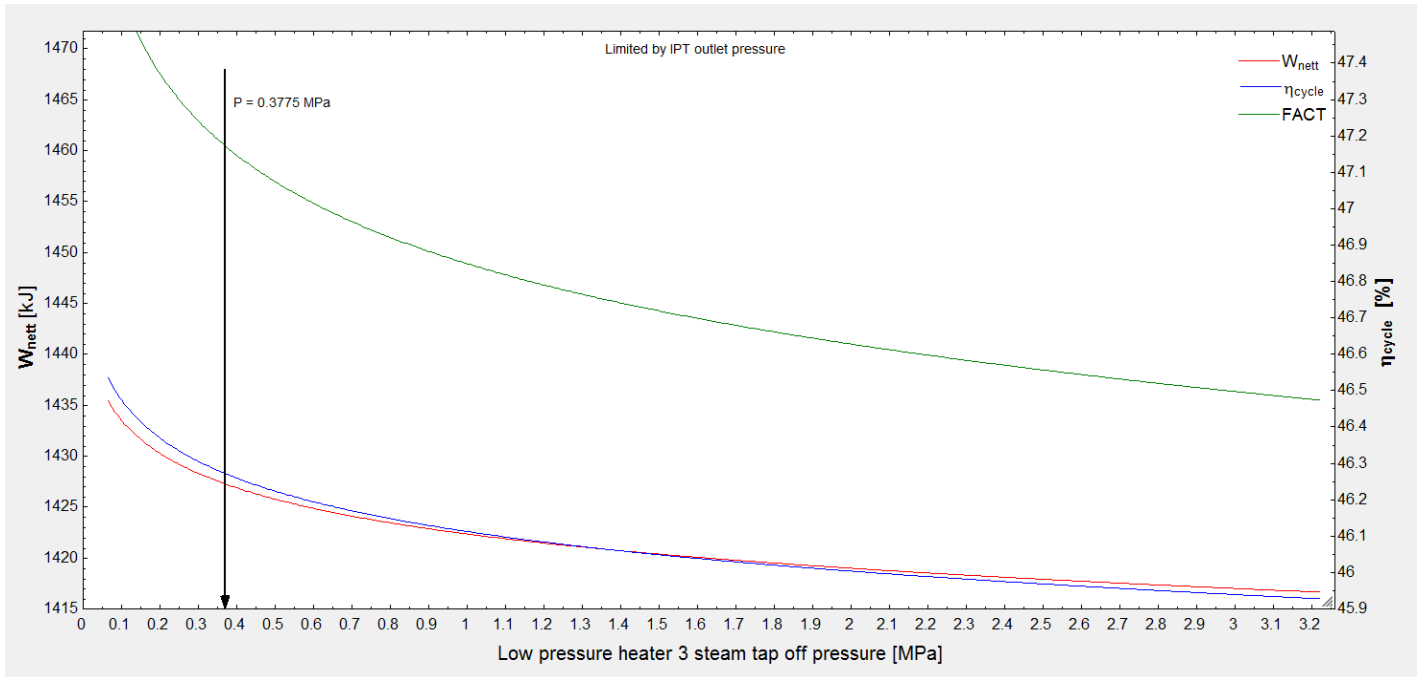


Figure 143: Cycle efficiency, net mechanical work and the factor line plotted against low pressure heater 3 steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
Low pressure heater 3 steam tap off pressure	0.3775 MPa	Low pressure heater 3 steam tap off pressure	0.3775 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.2 MW	Mega Watt output	906.2 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 101: Before and after optimisation results for low pressure heater 3 steam tap off pressure (2nd method, 3rd run)

6.2.3.3.7. De-aerator

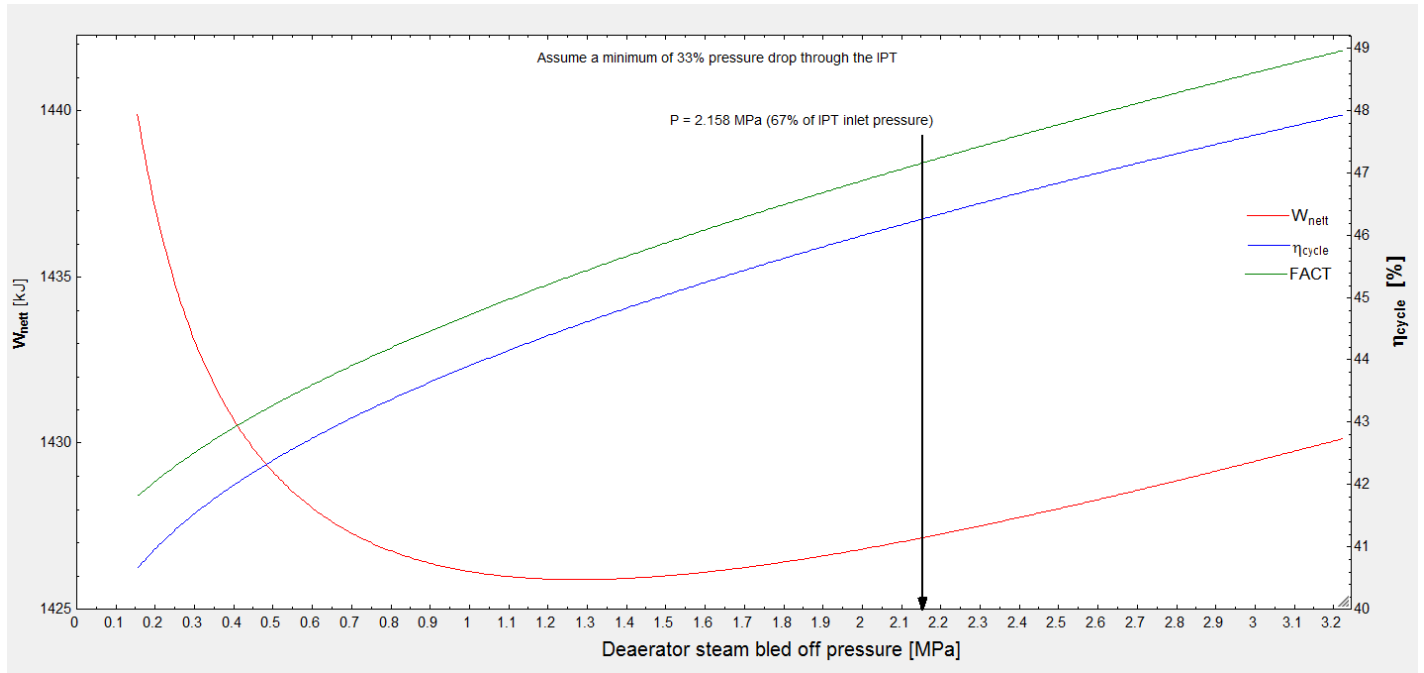


Figure 144: Cycle efficiency, net mechanical work and the factor line plotted against the de-aerator steam bled off pressure (2nd method, 3rd run)

Before optimisation		After optimisation	
De-aerator steam bled off pressure	2.158 MPa	De-aerator steam bled off pressure	2.158 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.2 MW	Mega Watt output	906.2 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 102: Before and after optimisation results for the de-aerator steam tap off pressure (2nd method, 3rd run)

6.2.3.3.8. Boiler feed pump pressure

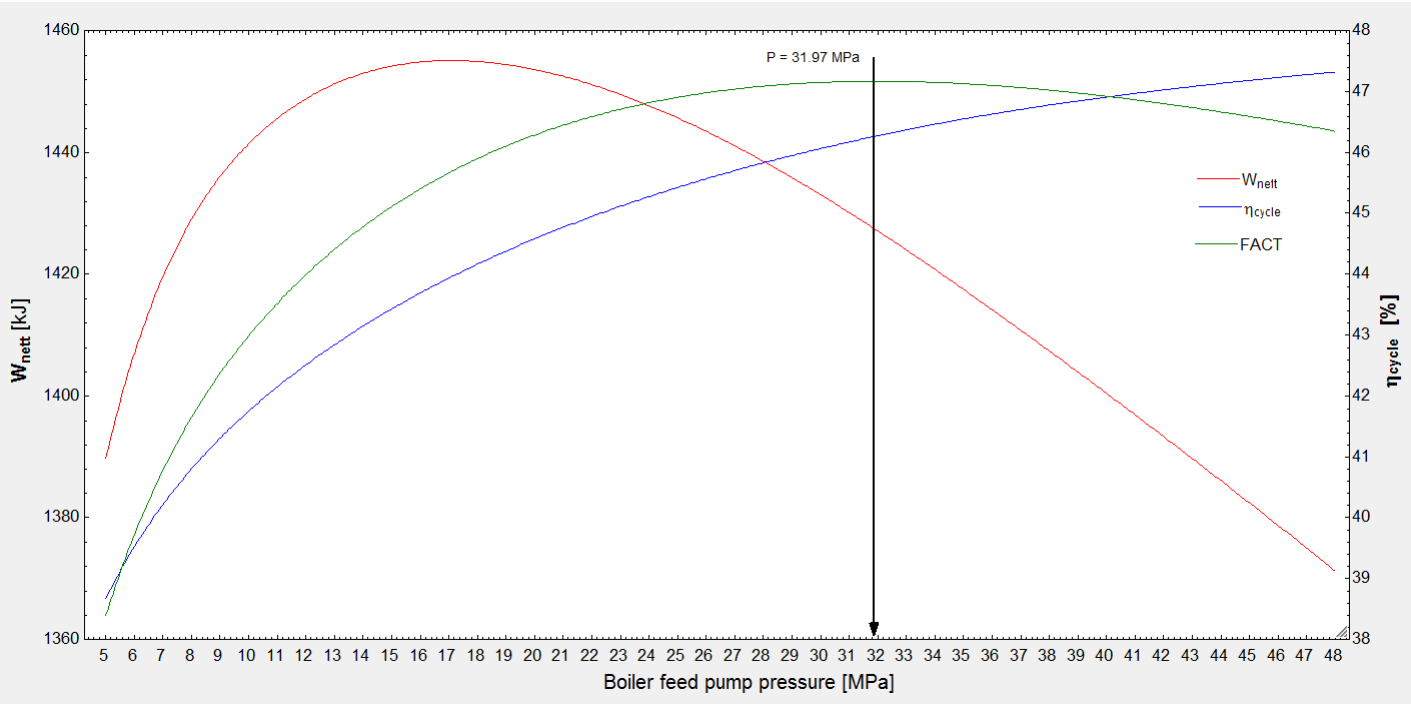


Figure 145: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (2nd method, 4th run)

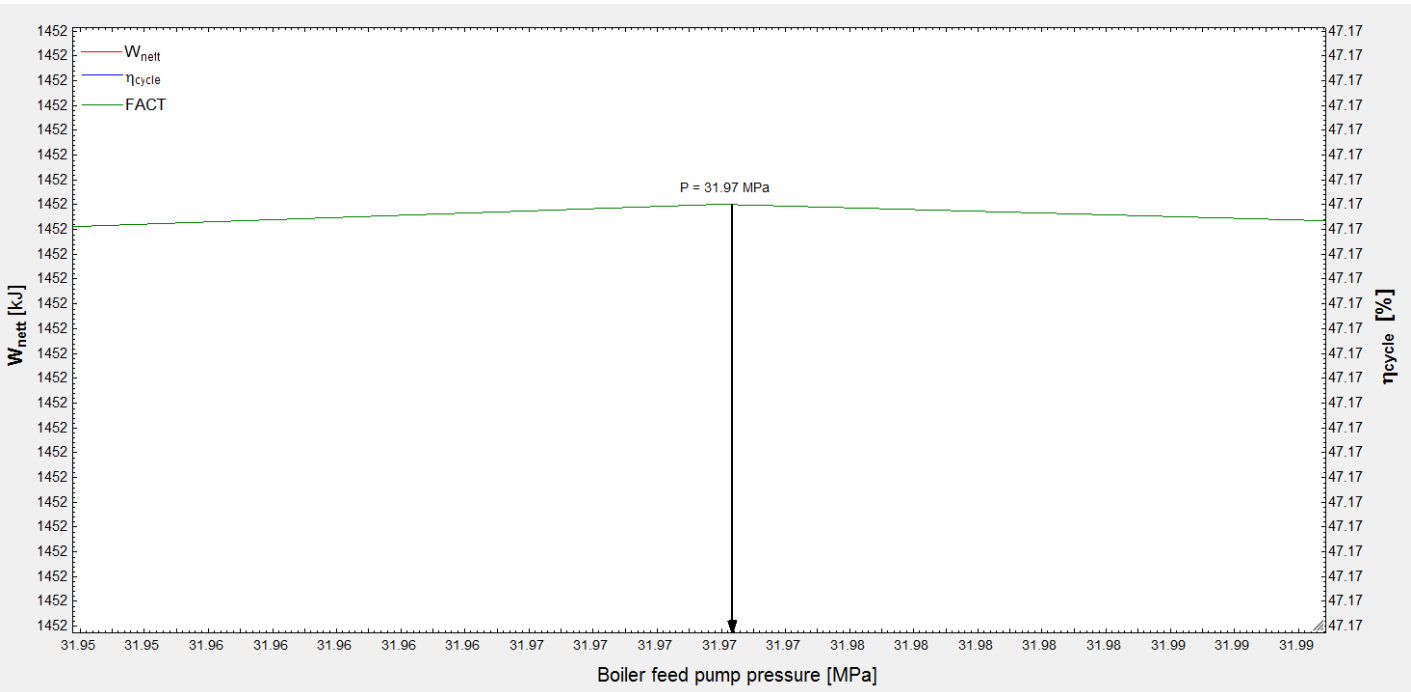


Figure 146: Cycle efficiency, net mechanical work and the factor line plotted against boiler feed pump discharge pressure (zoomed in) (2nd method, 4th run)

Before optimisation		After optimisation	
Boiler feed pump pressure	31.97 MPa	Boiler feed pump pressure	31.97 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.2 MW	Mega Watt output	906.2 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ

Table 103: Before and after optimisation results for the boiler feed pump discharge (2nd method, 4th run)

6.2.3.3.9. High pressure turbine expansion

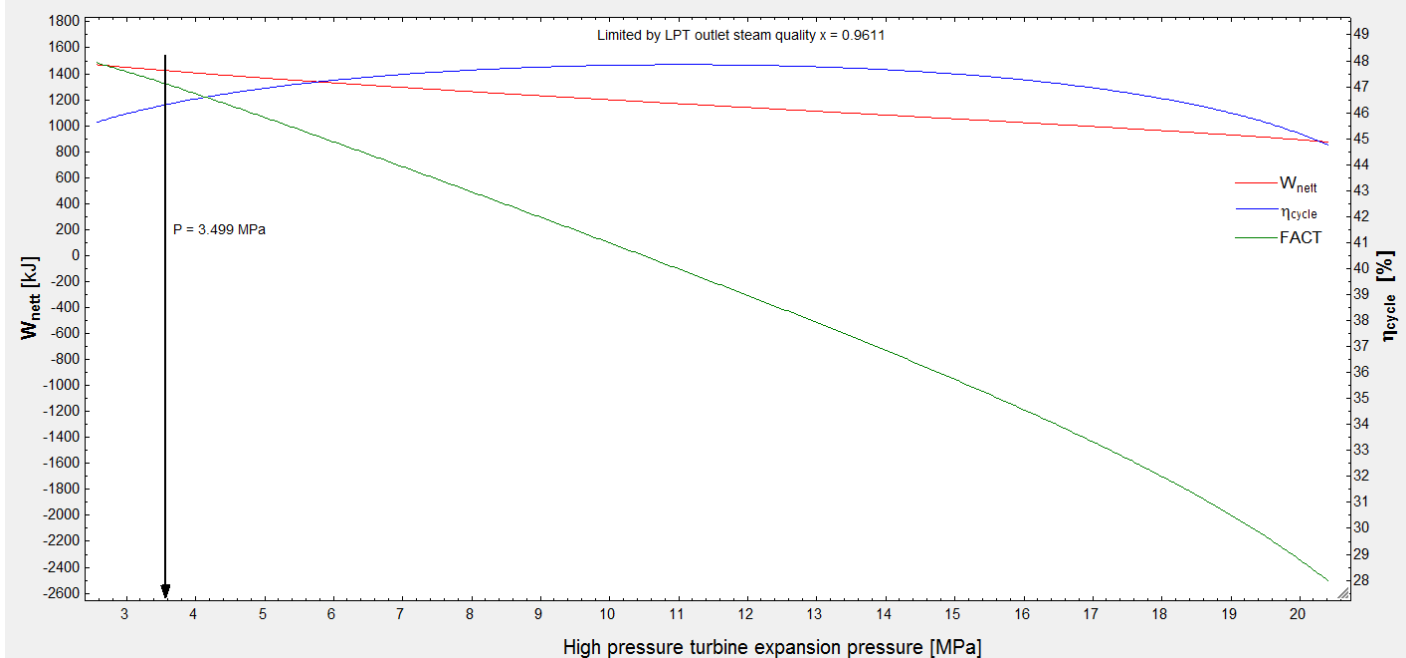


Figure 147: Cycle efficiency, net mechanical work and the factor line plotted against the high pressure turbine expansion pressure (2nd method, 4th run)

Before optimisation		After optimisation	
High pressure turbine expansion pressure	3.499 MPa	High pressure turbine expansion pressure	3.499 MPa
Cycle efficiency	46.27 %	Cycle efficiency	46.27 %
Mega Watt output	906.2 MW	Mega Watt output	906.2 MW
Net mechanical work	1427 kJ	Net mechanical work	1427 kJ
Heat added	3085 kJ	Heat added	3085 kJ
Heat rejected	1656 kJ	Heat rejected	1656 kJ
LPT outlet steam quality	X = 0.9611	LPT outlet steam quality	X = 0.9611

Table 104: Before and after optimisation results for the high pressure turbine expansion pressure (2nd method, 4th run)

6.2.3.4. Summary of super-critical results

First method			
	Boiler feed pump pressure	Efficiency	Mega Watt output
Current operating parameters	30.13 MPa	42.31%	831.6 MW
After first run	23.6 MPa	45.69%	889.3 MW
After second run	31.47 MPa	46.25%	905.6 MW
After third run	31.99 MPa	46.27%	906.3 MW
After fourth run	31.97 MPa	46.27%	906.3 MW

Second method			
	Boiler feed pump pressure	Efficiency	Mega Watt output
Current operating parameters	30.13 MPa	42.31%	831.6 MW
After first run	23.6 MPa	45.69%	889.3 MW
After second run	31.46 MPa	46.25%	905.6 MW
After third run	31.97 MPa	46.27%	906.2 MW
After fourth run	31.97 MPa	46.27%	906.2 MW

Table 105: Optimisation results for each run and method for the super-critical Rankine cycle

6.2.4. Adjustments to the final optimised cycles for super-critical

Optimisation of the first and second methods produces the same results.

The result of the tap off steam flow for high pressure heater 6 was negative, which means steam was flowing back into the intermediate pressure turbine. Thus high pressure heater 6 was no longer necessary and was removed. The cycle was calculated again and the following results were obtained. For this section the steam tap off flow m_6 is equal to 0.

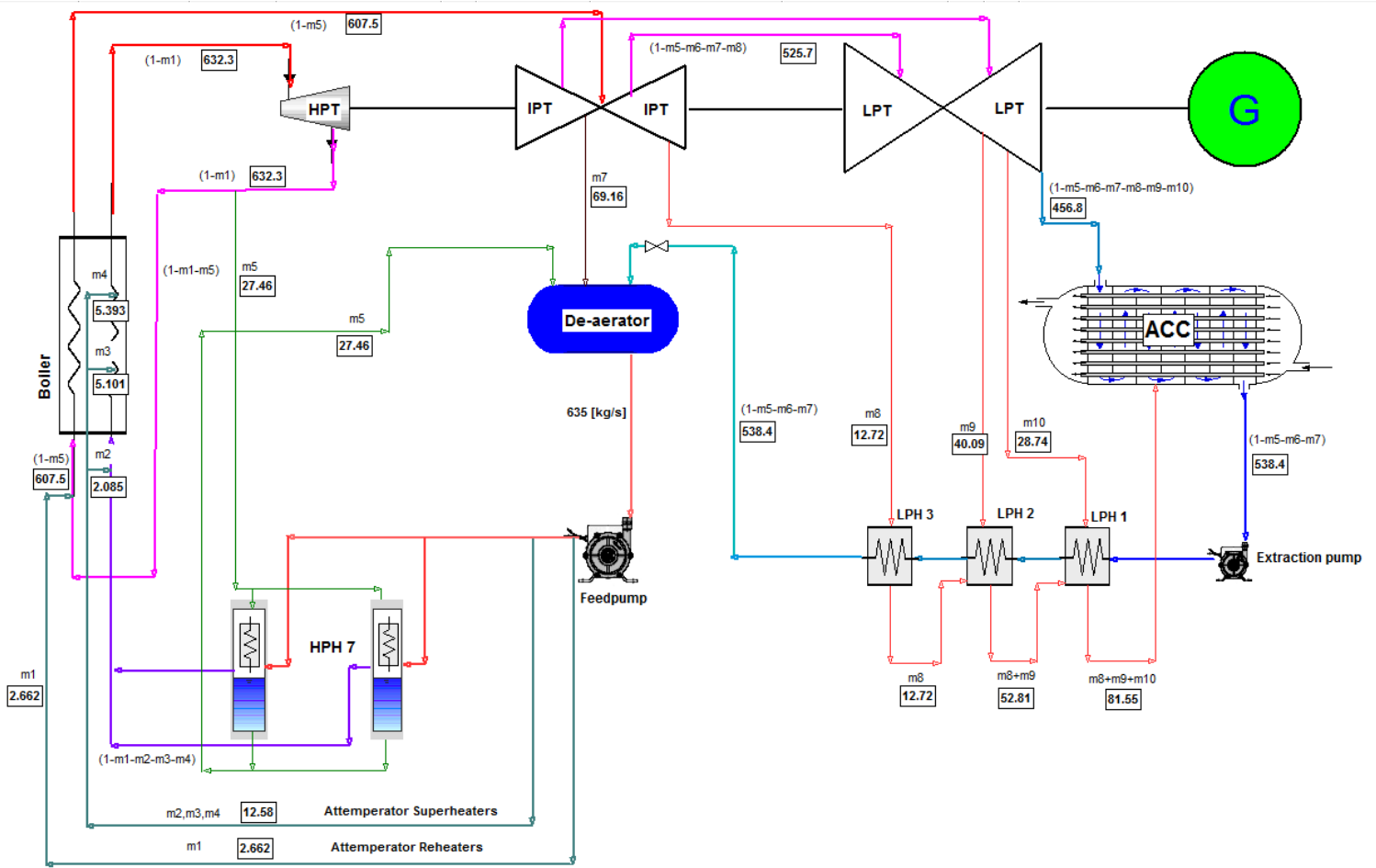


Figure 148: Super-critical plant layout without high pressure heater 6

After high pressure heater 6 was taken out	
Maximum boiler pressure	31.97 MPa
Cycle efficiency	46.53%
Mega Watt output	916.3 MW
Net mechanical work	1443 kJ
Heat added	3101 kJ
Heat rejected	1666 kJ

Table 106: Results for super-critical cycle after optimisation and after high pressure heater 6 was taken out

6.3. Sub-critical vs super-critical

6.3.1. Before optimisation

	Sub-critical	Super-critical
Feed pump	22.15 MPa	30.13 MPa
Maximum temperature	516°C	570°C
Minimum temperature	40°C	50°C
Maximum mass flow	415 kg/s	635 kg/s
Extraction pump discharge pressure	2.388 MPa	2.5 MPa
High pressure heater 6 bled steam tap off pressure	1.738 MPa	2.745 MPa
De-aerator bled steam tap off pressure	1.032 MPa	1.12 MPa
Low pressure heater 4 bled steam tap off pressure	0.4106 MPa	
Low pressure heater 3 bled steam tap off pressure	0.1743 MPa	0.591 MPa
Low pressure heater 2 bled steam tap off pressure	0.06046 MPa	0.2671 MPa
Low pressure heater 1 bled steam tap off pressure	0.02991 MPa	0.09048 MPa
Cycle efficiency	42.69%	42.31%
Mega Watt out	538.4 MW	831.6 MW
Total work in	28.68 kJ	38.65 kJ
Total work out	1326 kJ	1348 kJ
Net mechanical work	1297 kJ	1310 kJ
Heat added	3039 kJ	3096 kJ
Heat rejected	1553 kJ	1495 kJ

Table 107: Comparison between sub- and super-critical Rankine cycles before optimisation

6.3.2. After optimisation

	Sub-critical	Super-critical
Feed pump	29.29 MPa	31.97 MPa
Maximum temperature	516°C	570°C
Minimum temperature	40°C	50°C
Maximum mass flow	415 kg/s	635 kg/s
Extraction pump discharge pressure	2.388 MPa	2.5 MPa
High pressure heater 6 bled steam tap off pressure	0 MPa	0 MPa
De-aerator bled steam tap off pressure	1.969 MPa	2.157 MPa
Low pressure heater 4 bled steam tap off pressure	0.379 MPa	-
Low pressure heater 3 bled steam tap off pressure	0.2547 MPa	0.3775 MPa
Low pressure heater 2 bled steam tap off pressure	0.05612 MPa	0.2529 MPa
Low pressure heater 1 bled steam tap off pressure	0.02776 MPa	0.05782 MPa
Cycle efficiency	46.34%	46.53%
Mega Watt out	569.2 MW	916.3 MW
Total work in	37.76 kJ	41.31 kJ
Total work out	1409 kJ	1484 kJ
Net mechanical work	1372 kJ	1443 kJ
Heat added	2960 kJ	3101 kJ
Heat rejected	1600 kJ	1666 kJ

Table 108: Comparison between sub- and super-critical Rankine cycles after optimisation

7. Conclusion and recommendations

7.1. Conclusions

In this study optimisation was used to optimise the boiler pressure, high pressure turbine expansion and feed water heating steam tap of pressures.

The objective was to achieve higher efficiency and net mechanical work output with optimisation. Optimisation allows us to find the optimal point to get the most efficiency without losing too much net mechanical work.

After plotting the boiler feed pump optimised results the efficiency increased as the pressure increased and the net mechanical work increased up to a certain pressure then started to decrease. Thus finding an optimisation point was necessary to determine where the cycle would benefit both.

After plotting the turbine expansion pressure optimised results it can be concluded that the more the turbine expands the more efficiency and net mechanical work will be achieved. Thus the limit on expansion is determined by the low pressure outlet steam quality. If the steam quality exceeds the point where it transforms into liquid, more and bigger heat exchangers will be needed as the volume of steam is much larger than liquid

After plotting the high pressure heater and the de-aerator optimised results it can be concluded that the more tap off steam used the higher the efficiency and the net mechanical work, but a minimum pressure loss through the intermediate turbine must be taken into account.

After plotting the low pressure heater 4 (sub-critical) and low pressure heater 3 (super-critical) it can be seen that as the steam tap off pressure from the intermediate pressure turbine increases the efficiency and net mechanical work decrease. This is because the more energy that is used for feed water heating the less energy is supplied to the intermediate turbine. The minimum pressure is limited by the intermediate pressure turbine outlet pressure.

After plotting the optimised results for low pressure heater 3 (sub-critical) and low pressure heater 2 (super-critical) it can be concluded that the more steam that is tapped off the higher

the efficiency and net mechanical work output achieved. This is because so little pressure is used and more tap off pressure would result in an increase in efficiency and net mechanical work. This is limited by a minimum pressure loss through the low pressure turbine.

After plotting the optimised results for low pressure heater 2 and low pressure heater 1 (sub-critical) and low pressure heater 1 (super-critical) it can be concluded that the pressure is too low to make a difference.

After looking at the whole cycle the conclusion can be drawn that when the maximum temperature and pressure are raised the efficiency and net mechanical work rise as well. This is limited by metallurgical conditions of the material that handles the temperature and pressure. With better metallurgical conditions higher temperatures and pressure could be achieved, thus super-critical and ultra-super-critical Rankine cycles are possible.

The main difference between these sub- and super-critical cycles is the limit of metallurgical conditions. In the sub-critical cycle the temperature and pressure are limited and super-critical cannot be achieved.

The pressure of the sub-critical and the super-critical cycles is almost the same after optimisation. The reasons of the lower power generated output is because the mass flow of the steam produced was not taken into account and kept constant for each cycle. Super-critical cycles produce more mass flow of steam per second.

Temperature was also not taken into account and therefore the super-critical has a higher temperature.

Steam tap off points are all dependent on the final temperature and pressure. With higher temperature and pressure it is possible to tap off more steam and higher efficiencies could be achieved.

However, the efficiency was increased by 3.65% on the sub-critical cycle and 4.22% on the super-critical cycle with an increase of 30.8 MW and 84.7 MW respectively, the sub- and super-critical cycles face major challenges before implementing the optimisation results on the live plant.

- The metallurgical conditions should be improved to handle the temperature and pressure.
- Bigger turbines are required for further and full expansion.
- The steam tap off points need to be moved to accommodate the required pressure.
- Bigger condensers are necessary if the final stage turbine does not expand fully.

This would cost a lot and is not economical to do on older plants, but for future use it can be implemented in the design phase. If these results can be implemented it would save a lot of money over time and lower emissions would result.

7.2. Recommendations

The following recommendations can be made following the results achieved in the present study.

- Ambient temperatures were not taken in account in this study, thus it can be incorporated in future studies.
- Cooling down the water returning from the cooling towers.
- Simulating and optimising a Rankine cycle for ultra-super-critical boilers.
- Programming the steam feed pump into the Rankine cycle and optimising it.
- Optimising the maximum temperature for each cycle.
- If this were to be implemented, it would be recommended to optimise one feed water heater first to see how the cycle reacts before further optimisation is done.
- Exploring possibilities with different materials to reach higher temperatures and pressures.

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9. Appendix

9.1. Appendix A

9.1.1. Material development

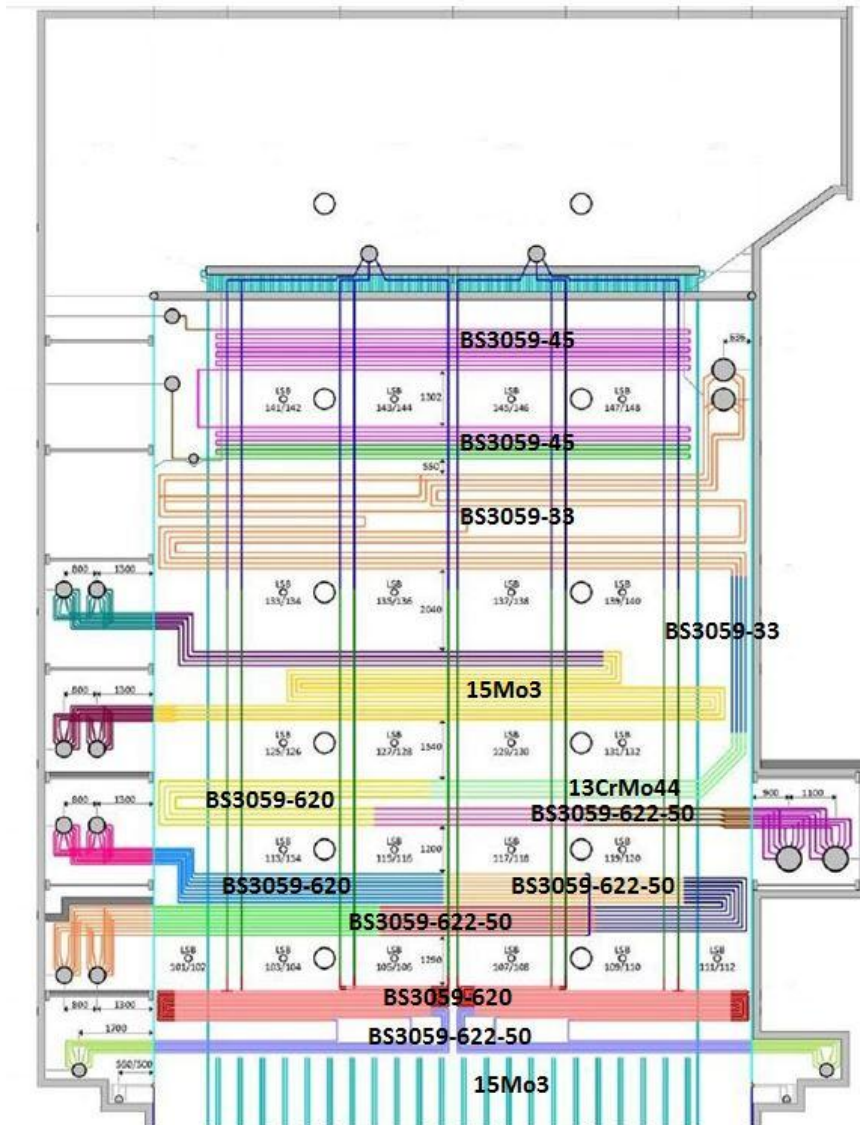


Figure 149: Kriel power station (sub-critical) boiler material

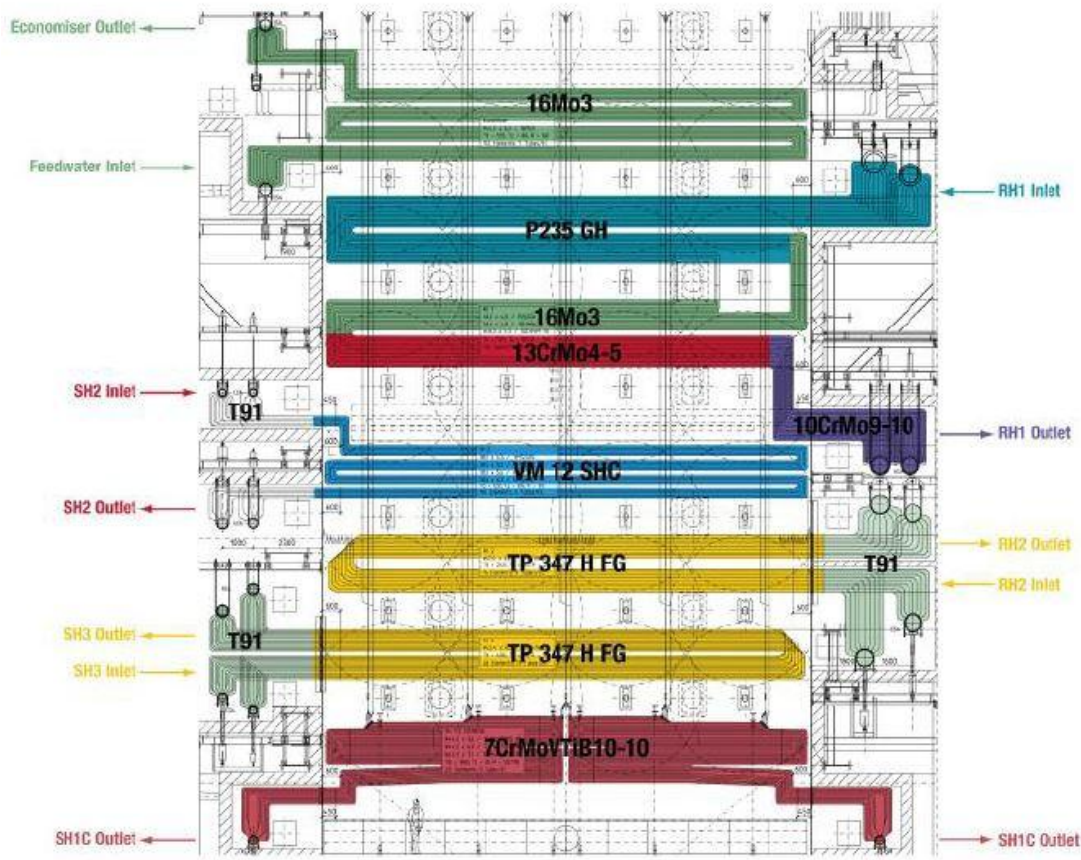


Figure 150: Medupi power station (super-critical) boiler material

The challenge was to develop material that can handle the intense heat and enormous pressure. P91 was developed (with 9% chromium) for the final stages of the superheat and reheat where the temperature is above 540°C. The new super-critical boilers use 13CrMo45 (with 1% chromium) material for the water walls (evaporation stage). 13CrMo45 was previously used in sub-critical boilers and performs outstandingly.

16Mo3 is used in the economiser. 16Mo3 is a chrome molybdenum steel alloy that is perfect for heat and corrosion resistance and is wear and fatigue resistant. 16Mo3 has a good tensile and yield strength with great toughness.

Reheater 1 is made up of four stages. The inlet and first sections are P235 GH. The second section is 16Mo3. The third section is 13CrMo45 with the outlet section as 10CrMo910.

The Superheater 2 is made up of VM12SHC.

The Reheater 2 and Superheater 3 are made up of TP347HFG.

Superheater 1 is made of 7CrMoVTi10-10.

Superheater 2, Reheater 2 and Superheater 3 inlet and outlet sections are made up of T91 material.

9.2. Appendix B

9.2.1. Sub-critical boilers

9.2.1.1. Introduction

Mark Benson wanted to generate steam with no pressure limit and in 1922 the once-through Benson boiler was born. In the mid-twenties Siemens took over and developed the Benson boiler further. Siemens recognised that a once-through evaporator could be used for sub-critical as well as super-critical boilers. Once-through boilers used smaller diameter tubes with a thinner wall thickness which promoted heat transfer more quickly and resulted in more steam produced in an hour.

Natural circulation is sacrificed in order to increase the boiler size and efficiency.

The once-through Benson boiler uses the spiral configuration for the evaporation walls. This ensures a higher steam quality with lower temperatures because it transfers heat better with an incline of 15 degrees.

The feed water pump is used to create force circulation of the water through the boiler.

9.2.1.2. Advantages of a sub-critical cycle

- High overall efficiency even at part loads.
- Can be used in a pressure range of 10 – 35 MPa without modifying the evaporator
- Short start-up time
- Suitable for all coal grades.
- Economical
- Low-stress operation
- Flexible operating modes

9.2.2. Super-critical boilers

9.2.2.1. Introduction

After the Second World War there was rapid economic development. The need for a more efficient and economical power plant was on the increase. This would mean that there had to be an improvement in material metallurgy and water chemistry as well as combustion to make operations at super-critical pressures possible.

The critical pressure for water is 22.06 MPa. Above 22.06 MPa there is no two phase stage (water and steam).

9.2.2.2. Cycle

This cycle is based on a boiler that is on load and has passed the start-up phase.

Demineralised water is pumped into the boiler economiser. At the economiser exit the water makes its way down and enters the spiral wall where it transforms halfway up into a vertical wall; the water and steam mixture accumulates again at the top outlet headers. The water and steam mixture is split into four separating vessels where the steam and water separate. The separated water accumulates in the collecting vessel (only during start-up) and is fed to the economiser again where the steam enters superheater 1 then superheater 2 and then superheater 3. After superheater 3 the steam condition is superheated steam and is ready for the high pressure turbine. The steam is sent to the high pressure turbine where it transforms into mechanical work. After the high pressure turbine the steam loses a lot of energy and is sent back to the boiler where it enters Reheater 1 and then Reheater 2 to turn it again into superheated steam. The steam is now ready for the intermediate pressure turbine and the low pressure turbine.

After the low pressure turbine the water and steam mixture goes through the condenser where heat transfer takes place and the energy is extracted out of the water to transform it into a liquid again. The water is pumped through the low pressure heaters that use tap off steam from the intermediate and low pressure turbines to increase the water temperature before it enters the de-aerator. More heat transfer

takes place in the de-aerator with tap off steam from the intermediate turbine to increase the temperature of the water. The de-aerator serves as an inlet for the boiler feed pump, where the water is pumped through the high pressure heaters that use tap off steam from the high pressure turbine exit and intermediate turbine for more heat exchange before it enters the economiser to start the cycle again.

9.2.2.3. Advantages

- High plant efficiency, high efficiency coal utilisation, low emission (CO₂)

9.3. Appendix C:

9.3.1. Boiler history and development

9.3.1.1. Introduction

A boiler is a structure designed to convert liquid into gas (water to steam). Boilers are used in generating power, energy is converted from the coal to the water/steam. The steam is used to turn the turbine and the turbine turns the generator. The demand for high pressure boilers rises because high temperatures and pressure increase the enthalpy drop in the expansion of the turbine which results in a higher efficiency of the power generating plants and reduces the cost of production. Flue gas can also be used to heat up air for combustion purposes.

At the end of the 1800s the focus was on increasing efficiency and safety of boiler systems.

9.3.1.2. Fairbairn Lancashire boiler

The Lancashire boiler was developed in 1844 by William Fairbairn. The Lancashire boiler is a double fire tube boiler. The Lancashire is a horizontal boiler using natural draught to get the fire through the tubes. It also uses natural circulation and is reliable because of the simplicity and ease of its design. It is approximately 7-9 metres in length and 2-3 meters in diameter. In 1848 they installed Galloway tubes to improve the water circulation.

Coal is fed through the fire hole onto the grate. The flue gas circulates from the back to the bottom flue pass, as in this way the bottom of the shell is first heated. The flue gas then travels all the way to the front where it splits into two streams and passes through the back again. The flue gas exits at the back of the boiler and enters the atmosphere through the chimney. Dampers control the flue gas flow rate and also the coal burn rate, and in this way the steam flow is controlled.

The boiler is fitted with two safety valves. One is to blow steam off if the pressure gets too high. The second valve is to blow off the steam if the water level drops below the working water level.

The Lancashire boiler produces a good steam quality with low maintenance and operating costs. The Lancashire boiler is widely used in the sugar and chemical industries where large reservoirs of water and steam are required.

The boiler has steel rings that are welded or riveted to form two parallel cylindrical shells that run through the boiler and carry the hot flue gas. The flue gas cylindrical shells lie below the water level to accommodate a grate of sufficient area with minimum length. The flue gas cylindrical shells are larger in diameter at the front and this helps with the control of gas flow and the amount of air entering the grate.

The Lancashire boiler uses about six tons of coal per day (about 250kg/h or 0.0694kg/s).

9.3.1.3. Babcock and Wilcox boiler

The Wilcox boiler was developed in 1867 by George Herman Babcock and Stephan Wilcox. This water tube boiler has a larger heating area and allows a much better water circulation which creates a much safer boiler. The tubes are placed in the furnace area at about 10 – 15 degrees angle which promotes water circulation. These tubes are heated by the flue gas radiation and convection. Steam is collected in a drum situated above the tubes. This drum contains water and steam with a safety valve.

The boiler is situated horizontally and is only filled halfway with water. The header is connected with tubes that are in a zig zag configuration in the combustion chamber as flue gas passes over tubes to transfer heat. There is a tube that takes steam back to the combustion chamber for more heating, thus becoming superheated steam. This boiler had a much greater heat transfer and was more effective than previous boilers. The water tube boilers became the standard for future boilers. With multiple tubes in the boiler and better quality steel higher pressure and temperatures could be achieved. This has become the boiler of choice for steam turbines. The tubes are tilted 5 – 15 degrees to achieve a natural water/steam flow. There are two baffles to deflect the hot flue gases. The boiler has a mud box that is situated at the end of the down comer. The mud box collects any impurities and mud that are found in the water. A blow off cock is used to get rid of the impurities in the mud box.

9.3.1.3.1. Operating method

Coal is fed onto the grate through the fire doors. The coal is burned and the flue gas rises upwards. The baffle box redirects the flue gas and travels in a zig-zag manner to cover as much as possible of the tube area. The flue gas exits the boiler and enters the atmosphere through the chimney.

The boiler makes use of natural circulation. The water in the drum travels down to the bottom into the tubes and is heated. The water turns into steam and the density decreases, which results in more steam in the drum. The steam is returned to the furnace through tubes known as the superheater. The superheater increases the temperature of the steam and becomes superheated.

There is a damper fitted on the flue gas outlet. This controls the burn rate and load of steam produced.

9.3.1.3.2. Advantages

- Higher steam pressures, resulting in higher cycle efficiency.
- Easy repairable and all components can be inspected.
- Can expand and contract freely.
- Hot gases flow at the same angle as the water, thus resulting in a maximum heat transfer.
- Tubes are angled at 10 - 15 degrees, resulting in better water circulation.

9.3.1.3.3. Disadvantages

- Not suitable for water with a lot of impurities and mud, as these easily form blockages and result in short term overheating of tubes. The tubes burst and cause a lot of damage.
- Higher maintenance cost.
- Water levels must be watched, and a failure in the power supply even for a small period can result in the boiler overheating.

9.3.1.4. Stirling boiler

This boiler was designed by Alan Stirling and operated from 1883 to 1927. The first design contains two upper drums and one lower drum (mud drum). The upper drums were connected to the lower mud drum with slightly bent tubes to connect the periphery to the drum. This was further developed and a fourth drum was added on top. This was known as a four drum boiler.

In 1906 the Babcock and Wilcox Company took over the Stirling boiler company. The three drums are set on top in parallel and are connected to the mud drum below. The middle drum is slightly higher than the front and rear drum. The top side of the middle drum is connected to the front and rear drum for the steam so move freely into the middle drum. The middle and

rear drum bottom parts are connected to maintain the water level. The front drum is where water enters and is fed to the tubes. All steam is collected at the middle drum. The safety valve can also be located on the middle drum. There is a blow-off valve on the bottom of the mud drum to get rid of impurities.

The boiler has a manhole in one end of each of the four drums. This makes it easy to enter the boiler for inspection and repairs. The furnace is built out of bricks and has an arch-shaped roof. This design made it possible to use almost any type of fuel. The baffle walls were built from the mud drum to almost each top drum. These baffle walls prevent the flue gas from passing straight through the furnace without going through all the tubes.

9.3.1.4.1. Operating method

Coal is fed onto the grating where it burns. The flue gas then rises to the top and with the help of baffle plates is sucked down before it goes up again and leaves the furnace to exit through a chimney into the atmosphere. Just before the flue gas exits the furnace there is a control valve that controls the rate at which gas exits, and in this way the boiling rate can be controlled as well as the amount of steam needed.

The three top drums are only half filled with water to separate the water and steam. The drums are connected to each other at the top, while the right hand top drum and middle top drum are connected to maintain the water level. The left drum is where the water is added to the system. There is also a drum at the bottom that is called the mud drum. The mud drum is there to collect mud and impurities and then let it drain out. A blow off valve is situated on top to blow off impurities in the water.

If a superheater is needed, it would be fitted between the right hand side top drum and middle top drum where the flue gas is still very hot.

9.3.1.4.2. Advantages

- Easy load changes because of the small water content.
- Tube and drum dimensions made it possible for high steam pressures (160 bar)
- Design made it possible for any burner arrangement - horizontal or vertical.
- Full control of temperature.
- Free to expand
- Can handle water with impurities better

9.3.1.4.3. Disadvantages

- More difficult to manufacture
- Limited steam amounts produced.

9.3.1.5. Tube walled furnace

The demand for more steam increased rapidly because more power needed to be generated. This would require a larger furnace, which led to the development of the tube walled furnace. With this design the combustion and heat transfer areas are in the same place. This means more savings in construction costs and a massive increase in boiler size. In 1955 the first fully tube welded furnace was constructed. With a tube walled boiler the heat is transferred through radiation in the combustion process.

9.3.1.6. La Mont boiler

9.3.1.6.1. Introduction

The La Mont boiler is a once through boiler with lots of small diameter tubes running from bottom to top in a helical configuration. The tubes are spaced with a little spacing between them. The La Mont boiler uses forced circulation for the water and therefore requires powerful pumps.

The La Mont boiler was the first forced circulation boiler introduced by La Mont in 1925 and was used in America and Europe. This was the first high pressure steam boiler. The La Mont concept was safer, lighter and more effective. This boiler was also used in World War 2 navy ships.

The La Mont Boiler consists of a feed pump to force water through the boiler. The water is pumped through the economiser and then enters the steam drum. The circulating pump was used to pump the water from the steam drum into the vertical wall tubes that are used as the first part of evaporation.

The boiler makes use of helical coil tubes from top to bottom. A little steam drum was used to separate steam and water. The water was fed back into the boiler with a circulation pump.

9.3.1.6.2. Operating method

The La Mont boiler is a once through boiler. Air is sucked through the boiler to extract hot flue gases. The flue gases pass through the furnace and transfer heat to the tubes. Helical configuration is used for the tubes to maximise the heat transfer. The gas passes from bottom to the top and then exits the boiler. Later the air heater was introduced and the gas would pass through the air heater to transfer heat to be used for combustion. This heat was transferred without mixing clean air with flue gas. This resulted in better combustion.

The La Mont boiler consists of an economiser, superheater and a set of convection tubes. The feed water is pumped into the economiser which is situated at the top of the boiler. The water enters the separating drum where the water and steam separate. The water is pumped through the water walls where it is exposed to radiant heat transfer; it then enters the convection tube area where it is exposed to convection heat transfer. At the end of the convection element it is all steam and enters the separating drum. The steam enters the superheater circuit where it absorbs more heat and becomes super-heated steam.

The La Mont uses fuel oil, crushed coal or gas as heat source.

9.3.1.6.2.1. Air circuit

A blower is used to blow cold air through the air heater to transfer heat for combustion secondary air. The air heater receives heat from the flue gas exiting the boiler.

9.3.1.6.2.2. Water circuit

The circuit starts with a de-aerator; water is pumped from the de-aerator into the economiser which acts as a heat exchanger. The vapour water mix is separated with the help of a separating drum. The remaining water in the separating drum is fed into a circulating pump to pump it through a primary and secondary evaporating coils system to return it to the steam drum.

9.3.1.6.2.3. Steam circuit

The separated steam in the drum is then fed through the superheater for more heat transfer and higher temperatures are achieved. The steam is then fed to the turbine.

9.3.1.6.3. Advantages

- Uses forced circulation, which helps with a positive circulation and evaporation.
- There are high heat transfer rates because of small diameter tubes, thus more steam is produced per hour
- Low carbon tubes were available, resulting in a safer boiler
- There is a short start-up time, thus it is perfect for peak times.

9.3.1.6.4. Disadvantages

- It requires a small vertical drum outside with a small circulation pump.
- There is a limited amount of steam
- Temperature and pressure are limited because of metallurgical conditions