How can Attero increase the energy efficiency and R1 mark by steam cycle improvements of the plant?
Preface

This report of a 30 ECTS bachelor thesis is written for the Hanze high school of applied since as part of process engineering study. The thesis project was held partly in collaboration with Attero Wijster. During the thesis efforts have been spent on studying steam processes, study methods and writing English. I experienced it as a very intensive learning time in which I had to face multiple times my own ability limits. I found it hard to create a total clear study method before I even started my study and even more did I struggle with the influences of my dyslexia in my English hand writing skills. It took me 9 months of fulltime study to create this thesis report. This thesis project did I develop in consultation with Attero, their strong drive to optimize their waste to energy plant and my interest for steam cycles came together in a steam cycle optimization study which satisfied both desires.

I would like to thank my supervisors Eldert Michels and Marietta de Rooij for their time spent to discuss with me about my thesis subject and their help by sharing their knowledge and experience with me. Lastly, I would like to thank my roommate Wilbert van de Kamp for helping me learning how to write English.
Summary

The waste to energy plant of Attero is paid to process waste in a environmentally way. Part of this waste will be converted into useful energy, in forms such as heat and electricity. This type of waste recovery has been encouraged greatly by the government over the last decade. This type of rewarding creates a strong drive that urges old waste separation plants to transform into a waste to energy plant. Practice shows that this leads to a national increase of waste demand, which will also decrease the waste price. Also of influence are the local initiatives to separate waste on local authority collection points, and the economic crisis. These influences will result in a further decrease of the supply of available amount of waste for the WTE plants. Ultimately, this will lead to a reduced income for Attero.

Attero wants to deal with the increase in waste demand on the one hand, and the decrease in waste supply on the other in the following ways: Firstly investing in optimizing the plant in order to create a higher energy efficiency which will bring a higher energy profit; secondly, by buying international waste to ensure a reliable en sufficient large waste stream. Therefore, it is necessary to achieve permission to purchase international waste by meeting the requirements for a specific energy efficiency - the so-called R1 mark - .

Increasing the heat and electricity output of the plant can increase these two efficiencies, which can be achieved by improving the steam cycle of the plant. The main study question will be: How can Attero increase the energy efficiency and R1 mark by steam cycle improvements of the plant? To answer this question, the study consists of 3 main chapters, which describe the current state, (where process constraints will be found), potential optimizations and the calculated highest potential optimization suggestion.

The current steam cycle is susceptible to several changing variables. The influences of the ambient temperature create in summer conditions a turbine work loss of 9.2[%](3.3[MW]) due to the capacity shortage of the air condenser capacity, and this loss will be reinforced by the heat extraction of the pre-heater 10 and 20. The operation of the boiler is subject to the variation of the furnace load, temperature (<1150[°C]) and the conditions of the heat exchanger surfaces. A changing furnace load will be managed by water injectors and a proportional pressure controller, but will go hand in hand with exergetic losses. The heat exchanger surface condition (pollution), - in a clean condition - creates disproportional large heat extraction in the Drum and lead to an instable boiler balance. The inner drum heat exchanger (IDHE) is able to level this imparity and will beside this prevent that the flue gas temperature does not become too low (<225[°C]). The economiser (first heating step) is subject to the minimum surface temperature of 150[°C] and leads to the need to pre-heat the supply water. This pre-heating process consist of 3 heating stages (PH10 PH20 and supply tank) and will be heated with overheated drained turbine steam. The influence of this part process is that it lowers the plant capacity with 26.5[MJ]. This part process is controlled by the level indicator of the main condensate tank and creates strong fluctuating heat demand on the heating stages, which eventually leads to turbine work variation. Besides, the supply tank, while heating the supply water, also evaporates (on 150[°C]) a certain amount of supply water in order to degas, which creates a heat loss 528[kJ]. The external steam supply (ESS) will currently extract 3[kg/s] MP-steam for an external consumer by bleeding 1.35[kg/s] out of the MP-drain and 1.65[kg/s] out of the turbine driven supply pump. This extraction will decrease the turbine work 1.7[MW] but will supply a heat energy of 6.8[MJ]. This creates a current total plant efficiency (TPE) of 26.2[%] and a R1 mark of 0.632.

The four optimization suggestions will be put to use to improve the steam cycle efficiencies by solving the steam cycle constrains or process boundaries. 1. Enlarging the air condenser capacity would solve the turbine work loss, due to high ambient temperatures, and solve the turbine work loss reinforcement due to the pre-heating heat energy demand variation. 2. Enlarging the MP-steam turbine drain by a new diaphragm valve set point. This valve will increase the outlet pressure of the high-pressure turbine in order to create MP-steam conditions on the current LP-drain. The benefit would be a larger MP-drain (50.1[kg/s] vs. 3.99[kg/s]) for external steam supply. 3. A change of managing the supply water flow through the pre-heater stages. By shifting the leading controller from mean condensate tank level indicator to the supply tank level indicator, it would possibly create a more stable and constant batch flow. 4. The following fourth optimization suggestion is selected as the
optimization with the highest potential for increasing the efficiencies. This optimization is extensively analyzed in order to provide a founded conclusion, which includes the technical feasibility and process benefit according to the TPE and R1 mark. Optimization 4 proposes a pre-heating process optimization, which is necessary to decrease the internal steam extraction for pre-heating, in order to create more space on the MP-drain for ESS and the ability to create more turbine work. An optimized operation setup in which the supply-tank heater replaces the pre-heater 20 and the supply tank heater is replaced by the IDHE will realize this. The new pre-heater set-up will shift a heat energy demand of 10.35[MJ] from the MP-drain to the drum. The supply-tank will heat the supply water from 80[°C] till 110[°C] with LP-steam where after a significant smaller quantity of heat will leak due to the degassing process (59.7[kJ]). The IDHE heat extraction out of the Drum in order to heat the supply water up to 150[°C] is larger then demanded. The heat transfer coefficient in combination with the surface and the logarithmic average temperature difference between the drum water and the IDHE water (∆T_{ln}) will lead to a heating capacity of 11.25[MJ]. This extra heat extraction out of the Drum will lead to a new boiler load balance and a need to increase furnace capacity. To realize this furnace capacity enlargement, the flue gas flow will increase with 14[kg/s] and the furnace temperature will be decreased to manage exactly on the demanded boiler capacity and the flue gas output temperature (>225[°C]). The boiler load balance will be occurred by enlarging the drum heat transfer surface and slightly decrease of the heat transfer coefficient of the OVO and ECO. A negative outcome of this furnace capacity enlargement is that the flue gas flow and the lower furnace temperature will decrease the boiler efficiency with 1.5[%] to 80.5[%].

The consumer side of the steam cycle will - due this pre-heating optimisation - no longer need to bleed 2.8[kg/s] steam out of the MP-drain to pre-heat. This would make generating 3[MW] electricity or enlarging the ESS with 2.8[kg/s] possible. However, due to the decrease of the boiler efficiency, in both options the TPE nor the R1 will increase. In order to increase the R1, mark is it necessary to increase the ESS supply to a steam supply of 12[kg/s]. The TPE will in that case increase 6.3[%] to an efficiency of 32.5[%].

Apart from the efficiency study, is it noteworthy that the capacity increase of the furnace will lead to a waste handling capacity enlargement, which will benefit the financial income.

Concluding: it is evident that optimizing the pre-heater process will lead till a TPE increase of 6.3[%] and has an unchanged R1 mark at that point. The final recommendation is: optimizing the pre-heater process when there are enough external steam consumers to extract a minimal flow of 12[kg/s].
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1. Introduction

The Attero Waste to energy plant processes waste generated by society in an environmentally friendly way, and are being paid for processing the waste. Part of the waste is converted into useful energy forms such as heat and electricity. In the last decade, the Dutch government encouraged to transform the old type of waste handing plants (Called Mechanical waste separation plants (MWS plants) into energy creating waste handling plants(WTE plants). The WTE plants with an electrical efficiency higher than 26% receive a grant of 1.45 ct/KWhe for a period of 10 years\(^1\). Practice shows that this way of rewarding creates a strong drive towards optimizing old WTE plants and building new plants. This will result in the increase of national waste demand, which causes a decrease of the waste price because of the competition between the WTE plants (from 100E/ton in 2003 to <60E/ton in 2009)\(^2\). Hence, a reduced income for Attero. In addition, the economic crisis will simply stagnate the amount of company and public waste supply. Also of influence are initiatives to separate waste on local authority collection points. This will result in a decrease of the supply of available amount of waste for the WTE plants.

Therefore, Attero wants to deal with the increase in waste demand, on the one hand, and the decrease in waste supply, on the other, in the following ways:
1. Invest in optimizing the plant to increase the energy efficiency. This optimization will bring a higher energy profit.
2. Buy international waste to ensure a waste stream which is stable and sufficiently large.

Therefore, it is necessary to achieve permission to purchase international waste by meeting the requirements for a specific energy efficiency, the so-called R1 mark \(^3\).

These efficiencies can be increased amongst others by increasing the heat and electricity output of the plant. This can be achieved by improving the performance of the processes of the steam cycle of the plant. Therefore, the research question is: How can Attero increase the energy efficiency and R1 mark by steam cycle improvements of the plant? To answer this research question, the following approach is used:

1. What is the current performance of the processes?
2. What is the best option to improve the current performance?
3. Which efficiency increase can be achieved with this option?

The structure is as follows: chapter 2 and 3 will introduce the study subject, the basic steam process and the different types of important steam cycle efficiencies in order to get to know the WTE plant steam cycle. Chapter 4 will clarify the current state of the steam cycle by studying the part processes on technical operation. The chapter will be split up in two paragraphs, which will be separated into the partial processes by the influences where they subject to, also known as internal and external influences. These two paragraphs will result in large overview of part process constrains and influences which are responsible for the current cycle performance. Chapter 5 will sum up the current process efficiency constrains and enrichments as base for the suggested optimizations where after will be decide which suggested optimization has the best potential for plant efficiency improvements. Chapter 6 will calculate the performance of the steam cycle in the optimized state in order to answer the last sub-question; Which efficiency increase can be achieved with this option? At last, the final conclusion will be drawn in chapter 7.

\(^{1}\) De Lange, T.J., Kosten Duurzame Eletriciteit (Petten, ECN, 2003)

\(^{2}\) De Vries, J. et al. Recyclingindustrie (Zaltbommel, 2009)

\(^{3}\) R1 state will be pointed in paragraph ‘Efficiencies’ page 5
2. Current steam process description

The WTE plant of Attero, which was built in 1995, is in its base an uncomplicated steam process (fig.1). This includes a Boiler, High pressure turbine, Low pressure turbine, Condenser and Supply water pump. However, in practice the steam cycle is much more complicated. Due to the extra components which will optimize the steam process, as shown in fig.3, it can be seen that beside the standard components this cycle also consists of pre-heaters, turbine bleeds, external steam delivery possibilities and a steam driven pump. The following paragraph will give a work explanation of the current steam cycle process.

The steam cycle process design guide (see fig. 3).

In the three boilers, the supply water absorbs the heat, which is generated by the burned waste. The boilers are designed to transform the supply water in three steps into overheated steam(400°C and 40bar). These three heat exchange steps/components (see fig.2) which are based in the boilers are called economizer(Eco), Drum and Over-heater(Ovo). The boiler design (fig.2) is based on the cross flow principle, which means that the entering water(cold composite) flow will start at the end of the boiler process en will move step by step closer to the heart of the combustion chamber. The overheated steam will, passing the High Pressure bar (HP-bar), enter the turbines where the steam will expand to a low pressure and temperature. Due to the isentropic expansion, the turbine will generate rotating mechanical work, of which the movement is being transformed to electrical power. The condenser will cool the relatively cold steam, until it is completely condensed. This is necessary to create the under-pressure at the entering side of the condenser⁴. The condensate flow will be pumped to the pre-heaters, to get pre-heated by two turbine steam bleeds⁵. The final preheat treatment takes place in the supply tank. The condensate will there be heated (with Mid-pressure steam) until the boiling point is reached(150°C), in order to make a degassing process possible and create a required supply water temperature. The preheated condensate, now called supply-water, will then enter the supply pumps, which will pump the water into the boiler. These different types of pumps (steam and electric driven) bring the possibility to create more process steam on a mid-pressure state with the steam driven pump, whereas the steam bleeds on the turbine could not supply enough steam on that particular specification.

The possibility for external steam supply (on mid-pressure state) can be created through the use of the turbine bleed, steam driven supply pump(TSP) and a high pressure steam reducer, or a combination of these techniques.

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⁴ see paragraph 4.1.1 Ambient Temperature Condenser
⁵ Bleeds are drains which tap steam on three possible stages out of the turbine
Figure 3. Principle Diagram Attero Steam cycle
3. Efficiencies

For the Attero WTE plant, waste handling is their main goal. Due to that goal, the efficiency of operating is highly important. Even more now that it is clear that the waste prices are still decreasing and the caloric energy density of the burnable waste will decrease as well, by the influence of the increase of segregating waste on local authority collection points. In this study, two types of efficiency are highly important. The two types are the total plant efficiency and the R1 mark, and both give an expression of usefulness. The following two sub-paragraphs will describe how the efficiencies will be calculated and what the meaning of the expressions are. The third sub-paragraph will clarify used method of this study.

Total Plant Efficiency

The total plant efficiency (TPE) is based on a basic principle, which imagines the plant as a black box, and divides the effective energy outcome by the total energy income to calculate the efficiency.

\[
\eta_{\text{Total}} = \frac{Q_{\text{out}}}{Q_{\text{in}}} 
\]

Eq. 1 Total Efficiency

\(Q_{\text{out}}\): Outgoing energy of the WTE Plant
\(Q_{\text{in}}\): Inputted waste in to the furnace

The \(Q_{\text{out}}\) would be in this case: the turbine work, the external steam supply (ESS) and the generated work out of the turbine driven supply water pump (TSP). The \(Q_{\text{in}}\) is the energy which will occur by burning the waste. However, it is hard to calculate the waste energy input. In this study, it will be calculated back from the absorbed heat in the boiler. Because the static boiler efficiency is known\(^6\) and the necessary heat for a full load steam process as well. In that way, Eq. 1 will be rewritten to the following equation (Eq.2):

\[
\eta_{\text{total}} = \frac{\left( Q_{\text{turbine}} \eta_{\text{generator}} + Q_{\text{ESS}} + \left( Q_{\text{TSP}} \eta_{\text{TSP}} \right) \right)}{\left( \frac{Q_{\text{boiler}}}{\eta_{\text{boiler}}} \right)}
\]

Eq. 2 Derived total efficiency eq.

\(Q_{\text{turbine}}\): Thermodynamic used energy in the turbine [kJ]
\(Q_{\text{ESS}}\): Inputted waste in to the furnace [kJ]
\(Q_{\text{TSP}}\): Turbine driven supply water pump work [kW]
\(Q_{\text{boiler}}\): absorbed heat out of the furnace by the heating steps [kJ]
\(\eta_{\text{boiler}}\): Boiler efficiency [%]
\(\eta_{\text{generator}}\): The efficiency of the electricity generator [%]
\(\eta_{\text{TSP}}\): Turbine driven supply water pump efficiency [%]
\(\eta_{\text{boiler}}\): Boiler efficiency [%]

The current total plant efficiency is calculated in chapter 4 paragraph 2.3 ‘current extern steam’ supply. The value of the current TPE is 26.5%

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\(^6\) Attero plant boiler description; efficiency of 0.823%
R1 Mark
The R1 mark is one of the guidelines created by the European commission to provide a legal certainty and a level playing field for energy producing waste power-plants, which results in the fact that only the certified WTE plant’s can trade international waste. The chief reason for this R1 efficiency is that the European commission wants to encourage innovation in the waste incineration industry. The R1 efficiency baseline for WTE plant’s in operation is 0.60 when build before 01-01-2009 and a baseline of 0.65 when the WTE plant is build after 31-12-2008. This R1 efficiency will be calculated with the same numbers as the total efficiency, only this number does not compare to the $\eta_{\text{Total}}$ because the R1 efficiency is strictly speaking not an expression of efficiency in physics, but an indicator for the level of recovery of energy from waste. Equation 13 will underpin that it is an indication for the level of recovery, by dividing and multiplying the equation with constants.

$$\eta_{R1} = \frac{E_p - (E_f + E_i)}{0.97(E_w + E_f)}$$

Eq. 3. R1 Mark

$E_p := E_{\text{elec}} + E_{\text{heat}}$

$E_{\text{elec}} := (Q_{\text{turbine}} \cdot \eta_{\text{generator}})^{2.6}$

$E_{\text{heat}} := [Q_{\text{ESS}} + (Q_{\text{TSP}} \cdot \eta_{\text{TSP}}) + Q_{\text{degass}}] \cdot 1.1$

$E_p$: means annual energy produced as heat or electricity. It is calculated with energy in the form of electricity being multiplied by 2.6 and heat produced for commercial use multiplied by 1.1 (GJ/year)

$E_f$: means annual energy input to the system from fuels contributing to the production of steam (GJ/year)

$E_w$: means annual energy input in the treated waste calculated using the net calorific value of the waste (GJ/year)

$E_i$: means annual energy imported excluding $E_w$ and $E_f$ (GJ/year)

0.97: is a factor accounting for energy losses due to bottom ash and radiation

Source: European commission R1 Guidelines

The equation shows that the energy produced minus the extra energy input, all dived by the waste energy plus the extra energy input multiplied with a correction factor. This equation encourages to create steam to supply for external and internal\(^7\) use, due to the reason that this type of energy supply has a far higher total efficiency. After all, the total used enthalpy range is much bigger than the enthalpy drop in the turbine (HD-external supply vs. Turbine is 2989.4 vs. 910.5 [kJ/kg]). The constants will level the giant differences between the $E_{\text{heat}}$ and the $E_{\text{elec}}$ by multiplying the $E_{\text{elec}}$ with 2.6 and the $E_{\text{heat}}$ with 1.1, but it is not large enough to recommend only the use of the turbine.

The variables $E_f, E_i, E_w, E_f$ are annual averages out of the Attero database\(^8\), only $E_p$ is variable in this study.

After all, the differences between the two efficiency expressions are not large. Only their aim is different. The TPE is a realistic value of affectivity, while the R1 state want to encourage the use of steam without making electricity out of it by multiplying the variables with constants. But in case of an increase or decrease of the energy output, both of the efficiency expressions change in the same direction.

The current R1 Mark is calculated in chapter 4 paragraph 2.3 ‘current extern steam supply’

The value of the current R1 mark 0.632

\(^7\) Internal use of steam will mean the $E_p$ heat energy for own purposes like TSP and Degassing

\(^8\) See appendix 2.1. R1 mark data
**Method description**

To study how the R1 mark and the TPE can be increased for the WTE plant, there will be made a separation between the current state operation method and the optimized state operation method. The current state of the plant is investigated on the basis of data of the plant (see appendix 1). This data (sheet) will supply the current process values. Which bring the possibility to calculate the current state efficiencies (TPE, R1mark). For this current state study (ch.3-4), there will be assumed that the steam cycle will constantly operate on full load and got an boiler efficiency ($\eta_{\text{boiler}}$) of 82.3%. This will uttering in a constant full load operating boiler. Which got as consequence that in the TPE equation 2 only the upper side of the equation will be responsible for the total efficiency. For the R1mark will this lead to a constant Ew value what will result in an equation in which only Ep is variable. The influence of the ambient temperature will be calculated in multiple stages for a couple specific part processes. However in the remaining study chapters will the ambient assumed to be constant on 10°C. The Turbine work is variable by multiple reasons as; output pressure, amount of turbine drain leak, and normal would also the isentropic turbine efficiency affect the turbine load. This should occur as a load reinforcement as well as debilitation by the turbine load change. Only in this study will the isentropic efficiency not change according to the load. The reason for this is that the isentropic efficiency is strongly depending on the quality and design of the turbine, which is unknown.

The differences in the optimization study method (for chapter5-6) instead of the current state study method are; that the boiler efficiency is calculated and the optimization are calculated at a constant ambient temperature.
4. The Current Steam Cycle State
The steam cycle of the WTE plant of Attero will be described in a number of sub-processes which describe their specific role within the steam cycle process. These sub-processes will be split up in two paragraphs which divide the sub-processes in processes which are particularly subject to the external variable influences, as well on internal variable influences. The motivation of separating the sub-processes in these two variables will be done in order to is to clarify the impact of these influences. What for the external influences inter alia would be the influences of the ambient temperature, boiler load and boiler pollution. Hereafter the internal process influences will describe where their process variables depend on (preheating, degassing and extern steam supply) in order to analyze what the current performance of the processes are.

4.1. External dynamic process influences
It is likely that a giant process is susceptible to changing variables. Instead of most energy plants, the WTE plant of Attero has a number of strongly deviating variables. For example, the Attero plant is based in an open field without stable cooling facilities, and the caloric energy density of the fuel depends heavily on the waste quality compared to the normal power-plants. In the following paragraphs, these variables will be described inter alia.

4.1.1. Ambient Temperature Condenser
The Attero plant is built in an open field right next to the VAM landfill. As a result of the geographic location, it needs an Air Condenser (AC) instead of a liquid chilled condenser, which is less susceptible to weather variations. On a summer day, the ambient temperature can raise to an average temperature of 30 degrees Celsius instead of an average temperature of 5 degrees on a winter day. These differences will have their effect on the cooling capacity of the condenser. The type of heat exchanger does not change that effect, because only one temperature changes in the air condenser (Tair) (see fig. 4). In that way, it would not make a difference whether it is a cross flow heat exchanger or another type of heat exchanger. The following paragraphs will calculate the influence of this varying temperature, which makes it possible to summarize the influence.

The Air condenser:
The main function of the Air condenser is to generate a pressure drop between the turbine output and the condenser entrance. The pressure drop will increase the turbine capacity, which create the ability to expand the steam to a lower pressure. The theory behind this physical reaction is that phase change from steam to water will occur isothermal. The extreme volume decrease (Vturb.out vs. Vcondensate is 4.67 vs 0.00102 [m3/kg]) will create the pressure drop, due to Clausius-Clapeyron equation (see eq. 4). This will occur because the face change from steam to condensate is coupled with a entropy decrease and will create an increase of the entropy difference (ΔScon). Knowing that the R, T and dT are constant in this calculation, will clarify that the condensate pressure (pcon) will decrease.

\[
\frac{d \ln (\frac{p}{p^0})}{dT} = \frac{\Delta H_{con}}{R \ast T^2} = \frac{\Delta S_{con}}{R \ast T} \quad \text{Eq. 4}
\]

---

9 Power plants usually based at giant rivers or sea’s so they can cool with a stable low temperature
10 VAM is a Dutch landfill Company called “Vuil Afvoer Maatschapj”
The condensate pressure, which will be achieved in the condenser, is an outcome which is directly related to the condensate temperature, according to the derived Ideal Gas Law (eq.5). This will mean that the air condenser provides the turbine outcome (under)pressure. The created under-pressure will bring the possibility for the turbine to expand the steam till a lower level and create a turbine work increase.

\[
P \cdot V = \text{Constant} \quad \text{Eq. 5}
\]

However, the air condenser (AC) operates by blowing ambient air over the AC surface (A). This creates a serious dependence on the ambient temperature, for the incoming air temperature determines the out-coming condensate temperature. The specific condenser equation (Eq.8) describes that.

Eq.6 and Eq.7 will display that in a condensation process, in which the cooling capacity demand \( Q_{\text{AC}} \) is constant and only the ambient temperature is changing, the condensate temperature will \( T_{\text{condensate}} \) react like the ambient temperature by rising or decreasing in a similar way.

\[
\Delta T_{\text{ln}} := \frac{(\Delta T_2 - \Delta T_1)}{\ln(\Delta T_2 / \Delta T_1)} \quad \text{Eq. 6}
\]

\[
\Delta T_2: \quad T_{\text{condensate}} - T_{\text{air\_in}} \text{ (see fig.4) \{°C\}}
\]

\[
\Delta T_1: \quad T_{\text{condensate}} - T_{\text{air\_out}} \text{ (see fig.4) \{°C\}}
\]

\[
Q_{\text{AC}} := M_{\text{Air}} \cdot c \cdot \Delta T_{\text{Air}} \quad \text{Eq. 7}
\]

\[
M_{\text{Air}}: \quad \text{Air flow over the condenser [kg/s]}
\]

\[
c: \quad \text{Specific air constant}
\]

\[
\Delta T_{\text{Air}}: \quad \text{the temperature difference between the in/out coming air flow \{°C\}}
\]

\[
Q_{\text{AC}} := K \cdot A \cdot \Delta T_{\text{ln}} \quad \text{Eq. 8}
\]

\[
K: \quad \text{Specific condenser constant [kcal/m².h.K]}
\]

\[
A: \quad \text{Condenser surface [m²]}
\]

\[
\Delta T_{\text{ln}}: \quad \text{The natural logarithm delta average temperature \{°C\}}
\]

The request would be to manage the \( \Delta T_{\text{ln}} \), so it could provide a smaller \( \Delta T_{\text{ln}} \) in times of high ambient temperatures. This would create a lower condensate pressure in order to participate on the sub-aim of the AC to enter as close as possible to the maximum under-pressure, which is given by the turbine specification (0.08[Bara]). Options which would anticipate on that aim could be for example an extension of the condenser surface (A) (Eq.8). This could provide an \( \Delta T_{\text{ln}} \) lowering. Furthermore, an air flow increase \( (M_{\text{Air}}) \) could bring the \( \Delta T_{\text{Air}} \) down (Eq.7) which would cause a decline of \( \Delta T_{\text{ln}} \) (eq.6-8). Those examples would result in the possibility to keep a more constant and low condensate pressure. However, the Attero plant does not have overcapacity on their air condenser surface or air fans, except when the ambient temperature lowers down to -5 Celsius. The controllers will then lower the air flow to retain the maximum under pressure.

To calculate the realistic condensate temperature, it is necessary to transform Eq.8 to the following expression (Eq.9), which will provide the incoming condensate temperature. These temperatures also represent a specific enthalpy which enables the turbine to extract more energy out of the steam. Equation 10 will show that it results in a turbine energy increase \( Q \).
\[
T_{\text{condenser}} = \frac{T_{\text{air\_in}} - T_{\text{air\_out}} e^{\Delta T_{\text{log}}}}{1 - e^{\Delta T_{\text{log}}}}
\]

\[
Q := M_{\text{steam}} (h_{\text{steam\_in}} - h_{\text{steam\_out}})
\]

Eq. 9. Condensate temperature form.

Eq. 10. Turbine Energy

\[M_{\text{steam}}: \text{Maximum steam flow}\]
\[h_{\text{steam\_in}}: \text{Enthalpy of entering steam in turbine}\]
\[h_{\text{steam\_out}}: \text{Enthalpy turbine output}\]

The different ambient temperatures are visualized versus the condensate temperature (fig. 5). The outcome shows that under the ambient temperature of -5 [°C], the condenser capacity will be reduced. This will tell that below the -5[°C] the maximum turbine under pressure is reached. The temperature of 39[°C] is related to the pressure of 0.08[Bara]. The consequences of the high condensate temperatures become evident in figure 6.

![Figure 5 Condensate temp. vs. ambient temp](image)

![Figure 6 Turbine work vs. ambient temp](image)

<table>
<thead>
<tr>
<th>T\text{ambient [°C]}</th>
<th>-15</th>
<th>-10</th>
<th>-5</th>
<th>0</th>
<th>5</th>
<th>10</th>
<th>15</th>
<th>20</th>
<th>25</th>
<th>30</th>
</tr>
</thead>
<tbody>
<tr>
<td>T\text{condensate [°C]}</td>
<td>39</td>
<td>39</td>
<td>39</td>
<td>43.7</td>
<td>48.7</td>
<td>53</td>
<td>58</td>
<td>63.7</td>
<td>68.7</td>
<td>73</td>
</tr>
<tr>
<td>P\text{C. Comp [Bar]}</td>
<td>0.070</td>
<td>0.070</td>
<td>0.070</td>
<td>0.090</td>
<td>0.156</td>
<td>0.143</td>
<td>0.182</td>
<td>0.236</td>
<td>0.295</td>
<td>0.355</td>
</tr>
<tr>
<td>h\text{C. Comp. in [kJ/kg]}</td>
<td>2367.04</td>
<td>2367.04</td>
<td>2367.04</td>
<td>2376.38</td>
<td>2386.27</td>
<td>2394.74</td>
<td>2404.54</td>
<td>2415.63</td>
<td>2425.28</td>
<td>2433.53</td>
</tr>
<tr>
<td>W_{\text{turbine [MW]}}</td>
<td>46.57</td>
<td>46.57</td>
<td>46.57</td>
<td>46.05</td>
<td>45.62</td>
<td>45.2</td>
<td>44.71</td>
<td>44.16</td>
<td>43.68</td>
<td>43.27</td>
</tr>
</tbody>
</table>

Table 1. Condenser data

Figure 6 also clarifies that the balance point is reached at the ambient temperature of -5 Celsius, because the turbine power is constantly beneath -5[°C]. Above -5 Celsius, the turbine work decreases. The condenser capacity shortage will lead to a turbine loss of 9.2% at an ambient temperature of 30 Celsius, which stands for a 3.3 [MW] turbine loss (figure 5-6).
4.1.2. Ambient temp. Influences on Pre-heater

The influence of the ambient temperature does not end at the condensation process step. Due to the temperature of condensate out of the condenser, it will also affect the first pre-heat process step. The magnitude of the impact of the ambient temperature will be pointed out in the following paragraph.

The process step of preheating sounds unnecessary in the first place, because after all the boiler has enough capacity to heat the supply water of a lower temperature. However, the pre-heating step is necessary for the following reasons; the economizer (first step in the boiler) has a minimum temperature of the pipe surfaces which are in contact with the flue gasses (145[°C]). Pumping a colder fluid would simply create a lower pipe surface temperature, what would lead to condensing flue gasses on the outer pipe surface, which causes low temperature corrosion due to the high pollution grade of the flue gasses. Besides this, the minimum economizer temperature is the second reason why the preheating is necessary, the need to degas the condensate flow before it may enter the boiler. By heating the condensate to a certain boiling point. Degassing\(^\text{11}\) will protect the boiler and turbine for cavitations for to the effect of harmful gasses.

The pre-heating process consists of three steps (see fig.7&8), and each step consists of a different heating steam flow. The last step is the supply tank in which the condensate will reach the boiling point (150[°C] at 4.8 [bar]), which makes degassing possible. The varying incoming condensate temperature has an effect on the first two heating steps (PH-10, PH-20). It is likely that the heating demand also will vary due to varying condensate temperature. But next to heat demand, will the condensate temperature also effects the heat extraction controlling step. This influence will be explained in the following text.

The pre-heater will get its heat energy supply from three turbine bleeds, and this will automatically result in a self reinforcing reaction. To clarify this effect, it will be described in two examples.

\textit{Example 1}; the condensate temperature would rise. This will decrease the heat demand of the first pre-heater, which will result in a flow lowering out of the ‘extra low pressure’ (ELP) steam bleed (see fig.1). However, the lowering in steam bleed will result in an increase of turbine work and condenser load. Due to shortage of condenser capacity, the increase of the condenser load create condenser temperature increase, which is negative.

\textit{Example 2}; the condensate temperature would decrease. Due to this decrease, the pre-heater heat demand would rise, which results in an increase of steam extraction out of the ELP steam bleed. This would result in a decrease of the steam flow through the last part of the turbine and creates a decrease of potential condensatable steam. Finally, this would result in a further decrease of the condensate temperature. Beside the vary on the steam flow extraction demand, would in both examples the pressure also vary on the steam bleeds due to the chancing condensate temperature. For example 2, the condensate temperature decrease would create a pressure decrease on the LP-turbine output. This would result in a larger pressure expansion range over the LP-turbine. Where after this would lead to a pressure increase on the ELP and the LP turbine. In their place, this would create a further lowering on pre-heater steam extraction flow for PH-10 as well PH-20.

These self reinforcing processes are hard to calculate, but what can be conclude is that in full load on an ambient temperature of 10[°C] the heating demand is 7.3 and 8.3[MJ](tab.2) in the first and second pre-heater (PH10, PH20). An increase of the ambient temperature will create an ELP bleed decrease of 2[kg/s](tab.3), what will result in a flow enlargement trough the last part of the LP-turbine and air condenser. This enlargement will underpin the significance size of the self reinforcing.

---

\(^{11}\) Condensate degassing will be described in paragraph 2.1
The consequences on the pre-heat demand due to the varying of the ambient temperature will be pointed out in table 3 and figure 9. These results display that the function of the first pre-heater (PH-10) in summer conditions (30°C) will be almost turned off. What is likely, due to the heating range of PH-10, which has the maximum of 94°C against a incoming condensate temperature of 74°C (tab.1)

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Q_{heating} [MW]</th>
<th>PH10 flow [kg/s]</th>
<th>PH20 flow [kg/s]</th>
<th>SuplyTank flow [kg/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-20</td>
<td>30.4</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
<tr>
<td>-25</td>
<td>30.4</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
<tr>
<td>-10</td>
<td>30.4</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
<tr>
<td>-5</td>
<td>29.1</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
<tr>
<td>0</td>
<td>27.8</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
<tr>
<td>5</td>
<td>26.5</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
<tr>
<td>10</td>
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<td>15</td>
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<td>4.31</td>
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<td>4.3</td>
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<td>20</td>
<td>22.4</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
<tr>
<td>25</td>
<td>21.3</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
<tr>
<td>30</td>
<td>20.0</td>
<td>4.31</td>
<td>3.6</td>
<td>4.3</td>
</tr>
</tbody>
</table>

4.1.3. **Boiler Input**

The burnable waste, which is the energy resource of the Attero plant, does not have stable caloric energy characteristics. It is a fact that waste burning is a dynamic process and depends to a great extent upon the quality of the waste. However, this study does not include the waste supply into the combustion furnace. This means that the fluctuation of the waste energy supply will be handled as a heat drop in the boiler. In this research, the boiler input is the available heat out of the black box, which is called combustion furnace. This paragraph will clarify how the vary of furnace heat energy supply will be managed in the boiler.

The plant operators try to manage the combustion furnace heat as closely as possible to the requested heat energy demand. Nevertheless, it is not possible to deliver a constant amount of heat out of three fluctuating furnaces, due to that fact there are three automatically acting safety systems implemented to prevent the steam cycle for steam excess, known as the Water injection system and the proportional...
process controller (PPC) system. The water injection system is based to provide grossing the maximum steam condition (called “overheating”) (400°C, 41 Bara). The PPC valve will provide exceeding the lower steam condition limit (325°C, 41 [bara]) which is called undersized overheating. The third system will prevent fluctuations in the boiler balance due to the pollution level of the boiler surfaces. The way of operating will be described separately in the following sub-paragraphs.

Excess of the overheating process
There are two causes for excessive high steam quality. The first cause is the possibility that there is a change of supply water, the second cause is an excessive increase of boiler load. Those two system performances could both bring a dangerous high steam condition out of the overheater into the steam network. A logical solution would be to stop absorbing heat out of the flue gasses. However, it is not possible to turn the heat-exchangers away from the heat or blow the flue gasses with a higher temperature into the flue gas filters. The boiler is stuck to the fact that the exchanger surface absorbs heat as long there is a temperature difference and a contact time. Therefore, there are water injectors based after the overheater (OVO) steps, which will cool the steam condition down to the demanded conditions (+/- 400°C) by injecting water. This type of regulating of steam temperature is highly effective, and has the possibility to react very resolute. The negative side of this solution is that the injector will create an exergy loss (E_loss) when it injects water into the overheated steam.

The following example (see figure 10) will show how a realistic heat excess will be managed with the use of the water injectors. The total amount of absorbed energy in the over-heater will be calculated with Eq.11. The necessary amount of injected water will be determined in equation 12. With the use of equation 13, it is possible to calculate the exergy-loss [kJ/kg]. Multiplying the E_loss with the steam mass flow will show what the heat-loss [kJ/s] is.

Eq.11 and 12 will determine the necessary amount of injected water. Eq.9 will solve the loss of exergy:

\[
Q_{\text{overheat total}} = M_{st} (h_2 - h_0)
\]

\[
Q_{\text{overheat total}} = (M_{st} h_1 - h_0) + [M_{water\text{-injection}} (H_1 - H_{water})]
\]

\[
E_{\text{loss}} = (S_2 - S_1) T_1
\]

In the suggested example, shown in figure 10, is an over-heated steam condition of 500°C (isobaric overheated) suggested. The injection of supply water will provide a cooling step by the same line, due to the “1st law of Gay Lussac” which states that a temperature decrease will simultaneously bring a volume decrease. This will suggests a decrease of the energy density of the steam and will expressed as an entropy [kJ/kg.K] decrease!

Result example: a steam mass flow of 21.3[kg/s]\(^{12}\) will be over-heated to 773 Kelvin (at 41 [Bara]). It should be cooled back to 673 [Kelvin] by injecting an amount of water 1.9 [kg/s], which creates an E_loss of 90.56[kJ/kg] ((7.08-6.76)*283), and a heat-loss of 1929[kJ/s].

\(^{12}\) This is a realistic load for one boiler
This exemplary heat loss indicates that injecting water can act adequate and has a large capacity. It shows that this type of safety system operates properly.

**Undersized over-heating**

When, due to an low heat capacity in the boiler, the steam conditions are too low, it is necessary to create an process step which will increase the steam conditions. The implementing of a smother valve, called “PPC” (proportional pressure controller) before the “high pressure bar”, will ensure the demanded characteristics. The theory behind this valve will be described in the following text.

The PPC valve

The task of the PPC valve is rather clear, by smothering the steam flow, the steam conditions will increase to the demanded steam temperature. The theory behind this valve is as follows: the PPC will act accurately on temperature differences. The steam pressure already reached the 41 [Bara] at the steam drum. The possibility that the steam might not get heated up far enough, will be solved by the PPC characteristics. The valve will smother the entering steam “isochoric” to a higher pressure. This pressure increase will, due to the 2nd law of Gay Lussac, create the temperature increase to 400 [°C], which is displayed in an entropy increase as well (fig.11).

At the other side of the valve, the high pressure (>41[Bara]) will be reduced isothermally to the demanded 41[Bara], and the requested temperature will be achieved. This process is displayed in figure 11. Although the temperature is reached, there is a small steam flow decrease due to the smother valve. However, in practice, the common pressure differences are small, which shows that the steam flow decrease is negligible in this process.

**Boiler pollution**

The boiler may be seen as one big heat exchanger which is constantly in contact with the flue gasses. The furnace heat will be transported by the flue gasses (hot composite) which still inhibit a large amount of ashes in the boiler. It is obvious that the “pipe walls” (the heat exchanger surfaces for all the heating steps) in the chimney will become polluted. There are a couple of different cleaning systems used to ensure the heat transfer, wherein each system has its own interval. Once in two weeks, the boiler will be cleaned with explosives, in order to clean the pipe walls entirely. After the explosive cleaning process, the common heat transfer will act much faster, and the specific thermal conductivity(k) constant will increase extremely due to the removal of ashes. The result of this is, that (on a full load furnace) the pipe walls which are the most closely placed to the furnace (drum heaters and Over-heater) will absorb an disproportionally large amount of heat, which will lead to large instable conditions and steam excesses (>400C), while at the end of the chimney the heating capacity decreases for the economizer(Eco). Fourier’s Law (eq.14) will describe this process behaviour.

\[ Q := \frac{(k \cdot A \cdot dT)}{s} \]

Eq. 14. Fourier’s Law, conductive heat transfer

\( A = \text{heat transfer surface (m}^2) \)
\( k = \text{thermal conductivity of the material (W/m.K)} \)
\( dT = \text{temperature difference across the material (K)} \)
\( s = \text{material thickness (m)} \)

The Fourier’s law shows that a decrease of the thermal heat transfer conductivity(k) automatically leads to a decrease in heat absorption, due to the fact that the heat transfer surface(A) and the material thickness(S) do not change. By this theoretic statement, it is possible to assume that the
temperature differences are also constant \((dT)\). The following calculation will show the decrease of wall efficiency due to the boiler pollution.

Example: pipe thickness is 1.2[cm] the transfer area 1[m\(^2\)] and the temperature difference of 200[\(^\circ\text{C}\)]. the thermal conductivity of the stainless steel pipe walls is 19[W/m.K] in clean condition, in extreme polluted situations will it be around the 10[W/m.K]. This will give an energy extracting of 316,6[kW] vs. 166,6[kW] what will create a wall efficiency loss of 47.3%. It is clear that a changing thermal conductivity will bring variation in the place where the heat will be extracted out of the flue gasses.

To ensure a stable boiler output without abnormally high maintenance intervals for cleaning the pipe walls, there is an extra heat exchanger placed in the steam-drum (fig.12). This additional heat exchanger (called inner drum heat exchanger, IDHE) will be connected to the economizer circuit for an extra economizer loop(fig.13), in order to enlarge the eco capacity. The supply water flow (cold composite) will in that case partly be preheated in the drum whereafter the water flow will return in the normal boiler cycle. The IDHE, which is based beneath the water level (258[\(^\circ\text{C}\)]) of the steam drum, will be managed proportionally by the output flue gas temperature controller(225[\(^\circ\text{C}\)]).

The maximum IDHE loop capacity is not known at the Attero databank. But due to the known pipe diameters it is possible to make a rough assumption. The pipe diameters from the supply tank is 150[mm] instead of the IDHE supply pipe which is 125[mm]. Knowing that the entire boiler heating process is acting on the same pressure, will bring the possibility to assume that the flow capacity of the IDHE loop is 83.3\% (125/150) of the economizer. For the calculation of the maximum heating capacity of the IDHE there is also the need to make an assumption for the output water temperature (+/-185[\(^\circ\text{C}\)]) of the IDHE. Now it is possible to calculate the heating capacity by Eq.10 and calculating the \(K*A\) factor by Eq.8 which will bring that the maximum heating capacity of 8.2[MJ] and a \(K*A\) factor of 91.7.

The Economizer and the steam drum are in this way protected against the variable boiler pollution by the possibility to shift 33\% of the total ECO heating capacity to the DRUM.
4.2. Internal process variables
Beside external influences, the steam cycle also depends on multiple internal process variables which influence the steam cycle. This paragraph will clarify the steam cycle pre-heating, the impact of the external steam supply, by describing the: pre-heaters 10 and 20, the supply-tank heater, external heat supply and the turbine set-up separately. The aim of this paragraph is to know what the internal influences are on the current steam cycle and what the process boundaries are.

4.2.1. Pre-heating 10 and 20
The pre-heater 10 and 20 got as main goal to pre-heat the incoming condensate up to the demanded incoming supply tank temperature. This will occur by extracting steam in two conditions out of the turbine drains. This sub-paragraph will explain how this process is controlled and how the energetic efficiency of this process is.

Containing two sub-paragraphs:
- Controlling the pre-heat process of PH10 and PH20
- Energetic efficiency

Controlling the PH10 and PH20
The pre-heaters PH10 and PH20 (fig.14) heat the condensate flow by the use of a low and extra-low steam pressure flow. The low-pressure steam bleed is slightly overheated (120°C and 2[Bar]), the Low-pressure steam bleed is 98% vapour at 94 Celsius. The pre-heaters will be controlled by the hot-well volume level controller, in order to preserve the condensate output set point temperatures. The hot-well controller acts on the slope of the volume level changes, due to time. These volume changes will be used to measure the demanded heat.

For example: When the pre-heater is heating the entering condensate, the volume of the hot-well tank will suddenly start to increase (normally, the hot well level is constant). This means that the condensate flow from a certain moment starts to absorb more heat out of the steam flow. The reason for this increase of heat extraction could be the decrease of the incoming condensate temperature or an increase of the condensate flow. The reaction of the controller turns more steam into the pre-heater to establish the rising slope level of the hot-well tank.

The theory behind controlling this process would be that the hot-well controller indirectly measures each logarithmic average temperature difference (ΔT_in) (Eq.5) between the steam and condensate flow.

A decrease in condensate flow would lower also the ΔT_in due to the created overcapacity in relation to the condensate output set-point temperature. This will increase the condensate output above the set point. The hot-well controller now will lower the steam flow in order to maintain the set-point value. In case the lowering incoming condensate temperature is lowered the ΔT_in increases, which suggests a heat demand increase. To keep a constant condensate output temperature, it needed to increase the steam input flow.

The following equations (Eq.15-16) will underpin that a lowering condensate input temperature increase the ΔT_in and will result in a heating demand growth. Equation 16 will clarify that the increase of the pre-heat heat energy demand will be created by an increase of the steam flow, because the extractable heat (Δh_st) out of the steam is constant.

13 the hot-well is a small collecting ‘tank’ at the underside of the pre-heaters where the condensed steam get collect before it will be pumped out of it.
\[ Q_{PH} := K \cdot A \cdot \Delta T_{log} \]  
Eq. 15 Specific pre-heater equation

\[ Q_{PH} := M_{st} \cdot \Delta h_{st} \]  
Eq. 16

\[ K: \text{ Specific condenser constant [kJ/m}^2 \cdot \text{K]} \]
\[ A: \text{ Condenser surface [m}^2 \text{]} \]
\[ \Delta T_{log}: \text{ The natural logarithm delta average temperature [K]} \]

However, due to the method of controlling, this type of controlling has a strong disadvantage. The method will manage the condensate output temperature by varying the steam input. However knowing that the condensate flow will enter the pre-heaters in batches out of the main condensate tank, will it result in a strong translating heat energy extraction out of the turbine drains. Eventually will this create a strongly varying turbine work, which is caused by the temporary large needed heat extraction capacity on the LP and ELP bleed for PH10 and PH20.

**Energetic efficiency of PH10 and PH20**

As described in appendix 1.1-2 the energetic efficiency tells how efficient the additional process will use the available heat energy. Exergy will mean the decrease of the energy content quality, in other words the loss of potential work. Figure 15 will display the way of heating, the red line which represent the steam input (hot composite). The blue line represent the heatable condensate (cold composite). The surface between those two lines will illustrate the exergy loss due to the heat transfer. Appendix 1.1-2 describes how this exergy loss and the efficiency will be calculated. The results are showed in table 5. The exergy loss of the PH10 and PH20 are calculated in full load on 10 Celsius. The exergy losses 144 and 114 [kJ/kg] are rather high, which is shown in the energetic efficiency (Eq.17). Due to the low temperature lift in the cold composite the energy density difference between the incoming and pre-heater output is rather low, in contrast to the hot-composite which condensates from (2602 [kJ/kg]) a relative high energy density. Equation 17 will clarify that the energetic efficiency of PH10 is negative (-26%) due to the bigger exergy loss than the heating demand (\( \Delta h_{\text{cold.comp.}} \)).

\[ \zeta := 1 - \left( \frac{E_{\text{loss}}}{\Delta h_{\text{cold.comp.}}} \right) \]  
Eq. 17

\[ \zeta: \text{ Energetic efficiency [%]} \]
\[ E_{\text{loss}}: \text{ Exergy loss [kJ/kg]} \]
\[ \Delta h_{\text{cold.comp.}}: \text{Pre-heating energy demand[kJ/kg]} \]

Summary: the energetic efficiency of the PH-10 and 20 could are really low(-26%, 12%).

<table>
<thead>
<tr>
<th>Pre-heater</th>
<th>( h_{av,\text{hot}} )</th>
<th>( T_{av,\text{hot}} )</th>
<th>( h_{av,cold} )</th>
<th>( T_{av,cold} )</th>
<th>( \Delta S_{irr} )</th>
<th>( Q )</th>
<th>( h_{loss} )</th>
<th>( \zeta )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[kJ/kg]</td>
<td>[K]</td>
<td>[kJ/kg]</td>
<td>[K]</td>
<td>[kJ/kg.K]</td>
<td>[kJ/kg]</td>
<td>[kJ/kg]</td>
<td>[%]</td>
</tr>
<tr>
<td>Pre-heater 10</td>
<td>2602.0</td>
<td>367.0</td>
<td>278.3</td>
<td>339.6</td>
<td>2323.7</td>
<td>0.51</td>
<td>144.6</td>
<td>-26%</td>
</tr>
<tr>
<td>Pre-heater 20</td>
<td>2723.0</td>
<td>393.0</td>
<td>401.3</td>
<td>367.8</td>
<td>2321.7</td>
<td>0.41</td>
<td>114.7</td>
<td>12%</td>
</tr>
</tbody>
</table>

Table 5 Energetic efficiency outcome of Pre-heater 10-20

![Figure 15 pre-heating displayed in a T/Q diagram](image-url)
4.2.2. Supply-tank pre-heating and degassing process

The main goals of the supply-tank are heating the condensate up to the demanded minimum temperature of 150[°C] and degassing the condensate water to avoid harmful gasses in the steam cycle. This ensures supplying clean condensate as supply water to the boilers. This paragraph will describe these two functions followed by the way of controlling this process.

Heating in the supply tank.
The supply tank (fig. 16) heats the condensate by injecting MP-steam (216[°C], 5.6[Bar]) beneath the condensate level. The condensate (4.8[Bar], 113[°C]) will after entering the tank be heated up till 150[°C] which is the boiling point of water on that particular pressure. The overheated MP-steam will directly after it is injected reduce from its current pressure till the prevailing pressure in the supply tank. This reduce of pressure will also result in reduce of the boiling point of the injected steam. It will cool back from 216[°C] till the prevailing boiling point where after it start to condensate, till it is finally completely merged. The energetic efficiency of the heating process will be calculated in appendix 1.3. The outcome of the appendix 1.3 calculations are displayed in table 6. In which, it showed that the average hot composite temperature ($T_{av,hot}$) is static on 150[°C]. Due to the average temperature differences between the hot and cold which are small the irreversible entropy loss ($\Delta S_{irr}$) become also small, what will result in a small exergy loss and an energetic efficiency of 51%, which is an sufficient percentage.

![Figure 16 Supply-tank](image)

Table 6 Energetic efficiency outcome of the Supply-tank heating

<table>
<thead>
<tr>
<th></th>
<th>$h_{av,hot}$</th>
<th>$T_{av,hot}$</th>
<th>$h_{av,cold}$</th>
<th>$T_{av,cold}$</th>
<th>$Q$</th>
<th>$\Delta S_{irr}$</th>
<th>$E_{loss}$</th>
<th>$\zeta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply-tank</td>
<td>[kJ/kg]</td>
<td>[K]</td>
<td>[kJ/kg]</td>
<td>[K]</td>
<td>[kJ/kg]</td>
<td>[kJ/kg.K]</td>
<td>[kJ/kg]</td>
<td>[%]</td>
</tr>
<tr>
<td>Heating (condensing steam)</td>
<td>2746.0</td>
<td>423.0</td>
<td>549.2</td>
<td>402.0</td>
<td>2196.8</td>
<td>0.27</td>
<td>76.9</td>
<td>51%</td>
</tr>
</tbody>
</table>

Degassing of the supply tank.
The supply-tank will degas the cold composite (condensate) by heating the cold composite flow up to the boiling point where after a certain amount of the cold composite will be evaporated, which will float the harm full gasses out of the supply-tank. To create a clear vision of the current degassing process, this paragraph describe the process characteristics such as; the energetic efficiency, the needed percentage of degassing [kg/s] and the needed heat energy for degassing.

To create this amount of evaporated cold composite, this degassing process uses the overheated part of the injected MP-steam flow. Due to the reduction of pressure (from MP till supply tank pressure) the overheated share will increase in relation to the condensation share. Which is enough for the degassing process. The energetic efficiency is calculated in appendix 1.3 and clarifies that the efficiency is extremely high (97%), due to the small exergy loss and the large heating range of the cold composite is it possible to create this high efficiencies (tab.10).

<table>
<thead>
<tr>
<th></th>
<th>$h_{av,hot}$</th>
<th>$T_{av,hot}$</th>
<th>$h_{av,cold}$</th>
<th>$T_{av,cold}$</th>
<th>$Q$</th>
<th>$\Delta S_{irr}$</th>
<th>$E_{loss}$</th>
<th>$\zeta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply-tank</td>
<td>[kJ/kg]</td>
<td>[K]</td>
<td>[kJ/kg]</td>
<td>[K]</td>
<td>[kJ/kg]</td>
<td>[kJ/kg.K]</td>
<td>[kJ/kg]</td>
<td>[%]</td>
</tr>
<tr>
<td>Degassing</td>
<td>2837.7</td>
<td>454.0</td>
<td>1439.1</td>
<td>423.0</td>
<td>1398.7</td>
<td>0.23</td>
<td>63.9</td>
<td>97%</td>
</tr>
</tbody>
</table>

(assume that the enthalpy ($h_{out}$) of the degassing output is 2791 [kJ/kg] at 170[°C])

Table 7 Energetic efficiency outcome of the Supply-tank degassing
The heat demand for degassing will be calculated by the ‘derivative continuity equation’ (Eq. 18). The needed values are: the cold composite steam exhaust factor for the orifice (0.9), the size of the degas-pipes (steam outlet pipes)(which are two pipes with an inner diameter of 10[mm]). The equations below will clarify the needed heat energy and the amount of steam ‘leak’.

\[ M \cdot v_{st} = A \cdot c \cdot \mu \]  
Eq. 18 continuity law

Eq. 19 will supply the needed cold composite steam flow for degassing which will be used in Eq.20 in which the ‘heat loss’ (due to the steam leak) will be calculated.

\[ M_s := \frac{(A \cdot c \cdot \mu)}{v_{st}} \]  
Eq. 19 derivative continuity law

\[ Q := M_s \left( h_d - h_w \right) + M_s \left[ C_w \left( T_{\text{steam}} - T_{\text{ambient}} \right) \right] \]  
Eq. 20 derivative Gas Law

\[ c := \frac{(2 \Delta p)}{\sqrt{\rho_{\text{steam}}} } \]  
Eq. 21

\[ \Delta p := p_{\text{tank}} - p_{\text{amb}} \]  
Eq. 22

\[ M_s: \text{ Steam leak mass flow [kg/s]} \]
\[ A: \text{ Pipe surface [m}^2\text{]} \]
\[ c: \text{ Steam leak velocity [m/s]} \]
\[ \mu: \text{ Orifice contraction factor [0.9]} \]
\[ v_{st}: \text{ specific steam volume [m}^3\text{/kg]} \]
\[ Q: \text{ Degas heat loss [kJ/s]} \]
\[ h_d: \text{ Enthalpy steam[kJ/kg]} \]
\[ h_w: \text{ Enthalpy water[kJ/kg]} \]
\[ C_w: \text{ Specific heat density water [kJ/kg.K]} \]
\[ T_{\text{steam}}: \text{ Steam temperature [°C]} \]
\[ T_{\text{ambient}}: \text{ Ambient temperature [°C]} \]
\[ \rho: \text{ Steam density [kg/m}^3\text{]} \]
\[ \Delta p: \text{ Liquid pressure [Pa]} \]
\[ p_{\text{tank}}: \text{ Supply-tank pressure [Pa]} \]
\[ p_{\text{amb}}: \text{ Ambient pressure [Pa]} \]

The outcome is displayed in tab.8, this shows that there will leakage of 0.196[kg/s] steam out of the supply-tank which result in an energy loss of 528.2[kJ/s]. Notable is that the energy loss is calculated till ambient conditions which will mean that this contains the total potential heat energy loss. The needed heat energy from the supply-tank to evaporate 0.196[kg/s] would only be 414 [kJ/s] (\(M_s \cdot (h_d - h_w)\)).

<table>
<thead>
<tr>
<th>Degas process data</th>
<th>(\Delta p)</th>
<th>Rho</th>
<th>Hd</th>
<th>Hw</th>
<th>(T_{\text{ambient}})</th>
<th>(C_{\text{water}})</th>
<th>Vs</th>
<th>A</th>
<th>c</th>
<th>(M_s)</th>
<th>Q</th>
</tr>
</thead>
<tbody>
<tr>
<td>[Pasc]</td>
<td>[kg/m}^3\text{]}</td>
<td>[kJ/kg]</td>
<td>[kJ/kg]</td>
<td>[°C]</td>
<td>[kJ/kg.K]</td>
<td>[m}^3\text{/kg]}</td>
<td>[mm}^2\text{]}</td>
<td>[m/s]</td>
<td>-</td>
<td>[kg/s]</td>
<td>[kJ/s]</td>
</tr>
<tr>
<td>376101</td>
<td>2.55</td>
<td>2745.9</td>
<td>632.3</td>
<td>10</td>
<td>4.18</td>
<td>0.393</td>
<td>157</td>
<td>543.36</td>
<td>0.9</td>
<td>0.196</td>
<td>528.2</td>
</tr>
</tbody>
</table>

Table 8 Degassing process data
Way of controlling the supply-tank
The supply tank heating and degassing process is controlled by the supply-tank level and pressure indicator. These two parameters are installed to indicate the demand of hot composite. The way of controlling the degas percentage and the heating process till 150[°C] is only based on the pressure indicator, because the increase of cold composite pressure will point that the there is an evaporated cold composite surplus. This will tell that the cold composite reached boiling point, and exceed the designed amount evaporated cold composite. However the amount cold composite, which enters the supply-tank, is controlled by the condensate pump (which is based beneath the main condensate tank) and will act proportionally on the main condensate tank level indicator. In the same way as pre-heater 10-20 will this also result in a variation of the condensate level in the main condensate tank what will create a variation in the heating demand supply-tank heater. Although the giant volume of the supply-tank will delimitate this variation of its own heating demand.

4.2.3. External steam supply
The idea of external steam supply came when the need for energy efficiency enhancing alternatives become more important. It is likely that selling the heat energy without turbine losses increases the plant efficiencies (η_total and R1 mark). The steam could be sold in all available conditions and the only influence on the efficiency is the way it is created. In this steam cycle, there are three different manners to provide external supply steam. The following paragraphs will describe these possibilities for steam extraction, and the process influences of the current external steam supply (ESS).

Possibilities for creating supply steam.
The current steam cycle has three different possibilities to create steam for external use, these options are; extracting steam out of the turbine, create MP-steam by the use of the turbine driven supply water pump(TSP) or by reducing high-Pressure steam to the requested conditions. Each option will be described in this paragraph to clarify the benefits and disadvantages of these options.
**Turbine drain**
The option to extract steam out of the turbine for external use is a good option due to the fact that the steam will be used from overheated steam till it is fully condensed. However this type of ESS is restricted by the turbine specifications and the internal heating demand. Table 9 shows the turbine bleed specification and the needed process heat. Notable is that only out of the LP an ELP drain still a sufficient amount of steam is available to extracted.

The thermodynamic efficiency of this partial process will be calculated but isn’t comparable with the total efficiency. This partial efficiency only points which of the steam extraction possibilities is the most thermal efficient. Equation 23 shows that the overall thermodynamic efficiency is the theoretical multiplication of the isentropic turbine efficiency (\( \eta_{\text{isen. turb}} \)) and the thermodynamic ESS efficiency. The isentropic turbine efficiency (\( \eta_{\text{isen. turb}} \))(75%) is depending on the technical performance of the turbine. The standard equation for the isentropic turbine efficiency (Eq.24) and figure 17 show that the irreversible enthalpy loss over the turbine expansion is responsible for the \( \eta_{\text{isen. turb}} \). The thermodynamic efficiency of the ESS (\( \eta_{\text{th ESS}} \)) will be almost 100%, because the steam leaves in MP conditions and will return as condensate on 5.3[bar] and 140[°C] (589[kJ/kg]) instead of the ambient conditions of water 224[kJ/kg]. The thermodynamic efficiency of this process will be calculated with equation 23.

\[
\eta_{\text{th.Turbine.ESS}} = \left( \frac{\eta_{\text{isen. turb}} \Delta h \text{Turbine.ideal} + \Delta h \text{th.ESS}}{\Delta h \text{total}} \right)
\]

Eq. 23

\[
\eta_{\text{isen. turb}} = \frac{h_1 - h_2}{h_1 - h_2'}
\]

Eq. 24

\( \eta_{\text{th.Turbine.ESS}} \): Thermodynamic efficiency of the part process of ESS creating with the main turbine [%]

\( \eta_{\text{isen. turb}} \): isentropic turbine efficiency 75[%]

\( \Delta h \text{Turbine.ideal} \): Enthalpy difference between the HP-steam and the ideal bledded MP-steam (point 2')(3212-2776)[kJ/kg]

\( \Delta h \text{th.ESS} \): Enthalpy difference between the bleeded MP-steam and the returned condensate (2885-589)[kJ/kg]

\( \Delta h \text{total} \): Enthalpy difference between the HP-steam and the returned condensate at ambient conditions(3212-224)[kJ/kg]

The overall thermodynamic efficiency for the ESS process in combination with a MP-turbine bleed is 86.3%
**ESS created with the Turbine driven supply-pump**

The turbine driven supply-pump operates on HP-steam (40[bar], 400[°C]) which will expand in the turbine till almost MP-steam. To add the TSP steam to the MP-steam bar it will be necessary to cool the steam further down till it has the same MP specs, this will be done by injecting supply water. The total isentropic efficiency of the TSP is 50% ($\eta_{\text{isen.TSP}}$), which is depending on the technical performance of the turbine-pump and is, compared to the main turbine, rather low. This will result due to the static ESS efficiency in an overall thermodynamic efficiency of 82.7% (Eq.25). The maximum flow enlargement on the MP-bar due to the TSP is 3.4 [kg/s] due to the restricted TSP capacity.

$$\eta_{\text{th.TSP.ES S}} := \frac{\eta_{\text{isen.TSP}} \Delta h_{\text{TSP.ideal}} + \Delta h_{\text{th.ES S}}}{\Delta h_{\text{total}}} \quad \text{Eq. 25}$$

$\eta_{\text{th.TSP.ES S}}$: Thermodynamic efficiency of the part process of ESS creating with the TSP [%]
$\eta_{\text{isen.TSP}}$: Isentropic efficiency of the TSP 50[%]
$\Delta h_{\text{th.ES S}}$: Enthalpy difference between the HP-steam and TSP MP-steam output (2885-632)[kJ/kg]

**Reducing HP-steam for ESS**

Reducing high pressure steam till ESS conditions will occur in two steps (see fig.18). Step one is an isothermic expansion to the 6.5 [bar] (point 2) after which the steam will be reduced isobarically in temperature by water injection (from point 2 to point 3). This part process consists of entropy loss (0.76[kJ/kg,k]) and a large enthalpy loss (327[kJ/kg]) because there is no work done in this part process.

The thermodynamic efficiency calculation of this part process becomes just the ESS enthalpy consumption divided by the total enthalpy use (Eq.26). The thermodynamic efficiency of this process is 75.4[%].

$$\eta_{\text{th.reduce.ESS}} := \frac{\Delta h_{\text{th.ESS}}}{\Delta h_{\text{total}}} \quad \text{Eq. 26}$$

$$\eta_{\text{th.m.turb}} \leq \frac{\eta_{\text{isen.m.turb}} \Delta h_{\text{mturb}}}{\Delta h_{\text{total}}} \quad \text{Fig. 18 h/s diagram HP-steam reduce}$$

$$\eta_{\text{th.m.turb}} \leq \frac{\eta_{\text{isen.m.turb}} \Delta h_{\text{mturb}}}{\Delta h_{\text{total}}} \quad \text{Eq. 27}$$

$\eta_{\text{th.m.turb}}$: Thermodynamic efficiency of the main turbine[%]
$\eta_{\text{isen.m.turb}}$: Isentropic efficiency of the main turbine 75[%]
$\Delta h_{\text{mturb}}$: Enthalpy difference between the HP-steam and turbine output (3212-2302)[kJ/kg]
Current extern steam supply
In the present state, there is one external steam consumer (ESC), based 800 meters from the Attero plant. The steam specifications of their needs are an amount of $11\text{[t/h]}$ ($3\text{[kg/s]}$) at 6.28 [bar] and 216 [°C]. The used steam returns as condensate on 5.3[bar] and 141 [°C] in the supply-tank.

Due to the thermodynamic efficiencies does Attero chose to extract as much steam out of the turbine MP-drain. However the current supply tank heater extract already the complete drain capacity of the MP-drain. To create the total amount of ESS Attero will make use of the TSP to fulfil the total amount of $3\text{[kg/s]}$.

Although the steam cycle is a closed circuit, the external steam consumer consumes a certain amount of condensate, which not will return. This amount of condensate loss will be corrected with the demi-water supply tanks. However this water does not have the high condensate temperature and therefore has to be pre-heated. The influences on the heating demand of the supply tank of this relatively small extra heating amount is negligible.

The ESS process will lower the total turbine steam flow output, which will result in a lower steam pressure output due to the fact that the capacity ratio of the air condenser versus the steam flow rises. This will increase the turbine work slightly which subsequently lead to proved the total efficiency and R1 mark a bit, only in the following efficiency calculation (total efficiency and R1 mark) will this not being implemented.

However the small work increase, due to the air condenser capacity ratio, will the current steam extraction create a large the turbine work decrease. But will also increase the thermodynamic efficiency for a part of the steam flow with circa 61[%] (~84%-22.8%).

The turbine work decrease caused a power drop $W_{\text{t, turb.elec}}$ of 1.7[MW]\(^{14}\) but will create an external heat supply of 6.8[MJ], which will bring a total plant efficiency of 26.5% (eq.28), and a R1 expression of 0.638 (eq.29-30)\(^{15}\). The values which are drawn below the equations are calculated values out of appendix 2.1. This Appendix will display the process values in the current state in use of the ESS (as write before by bleeding out of the turbine and using the TSP)

$$\eta_{\text{total}} = \frac{[Q_{\text{turbine}} \cdot \eta_{\text{generator}}] + Q_{\text{ESS}} + [Q_{\text{TSP}} \cdot \eta_{\text{TSP}}]}{Q_{\text{boiler}}}$$

Eq. 28

\[Q_{\text{turbine}}: \quad 47.13\text{[MJ]}\]
\[\eta_{\text{generator}}: \quad 98\%\]
\[Q_{\text{ESS}}: \quad 6.8\text{[MJ]}\]
\[Q_{\text{TSP}}: \quad 0.273\text{[MW]}\]
\[\eta_{\text{TSP}}: \quad 50\%\]
\[Q_{\text{boiler}}: \quad 165\text{[MJ]}\]
\[\eta_{\text{boiler}}: \quad 82.3\%\]
\[\eta_{\text{TPE}}: \quad 26.4\%\]

$$\eta_{\text{R1}} = \frac{E_p - (E_f + E_i)}{0.97(E_w + E_f)}$$

Eq. 29

\(^{14}\) Is the heat loss due to the steam extraction multiplied with an electric efficiency of 98 %

\(^{15}\) See chapter 3. Efficiencies
4.2.4. Turbine set-up

The challenge for the process operators is to deliver a stable and high steam amount out of the boilers by managing the furnace. Although they are highly qualified, it is not possible to ensure a max steam load [kg/s] at all times, which results in a varying turbine load. The aim of this paragraph is to describe the process influences on the turbine and how the turbine participates on those flow changes. Note: that the turbine is build up out of two turbines called high pressure and low pressure turbine (see fig. 3).

The risk of a lower steam amount is that the HP-turbine will already expand the output steam pressure before it enters the end of the LP-turbine. This will result in a leak of turbine work into the LP-turbine. After all, the turbine speed has to stay constantly at 3000[RPM] and only the turbine work has the possibility to fluctuate. Two other risks might be that due to the shortage of expandable steam, temperature excesses in the LP turbine could occur and the pressure on three steam drain stages could become too low. For these reasons, there is a diaphragm-valve placed between the HP and LP turbine (fig. 3 called throttle). This valve will smother the passing steam when the pressure becomes too low before the valve, to aim a balanced turbine work between both the turbines which also will ensure constant bleed pressure at the MP-drain. A disadvantage of the diaphragm-valve is that isentropic efficiency of the HP-turbine will decrease (fig.19-20). This will occur due to the pressure increase which causes an entropy increase. Eventually the LP turbine will operate at its current isentropic efficiency but cannot be avoided that there is an enthalpy loss (Δh_loss) (fig.14) increase. However this D-valve acts automatically on temporary turbine load variations, in this way it is not necessary to include this in the total efficiency equations.

\[
E_p = E_{elec} + E_{heat}
\]

\[
E_{elec} = \left( Q_{turbine} \cdot \eta_{generator} \right) \cdot 2.6
\]

\[
E_{heat} = \left[ Q_{ESS} + \left( Q_{TSP} \cdot \eta_{TSP} \right) + Q_{degass} \right] \cdot 1.1
\]

\[
E_{p} = E_{elec} + E_{heat}
\]

\[
E_{elec}^e \quad 3833699 \quad [\text{GJ/year}]
\]

\[
E_{elec}^h \quad 213152 \quad [\text{GJ/year}]
\]

\[
Q_{degass} \quad 13055 \quad [\text{GJ/year}]
\]

\[
E_{f} \quad 5048 \quad [\text{GJ/year}]
\]

\[
E_{i} \quad 50150 \quad [\text{GJ/year}]
\]

\[
E_{w} \quad 6322527 \quad [\text{GJ/year}]
\]

\[
E_{p} \quad 4066851 \quad [\text{GJ/year}]
\]

\[
R1mark \quad 0.63 \quad [-]
\]
5. Potential process optimizations
The aim of this chapter is to give an overview of the process influences which constrain the current plant efficiencies. Furthermore, to show which of these process influences offer the possibility to optimize the process in order to increase R1 and TPE of the plant. The comparison of the effect of these possibilities will lead to the answer of the sub-question “what is the best option to improve the current performance”.

5.1. Current state reflection
The performance of the Attero WTE plant steam cycle depends on the process influences that are described in detail in chapter 4 and summarized as follows:

1. Condenser capacity shortage.
The ambient temperature strongly influences the condenser power, which is responsible for the operation of the LP-turbine. An ambient temperature difference of 35[°C] will result in a turbine work loss of 9.75 [MW](17% of the total turbine power), which suggests that the condenser capacity is inadequate. The ambient temperature will also create a self-reinforcement on the turbine operation, due to the decrease of the heat energy demand in pre-heater 10, which eventually results in an increase in the required condenser capacity.

2. Inefficient partial processes
The caloric energy density of the burnable waste is a highly fluctuating process influence. To control the steam production two processes are installed: water injectors and a PPC valve. These processes operate in an energetic inefficient way.

3. Boiler pollution
The boiler itself becomes polluted by the passing flue-gasses, which create a thick layer of ashes on the pipe walls (surfaces of the ECO, DRUM, OVO) that decreases the heat transfer coefficient[K]. Therefore, an ‘ECO loop’ is applied to act in clean boiler conditions in order to manage the disproportional large heat extraction in the Drum. The ‘ECO loop’ has the ability to shift a large amount of the extracted heat, in the boiler, to the economiser. This loop will stabilize the boiler balance in clean conditions.

4. Economiser restriction (which caused the total preheating demand).
The Economizer restrict that the incoming supply water got a minimum temperature of 150[°C]. This result in a supply water pre-heat demand of 26[MJ], which lowers the total plant efficiency, due to the reason that it is currently heated with process steam in a low energetic inefficient way (PH-10 -26%, PH-20 12%, supply tank 51%).

5. Controlling the pre-heating process
The supply water pre-heating flow is managed by the main condensate tank level, and will be pumped by batch trough pre-heaters 10 and 20 into the supply tank. Due to the small storage capacity of the main condensate tank and in batch operating condensate pump will this result in high a fluctuating heat energy demand on the first two pre-heater steps, and will it cause temporary turbine work fluctuations.

6. Degassing process
The degassing process, which will occur in the supply tank, will create a heat loss of 528 [kJ/s] by evaporating a small percentage of the supply water. The quantity of heat loss is depending on the boiling temperature of the supply water.

7. Extern steam supply
The current external steam supply (ESS) extracts 3[kg/s] MP-steam, which has the benefit that the necessitated condenser capacity decreases, and the total plant efficiency and the R1 mark increases (TPE 26.5[%] and R1 mark of 0.638).
5.2. Potential optimizations

The goal of this paragraph is to analyze possible optimizations that will increase the total efficiency and R1 mark. This paragraph will describe each potential optimization in a way that makes clear which process constraint (described above) will be solved by the suggested improvement, and will describe the additional benefits which the optimization has.

The suggested optimizations are:

1. Enlarging the condenser capacity
2. Enlarging the MP-steam turbine drain (in order to supply more ESS out of the turbine).
3. Lowering the pre-heating steam (heat energy) demand.
4. A change in controlling the pre-heater process.

1. Enlarging the Condenser capacity.

The condenser is extremely sensitive for the ambient temperature, which results in high turbine losses due to the high condensate temperature. Enlarging the condenser capacity by increasing the condenser surface or air flow means that the temperature difference between the ambient and condensate temperature ($\Delta T_{in}$) will decrease, which will lead to an increase of the achievable minimum condenser pressure on a larger ambient temperature range. This could result in a maximum turbine work increase of $3.3[MW]$ in summer conditions ($T_{av, 30[°C]}$).

2. Enlarging the MP-steam turbine drain.

This optimization aims to enlarge the possibility to bleed MP-steam out of the turbine for external use by raising the diaphragm-valve pressure settings. In the current state, the MP-bleed is restricted by the size of the turbine drain, which causes that the current External Steam Supply (ESS) partially needs to be created by the TSP, which has a lower steam isentropic efficiency (75% vs. 50%). The possibility to increase the D-valve pressure settings would attempt to shift the turbine pressure states upwards to create an MP-state in order to provide a static higher inlet pressure (6.5bar instead of 2bar). This would make an enlargement of the ESS steam supply to a bleed of $51[kg/s]$ instead of $3.99[kg/s]$ possible. The disadvantage is that the current isentropic efficiency HP-turbine will decrease a certain percentage and that the ELP-drain will stop to exist.

3. Lowering the pre-heating steam demand.

Solving a part of the preheating demand would clearly result in a decrease of the internal process heat demand, what would bring the possibility to use that released heat energy demand to increase the total plant efficiency. This optimization would cause this efficiency increase by removing pre-heater 20 out of the steam cycle, and placing the supply tank in that position to heat the supply water up to the currently $110[°C]$ with LP-steam and degas on the same pressure. Hereafter, the partly preheated steam will be pumped up to the inner drum heat exchanger (which is used, currently, for the ‘Eco loop’) to be heated up to $150[°C]$. After this, the normal cycle will be resumed.

Questions are whether or not the heating capacity is sufficient, though, it is known that the logarithmic average temperature would increase and lead to an capacity increase. However, on the other hand, the supply water flow is significantly larger than the currently ECO loop flow.

The benefits of this optimization would be a decrease of the preheating heat energy extraction out of the turbine on MP-state (10.35[MJ]) and shift this heat energy demand to the boiler. This will result in stopping the use of MP-steam drain for preheating, which could account for a total plant capacity enlargement and efficiency increase by creating more turbine work or an ESS enlargement out of the turbine drain. The heat loss of the degas process would also decrease due to to lowering the degas temperature.
A change in controlling the preheating process.
The aim of this optimization is to fix the heat energy demand of PH10 and PH20 by flattening the flow variation of the cold composite flow. The suggested optimization will move the leading level indicator from the condensate tank to the supply tank, in order to achieve a more constant cold composite flow. This will happen because of two reasons. Firstly, due to the tank storage capacity differences between the supply tank and condensate tank (140 m$^3$ vs. 50m$^3$); secondly, because of the fact that the output flow of the supply tank is more constant than the input flow of the condensate tank. These two facts create an ability to manage the inlet flow of the supply tank more smoothly. This alteration will subsequently lead to a more constant cold composite flow over the pre-heaters, which indirectly will create a more constant turbine work, of which the total plant efficiency benefits.

5.3. Optimization selection
Each of the proposed optimizations aims to have efficiency growth as a result and is realizable. However, what is the best option to improve the current performance? By comparing the suggested optimizations with each other, it will become clear which one is the best optimization.

The first suggestion, enlarging the condenser capacity is a high potential optimization due to the benefit that it could increase the turbine work with 17% in hot summer times. The second suggestion, increasing the D-valve pressure settings is solely relevant when there is a concrete large external heat demand (next to the current ESS). The third suggestion is also a high potential because it solves multiple process constraints; it will raise the plant efficiencies due to the possibility to extract more steam out of the MP-bleed for the ESS, and will also increase the WTE plant capacity. The last suggestion will create a more established and constant flow though the pre-heaters, which subsequently will lead to an established heat energy demand. However, it does it not create an efficiency increase like suggestion 1 or 3.

The best optimization for Attero would be the third suggestion: Lowering the pre-heating steam demand, due to the reason that this option increases the efficiencies in each weather type; besides this, it will also increase general plant capacity.
6. Pre-heat optimization
In the following paragraphs, the feasibility of the steam cycle pre-heating process optimization will be examined in order to answer the third sub-question: which efficiency increase can be achieved through this option. To answer this question, the following structure is being followed: in the first paragraph, a detailed process explanation will be done. The second paragraph will calculate the new process numbers and efficiencies, which will be compared with the current state. Lastly, the conclusion will regard whether or not this optimization will benefit the process enough.

6.1. Process explanation
The most important improvement in this optimization is including the inner drum heat exchanger (IDHE) into the pre-heating cycle. This exchanger operates as last pre-heater by heating the cold composite up to 150[°C]. Due to this inclusion, the current pre-heat process needs to be restructured, which means that the supply tank will be shifted to the initial process step of the pre-heater 20 where the cold composite is being heated up to 110[°C]. This new supply tank heating point causes that the degassing process occurs at the boiling point of 110[°C] at 1.43[Bar], and will be heated with LP-steam. Pre-heater 20 operates as first pre-heater (PH-10) by heating the entering cold composite up to 80[°C]. This leads to the new structured preheating process, where Pre-heater 10 will no longer be used (fig.21). After the supply tank, the cold composite will be pumped on 40[bar] to the IDHE, where it will be heated up to demanded 150[°C] where after it continues into the current heating path.

Figure 21 optimized pre-heating process layout
6.2. Optimization results
In order to check the feasibility in a technical and thermo-dynamical way, the suggested optimization will be examined in the following subparagraphs:

- Pre-heater 20
- Supply-tank heating and degassing
- Inner drum heat exchanger
- New boiler operation
- Process benefits

The calculation will be made for a full load operation (steam flow 63.9[kg/s]) at an ambient temperature of 10 Celsius. The chosen temperature is the average temperature, which will clarify a realistic average plant operation; the final aim of this chapter is to supply the data for a well-founded conclusion in chapter 7.

**Pre-heater 20**
It is likely that the pre-heater tank will solely change this part of the optimization. The part process of the first heater in itself did not change. In that way, the first heater still heats the entering cold composite up to 80[°C] with ELP-steam, and needs 7.35[MJ] heat energy to complete this heating step. In table 10, the heat loss and the energetic efficiency are displayed (144[kJ/kg] and -26[%]). These values are compatible with the data of pre-heater 10 in the current state (appendix 1.2).

<table>
<thead>
<tr>
<th>Optimized state</th>
<th>(h_{av.\text{hot}}) [kJ/kg]</th>
<th>(T_{av.\text{hot}}) [K]</th>
<th>(h_{av.\text{cold}}) [kJ/kg]</th>
<th>(T_{av.\text{cold}}) [K]</th>
<th>(Q) [kJ/kg]</th>
<th>(\Delta S_{irr}) [kJ/kg.K]</th>
<th>(E_{loss}) [kJ/kg]</th>
<th>(\zeta) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pre-heater 20</td>
<td>2602.0</td>
<td>367.0</td>
<td>278.3</td>
<td>339.6</td>
<td>2323.7</td>
<td>0.51</td>
<td>144.6</td>
<td>-26%</td>
</tr>
</tbody>
</table>

Table 10. PH-20 energetic losses

**Supply-Tank heating and degassing**
The Supply-tank now operates at a lower heating level in the pre-heating process by heating the cold composite from 80[°C] up to 110[°C], and degasses at a new lower boiling point (110[°C]). Table 11 displays the new heating values which are calculated in the same way as displayed in appendix 1.3..

The degassing part will extract heat energy out of the overheated part of the hot composite (LP-steam), this is the part between the entering hot composite state (2[bar]) and the reduced hot composite boiling point at (1.4[bar]).

<table>
<thead>
<tr>
<th>Optimized state</th>
<th>(h_{av.\text{hot}}) [kJ/kg]</th>
<th>(T_{av.\text{hot}}) [K]</th>
<th>(h_{av.\text{cold}}) [kJ/kg]</th>
<th>(T_{av.\text{cold}}) [K]</th>
<th>(Q) [kJ/kg]</th>
<th>(\Delta S_{irr}) [kJ/kg.K]</th>
<th>(E_{loss}) [kJ/kg]</th>
<th>(\zeta) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply-tank heating</td>
<td>1260.4</td>
<td>401.0</td>
<td>401.3</td>
<td>367.8</td>
<td>859.1</td>
<td>0.19</td>
<td>54.8</td>
<td>58%</td>
</tr>
<tr>
<td>Supply-tank degassing</td>
<td>2706.0</td>
<td>391.8</td>
<td>1261.0</td>
<td>383.0</td>
<td>1445.0</td>
<td>0.0845</td>
<td>23.9</td>
<td>98.9%</td>
</tr>
</tbody>
</table>

(Assume that the enthalpy (h\text{out}) of the degassing output is 2690 [kJ/kg] at 110[°C])

Table 11 energetic supply-tank heating and degassing outcome

The differences between the current state and the optimized state are that the supply-tank heating process as well as the degassing process increase in energetic efficiency. This is displayed in table 12. This increase is a result of the smaller difference between hot and cold composite energy density, and thus, the heating step relays more closely to the ambient temperature.

<table>
<thead>
<tr>
<th>Optimized state</th>
<th>(\Delta S_{irr}) [kJ/kg.K]</th>
<th>(E_{loss}) [kJ/kg]</th>
<th>(\zeta) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heating (Current)</td>
<td>0.27</td>
<td>76.9</td>
<td>51%</td>
</tr>
<tr>
<td>Heating (Current)</td>
<td>0.19</td>
<td>54.8</td>
<td>58%</td>
</tr>
<tr>
<td>Degasing (Optimized)</td>
<td>0.23</td>
<td>63.9</td>
<td>97%</td>
</tr>
<tr>
<td>Degasing (Optimized)</td>
<td>0.08</td>
<td>23.9</td>
<td>99%</td>
</tr>
</tbody>
</table>

Table 12. comparison energetic efficiency of the supply tank in current state and optimized state
Degas leak
The degassing process will change too, due to the changing degas temperature and the pressure difference between the supply-tank and ambient pressure (Δp). The optimized data is pointed out in table 13 and shows the new degas flow and heat loss. The calculations are based on the same formulas as were used in paragraph 4.2.2(degassing).

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>40000</td>
<td>0.81</td>
<td>2689.9</td>
<td>458.4</td>
<td>10</td>
<td>4.22</td>
<td>1.230</td>
<td>0.0002</td>
<td>313.69</td>
<td>0.9</td>
<td>0.036</td>
<td>95.7</td>
<td>95.7</td>
</tr>
</tbody>
</table>

Table 13 degas leak outcome
The decrease of the needed degas flow (Ms) can be explained by the following two reasons (see eq.31); the decrease of C, which is caused by the Δp, will create a decrease of Ms. Due to the lowering pressure, the specific volume will increase, which will result in a Ms decrease. However, more important is that, due to the Ms and the lowering temperature difference between the cold composite and the ambient temperature (Eq.18), the degas heat energy leak will decrease from 528 [kJ](tab.18) to 95.7 [kJ].

\[ M_s \approx \frac{(A \cdot c \cdot \mu)}{v_{st}} \quad \text{Eq. 31} \]

Inner Drum Heat Exchanger (IDHE)
In its current stat, the IDHE is designed to heat 83.3% of the total economizer flow from 150[°C] to 180[°C]. The optimized state suggests heating the total cold composite flow (63.9[kg/s]) from 110[°C] to 150[°C]. This increase of flow and heat energy demand will result in hydrostatic and thermodynamic alterations. The resulting change in operation will be examined in the following two sub-paragraphs:

Hydrostatic changes
Isothermic changes

Hydrostatic changes
The IDHE cold composite flow will increase from the current 53.2[kg/s] to 63.9[kg/s] on the suggested pressure (40[bar]); besides, for this optimization, it is practically not possible to change the technical specifications (heat transfer surface). The cold composite flow increases velocity due to the static IDHE surface (Eq.28).

\[ v \approx \frac{M}{(\rho \cdot A)} \quad \text{Eq. 32} \]

\[ u: \quad \text{Cold composite velocity [m/s]} \]
\[ M: \quad \text{Mass flow [kg/s]} \]
\[ \beta: \quad \text{specific density of water 998[kg/m³]} \]
\[ A: \quad \text{IDHE cold composite inlet pipe surface[m²]} \]
By using the known inlet pipe diameter (0.125[m]), is it possible to calculate the current and the new cold composite velocity, which are 4.35 vs. 5.22[m/s]. It is obvious that a flow enlargement trough the IDHE will create a growth of cold composite fluid resistance. To overrule this resistance, the pressure before the IDHE will increase. However, it is not possible to calculate pressure increase without knowing the IDHE pipe specifications (resistance, length). Because the current cold composite pressure of 40[bar] does not reflect the needed work to pass the IDHE, but is related to the needed thermodynamic steam conditions. However, there can be assumed that due to the flow increase, the pressure before the IDHE will rise.

*Isothermic changes*

The IDHE has a capacity of 8.2[MJ] and an K*A number of 91.7(\(^16\)) at a cold composite flow of 53.2[kg/s] in the current state. However, the optimized process needs to heat the total flow from 110[°C] up to 150[°C]. Equation 33 shows that the optimized state needs a heat energy demand of 10.35[MJ] (table.14).

\[
Q_{\text{IDHE}} = M_{\text{Comp}} \Delta h_{\text{IDHE, Comp}} \tag{Eq. 33}
\]

- \(Q_{\text{IDHE}}\): heat energy demand [kJ]
- \(\Delta h_{\text{IDHE, cold comp}}\): Enthalpy difference between the entering and output cold composite flow [kJ/kg]
- \(M_{\text{Comp}}\): Mass flow of the cold composite [kg/s]

<table>
<thead>
<tr>
<th>Current state:</th>
<th>PH-10</th>
<th>PH-20</th>
<th>Supply-Tank</th>
<th>Total:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Qheat energy [kJ]</td>
<td>7348.5</td>
<td>7480.1</td>
<td>10351.8</td>
<td>413.7</td>
</tr>
<tr>
<td>Optimized state:</td>
<td>PH-20</td>
<td>Supply-Tank</td>
<td>IDHE</td>
<td>Total:</td>
</tr>
<tr>
<td>Qheat energy [kJ]</td>
<td>7348.5</td>
<td>8370.9</td>
<td>80.5</td>
<td>10351.8</td>
</tr>
</tbody>
</table>

Table 14. new pre-heating demand vs. current demand

This suggests that the IDHE has a heating shortage of 2.15[MJ]. The heating capacity of the IDHE will enlarge in the optimized state due to the fact that the optimized cold composite flow creates a larger \(\Delta T_{\log}\) because of to the lower income temperature (see fig.22 and eq.34,35)

\[
Q_{\text{IDHE}} = K \cdot A \cdot \Delta T_{\log} \tag{Eq. 34}
\]

\[
\Delta T_{\log} = \frac{(\Delta T_2 - \Delta T_1)}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \tag{Eq. 35}
\]

\(^{16}\) See chapter 4.1.3 (boiler pollution)
The problem that surfaces here is that the cold composite outcome temperature is unknown; however, in this case, it is determinative for the potential IDHE capacity. The outcome temperature is conversely depending on the IDHE capacity as well. This problem will be solved by the following equations.

By equating the formula for the demanded heat energy to heat the cold composite (Eq.36) with the specific heat exchanger formula (Eq.37)(which calculate the specific heat transfer capacity), it will result into a calculation of two equations with one unknown \( T_{C,Comp.out} \). This can be solved when one assumes that the average energy density \( C_w \) would be 3.6 [kJ/kg.k] and that the K and A factors are static. In practice, the heat transfer factor(K) would increase on the decreasing cold composite temperature, but for this equation it would only complicate the equation, and the influence would be negligible.

\[
Q_{IDHE} = M_{C,Comp} C_w (T_{C,Comp.out} - T_{C,Comp.in}) \quad \text{Eq. 36}
\]

\[
Q_{IDHE} = K \cdot A \cdot \frac{\left( T_{H,Comp} - T_{C,Comp.in} \right) - \left( T_{H,Comp} - T_{C,Comp.out} \right)}{\ln \left( \frac{T_{H,Comp} - T_{C,Comp.in}}{T_{H,Comp} - T_{C,Comp.out}} \right)} \quad \text{Eq. 37}
\]

\( C_w \): Average energy density 3.6 [kJ/kg.k], at 40[bar] and a average temp. of 132.5 [°C]
\( T_{H,Comp} \): The hot composite temperature [°C]
\( T_{C,Comp.out} \): Cold composite output temperature [°C]
\( T_{C,Comp.in} \): Cold composite input temperature [°C]
The equation in values:

\[ 63.9 \times 3.60 \times (x - 110) = 91.7 \times \frac{(258 - 110) - (258 - x))}{\ln(258 - 110 - \frac{258 - (258 - x))}{\ln(258 - x})} \rightarrow x = 158.7 \]

The result of this equation is that the outcoming cold composite temperature is 158.7[°C], which creates a heating capacity of 11.2[MJ] instead of the current 8.2[MJ]. It is positive that the IDHE capacity increase almost perfectly fulfilled the demanded heat energy. The slightly overheated (8.7[°C]) cold composite output is caused by the overcapacity of the IDHE, after all the necessary heat energy is 10.35[MJ](table 14)

Concluding, there could be argued that using the IDHE for heating the cold composite water up to 150[°C] is well applicable. However, there is extra work needed to pump the total amount of water trough the IDHE, and thus the IDHE would have an overcapacity of 0.85[MJ]. This causes that the cold composite output temperature will be lifted to an end temperature of 158.7[°C].
New boiler operation
Due to the enlargement of the drum capacity (11.2 MJ), the furnace capacity is forced to enlarge. This means that the produced heat energy in the furnace will have to increase. These enlargements result in a new boiler capacity balance. In the following paragraphs, the new boiler capacity balance will be calculated in order to check the feasibility in a technical and thermo-dynamical way. Firstly, the current boiler capacity balance will be calculated to provide the following variables; hot composite flow, temperatures and boiler efficiency, where-after the impact of the drum enlargement can be calculated. The impact of the Drum enlargement will be verified by calculating the new boiler balance, which provides new hot composite flow, temperatures and will make calculating the new boiler efficiency and furnace capacity possible.

The current boiler capacity balance
The current boiler capacity balance is displayed in figure 23 and 24. Only the input and output temperatures of the hot composite are known. The output temperature of the hot composite is managed on a minimum value and the maximum furnace temperature sets the input temperature. Equation 38 makes it be possible to calculate the mass flow that is needed to supply the total boiler heat energy demand.

\[
Q_{\text{boiler.total}} := M_{\text{H.Comp}} C_p \left( T_{\text{H.Comp.in}} - T_{\text{H.Comp.out}} \right) \quad \text{Eq. 38}
\]

\( Q_{\text{boiler.total}} \): total by the boiler extracted heat energy out of the hot composite flow 203150 [kJ]
\( M_{\text{H.Comp}} \): hot composite mass flow (flue gas) [kg/s]
\( C_p \): Specific heat energy density of the flue gas 1.3 [kJ/kg.k]

The boiler hot composite mass flow of 137[kg/s] will make calculating the intermediate hot composite temperatures possible. By deriving eq.38 to the following equations (eq.39-40), the intermediate temperatures become clear.

\[
Q_{\text{OVO}} := M_{\text{H.Comp}} C_p \left( T_{\text{H.Comp.Drum.out}} - T_{\text{H.Comp.OVO.out}} \right) \quad \text{Eq. 39}
\]

\[
Q_{\text{Drum}} := M_{\text{H.Comp}} C_p \left( T_{\text{H.Comp.in}} - T_{\text{H.Comp.Drum.out}} \right) \quad \text{Eq. 40}
\]

\( T_{\text{H.Comp.Drum.out}} \): Temperature of the hot composite after the Drum [°C]
\( T_{\text{H.Comp.OVO.out}} \): Temperature of the hot composite after the OVO [°C]
The intermediate temperatures are 526[^°C] after the drum and 360[^°C] after the OVO. The following equations, 41 and 42, will calculate the total boiler efficiency. The downside of equation 41 will calculate the total added energy from ambient air conditions to the maximum furnace temperature.

\[
\eta_{\text{boiler}} = \frac{Q_{\text{boiler.total}}}{Q_{\text{furnace.total}}} \\
Q_{\text{furnace.total}} = M_{\text{H.Comp}} C_p (T_{\text{H.Comp.in}} - T_{\text{amb}}) 
\]

Eq. 41

Eq. 42

\(T_{\text{amb}}: \text{ambient temperature [^°C]}\)

The total efficiency of the boiler is 82%, which means that 18% of the hot composite energy capacity will not be used and will be left in the output flow of hot composite. The complete heat balance of the current state will be displayed in figure 25.

![Figure 25. T/Q diagram current boiler process complete](image)

The optimized boiler capacity balance

Due to the enlargement of the drum capacity, the total boiler capacity balance will change. Equation 42 will underpin that the only option to enlarge the boiler capacity is by increasing the hot composite flow. The reason is that the hot composite boiler inlet and output temperatures are already operating on their technical maxima. However, this flow increase could have a negative influence on the boiler efficiency. By calculating the new hot composite flow and temperatures, it will be possible to conclude what the new boiler efficiency is, in order to conclude what the optimization disadvantages and benefits are.

The magnitude of the hot composite flow increase is dependent on the technical specifications of the drum optimization. On the first hand, equation 43 will clarify that an enlargement of the Drum heat energy \(Q_{\text{drum}}\) demand only can be occur when enlarging the heat transfer coefficient \(K\), heat transfer surface \(A\) or the average temperature difference between the hot and cold composite flow \(\Delta T_{\text{in}}\). On the other hand, equation 44 will describe that the drum heat energy enlargement will be created by a hot composite flow \(M\) increase or a larger difference between the incoming and output temperature of the hot composite flow. The relation between these two equations is that the hot composite output temperature is involved in both equations. The consequence of this is that when the drum heat energy demand increases and the \(K'A\) supposed to be constants (fig.26) the \(\Delta T_{\text{in}}\) needs to increase. Due to the constant cold composite temperature and the constant incoming hot composite, it will result in a hot
composite output temperature ($T_{H,\text{Comp.Drum.out}}$) increase. This has as reaction that the mass flow not only increases due to the $Q_{\text{Drum}}$, but also will lift further due to the higher $T_{H,\text{Comp.Drum.out}}$.

$$Q_{\text{Drum}} = K_A \Delta T_{\log}$$  \hspace{1cm} $Q_{\text{Drum}} = M_{\text{H,Comp}} C_p (T_{H,\text{Comp.in}} - T_{H,\text{Comp.out}})$  \hspace{1cm} Figure 26

When the heat transfer surfaces are enlarged to the size that the $T_{H,\text{Comp.Drum.out}}$ is equal to the current state, it will just lead to a mass flow increase due to the extra demanded heat energy (fig.27).

$$Q_{\text{Drum}} = K_A \Delta T_{\log}$$  \hspace{1cm} $Q_{\text{Drum}} = M_{\text{H,Comp}} C_p (T_{H,\text{Comp.in}} - T_{H,\text{Comp.out}})$  \hspace{1cm} Figure 27

These two options are calculated and drawn in table 15 in order to conclude which one of the options is the best.

$$Q_{\text{Drum}} = K_A \left\{ \frac{\left( T_{H,\text{Comp.out}} - T_{\text{C,Comp.in}} \right) - \left( T_{H,\text{Comp.in}} - T_{\text{C,Comp.out}} \right)}{\ln \left( \frac{T_{H,\text{Comp.out}} - T_{\text{C,Comp.in}}}{T_{H,\text{Comp.in}} - T_{\text{C,Comp.out}}} \right)} \right\}$$  \hspace{1cm} Eq. 43

$$Q_{\text{Drum}} = M_{\text{H,Comp}} C_p \left( T_{H,\text{Comp.in}} - T_{H,\text{Comp.Drum.out}} \right)$$  \hspace{1cm} Eq. 44

<table>
<thead>
<tr>
<th>Two different Drum optimizations</th>
<th>Current Drum</th>
<th>Optimization with current heat transfer surface</th>
<th>Optimization with enlarged heat transfer surface</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_{\text{Drum}}$</td>
<td>111</td>
<td>122</td>
<td>122</td>
</tr>
<tr>
<td>$T_{H,\text{Comp.out}}$</td>
<td>526</td>
<td>593</td>
<td>526</td>
</tr>
<tr>
<td>$M_{\text{H,Comp}}$</td>
<td>137</td>
<td>169</td>
<td>151</td>
</tr>
<tr>
<td>$\Delta T_{\text{in}}$</td>
<td>522</td>
<td>572</td>
<td>525</td>
</tr>
<tr>
<td>$K*A$ factor</td>
<td>213</td>
<td>213</td>
<td>233</td>
</tr>
</tbody>
</table>

Table 15. comparing the optimized boiler outcome with a boiler surface increase vs. the current surface(17)

The differences are evident. The unchanged drum heat exchanger surface will lead to a higher hot composite flow (169[kg/s]) and a higher output temperature (593 [°C]) compared to the drum with the enlarged heat exchanger surface. This drum with an enlarged heat transfer surface has a hot composite flow of 151[kg/s] and an unchanged hot composite output temperature (526[°C]). These differences are crucial for the boiler capacity balance, because the increase of mass flow due to the specific heat transfer equation (Eq.43) will higher the potential flue gas heat capacity further than the boiler demand. Table 16 will point that the new flue gas mass flows create a heat overcapacity of 27[MJ] in use of the current drum surface and an overcapacity of 5.2[MJ] in use of the optimized drum surface. This overcapacity will create a boiler efficiency decrease and boiler heat extraction excesses in the ECO and the OVO. The boiler efficiency decrease will occur due to equation 41, and becomes clear in a higher hot composite outcome temperature (fig.28).

The boiler heat extraction excess occurs due to the enlarged flue gas flow over the ECO and OVO who still has the current heat energy demand. In that way, the flow increase will create a smaller decrease of the flue gas temperature over the ECO and OVO. This will lead to $\Delta T_{\text{in}}$ increase and eventually result

---

17 The calculations behind the values are solved with the same equations as in 6.2.3.isthermic changes
in a heat extraction increase in the ECO an OVO (in the same way as described in thermo-dynamic changes of the IDHE).

<table>
<thead>
<tr>
<th>Heat energy demand Boiler vs. Flue gass heat supply</th>
<th>Optimization with current drum heat transfer surface</th>
<th>Optimization with enlarged drum heat transfer surface</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boiler heat demand [MJ]</td>
<td>[MJ]</td>
<td>[MJ]</td>
</tr>
<tr>
<td>Economiser</td>
<td>-28.9</td>
<td>-28.9</td>
</tr>
<tr>
<td>Overheater</td>
<td>-24.9</td>
<td>-24.9</td>
</tr>
<tr>
<td>Drum</td>
<td>-122.5</td>
<td>-122.5</td>
</tr>
<tr>
<td>Flue gas heat supply (between 1150°C and 225°C)</td>
<td>203.4</td>
<td>181.5</td>
</tr>
<tr>
<td>Total overcapacity</td>
<td>27.0</td>
<td>5.2</td>
</tr>
</tbody>
</table>

Table 16 boiler capacity surplus

Now that the influence of the flue gas flow is known, it could be concluded that the drum heat transfer surface enlargement is preferred. In that way the flue gas flow has the smallest increase \((137 --> 151 \text{ [kg/s]})\) which avoids a large boiler efficiency decrease and it would be far easier than creating a new boiler balance. Based on these outcomes, the suggestion would be to enlarge the drum heat exchanger surface to the current optimization.
New boiler balance point and efficiency

The drum capacity enlargement, including the heat transfer surface enlargement, creates a flue gas (called hot composite) mass flow increase of 13[kg/s] (9.5%). This increase results in a hot composite flow of 151[kg/s] trough the entire boiler. Instead of the way that the new ‘boiler hot composite flow’ is calculated, the new boiler balance will be calculated in the opposite way. The reason for these changes is that the hot composite boiler output temperature is restricted at 225[°C] and is a leading value in that way. This means that in the T/Q diagram (fig.29), the hot composite line will be fixed at 225[°C] and the new hot composite line increase angle is depending on the hot comp mass flow.

Figure 29. T/Q diagram new boiler balance

In order to calculate the new boiler balance and boiler efficiency, there is the need to make one statement: the K*A factor of heaters (Drum,OVO,ECO) are no static hard values. The K*A factors can change due to changing temperature differences over the heat transfer surfaces and by surface pollution. This means that by calculating the real new boiler hot composite using the new hot composite mass flow (151[kg/s]) the K*A factor of the Drum will change from the value calculated in table 15 (233).

Hot composite temperatures

By calculating the hot composite output temperature back to the furnace temperature, the missing in-between temperatures are calculable. The temperature outcomes are displayed in table 17 and calculated by equation 45.

\[ T_{\text{H.Comp.step.IN}} = \frac{Q_{\text{step}}}{M_{\text{H.Comp}}C_p} + T_{\text{H.Comp.step.out}} \]

Eq. 45

\( T_{\text{H.Comp.step.IN}} \): Incoming hot composite temperature of the particular heating step(Drum,OVO,ECO) [°C]
\( Q_{\text{step}} \): Heating demand of the particular heating step [kJ]
\( T_{\text{H.Comp.step.out}} \): the hot composite output temperature of the particular heating step [°C]
The new boiler balance is now complete. Compared to the current hot composite flow shows table 17 that the furnace input temperature need to be lower. The K*A factor will be calculated with equation 46. The outcome (table 18) shows that the K*A factor for the OVO and ECO decreases and the factor for the Drum increases. This means that the Drum surface still needs to be enlarged and the OVO and ECO are able to operate with the current heat exchangers, because the K factor (heat transfer coefficient), due to the polluting hot composite gasses, decreases and bring the brings the K*A factor down to the new OVO and ECO value. Because the Drum K*A factor is larger than the current, is it necessary to enlarge the heat transfer surface (A).

\[ K_A \triangleq \frac{Q_{\text{step}}}{\Delta T_{\ln, \text{step}}} \]  

Eq. 46

\( K_A: \) The heat transfer coefficient multiplied with the heat transfer surface  
\( \Delta T_{\ln, \text{step}}: \) The natural average logarithmic temperature difference over each heating step[^\text{C}].

<table>
<thead>
<tr>
<th>New K*A values</th>
<th>Current</th>
<th>Optimized</th>
</tr>
</thead>
<tbody>
<tr>
<td>( K_{A_{ECO}} )</td>
<td>316</td>
<td>306</td>
</tr>
<tr>
<td>( K_{A_{OVO}} )</td>
<td>245</td>
<td>313</td>
</tr>
<tr>
<td>( K_{A_{DRUM}} )</td>
<td>213</td>
<td>287</td>
</tr>
</tbody>
</table>

Table 18 K*A values

Boiler efficiency

Now that the new boiler balance is known, it is possible to calculate the new boiler efficiency by multiplying the total extracted heat out of the boiler (\( Q_{\text{Boiler.total}} \)) by the total extractible energy out of the hot composite flow till ambient conditions (\( Q_{\text{furnace.total}} \)).

<table>
<thead>
<tr>
<th>Boiler Efficiency [%]</th>
<th>( Q_{\text{Boiler.total}} )</th>
<th>( Q_{\text{furnace.total}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \eta_{\text{Boiler}} )</td>
<td>174.0 [MJ]</td>
<td>216.2 [MJ]</td>
</tr>
</tbody>
</table>

Table 19 boiler efficiency

The outcome of 80.5% (table 19) shows that the boiler efficiency decreases. The reason for this increase is the flow enlargement and the lowering in furnace temperature (the decrease of the furnace temperature (1150->1111.5[^\text{C}]) create a lower temperature drop (1111.5->225[^\text{C}]) in relation to the total possible temperature decrease (1111.5->10[^\text{C}]))
**New furnace capacity**

The furnace capacity is forced to increase by two reasons. The first reason is that the boiler efficiency is decreased, what suggests that there is more heat energy needed. The second reason is that the optimization suggests extracting 11.2[MJ] extra out of the Drum. Equation 47 enables to calculate the current and the new furnace capacity, which are drawn in table 20.

\[ Q_{\text{furnace}} = (Q_{\text{Drum}} + Q_{\text{IDHE}} + Q_{\text{ovo}} + Q_{\text{eco}}) \eta_{\text{boiler}} \]  \hspace{1cm} \text{Eq. 47}

<table>
<thead>
<tr>
<th>Boiler heating demand</th>
<th>DRUM</th>
<th>IDHE</th>
<th>OVO</th>
<th>ECO</th>
<th>total boiler demand</th>
<th>Boiler efficiency</th>
<th>Needed furnace capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Current state [MJ]</td>
<td>111.1</td>
<td>0</td>
<td>24.9</td>
<td>28.9</td>
<td>165.0</td>
<td>82%</td>
<td>201.2</td>
</tr>
<tr>
<td>Optimized state [MJ]</td>
<td>111.2</td>
<td>11.3</td>
<td>24.9</td>
<td>26.6</td>
<td>174.0</td>
<td>80.5%</td>
<td>216.2</td>
</tr>
</tbody>
</table>

**Table 20 Boiler capacity balance current vs. optimized**

**Process benefits**

What the specific technical process benefits are, depends on the way of technical operation. It is clear that the MP-turbine bleed will no longer used for pre-heating the condensate. This adds the possibility to increase the turbine work, or to enlarge the extern steam supply (ESS). Both options will be calculated and compared to the current operation in order to decide which way of operating has the best benefits. The first paragraph describes what the new total plant efficiency (TPE) and R1 mark will be after the turbine work enlargement. The second paragraph will calculate the needed ESS extraction to reach the suggested R1 mark increase.

**Technical new plant operation by the increase of the turbine work**

The turbine work will increase because of a couple of different process changes. Firstly by pre-heating in the IDHE (110->150[^\circ C]). Secondly, by shifting the degas process step to a lower steam pressure level. Thirdly, by extracting the ESS out of the MP-bleed instead of using the TSP.

The MP-steam flow will not leave the turbine, and expand in the turbine to 0.15[bar]. This extra work is calculated by equation 48 and will create an extra turbine work of: 1960[kW] by closing the 4[kg/s] MP-bleed.

\[ W_{\text{turbine}} = \eta_{\text{turbine}} \cdot M_{\text{MP\_reduction}} \left( h_{\text{MP\_steam}} - h_{\text{turbin\_out}} \right) \]  \hspace{1cm} \text{Eq. 48}

\[ W_{\text{turbine}} \] is the turbine work increase [kW]

\[ \eta_{\text{turbine}} \] is the turbine generator efficiency, 98[%]

\[ M_{\text{MP\_reduction}} \] is the steam mass flow which no longer will be drain for pre-heating, 2.8[kg/s]

\[ h_{\text{MP\_steam}} \] is the enthalpy of the MP-steam 2885[kJ/kg]

\[ h_{\text{turbin\_out}} \] is the enthalpy of the turbine output steam, 2395 [kJ/kg]

Shifting the heat energy demand for the degassing process from MP-level to LP-level will lead to an negligible turbine work increase. Currently, 0.196[kg/s] MP-steam would be extracted for degassing. In the optimized state there will be 0.033[kg/s] (tab.13) extracted for the gassing. The extra turbine work would result in a turbine work growth of ~ 71[kW].

Shifting the turbine supply pump (TSP) load of 3 [kg/s] to the MP-turbine bleed will bring an isentropic efficiency increase of 25% (TSP(50%) vs. turbine(75%)). The TSP creates a pump work of 770[kW] and by shifting this load to the steam turbine, it would create a work increase of 1040[kW], an increase of 270[kW]. In total, the result of stop heating the supply tank with MP-steam and shifting the ESS steam demand to the MP-bleed would lead to a turbine work increase of ~2300[kW].
**Influence on the total plant efficiency and R1 mark**

The influences of the extra turbine load and the lower boiler efficiency enables to calculate the new TPE and R1 mark. The TPE and the R1 mark are calculated, with the use of equation 49 and 50, in appendix 2.1. The outcome is pointed out in table 21.

![Equation Image]

**Eq. 49**

\[
\eta_{\text{total}} := \left( \frac{Q_{\text{turbine}} \cdot \eta_{\text{generator}} + Q_{\text{ESS}} + Q_{\text{TSP}} \cdot \eta_{\text{TSP}}}{Q_{\text{boiler}} \cdot \eta_{\text{boiler}}} \right)
\]

**Eq. 50**

\[
\eta_{\text{R1}} := \frac{E_{\text{p}} - (E_{\text{f}} + E_{\text{i}})}{0.97 \cdot (E_{\text{w}} + E_{\text{l}})}
\]

The outcome in table 21 shows that the optimized turbine operation does not create the suggested TPE and R1 mark increase. The TPE decreased with 0.8% and the R1 mark with 0.018. These declines of efficiencies are the result of the decline of the boiler efficiency. Equation 51-52 will substantiate this TPE decline. This equation shows that an increase of the turbine work $\pm 3\text{[MW]}$ (table 21) and an unchanged ESS will not surpass the increase of the produced boiler heat of $15\text{[MJ]}$, what the TPE decline caused.

![Equation Image]

**Eq. 51 Current Process**

\[
\frac{(48.18 \cdot 2.6) + (5.2 \cdot 1.1) - (0.2 + 1.6)}{0.97 \cdot (201.2 + 0.2)} = 0.627
\]

**Eq. 52 Optimized process, unchanged ESS**

\[
\frac{(48.07 \cdot 2.6) + (5.0 \cdot 1.1) - (0.2 + 1.6)}{0.97 \cdot (216 + 0.2)} = 0.614
\]

The reason for the R1 mark decline is partly the same as the reason for the TPE decline. The main reason is still the boiler efficiency decrease; beside this, the turbine efficiency and the ESS efficiency will also affect the R1 mark outcome. Because it seems to be that the multiplications of the $E_{\text{p,elec}}$ and $E_{\text{heat}}$ with the constants 2.6 and 1.1 are based to recommend a certain capacity correlation between the processes in the upper half of the equation. This means that the Turbine and the ESS capacity will be multiplied to create a curtain ‘fictive capacity’ in order to recommend a way of processing. The following example will point that, with the current turbine and ESS efficiencies, the benefits of processing ESS is seriously recommended.

---

18 The European commission created this equation and in that way recommended minimum plant efficiency and a way of operating.
Example\textsuperscript{19}: handling 1\,[kg/s]\, MP-steam in the turbine will produce 0.490\,[MW]. When this 1\,[kg/s] will be handled to for the ESS it would create 2.249\,[MJ]. For the ‘fictive capacity’, this would mean that EP\textsubscript{elec} will be 1.247\,[MW] and the EP\textsubscript{heat} 2.473\,[MJ]. This means a fictive capacity difference of 51%.

From this example follows that the best way of processing is not enlarging the turbine work. It will point out that the enlargement of the turbine work (-3\,[MW]) does not even approach the boiler capacity enlargement, let alone increasing the R1 mark.

Technical new plant operation by extracting more steam for the ESS
Out of the above made conclusion, that the R1 mark only will increase due to a large enlargement of the ESS, will rise the question how large this ESS need to be. In this paragraph there will be calculated what the minimum ESS enlargement need to be, to increase the R1 mark.

It is clear that the demanded ESS enlargement is necessary to overrule the large boiler capacity increase, which occurs due to the lowering in boiler efficiency. By modeling equation 50, it is possible to solve the minimum ESS flow, which is necessary to create the current R1 mark. This equation is solved in appendix 2.3, which shows that the minimum ESS flow to equal the R1 mark to the current value is 12\,[kg/s]. Table 22 will point the appendix outcome compared to the current values.

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Current state</td>
<td>201.2</td>
<td>45.18</td>
<td>3</td>
<td>0.77</td>
<td>3</td>
<td>6.75</td>
<td>26.2%</td>
</tr>
<tr>
<td>Optimized plant operation (minimum ESS enlargement)</td>
<td>216.2</td>
<td>41.18</td>
<td>8</td>
<td>2.05</td>
<td>12</td>
<td>26.99</td>
<td>32.5%</td>
</tr>
</tbody>
</table>

Table 22 minimal ESS to approach the current R1 mark

Table 22 shows that the needed minimum ESS flow will create a decrease of 4\,[MW] turbine work and an increase of ~20\,[MW] extra ESS capacity. The total plant efficiency will increase to 32.5\%, which is an excellent outcome. However, these outcomes suggest that the external heat extracting company could use an extra heat supply of 20\,[MW].

Summarized, the two different ways of operation, in order to adjust the IDHE optimization, create strongly varying outcomes. Both options are influenced by the decreased boiler efficiency(80.5\,[\%] vs. 82.3\,[\%]), which creates a strong enlarged boiler capacity. The result of this is that the first plant operation suggestion (turbine work increase) creates a decrease on the TPE and the R1 mark. Solely an increase of turbine work will not increase the plant efficiencies. The second option of operation (enlarging the ESS flow) will only bring a positive R1 mark(>0.632) by passing an ESS flow of 12\,[kg/s], which is a four times the current ESS flow. The TPE, at that point, has a very good efficiency of 32.5\,[\%]. The summarized outcomes are pointed in the table below (table 23).

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Current state</td>
<td>201.2</td>
<td>45.18</td>
<td>3</td>
<td>0.77</td>
<td>3</td>
<td>6.75</td>
<td>26.2%</td>
</tr>
<tr>
<td>Optimized plant operation (ESS not enlarged, TSP off.)</td>
<td>216.2</td>
<td>48.07</td>
<td>0</td>
<td>0</td>
<td>3</td>
<td>6.75</td>
<td>25.4%</td>
</tr>
<tr>
<td>Optimized plant operation (minimum ESS enlargement)</td>
<td>216.2</td>
<td>41.18</td>
<td>8</td>
<td>2.05</td>
<td>12</td>
<td>26.99</td>
<td>32.5%</td>
</tr>
</tbody>
</table>

Table 23 summarized ESS to approach the current R1 mark

\textsuperscript{19} The equations are based on the ‘static turbine data’ out of Appendix 2.1
7. Final conclusion
The disclosed problem that Attero wants to deal with, ‘the increase of the national waste demand’ - which lowers the financial income - can be dealt with by optimizing the WTE plant efficiencies. By answering the main question “how can Attero increase the energy efficiency (26.2%) and R1 mark (0.632) by steam cycle improvements of the plant”, it becomes possible to clarify the possibilities to solve the national waste demand problem. This study presents the outcome of the current process boundaries and a list of potential process optimizations, which ultimately leads to one complete calculated optimization.

In this conclusion, a sum of the outcome of this study will be made in three steps. The first step will describe the outcome of the current steam cycle state description; the second step will describe the potential optimizations; step three will describe the final optimization outcome in order to answer the main question.

The ‘current state steam cycle’ study displays the part process boundaries and the current energetic efficiencies. Out of this part study, it becomes clear that the following part processes caused a negative influence on the total plant efficiency.

1. The air condenser has a capacity shortage which will caused an turbine work increase of 9.2% on hot summer days.
2. The economizer restrictions for the cold composite input caused the need to pre-heat the cold composite (till 150° C) before it enters the economizer. This will reduce the plant capacity with 26.5[MJ] and will decrease the plant efficiency.
3. The pre-heaters 10 will create a self-reinforcement due to the capacity shortage of the condenser. This shows that the ambient temperature fluctuation will be reinforced by the extracted amount of steam for pre-heater 10. The suggested fluctuations result in turbine work fluctuation and a fluctuating TPE.
4. The way of managing the cold composite flow trough the pre-heaters will be controlled by the level indicator of the main condensate thank. This level will strongly vary due to pumping the cold composite by batch as well as the relatively small storage capacity. This results in a strong heat energy demand for the pre-heating steps, and will also result in fluctuations on the turbine work output; likewise, it will create fluctuations on the TPE.

The following part process is recommended for optimization: The external steam supply, because this part process will increase the TPE and R1 mark due to the high efficiency of this part process.

The suggested optimizations to increase the TPE and the R1 mark are based on the part process boundaries that are described above. The following suggestions where made:

1. Air condenser capacity enlargement to reduce the turbine work losses in hot summer conditions.
2. Enlarging the extern MP-steam supply possibilities, by changing the work balance between the high pressure and low-pressure turbine. In practice, the result of this would be that the diaphragm valve would manage on a far higher pressure than currently and likewise will lead to the possibility to extract MP-steam out of the LP-drain.
3. Changing the way of controlling the cold composite flow over the pre-heat part process by making the supply-tank level indicator a leading value instead of the mean condensate tank level indicator. This would establish the heat energy demand of the pre-heating steps and could lead to a more constant turbine work.
4. Lowering the pre-heat energy demand by shifting the supply tank heating demand to the IDHE. This would lead to the possibility to enlarge the turbine work or the ESS and would benefit the TPE and the R1 mark.

The last optimization suggestion is selected as the option with the most potential and has been calculated in order to check the technical feasibility. The outcome is drawn in the following text.
The optimization suggestion will affect the current process more than expected. The following changes occur during this process optimization:

The heat energy leak, which enables the degassing process, will decrease due to the lower boiling point. The flow over the IDHE will increase to 63.9[kg/s], but will create an increase of the pump work in order to squeeze the enlarged flow through the current pipes. Thermodynamically, the IDHE extracts more heat out of the drum than expected. The IDHE will heat the cold composite up to 158[°C] instead of the set point value of 150[°C]. This will bring an IDHE heat extraction 11.2[MW] instead of the demanded 10.35[MW]. This heat extraction enlargement leads to an increase of the total boiler heat demand, and causes a furnace heat supply increase. The balance of the new heat supply and heat extraction in the three boiler steps (Drum, OVO, ECO) will be realized in three ways: by a flue gas (hot composite) flow increase of 14[kg/s] (151-137), an increase of the Drum heat transfer surface, and a small decrease of the heat transfer coefficient(K) of the OVO and ECO heaters. This decrease will be caused by the natural flue gas pollution against the OVO and ECO heat transfer surfaces. To anticipate more precisely on the demanded heat energy and the constant hot composite output temperature, the hot composite input temperature shifts to a lower temperature of 1110[°C] instead of the current 1150[°C].

These boiler changes will lead to a boiler efficiency decrease of 1.5[%] (80.5 vs. 82)

At the consumer side of the steam cycle, there are two possible ways of managing the extra available steam on the MP-drain in order to raise the plant efficiencies. The first option is: creating more turbine work; the second option would be to use the available extracting space for an ESS enlargement. However, the study clarifies that the boiler efficiency decrease has a significant influence on the TPE and R1 mark, that optimization including the extra turbine work caused a decrease for the TPE and R1 Mark (25.4[%], 0.614). The second option is extracting more steam out of the MP-drain in order to enlarge the ESS. This option would neither increase the TPE and R1 mark when the MP-drain in combination with the TSP extract only the current steam amount of 7[kg/s]. To create a R1 mark increase (>0.632) it is necessary to enlarge the current ESS a minimum of four times to 12[kg/s]. The TPE would in that case increase to 32.5[%].

Concluding: it is possible to increase the TPE efficiency and the R1 mark by using the IDHE for pre-heating. Besides, the furnace capacity enlargement would increase the waste handling capacity, which would result in a financial income increase.

The overall planning of this study prescribed that, after 5 months, a founded optimization study could be made for the Attero WTE plant. However, it worked out in a different way. It took months to understand the WTE plant before I could start with calculating the potential optimization. Subsequently, I realized that I studied on process details that were too small process and would not lead to significance process optimizations, but to 4 months delay.
8. Recommendations
The conclusion describes that the pre-heat optimization is realistic to optimization which will bring a TPE efficiency and the R1 mark increase. However, there are still a couple outstanding questions which first need to be answered before this optimization could be implemented.

1. Is there a external company interested in the extra heat energy supply.
2. Is it possible to establish, in clean boiler conditions, the boiler balance without the use of the current IDHE operation? This could result in a new boiler cleaning schedule, an increase of the operation time of the water injectors and PPC valve.

Recommendations for further research:

1. To increase the benefits of this optimization it would be recommendable to search for a solution to the decrease of the boiler efficiency. An increase of the boiler efficiency would instant result in a further TPE efficiency and R1 mark increase. A suggestion to enable an increase the benefits could be: managing the furnace variables as oxygen inlet and furnace temperature.
Reference list

Articles:
  [https://brbs.webdog.nl/files/Naar%20duurzaam…def.%20versie.pdf](https://brbs.webdog.nl/files/Naar%20duurzaam…def.%20versie.pdf)
  [http://archief.knvk.nl/downloads/2k19704.pdf](http://archief.knvk.nl/downloads/2k19704.pdf)

Books:
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Appendix 1. Process data

Figure 30

Figure 31

Figure 32

Figure 33
Appendix 1. Current exergetic efficiencies

The energetic efficiency is an expression which will tell how efficient, an thermodynamic process will use their obtained potential heat energy. To rule on the energetic efficiency would it be necessary to compare the energetic qualification of the hot and cold composite (steam and condensate flow), this qualification is called Entropy in units of kJ/kg.K.. In other words, the entropy will be the specific energy content depending to the temperature in Kelvin, and will visualize the theoretical extractible energy till the 0 Kelvin (absolute zero point). The realistic extractible energy, which is the temperature difference to the ambient temperature, is called Anargy. The energy which is not extractable is called exergy, this generally is only the temperature difference between the ambient temperature and the 0 Kelvin point.

But applied to the current pre-heat cycle will there appear more exergy losses. These losses are create by the reducing entropy of the hot composite to the entropy value of the cold composite. Because this reduction will downgrade high qualified hot composite entropy, which is lost in the cold composite flow. This loss will called the irreversible entropy (\( \Delta S_{irr} \)). The specific energetic energy loss [kJ/kg] of this heating process is an multiplication of the \( \Delta S_{irr} \) with the ambient temperature(Eq.53), due to the fact that the \( \Delta S_{irr} \) is a certain heat capacity which is transformed from anargy to exergie. The energetic efficiency will be calculated by the exergy loss divided by the needed enthalpy increase of the cold composite(Eq.57).

In the following sub-paragraphs the energetic losses and efficiency calculated for the pre-heater 10,20 and the supply-tank in current state.

Appendix 1.2. Energetic losses Pre-heater 10,20

The exergy loss of pre-heater 10 and 20 will be calculated by the reference of the following figure 34. This figure displays the cold and hot pre-heating composites. The surface between the hot and cold composite will illustrate the exergy loss. Figure 34 will also show that after condensating of the hot composite it will further cool down. For calculating the exergy losses these small heating steps are not included, but assumed to be one horizontal condensating flow. This will bring the exergy loss a bit higher.

The calculation for the exergy loss:

\[
E_{loss} := \Delta S_{irr} \cdot T_{ambient}
\]

\( E_{loss} : \) Exergy loss [kJ/kg]

\( T_{ambient} : \) Ambient temperature (283[Kelvin])

\( \Delta S_{irr} : \) Irreversible entropy differences [kJ/kg.K]

As write before is the irreversible entropy loss the difference between the hot and cold composite difference this is visualized in Eq.54. Equation 55 and 56 will calculate the average temp. and heating energy. It is clear that the average hot composite temperature is equal to the condensation temp.

\[
\Delta S_{irr} := \left( -\frac{Q_1}{T_{av.hot}} \right) + \left( \frac{Q_1}{T_{av.cold}} \right)
\]

\( Q_1 := H_{av.hot} - H_{av.cold} \)
\[ T_{\text{av.cold}} := \frac{(T_{\text{out.cold}} - T_{\text{in.cold}})}{\ln \left( \frac{T_{\text{out.cold}}}{T_{\text{in.cold}}} \right)} \]

\[ H_{\text{av.cold}} := \frac{(H_{\text{out.cold}} - H_{\text{in.cold}})}{\ln \left( \frac{H_{\text{out.cold}}}{H_{\text{in.cold}}} \right)} \]

\( Q_1 \): enthalpy difference between the average hot and cold composite \([kJ/kg]\)

\( H_{\text{av.cold}} \): the cold composite logarithmic average enthalpy \([kJ/kg]\)

\( H_{\text{av.hot}} \): the hot composite average enthalpy \([kJ/kg]\)

\( T_{\text{av.hot}} \): Hot composite average temperature \([K]\)

\( T_{\text{av.cold}} \): cold composite average temperature \([K]\)

With the outcome of Eq. 57 and the demanded pre-heat energy it is possible to calculate the energetic efficiency(Eq.57).

\[ \zeta := 1 - \left( \frac{E_{\text{loss}}}{\Delta h_{\text{cold.comp.}}} \right) \]

\( \zeta \): Energetic efficiency \([\%]\)

\( E_{\text{loss}} \): Exergy loss \([kJ/kg]\)

\( \Delta h_{\text{cold.comp.}} \): Pre-heating energy demand \([kJ/kg]\)

The energetic efficiency as well as the exergy loss are for pre-heater 10 depending on the incoming cold composite temperature. Knowing that an lowering in incoming condensate temperature, due to a lowering ambient temperature, will increase the exergy loss and decrease the energetic efficiency.

<table>
<thead>
<tr>
<th>h_{\text{av.hot}} [kJ/kg]</th>
<th>T_{\text{av.hot}} [K]</th>
<th>h_{\text{av.cold}} [kJ/kg]</th>
<th>T_{\text{av.cold}} [K]</th>
<th>( \Delta S_{\text{irr}} ) [kJ/kg.K]</th>
<th>( E_{\text{loss}} ) [kJ/kg]</th>
<th>( \zeta ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pre-heater 10</td>
<td>2602.0</td>
<td>367.0</td>
<td>278.3</td>
<td>339.6</td>
<td>2323.7</td>
<td>0.51</td>
</tr>
<tr>
<td>Pre-heater 20</td>
<td>2723.0</td>
<td>393.0</td>
<td>401.3</td>
<td>367.8</td>
<td>2321.7</td>
<td>0.41</td>
</tr>
</tbody>
</table>

Table 24. energetic efficiency

The results which are displayed in table 24 show that the energetic efficiencies of pre-heater 10 and 20 are rather low or negative. This is caused by the relative small heating step of the pre-heaters versus a high exergy loss due to the high energy density of the hot composite.
Appendix 1.3. Energetic loss of heating in the supply-tank

The energetic efficiency of the supply-tank will be separated into efficiencies, one for the heating process and one for the degassing process. Assumed is that the demanded degassing heat is equal to the hot composite energy from 216°C to 170°C overheated steam. In that way is it possible to calculate the average temperatures of the hot composite in the two processes.

The calculations of the heating process are equal to the equations from appendix 1.1. (Eq 53-57), due to the condensing hot composite and the heating cold composite. For the degassing process is the process exactly the opposite, because the hot composite is cooling back and the cold composite is evaporating. This will result that the average hot composite heat supply (h_{av.hot}) is now longer constant, but need to be a logarithmic average enthalpy (Eq. 58) what also applies for the average hot composite temperature (T_{av.hot})(Eq.59).

\[
H_{av.hot} = \frac{H_{out.hot} - H_{in.hot}}{\ln \left( \frac{H_{out.hot}}{H_{in.hot}} \right)}
\]

Eq. 58

\[
T_{av.hot} = \frac{(T_{out.hot} - T_{in.hot})}{\ln \left( \frac{T_{out.hot}}{T_{in.hot}} \right)}
\]

Eq. 59

- \( H_{av.hot} \): the hot composite logarithmic average enthalpy [kJ/kg]
- \( H_{in.hot} \): the hot composite average enthalpy [kJ/kg]
- \( T_{av.hot} \): Hot composite average temperature [K]
- \( T_{in.hot} \): cold composite average temperature [K]

The current exergy loss is rather low (76.9, 63.9[kJ/kg])(table.25) due to the small temperature differences between the hot and cold composite(for both processes). The energetic efficiencies has on both the processes a high percentage(51%, 97%). This is the result of an low \( E_{loss} \) and for the degassing process also the large heating range (\( \Delta h_{cold} \)) which increases the energetic efficiency up to 97%, which is perfect!

<table>
<thead>
<tr>
<th>Supply-tank</th>
<th>( h_{av.hot} ) [kJ/kg]</th>
<th>( T_{av.hot} ) [K]</th>
<th>( h_{av.cold} ) [kJ/kg]</th>
<th>( T_{av.cold} ) [K]</th>
<th>( \Delta S_{irr} ) [kJ/kg.K]</th>
<th>( E_{loss} ) [kJ/kg]</th>
<th>( \zeta ) [%]</th>
</tr>
</thead>
</table>
| Heating (condensing steam) | 2746.0 | 423.0 | 549.2 | 402.0 | 2196.8 | 0.27 | 76.9 | 51%
| Degassing         | 2837.7 | 454.0 | 1439.1 | 423.0 | 1398.7 | 0.23 | 63.9 | 97%

(Assume that the enthalpy \( h_{out} \) of the degassing output is 2791 (170°C))

Table 25

51
Appendix 2.1 Current state TPE and R1 mark

### STATIC TURBINE DATA

<table>
<thead>
<tr>
<th></th>
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<tbody>
<tr>
<td>Turbine.in</td>
<td>400</td>
<td>40</td>
<td>3212.6</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine.MP</td>
<td>216.4</td>
<td>6.5</td>
<td>2885</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine(LP)</td>
<td>128</td>
<td>2</td>
<td>2723</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine.ELP</td>
<td>94.5</td>
<td>0.86</td>
<td>2602</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine.out</td>
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<td>0.15</td>
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<tr>
<td>TSP.out</td>
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<td></td>
<td>2951.3</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ambient</td>
<td>10</td>
<td>1</td>
<td>636</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Flow

- **Turbine.flow.in**: 63.9
- **Generator efficiency**: 98%

### Current state

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
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<tbody>
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<td>Boiler</td>
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</tr>
</tbody>
</table>

### Bleeds

- **HP-bleed**: 3.1
- **MP.ESS**: 4.15
- **LP.pre-heating**: 3.5
- **ELP.pre-heating**: 2.8

### Produced energy

- **HP/MP**: 19.95
- **MP/LP**: 9.22
- **LP/ELP**: 6.46
- **ELP/Turb.out**: 10.47

**Sub.total**: 46.10

**Turb.work.elec [MW]** 45.18

**ηTPE** 26.2%

### R1 mark

- **EP_react.intern [GJ/year]**
  - **Degas**: 8632.4
  - **EP_TOTACT1.1**: 9495.7

- **EP_elec.intern [GJ/year]**
  - **TSP**: 1424864.4
  - **EP_elec_intern.TOTAL**: 24226.64899

- **EP Comercial**
  - **ESS**: 157879.8
  - **EPelec.comercial.TOTAL**: 173667.8

- **EP_total**: 3950800.3
- **EF**: 5048.0
- **EI**: 50150.0
- **EW**: 6345266.26

**R1 Mark** 0.632
## Appendix 2.2 TPE and R1 mark for optimized state (ESS not enlarged, TSP out)

<table>
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<tr>
<th>Temp.</th>
<th>Prssure</th>
<th>Enthalpy</th>
<th>Mass flow</th>
<th>Energy</th>
<th>η</th>
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</thead>
<tbody>
<tr>
<td>[°C]</td>
<td>[Bar]</td>
<td>[kJ/kg]</td>
<td>[kg/s]</td>
<td>[MJ]</td>
<td>[%]</td>
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<td>Boiler</td>
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### Bleeds

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<tbody>
<tr>
<td>HP-bleed</td>
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<tr>
<td>MP, ESS</td>
<td>3</td>
</tr>
<tr>
<td>LP, pre-heating</td>
<td>3.5</td>
</tr>
<tr>
<td>ELP, pre-heating</td>
<td>2.8</td>
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</table>

### Produced energy

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<table>
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<tr>
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<tbody>
<tr>
<td>HP/MP</td>
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<tr>
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### Turb. work elec [MW]

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<tr>
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### R1 mark

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<tbody>
<tr>
<td>Degas</td>
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</tr>
<tr>
<td>EP_{TOTAL}*1.1</td>
<td>2792.8</td>
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<tr>
<td>EP_{elec.intern} [GJ/year]</td>
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<tr>
<td>TSP</td>
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</tr>
<tr>
<td>EP_{elec.intern.TOTAL}</td>
<td>394112.5</td>
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<tr>
<td>EP Comercial</td>
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</tr>
<tr>
<td>EPelec.comercial.TOTAL</td>
<td>173667.8</td>
</tr>
<tr>
<td>EP_{total}</td>
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<tr>
<td>EF</td>
<td>5048.0</td>
</tr>
<tr>
<td>EI</td>
<td>50150.0</td>
</tr>
<tr>
<td>EW</td>
<td>6818854.3</td>
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<tr>
<td>R1 Mark</td>
<td>0.614</td>
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### Optimized plant operation (minimum ESS enlargement)

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### Produced energy

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### Turb. work.elec [MW]

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<td><strong>ηTPE</strong></td>
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### R1 Mark

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<td><strong>EP\text{heat.intern} [GJ/year]</strong></td>
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</tr>
</tbody>
</table>

| **R1 Mark**          |            |                |                 |                  | 0.632      |       |