CSPBankability Project Report

Draft for an Appendix D - Power Block

to the SolarPACES Guideline for Bankable STE Yield Assessment

Document prepared by the project CSPBankability funded by the German Federal Ministry Economic Affairs and Energy under contract No. 0325293.

Gefördert durch:

Bundesministerium für Wirtschaft und Energie

aufgrund eines Beschlusses des Deutschen Bundestages

Document properties

Title	CSPBankability Project Report Draft for an Appendix D - Power Block to the SolarPACES Guideline Bankable STE Yield Assessment
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Date	January 9, 2017

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D. Power Block Modeling

This appendix aims to provide an overview of major factors/dependencies influencing the power block efficiency, which in turn are required to predict or verify the annual performance of a solar power plant. Therefore the basics of a Rankine-type water/steam cycle are laid out and specifics of different cooling technologies are described.

The sub-system power block (PB) transforms the thermal energy of the solar filed, the storage and the auxilillary heater into electricity. The efficiency of this transformation is a main factor to predict or to verify the annual performance of a solar power plant (SPP). The aim of this Appendix is to show the major influence factors on this efficiency and to give an overview of the different possibilities for modeling the PB.As the heat input into the PB is not constant over time, the PB load varies. The main purpose of the PB model is to describe the behavior of its efficiency which depends on the load, ambient air / wet bulb temperature as well as the temperatures of the Heat Transfer Fluid (HTF). For detailed calculations of water/steam cycle so-called heat and mass balance tools are used which are capable to calculate process data such as mass flows, pressures, temperatures, enthalpies, heat flows and pressure losses of such power cycles. Those tools are able to calculate steady-state in full load as well as in part load. For annual yield calculation purposes the most important characteristic of the PB is its gross efficiency besides a few more process data which are mainly temperatures and mass flows of the working fluids. Heat and mass balance models could be directly linked to the SPP performance model. This is rather impractical since the detailed PB model requires a lot of simulation time. This is mainly the reason why this option is not going to be discussed further in this Appendix D. Another option would be to use a detailed model of the subject PB configuration to determine efficiency matrices for all load points of interest comprising all necessary information to interface the PB model other sub-systems. A third option is to model the load depending behavior of the PB based on correction curves. The last two options are going to be discussed in detail lateron in this Appendix D.

The Solar Field (SF) and the Power Block (PB) are the main two systems of the SPP. The detailed block configuration and the steam parameters depend on the chosen CSP technology and Heat Transfer Fluid (HTF) such as Parabolic Trough Collector (PTC) based on synthetic oil, direct steam generation or molten salt concepts or solar tower systems either based on molten salt or and volumetric air receiver. The sub-system PB basically consists of the Solar Steam Generator (SSG), the Steam Turbine (ST), the condenser and the feedwater preheaters as well as of balance of plant systems.

For the yield calculation it is necessary to know as well the power block efficiency as the interface variable to the solar part of the plant like e.g. HTF outlet temperature of the SSG in case of a PTC.

The following chapters provide an overview of major factors/dependencies influencing the power block efficiency, which in turn are required to predict or check the annual performance of a SPP.

D.1. Fundamentals of Rankine Cycle Modeling

D.1.1. Thermodynamic fundamentals of the Clausius Rankine Cycle

This chapter intends to introduce the thermodynamics of the Rankine Cycle. Understanding the relevant effects and interdependencies is necessary for a comprehensive representation of this sub-system in annual yield analysis. The experienced reader might skip this chapter.

The Rankine cycle is used to convert thermal energy to electrical energy. Water is the most common working fluid for this process. The simplest Rankine Cycle is based on four steps represented by the four components shown in the heat flow diagram below.

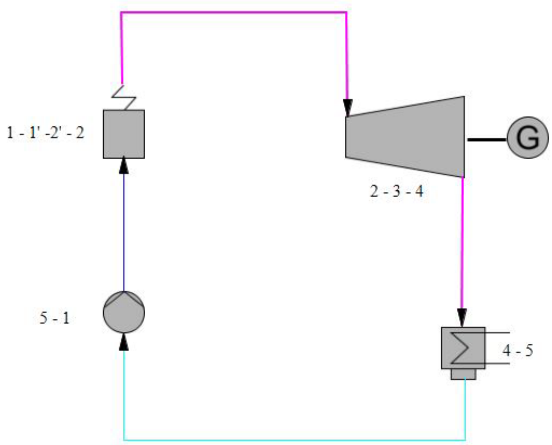

Figure D-1: Elementary Rankine cycle heat flow diagram

The Rankine Cycle as depicted above can be described with the following:

- The pressure of condensate at saturation temperature is increased with a pump (5 - 1).
- In the steam generator the water is heated, evaporated and the saturated steam is superheated (1 - 1’ - 2’ - 2).
- The superheated steam passes through the turbine and the turbine drives the generator (2 - 3 -4).
- Exhaust steam from the turbine is condensed in the condenser (4 -5).

This Rankine Cycle is usually shown in an enthalpy-entropy (h-s) or a temperature-entropy (T-s) diagram (see figures below).

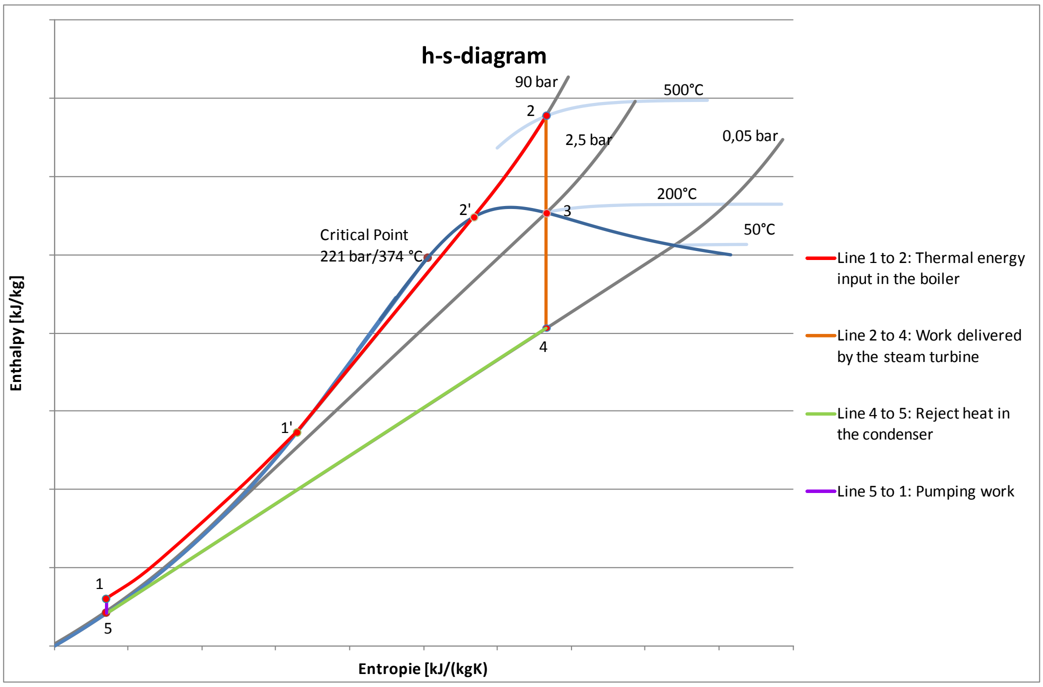

Figure D-2: Rankine Cycle in h-s diagram

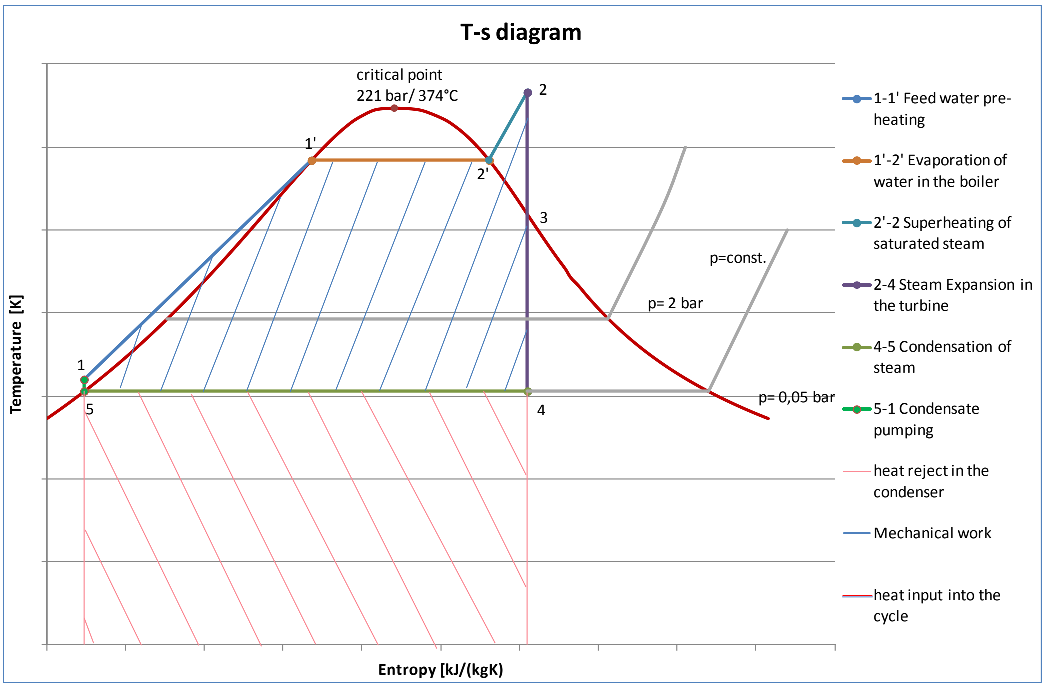

Figure D-3: Rankine Cycle in T-s diagram

Looking at the T-s-diagram, the heat input into the cycle is the integral of the curve 5-1-1’-2’-2-3-4. The losses in this case correspond to the rejected heat in the condenser, which is represented by the integral of the curve 4-5. The mechanical work is, according to the first law of thermodynamic, the difference between these two areas. The diagram shows that the mechanical work can be increased by using a higher “upper” temperature and the losses can be decreased by decreasing the “lower” temperature.

Figure D-4 and Figure D-5 show a heat cycle working with live steam parameters of 100 bar/383 °C and a condensate temperature of 41 °C with and without reheat. A comparison of the two figures shows that the mechanical work expressed by the area enclosed by 1-2-3-4-5-6-1 is increased while the losses do not increase in the same way. It is obvious that the efficiency with reheat is higher.

The entropy at point 6 is about 6.1 kJ/kg/K compared to 7.0 kJ/kg/K at point 8. This leads to a steam content of 0.84 instead of 0.72. In this example, it is assumed that the expansion is isentropic. In a real cycle process the turbine efficiency has to be taken into account. Therefore the entropy and the steam content will increase in both cases. To avoid damages on the turbine blades, wet steam with steam content lower than 86 % at the turbine exit should be avoided (tolerated wetness differs slightly among turbine suppliers).

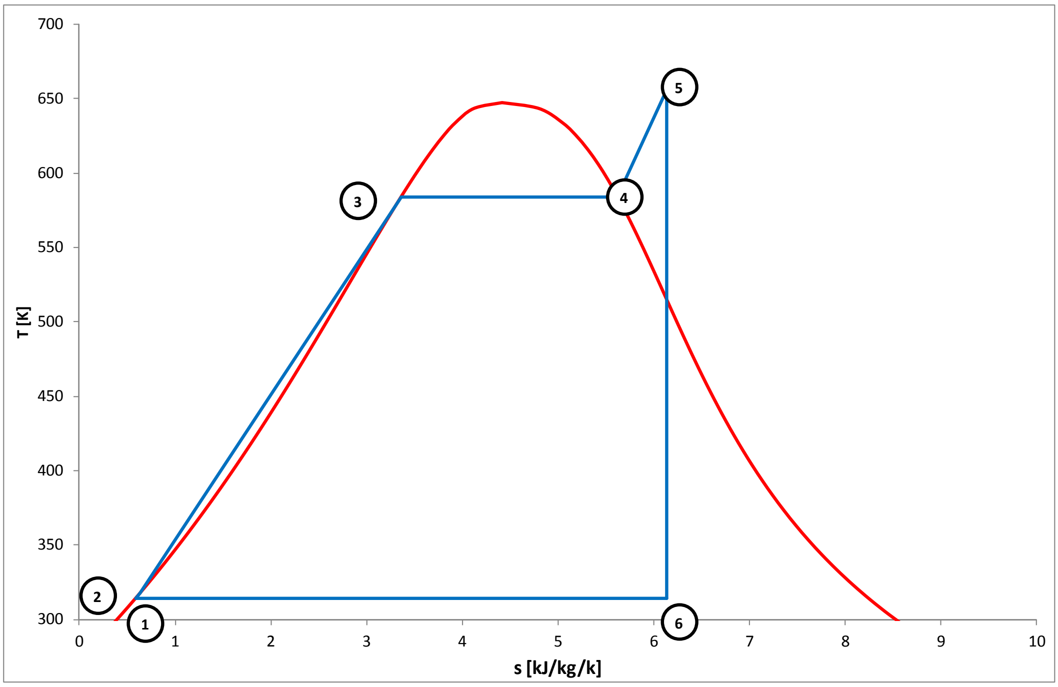

Figure D-4: T-s diagram without reheat

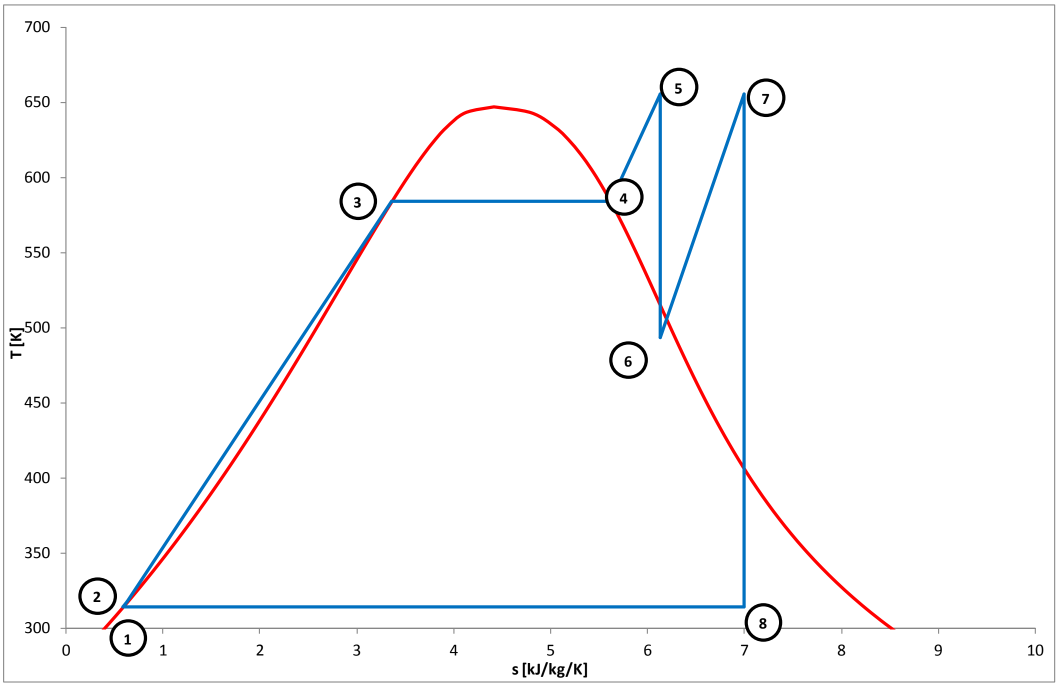

Figure D-5: T-s diagram with reheat

Figure D-6 and Figure D-7 depict the impacts on the heat cycle with and without reheat in the h-s diagram.

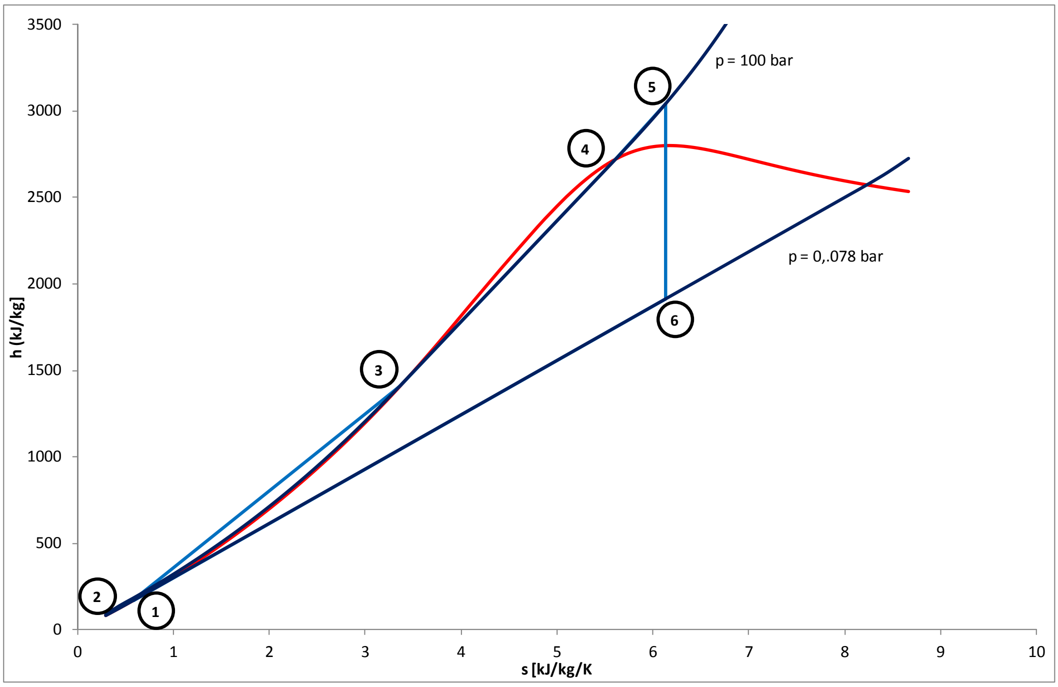

Figure D-6: h-s diagram without reheat

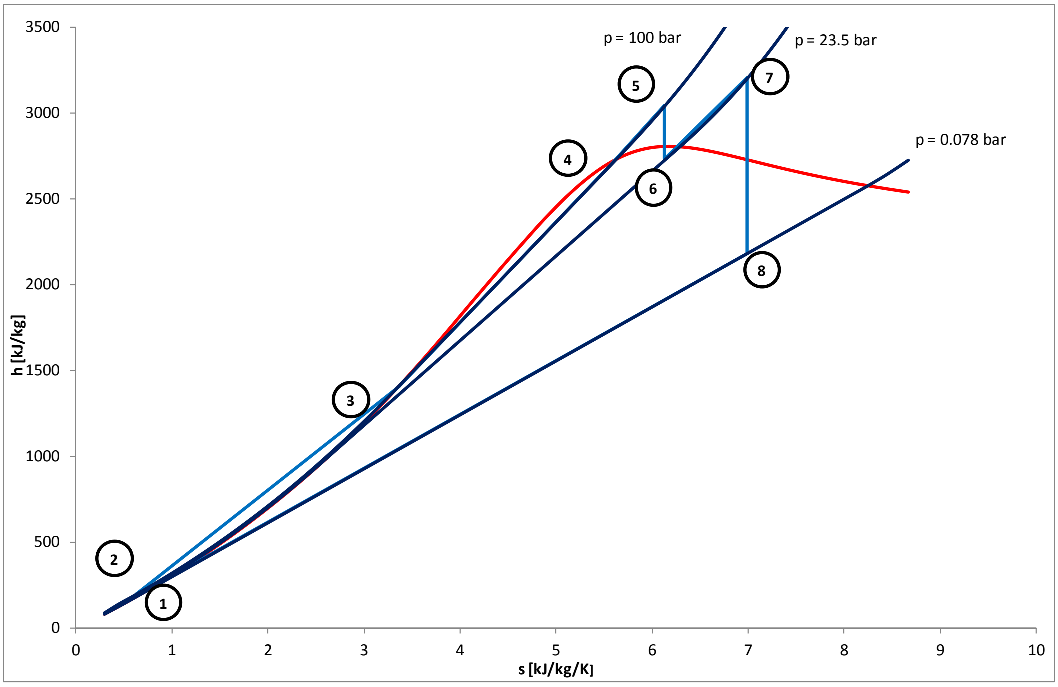

Figure D-7: h-s diagram with reheat

The Rankine Cycle is an irreversible process even if the turbine and the pumps work isentropic as shown in the figures above. This is related to the heating of the water in the steam generator. The process is irreversible due to the large temperature difference between the heating medium and the water/steam flow. In order to increase the cycle efficiency, feedwater pre-heating is applied, which results in lower temperature differences during heat transfer. Therefore, a regenerative cycle is applied (see Figure D-8 below). Steam from turbine extractions is forwarded to feedwater preheaters to increase the feedwater temperature step by step. The number of feedwater preheaters is essential for the power plant efficiency but basically more than nine (9) preheaters do not increase the efficiency significantly.

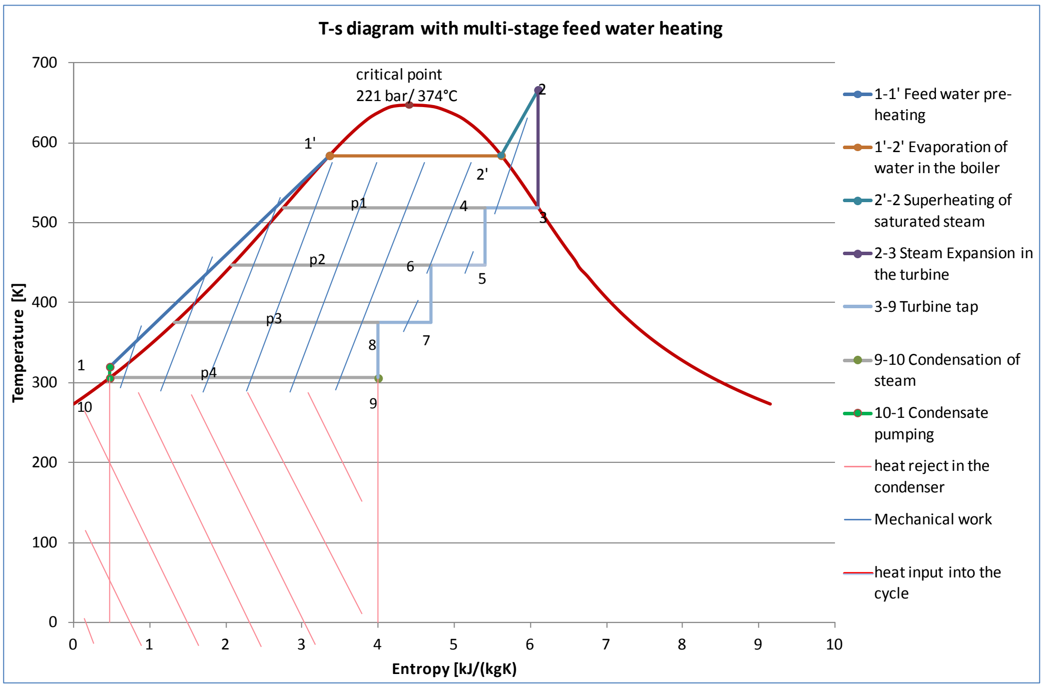

Figure D-8: Rankine Cycle with feedwater preheating

D.1.2. Impact of cooling conditions on the PB efficiency

The lower operation temperature of a heat cycle has a major influence on the cycle efficiency. It should be as low as possible. Basically, the use of once-through cooling would always result in the lowest temperature and thus result in the highest steam cycle efficiency. However, a large volume of cooling water is necessary. Due to the fact that STE plants are typically erected in desert regions with scarce water resources the dominating cooling applications are based on wet cooling tower and air cooled condensers (ACC). Further information concerning the water demand of different cooling technologies can be found in Appendix I (Water Demand, Quality and Supply).

Figure D-9 shows the gross efficiency curve of a parabolic trough power plant using synthetic oil as HTF versus load for a wet cooling option. The red curves have been determined based on heat cycle calculations for a PB with 6 feedwater preheaters and a steam temperature of 383 °C. For comparative analysis purposes, a reference Dry Bulb Temperature (DBT) of 42 °C has been considered. Two calculations with exhaust pressures of 150 mbar and 239 mbar lead to the lower two red curves shown in the diagram. The blue curves in-between represent the data from different designs for a 50 MW plant equipped with an ACC. The backpressure varies between 150 and 239 mbar. The variation of the gross efficiency is about 1.5 %.

The calculation with a wet cooling tower based on a wet bulb temperature (WBT) of 18 °C leads to approx. 2 % higher average gross efficiency compared to a plant with an ACC and 150 mbar backpressure.

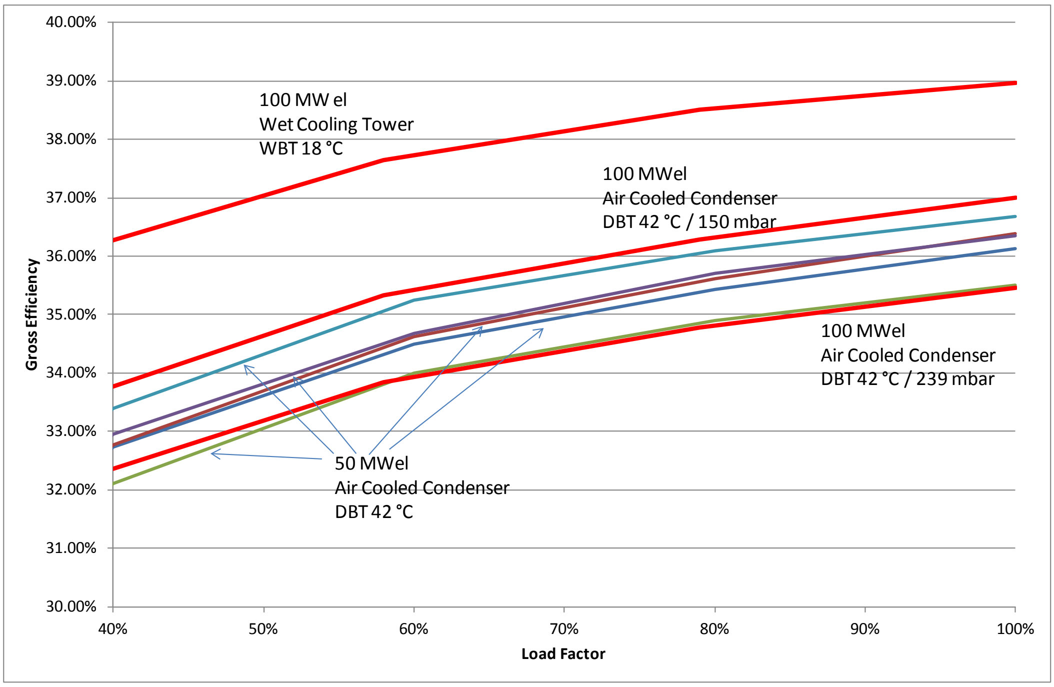

Figure D-9: Gross efficiency versus load factor and different backpressures in the range of 150mbar to 239 mbar

The efficiency of a power plant depends on the operation temperatures. The highest operation temperature is the live steam temperature of the cycle depending directly from the allowable HTF temperature. The lowest temperature of the cycle is the condensate temperature depending directly on the used cooling technology and the ambient conditions of the site.

The red curve in Figure D-10 shows the linear dependency between the PB efficiency and the condensate temperature of the reference design case. Changing the condensate temperature due to the ambient conditions leads to a part load operation of the power plant. The exhaust area of the Low Pressure (LP) turbine is fixed with the turbine design and therefore the exhaust losses are related to the volume flow and the exhaust pressure which determines the steam velocities. Lowering the exhaust pressure whilst maintaining the design mass flow constant leads to a higher volume flow and higher flow velocities. This increases the exhaust losses. For this reason the green curve representing the part load behavior of the power plant operated in a range of 2 °C to 18 °C WBT differs from the linear trend.

The condensate temperature is determined by the wet bulb temperature, the approach and the cooling range of the cooling tower plus the Terminal Temperature Difference (TTD) of the condenser.

In case of the reference design, the condensate temperature corresponds to 41 °C. This leads to a gross efficiency of about 39 %.

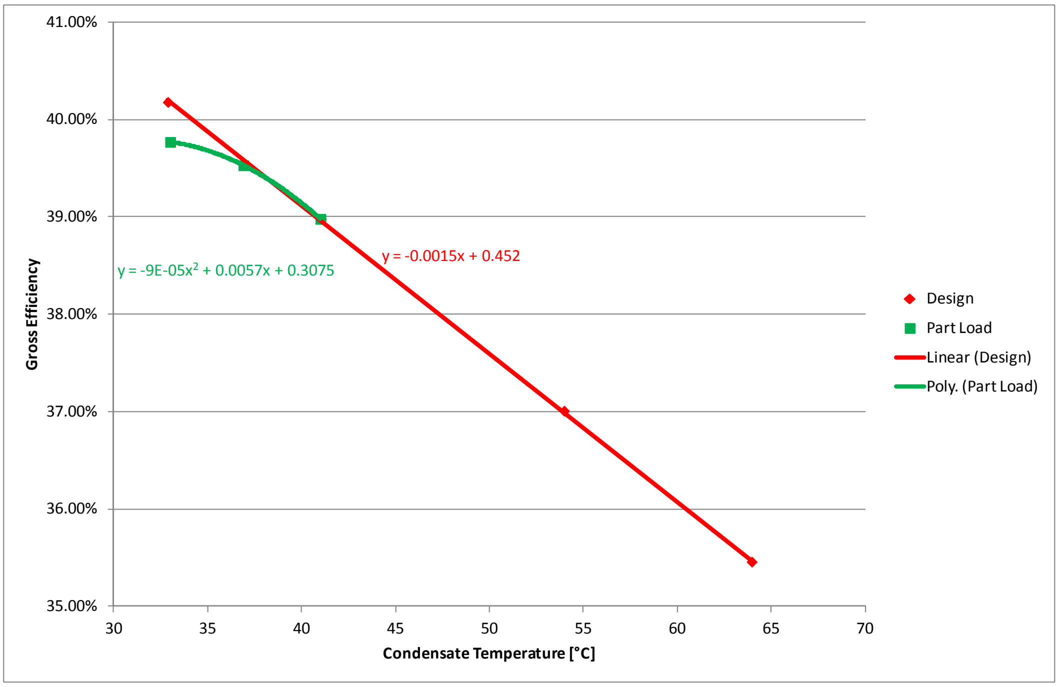

Figure D-10: Efficiency as a function of condensate temperature

D.1.3. Modeling of a wet cooling tower

The design of the cooling system is crucial for the performance of a Rankine Cycle plant. The following paragraphs illustrate the fundamentals of cooling tower design in order to work out the impact on the yield anlysis. For designing a power plant cooling system, some basic parameters have to be determined. The main parameters are as follows:

- Cooling tower heat load
- Cooling Range (CR)
- Approach (APP)
- TTD of the condenser

The temperatures relevant for the cooling tower performance can be obtained with the following equations:

$$T_{CW} = T_{WBT} + APP \quad (D.1)$$

$$T_{WW} = T_{CW} + CR \quad (D.2)$$

$$T_{Cond} = T_{WW} + TTD_{Cond} \quad (D.3)$$

with

T_{CW} .... Cold water temperature [K] @ outlet of cooling tower
T_{WBT} Wet bulb temperature [K]
APP .... Approach [K]
T_{WW} .... Warm water temperature [K] @ inlet of cooling tower
CR .... Cooling range [K]
TTD_{Cond} Terminal temperature difference of condenser [K]
T_{Cond} .... Condensate temperature [K]

The basis for the design of a cooling tower is the WBT. Figure D-11 shows the WBT duration curve on the basis of meteorological data for the reference site PSA (refer to Appendix B - Reference Systems). These data are used to describe the design criteria and the operation behavior of a wet cooling tower. The data obtained from this design are also used as input data for the modeling of the reference PTC SPP with wet cooling tower as well as to describe the behavior of a cooling tower in general.

Figure D-11 shows that 93 % of all WBT values are ≤ 18 °C. Therefore a WBT of 18 °C has been selected as the design temperature of the wet cooling tower. This 93%-criteria has been applied based on project experience with cooling tower suppliers. It must be noted that different criteria can be applied depending on the project-specific requirements. Basically, the selection of a design WBT intends to avoid oversizing and thus extra costs for the cooling system.

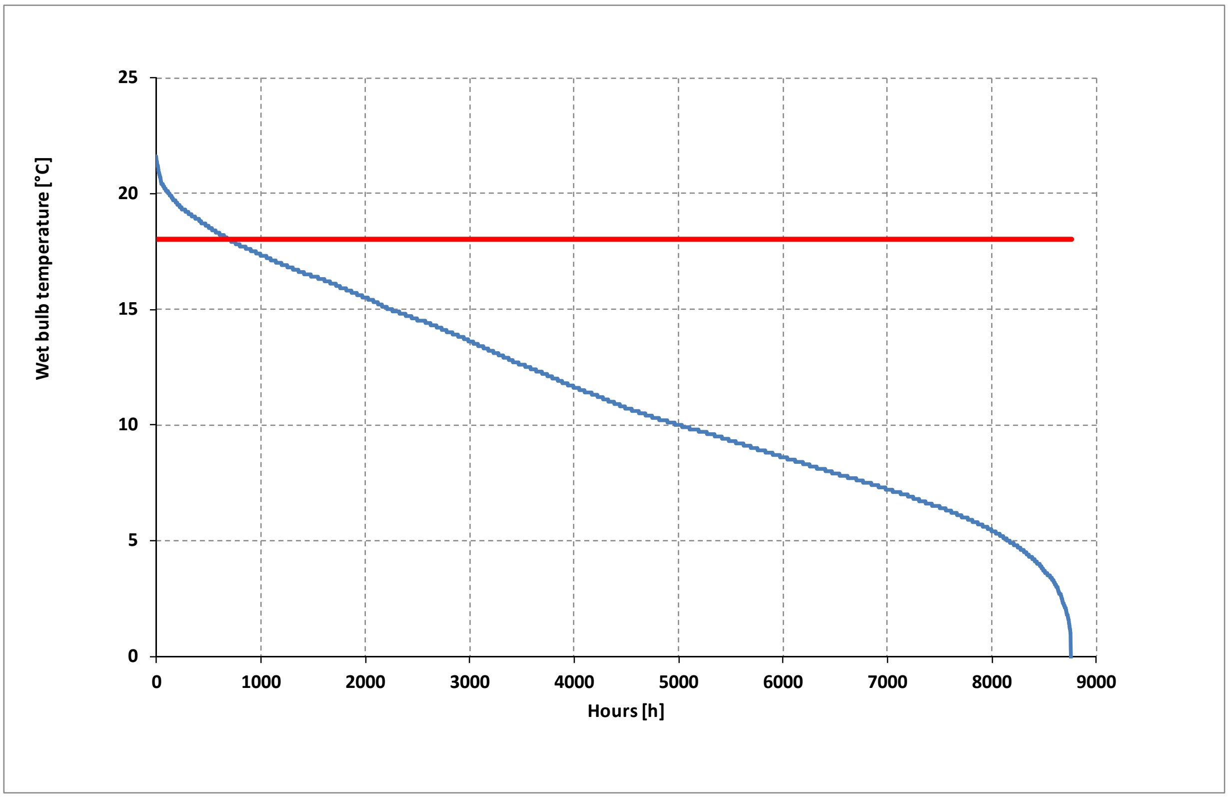

Figure D-11: WBT duration curve for reference site PSA

Cooling tower performance characteristics depend on the supplier. Figure D-12and Figure D-13 show two characteristic curves (nomogram) for different load factors versus WBT from different suppliers.

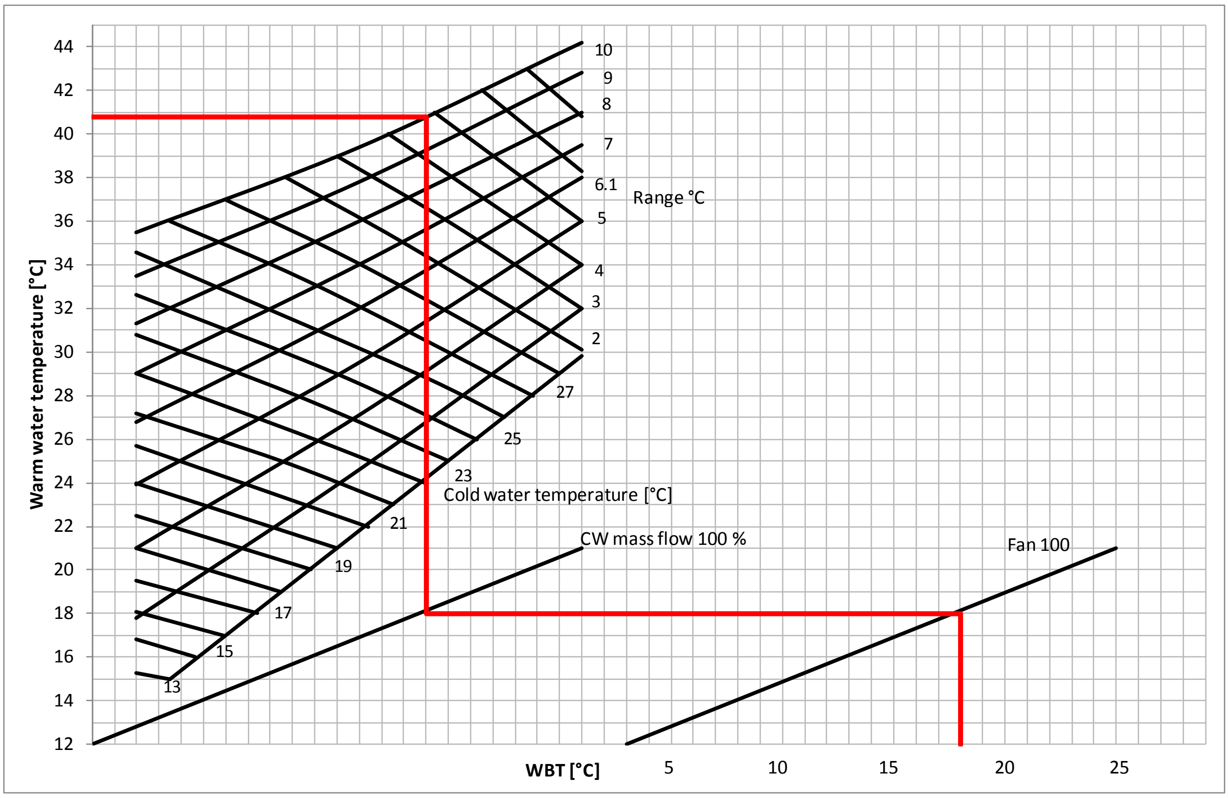

Figure D-12: Example 1 of a cooling tower characteristic curve for different load factors versus WBT

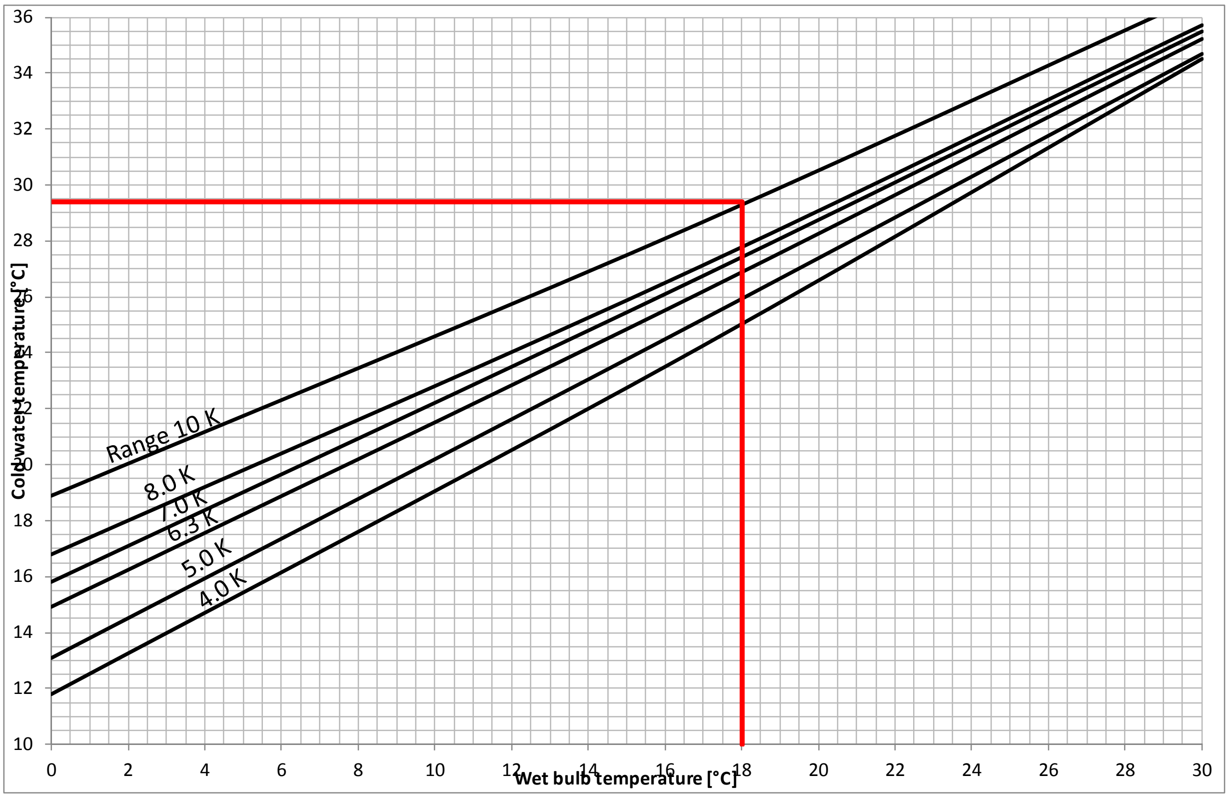

Figure D-13: Example 2 of a cooling tower characteristic curve for different load factors versus WBT

As an example, Figure D-13 depicts the cold water temperature as a performance parameter of the underlying wet cooling tower design (supplier-specific). The x-axis represents the WBTs, whereas on the y-axis the corresponding Cold Water Temperatures (CWTs) are shown in dependency on the selected cooling range (see indicated curves; e.g. top curve corresponds to a cooling range of 10 K). As can be seen, the CWT varies with the WBT. This dependency is also shown in Figure D-14 for the reference plant.

Due to the fact that the cooling function of the tower is based on evaporative cooling, the CWT depends strongly on the WBT. The CWT therefore can be below the dry bulb ambient air temperature. The water content of the air at 100 % humidity increases with increasing ambient temperature. Much more heat can be rejected by evaporation due to higher ambient temperatures and less heat has to be rejected by convection.

At the design point of 18 °C WBT the expected CWT is approx. 28 °C. This leads to an approach of 10 K. Increasing the WBT decreases the temperature difference between CWT and WBT (approach), whereas low WBTs lead to a higher approach. Therefore the CWT at 30 °C WBT is about 34 °C and at 5 °C WBT about 21 °C, the approach changes from 4 K to 16 K.

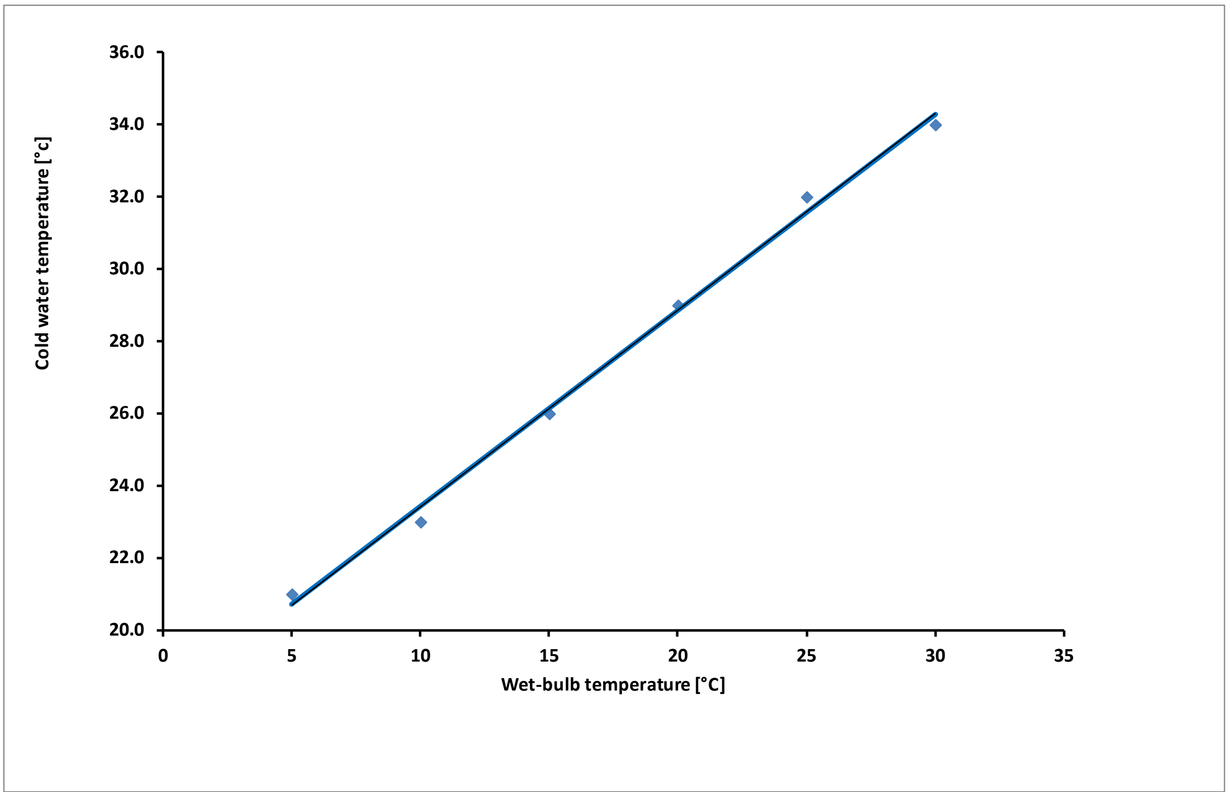

Figure D-14: Cold water temperature versus wet bulb temperature

Decreasing the load and keeping the cooling water flow rate constant leads to a decrease of the cooling range. Hence, the Warm Water Temperature (WWT) of the cooling water returning from the condenser to the cooling tower decreases. This influences the CWT which also decreases. This situation can also be seen in Figure D-15. Both effects stated above can be described by means of a linear function as shown in Figure D-14 and Figure D-15.

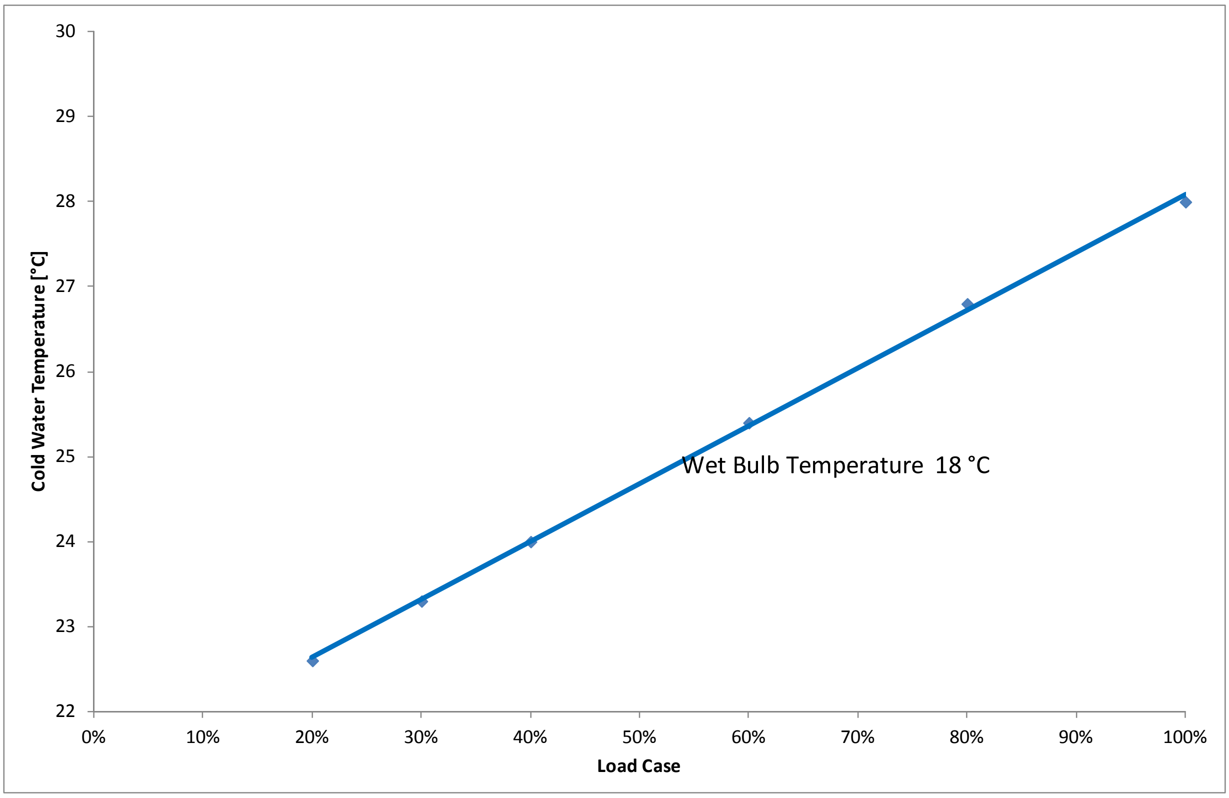

Figure D-15: Cold water temperature versus load

The approach changes with the WBT as shown in Figure D-16. Based on the approach calculated with this function, the dependency on the load can be further assessed (see Figure D-17). The approach for different part load cases is calculated with the linear function as shown in red which represents the curve for 11 °C WBT. Based on the WBT and the approach, the cold water inlet temperature of the condenser is determined. The cooling water flow is taken from the design case (18 °C WBT). The cooling range is determined by the rejected heat in the condenser and the TTD is calculated based on the k*A-value of the design case. These assumptions have been used to calculate the matrices in Section D.2.2.1.

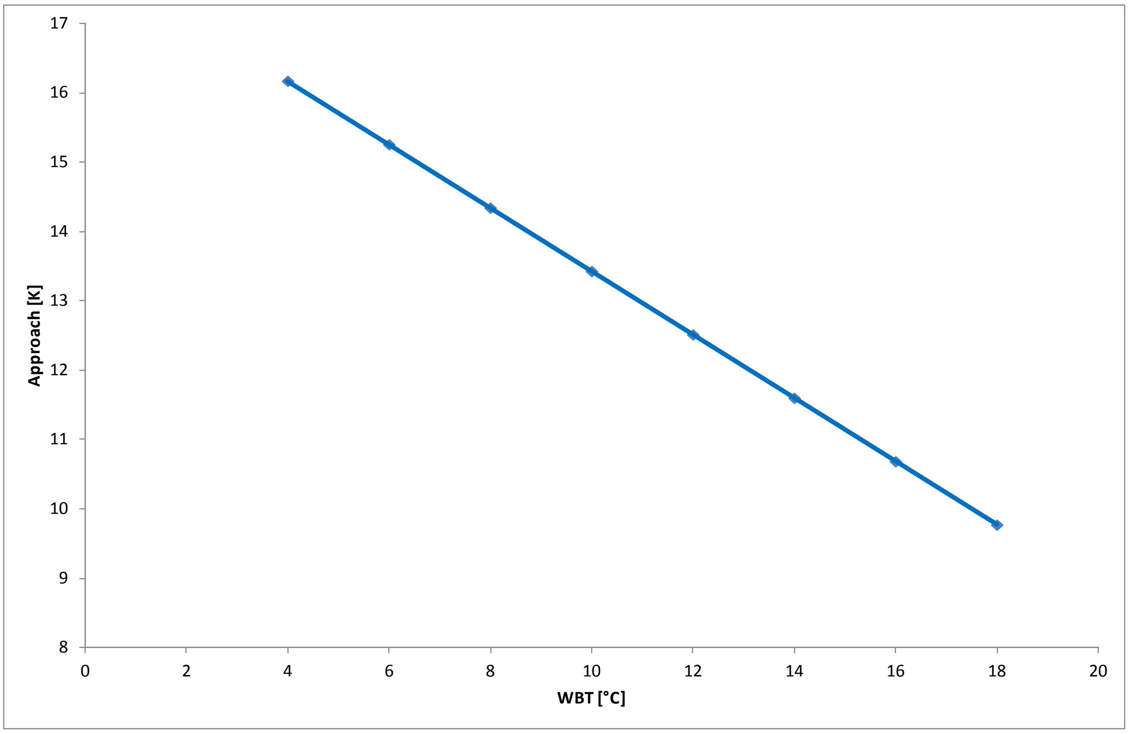

Figure D-16: Approach versus wet bulb temperature at 100 % load

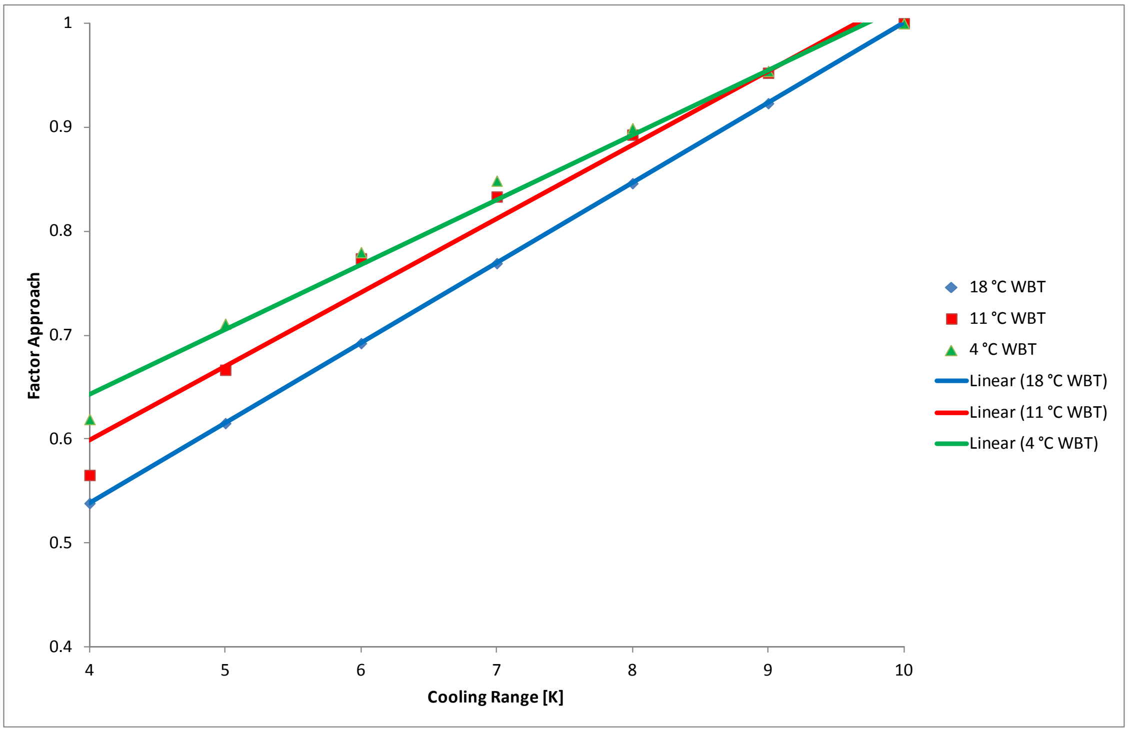

Figure D-17: Factor approach versus cooling range for different wet bulb temperatures

D.1.4. Modeling of air cooled condenser (ACC)

In many regionsthe availability of cooling water is limited and therefore restricted. In such a case an Air Cooled Condenser (ACC) is used. The condensate temperature which can be reached with this cooling option strongly depends on the ambient air temperature (dry bulb temperature). The design of an ACC is based on the Initial Temperature Difference (ITD). This is the temperature difference between condensate temperature and ambient air temperature. Normally the maximum ambient temperature is chosen as design temperature to make sure that the PB achieves the specified gross output. The ITD can be usually specified between 12 K and 30 K. The ITD influences the size and therefore the investment cost for the ACC. Conventional power plants have ACC with an ITD of about 25 K - 30 K, but in SPPs ITDs of about 15 K - 20 K are usual. This is due to the fact that the solar collectors are the major cost drivers and not the ACC which has comparatively less impact on the investment costs.

The dry bulb temperature is higher than the WBT and therefore even with small ITDs the condensate temperature is above the condensate temperature which can be reached using a wet cooling tower. The influence on the PB efficiency was already shown in Figure D-10.

Normally the range between the minimum and maximum ambient temperature on the site is higher than the variation of WBT. The approach decreases with lower WBTs. Therefore the variation of the condensate temperature using a wet cooling tower is smaller than the range which an ACC could theoretically reach. Due to the turbine design the minimum backpressure of the ST is limited. Hence, this pressure must be kept above minimum value by switching off some of the fans at low temperatures. This influences the trend of the efficiency of the PB at lower temperatures.

Figure D-18 shows a typical curve of the gross efficiency of a PB at 100 % load using an ACC. The design point of the ACC is 40 °C and the ITD is 15 K. All PB parameters are the same as shown in chapter D.2.1.2. In this case, ambient temperatures below 28 °C do not lead to higher efficiencies because the exhaust pressure is maintained based on the design of turbine last stage cross section. Therefore the efficiency is constant up to 28 °C. In contrast, with increasing ambient temperatures the efficiency is dropping following a parabolic function of the ambient temperature since the higher temperatures increase the exhaust pressure of the turbine.

This means that the approximation equations for an ACC are a function of ambient temperature at temperatures higher than 28°C and not a function of the WBT. At temperatures less than 28°C the efficiency and the plant output do not depend on the ambient temperature.

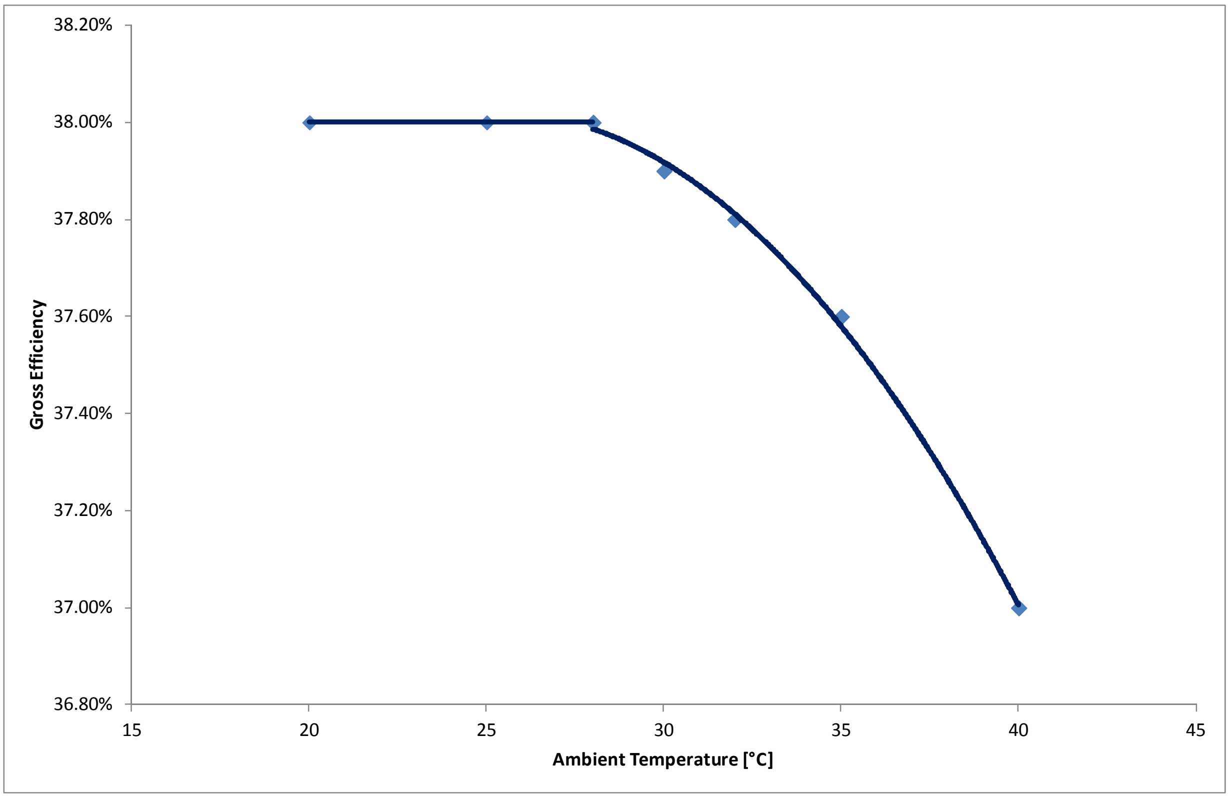

Figure D-18: Gross efficiency versus ambient temperature (ACC)

D.1.5. Modeling of an once-through cooling system

If there is a large amount of water available once-through cooling will be the option which leads to the highest PB efficiency. The design of the condenser is similar to the condenser required in case of cooling based on a wet cooling tower. The condensate temperature depends on the cooling range and the TTD of the condenser. Cooling ranges of 8 K up to 15 K are state-of-the-art, the TTD is between 2 K and 4 K. Since the variation of the (sea-) water temperature is smaller than the variation of the ambient air temperature, the range of the condensate temperature is smaller than in case of an ACC. In this case, the efficiency of the PB is a function of the cooling water inlet temperature.

D.1.6. Modeling transient effects

The start-up and shut down procedures of the power block are one of the main transient effects within a STE plant. Therefore, the mechanisms which lead to transient behavior need to be analyzed in detail so that their influence can be quantified. This chapter addresses procedures to be carried out regarding the main power block components steam generator, steam turbine and steam piping. Especially for thick-walled systems with high operating temperatures (live steam system, steam turbine) the impact on the lifetime is high due to highly transient operation. This leads to reduced

service life and increased operational costs . To mitigate lifetime consumption, power block operation must comply with maximum allowable gradients during normal operation modes.
For yield calculations only few transient situations of the power block are of interest. One of those is the start-up process. During start-up the external heat input can not be accounted for 100 % of electrical energy production, but must be used for removal of unusable energy and heating up of thermal masses.

D.1.6.1. Background: Illustration of rankine cycle power block start-up

A power block start-up procedure for rankine cycle systems is typically depending on the temperature state of the turbine and consequently on the downtime. Start-up processes can be classified into cold start, warm start, and hot start.
The start-up conditions given below have been derived from a STE plant in operation (parabolic trough, medium-sized TES) and can serve as orientation:

Table D-1: Exemplary classification of start-up processes

Start-up process[2)]	**Temperature HP turbine**	**Temperature LP turbine**	**Downtime duration**[1)]	**Start-up duration**[3)]
Cold start	< 180 °C	< 180°C	> 24 h	up to 120 min
Warm start	> 180 °C and < 280°C	> 200°C and < 320°C	up to 8 h	20 to 30 min
Hot start	> 280°C	> 320°C	up to 2 h	up to 15 min

1) Depends on TES size, location, insulation etc.; values indicative only
2) Please note the transition zones between cold and warm-start up where parameters are weighted from the extreme cases.
3) Start-up duration also depends on the irradiation situation during start-up, the size of solar field and TES size, values only indicative.

Note that the above given figures depend on the PB configuration and sizing as well as the supplier specification for the relevant equipment.

For the operation of a STE power block one warm start per day is typical, whereas this value depends on whether the plant is equipped with a large storage system which allows in certain seasons up to 24 h operation. The number of cold starts is limited and linked to the maintenance schedules which require plant shutdown for several days (downtime). Hot starts are assumed to have only minor impact on the yield calculation because of low thermal losses during the short downtime. The following description thus focusses on the warm start. Figure D-19 illustrates a typical rankine cycle layout for STE plants, including

- HTF heated steam system with economizer, evaporator, superheater and reheater
- Drum type evaporator with forced and/or natural circulation

- Start-up system of evaporator, attached between steam drum and feedwater system
- Bypass system for high pressure (HP) turbine including pressure reduction and attemperation spray section
- Bypass system for low pressure (LP) turbine
- Steam turbine condenser (various designs possible)

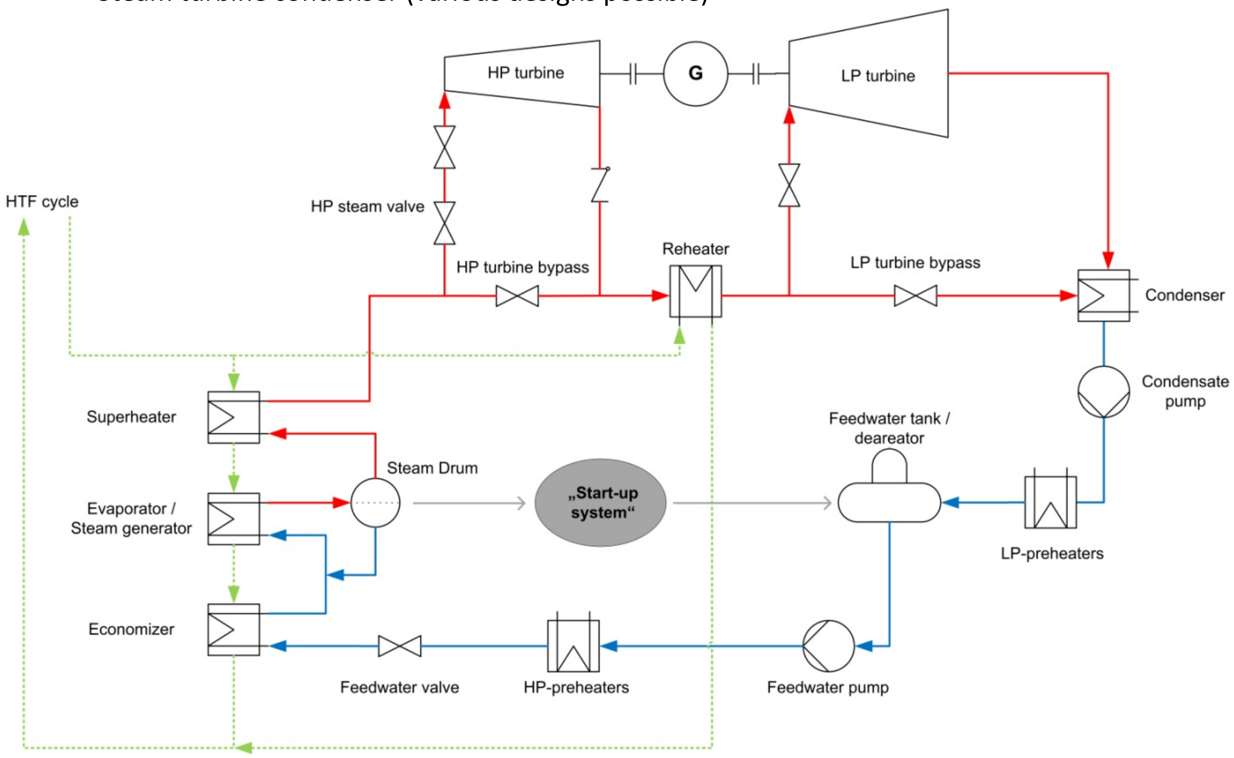

Figure D-19: Simplified power block scheme with exemplary measures used for start up (in green)

Before the warm start-up, the power block conditions can be assumed as follows:

- Economizer, evaporator, superheater and piping of the solar steam generator (SSG) are conserved by the main valves, i.e. feedwater and HP steam valve) maintaining steam pressure and temperature as well as material temperature at high level
- Steam turbine slowly rotated by turbine rotating gear, turbine bearings supplied with seal steam
- Optional steam turbine heating mats keeping turbine housing warm and therefore enabling higher load gradients

A typical warm start procedure passes through the following main steps. The order of described procedures may vary depending on power block design and operational concept.

Table D-2: Exemplary list of procedures for warm start of a power block

No.	procedure	effect

No.	procedure	effect
1	Initiation of steam generators circulation via circulation pump (forced circulation) or auxiliary pump (optional for natural circulation system)	Circulation initiated
2	Incorporation of turbine condenser and cooling tower	Vacuum realization, secure heat removal via cooling tower initiated
3	Incorporation of feedwater pump	Increase of feedwater pressure up to current SSG pressure level
4 / 5	Opening of feedwater valve Opening/initiation of start-up system at steam drum Initiation of HP and LP turbine bypasses	Feedwater flow to SSG Feedwater flow from steam drum to feedwater tank/deaerator, ensuring constant feedwater level in drum Dry staturated steam which is produced in the evaporator/drum flows through superheater, is then reduced and attempered and flows through reheater and enters the steam turbine condenser for condensation Piping and other condensate is evacuated via drainage system
5 / 4	Initiation of HTF circulation through superheater, reheater, evaporator and economizer under consideration of maximum allowable temperature pressure gradients.	Heat input to water-steam cycle, sliding increase of steam parameters (temperature and pressure) as well as increase of material temperature until acceptable parameter levels for steam turbine are reached
6	Opening of HP steam valve, quick-action stop valve, and slowly opening of control valves, simultaneously closing of HP turbine bypass, initiation of HP preheating	Increasing flow and heating up of HP turbine, switch from rotor turning gear operation to turbine operation, condensate is drained through drainage system
7	Opening of LP turbine inlet valves, closing of LP turbine bypass, initiation of LP preheating	Increasing flow and heating up of LP turbine, switch from rotor turning gear operation to turbine operation, drainage system removes condensate, increase of rotational speed

No.	procedure	effect
8	Reaching of reference rotational speed and synchronisation of generator with grid	Rotational speed remains constant
9	Increase of HTF mass flow circulation while keeping temperature and pressure gradients below limits	Start and increase power production

D.1.6.2. Calculation of the heat required to start the power block

Apart from time constraints describing the duration of the start-up the energy losses during the start-up has the most dominant impact on the yield of a STE plant. This section provides an approach to estimate the loss of thermal energy due to the start-up of the power block. The following effects are considered in this approach:

- Heating up of thermal masses (without component insulation)
- Heating up of water content of components
- Heat losses of start-up system at steam drum
- Heat losses over condenser during turbine bypassing

The thermal energy for heating up thermal masses for one single component i can be exemplarily expressed as follows:

$$Q_{\mathrm{heatup,wall,i}} = c_{\mathrm{pipe,i}} * m_{\mathrm{pipe,i}} * \left(T_{\mathrm{pipe,operation,i}} - T_{\mathrm{pipe,\ start,i}}\right) \quad \text{(D.4)}$$

$$Q_{\mathrm{heatup,water,i}} = c_{\mathrm{p,water,i}} * m_{\mathrm{water,i}} * \left(T_{\mathrm{water,operation,i}} - T_{\mathrm{water,\ start,i}}\right) \quad \text{(D.5)}$$

$$Q_{\mathrm{heatup,i}} = \sum_{i=1}^{n} Q_{\mathrm{heatup,water,i}} + Q_{\mathrm{heatup,wall,i}} \quad \text{(D.6)}$$

A compilation of the respective geometries and temperature differences in all relevant components provides the basis for an estimation of the required start-up thermal energy. For the example of a 100 MW power block driven by synthetic oil at 393°C the following tables list typical masses and temperature differences between the beginning of the start-up and the final condition after start-up.

Table D-3: Exemplary list of main components with high influence on transient behavior and "start-up losses" (based on an exemplary 100 MW power block operated with HTF at 390°C)

Aggregat / component	**Estimated dry weight in tons**	**Average material temperature before start-up in °C**	**Average material temperature after start-up in °C**
Economizer	55	190	290
Evaporator	115	250	330
Steam Drum	85	250	310
Superheater	70	240	360
Reheater	50	200	310

Table D-4: Exemplary list of water content of the steam system

Aggregat / component	**Estimated water content in tons**	**Average water temperature before start-up in °C**	**Average water temperature after start-up in °C**
Economizer	50	190	290
Evaporator	50	250	330
Steam Drum	40	250	310

Table D-5: Exemplary assumptions for 100 MW steam turbine mass and temperatures

Turbine part	**Estimated mass influenced by start up in tons**	**Average temperature increase in K**
HP-turbine casing and rotor	100[1]	80
LP-turbine casing and rotor	65	50

Table D-6: Exemplary assumptions for 100 MW steam turbine piping mass and temperatures

	Length in m	**Estimated weight per m in kg/m**
Live steam piping	100	360
Cold reheat piping	100	280
Hot reheat piping	100	280

With the exemplary values given in the tables above and an assumption of c = 500 J/(kg*K) for the steel, the inert start-up losses of the reference system sum up to about 7.6 MWh per warm start-up. Heating up of the water content in economizer, evaporator and steam drum sums up to approx.

[1] Only the weight of the material which undergoes a temperature change is considered in the turbine calculation. It is thus smaller then the total weight of the turbine.

13,3 MWh. These thermal energies are needed to heat up the inventory from the starting to the final temperature. Apart from this “ideal” start-up energy, a number of energy losses have to be taken into account. These are described and estimated in the following.

The start-up system of the evaporator, which is typically installed at the steam drum, prevents water penetration to the dry superheater as well as further downstream systems. After start up of the feedwater pumps, the start-up system is needed to remove feedwater from the drum in order to maintain the drum level below maximum limit. The design of the start-up system can vary, but it is assumed that for STE systems a very efficient, regenerative design approach will be used.

A simplified assumption for the effects on yield calculation can be described as follows:

- During the usage of the start-up system the start-up is performed with 10 % of nominal heat input to the SSG resulting in about 25 MW heat input for heating up.
- 70 % of the heat input is transported to the feedwater in the system, 10 % to the generated steam and the rest is used for heating up of the aggregats and piping.
- The start-up system of the evaporator generates losses of about 30 % related to feedwater heat input.
- The start-up system for the evaporator is used for about 8 minutes.

With these assumptions, the losses for the start-up system of the evaporator can be estimated:

$$Q_{\text{heatup,start-up system}} = 8\text{ min}*25\text{ MW} * 0{,}7 * 0{,}3 = 0{,}7\text{MWh} \tag{D.7}$$

Turbine bypass operation also produces losses since the thermal energy is bypassed and completely wasted for generating electricity. The bypass losses can be estimated with simplifications as follows:

- All produced steam is transported to the condenser.
- Total duration of use of turbine bypass is about 15 minutes
- During the first 8 minutes parallel operation with steam generators start-up system, the rest of the time all of the input heat of the water and steam has to be removed at the condenser

$$\begin{aligned} Q_{\text{heatup,condenser}} &= 8\text{ min} * 25\text{ MW} * 0{,}1 + 7\text{ min} * 25\text{ MW} * (0{,}7+0{,}1) \\ &= 0{,}33\text{ MWh} + 2{,}33\text{ MWh} = 2{,}66\text{ MWh} \end{aligned} \tag{D.8}$$

The total thermal losses for one start up can be summed up as follows:

$$\begin{aligned} Q_{\text{loss,start-up}} &= Q_{\text{heatup,start-up system}} \\ &+ Q_{\text{heatup,condenser}} + Q_{\text{heatup,pipe}} + Q_{\text{heatup,water}} = 24{,}8\text{ MWh} \end{aligned} \tag{D.9}$$

This amount of thermal energy either originating directly from the solar field or from the storage or an auxiliary heater is lost for electricity generation. A yield calculation model should therefore subtract this amount of energy from the available thermal energy.

D.2. Reference system - PTC with synthetic oil as HTF (PT-Oil 100)

In this chapter the modeling of a PB for a SPP based on PTC technology, synthetic oil as HTF and a large indirect two-tank molten salt Thermal Energy Storage (TES) is described. A first sub-chapter deals with design considerations typical for this kind of a power block. As illustrative example the PB for the reference plant configuration with synthetic oil is chosen together with a wet cooling tower as cooling option (see figure below). The following sections describe the modeling approaches for determining the PB gross efficiency, auxiliary electrical consumption and the HTF temperature at the outlet of the SSG.

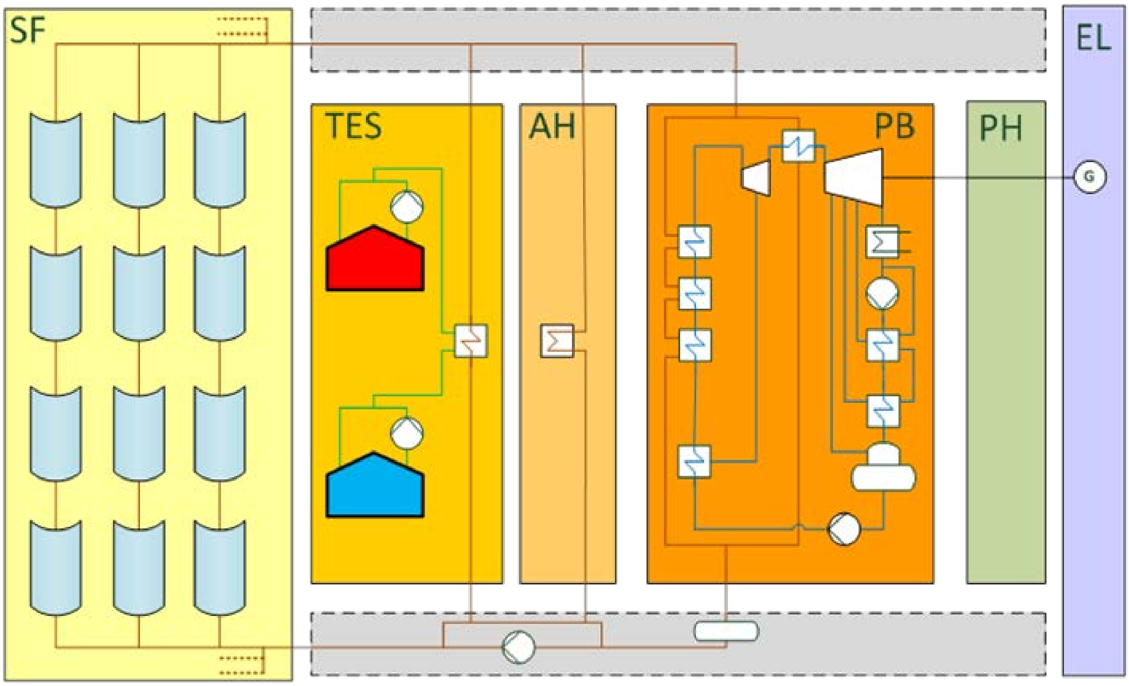

Figure D-20: Process flow diagram of a parabolic through power plant with sub-systems

D.2.1. Design considerations for the reference plant

The complex thermodynamics of the PB usually require performing detailed heat and mass balance calculations for design purposes. For the annual yield analysis it is sufficient to use the results of such detailed simulation in the form of gross efficiency correlation, HTF outlet temperature correlation and auxiliary electrical consumption. However, most of the required data are not always available. In such a case, it is necessary to recalculate the whole power cycle. The following sections describe the setup of typical design parameters required for heat cycle calculation of a Rankine cycle power block.

D.2.1.1. Typical power block configuration

The main components for modeling the PB are the following:

- Solar Steam Generators (SSGs)
- Steam Turbine (ST)
- Condenser
- Feedwater preheating

The modeling aspects/parameters of these main systems are as follows[2]:

Solar Steam Generator (SSG)

The SSG consists basically of two (2) trains (2 x 50%) and each train consists of the following four (4) main components:

- Economizer
- Evaporator
- Superheater and
- Reheater

The feedwater leaving the last High Pressure (HP) preheater is heated up in the economizers close to the saturation temperature, see Figure D-21. Subsequently, the preheated feedwater passes through the evaporator and leaves as saturated steam. This steam is superheated in the superheater and proceeds to the HP-Turbine. The exhaust steam from the HP turbine is returned to the SSG and superheated in the reheater. The reheater is arranged in parallel to the economizer, evaporator and superheater. Furthermore, a bypass of the SSG oil-side is considered in order to control the HTF's temperature returning to the SF.

The feedwater pump is one of the main auxiliary electrical consumers of the PB. For PBs in the range of 100 MW a motor driven pump is state-of-the-art. ST driven feedwater pumps are often applied in conventional power plants with higher capacity.

Steam Turbine (ST)

Typically the ST of a SPP with PTC is a condensing ST consisting of a an HP and a LP turbine. The ST is typically operated in sliding pressure mode, which means that live steam pressure decreases in part load. The lowest minimum live steam pressure / turbine inlet pressure of the HP-part depends on

[2] Please note that presented specific values are valid only fort he selected reference system and therefore may differ depending on the project and underlying plant concept.

suppliers technology. A typical value for this is 30 bar. For even lower part loads inlet pressure is kept constant and the turbine is operated in fixed pressure mode.

Condenser

A low condensation temperature is important for reaching a high efficiency of the Rankine Cycle. The technicalities of different cooling options have already been introduced in Appendix I of the Guideline. Typically, for SPPs wet cooling tower or ACCs are applied due to scarcity of water at potential sites, although a wet cooling tower consumes a relative high amount of water, too.
For the subject reference plant, a wet cooling tower has been selected (refer also to Appendix B).

Feedwater preheating

The efficiency of a Rankine Cycle can be improved by preheating the feedwater with steam from turbine extractions. The number of feedwater preheaters influences the efficiency but also the capital costs of a power plant. Basically, more than nine (9) feedwater preheaters do not further improve the efficiency significantly but increase the capital costs. The number of preheaters depends on the size of the power plant, e.g. 800 - 1000 MW supercritical coal fired power plants have up to 9 preheaters. For the investigated reference plant in total six (6) preheaters represent the state-of-the-art. This is a rather high number for a plant with a comparatively smaller capacity, however justifiable when considering the invest costs for the SF. In order to convert this “expensive” solar heat into electricity most efficiently, the PB is equipped with a variety of measures which maximize its efficiency.

The subject feedwater preheating system consists of three (3) LP-preheaters, one (1) deaerator and two (2) HP-preheaters. Depending on the load case, the feedwater temperature at the inlet of the economizer of the SSG is around 230 °C.

D.2.1.2. Heat cycle design parameters

Although detailed information for heat cycle calculations are often not available some major parameters are usually provided. These allow to roughly parameterize a heat cycle calculation.

The main heat cycle design parameters of the reference plant can be summarized as follows:

- HTF temperature - for SF only operation: 393 °C
- HTF temperature - for pure TES discharge operation: 379 °C
- HTF return temperature from SSG to SF with bypass control: minimum 250 °C
- Gross plant output: 100 MW at 18 °C WBT
- Live steam parameters: 103 bar/383 °C (SSG outlet), reheat 21.5 bar/383 °C
- Sliding pressure operation down to 30 bar; from then on fixed pressure operation

On the basis of the design assumptions as given above, heat and mass balances have been calculated for different

- WBTs
- load cases and
- HTF inlet temperatures

The corresponding simulation results are presented and assessed in the next sections.

Apart from the main design parameters a number of detailed design parameters are needed to carry out a heat cycle calculation. The input values for the calculation of the heat and mass balance diagrams of the reference plant are as follows[3]:

- Steam Turbine
  - Mechanical efficiency: 99.5 %
  - Thermal efficiency: 90 % HP / 88 % (IP/LP) (range 86 -91 %)
  - Pressure losses in the extraction lines: 2 % (referring to pressure)
- Generator
  - Total efficiency: 98.5 % (range 98 -99 %)
- Cooling tower
  - Approach: 10 K (range depending on WBT 5 - 15 K)
- Condenser
  - Cooling range: 10 K (range 6 - 15 K)
  - Terminal Temperature Difference (TTD): 3 K (range 2 - 4 K)
- LP-preheater
  - TTD preheater: 4 K (range 3 - 6 K)
  - TTD drain cooler : 7 K (range 5 - 8 K)
- HP-preheater
  - TTD preheater: 2 K (range -1 - 4 K)
  - TTD drain cooler: 7 K (range 5 - 8 K)
- Solar Steam Generator:
  - Pinch Point: 5 K (range 4 - 8 K)
  - Pressure drop: 3 bar
- Live steam pipeline
  - Pressure drop: 3 bar

The figure below shows a screenshot of one of the performed heat and mass balance calculations.

[3] Note that the values given in brackets indicate typical ranges.

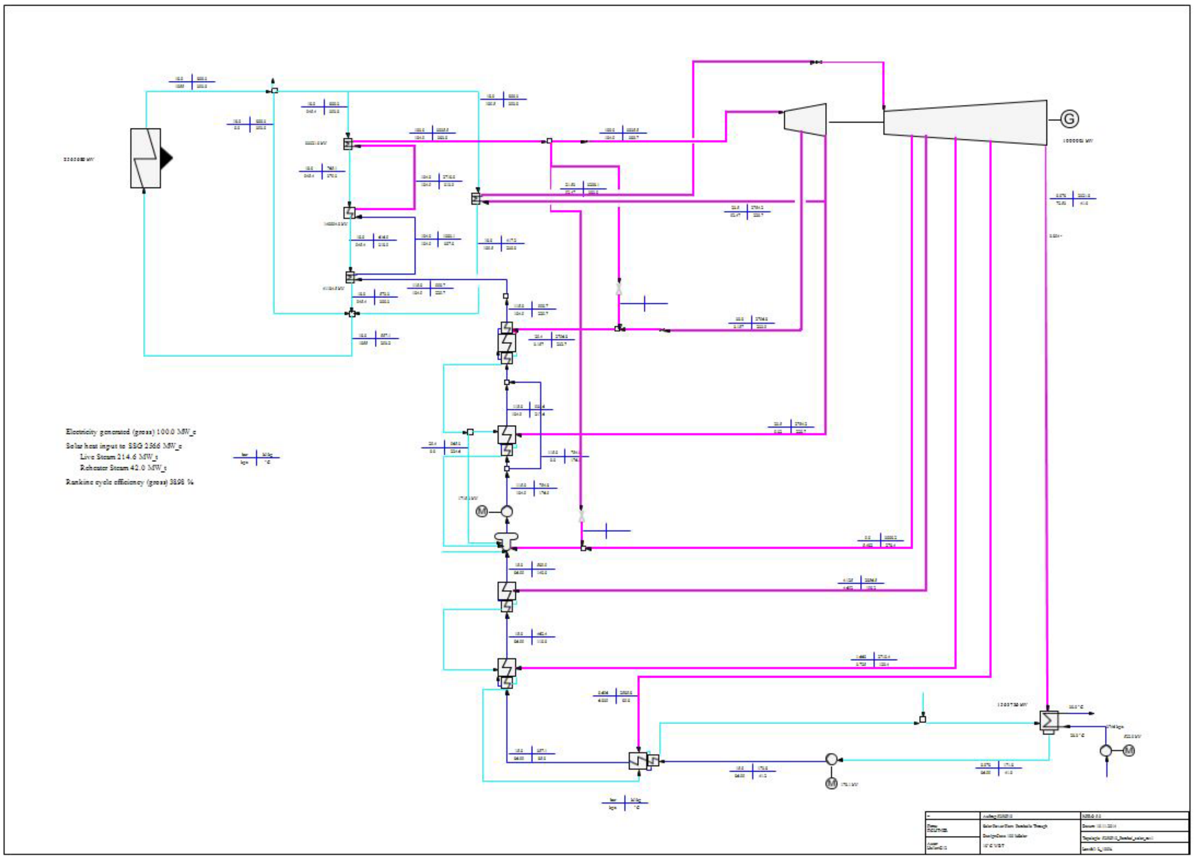

Figure D-21: Heat flow diagram

Heat and Mass Balance Diagrams (HMBDs) for the design case as well as some part load conditions in higher resolution can be found in the guideline attachments.

D.2.2. Power block gross efficiency

For annual yield prognosis simulations it is necessary to calculate the gross output of the plant for each time-step. The gross output is determined based on the solar heat input and the gross efficiency of the plant. The solar heat input is calculated based on the HTF mass flow and the inlet and outlet enthalpy at the SSG. The 100 % load case corresponds to the solar heat required for generating 100 MW gross output at design conditions. Part load cases are determined taking the 100 % load as reference. Changing the HTF inlet temperature and maintaining the design mass flow leads to a different transferred heat. As a consequence, this amount of heat is then taken as 100 % load for the load cases with different HTF temperatures.

As introduced in previous chapters, the PB gross efficiency depends on the the cooling conditions, the HTF inlet temperature at the SSG and the load which in turn corresponds to the HTF mass flow. There are two possibilities to specify the PB gross efficiency figures, either in form of matrices or as a set of approximation equations.

In case of matrices, it is important to use a step size for the input parameters which should take into account non-linear effects. Otherwise the accuracy of the results decreases. Operating the power plant with high solar input the curving of the efficiency function versus load is smaller compared to operation modes with low load factors. Therefore deviations are mostly visible in lower load cases. But the share of this load cases is small compared to the yearly operation hours and therefore the influence on the annual yield prognosis is negligible.

In case of the PTC reference case with synthetic oil the step size of the load is set to 10 %. Due to the use of a wet cooling tower, the wet bulb temperature is the main parameter for the cooling conditions. The figures in the matrices are shown with a step size of 2 K. The HTF inlet temperature influences the efficiency, too. To show this interdependency, matrices for three different HTF inlet temperatures have been elaborated.

Another possibility to describe the dependencies of the efficiency on the previously introduced parameters is the use of approximation equations which are determined on the basis of the figures in the matrices.

It is to be noted that the procedure which is chosen for the annual yield prognosis simulation depends on the simulation software and the possibility to introduce those equations.

D.2.2.1. Gross efficiency matrices

Based on the aforementioned design data and a wet bulb temperature of 18 °C, the solar heat input which is necessary to generate 100 MW gross output was calculated. The required solar heat input is approx. 256.6 MW and the corresponding HTF mass flow amounts to 1055 kg/s. Based on this design, part load calculations by means of solar heat input variation have been performed.

Table D-7 below shows the gross efficiency for different load cases at operation with design HTF temperature (@393°C). This matrix allows simulating the gross output of a CSP power plant based on the solar heat input obtained from the irradiation data. The values are calculated based on 10 %-steps of the thermal load (transferred heat) and 2 K-steps for the WBT. This resolution ensures a high accuracy since it is necessary to interpolate between calculated values and as the dependency of the gross efficiency and the HTF outlet temperature on the WBT and the load is non-linear.

Table D-7: Gross efficiency vs. WBT and load cases (HTF inlet temperature of 393 °C)

HTF Temperature @ SSG Inlet [°C]	393								
Wet Bulb Temperature [°C]	4	6	8	10	12	14	16	18	21
Heat transferred @ SSG (100% design)									
20%	35.02	34.67	34.31	33.92	33.51	33.08	32.63	32.16	
30%	36.92	36.69	36.43	36.14	35.83	35.48	35.11	34.71	
40%	37.73	37.59	37.41	37.21	36.96	36.69	36.38	36.03	
50%	38.24	38.14	38.01	37.84	37.64	37.41	37.14	36.84	
60%	38.75	38.69	38.60	38.47	38.31	38.12	37.90	37.64	
70%	39.06	39.00	38.92	38.81	38.67	38.50	38.30	38.07	
80%	39.37	39.32	39.24	39.14	39.02	38.87	38.70	38.50	
90%	39.57	39.53	39.46	39.36	39.24	39.10	38.93	38.73	
100%	39.78	39.74	39.67	39.58	39.46	39.32	39.16	38.97	38.80

The HTF temperature at the SSG outlet decreases towards lower load levels because of the lower pressure at the evaporator in sliding pressure mode and the lower feedwater inlet temperature decreasing with load, too. A SSG bypass enables to control the return temperature to avoid temperatures lower than 250 °C at the inlet to SF and the TES (see table below).

Table D-8: HTF outlet temperature vs. WBT and load cases [°C] (HTF inlet temperature of 393 °C)

HTF Temperature @ SSG Inlet [°C]	393								
Wet Bulb Temperature [°C]	4	6	8	10	12	14	16	18	21
Heat transferred @ SSG (100% design)									
20%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
30%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
40%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
50%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
60%	255.40	255.40	255.40	255.40	255.40	255.40	255.40	255.40	
70%	265.76	265.76	265.76	265.76	265.76	265.76	265.76	265.76	
80%	275.50	275.50	275.50	275.50	275.50	275.50	275.50	275.50	
90%	284.61	284.61	284.61	284.61	284.61	284.61	284.61	284.61	
100%	293.10	293.10	293.10	293.10	293.10	293.10	293.10	293.10	293.10

For Table D-9, the HTF mass flow was derived from the solar heat input and the HTF inlet and outlet temperature at the SSG.

Table D-9: HTF mass flow in kg/s vs. WBT and load cases (HTF inlet temperature of 393°C)

HTF Temperature @ SSG Inlet [°C]	**393**								
Wet Bulb Temperature [°C]	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**21**
Heat transferred @ SSG (100% design)									
20%	151.00	151.00	151.00	151.00	151.00	151.00	151.00	151.00	
30%	226.00	226.00	226.00	226.00	226.00	226.00	226.00	226.00	
40%	302.00	302.00	302.00	302.00	302.00	302.00	302.00	302.00	
50%	383.91	383.91	383.91	383.91	383.91	383.91	383.91	383.91	
60%	469.00	469.00	469.00	469.00	469.00	469.00	469.00	469.00	
70%	587.00	587.00	587.00	587.00	587.00	587.00	587.00	587.00	
80%	724.00	724.00	724.00	724.00	724.00	724.00	724.00	724.00	
90%	880.00	880.00	880.00	880.00	880.00	880.00	880.00	880.00	
100%	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00

In case the power block is fed with 50 % heat input from the SF and 50 % from the TES, the HTF temperature at the SSG inlet will have a mixing temperature of 386°C. This is due to the design SF outlet temperature of 393°C and the TES outlet temperature (HTF) of 379°C. This mixing temperature is chosen to add a further node as well as for interpolation by means of matrices and to determine approximation equations. Of course load cases with other temperatures are possible as well. In this case 100 % transferred heat means the same HTF mass flow as in the design case. The HTF outlet temperature at the SSG decreases with decreasing inlet temperature but the difference between these temperatures becomes smaller (see Table D-11). Due to the decreased ΔT, the heat input is reduced. All load points for this operation mode have been calculated based on the reduced heat input.

Table D-10: Gross efficiency in [%] vs. WBT and load cases (HTF inlet temperature of 386°C)

HTF Temperature @ SSG Inlet [°C]	**386**								
Wet Bulb Temperature [°C]	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**21**
Heat transferred @ SSG (100% design)									
20%	34.72	34.37	33.99	33.60	33.19	32.76	32.30	31.84	
30%	36.54	36.30	36.03	35.74	35.41	35.05	34.66	34.24	
40%	37.48	37.32	37.13	36.91	36.66	36.37	36.04	35.68	
50%	38.00	37.89	37.74	37.57	37.35	37.11	36.83	36.52	
60%	38.52	38.45	38.35	38.22	38.05	37.85	37.61	37.35	
70%	38.84	38.78	38.69	38.58	38.45	38.30	38.13	37.93	
80%	39.17	39.11	39.03	38.95	38.85	38.75	38.64	38.52	
90%	39.38	39.32	39.25	39.16	39.06	38.94	38.81	38.66	
100%	39.59	39.54	39.47	39.38	39.27	39.13	38.97	38.80	

Table D-11: HTF outlet temperature [°C] vs. WBT and load cases (HTF inlet temperature of 386°C)

HTF Temperature @ SSG Inlet [°C]	**386**								
Wet Bulb Temperature [°C]	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**21**
Heat transferred @ SSG (100% design)									
20%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
30%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
40%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
50%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
60%	252.70	252.70	252.70	252.70	252.70	252.70	252.70	252.70	
70%	263.05	263.05	263.05	263.05	263.05	263.05	263.05	263.05	
80%	272.80	272.80	272.80	272.80	272.80	272.80	272.80	272.80	
90%	281.95	281.95	281.95	281.95	281.95	281.95	281.95	281.95	
100%	290.50	290.50	290.50	290.50	290.50	290.50	290.50	290.50	290.5

Table D-12: HTF mass flow [kg/s] vs. WBT and load cases (HTF inlet temperature of 386°C)

HTF Temperature @ SSG Inlet [°C]	**386**								
Wet Bulb Temperature [°C]	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**21**
Heat transferred @ SSG (100% design)									
20%	152.00	152.00	152.00	152.00	152.00	152.00	152.00	152.00	
30%	228.00	228.00	228.00	228.00	228.00	228.00	228.00	228.00	
40%	303.00	303.00	303.00	303.00	303.00	303.00	303.00	303.00	
50%	382.72	382.72	382.72	382.72	382.72	382.72	382.72	382.72	
60%	464.00	464.00	464.00	464.00	464.00	464.00	464.00	464.00	
70%	581.38	581.38	581.38	581.38	581.38	581.38	581.38	581.38	
80%	719.00	719.00	719.00	719.00	719.00	719.00	719.00	719.00	
90%	876.88	876.88	876.88	876.88	876.88	876.88	876.88	876.88	
100%	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00

Table D-13 and Table D-15 show the results based on a HTF inlet temperature of 379 °C. This represents the HTF temperature for a TES discharge operation mode, when the heat for the SSG is provided from the TES only. HTF inlet temperatures lower than 379°C are also applicable down to360 °C, where the water content in the exhaust line of the ST is lower than 14 %. As a consequence, the efficiency will decrease even more than in the presented cases.

Table D-13: Gross efficiency [%] vs. WBT and load cases (HTF inlet temperature of 379°C)

HTF Temperature @ SSG Inlet [°C]	379								
Wet Bulb Temperature [°C]	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**21**
Heat transferred @ SSG (100% design)									
20%	34.29	33.92	33.54	33.13	32.71	32.27	31.82	31.35	
30%	36.20	35.93	35.64	35.32	34.98	34.62	34.23	33.82	
40%	37.21	37.04	36.84	36.61	36.34	36.04	35.71	35.35	
50%	37.74	37.62	37.46	37.27	37.05	36.80	36.52	36.20	
60%	38.28	38.20	38.09	37.94	37.77	37.56	37.32	37.04	
70%	38.61	38.54	38.45	38.33	38.17	37.99	37.77	37.53	
80%	38.94	38.89	38.81	38.71	38.58	38.42	38.23	38.01	
90%	39.16	39.11	39.04	38.94	38.81	38.67	38.49	38.29	
100%	39.39	39.34	39.26	39.17	39.05	38.92	38.76	38.57	

Table D-14: HTF outlet temperature [°C] vs. WBT and load cases (HTF inlet temperature of 379°C)

HTF Temperature @ SSG Inlet [°C]	379								
Wet Bulb Temperature [°C]	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**21**
Heat transferred @ SSG (100% design)									
20%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
30%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
40%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
50%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
60%	250.00	250.00	250.00	250.00	250.00	250.00	250.00	250.00	
70%	261.53	261.53	261.53	261.53	261.53	261.53	261.53	261.53	
80%	270.30	270.30	270.30	270.30	270.30	270.30	270.30	270.30	
90%	279.03	279.03	279.03	279.03	279.03	279.03	279.03	279.03	
100%	287.70	287.70	287.70	287.70	287.70	287.70	287.70	287.70	287.70

Table D-15: HTF mass flow [kg/s] vs. WBT and load cases (HTF inlet temperature of 379°C)

HTF Temperature @ SSG Inlet [°C]	**379**								
Wet Bulb Temperature [°C]	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**21**
Heat transferred @ SSG (100% design)									
20%	152.60	152.60	152.60	152.60	152.60	152.60	152.60	152.60	
30%	228.90	228.90	228.90	228.90	228.90	228.90	228.90	228.90	
40%	305.20	305.20	305.20	305.20	305.20	305.20	305.20	305.20	
50%	381.39	381.39	381.39	381.39	381.39	381.39	381.39	381.39	
60%	457.80	457.80	457.80	457.80	457.80	457.80	457.80	457.80	
70%	576.73	576.73	576.73	576.73	576.73	576.73	576.73	576.73	
80%	715.90	715.90	715.90	715.90	715.90	715.90	715.90	715.90	
90%	875.33	875.33	875.33	875.33	875.33	875.33	875.33	875.33	
100%	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00	1055.00

Note that for all matrices presented in this section, the operation at 21°C WBT was considered as extreme operation case and hence was only analyzed for the 100% load case.

D.2.2.2. Modeling based on approximation equations

Another possibility to describe the dependencies of the efficiency on the previously introduced parameters in performance models could be using approximation equations instead of matrices. The derivation of approximation functions requires a data base like the one available for the efficiency matrices. The examples given in this section are based on the efficiency values obtained from the matrices in chapter D.2.2.1. In this section, the different approximations will be presented for the determination of the gross efficiency in dependency of the WBT, load and HTF temperature.

$$\eta_{gross}^{PB} = f_1\left(\dot{Q}^{PB}, T_{in}^{PB}, T^*\right) = f_2(Load, \vartheta_{HTF}, \vartheta_{WBT}) \tag{D.10}$$

$$Load = \dot{Q}^{PB}/\dot{Q}_0^{PB} \tag{D.11}$$

$$\vartheta_{HTF,in} = T_{in}^{PB}\ [°C] \tag{D.12}$$

$$\vartheta_{WBT} = T^*\ [°C] \tag{D.13}$$

The power block gross efficiency changes with the load. Figure D-22 shows the calculated gross efficiency based on a HTF temperature of 393 °C for three different wet bulb temperatures in dependency with the load. The values are scaled so that the factor for 18 °C WBT (design) and 100 % load corresponds to 1. Additionally, best fit curves are indicated for the subject case.

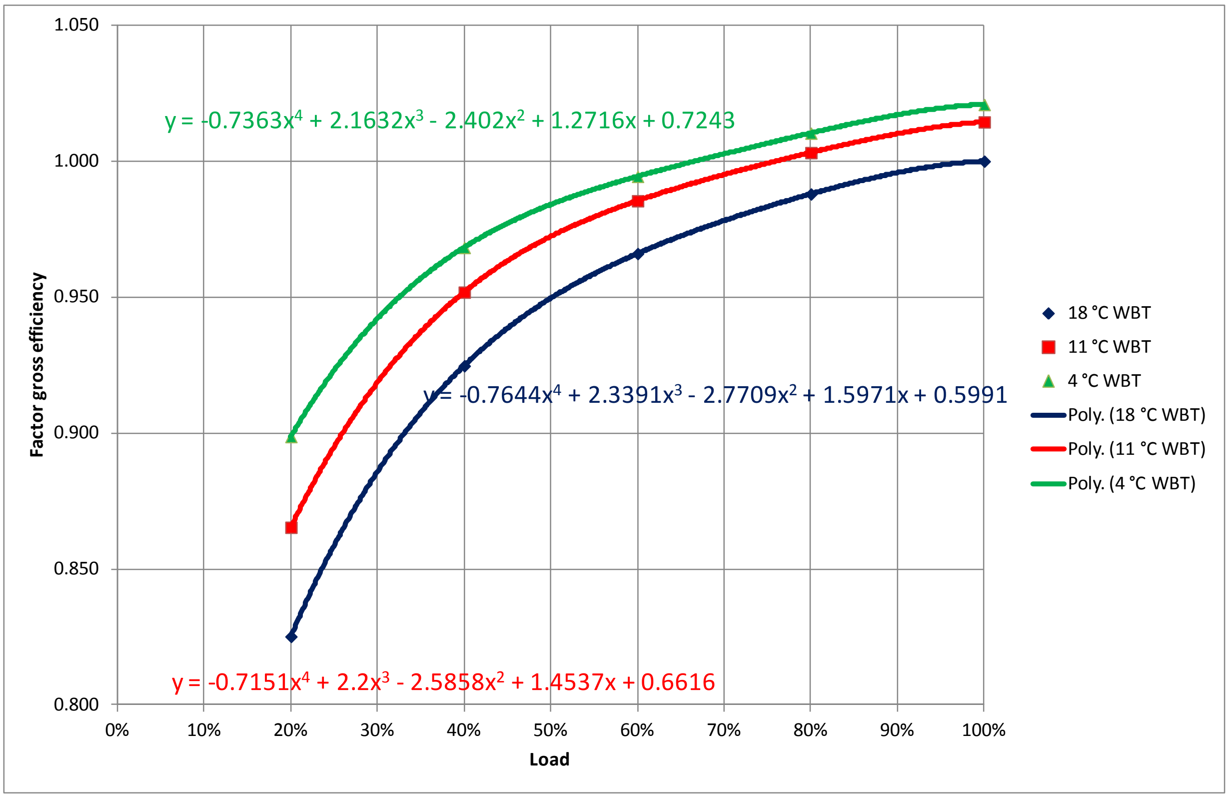

Figure D-22: Factor gross efficiency versus thermal load

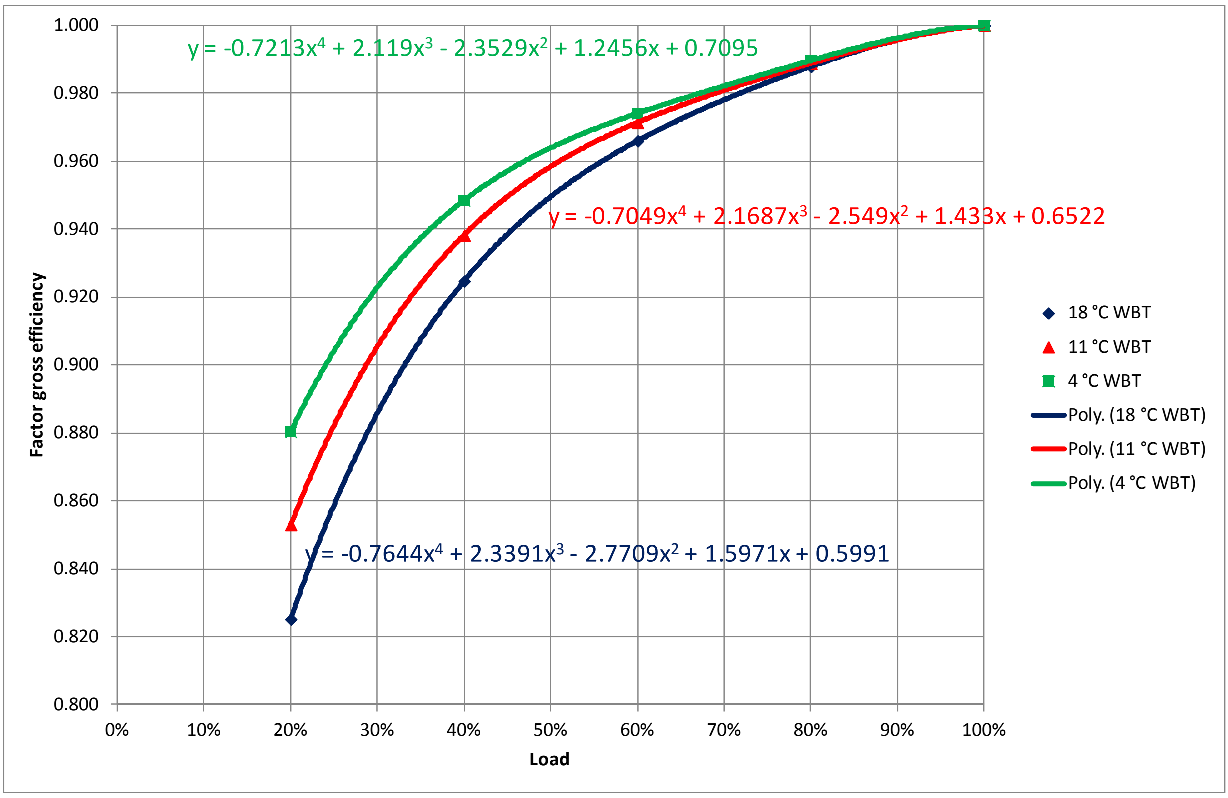

Figure D-23: Factor gross efficiency versus thermal load (different scaling)

Figure D-23 shows the same data as Figure D-22 but with a different scaling factor. Each curve is normalized with its efficiency at 100 % load. So each curves start with the value 1 at 100 % load. In this diagram it can be seen that the curves for different WBTs are fanned out with decreasing load. This relies on the change in cold water temperature (condenser inlet temperature) which is higher in case of a lower WBT. Because of the higher approach in case of lower WBTs, halving the approach creates a stronger decrease in the cold water than in case of higher WBTs.

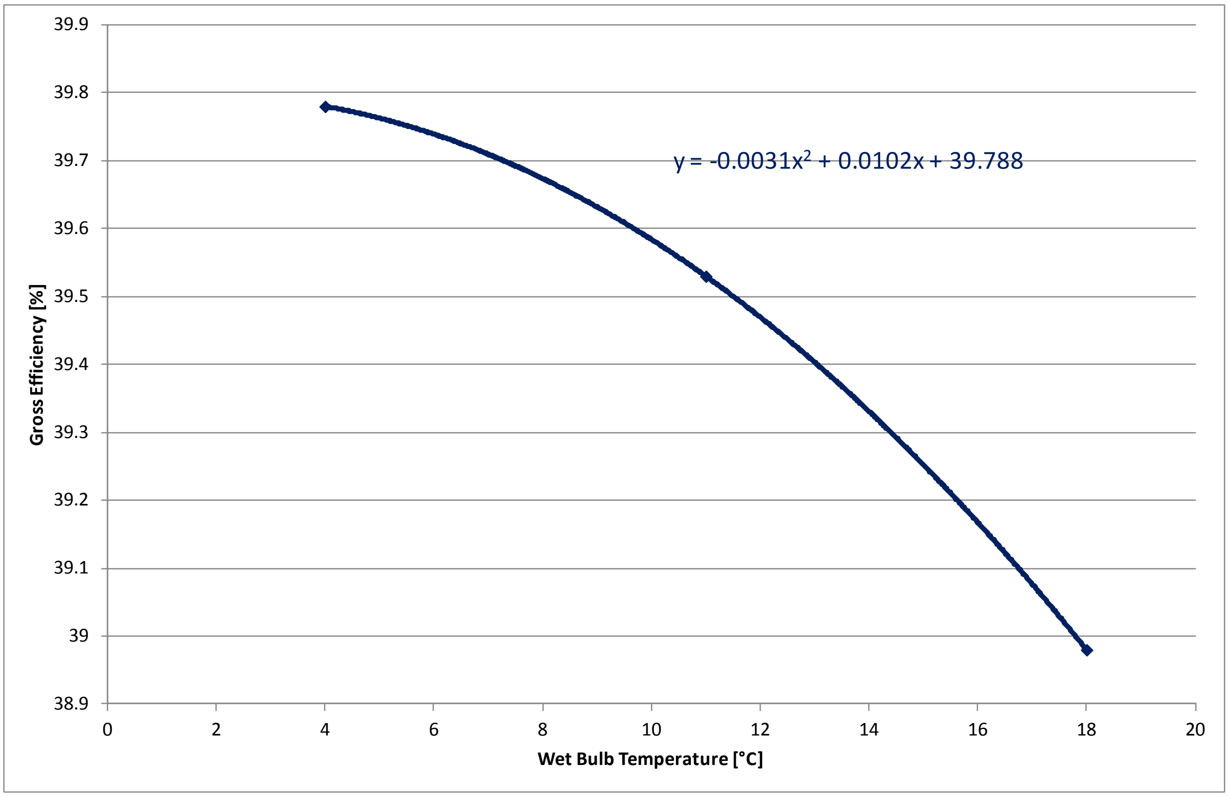

Figure D-24: Gross efficiency versus wet bulb temperature (100%-load)

Figure D-24 shows the course of gross efficiency of the power plant over the WBT at 100 % load. This curve can be described by means of a polynomial function. Using this equation to calculate the efficiency at 100 % load and taking the normalized dependency of the load at 11 °C WBT (see Figure D-23) an equation can be obtained to approximate the behavior of the gross efficiency versus load for different WBTs (see (D.14) below).

$$\eta_{gross,393}^{PB} = f(\vartheta_{WBT}, Load) \tag{D.14}$$

with

$$\eta_{gross,393,100\%}^{PB} = (-0.0031 * \vartheta_{WBT}{}^{2} + \ 0{,}0102 * \vartheta_{WBT} + 39{,}788)/100$$

and

$$\eta_{gross,393}^{PB} = \eta_{gross,393,100\%}^{PB} * (-0.7049 * Load^{4} + 2.1687 * Load^{3} \ - 2.549 \\ * Load^{2} \ + 1.433 * Load \ + 0.6522 \,)$$

The comparison of the results from the cycle process calculations and the results of the approximation equations are shown in Figure D-25. The dashed lines are calculated based on the approximation equation (D.14) and differ from the results from the heat cycle equations especially at lower load cases.

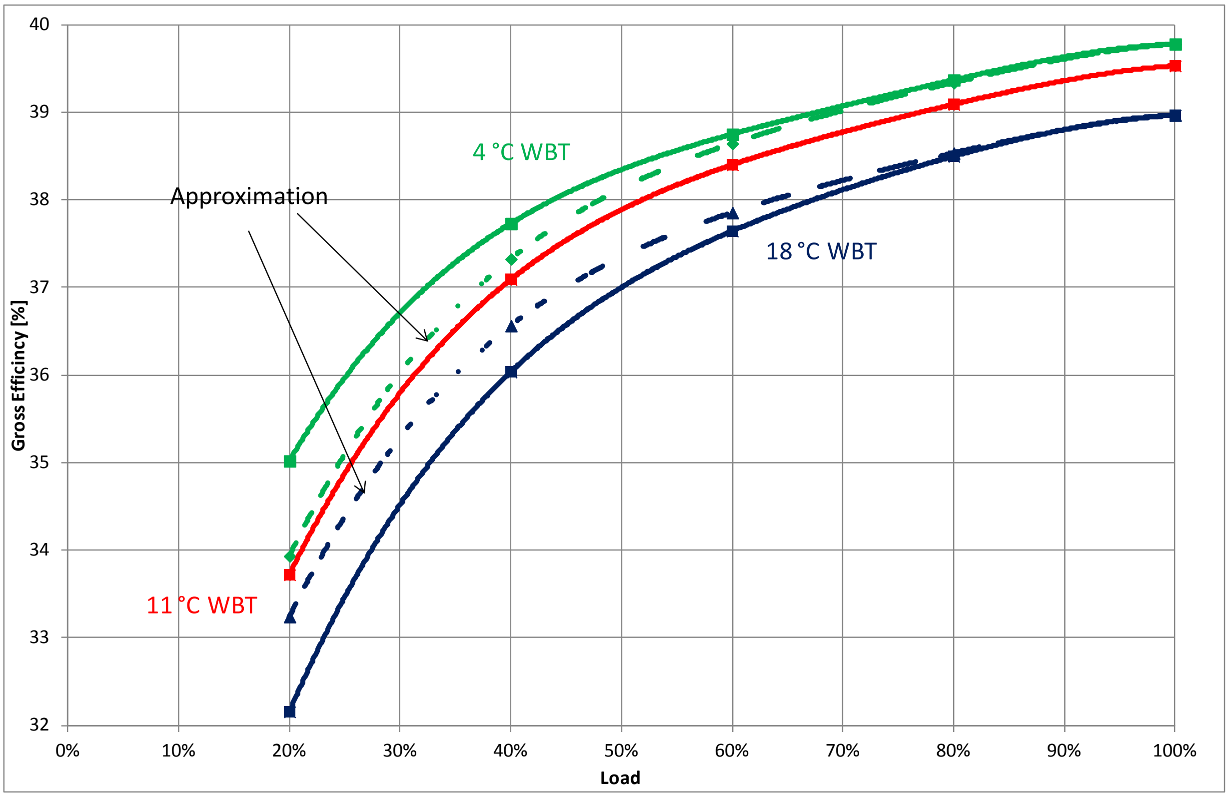

Figure D-25: Approximation of gross efficiency

The difference between efficiency calculated based on approximation and efficiency from the cycle process calculation divided by the efficiency of the cycle process calculation determines the accuracy of the approximation. This accuracy at 4 °C WBT and 18 °C WBT is shown in Figure D-26.

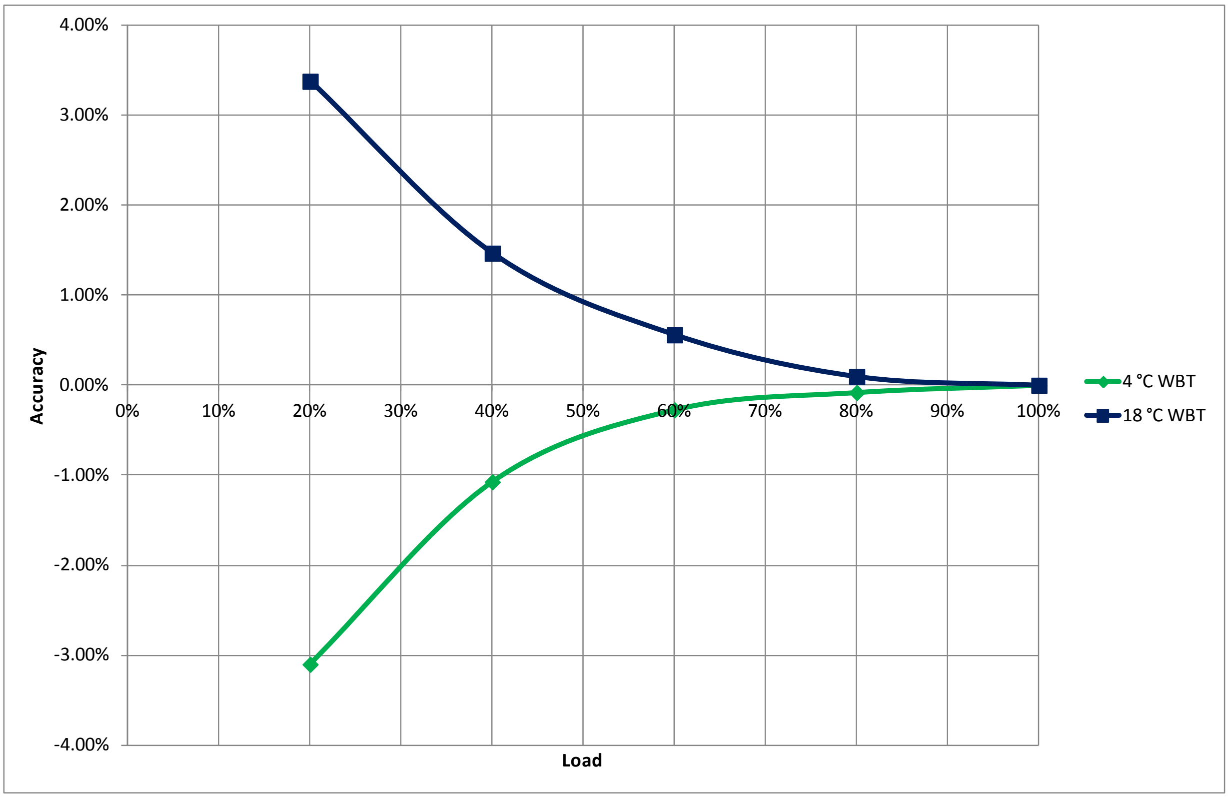

Figure D-26: Accuracy of approximation

Decreasing the load leads to higher discrepancy between the results obtained with a cycle process calculation and the values calculated with the help of the approximation equations. Due to the fact that the approximation curve is based on the gross efficiencies at a WBT of 11 °C, the deviations and thereby the accuracies at 4 °C WBT and 18 °C WBT have a different algebraic sign.

The HTF-temperature determines the live steam temperature and therefore also the gross efficiency of the plant. The HTF inlet temperature of the SSG depends on the operation mode of the plant. HTF directly coming from the SF has a higher temperature than from the TES resulting in higher live steam temperatures and higher gross efficiency.

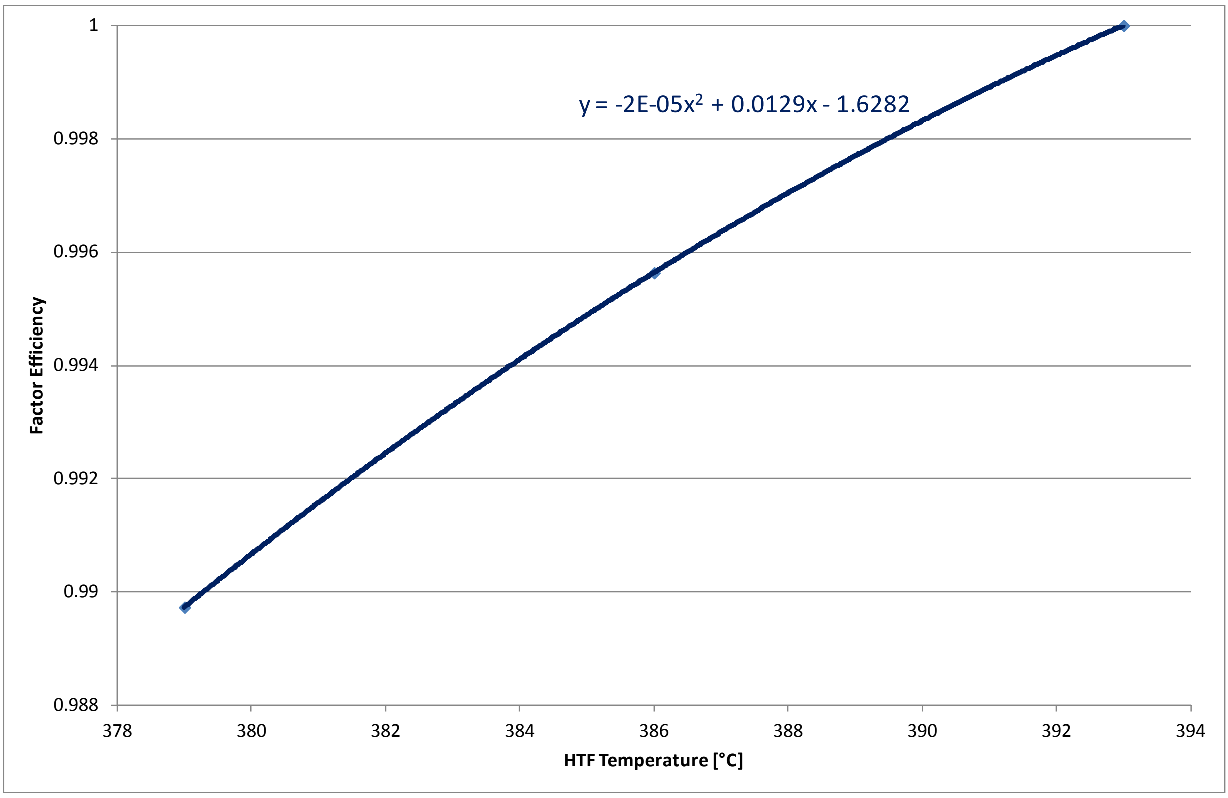

Figure D-27: Factor gross efficiency versus HTF temperature at 18 °C WBT

Based on the fit shown in Figure D-27 the PB gross efficiency for lower HTF temperatures can be obtained either using the values from Table D-7 or the approximation equations multiplied with this factor.

The gross efficiency of the PB is a function of WBT, load and HTF inlet temperature at the SSG:

$$\eta_{gross}^{PB} = f\left(Load, \vartheta_{HTF,in}, \vartheta_{WBT}\right) \quad \text{(D.15)}$$

Based on the equations shown in this Chapter which take into account the dependency with WBT and load, the efficiency approximation including the HTF inlet temperature can be determined with following equation:

$$\eta_{gross}^{PB} = \eta_{gross,393}^{PB} * (-0.000015807 * \vartheta_{HTF,in}^{2} + 0.0129 * \vartheta_{HTF,in} - 1.6282) \quad \text{(D.16)}$$

The accuracy of this approximation equation depends mainly on the load and WBT (Figure D-26) and presents the gross efficiency of a SPP based on PTC technology, synthetic oil as HTF, an indirect two-tank molten salt TES and a wet cooling tower. Besides the design data as given in Section D.2.1.2, the mode of operation (sliding pressure/ fixed pressure) is a further important factor for the efficiency. Hence, the equations shown above are only applicable for this special case. Amongst other factors, different turbine efficiencies change the coefficients in the presented equations, whereas different modes of operation or different condensers (e.g. Air Cooled Condenser (ACC)) will influence the kind of function. If approximation equations will be used to calculate the electrical output of a SPP, it is necessary to determine these equations based on particularly performed heat cycle calculations.

Table D-16, Table D-17 and Table D-18 show the deviations of the PB gross efficiency calculated based on the approximation equations and directly determined with cycle process calculations. It can be seen that from a load range from 100 - 60 % the accuracy is high, while the deviation increases rapidly for part load levels below 60 %. At low WBTs the efficiency obtained from the approximation formula is lower and at high WBTs the efficiency is higher compared to the values determined by a heat cycle calculation.
These effects are caused by different reasons:

- Controlled HTF outlet temperature up to a load between 50 and 60 % depending on the HTF inlet temperature
- Fixed pressure operation for part loads
- Changing cooling water inlet temperatures not only depending on WBT but also on load

Table D-16: Accuracy of gross efficiency calculated based on approximation equations (393 °C HTF inlet temperature)

HTF Temperature @ SSG Inlet [°C]	**393**								
Wet Bulb Temperature [°C]	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**21**
Heat transferred @ SSG (100% design)									
20%	-3.19%	-2.27%	-1.36%	-0.44%	0.48%	1.41%	2.34%	3.28%	
30%	-2.48%	-1.94%	-1.39%	-0.82%	-0.24%	0.37%	0.99%	1.64%	
40%	-1.07%	-0.80%	-0.50%	-0.17%	0.19%	0.58%	1.00%	1.46%	
50%	-0.28%	-0.12%	0.06%	0.27%	0.51%	0.77%	1.06%	1.38%	
60%	-0.26%	-0.21%	-0.14%	-0.05%	0.07%	0.21%	0.38%	0.57%	
70%	-0.09%	-0.05%	0.00%	0.05%	0.13%	0.21%	0.30%	0.41%	
80%	-0.07%	-0.05%	-0.02%	0.00%	0.03%	0.05%	0.08%	0.11%	
90%	0.10%	0.11%	0.12%	0.13%	0.15%	0.16%	0.17%	0.19%	
100%	0.01%	0.01%	0.01%	0.01%	0.01%	0.01%	0.01%	0.01%	

Table D-17: Accuracy of gross efficiency calculated based on approximation equations (386 °C HTF inlet temperature)

HTF Temperature @ SSG Inlet [°C]	386								
Wet Bulb Temperature [°C]	4	6	8	10	12	14	16	18	21
Heat transferred @ SSG (100% design)									
20%	-2.71%	-1.77%	-0.83%	0.10%	1.04%	1.98%	2.92%	3.86%	
30%	-1.83%	-1.28%	-0.70%	-0.10%	0.53%	1.19%	1.87%	2.58%	
40%	-0.80%	-0.49%	-0.15%	0.22%	0.63%	1.06%	1.53%	2.03%	
50%	-0.05%	0.14%	0.35%	0.60%	0.87%	1.17%	1.50%	1.86%	
60%	-0.07%	0.00%	0.10%	0.22%	0.36%	0.54%	0.74%	0.96%	
70%	0.06%	0.13%	0.19%	0.24%	0.28%	0.32%	0.35%	0.38%	
80%	0.04%	0.10%	0.12%	0.11%	0.05%	-0.04%	-0.17%	-0.34%	
90%	0.19%	0.23%	0.25%	0.24%	0.21%	0.16%	0.08%	-0.02%	
100%	0.09%	0.11%	0.12%	0.12%	0.12%	0.10%	0.08%	0.05%	

Table D-18: Accuracy of gross efficiency calculated based on approximation equations shown above (379 °C HTF inlet temperature)

HTF Temperature @ SSG Inlet [°C]	379								
Wet Bulb Temperature [°C]	4	6	8	10	12	14	16	18	21
Heat transferred @ SSG (100% design)									
20%	-2.02%	-1.03%	-0.05%	0.92%	1.89%	2.86%	3.83%	4.80%	
30%	-1.46%	-0.81%	-0.16%	0.50%	1.17%	1.85%	2.54%	3.25%	
40%	-0.65%	-0.29%	0.08%	0.49%	0.92%	1.38%	1.86%	2.38%	
50%	0.05%	0.28%	0.53%	0.80%	1.10%	1.42%	1.77%	2.15%	
60%	-0.01%	0.10%	0.22%	0.37%	0.55%	0.74%	0.96%	1.20%	
70%	0.10%	0.16%	0.24%	0.34%	0.45%	0.57%	0.72%	0.88%	
80%	0.06%	0.08%	0.11%	0.15%	0.20%	0.26%	0.33%	0.41%	
90%	0.17%	0.20%	0.22%	0.25%	0.27%	0.30%	0.33%	0.36%	
100%	0.03%	0.06%	0.07%	0.09%	0.09%	0.09%	0.08%	0.06%	

D.2.3. HTF outlet temperature (@SSG) in dependency on load and inlet temperature

The design HTF mass flow is determined based on the load case with 18 °C WBT and 100 $MW_{e,gross}$ PB output. This mass flow is maintained for all 100 % load cases at different HTF inlet temperatures at the SSG. Based on this mass flow the transferred heat at the SSG is calculated. All part load calculations are referred to this amount of heat, e.g. 50 % load corresponds to 50 % of this heat.

The HTF outlet temperature varies with the HTF inlet temperature at the SSG as shown in Figure D-28 below. This figure shows that a reduction of the HTF inlet temperature of about 7 K causes a change in HTF outlet temperature of about 2.6 K. Thereby the range between inlet and outlet temperature decreases.

The dependency on load is almost linear until 250 °C HTF outlet temperature is reached. This return temperature is controlled via the aforementioned SSG bypass.

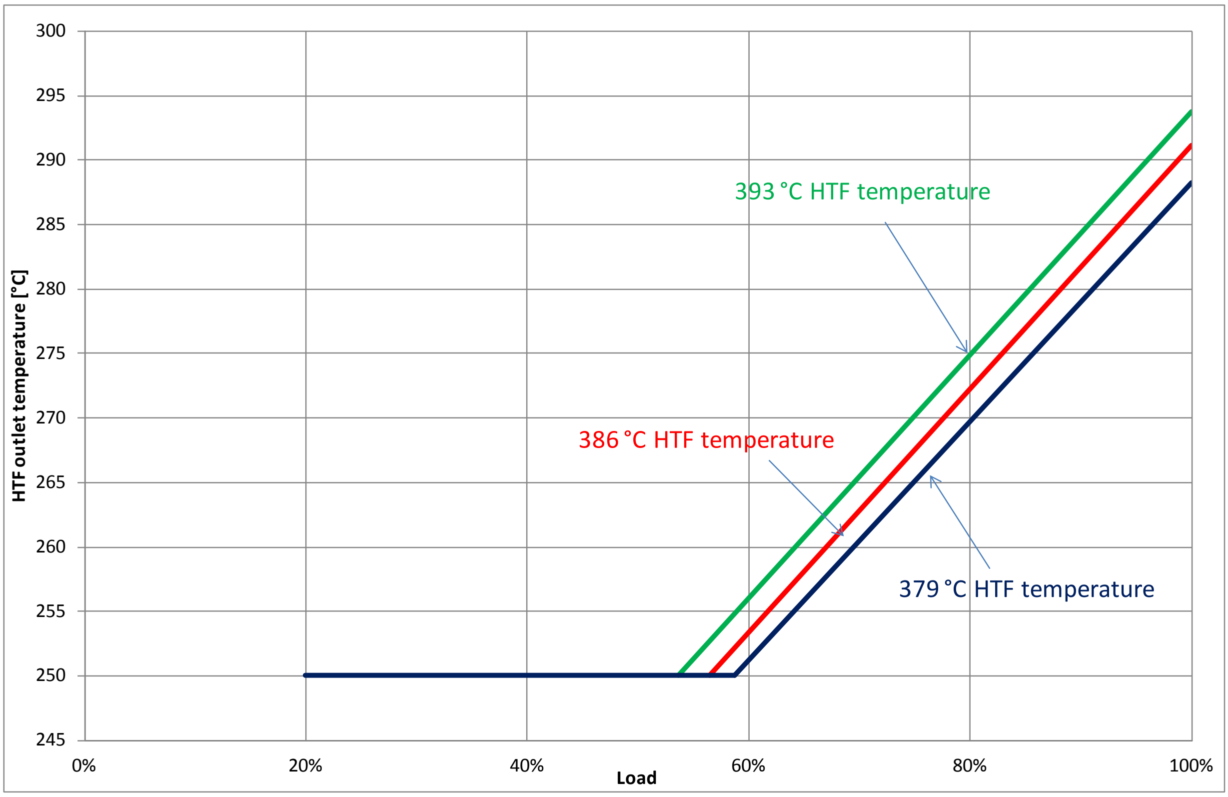

Figure D-28: HTF outlet temperature vs. thermal load at different HTF inlet temperatures

The approximation equation for the HTF outlet temperature depending on the HTF inlet temperature and the load can be written as

$$\vartheta_{HTF,out} = 94.25 * Load + 0.3857 * \vartheta_{HTF,in} + 47.2695 \quad (D.17)$$

If the HTF outlet temperature is fixed between 20% and approx. 60% load, the HTF mass flow can be considered as linear function of the load (see Figure D-29). In this range, the HTF inlet and outlet temperature and therefore the temperature range are fixed and the transferred heat is proportional to the mass flow. Looking at the figure below, it can be seen that this correlation on higher load is characterized by a polynomial function because of the change of temperature range between inlet and outlet of the HTF at the SSG. The transferred heat (load) is a function of enthalpy difference between inlet and outlet and mass flow. The temperature range increases and therefore the mass flow does not decrease proportionally to the load.

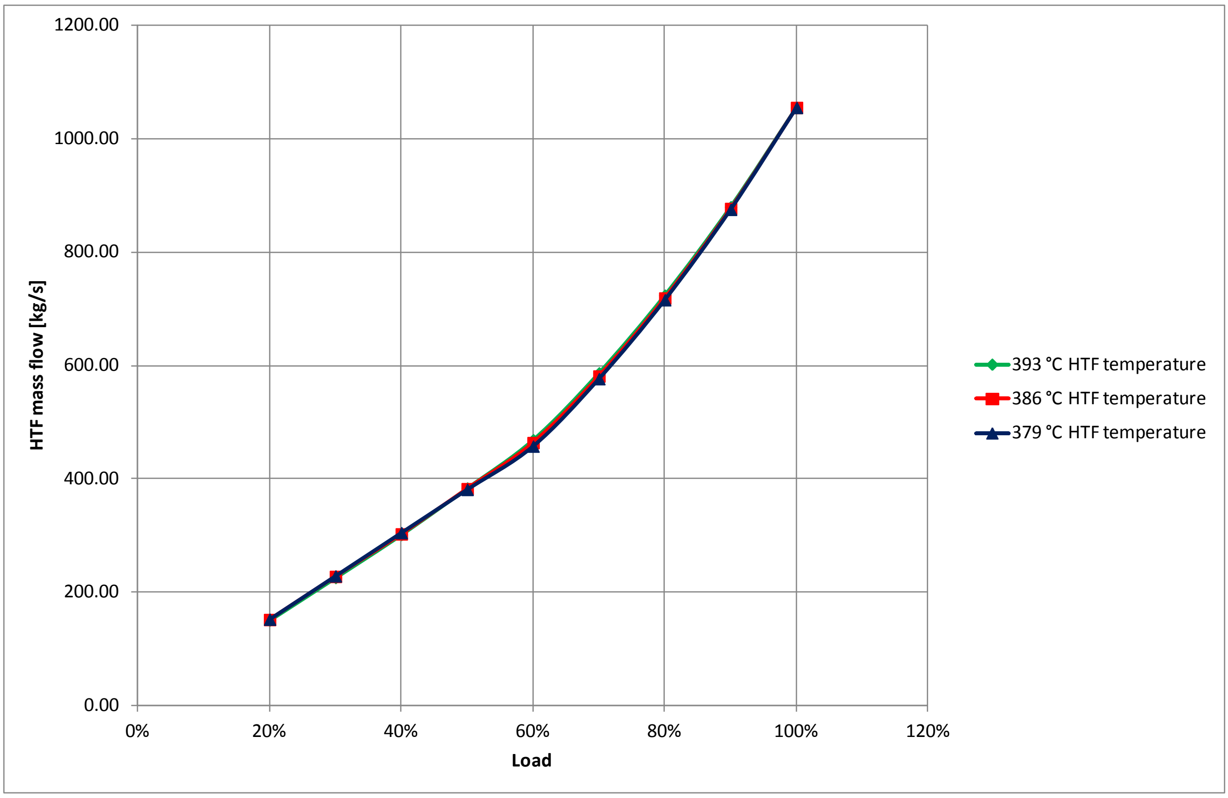

Figure D-29: HTF mass flow vs. load for different HTF inlet temperatures

Using the HTF mass flow as an interface variable between the SF and PB model is possible. Figure D-30 shows the HTF outlet temperature in dependency on the HTF mass flow. The mass flow has a lower limit at approx. 460 kg/s. Smaller mass flows lead to constant temperatures of 250 °C.

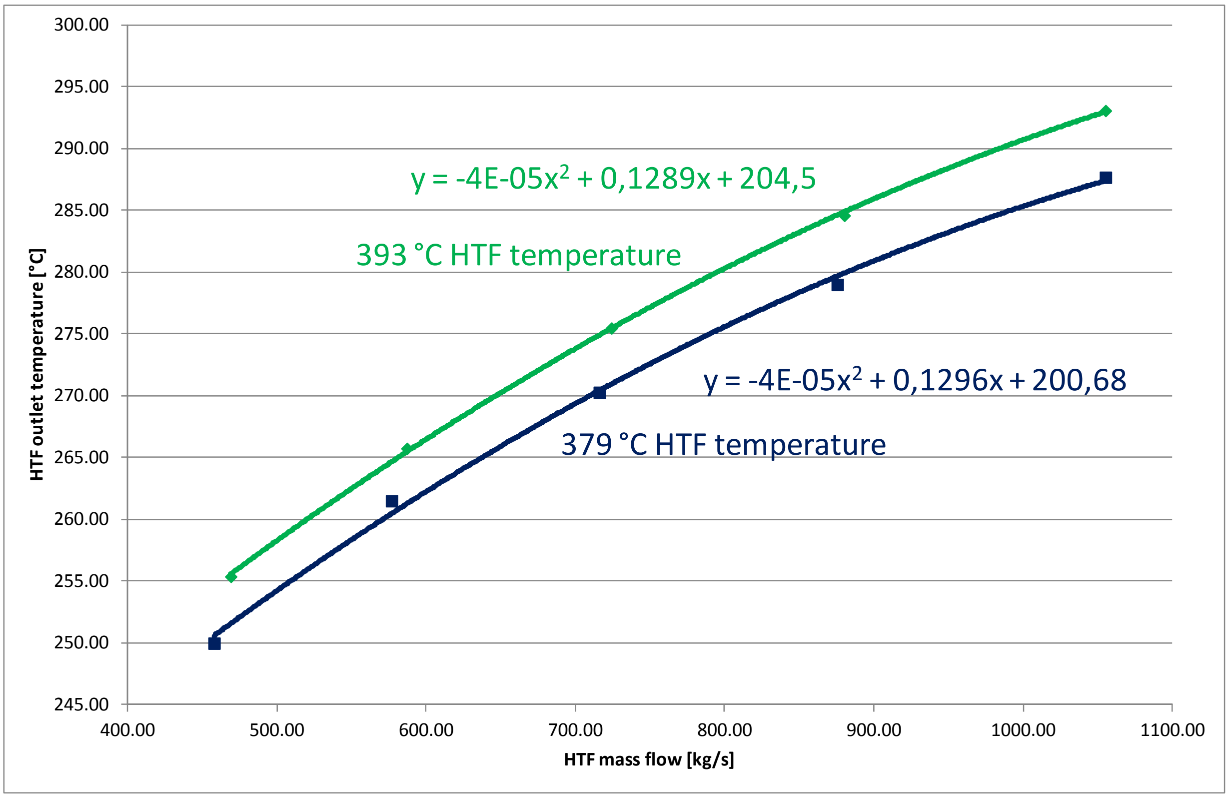

Figure D-30: HTF outlet temperature versus HTF mass flow for different HTF inlet temperatures

The approximation equation of the HTF outlet temperature as a function of different HTF inlet temperatures and mass flows can be written as

$$T_{out}^{PB} = f_2(\dot{m}^{PB}, T_{in}^{PB}) \tag{D.18}$$

$$\vartheta_{HTF,out} = T_{out}^{PB} \; [°C] \tag{D.19}$$

$$\vartheta_{HTF,out} = -0.000042 * (\dot{m}^{PB})^2 + 0.1296 * \dot{m}^{PB} + 203.12 - 0.285 * (393 - \vartheta_{HTF,in}) \tag{D.20}$$

D.2.4. Auxiliary electrical consumption of power block

Basically, the main electrical consumers of the power block in case of the reference configuration based on PTC with synthetic oil and wet cooling tower are the feedwater pumps of the water/steam cycle as well as the cooling water pumps and the fans of the cooling tower.

The auxiliary electrical consumption of the mentioned main consumers for the reference plant is given below.

Table D-19: Auxiliary electrical consumption of PB main equipment at 100 % load (Reference case PTC with synthetic oil and wet cooling tower)

Item	***Unit***	***Value***
Feedwater Pumps	**kW**	**1,800**
Cooling Tower	**kW**	**1,200**
Cooling Water Pumps	kW	500
Fans	kW	700

Basically, the PB has further electrical consumers which are not depicted in this Appendix, as they either have minor consumption which can be neglected and/or are difficult to determine due to dependency on the varying operation modes as well as off-line and online status of the plant (refer also to **Appendix G - Electrical Systems**). For the estimation of consumption of minor systems practically a percentage of the gross electrical production can be applied for annual simulations.

From experience (e.g. annual performance data), it can be stated that for a SPP with such a configuration the PB auxiliary electrical consumption amounts to approx. 35-40 % of the total auxiliary electrical consumption of the plant.

Figure D-31 and Figure D-32 show the auxiliary electrical consumption of the PB depending on the load for a 50 MW SPP based on PTC technology, synthetic oil, TES and ACC. The data are taken from five (5) supplier yield assessments (see Figure D-9 and explanation inD.1.2).

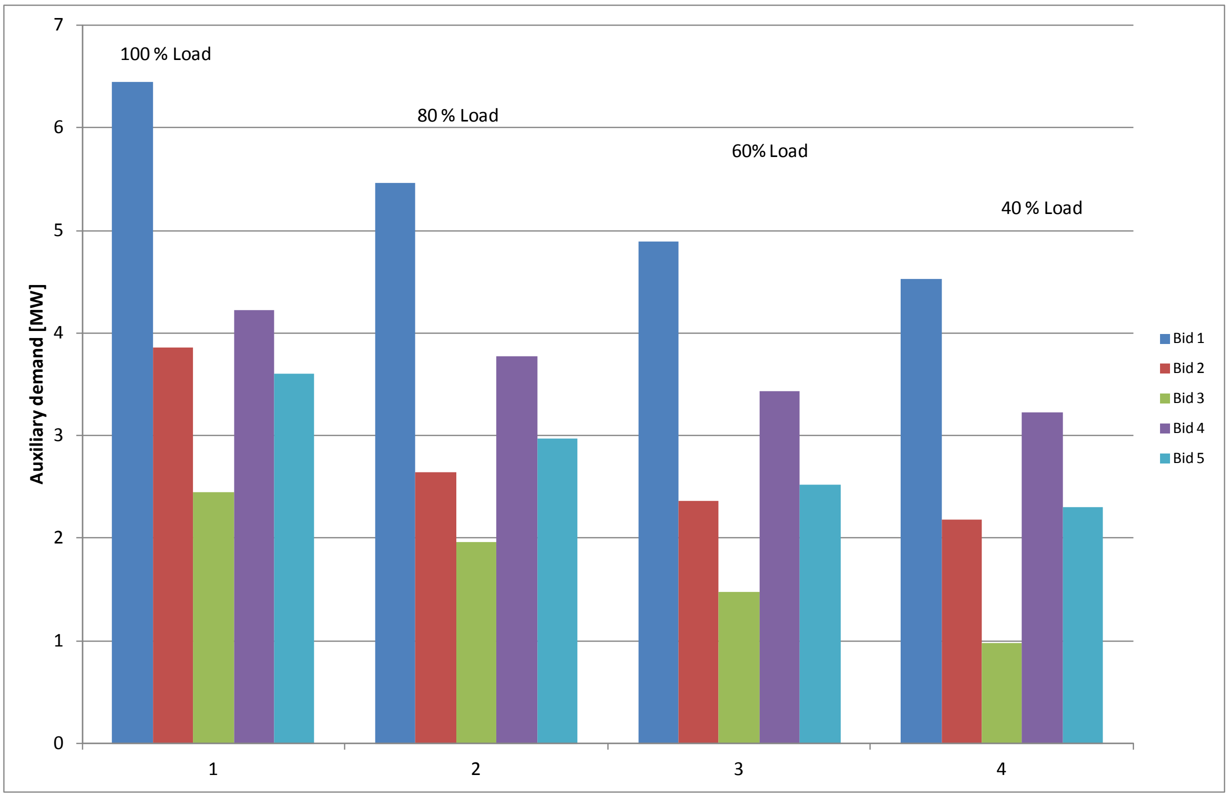

Figure D-31: Auxiliary electrical consumption of a 50 MW SPP

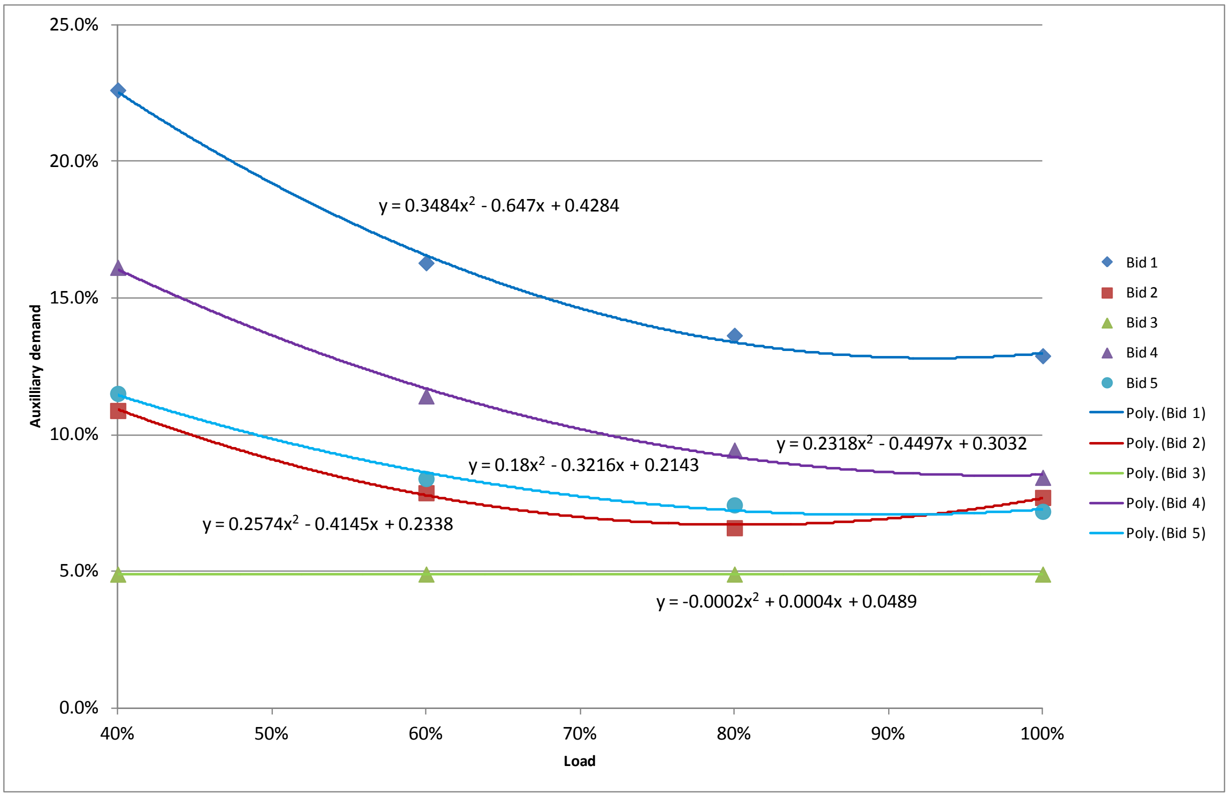

Figure D-32: Auxiliary electrical consumption of a 50 MW SPP in %

The first bar of the bar chart shown in Figure D-31 shows a very high auxiliary electrical consumption and is neglected for the further consideration. As shown in Figure D-32 consumption indicated by the third supplier has always the same percentage of the gross output and does not depend on the load. The corresponding figures are neglected for the further consideration, too. The values of suppliers no. 2 and no. 5 are close together and can therefore be used to illustrate the typical course of the PB auxiliary electrical consumption.

The electrical consumption can be described by a nominal value and a load-dependent correction and a constant consumption for load independent consumers:

$$P_{aux}^{PB} = P_{aux,const}^{PB} + P_{aux,0}^{PB} * f_{aux}\,(P_{gross}^{PB}, T^{*}) \quad \text{(D.21)}$$

$$Aux.demand\ [\%] = P_{aux}^{PB} / P_{gross,0}^{PB} \quad \text{(D.22)}$$

The trend can be described by a polynomial function of the load:

$$Aux.demand\ [\%] = 0.18 * Load^{2} - 0.3216 * Load + 0.2143 \quad \text{(D.23)}$$

The function above is based on an auxiliary electrical consumption of 7.2 MW at 100 % load which proportionally can be considered as an average value for this type of PB.

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APP	Approach [K]
$\bar{c}_{p\ HTF}$	Average specific heat capacity of HTF
CR	Cooling Range [K]
$\dot{m}^{PB}$	HTF mass flow through SSG
P_{aux}^{PB}	PB auxiliary electrical consumption
P_{gross}^{PB}	PB gross output
P_{net}^{PB}	PB net output
$\dot{\mathrm{Q}}^{\mathrm{PB}}$	Heat transferred to PB
$Range$	Range [K]
T_{Cond}	Condensate temperature condenser [K]
T_{CW}	Cold water temperature cooling tower [K]
$T_{Live\ steam}$	Live steam temperature [K]
T_{WWT}	Warm Water Temperature cooling tower [K]
T_{WBT}	Wet Bulb Temperature [K]
TTD_{Cond}	Terminal Temperature Difference condenser [K]
η_{gross}^{PB}	PB gross efficiency
$\eta_{gross,393}^{PB}$	PB gross efficiency at HTF inlet temperature of 393 °C
$\eta_{gross,393,100\,\%}^{PB}$	PB gross efficiency at HTF inlet temperature of 393 °C, 100 % load
$\vartheta_{\ WBT}$	Wet Bulb Temperature [°C]
$\vartheta_{\ HTF,in}$	HTF inlet temperature at SSG [°C]
$\vartheta_{\ HTF,out}$	HTF outlet temperature at SSG [°C]

List of abbreviations

ACC	Air Cooled Condenser
AH	Auxiliary Heater
APP	Approach
CWT	Cold Water Temperature
CR	Cooling Range
DBT	Dry Bulb Temperature
DSG	Direct Steam Generation
HMB	Heat and Mass Balance Diagram
HP	High Pressure
HTF	Heat Transfer Fluid
ITD	Initial Temperature Difference
LP	Low Pressure
PB	Power Block
PTC	Parabolic Trough Collector
SF	Solar Field
SPP	Solar Power Plant
SSG	Solar Steam Generator
S/S	Electrical Substation
ST	Steam Turbine
TES	Thermal Energy Storage
TTD	Terminal Temperature Difference
WBT	Wet Bulb Temperature
WWT	Warm Water Temperature