TECHNICAL UNIVERSITY OF LIBEREC
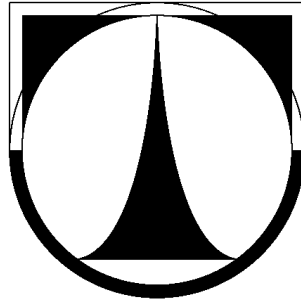
Faculty of Mechatronics, Informatics and Interdisciplinary Studies

Modeling and Simulation of Power Plant Components

Diploma thesis

Bc. Michal Jadrný

Liberec   2015

TECHNICAL UNIVERSITY OF LIBEREC
Faculty of Mechatronics, Informatics and Interdisciplinary Studies

MODELING AND SIMULATION OF POWER PLANT COMPONENTS

Diploma thesis

DIPLOMA THESIS ASSIGNMENT

(PROJECT, ART WORK, ART PERFORMANCE)

First name and surname: **Bc. Michal Jadrný**

Study program: **N2612 Electrical Engineering and Informatics**

Identification number: M12000268

Specialization: **Mechatronics**

Topic name: **Modeling and Simulation of Power Plant Components**

Assigning department: **Institute of Mechatronics and Computer Engineering**

R u l e s   f o r   e l a b o r a t i o n :

1. Literature research relating the basics, modeling and simulation of thermo- dynamical power plant processes.

2. Familiarization with the simulation system.

3. Determination of balance equations for the individual power plant components.

4. Development and realization of the power plant components.

5. Designing a simulation model for static behavior analysis of the overall model (thermo-dynamic cycle).

6. Performing the simulation calculations to prove correct functionality of the overall model.

7. Documentation of the results.

Scope of graphic works: **In respect to the documentation needs**

Scope of work report
(scope of dissertation):   **c. 50–60 pages**

Form of dissertation elaboration: **printed/electronical**

Language of dissertation elaboration: **English**

List of specialized literature:

[1] BECKMANN, G.; GILLI, P. V. Thermal Energy Storage: Basics, Design, Applications to Power Generation and Heat Supply. Wien: Springer-Verlag, 1984.

[2] HORLOCK, John H. Combined power plants : including combined cycle gas turbine (CCGT) plants. Oxford: Pergamon Press, 1992.

Tutor for dissertation: **prof. Dr. Ing. Alexander Kratzsch**
                                     Hochschule Zittau/Görlitz, IPM, Germany

Dissertation Counsellor: **prof. RNDr. Stefan Bischoff**
                                     Hochschule Zittau/Görlitz, IPM, Germany

Date of dissertation assignment: **10 October 2014**

Date of dissertation submission: **15 May 2015**

prof. Ing. Václav Kopecký, CSc.   doc. Ing. Milan Kolář, CSc.
Dean   Department Manager

L.S.

Liberec, dated: 10 October 2014

Declaration

I hereby certify that I have been informed the Act 121/2000, the Copyright Act of the Czech Republic, namely § 60 - Schoolwork, applies to my diploma thesis in full scope.

I acknowledge that the Technical University of Liberec (TUL) does not infringe my copyrights by using my diploma thesis for TUL's internal purposes.

I am aware of my obligation to inform TUL on having used or licensed to use my diploma thesis; in such a case TUL may require compensation of costs spent on creating the work at up to their actual amount.

I have written my diploma thesis myself using literature listed therein and consulting it with my thesis supervisor and my tutor.

Concurrently I confirm that the printed version of my diploma thesis is coincident with an electronic version, inserted into the IS 5TAG.

In Liberec 5th January 2015

Signature:

Acknowledgements

Abstract

This diploma thesis deals with development of simulation model for the thermal energy storage facility located in Zittau, Germany, and with the extension of an existing model of real power plant.

In the first section of thesis the basic types of thermal power plants are described. Then the main focus of work is given to thermal energy storages, to their principles, types and examples of use.

In practical part of diploma thesis the mathematical model for one mode of the THERESA facility is derived. The development of simulation models for facility is shown, firstly for individual modes that are relevant for the thermal energy storage, and then is shown a simulation model for the entire facility and all modes. At the last part of the theses are explained the implementation of the thermal energy storage to the existing simulation model of reference power plant and necessary modifications of model, which were made to achieve the valid results.

Key words

Thermal power plant, thermal energy storage, heat storing, thermal cycle process, mathematical model, THERESA facility, simulation model, EBSILON

Abstrakt

Diplomová práce se zabývá vývojem simulačního modelu pro zařízení s úložištěm tepelné energie nacházejícího se ve městě Zittau v Německu a dále rozšířením existujícího simulačního modelu reálné elektrárny.

V první části práce jsou popsány základní typy tepelných elektráren a poté je hlavní pozornost věnována úložištím tepelné energie, jejich principům, typům a příkladům jejich použití v praxi.

V praktické části práce je nejprve odvozen matematický model pro jeden z režimů zařízení THERESA. Vývoj simulačních modelů je nejprve ukázán pro jednotlivé režimy zařízení, které využívají úložiště tepelné energie, a dále je popsán vývoj simulačního modelu pro celé zařízení a všechny režimy. V poslední části práce je vysvětlena implementace modelu úložiště tepelné energie do existujícího simulačního modelu referenční tepelné elektrárny a jsou popsány nezbytné úpravy modelu, které byly provedeny pro dosažení platných výsledků simulací.

Klíčová slova

Tepelná elektrárna, úložiště tepelné energie, skladování tepla, tepelný proces, matematický model, zařízení THERESA, simulační model, EBSILON

Table of contents

List of figures

List of tables

List of used symbols and abbreviations

Equation symbol	Description	Unit
g	specific Gibbs free energy	[J/mol]
h	specific enthalpy	[kJ/kg]
m_f	mass flow	[kg/s]
n	polytrophic exponent	[-]
p	pressure	[bar], [MPa]
P	power	[kW]
Q	heat	[kJ]
R	specific gas constant	[kJ/kg·K]
s	entropy	[kJ/kg·K]
T	temperature	[K]
W	work	[kJ]
x	vapor fraction	[kg/kg]
γ	dimensionless Gibbs free energy	[-]
η	efficiency	[-]
π	reduced pressure	[-]
τ	inverse reduced temperature	[-]

Abbreviation	Description
CCGT	combined cycle gas turbine
ESD	energy storage device
TES	thermal energy storage
PV	pressure-volume
T-s	temperature-entropy
Nr.	number

Introduction

The largest share of electric power in the world is produced in conventional thermal power plants. To the category of thermal power plants can be classified fossil-fuels, nuclear and solar power plants. This method of electricity generation has, in general, several disadvantages and limitations in flexibility, which can be removed or reduced by the using of the thermal energy storage device placed directly in the power plant.

After introducing the basic methods of obtaining thermal energy, which are used to produce electricity, is therefore a major part of the diploma thesis devoted to the thermal energy storages. The reasons for using these storage devices are explained, they are presented the basic principles and at the end the examples of use in different types of power plant are described.

In practical part of diploma thesis the calculations of thermal cycle's processes are shown first. The Carnot cycle is described and the calculations of the state variables are derived. Then the THERESA research facility located in Zittau, Germany is introduced and for the selected mode containing thermal energy storage the existing ideal mathematical model for static behavior is presented.

The calculated mathematical model serves as the basis for the simulation model, which is constructed in the EBSILON program. The first is learning how to work with the simulation tool and next the ways to select the individual components and how the simulation model was created. There are shown different models for different modes of facility as well as one complex model for whole facility.

The last part of thesis describes how the thermal energy storage from THERESA model was implemented to the existing model of Lippendorf power plant. The research objective was to determine what impact the thermal energy storage on the selected plant parameters will have at different modes and loads. The results of the implementation are discussed in the last chapter and these results will serve as the basis for discussions with the company, which owns and operates Lippendorf power plant, about possible future implementation of the real thermal energy storage to the power plant.

1. Thermal power plants

Thermal power plant is the general term for the power stations, where the electricity is produced by conversion from energy bound in fuel. Often by this term are described power stations that using fossil fuels. For the purpose of this diploma thesis is further used the principle of steam power plant described in next chapter.

1.1. Principles of thermal power plants

1.1.1. Coal power plant

The most commonly used type of thermal power plant is steam power plant, which using coal, gas, oil or biomass as a fuel. The principle of a coal power plant partially uses well-known principle of the steam engine. Combustion of fuel leads to the release of chemical energy bound in fuel. Water is first preheated in the feed water tank. Then the water is heated up by residual heat of combustion gases in preheaters and this water supplies the steam generator, where it is vaporized. Saturated steam that leaves the steam generator is converted into superheated steam in superheater. Superheated steam flows to the steam turbine, which is divided into several blocks to achieve higher efficiency:

    a) high pressure part,

    b) heating to the initial temperature,

    c) middle pressure part,

    d) low pressure part.
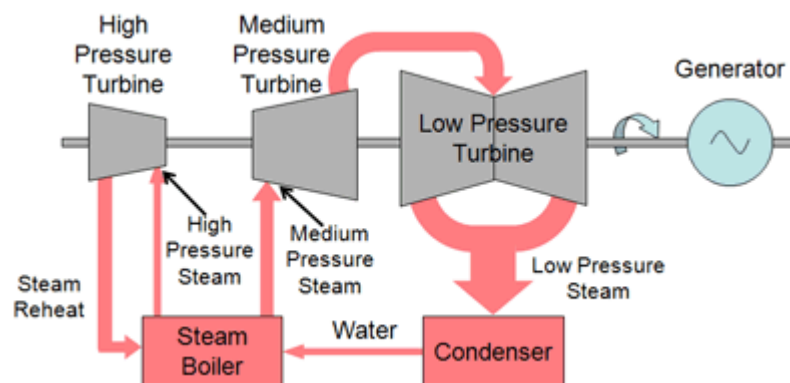
Figure 1: Principle of multistage steam turbine [22]

The basic principle of multistage steam turbine is shown in Figure 1. Additionally, the middle-pressure and low pressure part of turbine are used for feed water heating. This allows the higher efficiency of thermal cycle. The turbine is connected to electrical generator through the common shaft. After conversion of energy the steam flows from low-pressure part of turbine to condenser, where the steam is converted back into liquid state, and flows to feed water tank. The principle of steam power station is shown in Figure 2. [1][2][4]
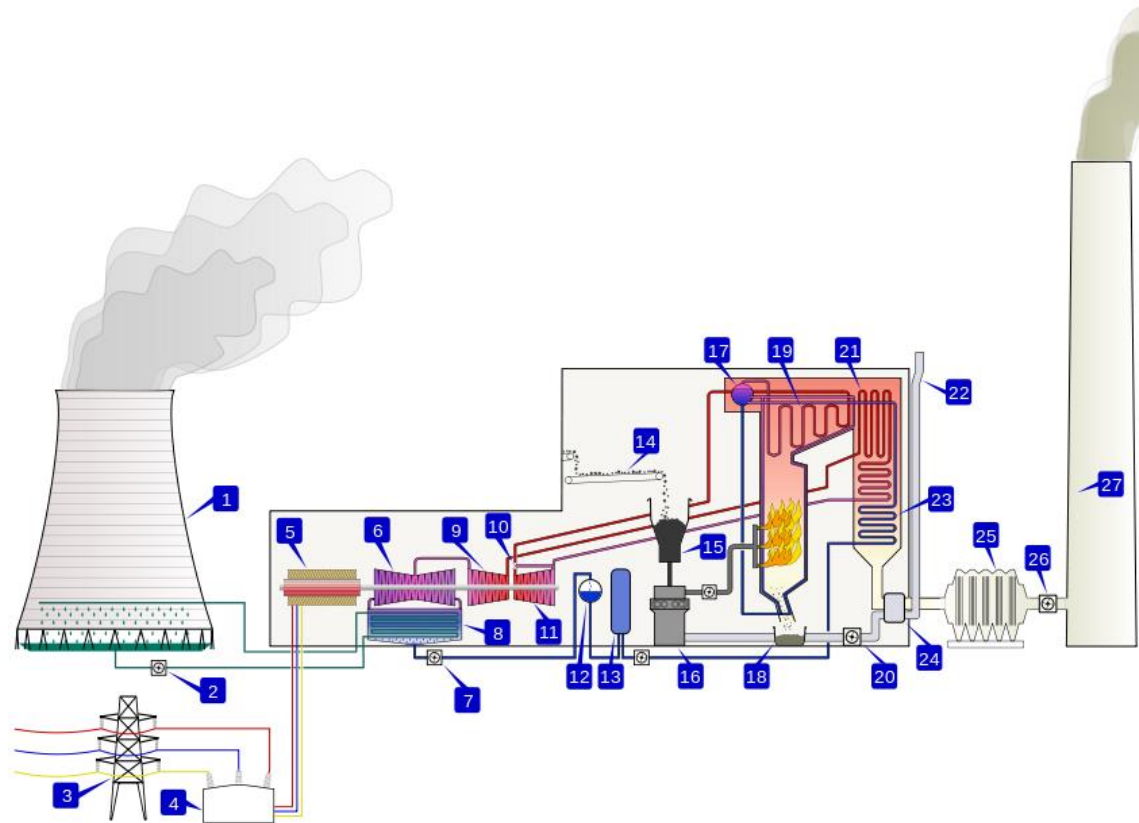
1. Cooling tower	10. Steam control valve	19. Superheater
2. Cooling water pump	11. High pressure turbine	20. Forced draught fan
3. Transmission line	12. Deaerator	21. Reheater
4. Step-up transformer	13. Feed water heater	22. Combustion air intake
5. Electrical generator	14. Coal conveyor	23. Economizer
6. Low pressure turbine	15. Coal hopper	24. Air preheater
7. Condensate pump	16. Pulverized fuel mill	25. Precipitator
8. Surface condenser	17. Boiler steam drum	26. Induced draught fan
9. Middle pressure turbine	18. Bottom ash hopper	27. Flue gas stack

Figure 2: Coal power plant scheme [23]

1.1.2. Gas power plant

Combined cycle gas turbine (CCGT) plant is more ecological variant of coal-fired plant. High efficiency of electricity production is ensured by the two thermal cycle's synergy, steam and gas. By the burning of gas is released its chemically bound energy, and the energy is used first in gas turbine and subsequently in the steam generator to generate steam, which drives the steam turbine. Electrical energy is obtained for both, gas and steam turbine. Heat flow of the gas turbine consists of a compression inlet air, its mixing with the fuel that is burnt, and the expansion of the combustion gases in a gas turbine. Heat flow of the steam turbine consists of heating pressurized water to boiling state, evaporation, superheating of steam to the operating temperature and the expansion of the steam in the steam turbine. Figure 3 shows the principle of CCGT power plant.
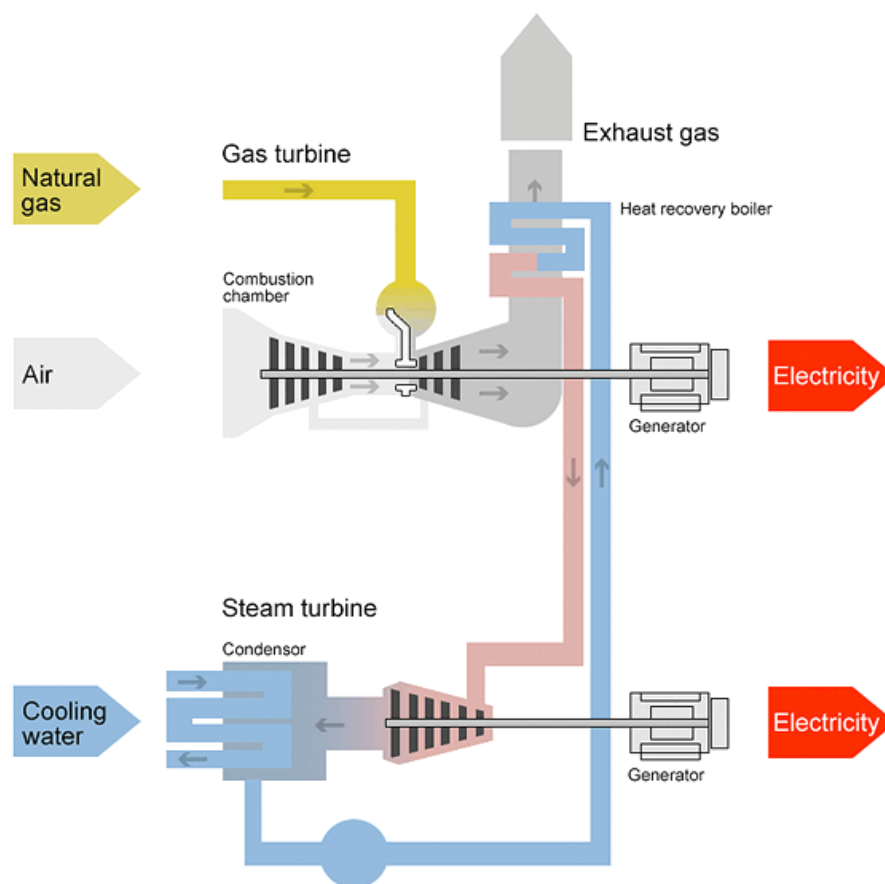
Figure 3: CCGT plant scheme [24]

The advantage of combined circulation is nearly twice the efficiency of electricity generation compared to conventional coal-fired power plant at achieving minimum environmental burden, which is compared with coal-fired plants by up to 70% lower. [3]

1.1.3. Nuclear power plant

Another way how to obtain the thermal energy is by process of fission of atomic nuclei. As fuel uranium or the plutonium is used. The cleavage reaction of the fuel takes place inside a sealed reactor. The commonly used type of reactor is a pressurized boiling water reactor. Fission of nuclei is accompanied by release of a large amount of heat. The released heat is transferred by water in primary thermal circuit. This hot water is used to produce saturated steam in the steam generator. Steam circuit is referred as secondary thermal circuit. Furthermore, the principle of electricity production is the same as a principle of steam power plant and the principle is shown in Figure 4. [2]
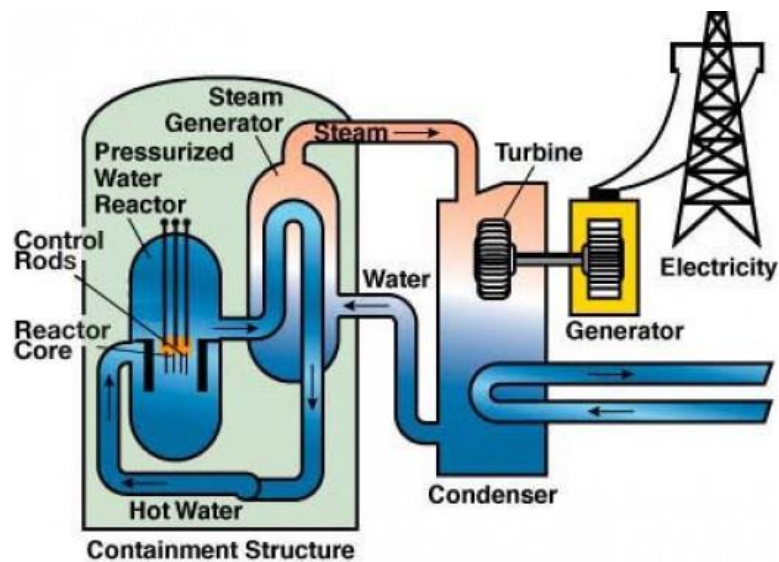
Figure 4: Nuclear power plant scheme with boiling water reactor [25]

1.1.4. Geothermal and solar power plant

The thermal energy can be also obtained from renewable resources. In case of geothermal energy is used thermal energy of the Earth's core. This internal energy is generated from radioactive decay and continual heat loss from Earth's formation and can be used to generate electricity in geothermal power plant or on a smaller scale for heating. Greater extension of this king of electricity generation prevents technologically sophisticated solution due to frequent accumulation of dirt. The second problem is that the sufficient thermal gradient can be achieved almost exclusively in geologically instable areas, which places high demands on the construction of the power plant. Figure 5 shows the simplified principle of geothermal power plant.
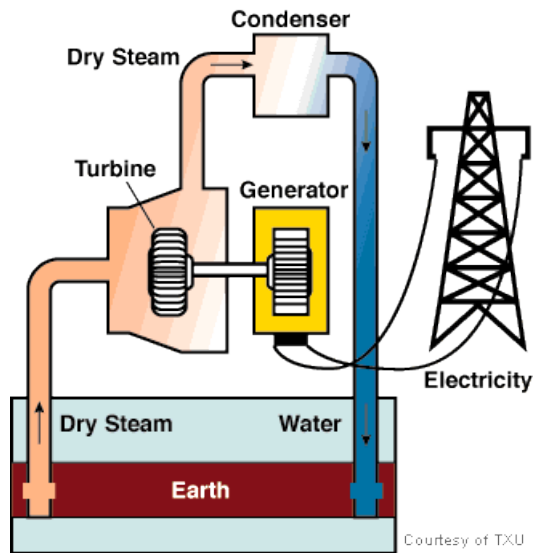
Figure 5: Geothermal power plant scheme [26]

Second way for using renewable energy resources is the use of solar energy. Solar energy is concentrated into a single point, where the water is converted to the steam as shows the Figure 6. Steam is then used to drive the turbine. Water from the condenser after the turbine is fed back into the solar field. [2][5]
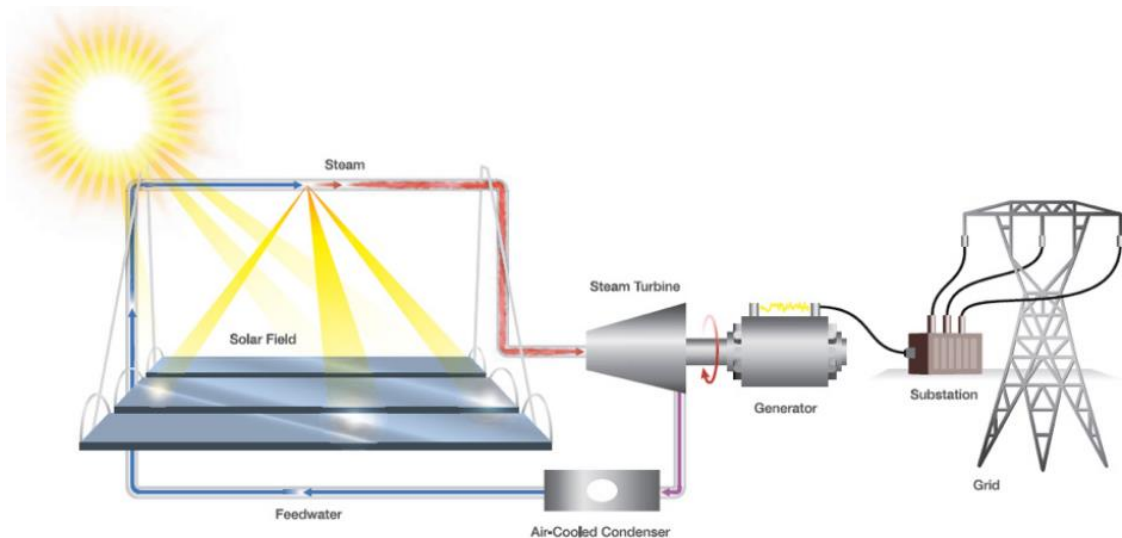
Figure 6: Solar thermal power plant scheme [27]

1.1.5. Characteristics of thermal power plants

Thermal power plants are the most used power stations in the world. Their biggest advantages are the well-known principle, because heat conversion to another type of energy is used since the first steam engine. Thermal power plants are relatively independence on geographical location, unlike for example solar and hydro power plants. Of course, the fuel must be available as well as cooling water for power plant.

These power stations have possibility to work 24/7 and the installed power is high. Their disadvantages are non-environmental friendly operation of coal power plants, high fuel consumption and low flexibility. Thermal power plants are suited for continuous operation in the area of plant nominal power, so called base load power stations. The rise time of a coal power plant from zero to the nominal value of output power is around 24 hours and in a matter of hours also varies the control time for the output power control. Frequent changes of the plant output power leads to increased wear of components, the probability of failure is higher and costs of maintenance are also increasing.

Problem is that there will always be the difference between the amount of electricity produced and consumed. The result of this difference is production peaks and production valleys. In case of peaks the amount of consumed electricity is higher than amount of produced electricity. In case of production valleys the situation is reversed. This problem is increasingly important and it is related with the increasing number of renewable energy sources, whose electricity production is fluctuating during day, because of dependency on weather conditions.

Grid-wide control is actually possible by water or gas-burning power plants, because these plants have a startup time in minutes. By this method it is possible to control network peeks. Production valleys can be covered by pumped water storage plants. Both of these methods are using for balancing production and consumption of electricity different kind of energy source than the primary energy source. The difference between supply and demand can be also partly solved directly in thermal power plant by installation of energy storage device (ESD) inside the power plant. [1][3][4][6]

2. Thermal energy storages

The task of the energy storage device in power plant is to store the energy when the production is higher than consumption and use the stored energy when the demand rises. For example the function of battery storage can be described so that the energy store device takes the cheap electricity from the grip and after some time supplies the expensive electricity back to the grid. Unlike thermal pipelines the electrical grid itself has almost no ability to store energy in any form. Energy can be stored with the use of external device, which can by placed:

a) directly in the power plant – for this purpose are used ether thermal or mechanical storage,

b) in the distribution grid – the direct storage (electromagnetical, electrochemical storage) or indirect storage (pneumatic, hydraulic or pumped thermal storage),

c) at the customer side – thermal, mechanical or electrochemical storage.

Thermal energy storage (TES) is generally suitable as short time storage (a matter of hours). On the other hand the hydraulic storage is suitable like long time storage with possibility to supply by electricity up to few hundred hours. Other text will deal only with thermal energy storages, because it's the efficient way how to store energy in thermal power plant. There is no need of complicated mechanisms for storing; energy is stored before conversion to electricity, thus reducing the operation time of the device and the cost of the storage device. The effects of TES are shown on example of power plant block-load curve in Figure 7. The main positive effects of implementation the TES to the power plant are increasing energy efficiency of power plant, overload capacity (marked as 1) and minimum load reduction (marked as 2).
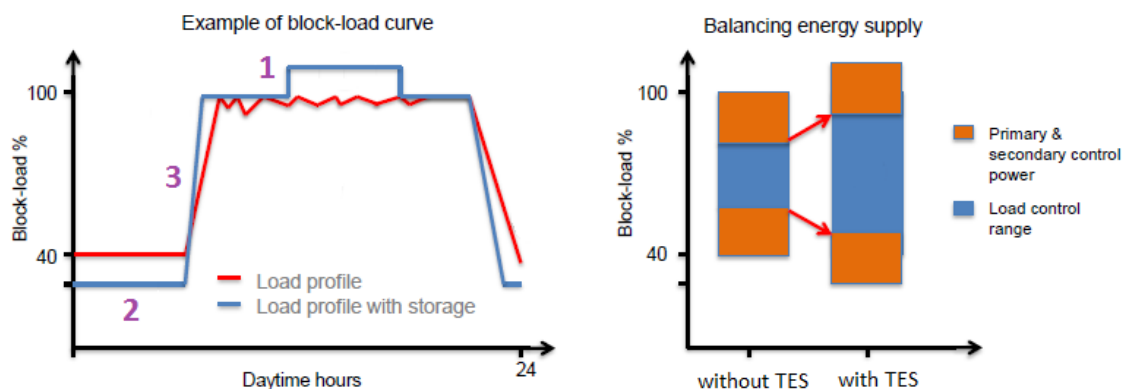
Figure 7: Block-load curve of power plant with embedded TES [8]

22

Other effects are better energy supply flexibilization and optimization of the startup and shutdown processes (marked as 3). Also the reliability of the power plant is enhanced. [6][7][8]

2.1. Basic description of thermal energy storage

Thermal energy storage system can be defined like system, which provides the storage of thermal energy, in case of displacement storage this system also contains the charging and discharging devices. Thermal energy storage system includes the thermally insulated storage vessel, the storage medium, the charging and discharging devices and all necessary auxiliary circuits. Thermal energy storage itself is defined like chemical or physical process, which allows storing of the energy. The basic function of TES is possible to describe with respect to energy balance like

$$stored\ energy = input\ enegry - output\ energy \qquad (1)$$

As storage medium are used solid materials (metals, ceramics or rocks), liquids (water, oil or liquid metal), a phase change media (liquid and vapor) or gasses. The heat transfer media are typically a phase change media or gasses. TES can be categorized according to the used medium in this way:

a) direct storage – storage medium and heat transfer medium are the same,
b) indirect storage – thermal energy is passed either by heat transfer or by mass transfer through different transfer medium,
c) sorption storage – this fits for special storage media, which releases heat during mass absorption and releases mass during cooling. The gasses are often used as a transfer medium. [6][7]

2.2. Types of thermal energy storage

The basic division of the TES is shown in Figure 8. The thermal energy may be stored by temperature increasing or lowering of the storage substance, these storages are called sensible heat storages. Latent heat storages are based on the phase changing energy of substance. The last method uses thermochemical reaction between transfer and storage medium. The monitored parameters in the selection of the TES are thermal capacity, thermal conductivity, stability of the storage medium and its cost and cost of the whole TES technology.
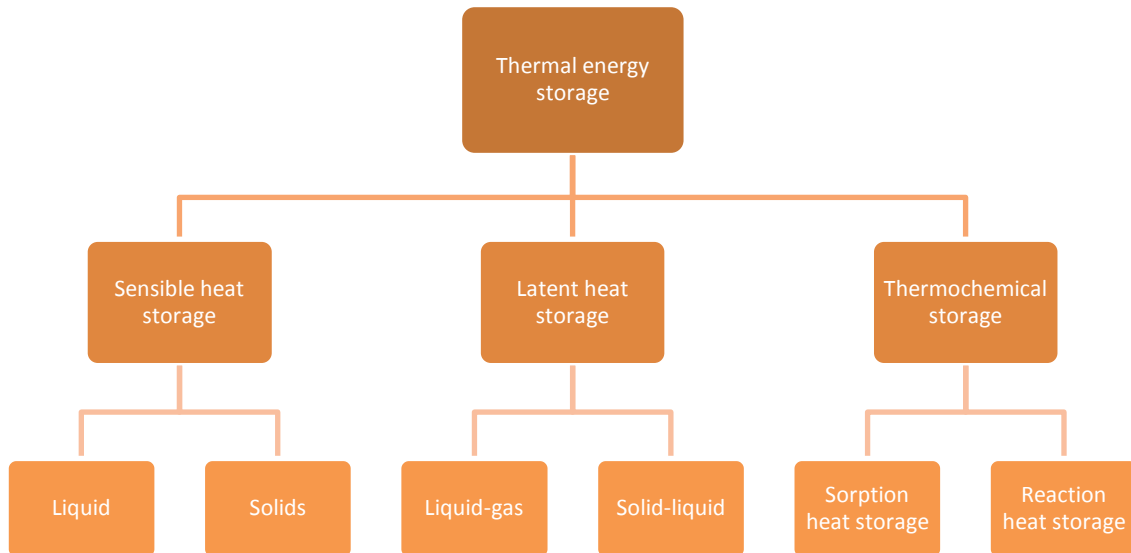
Figure 8: Division of the TES

The sensible heat storages are most commonly used. Their technological solution is simpler then for both other methods. Another advantage is that they are capable of unloading in wide temperature range. The disadvantage is the need for the large pressure storage vessel, because the capacity of storage is dependent on the volume of storage medium. The latent heat storages have higher storage density and can be stored without pressure vessel, depending on the used material. On the other hand the cost of storage media is several times higher. Thermochemical storages theoretically offer the highest storage density and wide temperature range, but they are not actually ready for the practical applications. Individual types are further discussed. [6][7]

2.2.1. Energy storing in saturated fluids

The first media used for the sensible heat storing are saturated fluids. For this type the water or steam as a storage medium are used. TES contains thermally insulated pressure vessel, where the water is in lower part and steam is in upper part of vessel. Water and steam are in thermodynamics equilibrium. Example of this type of storage is shown in Figure 9. Three quarters are filled with saturated water and last quarter is filled by steam.

Charging takes place by injecting superheated steam into lower part of the vessel. Steam must have the higher pressure than the water, into which is injected. Second way how to achieve the charging is by heat exchange surface located in the middle of vessel.

In this case the medium, which is fed to heat exchange surface, must have higher temperature than the temperature inside the vessel. The discharge valve is located at the top of the vessel. During discharging the wet or saturated steam flows from the vessel. The discharging is accompanied by a significant decrease of pressure inside vessel. Because of this pressure change during charging and discharging, this type of storage is sometimes called sliding-pressure storage. The advantages of this type are possibility of charging by the steam, steam at the output of storage and low investment costs.

1. Pressure vessel
2. Water content
3. Steam cushion
4. Discharge line for saturated steam
5. Discharge line for superheated steam
6. Discharge heat exchanger surface
7. Discharge line for hot water
8. Charge line for steam
9. Charge heat exchanger surface
10. Charge line for (hot) water
11. Charge line for heating steam
12. Internals
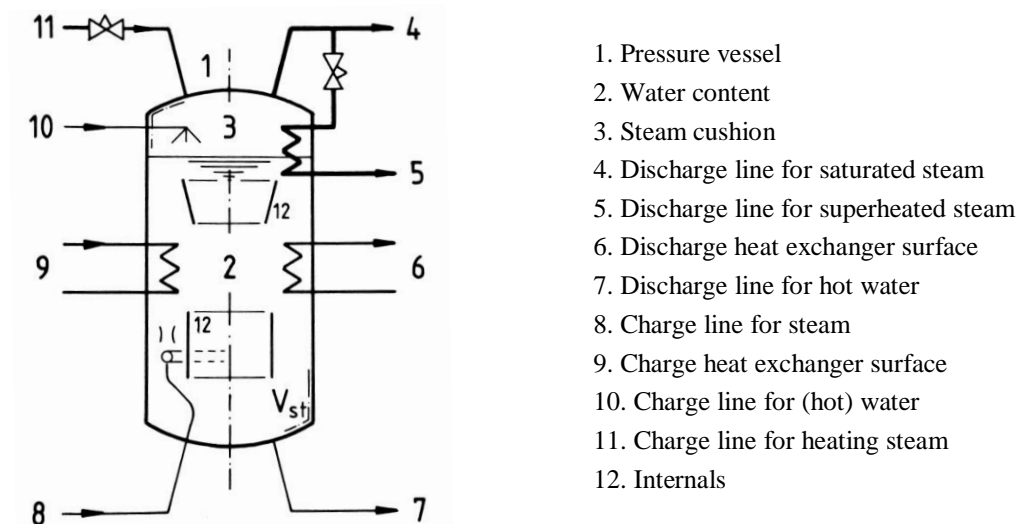
Figure 9: Sliding-pressure thermal energy storage [28]

On the very similar principle works so-called expansion storage. The main differences are between charging and discharging. The vessel is largely filled by the water; the volume of steam is lower than in previous case. The charging is achieved by injecting of hot water. The discharge valve is located at the bottom of vessel and during discharging the hot water flows out from the vessel. The pressure drop is not as significant as in sliding-pressure storage because of the additional steam production inside the vessel.

Last representative of heat storing in saturated liquids is so-called indirect sliding-pressure storage. Storage contains a closed pressure vessel with constant volume of storage medium. Charging and discharging is realized through heat exchange surfaces. [6]

2.2.2. Energy storing in solids

Another interesting method of sensible heat storage is possibility to store the heat inside solid materials. TES again contains thermally insulated pressure vessel, inside of which is storage medium in the form of exchange beds. In the beds are channels, in which transport medium flows through. The path must be designed so that the medium flows through the sufficient quantity of heat transfer surfaces. The shape of the bed is dependent on the storage medium. The solid body is heated during loading of TES and cooled during unloading, both without phase change.

1. Vessel
2. Upper discharge/charge line
3. Lower discharge/charge line
SSM - Stationary solid storage medium
HTM - Heat transfer medium
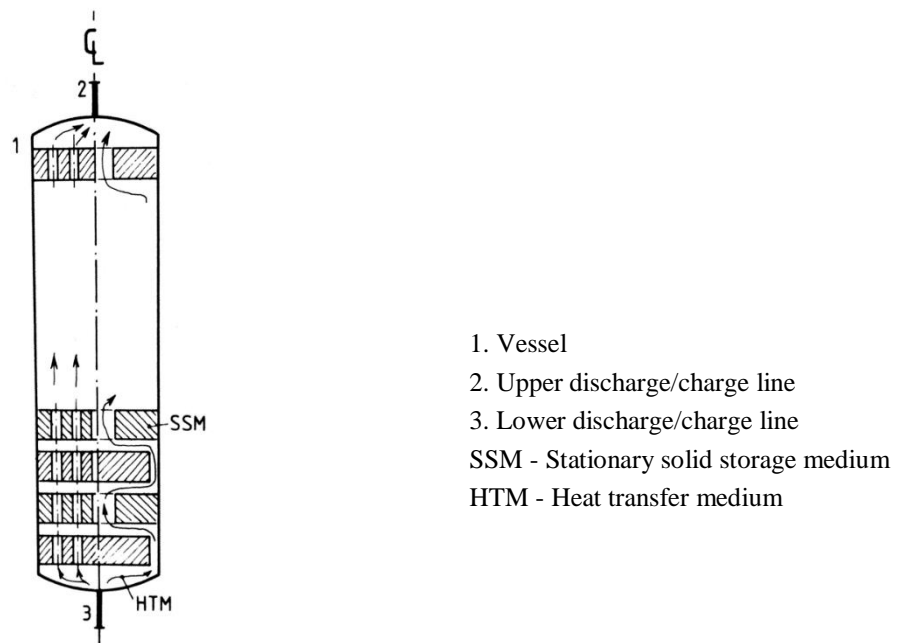
Figure 10: Solid thermal energy storage [29]

In terms of materials, the logical way is to use some of the metal. Cast iron is the metal with the largest thermal capacity while having the highest thermal conductivity. For nonmetallic materials the fireclay has the greatest heat capacity, but his disadvantage is the low thermal conductivity. It is better (also regarding to economic reasons) use of magnesium oxide, which has a little lower heat capacity, but four times greater thermal conductivity than fireclay. [6]

2.2.3. Energy storing in pressurized liquids and pressurized gas

The last two types of sensible heat storage are storing in pressurized liquids and storing in pressurized gasses. Pressurized liquid, where the pressure of the liquid is higher than saturation pressure, or supercooled liquid, where the temperature of the liquid is lower than saturation temperature, are used as the storage media. The storage

system is maintained at a constant pressure. This type of TES is widely used in solar power plants and the storing is based on liquid phase in form of pressurized liquid.

Single phase gas is used as the storage medium for storing in pressurized gas. The gas, which is charged, stored and discharged, is the same. This type of storage is not suitable for heat storing, because of low heat storage capacity, but it is commonly used in gas turbine power plants like additional power storage. The storage system consists of the pressure vessel and if the wet gas is used, also the water draining is necessary. [6]

2.2.4. Latent heat storage

Unlike previous cases the principle of latent heat storage is based on the phase-change of the storage medium. The temperature still the same during the change and the change leads to the release or absorption of latent heat. Phase change from liquid to gas phase is accompanied by the change in volume. Therefore it is used only a small part of the media for change of phase, so there are only a minor changes in volume.

Latent heat storage has, in comparison with the previous principles, several advantages. Latent heat storage has higher energy density per unit volume and also higher energy density per unit mass. Materials that can be used for latent heat storage is a large amount. Due to cost aspects, metallic materials like tin or lead can't be used, instead salts like sodium nitrate or potassium nitrate are typical materials. On the other hand, it is unable to determine, which material is most suitable, and therefore it is always necessary to select a material of storage with respect to the entire facility, optionally the entire power plant, where will be use. Charging and discharging of the latent heat storage is not so easy like in the case of sensible heat storage. It is because convection in the solid state is prevented, volume is changing and the thermal conductivity of the storage medium is generally low. The development in recent years is concentrated right on the latent storages to overcome these obstacles and to help higher deployment in practice. [6][7]

2.2.5. Thermochemical storage

All previously mentioned principles of TES can be marked in summary as direct heat storages. Direct heat storage is characterized so that the energy (heat) is transferred directly to the storage media. The heat inside the storage is stored together with the corresponding amount of entropy. From this it follows a disadvantage because

the maximum amount of stored energy is limited by the maximum entropy of the material and limited storage capacity. This disadvantage is removed by the thermochemical storage. Thermochemical storage is based on the suitable reversible reaction and it offers indirect storing by the separate storing of reaction partners and it means by the separate storing of entropy flow and energy. This type of storage typically uses two different principles. The first principle is based on heat absorption and energy storage is called a sorption heat storage or absorption storage.

As an example it is given energy storage in the form of silica gel. Principle of this sorption heat storage is shown in Figure 11. Charging takes place so that the steam at the input heats the silica gel and during this process some steam is released. Steam is captured and converted into water. The cooled water passes from the input to the output of the TES. Condensed water and dry silica gel is separated during energy storing. During discharging the liquid water is evaporated with help of the input water with higher temperature than stored water. The steam is absorbed by the silica gel and the absorption heat is transferred to input steam. Development of this principle is still continuing by searching of suitable pairs and by studying their reactions.
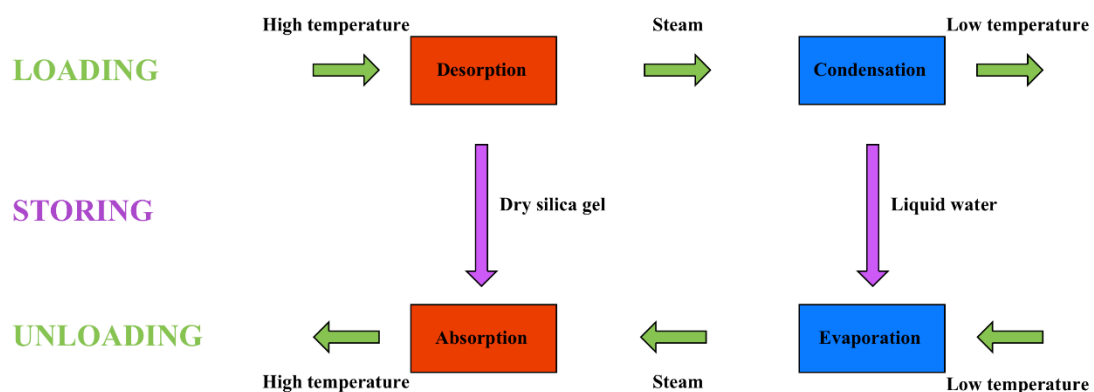
Figure 11: Sorption heat storage [9]

The second principle is based on merging and separation of chemical compounds. The reaction may or may not be started by the catalysts. Example of the compound can be the salt NH_4HSO_4, which consists of the reaction products NH_3, SO_3 and H_2O. Products have a high energy density, they can be stored in liquid phase and all three products can be easily separated and stored. This principle is not further described, despite the fact that it provides a significant improvement potential for storing of thermal energy in comparison with the sensible and latent heat storages. It is mainly

because of this principle is currently in the stage of research and development and its implementation in practice is still in the future. [6][9]

Some types of TES are known from the past, the development of other types of TES is given by the further research. Figure 12 shows the number of research projects dealing with the storage of thermal energy in Germany. Data are from June 2013 [10]. From the data we can see that the main interest is focused in latent heat storages and increasing the density of heat storages. As described above, thermochemical heat storage offers greater potential for storing, but its practical use is in the distant future. In contrast, the latent heat storages are actually often used and their current research focuses primarily on finding suitable pairs of materials.
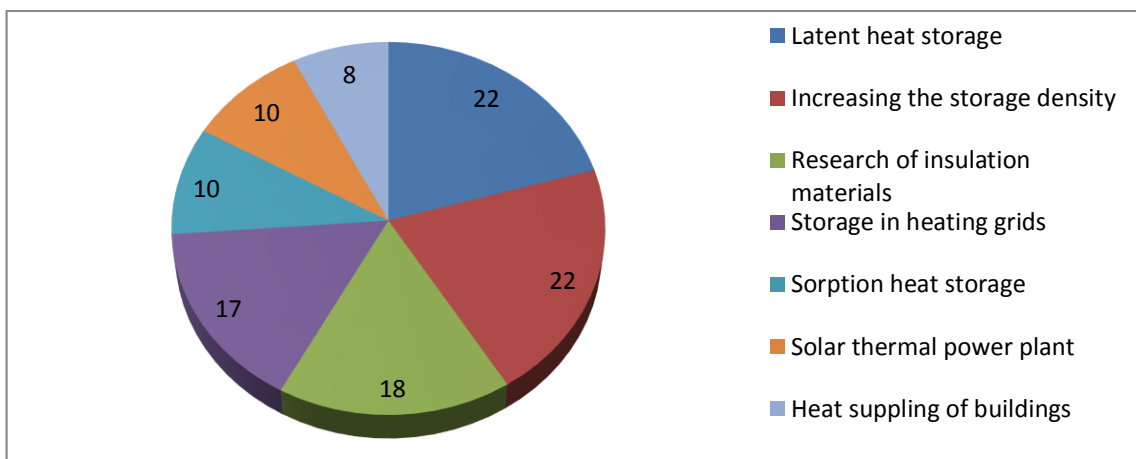
Figure 12: Research projects dealing with TES in Germany (June 2013) [10]

2.3. Thermal energy storage in power plants

Benefits and reasons for deployment of the TES in power plants are discussed in more detail in the introduction of the chapter *2. Thermal energy storage*. In short, the TES provides compensation of consumption peaks and allow filling the valleys in consumption. Additionally they are improving the response time to changes in demand of power plant.

The TES in lignite power plants was used for a quite long time. Deployment in the past was necessary for the steam generator output power correction. This need was partially solved by new steam generators for a milled coal and partly solved by the extension of the distribution grid and increasing consumption of electrical energy, when the power regulation was not so much needed. Renewed interest in TES in coal power plants is mainly due to the effort to save gas and the oil, because these fuels were formerly used to cover peak loads. Another economic reason is maximum utilization of

the power plant, because operations of the environmental control equipment such as electrostatic precipitators and flue gas desulphurization plants are expensive. The equipment must be used always and the costs of operating that equipment are fixed.

The usage of TES in nuclear power plants is even more crucial. The first reason is that the nuclear fuel is cheap with respect to the necessary quantities of the fuel. The second reason is the high cost of nuclear power plant construction, therefore, is the pursuit of maximum utilization of power generation and with this is closely related the last reason. Cyclic operation of the reactor damages its fuel and causes greater wear of components; therefore, it is needed to maintain the fixed reactor power near its rated power. In nuclear power plant is usually a separate production circuit and the storage circuit. The main reason is safety. The water in the primary circuit is radioactive and the primary circuit is unsuitable for energy storage. The storage should takes place in secondary water circuit. Storage circuit can use the main turbine or can have its own. Decision depends on the overload capacity of main steam turbine.

The fuel flow cannot be controlled at solar power plant and it is not even possible to store the fuel. TES in this case may partly serve as the output regulator of solar power plant. Solar power plant is dependent on sunlight, which intensity is during the day yields to step changes, for example during increased cloudiness. TES is appropriate to cover short-term climatic changes in order to avoid sudden and unattached changes in power generation. Opposite problem occurs during very sunny weather in places, where are built a so-called solar farms. There are cases, where it is produced excess of electric energy, which is used for charging of the TES, to avoid overloading of the grid. For solar power plants are used oils, liquid metals or molten salt as transport media. Exactly the same media are used as storage media. Therefore, it is common to use for solar power plants the same medium for transport and storage of energy. This can simplify the technology of the TES.

Among the requirements on power plants certainly include high process efficiency and eco-plant operation. Together with the development of industry and the increasing standard of living is increasing the power consumption. It is a growing need to regulate the grid feeding for example due to cyclic renewables resources. Thermal energy storage is one way to meet these requirements. It is possible to increase the energy efficiency of thermal power plant and also fix problems with the peaks and the valleys in electricity consumption, as well as overall power control. [6][7][8]

3. Mathematical model of thermal power plant

The first chapter of the practical part of thesis deals with the description of used model and basic calculations of thermal cycle process. As the reference model the model of test facilities THERESA is used.

3.1. Calculations of thermal cycle process

Thermal process is defined as the change of condition in real thermal or thermodynamic system due to changing the internal energy and characterized by non-uniformity of the temperature field in the system. Thermodynamic system can be described by the state variables (temperature, volume, pressure, mass, etc.) or state functions (internal energy, enthalpy, entropy). Heat transfer is possible through conduction, convection or radiation and be either stochastic or deterministic.

If thermodynamic condition of fluids passes through several changes and eventually returns to its original condition, then the system is circular and his PV (pressure-volume) and T-s (temperature-entropy) diagrams forms closed cycles. Thus defined cycle can be reversible or irreversible. The ideal and reversible cycle with maximum efficiency is called Carnot cycle. The pV, T-s diagrams of Carnot cycle are shown in Figure 13 and in Figure 14.
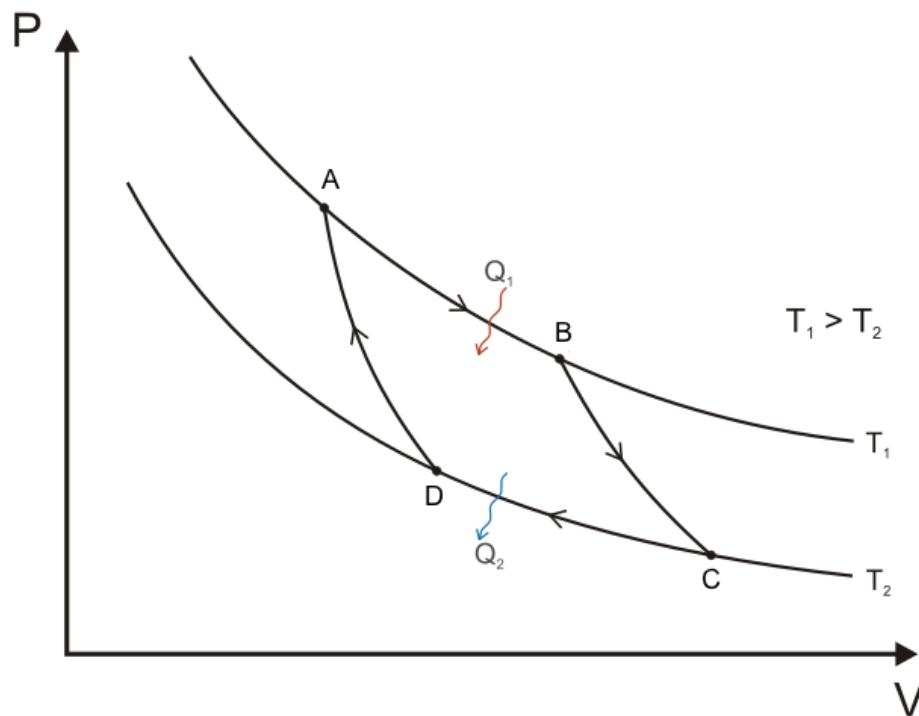
Figure 13: pV diagram of Carnot cycle [30]
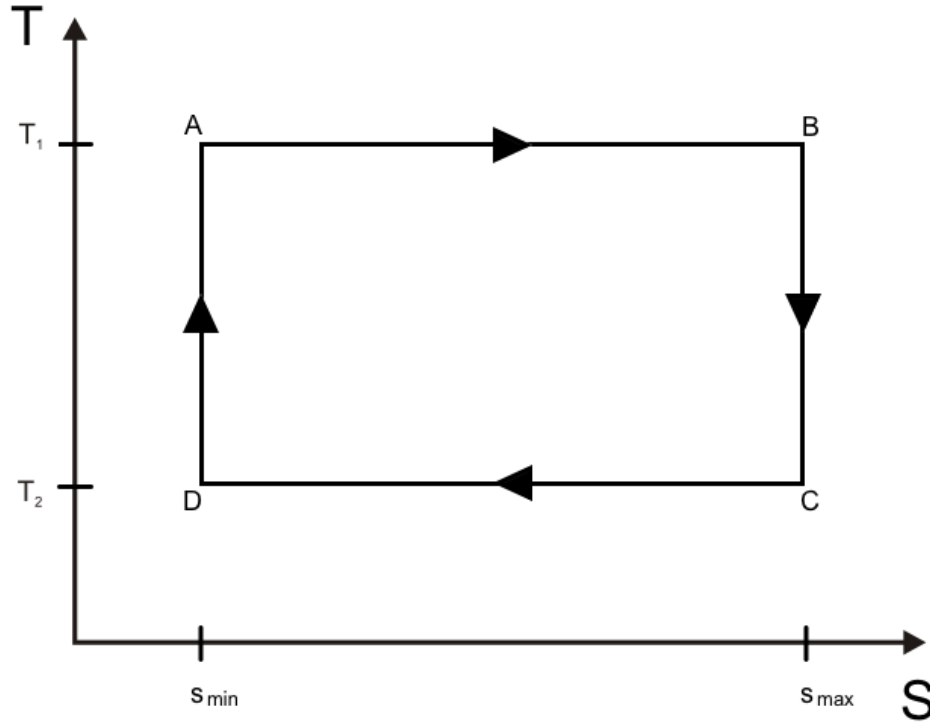
Figure 14: T-s diagram of Carnot cycle

Carnot cycle consists of two isothermal and two adiabatic processes. During isothermal expansion of A - B working medium receives heat Q_1 from the heater with constant temperature T_1. In the isothermal compression C - D working substance gives heat Q_2 to cooler with constant temperature T_2, where the temperature T_2 is lower than temperature T_1. The thermal efficiency is possible to calculate like

$$\eta_{CARNOT} = \frac{W}{Q_1} = \frac{Q_1 - Q_2}{Q_1} \tag{2}$$

where W is the total work executed during the cycle system. Received and disposed of heat can be expressed in relation to temperatures as

$$Q_1 = T_1 \cdot (s_{max} - s_{min}) = T_1 \cdot \Delta s \tag{3}$$

$$Q_2 = T_2 \cdot (s_{max} - s_{min}) = T_2 \cdot \Delta s \tag{4}$$

where s_{max} is the maximum system entropy, s_{min} is the minimum system entropy and Δs is the difference between them. Efficiency of Carnot cycle can be then expressed as

$$\eta_{CARNOT} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1} \tag{5}$$

On the other side the irreversible processes are those processes, which work without external action in one direction only. It means that the original state cannot be achieved using the same procedure in reverse order. It is necessary to supply some energy, which doesn't belong to given system, to achieve the original state. All real processes in nature are almost exclusively irreversible. Carnot cycle is therefore only an intellectual structure. [11]

The probably most significant state function is enthalpy, which is one of the four fundamental thermodynamic potentials and its value expresses the energy stored in a thermodynamic system. Enthalpy is defined as

$$H = U + V \cdot p \tag{6}$$

where U is the internal energy of the system, p is the pressure and V is volume of the system.

The specific enthalpy of a uniform system is defined like

$$h = \frac{H}{m} \tag{7}$$

$$h = u + v \cdot p \tag{8}$$

where m is the mass of the system, u is the specific internal energy, p is the pressure, and v is specific volume, which is equal to inverted density. [12]

For the calculation of power plants properties, where water and steam are used as a working medium, the standard IAPWS-IF97 Industrial Formulation for Thermodynamic Properties of Water and Steam must be used. The norm was defined in 1997 by the International Association for the Properties of Water and Steam and this norm replaced the older IFC-67 norm. By this norm the phase (pressure-temperature) diagram of water is divided into the five regions as is shown in Figure 15. For the needs of THERESA facility those three regions are valid:

   a) region 1 – the liquid state from low to high pressures,
   b) region 2 – the vapor and ideal gas state,
   c) region 4 – the saturation curve (vapor-liquid equilibrium).
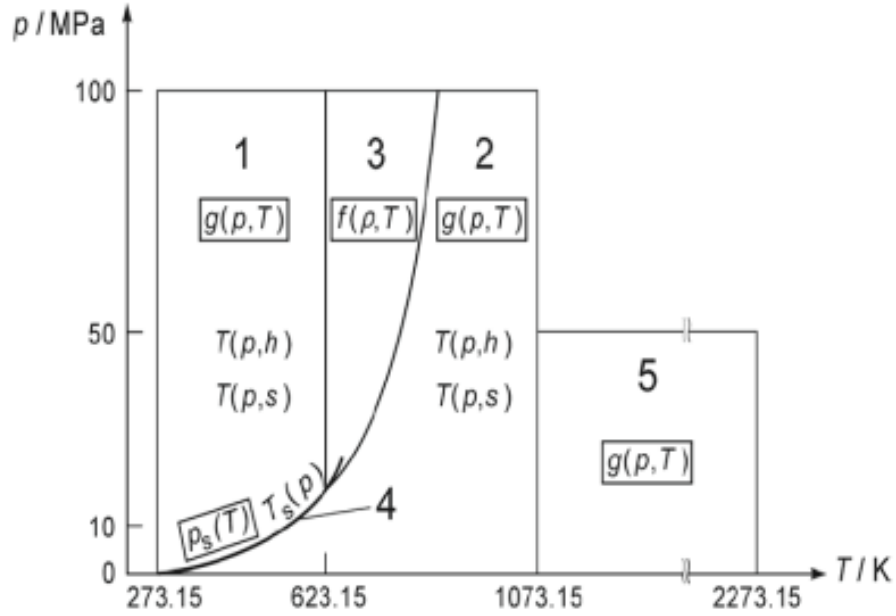
Figure 15: Regions of IAPWS-IF97 norm [31]

The starting point is the specific enthalpy calculation for medium in the region 1. All thermodynamics properties are possible to calculate from a fundamental equation for the specific Gibbs free energy g, whose dimensionless form is expressed as

$$\frac{g(p,T)}{R \cdot T} = \gamma(\pi,\tau) = \sum_{i=1}^{34} n_i (7,1 - \pi)^{I_i} \cdot (\tau - 1,222)^{J_i} \tag{9}$$

where R is the specific gas constant and for ordinary water is equal to

$$R = 0,461526 \; kJ \cdot kg^{-1} \cdot K^{-1} \tag{10}$$

and where π is reduced pressure and τ is reversed reduced temperature

$$\pi = \frac{p}{p^*} \;, with \; p^* = 16,53 \; MPa \tag{11}$$

$$\tau = \frac{T^*}{T} \;, with \; T^* = 1386 \; K \tag{12}$$

The values of coefficients n_i and exponents I_i and J_i are listed in Appendix B.

The specific enthalpy of liquid water is after that calculated by the formula

$$h(\pi,\tau) = R \cdot T \cdot \tau \cdot \gamma_\tau \tag{13}$$

where

$$\gamma_\tau = \sum_{i=1}^{34} n_i (7,1 - \pi)^{I_i} \cdot J_i (\tau - 1,222)^{J_i-1} \tag{14}$$

The second point is calculation of specific enthalpy of the steam with respect to region 2 of IAPWS-IF97 norm. The calculation is again based the fundamental equation for the specific Gibbs free energy g. Equation is separated into part for ideal gas and residual part and is expressed in dimensionless as

$$\frac{g(p,T)}{R \cdot T} = \gamma(\pi, \tau) = \gamma^o(\pi, \tau) + \gamma^r(\pi, \tau) \tag{15}$$

where R is given by equation 10. The equation for ideal-gas part is

$$\gamma^o = \ln \pi + \sum_{i=1}^{9} n_i^o \cdot \tau^{J_i^o} \tag{16}$$

and the equation for residual part is

$$\gamma^r = \sum_{i=1}^{43} n_i \cdot \pi^{I_i} (\tau - 0,5)^{J_i} \tag{17}$$

for both equations applies that

$$\pi = \frac{p}{p^*} \text{, with } p^* = 1 \, MPa \tag{18}$$

$$\tau = \frac{T^*}{T} \text{, with } T^* = 540 \, K \tag{19}$$

The values of coefficients n_i^o and exponents J_i^o are listed in Appendix C, where are also listed the values for coefficients n_i and exponents I_i and J_i.

The specific enthalpy of steam is after that calculated by the formula

$$h(\pi, \tau) = R \cdot T \cdot \tau(\gamma_\tau^o + \gamma_\tau^r) \tag{20}$$

where

$$\gamma_\tau^o = 0 + \sum_{i=1}^{9} n_i^o \cdot J_i^o \cdot \tau^{J_i^o - 1} \tag{21}$$

$$\gamma_\tau^r = \sum_{i=1}^{43} n_i \cdot \pi^{I_i} \cdot J_i (\tau - 0,5)^{J_i - 1} \tag{22}$$

All the above mentioned formulas are included in the documentation for the IAPWS-IF97 norm [13]. Due to the complexity of the calculation of specific enthalpy is in the thesis the specific enthalpy calculated by the Mathcad program with FluidMAT library. This library has been developed to calculate thermodynamics properties with respect to IAPWS-IF97 norm in regions 1, 2 and 4 (wet steam region). Input values for library subroutines are temperature T, pressure p and vapor fraction x, which represents ratio between defined mass of saturated steam and mass of wet steam. All functions, which can be calculated with use of FluidMAT library, are listed in Table 1.

Table 1: List of FluidMAT functions [14]

Function	Thermodynamics property	Unit of result	Function name
$c_p = f(p, T, x)$	Specific isobaric heat capacity	kJ/kg·K	cp_pTx_97
$\eta = f(p, T, x)$	Dynamic viscosity	kg/m·s	eta_pTx_97
$h = f(p, T, x)$	Specific enthalpy	kJ/kg	h_pTx_97
$\lambda = f(p, T, x)$	Thermal conductivity	W/m·K	lambda_pTx_97
$p_s = f(T)$	Saturation pressure from temperature	Mpa	ps_T_97
$s = f(p, T, x)$	Specific entropy	kJ/kg·K	s_pTx_97
$T_s = f(p)$	Saturation temperature from pressure	K	Ts_p_97
$v = f(p, T, x)$	Specific volume	m³/kg	v_pTx_97

3.2. Description of THERESA

THERESA (<u>Ther</u>mische <u>E</u>nergie<u>s</u>peicher<u>a</u>nlage – Thermal Energy Storage Facility) is test facility located in Zittau, Germany. The main purposes of this facility are simulations and analyses of processes, which are relevant for thermal power plants. One of the main tasks is simulation of thermal energy storage system and subsequent integration of thermal energy storage into the real power plant. Test facility THERESA allows simulating power plant processes up to 350 °C and 160 bars with a maximum mass flow 0,1 kg/s of steam and 0,5 kg/s of water.
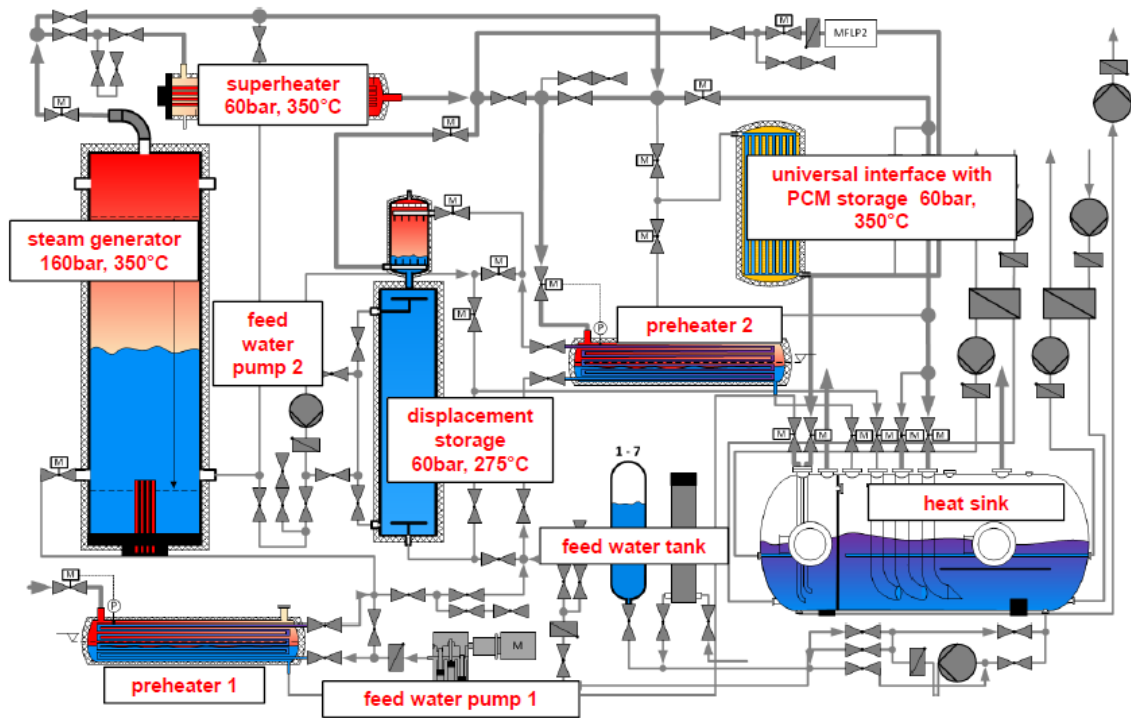
Figure 16: Scheme of test facility THERESA [15]

Scheme of test facility THERESA is shown in Figure 16. The facility is divided into four subsystems according to their primary functions. Pressure vessel subsystem contains feed water pump 1, preheater 1, steam generator and superheater. The input medium to the subsystem is water from feed water tank. Feed water pump 1 increases the water pressure to the working (system) pressure. This water is preheated and then converted to the steam. The saturated steam, which flows from the steam generator, is further converted to the dry steam by superheater. The output medium from pressure vessel subsystem is dry steam.

The second subsystem is storage subsystem. This subsystem contains universal interface, preheater 2, displacement storage and feed water pump 2 in form of circulation pump. Universal interface can be used for connecting of the different investigated elements into the test facility. The interface can have on its output water or steam with values up to the nominal values of temperature and pressure (350 °C and 160 bars).

Third subsystem is cooling subsystem, which unites heat sink and its auxiliary circuits. The subsystem ensures the function of the heat sink. Last subsystem is feed water subsystem. The only task of this subsystem is to supply test facility by deionized water with constant temperature. [15]

3.3. Mathematical model of THERESA

Test facility THERESA is able to work in several different operation modes. The aim of this thesis is to describe modes, which operates with the thermal energy storage. These selected modes can be split into two groups. The first group of modes serves for loading of the TES and second group serves for unloading of the TES. In Table 2 are listed and shortly described all selected modes.

Table 2: Modes of THERESA

	Mode name	Mode description
Loading of TES	FW1	loading by hot water from the steam generator
	FW8	loading by superheated steam; the cold water is taken from TES, heated in preheater 2 and returned to TES - cyclic mode
	FW9	loading by hot water from preheater 2, which is fed by superheated steam; a part of the water from TES feeds the steam generator; other part is for the loading of TES
	FW10	loading by hot water from mixing preheater; where the hot water from preheater 2 and superheated steam are mixed; a part of the water from TES feeds the steam generator; other part is for the loading of TES
Unloading of TES	FW2	unloading by water from preheater 1; the heated water flows to the heat sink 1
	FW11	unloading by water from preheater 1; the heated water is used to feed the steam generator and the steam after superheating flows to the heat sink 1, where condenses

Mode FW10 is the most complex mode of THERESA facility, which combines all loading modes into one mode and which is simplification of the real power plant. Calculations of the parameters of the thermal cycle will be therefore explained on this mode. Connection scheme is shown in Figure 17. The input water is first heated by preheater 1 to the temperature 176 °C and then heated by preheater 2 to the temperature 265 °C. Water is mixed with steam of temperature 350 °C in mixing preheater. Part of the mixed water supplies the steam generator, which produced the saturated steam. Saturated steam is converted to the superheated steam by superheater. At the output of superheater is steam with temperate 350 °C, which is splitted to the three pipes. Part of

steam serves as a supply for the preheater 2, part flows to mixing preheater and small part flows directly to the heat sink 1. To the heat sink 1 also flows the cold water from the TES with temperature 176 °C and the condensed water from the preheater 2. Water inside heat sink 1 is cooled to temperature 90 °C and the water in second heat sink is also cooled to 90 °C by heat sink 2. In next subchapters are shown the calculations of THERESA thermal cycle, which are divided by the individual components.
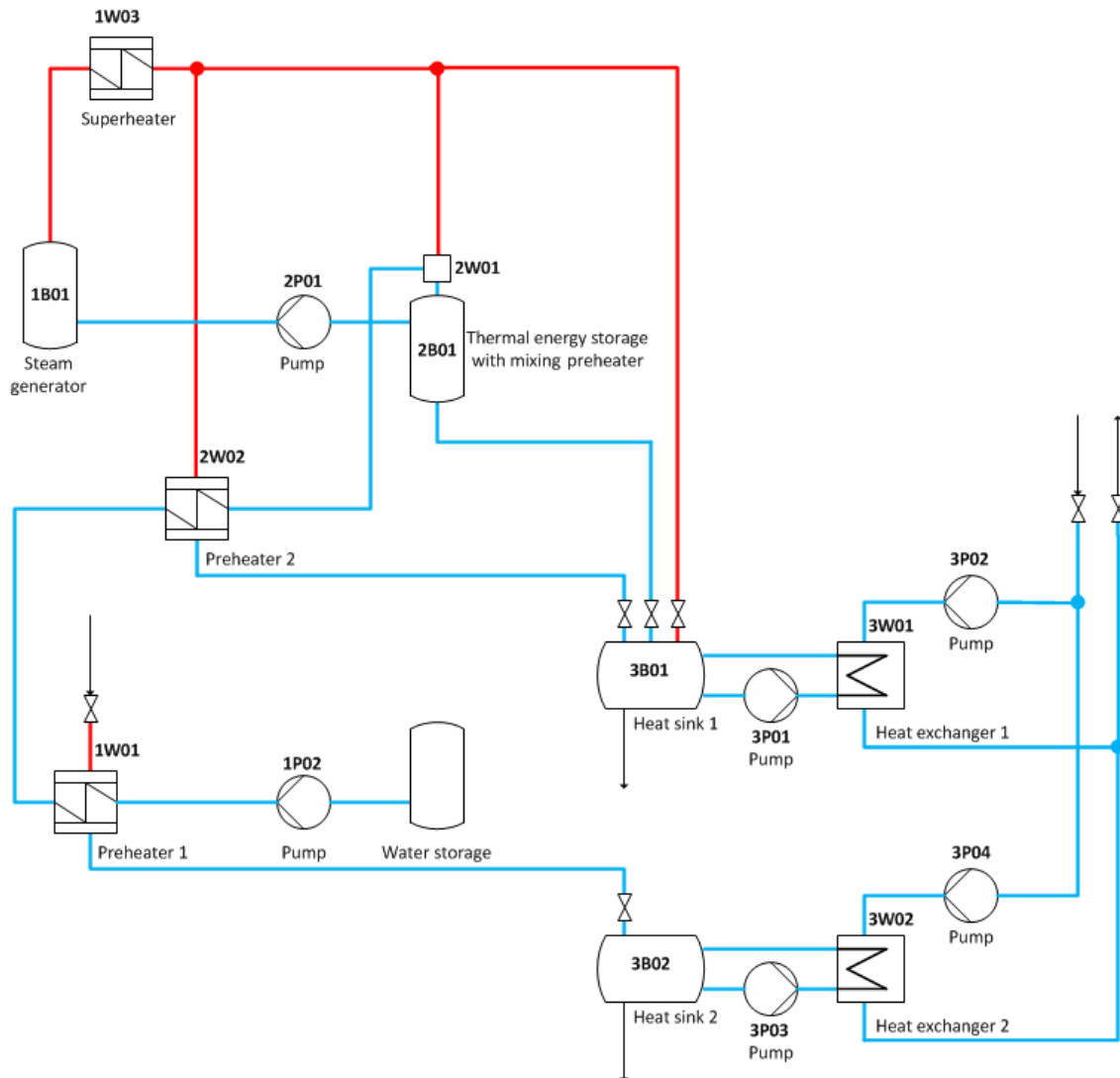
Figure 17: Test facility connected to mode FW10

3.3.1. Preheater 1 (1W01)

Preheater is device with primary and secondary circuits. Secondary circuit is source of thermal energy, which is given to medium in primary circuit. In this particular case, the input to the secondary circuit is steam with temperature 190 °C and pressure 11,5 bars. The primary circuit's input is water with the temperature 5 °C, which is

heated up by preheater to the temperature 176 °C. Output from secondary circuit is condensed steam. The pressures between input and output in both circuits after the energy exchange are the same.

From the knowledge of all input and output temperatures can be calculated the specific enthalpy of all preheater's inputs and outputs. Heat transferred by the preheater from second circuit to primary circuit is defined like

$$Q_{1W01} = \dot{m}_{f\_prim} \cdot \frac{h_{prim\_out} - h_{prim\_in}}{\eta} \tag{23}$$

where $\dot{m}_{f\_prim}$ is primary circuit mass flow, h_{prim_out} is primary circuit output specific enthalpy, h_{prim_in} is primary circuit input specific enthalpy and η is transmission efficiency, which is in our case equal to 1.

Formula for calculation of necessary secondary circuit's mass flow is

$$\dot{m}_{f\_sec} = \dot{m}_{f\_prim} \cdot \frac{h_{prim\_out} - h_{prim\_in}}{\eta(h_{sec\_in} - h_{sec\_out})} \tag{24}$$

where $\dot{m}_{f\_prim}$ is primary circuit mass flow, h_{prim_out} is primary circuit output specific enthalpy, h_{prim_in} is primary circuit input specific enthalpy, h_{sec_in} is secondary circuit input specific enthalpy, h_{sec_out} is secondary circuit output specific enthalpy and η is transmission efficiency, which is again equal to 1.

In Table 3 are shown for comparison values for the preheater 1 calculated using the procedure described in chapter *3.1. Calculations of thermal cycle process* and values calculated in Mathcad. Mathcad calculations are shown in Appendix D and all given and calculated input and output values related to preheater 1 are shown in Figure 18.

Table 3: Comparison of preheater's 1 values

	Primary circuit		Secondary circuit	
	Input	Output	Input	Output
Medium	water	water	steam	water
Temperature T [°C]	5	176	190	186,05
Pressure p [bar]	60	60	11,5	11,5
Specific enthalpy h [kJ/kg] (calculated by IAPWS-IF97 formulas)	26,968	748,228	2793,096	789,986
Specific enthalpy h [kJ/kg] (calculated by Mathcad)	26,968	748,228	2793	789,988

$$p_{sec} = 11.5\,\text{bar}$$

$$T_{sec\_in} = 190\,°C$$

$$h_{sec\_in} = 2.793 \times 10^3\, \frac{kJ}{kg}$$

$$m_{f\_sec} = 0.18\, \frac{kg}{s}$$

$$T_{prim\_out} = 176\,°C$$

$$h_{prim\_out} = 748.228\, \frac{kJ}{kg}$$

$$p_{prim} = 60\,\text{bar}$$

$$T_{prim\_in} = 5\,°C$$

$$h_{prim\_in} = 26.968\, \frac{kJ}{kg}$$

$$m_{f\_prim} = 0.5\, \frac{kg}{s}$$

1W01

$$Q_{1W01} = 360.63\,\text{kW}$$

$$T_{sec\_out} = 186.05\,°C$$

$$h_{sec\_out} = 789.988\, \frac{kJ}{kg}$$
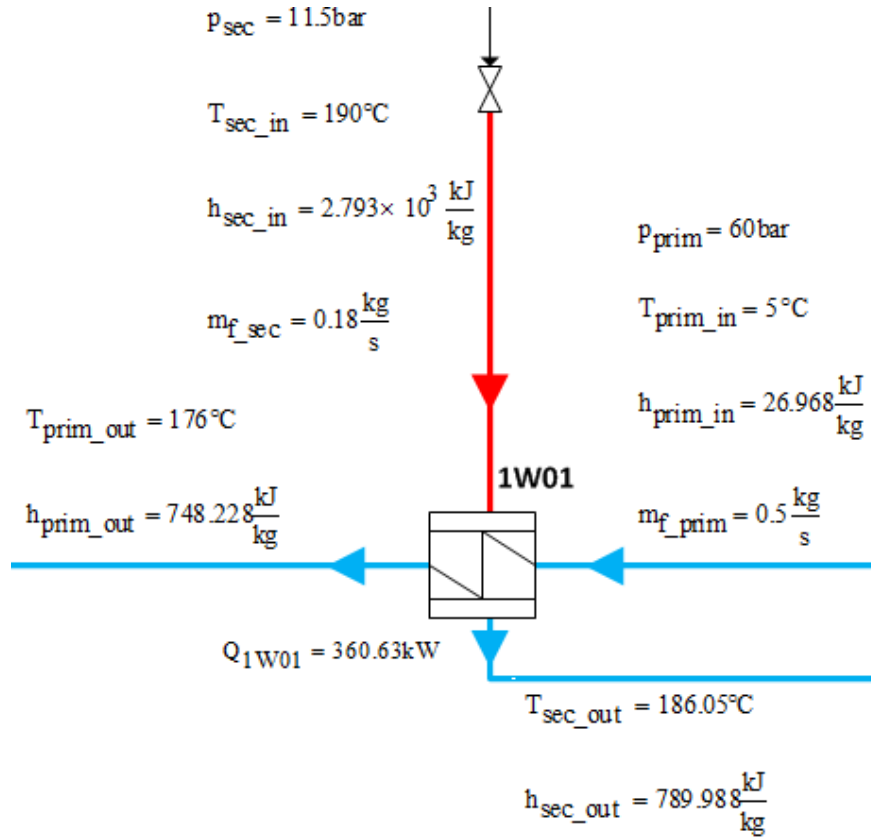
Figure 18: Values of preheater 1

3.3.2. Steam generator (1B01)

Within the steam generator enters part of the saturated water from mixing preheater, which temperature is given by the saturation temperature of water at pressure 60 bars. Steam generator has a two operation modes. In the first mode converts input water into saturated steam. In the second mode the input water is only heated up to higher temperature. In case of FW10 mode the first case is used. The generator in THERESA facility is powered from the mains. Generator power is calculated as

$$P = \eta \cdot P_{max} \tag{25}$$

where η is efficiency equal to 1 and P_{max} in maximum power of generator.

For the calculations it is assumed that the temperature of the saturated output steam is the same as the temperature of the saturated input water from mixing preheater and all delivered energy is used for phase-change conversion.

Output mass flow from steam generator is defined like

$$m_{f\_out} = \frac{P}{h_{out} - h_{in}} \tag{26}$$

41

where P is generator power, h_{in} is input specific enthalpy and h_{out} is specific enthalpy of output.

All given and calculated input and output values related to steam generator are shown in Figure 19 and calculations are shown in Appendix E.

$$T_{out} = 275.586°C$$

$$h_{out} = 2.785 \times 10^3 \frac{kJ}{kg}$$

$$p = 60 \, bar$$

$$T_{in} = 275.586°C$$

$$m_{f\_out} = 0.127 \frac{kg}{s}$$

$$h_{in} = 1.214 \times 10^3 \frac{kJ}{kg}$$

1B01

$$m_{f\_in} = 0.127 \frac{kg}{s}$$

$$P = 200 \, kW$$

Figure 19: Values of steam generator

3.3.3. Superheater (1W03)

In superheater the saturated steam from the steam generator is converted to the superheated steam. Superheater is powered from the electrical mains and steam on its output has the temperature 350 °C. Calculations are shown in Appendix F and all values are listed in Figure 20.

$$h_{out} = 3.044 \times 10^3 \frac{kJ}{kg}$$

1W03

$$m_{f\_out} = 0.127 \frac{kg}{s}$$

$$p = 60 \, bar$$

$$T_{in} = 275.586°C$$

$$P_{1W03} = 32.931 \, kW$$

$$h_{in} = 2.785 \times 10^3 \frac{kJ}{kg}$$

$$m_{f\_in} = 0.127 \frac{kg}{s}$$
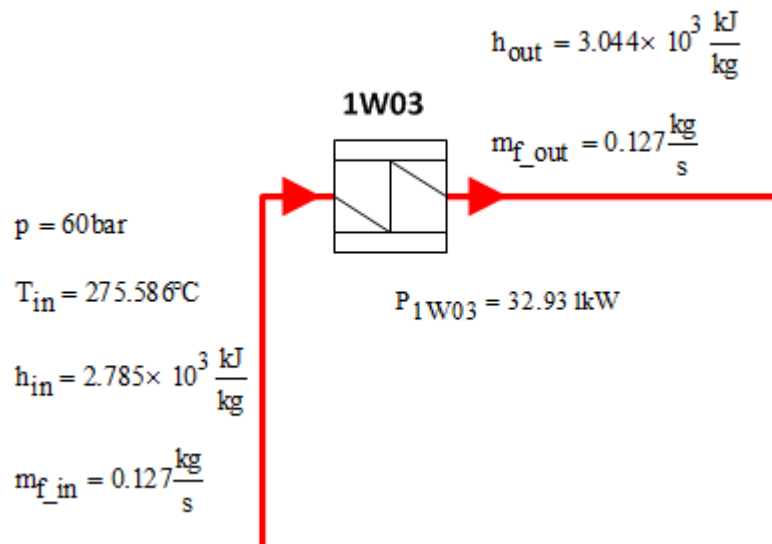
Figure 20: Values of superheater

3.3.4. Preheater 2 (2W02)

The input to the secondary circuit is the superheated steam with temperature 350 °C and pressure 60 bars. To primary circuit flows the water from preheater 1 with the temperature 176 °C, which is heated up to the temperature 265 °C. Calculations are shown in Appendix G and all given and calculated values related to the preheater 2 are shown in Figure 21.

$$p_{sec} = 60\,bar$$

$$T_{sec\_in} = 350\,°C$$

$$h_{sec\_in} = 3.044 \times 10^3 \frac{kJ}{kg}$$

$$p_{prim} = 60\,bar \qquad m_{f\_sec} = 0.112 \frac{kg}{s}$$

$$T_{prim\_in} = 176°C$$

$$h_{prim\_in} = 748.228 \frac{kJ}{kg} \qquad T_{prim\_out} = 265°C$$

2W02

$$m_{f\_prim} = 0.5 \frac{kg}{s} \qquad h_{prim\_out} = 1.16 \times 10^3 \frac{kJ}{kg}$$

$$Q_{2W02} = 205.677 kW$$

$$T_{sec\_out} = 275.586°C$$

$$h_{sec\_out} = 1.214 \times 10^3 \frac{kJ}{kg}$$
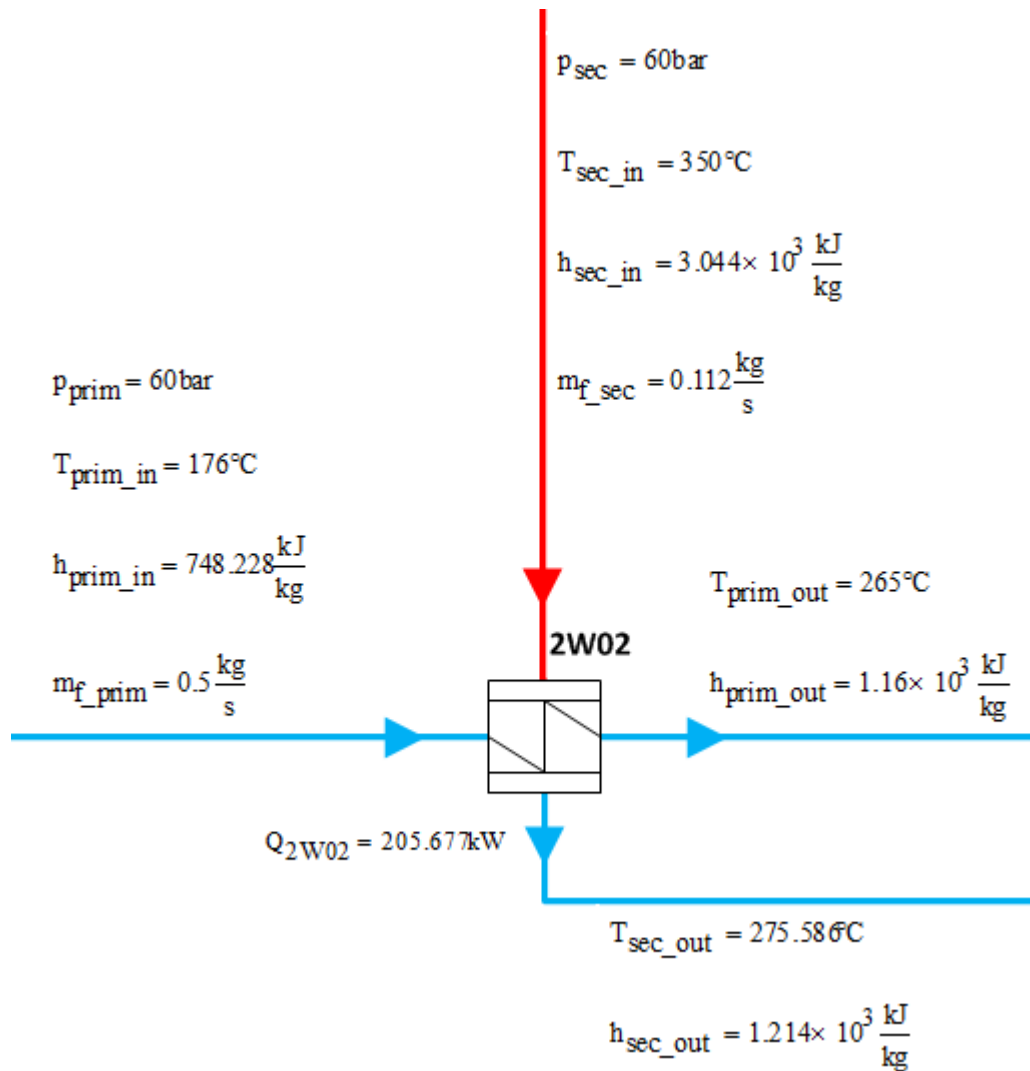
Figure 21: Values of preheater 2

3.3.5. Mixing preheater (2W01)

In mixing preheater the saturated steam from the superheater and the hot water from the preheater 2 are mixed together. The output temperature of the water is given by the saturation temperature of water at pressure 60 bars. The values of mixing preheater are shown in Figure 22 and calculations are listed in Appendix H.

$T_{steam\_in} = 350\,°C$

$h_{steam\_in} = 3.044 \times 10^3 \dfrac{kJ}{kg}$

$m_{f\_steam\_in} = 0.015 \dfrac{kg}{s}$

2W01

$p = 60\,bar$

$T_{water\_in} = 265\,°C$

$h_{water\_in} = 1.16 \times 10^3 \dfrac{kJ}{kg}$

$m_{f\_water\_in} = 0.5 \dfrac{kg}{s}$

$T_{water\_out} = 275.586\,°C$

$h_{water\_out} = 1.214 \times 10^3 \dfrac{kJ}{kg}$

$m_{f\_water\_out} = 0.515 \dfrac{kg}{s}$
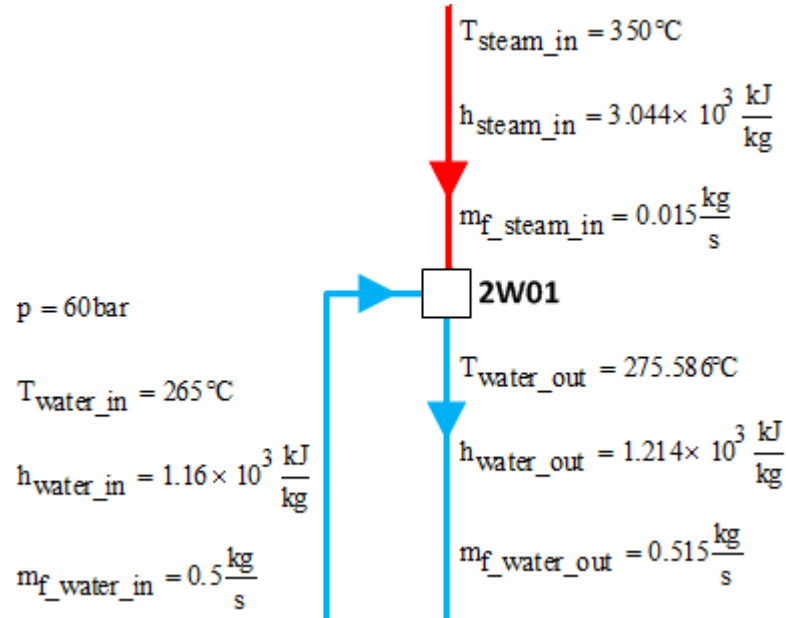
Figure 22: Values of mixing preheater

3.3.6. Thermal energy storage (2B01)

From knowledge of input and output temperature the input and output enthalpy can be calculated. The mass flow is the same at the input and output site so the power, which is taken by TES for charging, can be calculated by the use of modified formula 26.

$$P_{TES} = m_{f\_in}(h_{out} - h_{in}) \tag{27}$$

All given and calculated input and output values related to TES are shown in Figure 23. There are additional values, which can be calculated, but for the needs of FW10 mode only the necessary values are calculated. The calculation procedures are shown in Appendix I and resulting values are shown in Figure 23.

$p = 60 \, bar$

$T_{in} = 275.586°C$

$h_{in} = 1.214 \times 10^3 \, \frac{kJ}{kg}$

$m_{f\_in} = 0.515 \frac{kg}{s}$

2B01

$T_{out\_gen} = 275.586°C$

$h_{out\_gen} = 1.214 \times 10^3 \, \frac{kJ}{kg}$

$m_{f\_out\_gen} = 0.127 \frac{kg}{s}$

$P_{2B01} = 180.369 kW$

$T_{out} = 176°C$

$h_{out} = 748.228 \frac{kJ}{kg}$

$m_{f\_out} = 0.387 \frac{kg}{s}$
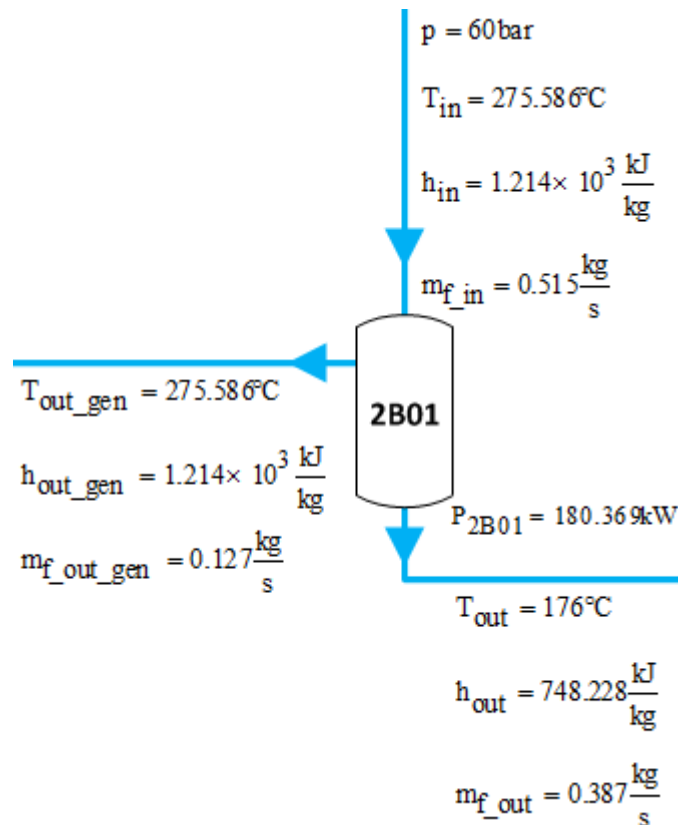
Figure 23: Values of thermal energy storage

3.3.7. Heat sink 1 (3B01)

Heat sink 1 provides cooling of water from the TES and preheater 2 and cooling of residual steam from superheater. Before water and steam enter to the heat sink, their pressure is reduced to 1,013 bars by the valve. During this change the enthalpy is the same and the temperature and pressure of the water are reduced.

Heat sink consists of two main parts. The first part is vessel with several inputs and outputs. The output mass flow is equal to sum of all input mass flows to maintain the balance. The second part is cooling part, which connects the vessel with cooling circuit. The maximum temperature inside the vessel is given and it's equal to 90 °C. All other heat above this value must be dissipated by cooling circuit.

Heat exchanger is used for cooling, where the water from vessel is in secondary circuit and heats up the water in primary circuit. Into primary circuit flows water with the temperature 70 °C and pressure 11,5 bars and the required temperature at the output is 85 °C with pressure 11,5 bars. From these values the necessary mass flows through primary and secondary circuit for heating water can be calculated. All given and

calculated input and output values related to heat sink 1 are shown in Figure 24 and calculations are listed in Appendix J.

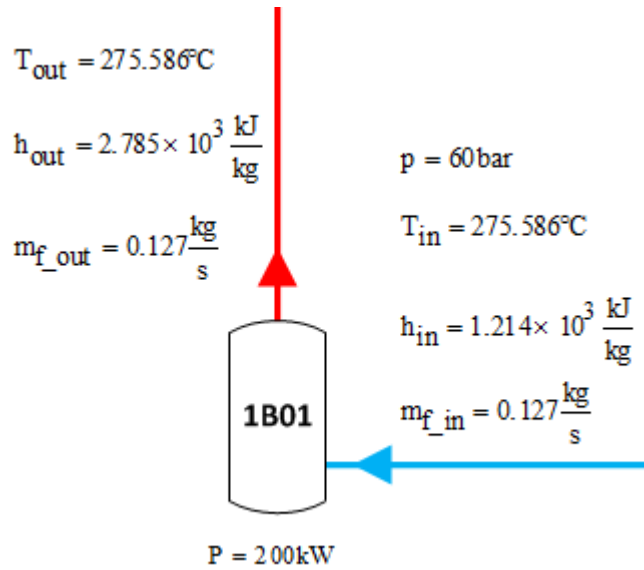$T_{in\_superheater} = 350 \,^{\circ}C$

$h_{in\_superheater} = 3.044 \times 10^3 \,\frac{kJ}{kg}$

$m_{f\_in\_superheater} = 1.44 \times 10^{-4} \,\frac{kg}{s}$

$T_{in\_TES} = 176 \,^{\circ}C$

$h_{in\_TES} = 748.228 \,\frac{kJ}{kg}$

$m_{f\_in\_TES} = 0.387 \,\frac{kg}{s}$

$T_{in\_preheater2} = 275.586 \,^{\circ}C$

$h_{in\_preheater2} = 1.214 \times 10^3 \,\frac{kJ}{kg}$

$m_{f\_in\_preheater2} = 0.112 \,\frac{kg}{s}$

$P_{primary\_in} = 6.5 \,bar$

$T_{primary\_in} = 70 \,^{\circ}C$

$P_{primary\_out} = 11.5 \,bar$

$T_{primary\_out} = 85 \,^{\circ}C$

3P02

3W01

$h_{primary\_in} = 293.933 \,\frac{kJ}{kg}$

3B01

3P01

$P_{secondary\_in} = 1.013 \,bar$

$T_{secondary\_in} = 90 \,^{\circ}C$

$h_{secondary\_in} = 376.993 \,\frac{kJ}{kg}$

$m_{f\_secondary} = 3.787 \,\frac{kg}{s}$

$h_{primary\_out} = 356.803 \,\frac{kJ}{kg}$

$m_{f\_primary} = 3.787 \,\frac{kg}{s}$

Figure 24: Values of heat sink 1

3.3.8. Heat sink 2 (3B02)

The heat sink 2 works on the same principle as heat sink 1 with the difference that provides cooling of condensed steam from preheater 1. Calculations of heat sink 2 values are shown in Appendix K and all values related to heat sink 2 are shown in Figure 25.
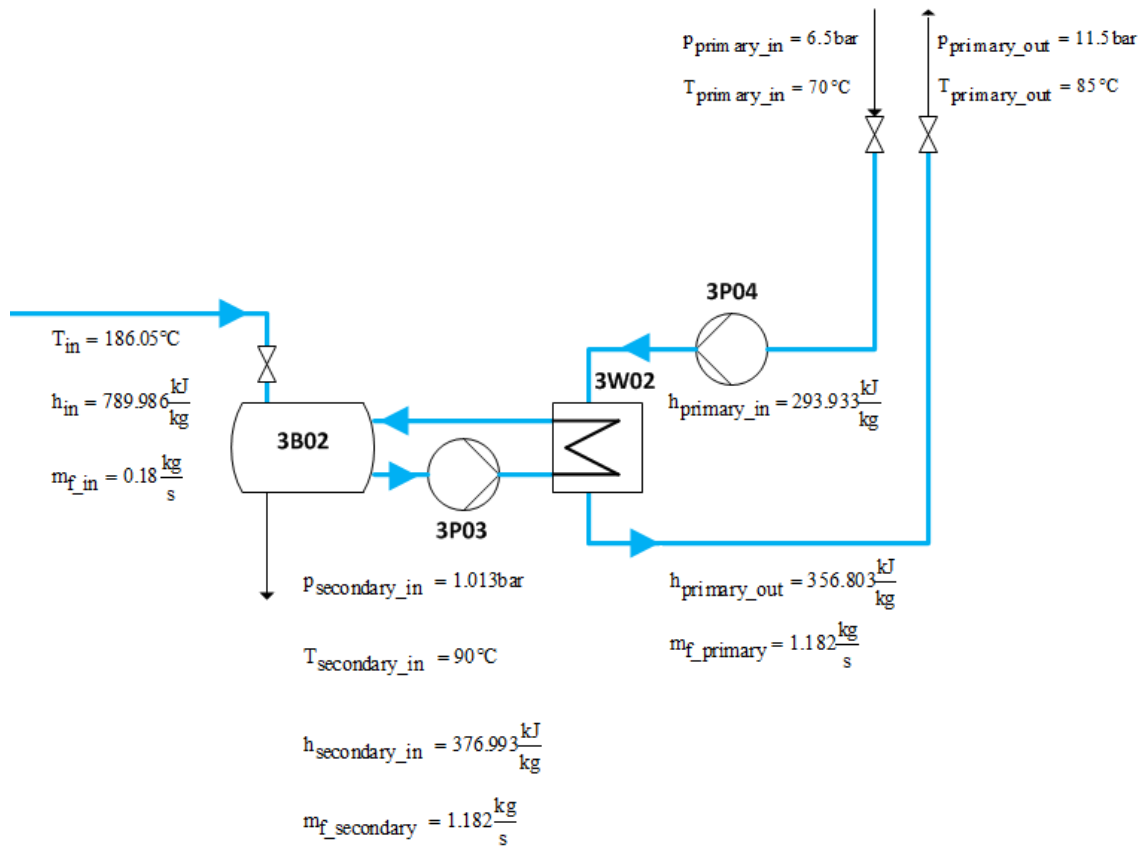
Figure 25: Values of heat sink 2

By calculation the heat sink 2 parameters are obtained last needed values of mathematical model of THERESA test facility. Mathematical model is derived for ideal static conditions. During calculation is not taken into consideration any pressure or heat loss in the pipes and no pressure losses in the heat exchangers. Of course in the real test facility these losses occur.

4. Simulation of THERESA facility

Simulation and computational program EBSILON Professional (version 9.00) is used for development of simulation model. EBSILON is being developed by German company STEAG Energy Services GmbH. Program is suitable for static engineering and designing energy and power plant systems and allows to simulate any thermodynamic cycle process. Program contains over a hundred different components and about twenty media that can flow through these components.

Components are divided into groups according to their function. For modeling of thermodynamic cycles is possible to use components like throttles, mixers/splitters, steam generators, turbines, pumps, heat exchangers and many more. As the transfer media can be used water, steam, air, oil, coal, gas, 2-phase liquid and others. For needs of this thesis the liquid water and steam will be used as the transfer media and components, which are used for modeling of test facility, are described below.

4.1. Simulation model for FW10 mode

The first part of the simulations is to simulate the individual modes of THERESA facility separately. Mode FW10 is again used as a reference model because of its complexity. Unlike the mathematical model in the simulation model are taken into account the pressure drops in the circuit. Furthermore, the simulation model includes several pumps against the mathematical model. One serves to increase the pressure of the input media to system pressure and others serve as a circulatory pumps in specific parts of the circuit. In next subchapters are described individual components and their settings.

4.1.1. Input part of the circuit

The input values of the circuit are in mathematical model defined after the pump 1P02. For this reason it is assumed that water pressure before pump is 4 bars and the pump serve for increasing the water pressure to the value of system pressure, which is 60 bars in this case. The result is that the water after the pump 1P02 has the pressure of 60 bars, temperature of 5 °C and mass flow of 0,5 kg/s. For the definition of the pressure before pump was used component Nr. 1 – *Boundary value input*. Mass flow, temperature and pressure after pump are defined in component Nr. 33 – *General value input*. For the pump 1P02 is used component Nr. 8 – *Pump* with default

parameters, for which the pressure P_2 is result of the system pressure. This means that the pump increases the pressure P_2 up to the value that is set after the pump. The temperature after pump is defined so the temperature before pump is calculated by EBSILON.

60.000 bar	26.968 kJ/kg
5.000 °C	0.500 kg/s
0.000 -	13.484 kW

3.502 kW

1P02

4.000 bar	19.979 kJ/kg
4.658 °C	0.500 kg/s
0.000 -	9.989 kW
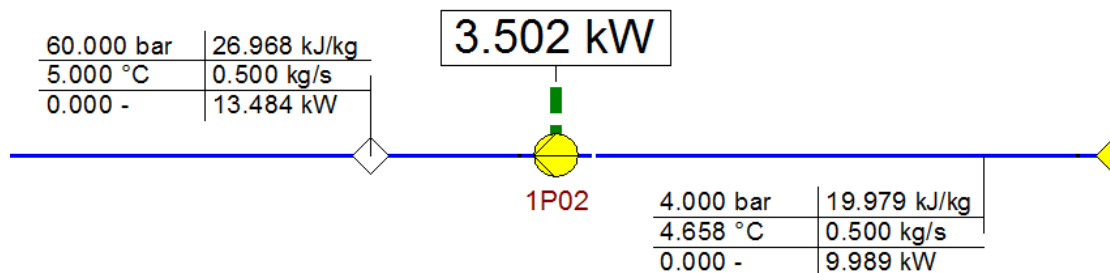
Figure 26: Input part of facility

4.1.2. Preheaters 1 and 2 (1W01 and 2W02)

The water with these parameters flows to the primary circuit of preheater 1. Input to the secondary circuit of the preheater 1 is steam with pressure 11,5 bars and the temperature 190 °C. As the preheater 1 the component Nr. 10 − *Heating condenser* is used. *Upper terminal temperature difference* parameter is set to value 10 K, primary and secondary circuit's pressure drops are set to value 0,05 bars. Values of preheater 1 are shown in Figure 27.

| 11.500 bar |
| 190.000 °C |
| 2793.096 kJ/kg |
| 0.180 kg/s |

59.950 bar		60.000 bar
176.050 °C		5.000 °C
748.445 kJ/kg		26.968 kJ/kg
0.500 kg/s		0.500 kg/s

| 11.450 bar |
| 185.855 °C |
| 789.122 kJ/kg |
| 0.180 kg/s |

Figure 27: Preheater 1

Preheater 2 is again represented by component Nr. 10 − *Heating condenser* with the very same parameters as the preheater 1. The water from primary circuit of

preheater 1 flows to the primary circuit of preheater 2 and it is heated up by the steam from superheater, which condensate in secondary circuit of the preheater 2. Values of preheater 2 are shown in Figure 28.
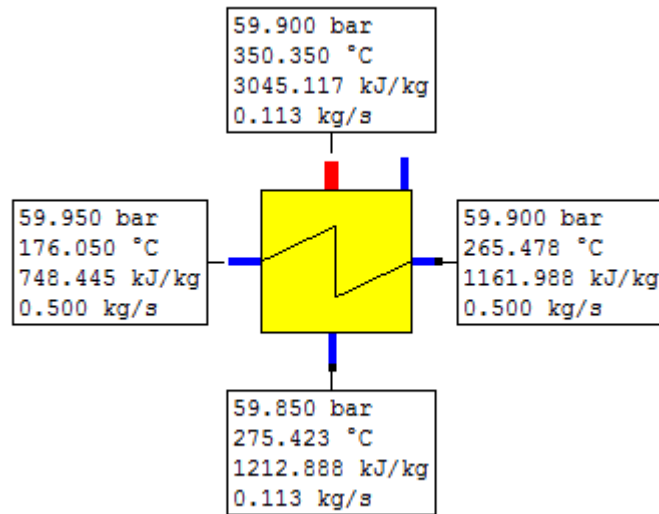
Figure 28: Preheater 2

4.1.3. Mixing preheater (2W01) and thermal energy storage (2B01)

The mixing preheater can be seen as part of thermal energy storage. TES itself operates only with medium in form of the liquid water and mixing preheater is therefore used in cases, if the TES is loaded by the steam. The appearance of TES is shown in Figure 29. T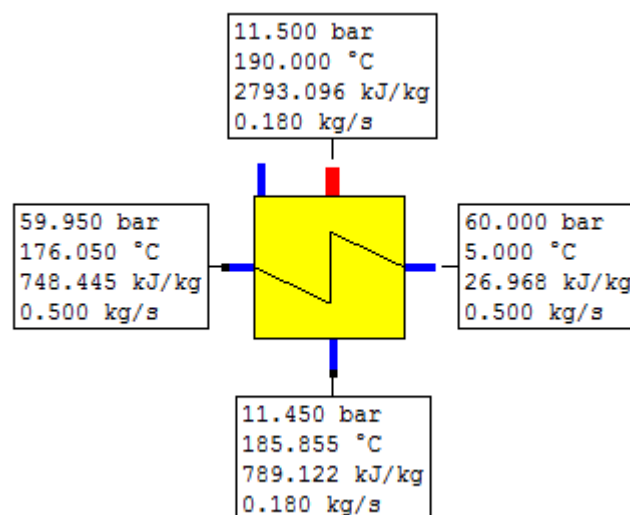ES is complemented with a circulation pump 2P01, which is used for exhausting water from the TES to the steam generator. For the pump 2P01 is used component Nr. 8 – *Pump* with default parameters, where the pressure P_2 is result of the system pressure, which is the same before and after the pump.

The mixing preheater is represented by the component Nr. 9 – *Feed water tank*. Inside the mixing preheater the steam is condensed by mixing with the water. At the output of the component is saturated water. Output water is next divided to two pipes by the splitter (component Nr. 4 – *Mass defined splitter*). Part of the water flows through the pump 2P01 to the steam generator and the part of water flows to TES, which is represented by the component Nr. 28 – *Water tank*.
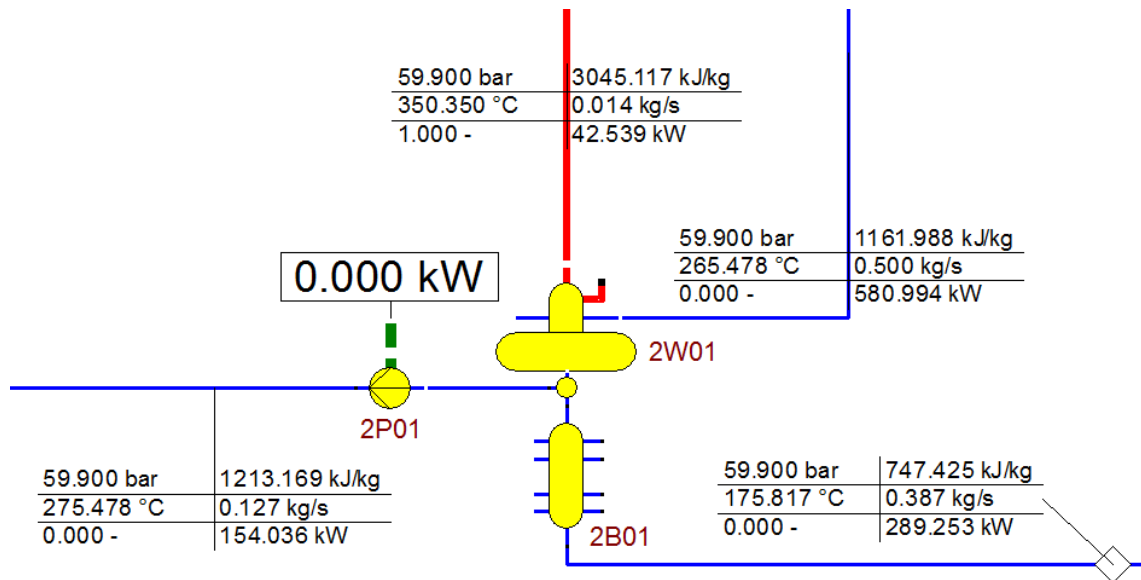
Figure 29: Thermal energy storage part of circuit

Big advantage of this component is that allows user to set value of *Heat loss* parameter. This parameter is used for charging and discharging power of the TES. In case of charging the value of heat loss is positive and represents the endothermic power. This means that there is some loss of energy, which corresponds to charging, because the part of energy is given to medium inside TES during charging. In case of discharging the value of heat loss is negative, represents the exothermic power and component produced additional energy. The behavior is the same like for the real TES. Value, which is used for heat loss, is value, which was calculated by the use of formula 27. All values of input and output media for the mixing preheater and TES are shown in Figure 29.

4.1.4. Steam generator (1B01) and superheater (1W03)

Function of steam generator is covered by the component Nr. 5 – *Steam generator*. Component has the feed water inlet, cold reheat inlet, high pressure steam outlet, hot reheat outlet, auxiliary spray inlets and blow down outlet. The focus is on feed water inlet and high pressure steam outlet. At component properties can be set for high pressure steam outlet values of temperature, pressure and pressure drop. Output temperature is equal to value calculated in Appendix E. Pressure drop is neglected and equal to zero. For the value of output steam pressure there are two ways how to set it. First way is to use value of pressure, which is at the input of steam generator. This value is equal to system pressure minus pressure drops, which occur before steam generator.

This method is used for mode FW10. System pressure is equal to 60 bars and before steam generator are two preheaters, both with pressure drop 0,05 bars. Resulting pressure at the output of steam generator is equal to

$$p_{out} = p_{sys} - \Delta p_{1W01} - \Delta p_{2W02} \tag{28}$$

$$p_{out} = 60 - 0{,}05 - 0{,}05$$

$$p_{out} = 59{,}9 \; bar$$

Second way is to set value of output steam to 60 bars and then delete value of system pressure at the input part of the circuit. This procedure will increase the pressure after the pump 1P02. The pump 1P02 will compensate pressure drops in the preheaters 1 and 2, so that the water at the inlet of the steam generator will have a system pressure 60 bars. Pressure after pump 1P02 will be equal to

$$p_{1P02} = p_{sys} + \Delta p_{1W01} + \Delta p_{2W02} \tag{29}$$

$$p_{1P02} = 60{,}1 \; bar$$

The steam generator settings are shown in Figure 30.

Specification-Values	Charlines	Basic Properties	Fluids	View Properties	Extended Options	SRx

Life steam pressure (nominal)	P2N	59.9		bar
Life steam temperature	T2	275.586		°C
Hot RH temperature	T4	520		°C
HP pressure drop (nominal)	DP12N	0		bar
RH pressure drop (nominal)	DP34N	5		bar
HP-injection mass flow (relative)	M6M1	0		-
Draining mass flow (relative)	M8M1	0		-
ECO pressure drop (nominal)	DPECON	0		bar
Specification	FSPEC	Input P2,T2,T4 as spec. : 0		
Partload pressure drop (reheat)	FVOL	depending on mass and volume		
Calculation mode (design / off-design)	FMODE	GLOBAL : 0		
Handling of spray and blow down flows	FINJ	M6, M7 and M8 given by interna		
Feed water mass flow (nominal)	M1N	0.126969665583355	kg/s	
Spec. volume of feed water (nom.)	V1N	0.001318946778336	m³/kg	
Spec. volume at RH inlet (nominal)	V3N	0.001000206115665	m³/kg	

Figure 30: The steam generator settings

For the superheater the component Nr. 9 – *Feed water tank* is used. This component allows connecting steam on its input (and output). To ensure the function of the superheater is used the same approach as for power of the TES component. The value of heat loss is negative and equal to the value calculated in Appendix F. Additional energy given to the superheater increases the enthalpy (and temperature) of steam. Part of circuit with steam generator and superheater is shown in Figure 31.
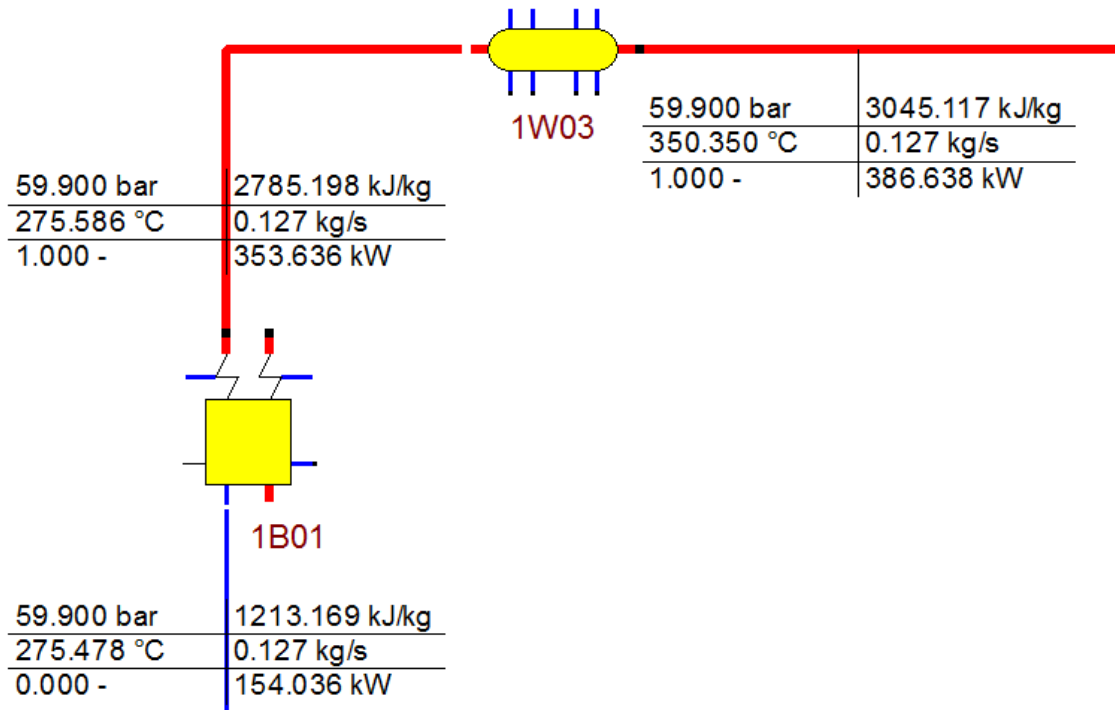
Figure 31: The steam generator and superheater

4.1.5. Heat sink 1 (3B01)

In chapter *3.3.7. Heat sink 1 (3B01)* is described the division of the heat sink into two parts – heat sink vessel and cooling circuit. As the vessel the component Nr. 28 – *Water tank* is used. Before the media enter to the vessel the pressure must be decreased to the value 1,013 bars. To reduce the pressure the component Nr. 14 – *Control valve* is used. After this valve the *General value input* component (Nr. 33) is used and the wanted pressure inside the water tank is set in this component. Another way is to use component Nr. 2 – *Throttle*. Throttle has more things to configure. Relative or absolute pressure drop can be set or can be used as well as control valve. Heat sink 1 is shown in Figure 32.
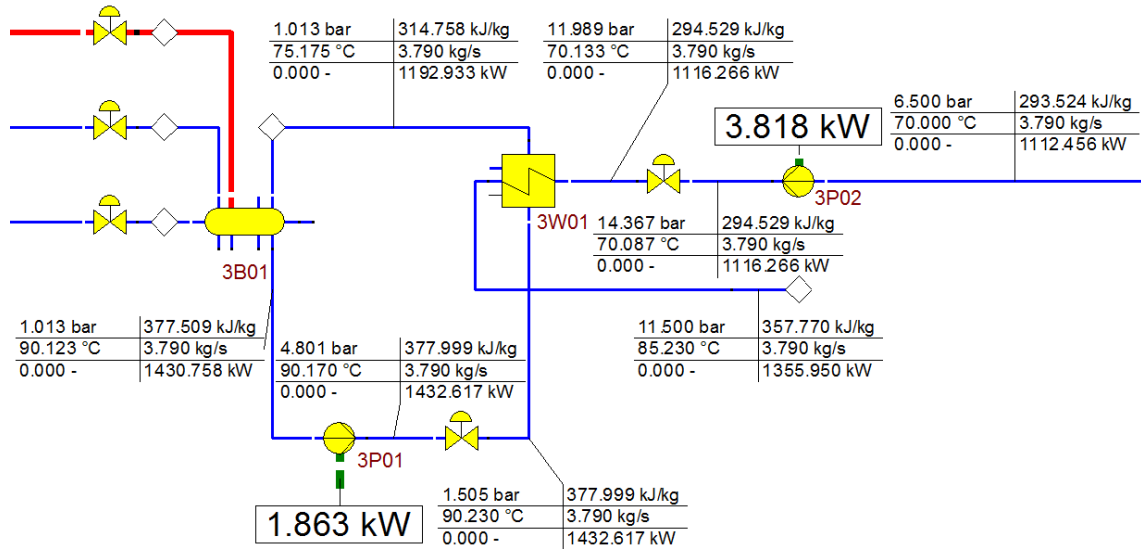
Figure 32: Heat sink 1

The main component of the cooling part is component Nr. 55 – *Universal heat exchanger*. Secondary circuit of the heat exchanger is connected to heat sink vessel. Circuit also contains an additionally auxiliary components such as 3P01 pump and control valve 3A05. Pump serves for circulatory function and for the pressure drop compensation of the heat exchanger. Unlike previous pumps all the pumps in the cooling circuits (3P01, 3P02, 3P03 and 3P04) have implemented the exact delivery head characteristics from the datasheets, which is shown in Figure 33. Figure 34 then shows implemented characteristics in component properties. For the use of imported characteristics it is necessary to switch item *Type of characteristic* on the tab *Specification-Values* to the value *P2: taken from delivery head line*.

Figure 33: Pump characteristics from datasheet [Appendix N]
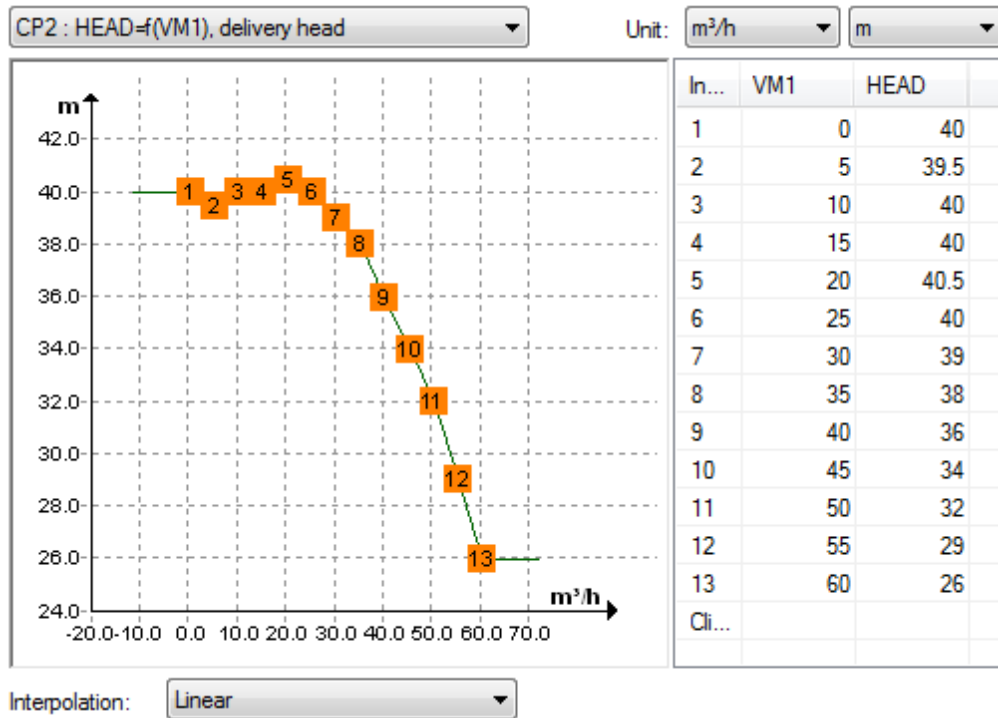
Figure 34: Implemented characteristics in pump component

EBSILON program has three different methods how to interpolate the imported points of characteristic.

a) Linear interpolation – two adjacent points are connected by the straight line

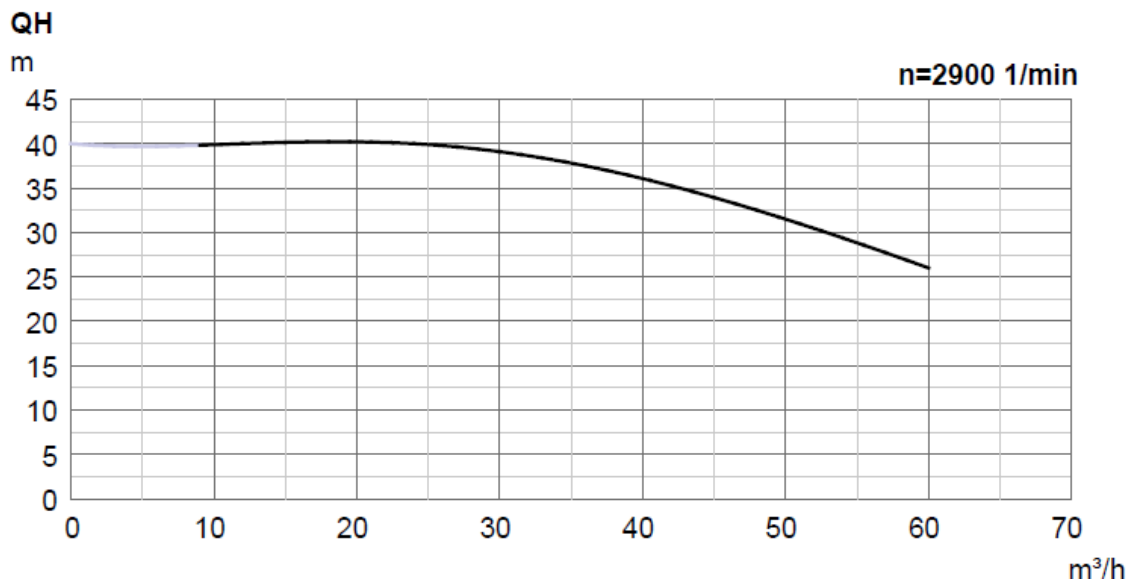b) Interpolation by specified orders – two adjacent points are connected by a polynomial function of given order

c) Spline interpolation – instead of a polynomial function is used piecewise polynomial function

With interpolation is closely related the extrapolation, which can be also performed by EBSILON. For linear interpolation the extrapolation function is equal to constant function in axis y with value equal to the last point. In case of interpolation of specified order the extrapolation is given by the specified order of extrapolation. For spline interpolation the extrapolation is specified in form of derivation value of function's boundary points and can be choose between f'' = 0, f' = constant or f' = constant. For all pumps in THERESA models the linear interpolation is used.

From the values in the Figure 32 can be seen that pressure at the output of the pump is higher than it is necessary. The higher pressure is compensated by control valve, which is placed directly after the pump. Pressure drop on the valve is given by

pressure value, which is set in *General value input* component (Nr. 33). This component is located just before water from secondary circuit enters to the cooling vessel. In the *General value input* component is also set the secondary circuit's mass flow value, which is calculated in Appendix J.

Parameters of heat exchanger 3W01 are taken from datasheet, which is listed in Appendix L. *Upper terminal temperature difference* parameter is set to value 5 K, primary circuit's pressure drops are set to value 0,489 bars and secondary circuit's pressure drop is set to value 0,492 bars. Primary circuit of heat exchanger is supplied by the water with temperature 70 °C and pressure about 12 bars.

Before water flows to the heat exchanger the pressure is increased by pump 3P02. This pump has its characteristics listed in Appendix N. As in previous case, the resulting pressure is higher, so pressure must be reduced by control valve 3A03. Required output pressure from heat exchanger is equal to value 11,5 bars. Heat exchanger 3W01 is shown in Figure 35. Mass flow through primary circuit is the same like for secondary circuit and it is set in *General value input* component at the end of the primary pipeline.
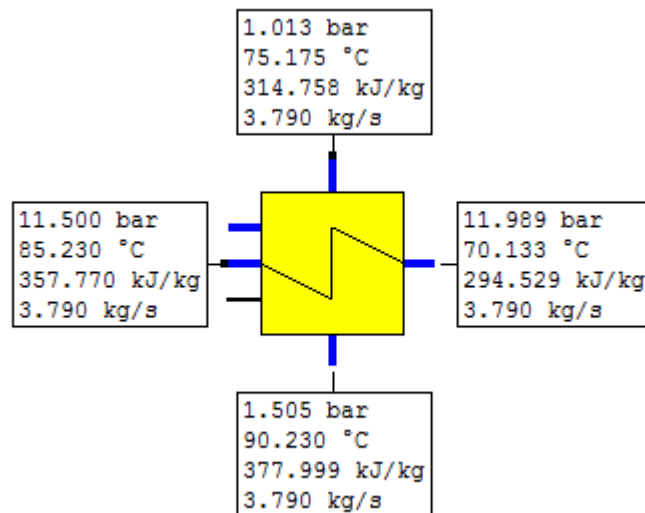

Figure 35: Heat exchanger 3W01

4.1.6. Heat sink 2 (3B02)

Heat sink 2 works the same as heat sink 1. Because heat sink 2 serves only for cooling condensed steam from secondary circuit of preheater 1 the requirements are

lower than in case of heat sink 1. From this reason the pumps aren't the same and have different characteristics, which are listed in Appendix N. *Upper terminal temperature difference* parameter of heat exchanger is set to value 5 K, primary circuit's pressure drop is set to value 0,46 bars and secondary circuit's pressure drops are set to value 0,464 bars. Datasheet for the heat exchanger 3W02 is listed in Appendix M. Mass flow through primary and secondary circuits is calculated in Appendix K. Part of the circuit with heat sink 2 is shown in Figure 36.
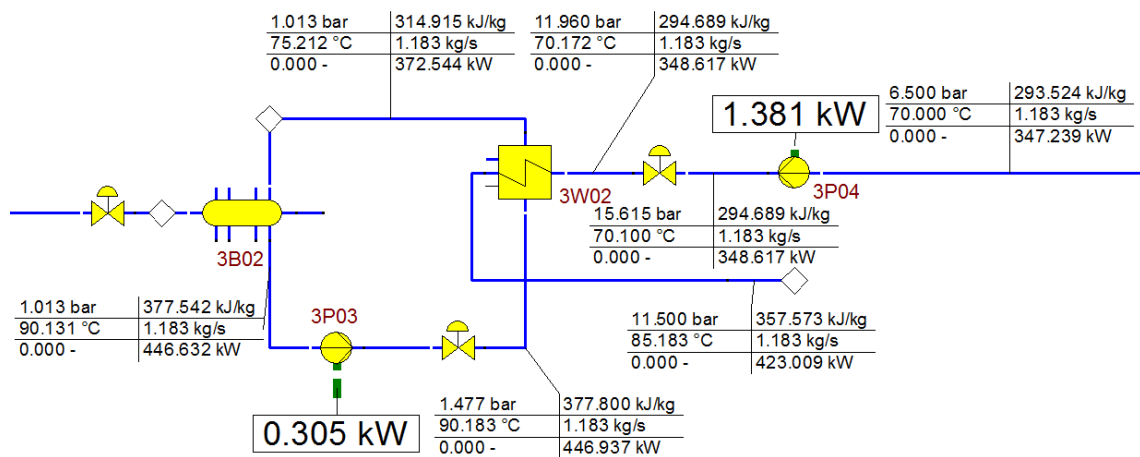
Figure 36: Heat sink 2

From the values at the outputs of primary circuits of heat sinks 1 and 2 can be seen, that even though the used values were used from an ideal mathematical model, the simulation model is balanced. The water inside both heat sinks has little higher temperature then 90 °C. Temperature is possible to reduce by increasing mass flow through primary and secondary circuit of both heat exchangers.

Also the temperature at the output of primary circuits of heat exchangers is higher than expected 85 °C. This is caused by additional components in primary circuits specifically by pumps and control valves, whose outside pressure also increase the temperature of the media. On the other hand, higher temperature would be compensated in the real thermodynamics system by heat losses in the pipelines and other components.

By combining of the individual blocks the model of THERESA facility for FW10 mode is created. Simulation model for whole FW10 mode is shown in Figure 37.
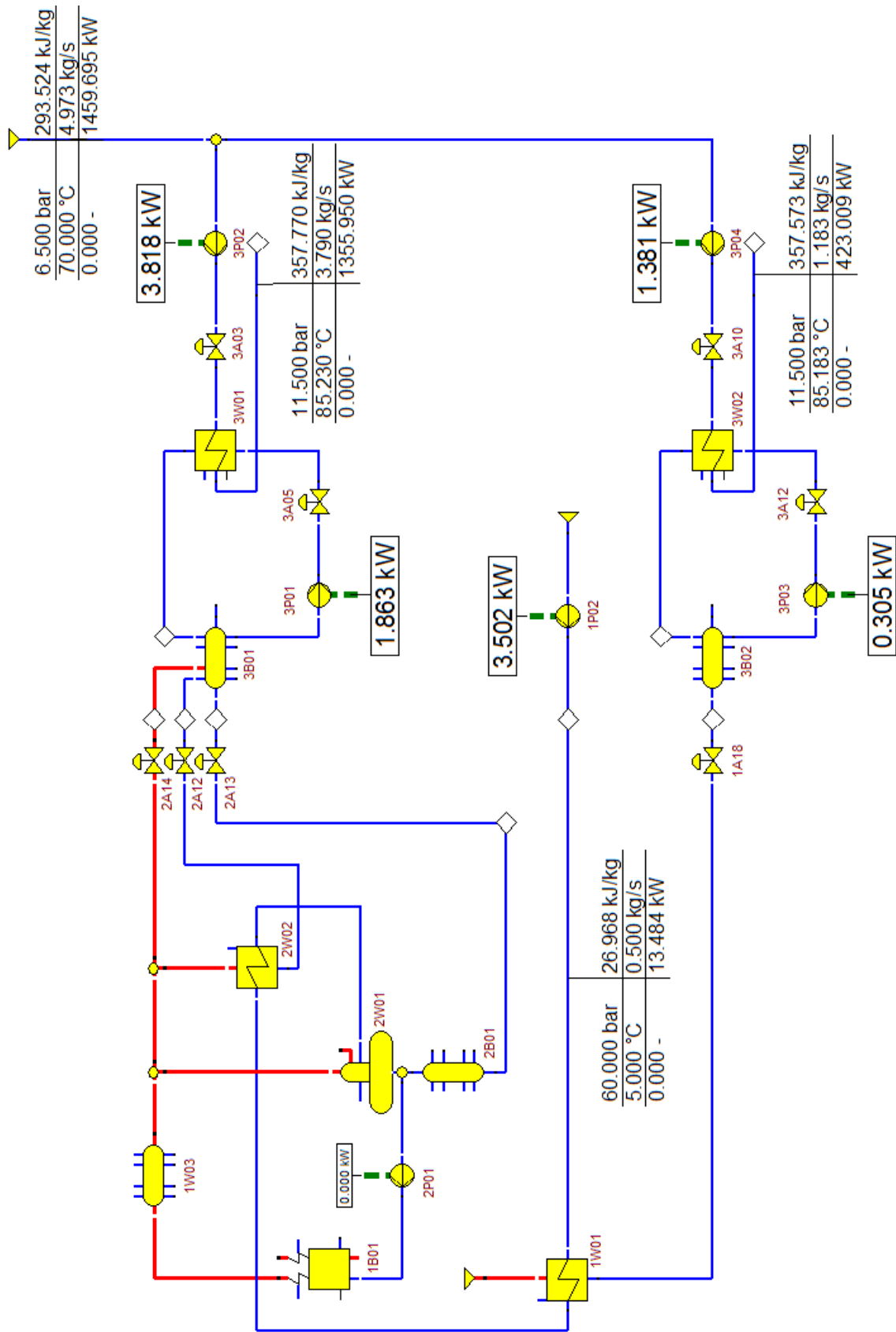
Figure 37: Simulation model for FW10 mode

In next chapters are shown simulation models for other modes of facility. All models are based on this model; therefore, in description are described only changes against the FW10 mode. Used parameters are again the results of ideal mathematical models. Usually, the only parameters, which are changing, are the mass flow and the pressure of water at the input before preheater 1 and the mass flows in the primary and secondary circuits of heat sinks 1 and 2. All changing parameters for all simulation models are listed in Table 4.

Table 4: Changing parameters of simulation models

Parameter	Mode					
	FW1	FW2	FW8	FW9	FW10	FW11
system pressure [bar]	51	60	60	60	60	60
input mass flow [kg/s]	0,484	0,5	0,098	0,5	0,5	0,127
mass flow from TES [kg/s]			0,43	0,123	0,127	
output pressure of steam generator [bar]			59,95	59,9	59,9	59,95
heat loss of superheater [kW]			-25,434	-31,918	-33,014	-32,95
heat loss of TES [kW]	200,013	-213,23	178,034	156,192	180,24	-58,766
mass flow from heat sink 1 [kg/s]	2,856	6,63	1,347	4,176	3,79	5,39
mass flow from heat sink 2 [kg/s]	1,145	1,183	0,232	1,183	1,183	0,3

As is described in chapter *4.1.4. Steam generator (1B01) and superheater (1W03)* the value of the *system pressure [bar]* is related with the value of output pressure of *steam generator [bar]*. In modes, which do not contain the steam generator, the system pressure must be defined in *General value input* component behind pump 1P02. In other modes, the pressure must be defined for *Live steam pressure* parameter in the steam generator properties, and then the value of pressure in *General value input* component behind pump 1P02 can be removed, because the pressure behind the pump is calculated automatically by EBSILON.

4.2. Simulation model for FW1 mode

Mode FW1 is the basic mode of THERESA facility. TES is charged by the hot water. The input water is first heated up to temperature 176 °C by preheater 1 and then heated up again in the steam generator to temperature 265 °C. The water from steam generator flows to the TES, where the part of energy is given for loading of the TES. The output water from the TES has temperature 176 °C. This water is cooled by heat sink 1 to the temperature 90 °C and the condensed steam from secondary circuit of preheater 1 is also cooled to the temperature 90 °C by heat sink 2.

Simulation model of FW1 mode is shown in Figure 38. The component Nr. 5 – *Steam generator*, which is used in FW10 mode, cannot be used for FW1 mode. This component does not contain the output that would allow draining of hot water. For this reason the component Nr. 9 – *Feed water tank* is used. *Heat loss* parameter is set to value -200 kW, which ensures desired water heating.
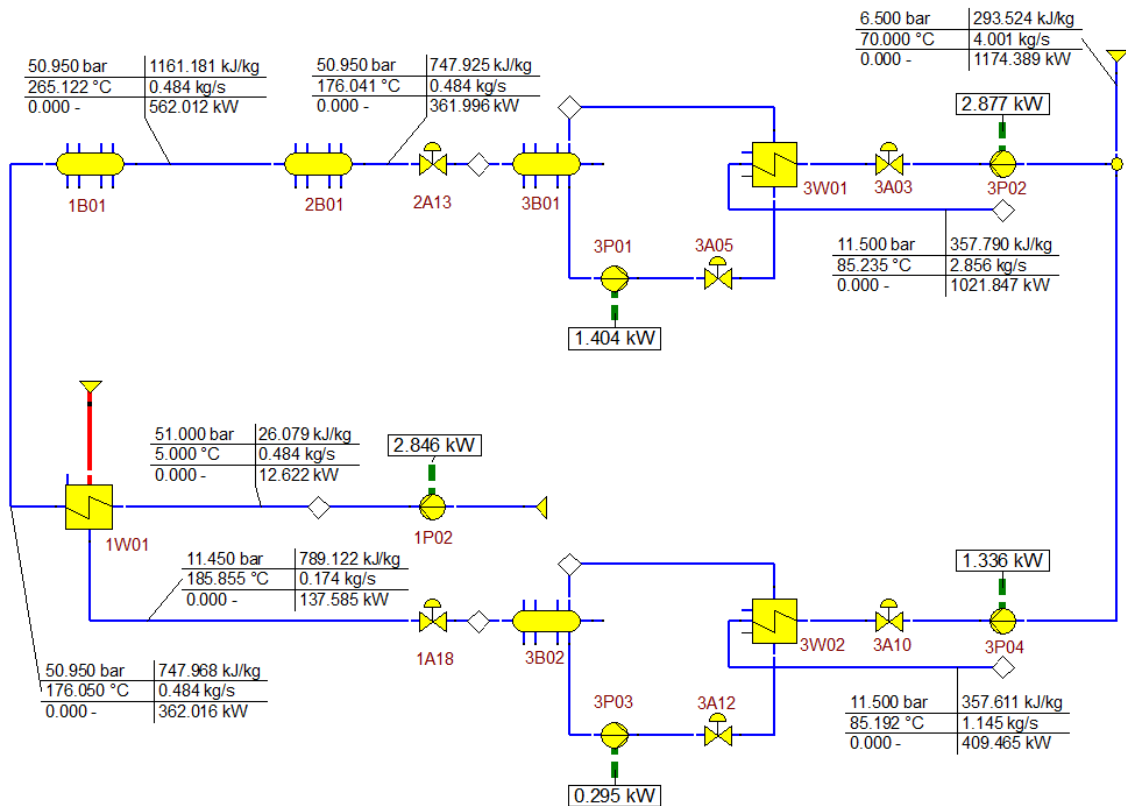
Figure 38: Simulation model for FW1 mode

4.3. Simulation model for FW2 mode

Mode FW2 is opposite to the FW1 mode, because it provides unloading (discharging) of the TES. The input water is again first heated up by preheater 1 to the temperature 176 °C and then heated up by the TES to final temperature 275 °C. This water is cooled by heat sink 1 to the temperature 90 °C and the condensed steam from preheater 1 is also cooled to 90 °C by heat sink 2. Simulation model for FW2 mode is shown in Figure 39.
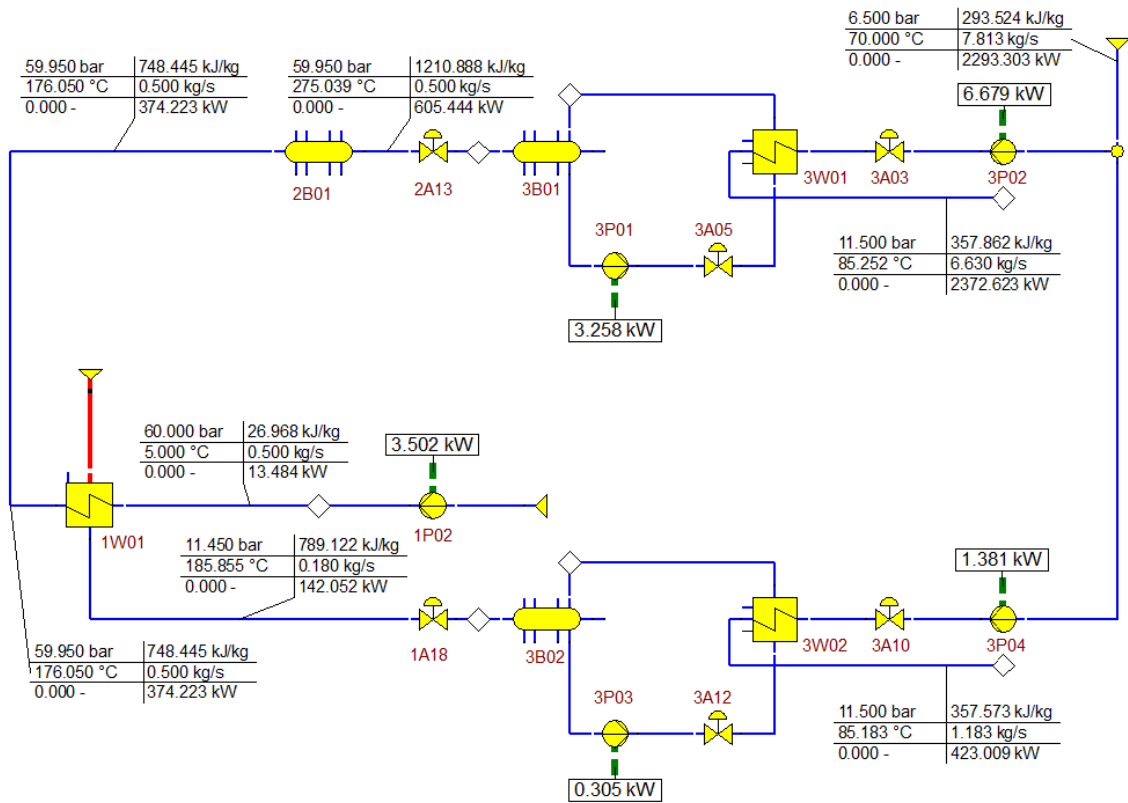
Figure 39: Simulation model for FW2 mode

4.4. Simulation model for FW8 mode

FW8 mode serves for loading TES by the superheated steam. The input water is heated up to the temperature 176 °C by preheater 1 and then is converted to the steam inside steam generator. Saturated steam from generator is converted by superheater to superheated steam with temperature 350 °C. Part of superheated steam is used for the heating of cold water from the TES in preheater 2. Water is heated up from 176 °C to 265 °C and then flows back to the TES. Condensed steam from preheater 2 and rest of the superheated steam is cooled in heat sink 1 and condensed steam from preheater 1 is cooled in heat sink 2. Model of FW8 mode is shown in Figure 40.
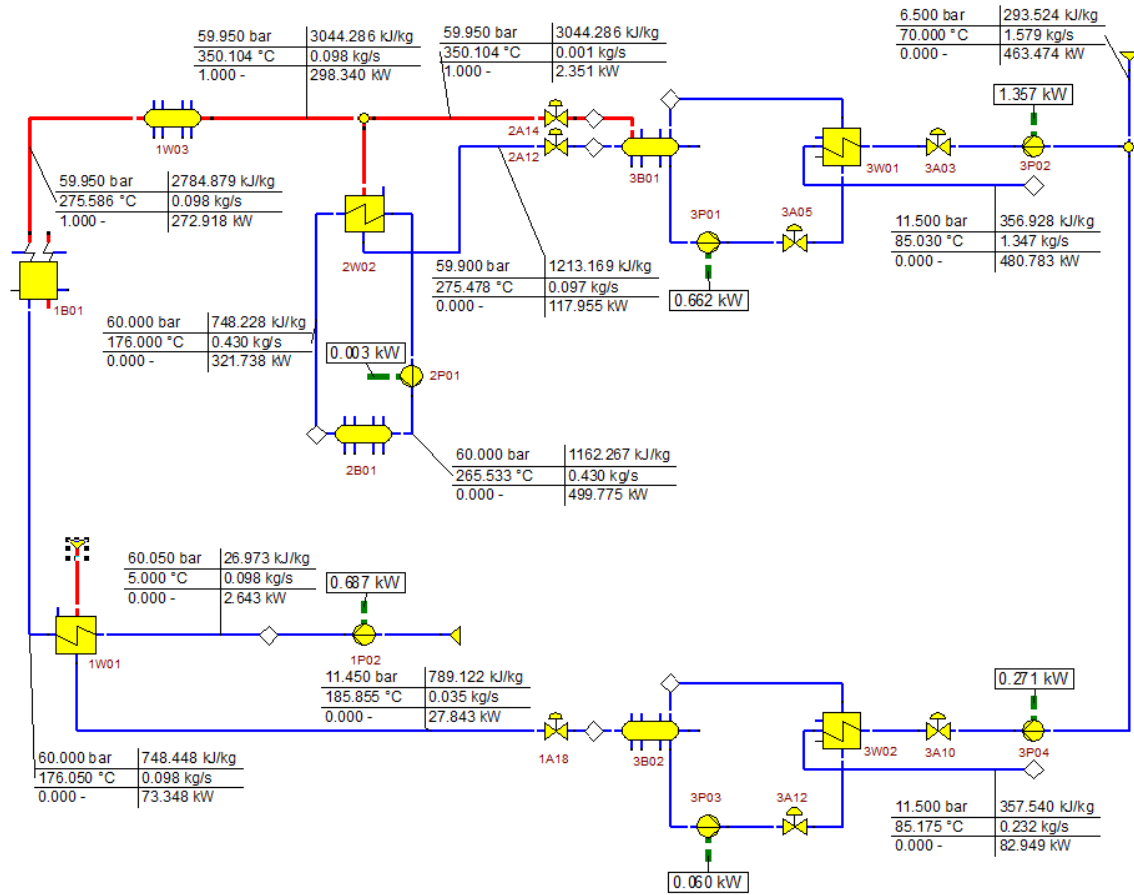
Figure 40: Simulation model for FW8 mode

4.5. Simulation model for FW9 mode

Simulation model for FW9 mode is shown in Figure 41. In mode FW9 the TES is charged by hot water from preheater 2. The input water is heated up to the temperature 176 °C by preheater 1 and after that is heated up by preheater 2 to the temperature 265 °C, which flows to the TES. Part of the water with same temperature feeds steam generator, where is converted to saturated steam, and then is converted by superheater to superheated steam with the temperature 350 °C. By this steam the preheater 2 is supplied. The water with temperature 176 °C flows from the TES to heat sink 1 together with the rest of the superheated steam and condensed steam from preheater 2. Condensed steam from preheater 1 flows to heat sink 2.
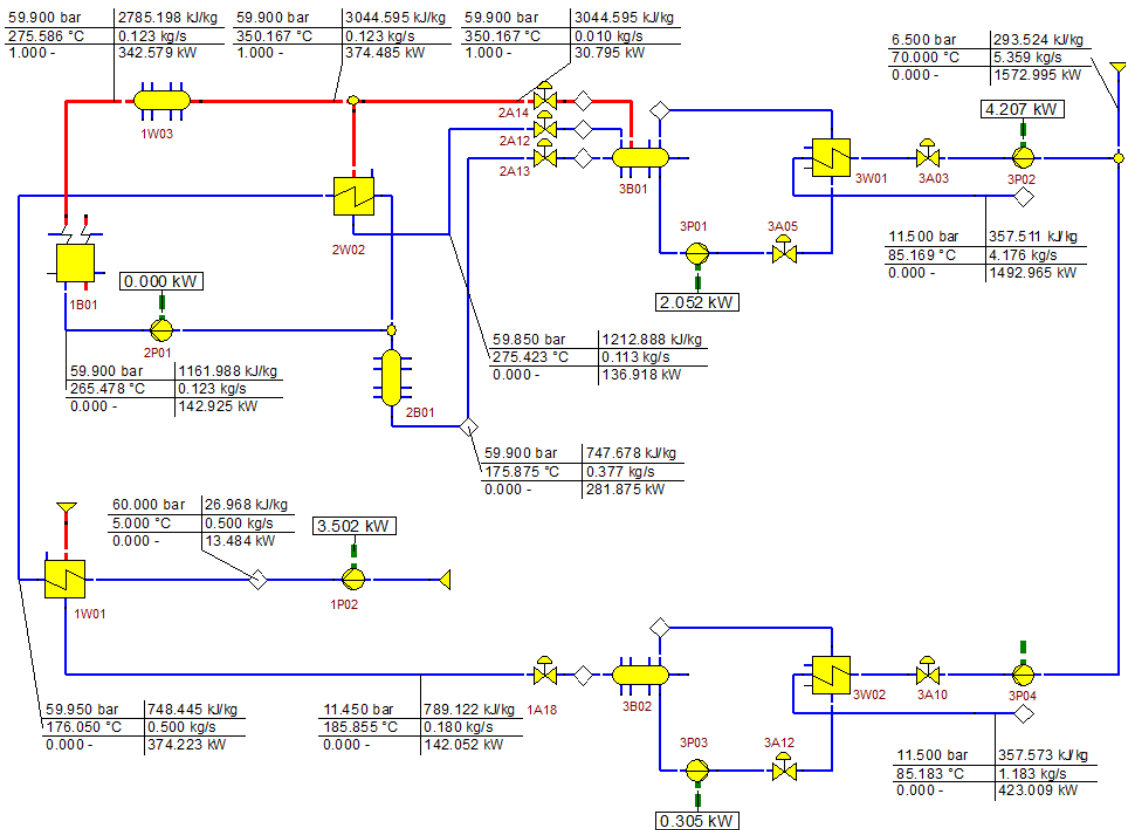
Figure 41: Simulation model for FW9 mode

4.6. Simulation model for FW11 mode

In the last mode FW11 the hot water from the TES is unloaded. The input water is heated up to the temperature 176 °C by the preheater 1 and then flows into the TES. Here the water is by unloading heated up to temperature 275 °C, which feeds the steam generator. The steam from generator flows to the superheater and then to the heat sink 1. The condensed steam from preheater 1 flows into the heat sink 2. Simulation model for FW11 mode is shown in Figure 42.
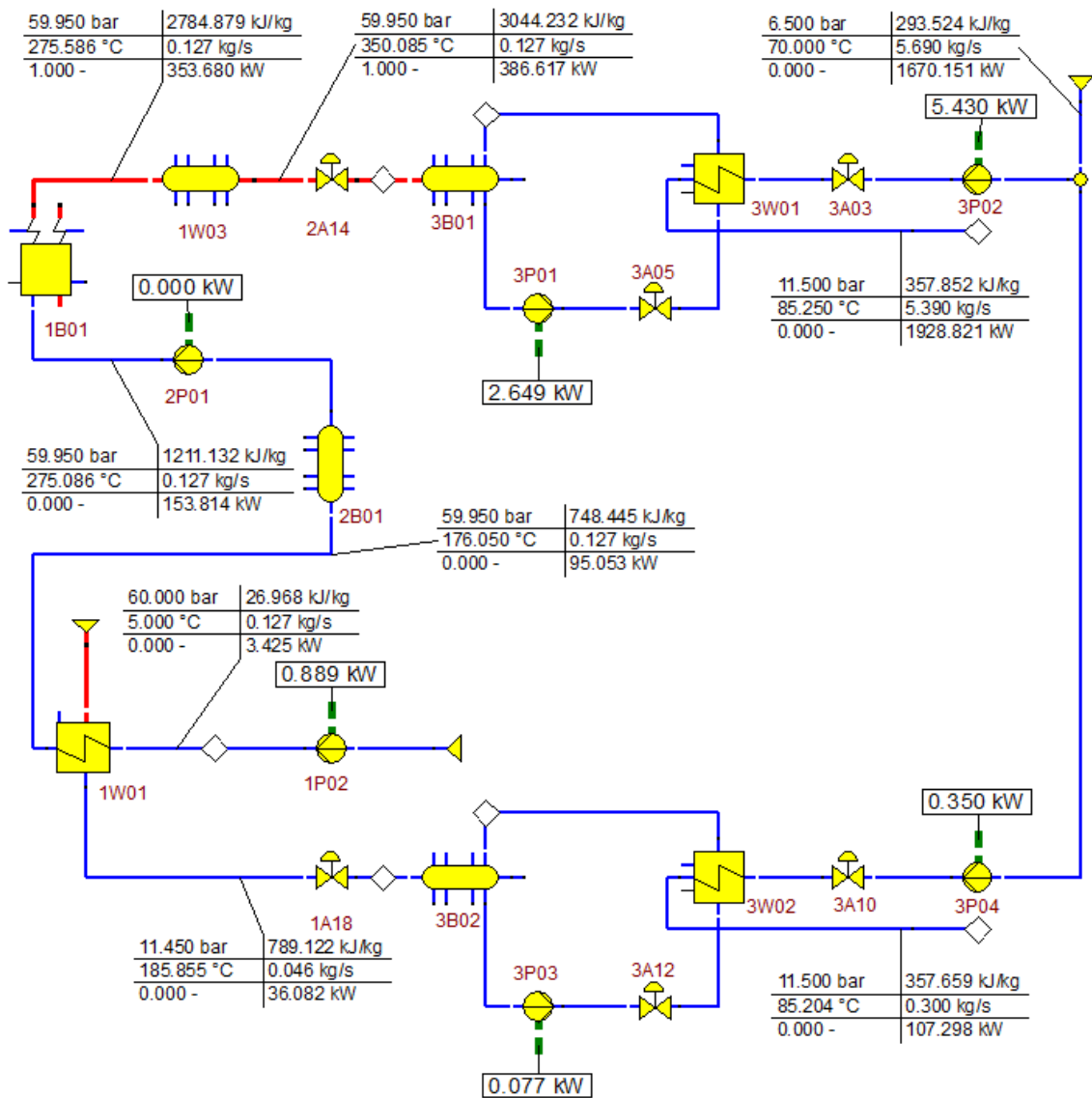

Figure 42: Simulation model for FW11 mode

4.7. Simulation model of THERESA test facility

One of the requirements on the simulation model was its universality and the possibility to simulate all modes within a single model. After finishing the simulation of each mode was therefore necessary to find a way on how to modify existing models and put them into one universal model for whole THERESA facility. The idea is that, the model includes all components and the choice of mode takes a place by routing the mass flows of transport media.

4.7.1. Modifications of existing models

The components from the each model are used. Many components have the same settings for all modes, for example pumps, heat exchangers and preheaters. In contrast to model shown in Figure 37 a simulation model for whole facility contains one extra component. Steam generator is in the individual simulation models represented by two components – *Steam generator* component (Nr. 5) and *Feed water tank* component (Nr. 9). Feed water tank is used for heating the water in mode FW1 and steam generator is used in other modes for converting the water to steam. All other components are the same like in existing models.

The first version of routing was made by components Nr. 4 – *Mass defined splitter* and component Nr. 37 – *Simple mixer*. The example of this solution is shown in Figure 43. The use of this type splitter requires using of additional *General value input* component (Nr. 33) at least at one of the output to define the division of the mass flow between the outputs. This solution is indeed functional, but the disadvantage is the high number of *General value input* components, which is undesirable.
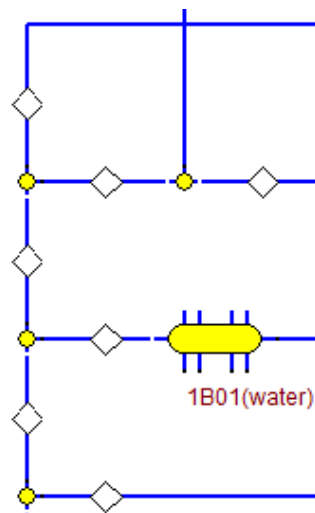
Figure 43: Routing with the use of Mass defined splitter

This disadvantage was removed after several trials by replacing the component Nr. 4 – *Mass defined splitter* with component Nr. 18 – *Splitter with % of flow*. This component has one input and two outputs – primary and secondary output. Only parameter, which can be set in properties of this element, is the relative ratio of the mass flow between secondary and primary output. In the case of model for THERESA facility the value of relative ratio is either zero or one, because this component here doesn't serves for dividing of mass flow, but for routing of the whole mass flow. For the value zero the entire mass flow passes through the primary output and for the value one through the secondary output. No *General value input* components are needed. The resulting model of THERESA facility, which is shown in Figure 44 is then combination of separate models for individual modes with mass flow routing between components.
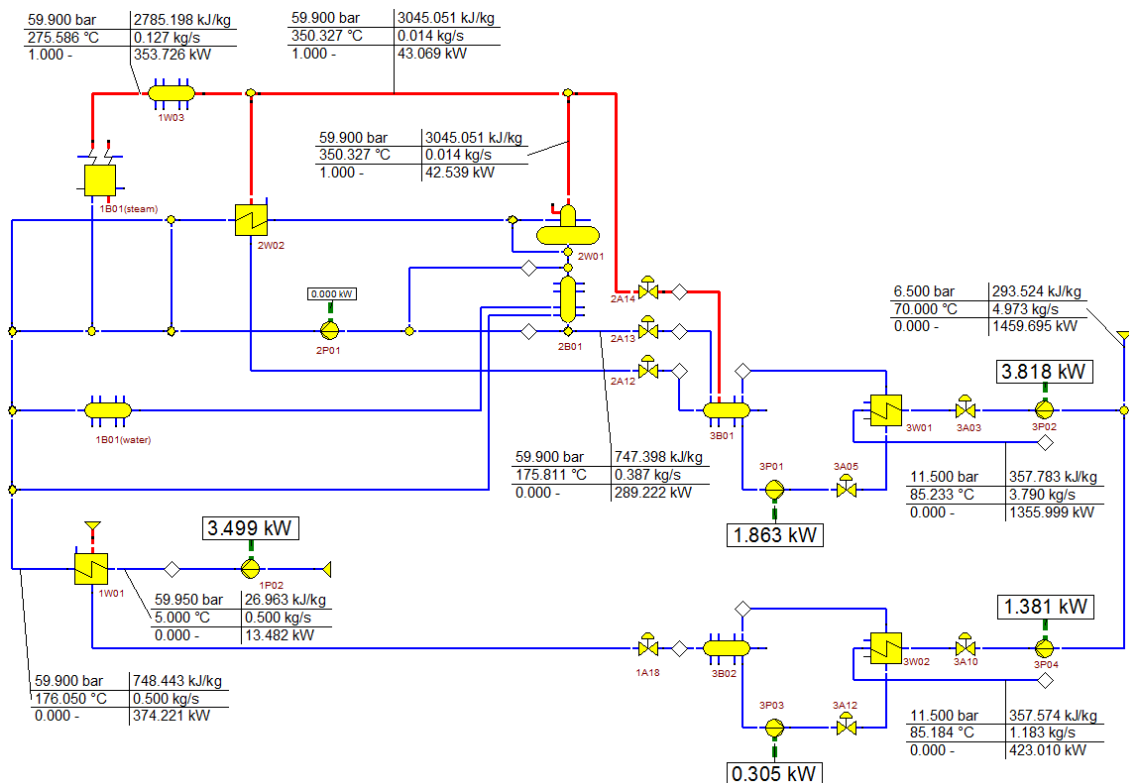
Figure 44: Simulation model of THERESA test facility

4.7.2. Demonstration of model reconfiguration

The following example demonstrates model reconfiguration from one mode to another mode of facility. The values from Table 5 are used and the marking corresponds to the violet letters in Figure 45. It is recommended to perform the individual steps in the specified order.
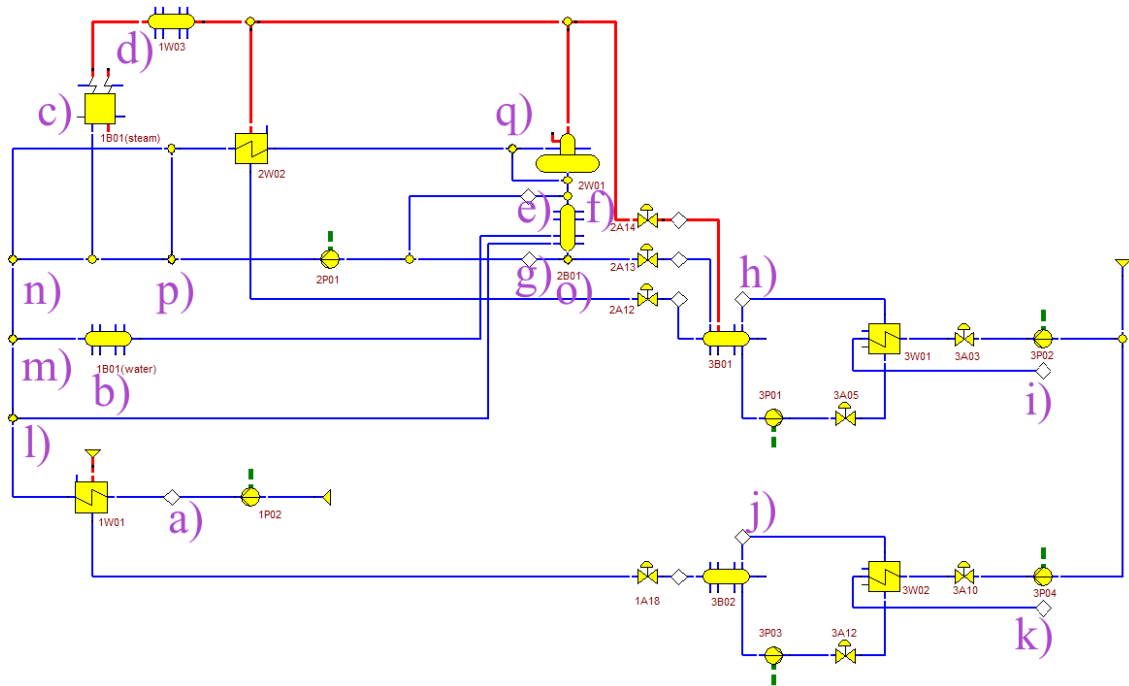
Figure 45: Model with marked points for parameters change

The first part refers to the parameter's settings of the individual components. The change is only for mentioned parameters, the other parameters (if any) don't change.

a) Input mass flow [kg/s].

b) Heat loss of water heater [kW] – value is negative.

c) Output (live stream) pressure of the steam generator [bar].

d) Heat loss of the superheater [kW] – values are negative, so it is actually the power of the superheater.

e) Mass flow, which flows to the steam generator [kg/s].

f) Heat loss of the TES [kW] – for loading the values are positive (power loss), for unloading the values are negative (power generation).

g) Mass flow, which leaves the TES [kg/s]. This value is used only for the cyclic charging of the TES (FW8 mode). In other cases, this value field is empty (not zero).

h) Mass flow of the cooling circuit of heat sink 1 [kg/s].

i) The same value of mass flow like in case h) [kg/s].

j) Mass flow of the cooling circuit of heat sink 2 [kg/s].

k) The same value of mass flow like in case j) [kg/s].

The second part refers to the mass flow routing. As it is described above, the routing takes place in a mode, where the entire input stream passes either to primary output or to the secondary output of the splitter, but only to one of them. There isn't flow division into two pipelines.

l) first splitter at the input side

- 1 – mass flow is directed to the TES (unloading, modes FW2 and FW11)
- 0 – mass flow is directed to the next splitter

m) second splitter at the input side

- 1 – mass flow is directed to the water heater (loading, mode FW1)
- 0 – mass flow is directed to the next splitter

n) third splitter at the input side

- 1 – mass flow is directed to the steam generator (loading, mode FW8)
- 0 – mass flow is directed to the preheater 2 (loading, modes FW9 and FW10)

o) splitter at the output of the TES

- 1 – mass flow is directed to the steam generator or preheater 2 (modes FW8 and FW11)
- 0 – mass flow is directed to the heat sink 1 (modes FW1, FW2, FW9 and FW10)

p) splitter between the steam generator and the preheater 2

- 1 – mass flow is directed to the preheater 2 (mode FW8)
- 0 – mass flow is directed to the steam generator (modes FW9, FW10 and FW11)

q) splitter before the TES

- 1 – mass flow is directed to the TES (modes FW8 and FW9)
- 0 – mass flow is directed to the mixing preheater (mode FW10)

Table 5: Values for THERESA model

Parameter	Mode					
	FW1	FW2	FW8	FW9	FW10	FW11
system pressure [bar]	51	60	60	60	60	60
input mass flow [kg/s]	0,484	0,5	0,098	0,5	0,5	0,127
heat loss of water heater [kW]	-200	0	0	0	0	0
output pressure of steam generator [bar]	50,95	59,95	59,95	59,9	59,9	59,95
heat loss of superheater [kW]	0	0	-25,434	-31,918	-33,014	-32,95
mass flow to steam generator [kg/s]	0	0	0	0,123	0,127	0
heat loss of TES [kW]	200,013	-213,23	178,034	156,192	180,24	-58,766
mass flow to steam generator [kg/s]			0,43			
mass flow from heat sink 1 [kg/s]	2,856	6,63	1,347	4,176	3,79	5,39
mass flow from heat sink 2 [kg/s]	1,145	1,183	0,232	1,183	1,183	0,3

By following the steps above is able to change simulation model of THERESA facility to another mode. Due to the versatility of the model is possible its future expansion by other components such as universal interface with phase-change material and others. Component with appropriate settings (for example from the datasheet) can be added from the component's menu in EBSILON and to the current model can connected by suitable routing the mass flow of the transport medium.

System pressure [bar] parameter is listed in Table 5 only for orientation. The value of system pressure is not saved in *General input value* component (a)), but it must be set in *Steam generator* component (c)) for all modes. Pressure value in *General input value* component (a)) should stay empty. Here is the difference compared with the models for individual modes. The value of *Output pressure of steam generator [bar]* in Table 5 is calculated with the use of formula 28 (page 50) and the same formula can be used in case of pressure change.

The EBSILON contains the feature called *Profile*. Using this function the multiple modes can be contained in single simulation model. This can be explained on the example of THERESA model so that one file contains all of the modes, between them is possible to switch. One profile is defined as parental profile and all its sub profiles in default state inherit the component's parameters. It is of course possible to define the different value parameters for each profile, so for example the value of TES power can be variable. This holds for all parameters of all components. The resulting simulation model of THERESA facility includes a parental profile named THERESA and six sub profiles for each of the modes, as is shown in Figure 46.
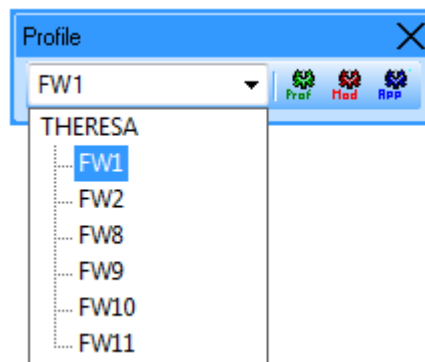
Figure 46: List of profiles of THERESA simulation model

5. Simulation model of power plant with thermal energy storage

The second part of simulations is to implement the thermal energy storage, which was developed for THERESA facility, to the simulation model of the real power plant. The research aims to determine the effect of the implementation of TES on the selected power plant parameters like necessary energy from the fuel, output electrical power and the efficiency.

The model of Lippendorf Power Station is used as the reference power plant. Lippendorf Power Station is a lignite-fired (also known as brown coal-fired) power station, which is located near Leipzig in Saxony, Germany. The power plant is co-owned and operated by Swedish power company Vattenfall Europe AG and by German electric utilities company Energie Baden-Wurttemberg AG (EnBw). Lippendorf Power Station has two units with the installed capacity of 933 MW each and also a district heating capability of 300 MW. The power station has the fuel efficiency about 43 %.

5.1. Simulation model for 100 % of mass flow (nominal mass flow)

The simulations are based on very detailed simulation model of Lippendorf Power Station, which was developed by Vattenfall Europe AG. The starting point for simulations is case with nominal value of mass flow of steam through the turbine. The value of mass flow is stored in component Nr. 39 – *Controller with internal set value*, in the simulation model named as *Controller_3* and placed near an output generator. In Figure 47 the *Controller_3* is marked by a green arrow.
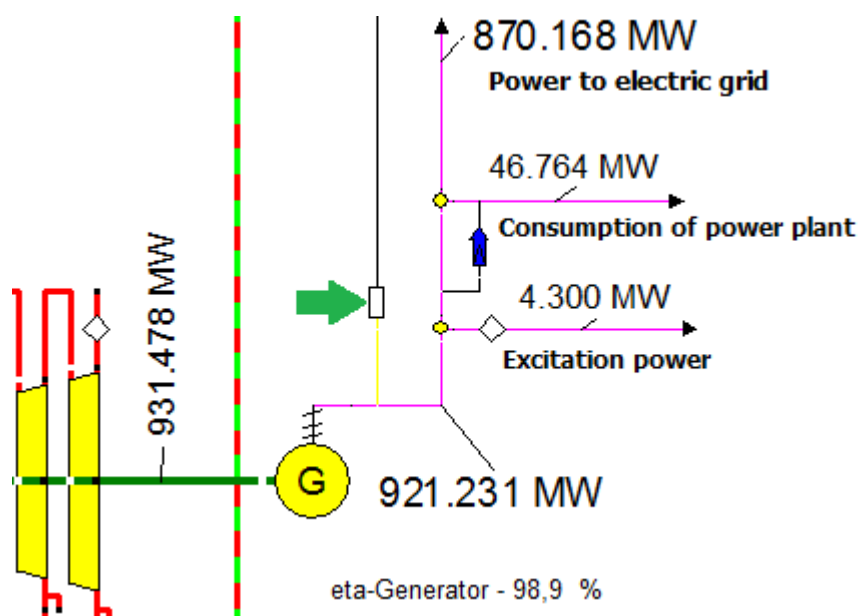

**Figure 47: Output part of Lippendorf plant simulation model**

71

The value of the mass flow corresponding to the nominal mass flow is equal to 672,2 kg/s. Some output parameters of the power plant can be seen in Figure 47, these parameters together with others are summarized in Table 6.

Table 6: Parameters of original simulation model for 100% of mass flow

Parameter	Value
mass flow through turbine	672,2 kg/s
steam generator power	1874,237 MW
mechanical energy	931,482 MW
generated electrical energy	921,235 MW
fuel efficiency	42,6132 %

The basic simulation model is modified by adding of the TES developed in previous chapter for THERESA facility. Method of implementing to the simplified model is shown in Figure 48. The TES is placed behind a pair of preheaters HDV7 and HDV8 and before preheater (steam cooler) ENTH7. An original model contains only one pump P1, which is powered by boiler feed pump turbine (in Figure 48 marked as BFPT). The TES operates at a lower pressure, so the model is extended by the second pump. Pump P1 increases the pressure of water from the feed water tank to the pressure about 50 bars. This pump is driven by an electric motor. Pump P2, which is located behind TES, increases water pressure to the pressure about 300 bars. Preheated water with the temperature about 275 °C and the pressure of 300 bars is used to as a supply of the steam generator.

The high pressure steam generated by the steam generator flows to high pressure turbine (marked as HP). The steam from the high pressure turbine output, which has a much lower pressure, is returned to the reheater circuit inside the steam generator, where is increased its temperature. The steam then passes through medium-pressure turbine (marked as MP) at first and low pressure turbine (marked as LP) after. The steam from all parts of the turbine is first converted into water in condensers and then stored in the feed water tank. To increase the efficiency of the cycle are all preheaters supply by outgoing steam from high pressure or middle-pressure turbine. The electric power generator is on a common shaft with the steam turbine. From Figure 47 can be seen that the part of generated energy is used as an excitation power, part of energy is used to cover own consumption of power plant and the rest of electrical energy is fed into the grid.
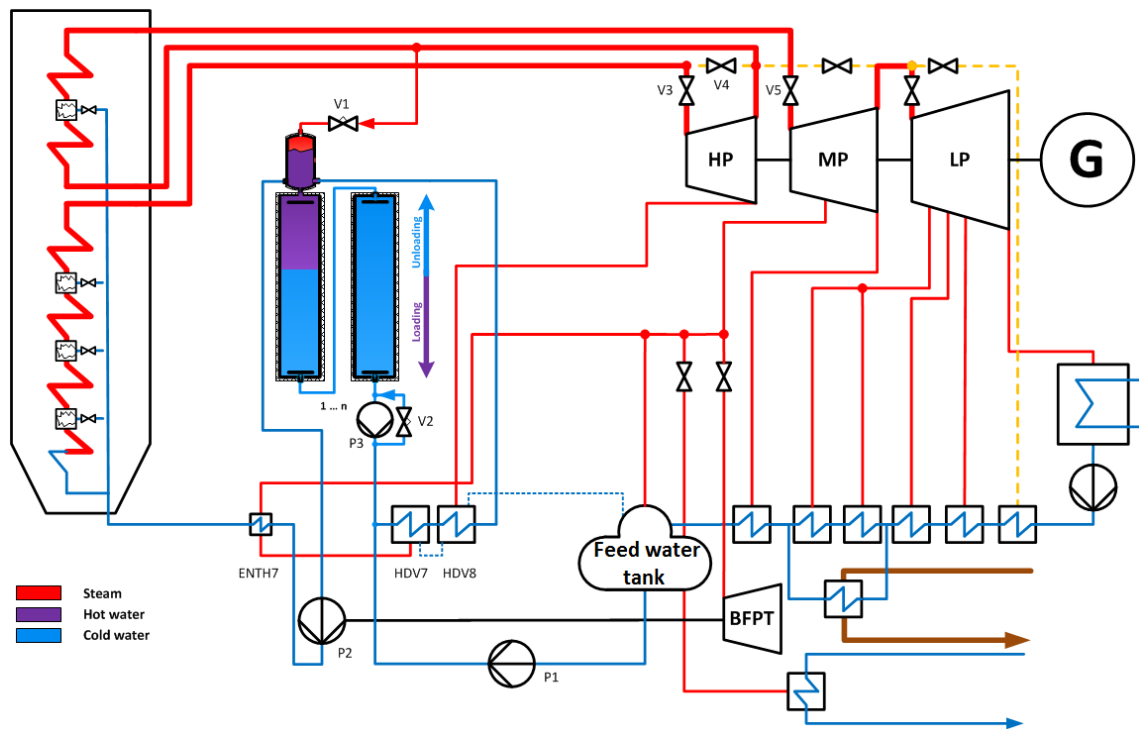
Figure 48: Scheme of power plant with TES [20]

The power plants with an embedded TES can operate in three different modes – normal, loading of TES and unloading of TES.

a) Normal mode – the cold water from the feed water tank flows through preheaters HDV7 and HDV8 and then directly to preheater ENTH7.

b) Loading mode

    a. Loading – the cold water from feed water tank is mixed with cold water from the TES and then is heated up by preheaters HDV7 and HDV8. The same mass flow of water, which flowed from the TES, now flows to TES after preheating. The rest of water flows to the preheater ENTH7 and subsequently to steam generator.

    b. Loading with mixing – the beginning is the same as in the previous case, but the valve V1 is opened and mixed water flows through mixing preheater, where the water is mixed with the steam. The rest is again the same as in previous case.

c) Unloading mode – the preheaters HDV7, HDV8 and ENTH7 are switched off. The cold water from the feed water tank is pumped to the TES and hot water flows out of the TES. The function of preheaters is replaced by the TES; the mass flow of steam, which was used by

preheaters is now used after reheating for middle-pressure turbine, where the mass flow of steam is higher than in normal mode.

The components, which together create the TES, are with one exception the same as in case of TES for THERESA facility. The comparison is shown in Figure 49. The TES contains mixing preheater (marked as 1, component Nr. 9 – *Feed water tank*), thermal energy storage vessel (marked as 2, component Nr. 28 – *Water tank*), splitters (marked as 3, component Nr. 18 – *Splitter with % of flow*) and mixers (marked as 4, component Nr. 37 – *Simple mixer*). Mass defined splitter, which is used in THERESA model (marked as 5, component Nr. 4 – *Mass defined splitter*), is replaced by splitter with % of flow (marked as 3, component Nr. 18).
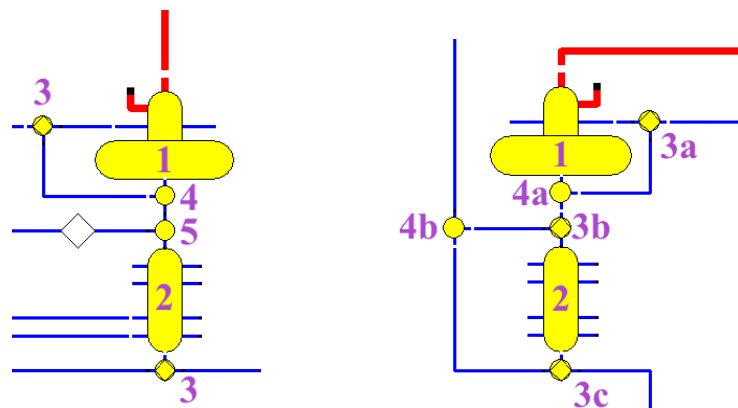
Figure 49: TES for THERESA facility (left) and for Lippendorf plant (right)

In normal mode the hot water flows through the secondary output of the splitter 3a and then through the secondary output of the splitter 3b (in both components the value of *Ratio of mass flow M3 to M1* is equal to 1). At the end the water flows through mixer 4b to preheater ENTH7.

In loading mode the hot water flows again through the secondary output of splitter 3a. The splitter 3b divides the mass flow into two equal flows (*Ratio of mass flow M3 to M1* is equal to 0,5). It means that the mass flow for the loading is the same as mass flow for the steam turbine. The water for the turbine flows through mixer 4b to preheater ENTH7. The water, which is used for loading, flows through the TES and through the primary output of splitter 3c (*Ratio of mass flow M3 to M1* is equal to 0).
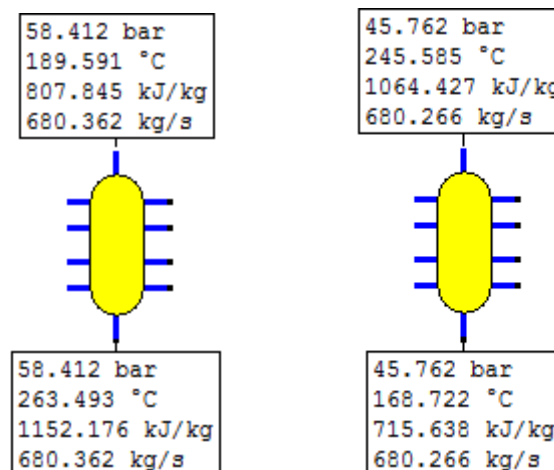
The difference in loading with mixing mode is given by parameter of splitter 3a. The preheated water flows through the primary output of this splitter and it is mixed in the mixing preheater with the steam from the output of high pressure turbine. The rest is the same like as in loading without mixing mode.

In unloading mode the cold water flows through the secondary output of the splitter 3a and through the primary output of the splitter 3b to the TES. The hot water from the TES flows through the secondary output of splitter 3c directly to the pump P2. The water heating in the TES is given by the negative value of heat loss. In ideal case the cold water, which flows to the TES, should be heated up to the temperature, which has water after the preheaters HDV7, HDV8 and before pump P2 in normal mode. The calculation of this value in Mathcad is shown in Code 1.

$$T_{in} := 189.591 °C$$

$$p_{unloading} := 58.412 bar$$

$$T_{out} := 263.495 °C$$

$$p_{normal} := 51.767 bar$$

$$m_f := 680.362 \frac{kg}{s}$$

$$h_{in} := h\_ptx\_97\left(\frac{p_{unloading}}{bar}, \frac{T_{in}}{°C}, -1\right) \cdot \frac{kJ}{kg} = 807.879 \frac{kJ}{kg}$$

$$h_{out} := h\_ptx\_97\left(\frac{p_{normal}}{bar}, \frac{T_{out}}{°C}, -1\right) \cdot \frac{kJ}{kg} = 1.152 \times 10^3 \cdot \frac{kJ}{kg}$$

$$P_{TES} := m_f \cdot (h_{out} - h_{in}) = 234.269 MW$$

Code 1: Calculation of TES power

The resulting value of power is used as a tank heat loss value with a negative sign for unloading and with a positive sign for loading. The opposite value for loading is given by the requirement to have the same or nearly same value for both, loading and unloading. The values of TES for unloading and loading are shown in Figure 50.

```
58.412 bar        45.762 bar
189.591 °C        245.585 °C
807.845 kJ/kg     1064.427 kJ/kg
680.362 kg/s      680.266 kg/s
```

```
58.412 bar        45.762 bar
263.493 °C        168.722 °C
1152.176 kJ/kg    715.638 kJ/kg
680.362 kg/s      680.266 kg/s
```

Figure 50: Values for calculated power for unloading (left) and loading (right)

When comparing the input water temperature of TES during loading and output water temperature during unloading is clear that the value of TES power is not entirely correct. Problem is that the output temperature during unloading is higher than input temperature during loading. This is not possible due to physical laws and the TES construction. The value of power is necessary to recalculated with respect to input temperature during loading. The recalculation is shown in Code 2.

$$h_{in} := 807.879 \frac{kJ}{kg}$$

$$T_{out} := 245.585 °C$$

$$p_{normal} := 51.767 \, bar$$

$$m_f := 680.362 \frac{kg}{s}$$

$$h_{out} := h\_ptx\_97\left(\frac{p_{normal}}{bar}, \frac{T_{out}}{°C}, -1\right) \cdot \frac{kJ}{kg} = 1.064 \times 10^3 \cdot \frac{kJ}{kg}$$

$$P_{TES} := m_f \cdot \left(h_{out} - h_{in}\right) = 174.499 \, MW$$

Code 2: Recalculation of TES power

The changing of TES power causes the changing of water input temperature for loading. After rounding the value of the power was set to

$$P_{TES} = 180 \; MW$$

and the values of TES are shown in Figure 51. The output temperature during unloading is calculated with respect to the input temperature to TES and the power of TES. It isn't possible to directly change this value by user.

58.412 bar	46.640 bar
189.696 °C	248.294 °C
808.310 kJ/kg	1077.496 kJ/kg
680.361 kg/s	680.269 kg/s

58.412 bar	46.640 bar
247.332 °C	190.856 °C
1072.876 kJ/kg	812.895 kJ/kg
680.361 kg/s	680.269 kg/s
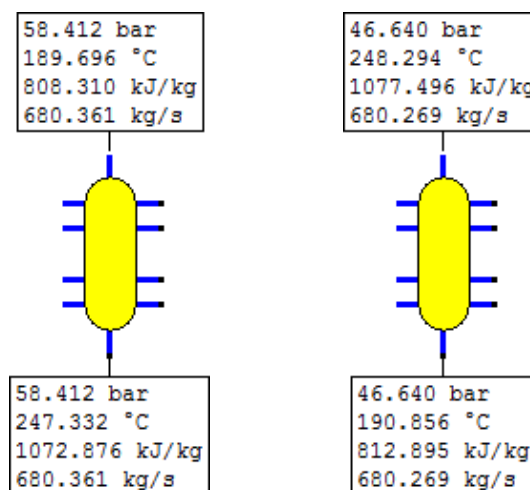
Figure 51: Values for recalculated power for unloading (left) and loading (right)

The power increasing occurs in case of loading with mixing mode, because in this case the input temperature to TES during loading is higher about 10 °C. Resulting TES power after recalculation for loading with mixing mode is

$$P_{TES} = 205\ MW$$

With calculated value of TES power is possible to start with simulation. In original simulation model with embedded TES a several error has occurred, so the further model modifications are necessary.

5.1.1. Modifications of the model to eliminate errors

The first error occurs for loading with mixing mode in component Nr. 8 – *Pump*, in the simulation model named as *Pump_5*. The water pressure before pump is higher than behind pump, the difference is about 0,3 bars. The pump serves as a pressure adapting element for water in two parts of the circuit. After pressure is equalized, the water is mixed together. The error is solved by adding the component Nr. 3 – *Mixer with throttle* parallel to the pump. By this modification can be pressure equalization done in both ways. The mass flow routing is ensured by component Nr. 18 – *Splitter with % of flow*. Model modification is shown in Figure 52.
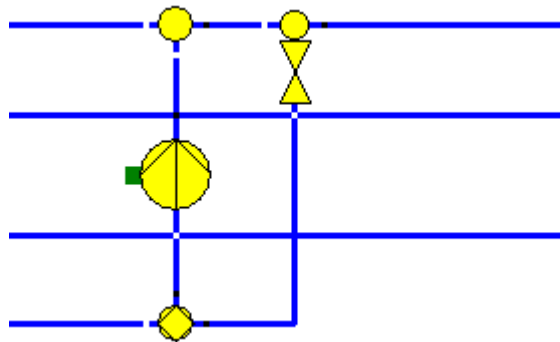

Figure 52: Modification of pressure adapting element

Another error occurs again in loading with mixing mode. The component Nr. 14 – *Control valve*, named *RV_Spat_1*, serves for limiting of steam pressure before BFPT turbine on the maximum pressure value to 9,7 bars. In original model the maximum value is stored component named *Start_value_6* (Nr. 33 – *General value input*). Problem is that in case of loading with mixing mode the pressure before valve is lower than pressure value set in *Start_value_6* component. The modification is based on replacing the *Start_value_6* component by component Nr. 36 – *Value transmitter*.

In value transmitter component is defined the maximum transmitting value equals to 9,7 bars. If the pressure before valve is lower than 9,7 bars the output behind valve is equal to value before valve. If the pressure is higher than 9,7 bars the output from valve is decreased to 9,7 bars. Modification is shown in Figure 53.
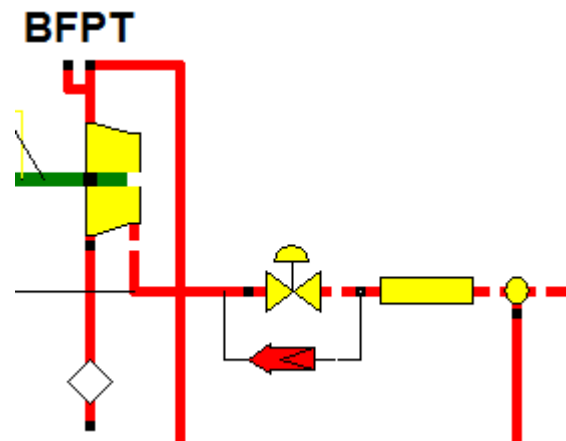

Figure 53: Modification of control valve

The rest of errors occur in unloading mode of power plant. Two errors, that are closely related, occur in pump (component Nr. 44 – *Extraction pump*) in the simulation model named as *Speisepumpe_1*, and in steam generator (component Nr. 5 – *Steam generator*) named as *Kessel*. The error in pump is caused by pressure of water, which is extracted from pump and which wanted value is too low and cannot be extracted. Extracted water is sprayed in the generator. The error in generator is again caused by pressure of sprayed water, because this pressure must be higher than steam pressure, in which the water is sprayed. Pressure of extracted water before modification is stored in component Nr. 33 – *General value input*, named as *Start_value_2*, and is equal to 55 bars. The minimum water pressure needed by steam generator is about 55,2 bars and minimum pressure of extracted water from pump is about 58,5. The new value of extracted water pressure is equal to

$$p = 58,5 \ bar$$

The change of this value solves errors in pump and also in steam generator.

The last error is missing pressure before middle-pressure turbine, specifically before blade wheel A7. In original model the necessary pressure before turbine is calculated by EBSILON from the knowledge of basic turbine parameters and output pressure from turbine. Individual parts of the turbine are partially connected together

and so the output pressure is given by the balance between them. Calculation of input pressure is based on Stodola's cone law (also known as Ellipse Law). The Stodola's law is defined like

$$\frac{\dot{m}_f}{\dot{m}_{f\_nom}} = \frac{p_{in}}{p_{in\_nom}} \cdot \sqrt{\frac{p_{in\_nom} \cdot V_{m\_nom}}{p_{in} \cdot V_m}} \cdot \sqrt{\frac{1 - \left(\frac{p_{out}}{p_{in}}\right)^{\frac{n+1}{n}}}{1 - \left(\frac{p_{out\_nom}}{p_{in\_nom}}\right)^{\frac{n+1}{n}}}} \tag{30}$$

where $\dot{m}_f$ is input mass flow, V_m is input volume flow, p_{in} is input pressure, p_{out} is output pressure and n is polytrophic exponent. Variables contain '*nom*' in the sub index relate to the nominal values of variables for the specific turbine. If the steam is considered as an ideal gas the Stodola's law is simplified to the form

$$\frac{\dot{m}_f}{\dot{m}_{f\_nom}} = \frac{p_{in}}{p_{in\_nom}} \cdot \sqrt{\frac{T_{in\_nom}}{T_{in}}} \cdot \sqrt{\frac{1 - \left(\frac{p_{out}}{p_{in}}\right)^2}{1 - \left(\frac{p_{out\_nom}}{p_{in\_nom}}\right)^2}} \tag{31}$$

where T_{in} is input temperature of the steam and T_{in_nom} is its nominal value. The input pressure is then equal to

$$p_{in} = \pm \sqrt{p_{out}^2 + \left(\frac{\dot{m}_f^2}{\dot{m}_{f\_nom}^2} \cdot \frac{T_{in}}{T_{in\_nom}} \cdot \left(p_{in\_nom}^2 - p_{out\_nom}^2\right)\right)} \tag{32}$$

All input pressures for all part of the steam turbine are calculated with the using of this formula. The solution of too low pressure before turbine is to switch the value of *Inlet pressure handling* parameter from value *P1 calculated from P1NSET (by Stodola equation)* to the value *P1 given from outside* for the unloading mode. Than is necessary to add component Nr. 36 – *Value transmitter* to transmit the value of pressure from the pipe before the valve (component Nr. 14 – *Control valve*, named *Control_valve*) to the pipe behind the valve. This modification causes that the steam flows through the valve without the changing of pressure. The second option would be to remove the valve, but this solution is not appropriate for the reason described below. The modification is shown in Figure 54.
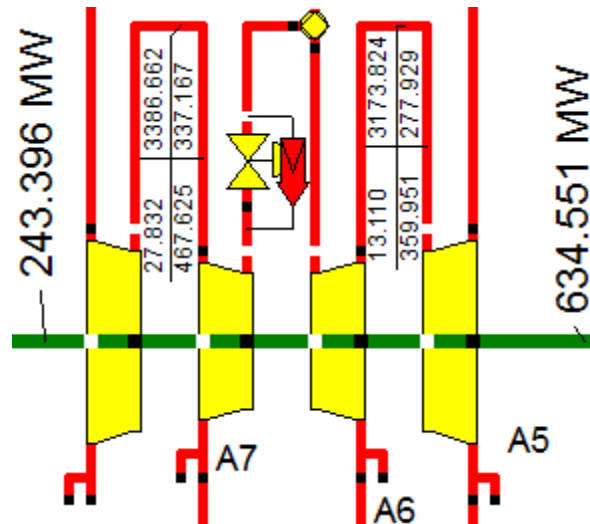
Figure 54: Modification of the middle-pressure turbine

The last modification of simulation model consists in adding the pipe parallel to the preheaters HDV7 and HDV8. During unloading mode the preheaters are switched off and the water flows through them without changing of temperature, but with pressure drop, which is in total equals to 4 bars. The purpose of the pipeline is to bypass the preheaters so that in case of unloading the loss of water pressure was prevented. The water enters to the pipe through splitter named *Bypass_1* (component Nr. 18 – *Splitter with % of flow*) and the pipeline is concluded by component Nr. 3 – *Mixer with throttle*, named *Bypass_2*.

To make the simulation model more user friendly the profile feature of EBSILON is used. As described in chapter *4.7.2. Demonstration of model reconfiguration* the profile feature allows having the different modes of a simulation model in one file. The simulation model contains one main profile with five sub profiles. The sub profiles represent individual modes of power plant with TES.

      a) Normal mode
      b) TES loading
      c) TES unloading
      d) TES loading with mixing
      e) TES unloading 2

The difference between profile *TES unloading* and *TES unloading 2* is in power of TES. Profile *TES unloading 2* has higher TES power corresponding to the higher power of loading with mixing mode. This is related to the need have for certain components different settings for each profile. For example in unloading profiles the middle-

pressure turbine A7 is switched to mode, where the input pressure is given from outside, but in other modes the same turbine is switched to mode, where the pressure is calculated..

5.1.2. Results for 100% of mass flow

The Figure 55 shows the part of the modified simulation model, which includes embedded TES, without error and with bypass of preheaters HDV7 and HDV8. The values of selected power plant parameters are then listed in Table 8.
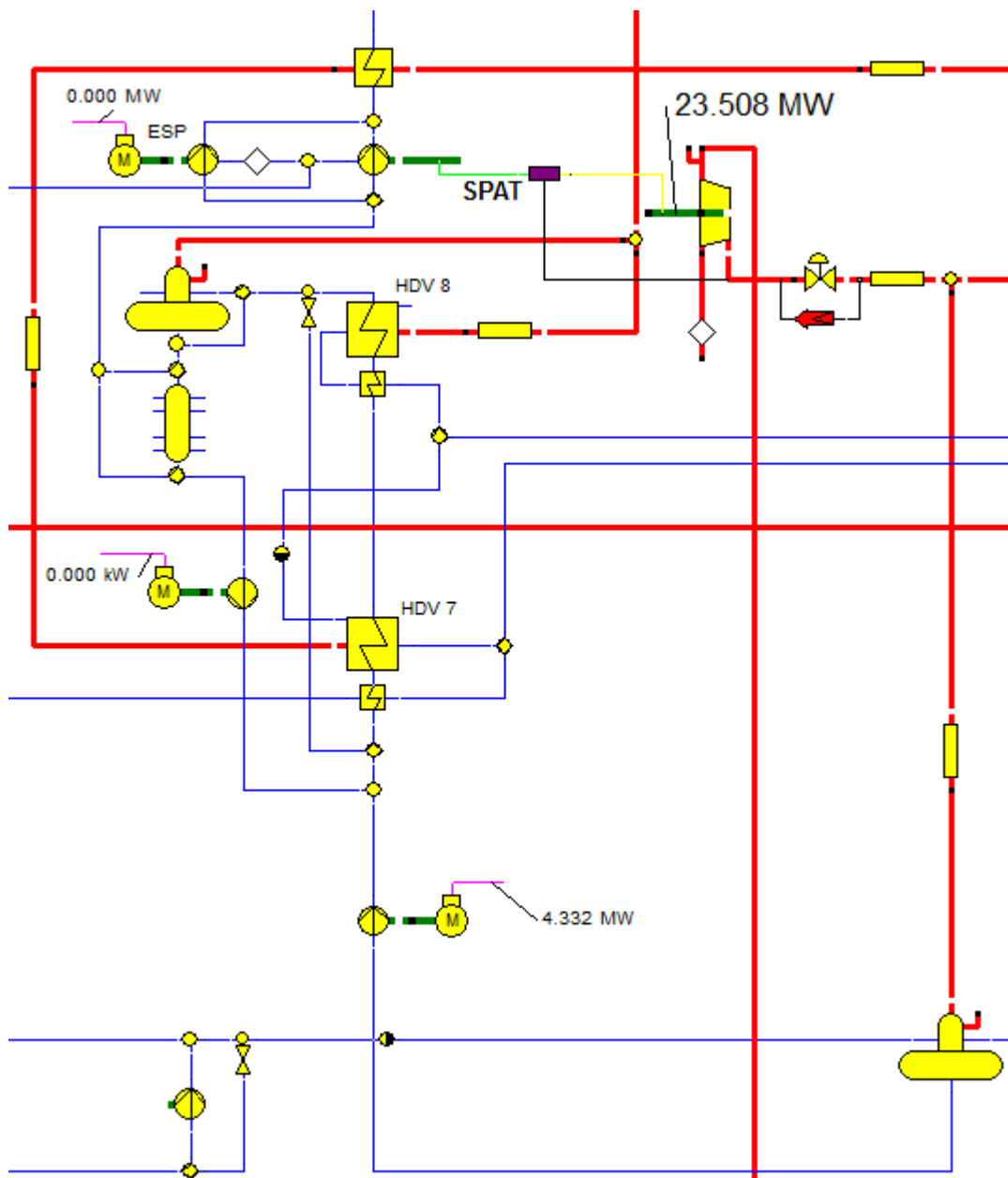

Figure 55: Simulation model of power plant with TES

Table 7: Selected values of power plant for 100% of mass flow

Mode	η [%]	P_out [MW]	P_steam [MW]	TES [MW]
Normal	42,8950	913,008	1845,799	0
Loading	39,5646	851,486	1865,286	180
Unloading	45,2951	1018,402	1948,591	-180
Unloading with bypass	45,2864	1018,405	1948,964	-180
Loading with mixing	39,1471	823,968	1824,600	205
Unloading 2	45,9150	1018,966	1923,784	-205
Unloading 2 with bypass	45,9061	1018,969	1924,157	-205
Original model	*42,6132*	*921,235*	*1874,237*	-

5.2. Simulation model for 80% mass flow

The second state, for which the effect of embedded TES in the power plant is simulated, is state with the value of turbine mass flow corresponding to 80% of nominal mass flow. The original model for this mass flow isn't possible to simulate without further modifications. The problem is the low steam pressure before high pressure turbine. It is the same problem as for nominal mass flow and unloading mode. The only difference is in the part of turbine, where the pressure is missing. The solution of this error is therefore the same. It is necessary to switched the *Inlet pressure handling* parameter to the value *P1 given from outside* and add the *Value transmitter* component as is shown in Figure 56. With this modification the simulation model works without errors. The parameters for modified power plant model without TES are listed in Table 9.
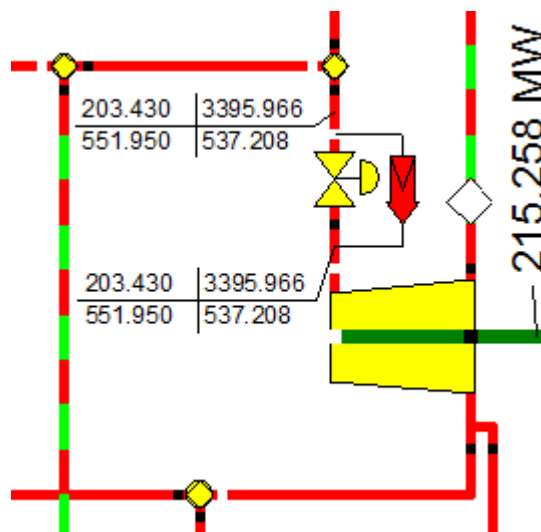
Figure 56: Modification of the high pressure turbine

Table 8: Parameters of original simulation model for 80% of mass flow

Parameter	Value
mass flow through turbine	537,8 kg/s
steam generator power	1544,236 MW
mechanical energy	763,943 MW
generated electrical energy	751,962 MW
fuel efficiency	42,2792 %

The simulation model with embedded TES and with steam mass flow through turbine equal to 80% of nominal mass flow reports the same errors like previous model for nominal mass flow. Only error, which doesn't occur, is the error, where the wanted pressure of water extracted from *Speisepumpe_1* is too low and the pressure of sprayed water in *Kessel* is also too low. All the other occurred errors are

a) higher pressure before *Pump_5* than behind the pump,

b) the pressure before valve *RV_Spat_1* is lower than value of pressure set in *Start_value_6* component,

c) the pressure before *MDE_A7* is too low,

d) the pressure before *HP_turb* is too low.

All of these errors are solved by combination of solutions describes at the beginning of this chapter and in chapter *5.1.1. Modifications of the model to eliminate errors*. Simulation model again contains every mode of the power plant with help of profile feature. The resulting values of selected power plant parameters are listed in Table 10.

Table 9: Selected values of power plant for 80% of mass flow

Mode	η [%]	P_{out} [MW]	P_{steam} [MW]	TES [MW]
Normal	42,4651	747,072	1527,679	0
Loading	39,3810	696,159	1534,300	140
Unloading	45,0181	826,221	1593,513	-140
Unloading with bypass	45,0128	826,222	1593,700	-140
Loading with mixing	39,1059	683,023	1516,045	155
Unloading 2	45,4656	826,732	1579,040	-155
Unloading 2 with bypass	45,4602	826,733	1579,227	-155
Original model	*42,2792*	*751,962*	*1544,236*	-

5.3. Summary of the results for Lippendorf power plant

The simulation results are compared in Figure 57 with respect to the achieved efficiencies of the power plant for the individual modes. The results show that for the basic power plant mode with and without embedded thermal energy storage the efficiencies are almost equal. The small difference is caused by the using two pumps for model with embedded thermal energy storage, where the one pump is powered by steam and the second pump is driven from the electrical mains. The model without the thermal energy storage has only one pump powered by the steam, so the used quantity of the steam for the pump is higher.
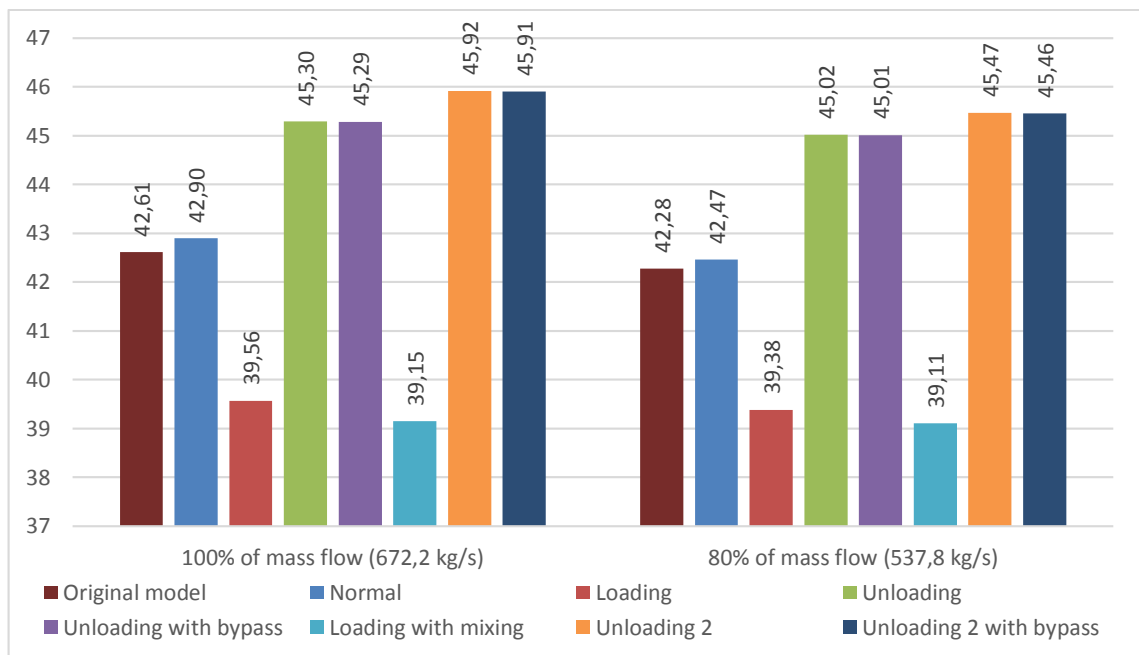

Figure 57: Comparison of efficiencies between different modes

The efficiencies of the basic loading mode for both mass flows are higher about 0,35% compared with loading with mixing mode. It is logical, because the additionally quantity of steam for mixing is taken from the output of the high pressure turbine and this quantity is not further used for next parts of turbine. On the other hand the energy taken by the TES during loading with mixing mode is higher by 13% and this energy can be then used for unloading, because from the results can be seen that the efficiency difference between unloading modes with different values of TES power is about 0,54%.

Noticeable change during unloading with bypass almost doesn't appear in results, even with bypass the efficiency decreases. The decrease is given by the lower

temperature of water, which flows to the steam generator. The temperature rises when the pressure increases, but still the same during pressure drops in preheaters, and so the water temperature before pump 2 is for the same pressure higher in case of unloading mode without bypass.

Conclusion

Thermal energy storage can be labeled as a key technology for balancing the demand and supply of electricity in thermal power plants. Selection of the storage type depends on number technical and economic aspects. Nowadays are widely used storages that using oil or water-based solutions as a storage medium. Water has up to three time's greater heat capacity than metals and it's cheaper. Latent storages are also used, but are still in the development stage as well as thermochemical storages.

The several simulation models for the THERESA test facility, which serves for simulation and analysis of processes relevant for thermal power plants, have been developed. They are prepared individual models for individual facility modes as well as a universal model able to simulate any other modes. A universal model also contains predefined profiles for all known modes, which are using the TES. Due to the versatility of the model is possible its future expansion by new components. The results of simulations show that even if the ideal mathematical model was used, as the basis for simulation, the models are thermodynamically balanced. Small deviations are due to the fact that the components have properties identical or similar to those of the real system components. In cases, where it was possible, the parameters of components were used from the datasheets.

The TES developed for the THERESA facility has been implemented to the existing simulation model of Lippendorf Power Station. Necessary modifications to correct the errors, which are described in thesis, were made and the final simulations were performed for the nominal value and for 80% of nominal value of the turbine mass flow. From the simulation results can be concluded that by the implementation of the TES the output parameters of the plant remain unchanged for normal mode.

The fuel efficiency is decreased for loading mode and increased for unloading mode. The better results were obtained for loading mode with use of mixing preheater. In this mode the quantity of saved energy is higher and the TES can be charged faster. Also for unloading mode, where the higher energy from TES is used, the results are better and the efficiency is higher about 0,5% than in case of unloading with lower energy. The difference between normal unloading and unloading with bypass of preheaters is very low and in the real power plant, because of additional heat and pressure losses, the difference can be neglected.

References

[1] KADRNOŽKA, Jaroslav. Tepelné elektrárny a teplárny. 1st edition, Praha: Státní nakladatelství technické literatury, 1984. 607 pages.

[2] Thermal power station [online]. San Francisco (CA): Wikimedia Foundation, 2014. [cit. 2014-04-29]. Available from: <http://en.wikipedia.org/wiki/Thermal_power_station>

[3] HORLOCK, John H.. Combined power plants : including combined cycle gas turbine (CCGT) plants. 1st edition, Oxford: Pergamon Press, 1992. 288 pages. ISBN: 0-08-05052-9

[4] KERCELLI, L. I.; RYŽKIN, A. V.. Tepelné elektrárny – díl I. 1st edition, Praha: Státní nakladatelství technické literatury, 1955. 374 pages.

[5] Geothermal energy [online]. San Francisco (CA): Wikimedia Foundation, 2014. [cit. 2014-04-29]. Available from: <http://en.wikipedia.org/wiki/Geothermal_energy>

[6] BECKMANN, G.; GILLI, P. V.. Thermal Energy Storage: Basics, Design, Applications to Power Generation and Heat Supply. 1st edition, Wien: Springer-Verlag, 1984. 230 pages. ISBN: 3-211-81764-6

[7] DINCER, Ibrahim; ROSEN, Marc. Thermal Energy Storage : Systems and Applications. 2nd edition, Chichester: Wiley, 2011. 599 pages. ISBN: 978-0-470-74706-3

[8] SCHNEIDER, Clemens. Investigation of thermal energy storage for power plants [presentation]. 2013-05-28.

[9] GARTLER, Günter. Development of a High Energy Density Sorption Storage System [online]. [cit. 2014-06-03]. Available from: < http://download.aee-intec.at/0uploads/dateien7.pdf >

[10] Storing heat [online]. [cit. 2014-06-04]. Available from: <http://forschung-energiespeicher.info/en/waerme-speichern/ueberblick>

[11] Carnot cycle [online]. San Francisco (CA): Wikimedia Foundation, 2014. [cit. 2014-05-06]. Available from: <http://en.wikipedia.org/wiki/Carnot_cycle>

[12] Enthalpy [online]. San Francisco (CA): Wikimedia Foundation, 2014. [cit. 2014-05-06]. Available from: <http://en.wikipedia.org/wiki/Enthalpy>

[13] Revised Release on the IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam [online]. 2007. [cit. 2014-05-06]. Available from: <http://www.iapws.org/relguide/IF97-Rev.pdf>

[14] FluidMAT LibIF97 for Mathcad (Student's Version) [online]. 2013. [cit. 2014-05-07]. Available from: <http://thermodynamics.hs-zigr.de/cms/_data/FluidMAT_LibIF97_Stud_Docu_Eng.pdf>

[15] Test Facility THERESA at the Zittau Power Plant Laboratory (Zittauer Kraftwerkslabor ZKWL) [online]. 2007. [cit. 2014-05-15]. Available from: <http://ipm.hszg.de/fileadmin/Redakteure/Institute/IPM/Projekte/fg-mt-pa/THERESA/20130905_presentation_VA_THERESA.pdf>

[16] KUNEŠ, Josef; VAVROCH, Otakar; FRANTA, Václav. Základy modelování. 1st edition, Praha: Státní nakladatelství technické literatury, 1989. 263 pages. ISBN: 80-03-00147-1

[17] KUNEŠ, Josef. Modelování tepelných procesů. 1st edition, Praha: Státní nakladatelství technické literatury, 1989. 423 pages. ISBN: 80-03-00134-X

[18] Lippendorf [online]. 2012. [cit. 2014-07-02]. Available from: <http://powerplants.vattenfall.com/node/293>

[19] Lippendorf Power Plant, Germany [online]. 2012. [cit. 2014-07-02]. Available from: <http://www.power-technology.com/projects/lippendorf/>

[20] BRAUN, Sebastian. Arbeitsbesprechung zum gemeinsamen Vorhaben Hochschule Zittau/Görlitz (IPM) und Vattenfall Europe Generation (Cottbus) [presentation]. 2014-03-12.

[21] BASSAS, Jordi. Development and implementation of a Nuclear Power Plant steam turbine model in the system code ATHLET [online]. 2011. [cit. 2014-07-10]. Available from: <http://upcommons.upc.edu/pfc/bitstream/2099.1/14161/1/JORDI-BASSAS-MasterThesis-110722.pdf>

[22] File: steam_turbine.gif. [online] [cit. 2014-04-29]. Available from: <http://www.mpoweruk.com/steam_turbines.htm>

[23] File: PowerStation2.svg. [online] [cit. 2014-04-29]. Available from: <http://en.wikipedia.org/wiki/Thermal_power_station>

[24] File: 1332124197159.gif. [online] [cit. 2014-04-29]. Available from: <https://www.eon.com/en/business-areas/power-generation/natural-gas-and-oil/combined-cycle-gas-turbine-how-it-works.html>

[25] File: nuclear+power+plants.jpg. [online] [cit. 2014-04-29]. Available from: <http://vikashkumarsharma.blogspot.de/2011/03/types-of-power-plants-generation-of.html>

[26] File: geothermal_schema.gif. [online] [cit. 2014-04-29]. Available from: <http://www.stork-gears.com/news_maag_gearbox_repair_geothermal_powerplant.php>

[27] File: solar-thermal-power-plant.jpg. [online] [cit. 2014-04-29]. Available from: <http://www.practical-alternative-energy.com/how-does-solar-energy-work.html>

[28] Figure: Fig. 2.2. Sliding-pressure accumulator. Page 26. BECKMANN, G.; GILLI, P. V.. Thermal Energy Storage: Basics, Design, Applications to Power Generation and Heat Supply. 1st edition, Wien: Springer-Verlag, 1984. 230 pages. ISBN: 3-211-81764-6

[29] Figure: Fig. 2.10. Sensible heat accumulator with stationary solid storage medium. Page 42. BECKMANN, G.; GILLI, P. V.. Thermal Energy Storage: Basics, Design, Applications to Power Generation and Heat Supply. 1st edition, Wien: Springer-Verlag, 1984. 230 pages. ISBN: 3-211-81764-6

[30] File: Carnot_cycle_p-V_diagram.svg. [online] [cit. 2014-05-06]. Available from: <http://en.wikipedia.org/wiki/Carnot_cycle>

[31] File: diag97.gif. [online] [cit. 2014-05-06]. Available from: <http://www.steamtablesonline.com/help.aspx>

Appendix A – Contents of enclosed CD

- Text – text of the diploma thesis
  - Diploma_thesis_2015_Michal_Jadrny.pdf
  - Diploma_thesis_2015_Michal_Jadrny.docx
  - Assignment.pdf

- Simulation model of THERESA – source files for THERESA model
  - FW1 mode.ebs
  - FW10 mode.ebs
  - FW11 mode.ebs
  - FW2 mode.ebs
  - FW8 mode.ebs
  - FW9 mode.ebs
  - THERESA.ebs (model without modes)
  - THERESA_with_modes.ebs (model with modes)

- Tables
  - Changing values of THERESA.xlsx
  - Results of Lippendorf model.xlsx
  - Values of original Lippendorf model.xlsx

- Datasheets – used datasheets for heat exchangers and cooling pumps

Appendix B – Coefficient for calculating the enthalpy of water

i	I_i	J_i	n_i
1	0	-2	1,4632971213167E-01
2	0	-1	-8,4548187169114E-01
3	0	0	-3,7563603672040E+00
4	0	1	3,3855169168385E+00
5	0	2	-9,5791963387872E-01
6	0	3	1,5772038513228E-01
7	0	4	-1,6616417199501E-02
8	0	5	8,1214629983568E-04
9	1	-9	2,8319080123804E-04
10	1	-7	-6,0706301565874E-04
11	1	-1	-1,8990068218419E-02
12	1	0	-3,2529748770505E-02
13	1	1	-2,1841717175414E-02
14	1	3	-5,2838357969930E-05
15	2	-3	-4,7184321073267E-04
16	2	0	-3,0001780793026E-04
17	2	1	4,7661393906987E-05
18	2	3	-4,4141845330846E-06
19	2	17	-7,2694996297594E-16
20	3	-4	-3,1679644845054E-05
21	3	0	-2,8270797985312E-06
22	3	6	-8,5205128120103E-10
23	4	-5	-2,2425281908000E-06
24	4	-2	-6,5171222895601E-07
25	4	10	-1,4341729937924E-13
26	5	-8	-4,0516996860117E-07
27	8	-11	-1,2734301741641E-09
28	8	-6	-1,7424871230634E-10
29	21	-29	-6,8762131295531E-19
30	23	-31	1,4478307828521E-20
31	29	-38	2,6335781662795E-23
32	30	-39	-1,1947622640071E-23
33	31	-40	1,8228094581404E-24
34	32	-41	-9,3537087292458E-26

Appendix C – Coefficient for calculating the enthalpy of steam

i	J_i^o	n_i^o
1	0	-9,6927686500217E+00
2	1	1,0086655968018E+01
3	-5	-5,6087911283020E-03
4	-4	7,1452738081455E-02
5	-3	-4,0710498223928E-01
6	-2	1,4240819171444E+00
7	-1	-4,3839511319450E+00
8	2	-2,8408632460772E-01
9	3	2,1268463753307E-02

i	l_i	J_i	n_i	i	l_i	J_i	n_i
1	1	0	-1,7731742473213E-03	23	7	0	-5,9059564324270E-18
2	1	1	-1,7834862292358E-02	24	7	11	-1,2621808899101E-06
3	1	2	-4,5996013696365E-02	25	7	25	-3,8946842435739E-02
4	1	3	-5,7581259083432E-02	26	8	8	1,1256211360459E-11
5	1	6	-5,0325278727930E-02	27	8	36	-8,2311340897998E+00
6	2	1	-3,3032641670203E-05	28	9	13	1,9809712802088E-08
7	2	2	-1,8948987516315E-04	29	10	4	1,0406965210174E-19
8	2	4	-3,9392777243355E-03	30	10	10	-1,0234747095929E-13
9	2	7	-4,3797295650573E-02	31	10	14	-1,0018179379511E-09
10	2	36	-2,6674547914087E-05	32	16	29	-8,0882908646985E-11
11	3	0	2,0481737692309E-08	33	16	50	1,0693031879409E-01
12	3	1	4,3870667284435E-07	34	18	57	-3,3662250574171E-01
13	3	3	-3,2277677238570E-05	35	20	20	8,9185845355421E-25
14	3	6	-1,5033924542148E-03	36	20	35	3,0629316876232E-13
15	3	35	-4,0668253562649E-02	37	20	48	-4,2002467698208E-06
16	4	1	-7,8847309559367E-10	38	21	21	-5,9056029685639E-26
17	4	2	1,2790717852285E-08	39	22	53	3,7826947613457E-06
18	4	3	4,8225372718507E-07	40	23	39	-1,2768608934681E-15
19	5	7	2,2922076337661E-06	41	24	26	7,3087610595061E-29
20	6	3	-1,6714766451061E-11	42	24	40	5,5414715350778E-17
21	6	16	-2,1171472321355E-03	43	24	58	-9,4369707241210E-07
22	6	35	-2,3895741934104E+01				

Appendix D - Calculations of preheater 1 properties

$T_{prim\_in} := 5°C$

$T_{prim\_out} := 176°C$

$m_{f\_prim} := 0.5\dfrac{kg}{s}$

$p_{prim} := 60bar$

$T_{sec\_in} := 190°C$

$p_{sec} := 11.5bar$

$\eta := 1$

$$h_{prim\_out} := h\_ptx\_97\left(\frac{p_{prim}}{bar},\frac{T_{prim\_out}}{°C},-1\right)\cdot\frac{kJ}{kg} = 748.228\frac{kJ}{kg}$$

$$h_{prim\_in} := h\_ptx\_97\left(\frac{p_{prim}}{bar},\frac{T_{prim\_in}}{°C},-1\right)\cdot\frac{kJ}{kg} = 26.968\frac{kJ}{kg}$$

$$T_{sec\_out} := ts\_p\_97\left(\frac{p_{sec}}{bar}\right)\cdot°C = 186.05°C$$

$$h_{sec\_in} := h\_ptx\_97\left(\frac{p_{sec}}{bar},\frac{T_{sec\_in}}{°C},-1\right)\cdot\frac{kJ}{kg} = 2.793\times 10^3\cdot\frac{kJ}{kg}$$

$$h_{sec\_out} := h\_ptx\_97\left(\frac{p_{sec}}{bar},\frac{T_{sec\_out}}{°C},0\right)\frac{kJ}{kg} = 789.988\frac{kJ}{kg}$$

$$m_{f\_sec} := m_{f\_prim}\cdot\left[\frac{h_{prim\_out}-h_{prim\_in}}{\eta\cdot\left(h_{sec\_in}-h_{sec\_out}\right)}\right] = 0.18\frac{kg}{s}$$

$$Q_{1W01} := m_{f\_prim}\cdot\frac{h_{prim\_out}-h_{prim\_in}}{\eta} = 360.63kW$$

Appendix E - Calculations of steam generator properties

$P_{max} := 200 kW$

$\eta := 1$

$T_{in} = 275.586°C$

$p := 60 bar$

$\quad P := \eta \cdot P_{max} = 200 \cdot kW$

$$T_{out} := ts\_p\_97\left(\frac{p}{bar}\right) \cdot °C = 275.586°C$$

$$h_{in} := h\_ptx\_97\left(\frac{p}{bar}, \frac{T_{in}}{°C}, 0\right) \cdot \frac{kJ}{kg} = 1.214 \times 10^3 \cdot \frac{kJ}{kg}$$

$$h_{out} := h\_ptx\_97\left(\frac{p}{bar}, \frac{T_{out}}{°C}, -1\right) \cdot \frac{kJ}{kg} = 2.785 \times 10^3 \cdot \frac{kJ}{kg}$$

$$m_{f\_out} := \frac{P}{h_{out} - h_{in}} = 0.127 \frac{kg}{s}$$

Appendix F - Calculations of superheater properties

$T_{in} := 275.586°C$

$T_{out} := 350°C$

$m_f := 0.127\dfrac{kg}{s}$

$p := 60bar$

$\eta := 1$

$$h_{in} := h\_ptx\_97\left(\dfrac{p}{bar}, \dfrac{T_{in}}{°C}, 1\right) \cdot \dfrac{kJ}{kg} = 2.785 \times 10^3 \cdot \dfrac{kJ}{kg}$$

$$h_{out} := h\_ptx\_97\left(\dfrac{p}{bar}, \dfrac{T_{out}}{°C}, -1\right) \cdot \dfrac{kJ}{kg} = 3.044 \times 10^3 \cdot \dfrac{kJ}{kg}$$

$$P := m_f \cdot \dfrac{\left(h_{out} - h_{in}\right)}{\eta} = 32.931 kW$$

Appendix G - Calculations of preheater 2 properties

$T_{prim\_in} := 176°C$

$T_{prim\_out} := 265°C$

$m_{f\_prim} := 0.5\dfrac{kg}{s}$

$p_{prim} := 60bar$

$T_{sec\_in} := 350°C$

$p_{sec} := 60bar$

$\eta := 1$

$h_{prim\_out} := h\_ptx\_97\left(\dfrac{p_{prim}}{bar}, \dfrac{T_{prim\_out}}{°C}, -1\right)\cdot\dfrac{kJ}{kg} = 1.16\times 10^3\cdot\dfrac{kJ}{kg}$

$h_{prim\_in} := h\_ptx\_97\left(\dfrac{p_{prim}}{bar}, \dfrac{T_{prim\_in}}{°C}, -1\right)\cdot\dfrac{kJ}{kg} = 748.228\dfrac{kJ}{kg}$

$T_{sec\_out} := ts\_p\_97\left(\dfrac{p_{sec}}{bar}\right)\cdot°C = 275.586°C$

$h_{sec\_in} := h\_ptx\_97\left(\dfrac{p_{sec}}{bar}, \dfrac{T_{sec\_in}}{°C}, -1\right)\cdot\dfrac{kJ}{kg} = 3.044\times 10^3\cdot\dfrac{kJ}{kg}$

$h_{sec\_out} := h\_ptx\_97\left(\dfrac{p_{sec}}{bar}, \dfrac{T_{sec\_out}}{°C}, 0\right)\dfrac{kJ}{kg} = 1.214\times 10^3\dfrac{kJ}{kg}$

$m_{f\_sec} := m_{f\_prim}\cdot\left[\dfrac{h_{prim\_out} - h_{prim\_in}}{\eta\cdot\left(h_{sec\_in} - h_{sec\_out}\right)}\right] = 0.112\dfrac{kg}{s}$

$Q_{2W02} := m_{f\_prim}\cdot\dfrac{h_{prim\_out} - h_{prim\_in}}{\eta} = 205.677kW$

Appendix H - Calculations of mixing preheater properties

$T_{water\_in} := 265°C$

$T_{steam\_in} := 350°C$

$m_{f\_water\_in} := 0.5\dfrac{kg}{s}$

$p := 60bar$

$h_{water\_in} := h\_ptx\_97\left(\dfrac{p}{bar}, \dfrac{T_{water\_in}}{°C}, -1\right)\cdot\dfrac{kJ}{kg} = 1.16\times 10^3\cdot\dfrac{kJ}{kg}$

$h_{steam\_in} := h\_ptx\_97\left(\dfrac{p}{bar}, \dfrac{T_{steam\_in}}{°C}, -1\right)\cdot\dfrac{kJ}{kg} = 3.044\times 10^3\cdot\dfrac{kJ}{kg}$

$T_{out} := ts\_p\_97\left(\dfrac{p}{bar}\right)\cdot°C = 275.586°C$

$h_{water\_out} := h\_ptx\_97\left(\dfrac{p}{bar}, \dfrac{T_{water\_out}}{°C}, -1\right)\cdot\dfrac{kJ}{kg} = 1.214\times 10^3\cdot\dfrac{kJ}{kg}$

$m_{f\_steam\_in} := m_{f\_water\_in}\cdot\left(\dfrac{h_{water\_in} - h_{water\_out}}{h_{water\_out} - h_{steam\_in}}\right) = 0.015\dfrac{kg}{s}$

$m_{f\_water\_out} := m_{f\_water\_in} + m_{f\_steam\_in} = 0.515\dfrac{kg}{s}$

Appendix I - Calculations of thermal energy storage properties

$T_{in} := 275.586°C$

$T_{out} := 176°C$

$p := 60 \cdot bar$

$m_{f\_MP} := 0.514793 \dfrac{kg}{s}$

$m_{f\_steam\_gen} := 0.127321 \dfrac{kg}{s}$

$m_f := m_{f\_MP} - m_{f\_steam\_gen} = 0.387 \dfrac{kg}{s}$

$h_{in} := h\_ptx\_97\left(\dfrac{p}{bar}, \dfrac{T_{in}}{°C}, -1\right) \cdot \dfrac{kJ}{kg} = 1.214 \times 10^3 \cdot \dfrac{kJ}{kg}$

$h_{out} := h\_ptx\_97\left(\dfrac{p}{bar}, \dfrac{T_{out}}{°C}, -1\right) \cdot \dfrac{kJ}{kg} = 748.228 \dfrac{kJ}{kg}$

$P_{TES} := m_f \cdot \left(h_{in} - h_{out}\right) = 180.369 kW$

Appendix J - Calculations of heat sink 1 properties

$$h_{2B01} := 748.228 \frac{kJ}{kg}$$

$$m_{f\_2B01} := 0.387 \frac{kg}{s}$$

$$h_{2W02} := 1.214 \times 10^3 \frac{kJ}{kg}$$

$$m_{f\_2W02} := 0.112384 \frac{kg}{s}$$

$$h_{1W03} := 3.044 \times 10^3 \cdot \frac{kJ}{kg}$$

$$m_{f\_1W03} := 0.127321 \frac{kg}{s}$$

$$m_{f\_2W01} := 0.014793 \frac{kg}{s}$$

$$m_{f\_steam} := m_{f\_1W03} - m_{f\_2W02} - m_{f\_2W01} = 1.44 \times 10^{-4} \frac{kg}{s}$$

$$T_{vessel\_max} := 90^\circ C$$

$$p_{vessel} := 1.013 bar$$

$$m_{f\_vessel\_in} := m_{f\_2B01} + m_{f\_2W02} + m_{f\_steam} = 0.5 \frac{kg}{s}$$

$$h_{vessel\_out} := h\_ptx\_97 \left(\frac{p_{vessel}}{bar}, \frac{T_{vessel\_max}}{^\circ C}, -1 \right) \cdot \frac{kJ}{kg} = 376.993 \frac{kJ}{kg}$$

$$P_{in} := \left(m_{f\_2B01} \cdot h_{2B01} \right) + \left(m_{f\_2W02} \cdot h_{2W02} \right) + \left(m_{f\_steam} \cdot h_{1W03} \right) = 426.437 kW$$

$$P_{cooler} := P_{in} - \left(m_{f\_vessel\_in} \cdot h_{vessel\_out} \right) = 238.118 kW$$

$$T_{primary\_in} := 70^\circ C$$

$$T_{primary\_out} := 85^\circ C$$

$$p_{primary\_out} := 11.5 bar$$

$$T_{secondary\_in} := T_{vessel\_max} = 90 \cdot ^\circ C$$

$$p_{secondary\_in} := 1.013 bar$$

$$h_{primary\_in} := h\_ptx\_97 \left(\frac{p_{primary\_out}}{bar}, \frac{T_{primary\_in}}{^\circ C}, -1 \right) \cdot \frac{kJ}{kg} = 293.933 \frac{kJ}{kg}$$

$$h_{primary\_out} := h\_ptx\_97 \left(\frac{p_{primary\_out}}{bar}, \frac{T_{primary\_out}}{^\circ C}, -1 \right) \cdot \frac{kJ}{kg} = 356.803 \frac{kJ}{kg}$$

$$m_{f\_primary\_and\_secondary} := \frac{P_{cooler}}{h_{primary\_out} - h_{primary\_in}} = 3.787 \frac{kg}{s}$$

Appendix K - Calculations of heat sink 2 properties

$T_{in} := 186.05\,°C$

$p_{in} := 11.5\,bar$

$p_{out} := 1.013\,bar$

$$h_{in} := h\_ptx\_97\left(\frac{p_{in}}{bar}, \frac{T_{in}}{°C}, -1\right)\cdot\frac{kJ}{kg} = 789.986\frac{kJ}{kg}$$

$$T_{out} := t\_ph\_97\left(\frac{p_{out}}{bar}, h_{in}\cdot\frac{kg}{kJ}\right)\cdot°C = 99.967\,°C$$

$T_{vessel\_in} := 99.967\,°C$

$h_{vessel\_in} := 789.986\frac{kJ}{kg}$

$m_{f\_vessel\_in} := 0.18\frac{kg}{s}$

$T_{vessel\_max} := 90\,°C$

$p_{vessel} := 1.013\,bar$

$$h_{vessel\_out} := h\_ptx\_97\left(\frac{p_{vessel}}{bar}, \frac{T_{vessel\_max}}{°C}, -1\right)\cdot\frac{kJ}{kg} = 376.993\frac{kJ}{kg}$$

$$P_{cooler} := m_{f\_vessel\_in}\cdot\left(h_{vessel\_in} - h_{vessel\_out}\right) = 74.339\,kW$$

$T_{primary\_in} := 70\,°C$

$T_{primary\_out} := 85\,°C$

$p_{primary\_out} := 11.5\,bar$

$T_{secondary\_in} := T_{vessel\_max} = 90\,°C$

$p_{secondary\_in} := 1.013\,bar$

$$h_{primary\_in} := h\_ptx\_97\left(\frac{p_{primary\_out}}{bar}, \frac{T_{primary\_in}}{°C}, -1\right)\cdot\frac{kJ}{kg} = 293.933\frac{kJ}{kg}$$

$$h_{primary\_out} := h\_ptx\_97\left(\frac{p_{primary\_out}}{bar}, \frac{T_{primary\_out}}{°C}, -1\right)\cdot\frac{kJ}{kg} = 356.803\frac{kJ}{kg}$$

$$m_{f\_primary\_and\_secondary} := \frac{P_{cooler}}{h_{primary\_out} - h_{primary\_in}} = 1.182\frac{kg}{s}$$

Appendix L – Datasheet for heat exchanger 3W01

Spezifikationsblatt

Projekt: 3W01   Datum: 29.08.2013

Plattenwärmeaustauscher, geschraubt   Version: 4.08g.5.1f

Pos.:	Gerät: FP 22-61-1-NH				

Technische Daten	warm			kalt	
Medien	Water			Water	

			warm		kalt
Massenstrom	[kg/s]	7.66		[kg/s]	7.66
Volumenstrom	[m³/h]	28.435		[m³/h]	28.369
Eintrittstemperatur	[°C]	90.00		[°C]	70.00
Austrittstemperatur	[°C]	75.00		[°C]	85.00
Arbeitsdruck	[kPag]	0.00		[kPag]	0.00

Physikalische Stoffeigenschaften		Eintritt	Austritt	Eintritt	Austritt
Dyn. Viskosität	[cP]	0.313	0.378	0.404	0.333
Dichte	[kg/m³]	964.8	973.7	976.4	967.9
Wärmekapazität	[kJ/kgK]	4.184	4.176	4.174	4.181
Wärmeleitfähigkeit	[W/mK]	0.656	0.651	0.649	0.655

Wärmeaustauscher Kenndaten					
Wärmeleistung	[kW]		480.00		
Übertragungsfläche	[m²]		12.39		
Log./Effektive Temperaturdifferenz	[K]		5.00 / 5.00		
Wärmedurchgangszahl, erf./vorh.	[W/m²K]		7748 / 7767		
Verschmutzungsfaktor gesamt	[m²K/W]		0.0000003		
Flächenreserve	[%]		0.24		
Druckverlust	[kPa]	49.212		[kPa]	48.878
Spaltgeschwindigkeit	[m/s]	0.46 / 0.98		[m/s]	0.46 / 0.97
Stutzengeschw.	[m/s]	3.32		[m/s]	3.32
Anzahl der Wege in Serie		1			1
Anzahl der Spalten gesamt			60		
Spalttypen			24*HH + 6*HL		

Konstruktion					
Leergewicht/Füllgewicht	[kg]		ca. 287/312		
Inhalt	[dm³]	13.200			13.200
Auslegungsdruck	[barg]		25.0		
Max. Betriebstemperatur	[°C]		110.00		

Material		DIN			
	Platten	AISI 316L / 1.4404 (0.60 mm)			
	Dichtungen	NBR HT			
	Gestell	S 355 J2+N (Druckplatten), max.Anz.Platten: 110, Erweiterbarkeit 80%			
	Lackierung	C2, RAL 5012 Standard, <= 150°C, Osnapox Z 1K sg (80µm)			

Anschluss	warm	F1 => F4 : Vorschweißflansch, DN 50, C-Stahl, C 22.8, PN 40, DIN 2635			
	kalt	F3 => F2 : Vorschweißflansch am Rohrbogen, DN 50, C-Stahl, C 22.8, PN 40, DIN 2635			

Appendix M – Datasheet for heat exchanger 3W02

DieTerHoeven GmbH
Am Mühlenberg 11
47638 Straelen
Fon +49 (0) 2839-5686-84
E-Mail info@dieterhoeven.de

Spezifikationsblatt

Projekt: 3W02   Datum: 29.08.2013

Plattenwärmeaustauscher, geschraubt   Version: 4.08g.5.1f

Pos.:		Gerät: FP 22-39-1-NH			

Technische Daten		warm		kalt	
Medien		Water		Water	
Massenstrom	[kg/s]	4.79		[kg/s]	4.79
Volumenstrom	[m³/h]	17.772		[m³/h]	17.731
Eintrittstemperatur	[°C]	90.00		[°C]	70.00
Austrittstemperatur	[°C]	75.00		[°C]	85.00
Arbeitsdruck	[kPag]	0.00		[kPag]	0.00

Physikalische Stoffeigenschaften		Eintritt	Austritt	Eintritt	Austritt
Dyn. Viskosität	[cP]	0.313	0.378	0.404	0.333
Dichte	[kg/m³]	964.8	973.7	976.4	967.9
Wärmekapazität	[kJ/kgK]	4.184	4.176	4.174	4.181
Wärmeleitfähigkeit	[W/mK]	0.656	0.651	0.649	0.655

Wärmeaustauscher Kenndaten					
Wärmeleistung	[kW]		300.00		
Übertragungsfläche	[m²]		7.77		
Log./Effektive Temperaturdifferenz	[K]		5.00 / 5.00		
Wärmedurchgangszahl, erf./vorh.	[W/m²K]		7722 / 7843		
Verschmutzungsfaktor gesamt	[m²K/W]		0.0000020		
Flächenreserve	[%]		1.56		
Druckverlust	[kPa]	46.369		[kPa]	45.969
Spaltgeschwindigkeit	[m/s]	0.47 / 1.00		[m/s]	0.47 / 0.99
Stutzengeschw.	[m/s]	2.08		[m/s]	2.07
Anzahl der Wege in Serie		1			1
Anzahl der Spalten gesamt			38		
Spalttypen			16*HH + 3*HL		

Konstruktion					
Leergewicht/Füllgewicht	[kg]		ca. 247/264		
Inhalt	[dm³]	8.360			8.360
Auslegungsdruck	[barg]		25.0		
Max. Betriebstemperatur	[°C]		110.00		

Material		DIN	
	Platten	AISI 316L / 1.4404 (0.60 mm)	
	Dichtungen	NBR HT	
	Gestell	S 355 J2+N (Druckplatten), max.Anz.Platten: 60, Erweiterbarkeit 54%	
	Lackierung	C2, RAL 5012 Standard, <= 150°C, Osnapox Z 1K sg (80µm)	

Anschluss	warm	F1 => F4 : Vorschweißflansch, DN 50, C-Stahl, C 22.8, PN 40, DIN 2635	
	kalt	F3 => F2 : Vorschweißflansch am Rohrbogen, DN 50, C-Stahl, C 22.8, PN 40, DIN 2635	

Appendix N – Delivery head characteristics for the cooling pumps
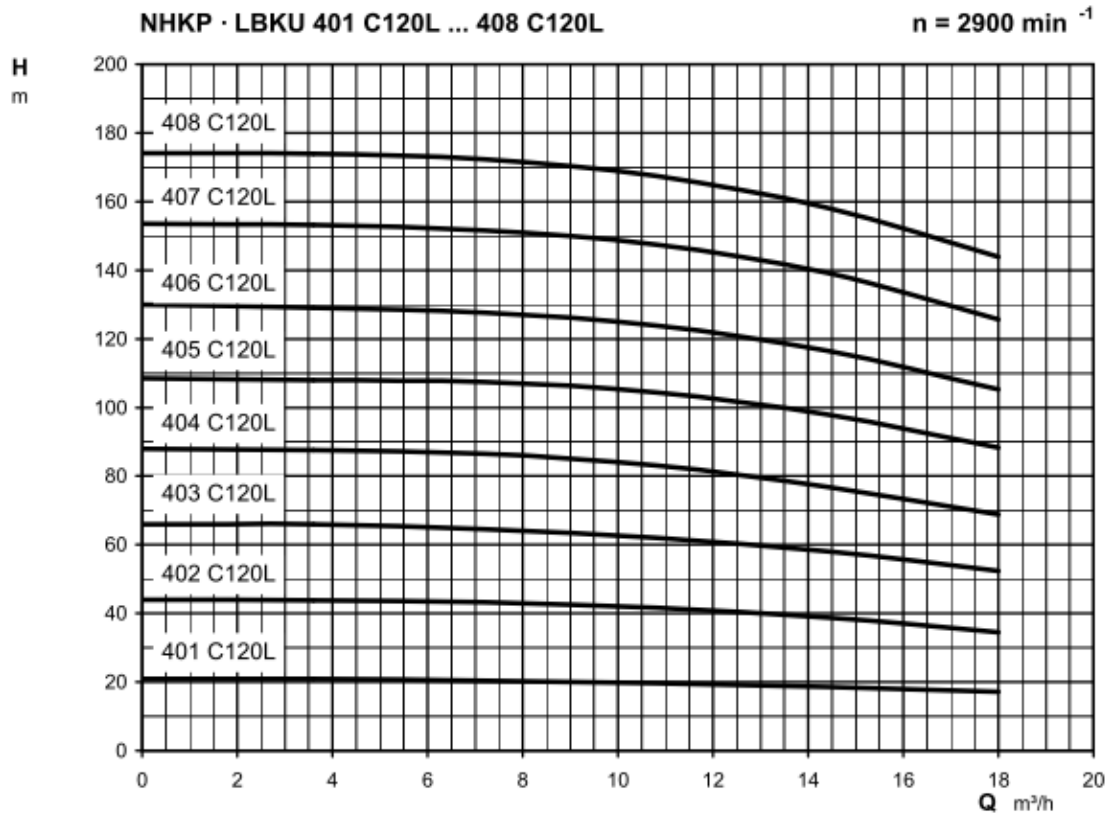
Pump 3P01 (LBU601E162L/7,5 kW)
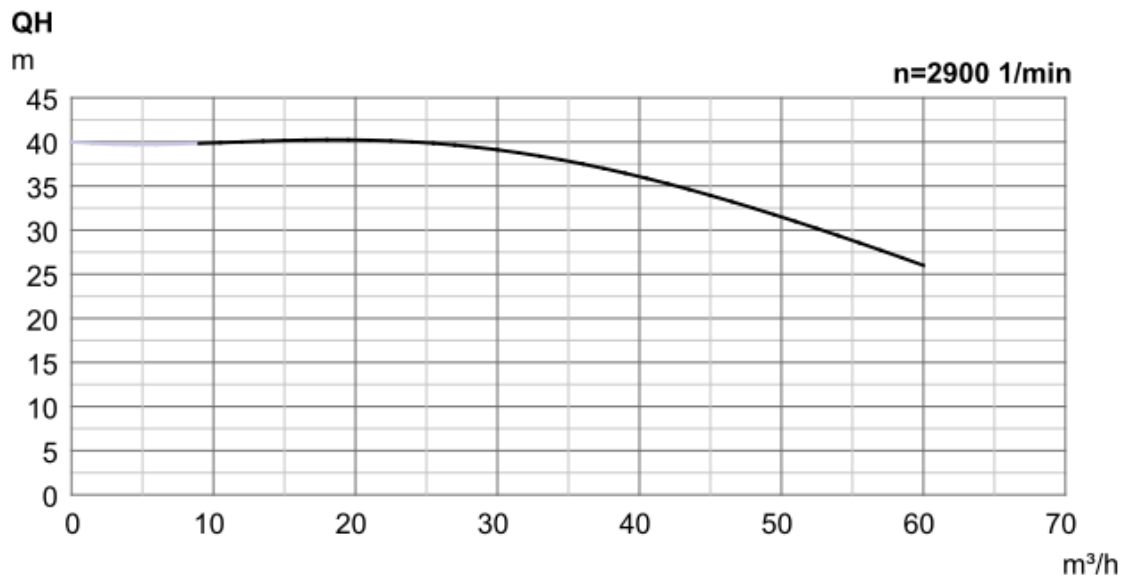
Pump 3P02 (LBU602D160L/11 kW)

Pump 3P03 (LBKU 401 C120L)

NHKP · LBKU 401 C120L ... 408 C120L   n = 2900 min⁻¹
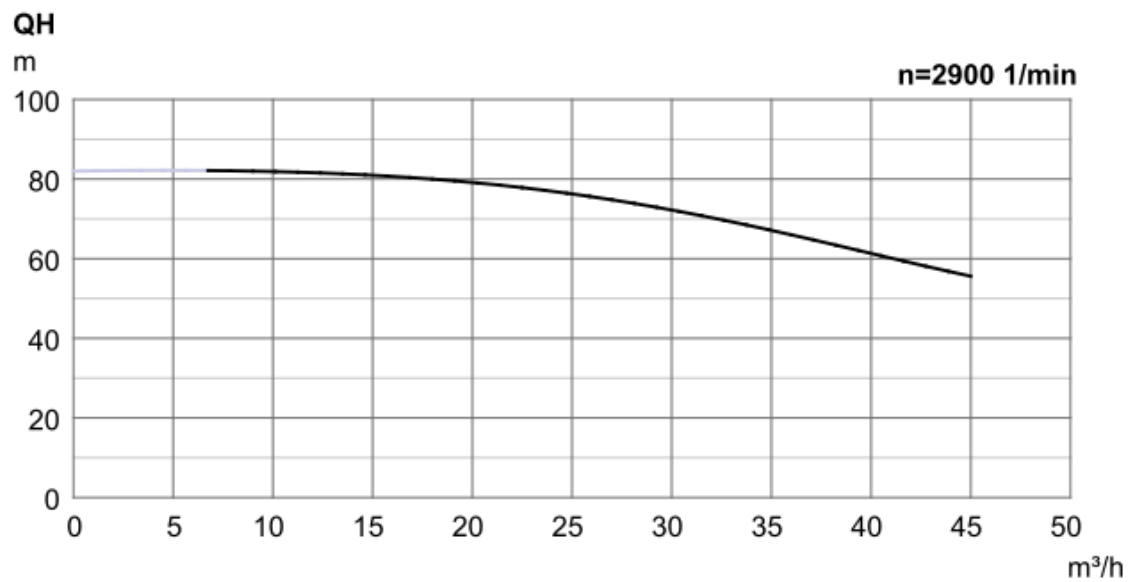
Pump 3P04 (LBU403B150L/7,5 kW)