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DIPLOMA THESIS

Steam Tail – Optimization of cycle between HRSG and steam turbine in combined cycle power plants

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Zásady pro vypracování:

1. Description of the combined cycle power plants.
2. Detailed description of the optimized parts of the steam turbine and HRSG.
3. Description of optimization and its calculation.
4. Impact of the optimized components on the improvement of the power cycle.

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- **Heinz P. Bloch, Murari P. Singh: Steam Turbine: design, application and reratin. 2nd ed. New York: MeGraw-Hill, 2009, 433 s., ISBN 978-0-07-164100-5**
- **Kehlhofer R.: Combined-cycle power gas & steam turbine power plants Tulsa: PenWell, 1997, 388 s., ISBN 0-87814-736-5**

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V Plzni dne 3. října 2016

Prohlášení o autorství

Předkládám tímto k posouzení a obhajobě diplomovou práci zpracovanou na závěr studia na Fakultě strojní Západočeské univerzity v Plzni.
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Declaration of authorship

By this I submit the diploma thesis written at the end of my studies at Faculty of Mechanical Engineering of the University of West Bohemia to be revisited and defensed.
I proclaim that I have done this thesis on my own, with the use of literature and other sources listed inside the thesis.

In Pilsen, 2^{nd} June 2017. ....

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CELKEM	85	**TEXTOVÁ ČÁST**	60	**GRAFICKÁ ČÁST**	25

STRUČNÝ POPIS (MAX 10 ŘÁDEK) **ZAMĚŘENÍ, TÉMA, CÍL POZNATKY A PŘÍNOSY**	Následující text je zaměřen na optimalizaci referenční paroplynové elektrárny. Optimalizace je dělána z pohledu parního cyklu, který je nabízen výrobcem parních turbín jako „přednavržený“ balíček. Navržené úpravy berou v potaz jak ekonomickou stránku, tak i výkon (účinnost) elektrárny. Výkon parní turbíny byl zvýšen použitím dvoutlakého kondenzátoru, který nahradil původní jednotlaký. Snížení ceny celého projektu bylo dosaženo pomocí optimalizace dispozic strojovny.
KLÍČOVÁ SLOVA **ZPRAVIDLA JEDNOSLOVNÉ POJMY, KTERÉ VYSTIHUJÍ PODSTATU PRÁCE**	paroplynová elektrárna, parní cyklus, spalinový kotel, kondenzátor, dispozice, strojovna

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BRIEF DESCRIPTION **TOPIC, GOAL, RESULTS AND CONTRIBUTIONS**	Following pages are focused on optimization of a reference combined cycle power plant. Optimization is done from point of view of bottoming (steam) process, which is being actively offered by steam turbine manufacturer as pre-engineered package. Suggested improvements take into account both economic part of project and power output of plant. Power output of the steam turbine was improved by using a double-pressure condenser which replaces a single-pressure one. Cost reduction of the project was achieved by optimizing the layout of machine hall.
KEY WORDS	combined cycle power plant, steam tail, hrsg, condenser, layout, machine hall

Contents

Nomenclature

Quantity	**Unit**	**Name**
A	$[\mathrm{m}^2]$	Surface area
A_E	$[\mathrm{m}^2]$	Exhaust area
A_T	$[\mathrm{m}^2]$	Inside flow tube area
c_{pw}	$\left[\frac{\mathrm{kJ}}{\mathrm{kg}\cdot\mathrm{K}}\right]$	Specific heat of cooling water
d_i	$[\mathrm{m}]$	Internal diameter
d_o	$[\mathrm{m}]$	Outside diameter
F_C	$[-]$	Tube cleanliness correction factor
F_M	$[-]$	Tube material and gauge correction factor
F_W	$[-]$	Cooling water inlet temperature correction factor
h	$\left[\frac{\mathrm{kJ}}{\mathrm{kg}}\right]$	Specific enthalpy drop
h_1	$\left[\frac{\mathrm{kJ}}{\mathrm{kg}}\right]$	Specific enthalpy before last stage
h_2	$\left[\frac{\mathrm{kJ}}{\mathrm{kg}}\right]$	Specific enthalpy behind last stage
h'_c	$\left[\frac{\mathrm{kJ}}{\mathrm{kg}}\right]$	Condensing specific enthalpy
K	$[\mathrm{m}]$	Roughness of tubes
L_t	$[\mathrm{m}]$	Length of tubes
$\dot{m}_s$	$\left[\frac{\mathrm{kg}}{\mathrm{s}}\right]$	Mass flow of steam
$\dot{m}_w$	$\left[\frac{\mathrm{kg}}{\mathrm{s}}\right]$	Mass flow of cooling water
N_{el}	$[\mathrm{W}]$	Electric power output
n_t	$[-]$	Number of tubes
P	$[\mathrm{W}]$	Power
P_{LS}	$[\mathrm{W}]$	Power of last stage
p	$[\mathrm{bar(a)}]$	Pressure (absolute)
p_1	$[\mathrm{bar(a)}]$	Pressure (absolute) before last stage
p_2	$[\mathrm{bar(a)}]$	Pressure (absolute) behind last stage
p_c	$[\mathrm{bar(a)}]$	Condensing pressure (absolute)
p_w	$[\mathrm{bar}]$	Pressure of cooling water
Δp	$[\mathrm{bar(a)}]$	Pressure drop
$\dot{Q}$	$[\mathrm{W}]$	Heat flow
$\dot{Q}_{in}$	$[\mathrm{W}]$	Supplied heat flow
$\dot{Q}_{out}$	$[\mathrm{W}]$	Taken out heat flow
Q_V	$\left[\frac{\mathrm{m}^3}{\mathrm{s}}\right]$	Volumetric flow rate
Re	$[-]$	Reynolds number
s	$\left[\frac{\mathrm{kJ}}{\mathrm{kg}\cdot\mathrm{K}}\right]$	Specific entropy
t_c	$[^\circ\mathrm{C}]$	Condensing temperature
t_{w1}	$[^\circ\mathrm{C}]$	Inlet temperature of cooling water
t_{w2}	$[^\circ\mathrm{C}]$	Outlet temperature of cooling water
t_{wm}	$[^\circ\mathrm{C}]$	Mean temperature of cooling water
T_H	$[^\circ\mathrm{C}]$	Higher temperature
T_L	$[^\circ\mathrm{C}]$	Lower temperature
U_1	$\left[\frac{\mathrm{W}}{\mathrm{m}^2\cdot\mathrm{K}}\right]$	Uncorrected heat transfer coefficient
U	$\left[\frac{\mathrm{W}}{\mathrm{m}^2\cdot\mathrm{K}}\right]$	Heat transfer coefficient

v	$\left[\frac{\mathrm{m}^3}{\mathrm{kg}}\right]$	Specific volume
w	$\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$	Velocity
W	$[\mathrm{W}]$	Work
x	$[-]$	Steam dryness fraction
ζ	$[-]$	Pressure drop coefficient
η	$[-]$	Efficiency
η_C	$[-]$	Carnot efficiency
η_t	$[-]$	Thermal efficiency
η_{td}	$[-]$	Thermodynamic efficiency
λ_{fr}	$[-]$	Friction coefficient
ν	$\left[\frac{\mathrm{m}^2}{\mathrm{s}}\right]$	Kinematic viscosity
π	$[\mathrm{USD}]$	Economic profit
ρ	$\left[\frac{\mathrm{kg}}{\mathrm{m}^3}\right]$	Density

Abbreviations

CCPP	Combined cycle power plant
G	Generator
GE	General Electric
GT	Gas turbine
HP	High-pressure
HRSG	Heat recovery steam generator
IP	Intermediate-pressure
ISO	International Organization for Standardization
ITD	Initial temperature difference
LP	Low-pressure
SF	Supplementary firing
ST	Steam turbine
TR	Temperature rise
TTD	Terminal temperature difference

Introduction

Modern world requires still more and more energy due to the increase in consumption. Especially it is electrical energy that is being requested. There is of course an effort (and due to strict emission standards a necessity) to raise the efficiency as much as possible and to optimize these sources of power from the economic point of view. A predominant portion of this kind of energy is generated by steam turbines. If we consider classic thermal (fossil, biomass) power plants, it is not possible to attain as high efficiency as in a combination of two thermal cycles. This fact is successfully used in combined cycle power plants (CCPP) which combine gas turbine and steam turbine cycles. From the energetic point of view it is not reasonable to release hot exhaust gas from the gas turbine directly to the stack. This exhaust gas can be further utilized since it has appropriate parameters for steam generation in a heat recovery steam generator (HRSG). The heat is brought in the cycle at a very high temperature and taken out at a very low temperature (in comparison to a single cycle configuration). This diploma thesis focuses on optimizing a reference steam cycle (steam tail) which is being actively offered to potential clients by the steam turbine manufacturer.

There are many ways to improve the efficiency. From the point of view of thermal efficiency it can be: raising inlet parameters of live steam (temperature and pressure), implementing regeneration and reheating, reducing backpressure, etc. Optimizing the flow path improves thermodynamic efficiency of the turbine. Mechanical efficiency can be improved for example by reducing the friction in bearings or by using electromagnetic bearings (in the future). Above mentioned methods for improving the efficiency usually contributes to economic optimization because additional investment costs can really pay off. Nevertheless, these investment costs can be cut down by optimal material utilization, by saving material, working hours and so forth.

A frequently neglected way to reduce the backpressure (vacuum) in a condenser is to use a double-pressure (or generally multipressure) condenser instead of a single-pressure one. The temperature rise of cooling water is split into sections according to the number of condenser compartments. The condensing pressure in the first compartment is thus lower than the pressure in the second compartment. This provides lower average condensing pressure, longer expansion of steam and higher power output (efficiency), even if the original heat exchange surface is preserved.

By optimizing the layout of a power plant, a huge amount of investment cost can be saved. This thesis shows the influence of flattening a machine hall. It can be done by changing a downward exhaust from a steam turbine to a lateral or axial one.

Firstly, the reader is introduced into the issue of combined cycle power plants. The fundamentals of thermodynamic cycles and their combination are delineated. The next part deals with detailed description of combined cycle power plants and their major parts. Examples of various arrangements of these plants and their comparison are mentioned as well. Then the actual triple-pressure system to be optimized is described. The ways to optimize the reference steam tail are listed in the fourth section. The power gain of using a double-pressure condenser and the comparison of individual options are calculated. Changes in the layout of the reference plant are described too. Finally, a technical and economic evaluation of suggested improvements is done.

1 Basic considerations

1.1 Fundamentals of thermodynamic cycles

1.1.1 Work and thermal efficiency of thermodynamic cycle

Thermodynamic cycle is a series of changes returning a thermodynamic system to its default thermodynamic state. Generally the heat is supplied at one end and taken out at the other end of the cycle. The difference between the heat supplied and taken out of the cycle expresses the work (1).

$$W = \dot{Q}_{in} - \left|\dot{Q}_{out}\right| \tag{1}$$

Thermal efficiency is defined by the quotient of work and supplied heat.

$$\eta_t = \frac{W}{\dot{Q}_{in}} = \frac{\dot{Q}_{in} - \left|\dot{Q}_{out}\right|}{\dot{Q}_{in}} = 1 - \frac{\left|\dot{Q}_{out}\right|}{\dot{Q}_{in}} \tag{2}$$

The main target is to reach as high thermal efficiency as possible.

1.1.2 Carnot efficiency

Carnot cycle thermal efficiency is defined by the equation (3).

$$\eta_C = \frac{T_H - T_L}{T_H} \tag{3}$$

As can be seen in Figure1 the Carnot cycle starts with isothermic expansion followed by adiabatic expansion.

This is an ideal process which is not realizable in real cycle. This is mainly caused by energetic and exergetic losses. Energetic losses are heat losses, such as radiation and convection. Exergetic losses are associated with the Second Law of Thermodynamics. Throttling of steam can be a typical example.

That is the major reason for combining cycles. Heat potential of hot exhaust gas from the gas turbine is irretrievably lost if it is not further utilized in a heat recovery steam generator (HRSG). The parameters of live steam are not high compared to conventional fossil power plants, however, the gas leaving the stack has very low temperature level. Achievable thermal efficiency of the whole combined cycle power plant is therefore about 60%, whereas in case of single steam turbine cycle it is roughly 50%, and gas turbine can achieve 40%.

1.1.3 Brayton cycle

This thermodynamic cycle described by George Bailey Brayton consists of two adiabatic processes and two isobaric processes. Compression of air (1-2) is followed by isobaric heat intake (2-3). Hot gas from the combustion chamber then expands in the gas turbine (3-4) to the atmosphere. Figure 2, respectively Figure 3 shows the process of thermodynamic quantities in such a cycle. The dashed line represents a cycle open to the atmosphere.

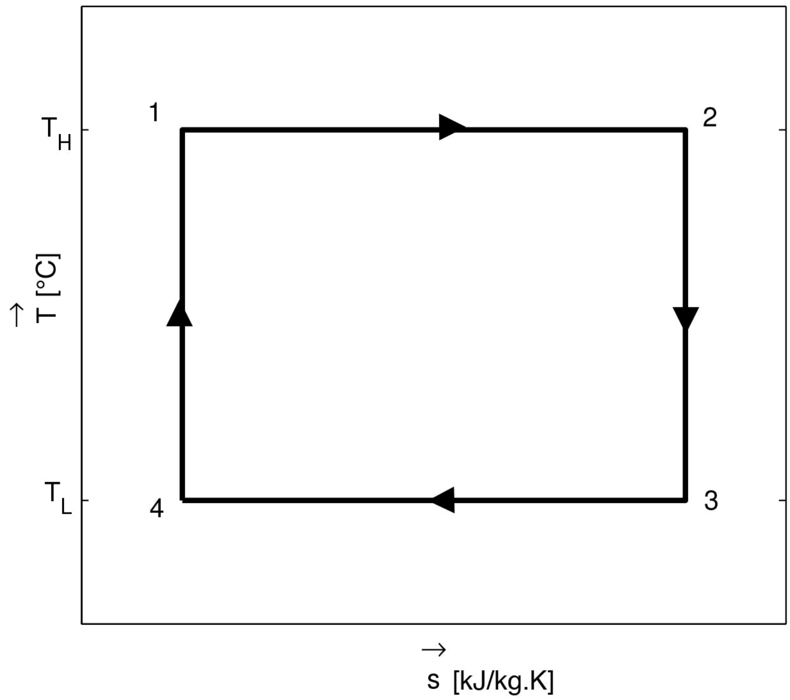

Figure 1: T-s diagram of Carnot cycle

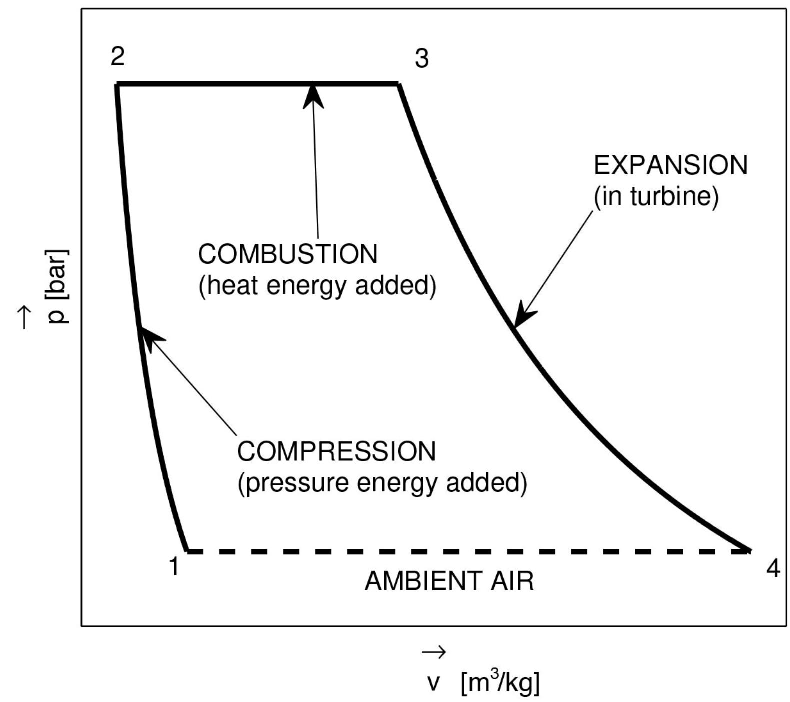

Figure 2: p-v diagram of Brayton cycle

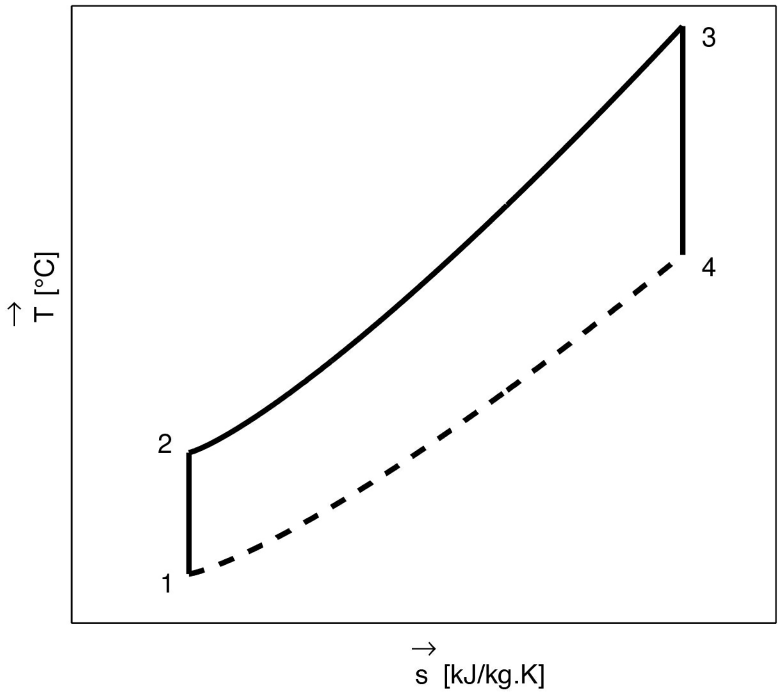

Figure 3: T-s diagram of Brayton cycle

1.1.4 Rankine cycle

Rankine cycle is a model for the calculation of performance and efficiency of thermal engine using water and steam as working fluid (Figure 4). Every single thermal power plant using a steam turbine works on this principle.

Superheated or saturated (nuclear power plants) water steam expands in a steam turbine (1-2) while it is generating mechanical work. At the end of a steam turbine the water steam condenses in a condenser (2-3). Formed condensate is then pressurized by feed water pumps (3-4) and enters a boiler where it is heated (4-5), evaporated (5-6) and superheated (6-1). The whole process is repeated again and again in this closed cycle.

For increasing the thermal efficiency and thus for approaching the Carnot cycle, a regeneration can be implemented. Part of the water steam extracted from the steam turbine heats the condensate in low pressure heaters (LPHs) and high pressure heaters (HPHs). Another possible way to improve the thermal efficiency is reheating. Partly expanded water steam in the steam turbine is reheated in the boiler to avoid the moisture and get additional heat potential.

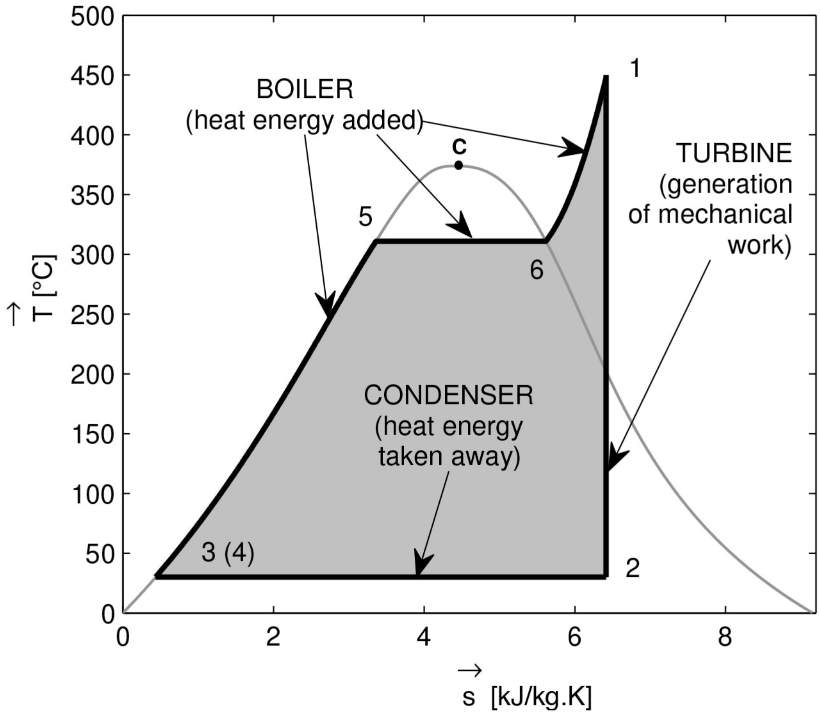

Figure 4: T-s diagram of Rankine cycle

1.2 Combining thermodynamic cycles

Main idea of combining different thermodynamic cycles is utilizing the heat potential in a fluid leaving the topping cycle which has higher temperature level, in the bottoming cycle having lower temperature level (Figure 5). This fact is used with benefits in coupling a gas turbine as a topping cycle and a steam turbine system as a bottoming cycle. There are not any extra costs for developing these machines. Gas turbines are standardized machines that had been developed from aircraft engines. Steam turbines have been in use for decades as well.

A gas turbine connected to a generator produces electrical energy and hot exhaust gas generates water steam in a heat recovery steam generator. This steam can be used as process steam and/or expands in a steam turbine which generates electrical energy as well, and eventually additional process steam from extraction e.g. for district heating.

Efficiency of a gas turbine is

$$\eta_{GT} = \frac{P_{GT}}{\dot{Q}_{GT}} \tag{4}$$

and efficiency of a steam turbine (including a possibility of supplementary firing) is

$$\eta_{ST} = \frac{P_{ST}}{\dot{Q}_{EG} + \dot{Q}_{SF}}, \tag{5}$$

where

$$\dot{Q}_{EG} \cong \dot{Q}_{GT} \cdot (1 - \eta_{GT}) \,. \tag{6}$$

By combining equations (8), (5) and (6) we obtain gross thermal efficiency of a CCPP:

$$\eta_{CCPP} = \frac{P_{GT} + P_{ST}}{\dot{Q}_{GT} + \dot{Q}_{SF}}. \tag{7}$$

Reducing the total power output by plant auxiliary consumption yields net thermal efficiency:

$$\eta_{CCPP} = \frac{P_{GT} + P_{ST} - P_{aux}}{\dot{Q}_{GT} + \dot{Q}_{SF}}. \tag{8}$$

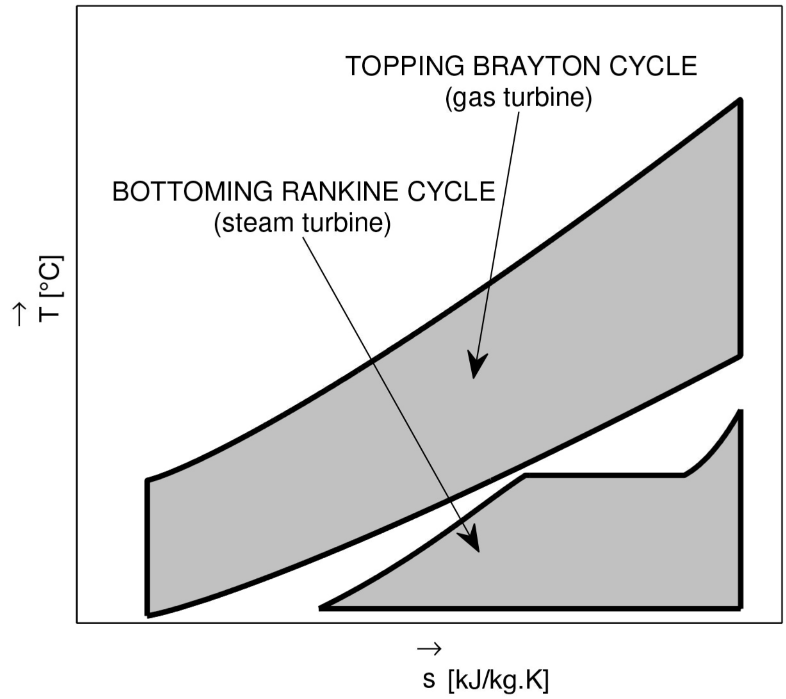

Figure 5: T-s diagram of combined cycle power plant

2 Combined cycle power plants

There are several possible arrangements of these combined cycle power plants (CCPPs) which will be introduced and explained below. A simplified arrangement of basic CCPP is shown in Figure 6. As said above, this type of power plant consists of a gas turbine **4** with a compressor **1** and a combustion chamber **3** connected to a generator. Stack gas from a gas turbine flows to an HRSG **5** where water is heated, evaporated and superheated. Formed water steam proceeds to a steam turbine **6** which also has a generator **7** on the same shaft. Another necessary equipment to close the Rankine cycle is a condenser **8**. Major parts of CCPP will be described below.

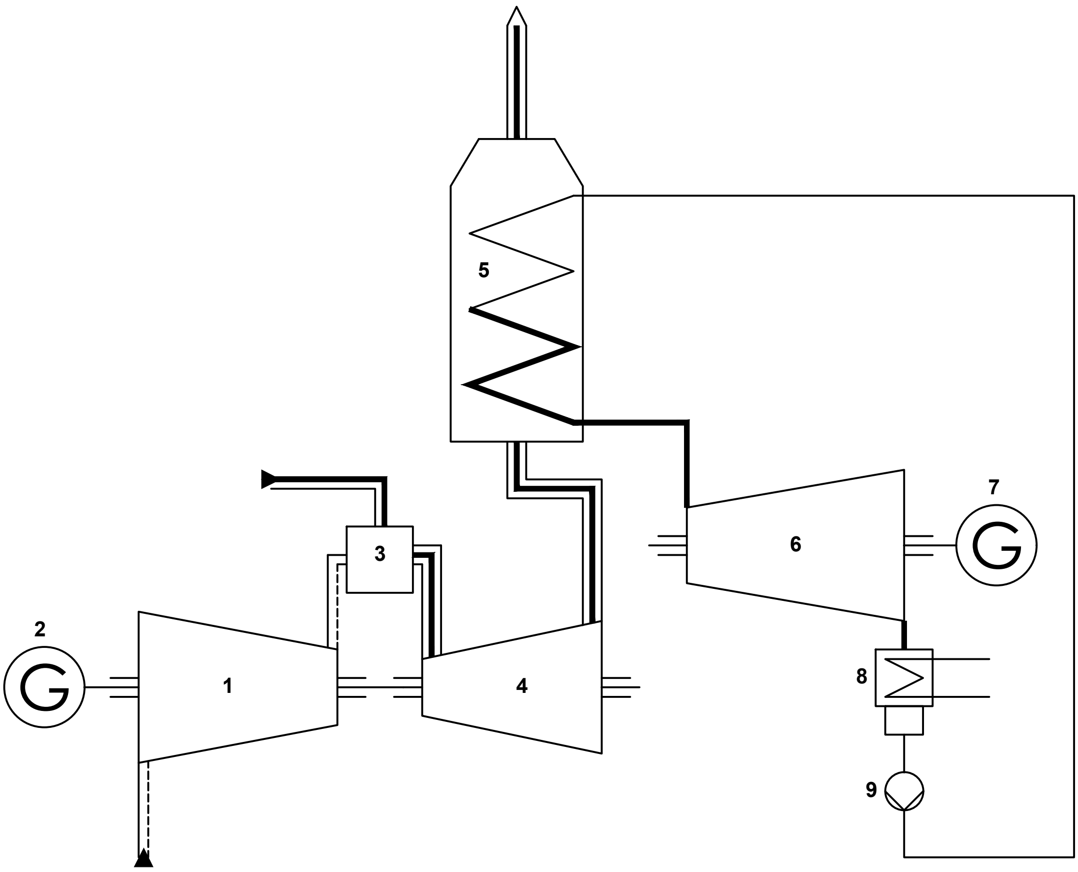

Figure 6: Simplified arrangement of combined cycle power plant

2.1 Major parts of CCPP

2.1.1 Gas turbine

A gas turbine is an internal combustion engine, which changes chemically bonded energy in a fuel into a mechanical energy using turbo-machines. Ambient air flows through a centrifugal or axial compressor which brings it to higher pressure. Heat energy is added in a combustion chamber where is it mixed with fuel and ignited. This gas of high-pressure and high-temperature expands in a gas turbine, producing rotational mechanical energy of a shaft.

Gas turbines can be classified into five groups [2]:

- *Frame type Heavy-Duty gas turbines.* These large power generation units ranging from 3 MW to 480 MW attain efficiencies 30-46 %.
- *Aero-derivative gas turbines.* The origin of these units is in aerospace industry as the prime mover of an aircraft. They have been adapted for a land use and generating electricity by removing the by-pass fans and adding a power turbine at their exhaust. The power range of these units is from 2.5 MW to approx. 50 MW and efficiencies are between 35 and 45 %.
- *Industrial type gas turbines.* These turbines drive compressors in petrochemical plants. With efficiencies in the low 30s they are attaining the power output in the range 2.5-15 MW.
- *Small gas turbines.* Machines with centrifugal compressors and radial inflow turbines. The power range of these turbines is from 0.5-2.5 MW.
- *Micro-turbines.* In the distributed generation market in the late 1990s there was an upsurge of these types of turbines in the range from 20 kW to 350 kW.

This thesis will deal with a member of the first group - Heavy-Duty gas turbine.

2.1.2 Heat recovery steam generator

HRSG is an analogy to a conventional boiler but there is a difference in the way of supplying the heat. If there is no supplementary firing considered, the heat is supplied from a gas turbine exhaust. This heat is transferred over a heat transfer surface which consists of tubes (bare or finned). There are three types of these heat exchangers used in HRSGs:

- economizer,
- evaporator,
- superheater.

Heat distribution between two cycles entails losses. Therefore heat transfer cannot be ideal, since there are three main restrictions:

- physical nature of evaporation at constant temperature,
- impossibility to have an infinitely large heat exchanger,
- danger of low temperature corrosion at the cold end of a heat exchanger.

Even if we consider an infinitely large heat exchanger, which means unlimitedly large heat exchange surface, evaporation at constant temperature excludes thermodynamically optimal utilization of heat energy of waste heat from a gas turbine, as can be seen in Figure 7. It means that an HRSG can never be an ideal heat exchanger. Exhaust gas temperature drops downstream while water temperature contradictory rises.

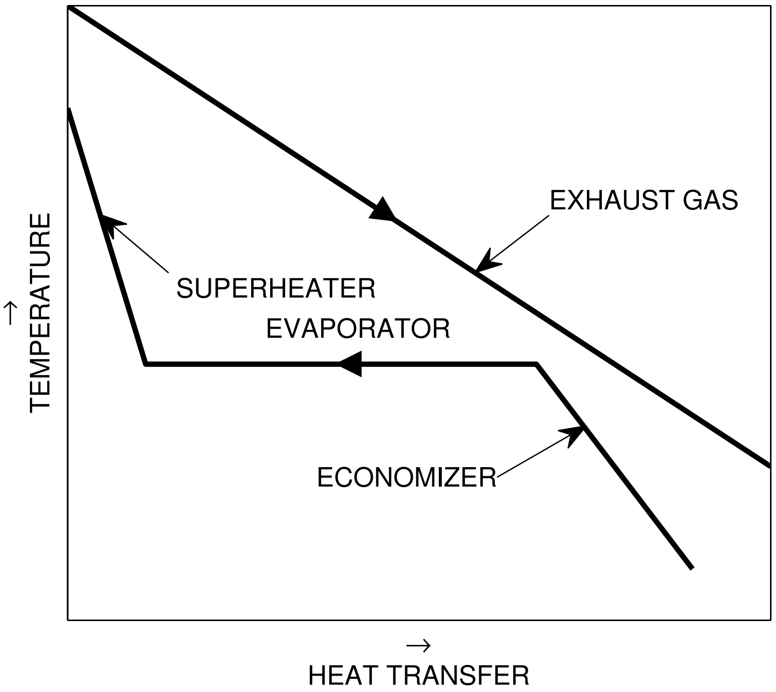

Figure 7: Temperature/heat transfer diagram of HRSG

The decrease in temperature in a HRSG is limited by the low temperature corrosion. This process occurs sulphur when the exhaust gas is cooled below the sulphur acid dewpoint. The surface temperature of water and steam pipes on the exhaust gas side in a HRSG is almost the same as that of water or steam flowing inside. It is due to the fact that the heat exchange on the exhaust gas side is not as good as on the water or steam side. For that reason the temperature of feed water should be never as low as the sulphur acid dewpoint. A similar problem arises even when using sulphur free fuel. If the temperature drops below the water dewpoint, low temperature corrosion can occur as well.

The most significant difference between conventional thermal power plant and steam tail in CCPP is approach to the feed water temperature. In a conventional power plant feed water is preheated in low and high pressure heaters and is conveyed to the a boiler at high temperature level to attain higher efficiency. However, in CCPP the process is utterly inverse. The temperature of the feed water is kept as low as possible, regarding the limit set by low temperature corrosion, but it must not be below the sulphuric acid dewpoint or water dewpoint. This fact has two substantiations:

- There is not any air preheater behind an economizer in the HRSG compared to a boiler in conventional thermal power plants. If the feed water temperature is similarly high, the rate of waste heat utilization is then poor.
- As can be seen in Figure 7 the lowest difference between temperature of the exhaust gas and feed water is in the stage where the evaporation starts. In a conventional boiler this difference is at the other end of the economizer. It

means that the amount of generated steam depends mainly on the feed water temperature since the water flow is much larger in proportion to flue gas flow.

Noteworthy are also parameters called the *pinch point* and the *approach point*. The first mentioned represents the smallest temperature difference between water and exhaust gas. If the pinch point is reduced and the other parameters are fixed, the rate of waste heat utilization is better as well as the amount of generated steam. Nevertheless, by exceeding a certain limit the heat exchange surface increases exponentially. This sets limit of such a utilization. The *approach point* is the difference between the economizer outlet temperature and the saturation temperature in the drum. Its decreasing has the same impact as in case of the *pinch point*. The limit of reducing the *approach point* is set by steaming in economizer [7].

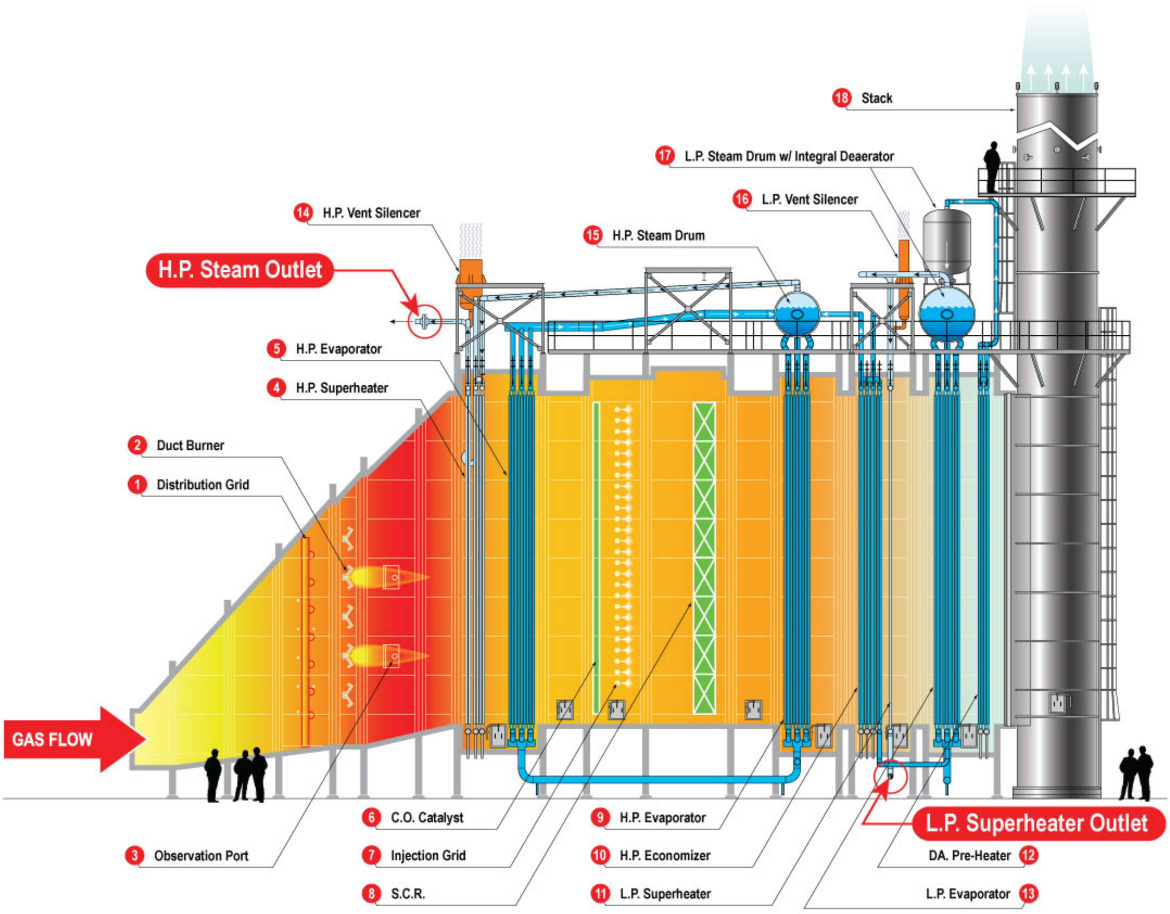

Figure 8: Double-pressure HRSG [3]

2.1.3 Steam turbine

A steam turbine is a thermal rotating machine. The nozzles and diaphragms in a steam turbine are designed to direct the steam flow into well-formed, high-speed jets as the steam expands from inlet to exhaust pressure. These jets strike moving rows of blades mounted on the rotor. The blades convert kinetic energy of the steam into rotation energy of the shaft [1].

According to [14] and [8], steam turbines can be divided from various points of view:

1. according to the field of use:
    - for generating electricity,
    - for propulsion of ships,
    - to drive turbochargers, compressors and other rotating machines,
2. according to the principle of energy conversion:
    - impulse turbines (the pressure before a row of moving blades is almost the same as behind it),
    - reaction turbines (there is a pressure drop on a row of moving blades),
3. according to the number of stages:
    - single-stage (using an impulse or Curtis stage),
    - multistage (impulse staging, reaction staging or a combination of both)
4. according to the direction of steam flow:
    - with axial steam flow (the most commonly used),
    - with radial steam flow,
5. according to the inlet parameters of steam:
    - with saturated steam,
    - with superheated steam,
    - with supercritical parameters,
6. according to the pressure behind last stage:
    - condensing,
    - back pressure (non-condensing),
    - extracting.

The steam turbine which will be dealt with is:

- for generating electricity,
- condensing,
- multistage,
- with combination of impulse and reaction blades,
- axial flow,
- with superheated steam.

The steam turbines used in CCPPs have a few specifics [8]:

- the regenerative heaters (LPHs and HPHs) are usually not used since there is a huge amount of low potential heat for this purpose (the only exception is made in case of restriction by a required higher temperature of the exiting stack gas),
- addition of second and third steam increases the amount of the steam flowing downstream the turbine unlike the steam turbines for fossil fuel units, where the mass flow decreases downstream the flow path due to the extraction of steam for regeneration,
- on the one hand the steam turbine in CCPPs has a lower mass flow of steam in the HP part compared to the fossil fuel units → higher secondary losses,
- on the other hand the mass flow in LP part is higher (again compared to the fossil units), which results in the longer last stage blades or higher leaving velocity losses.

2.1.4 Condenser

Condensation of the steam at the turbine exhaust is an essential condition for realization of the Rankine cycle. The equipment used for condensing the steam at the turbine exhaust and for closing the cycle is called a condenser. A condenser is a large heat exchanger of the shell and tube type. A typical steam surface condenser is shown in Figure 9. Cooling water enters through the waterbox and the tubesheet into the tubes. The shell side of the condenser receives steam from the turbine exhaust. The steam is condensed to a liquid phase by passing over the tubes where the cooling water is circulating. Heat is transferred from the steam to the cooling water [4].

Condensing is not the only function of a condenser. Moreover it provides for example:

- continual venting of non-condensable gases to prevent air binding and the loss of heat transfer capability (non-condensable gases come from air in-leakage),
- conserving the condensate for re-use as feedwater,
- maintaining vacuum at the turbine exhaust, since the vacuum level is given by the temperature of cooling water,
- the draining receptacle for condensate, because the condenser is the lowest pressure point in a steam cycle, it is the most logical collection point for various condensate vents and drains,
- a convenient place for feedwater makeup.

There are several types of condensers used in power plants. The choice depends on the design of the cooling system, plant and turbine configuration, desired temperature rise and cooling water system optimization.

Classification domains (according to [10] and [4]) are:

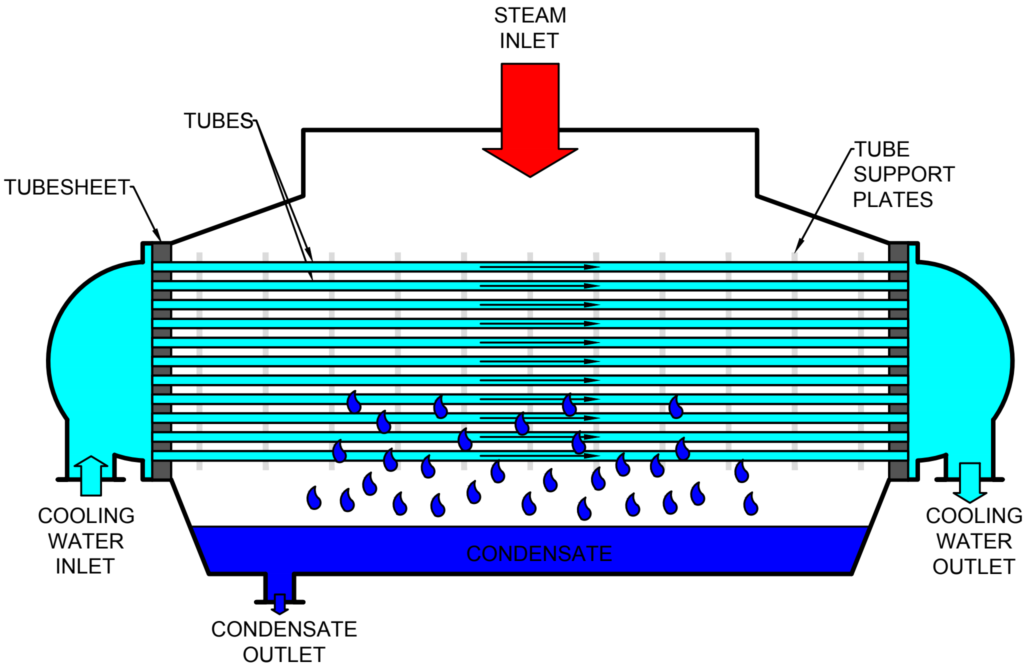

Figure 9: Water-cooled condenser

1. number of compartments:
   - one,
   - two,
   - three,
2. number of tube passes:
   - one,
   - two,
3. distribution of cooling water flow:
   - non divided,
   - divided,
4. orientation of the condenser tubes relative to the turbine axis:
   - transverse,
   - parallel,
5. system of connection of the cooling water:
   - parallel (single-pressure condenser),
   - serial (multipressure condenser),
6. placement of the condenser relative to the turbine:

- radial placement (condenser under the turbine),
- lateral placement (condenser next to the turbine),
- axial placement (condenser behind the turbine).

Further information can be found for instance in [4], [6] and [10]. The original condenser which will be mentioned below is a single-pressure, double-pass, divided one with radial placement. This type of condenser will be replaced by a double-pressure, single-pass, divided one with radial (or lateral) placement.

2.2 Single-pressure system

The simplest possible arrangement of CCPP is a single-pressure system without any additional equipment (Figure 10). There are one or more gas turbines and usually the same number of HRSGs. A steam tail consists of a condensing steam turbine, air or water cooled condenser and deaerator in the feed water tank. The steam for the deaerator is extracted from the turbine.

2.2.1 Example of a single-pressure system

An example of this single-pressure cycle with two GE 7241FA gas turbines was calculated in GT PRO program (part of the Thermoflow software package). As can be seen in Figure 12, the net efficiency of this cycle is 53.33 % and net heat rate is 6750 kJ/kWh. That is the reason why this type of a cycle is not widespread. It is reasonable to split heat exchange surfaces to more pressure levels (two or three) for better utilization of the exhaust heat of gas turbine(s).

2.3 Double-pressure system

Another arrangement of a CCPP is using a double-pressure system. HRSG generates steam in two pressure levels - HP and LP. The LP steam can be added to the steam turbine in a suitable place and/or can be also used for deaerating.

2.3.1 Example of a double-pressure system

This example of the double-pressure system also uses two GE 7241FA gas turbines. The HRSG has - in addition to a single-pressure one - an LP economizer, an LP evaporator and a LP superheater for better heat utilization. This makes it possible to cool down the exhaust gases to 91 °C instead of approx. 143 °C in previous example. The net efficiency reaches 55.2% and the net heat rate is lowered to 6521 kJ/kWh.

2.4 Considerations about various arrangements of CCPP

Table 5 shows different arrangements of a combined cycle power plant. Poor utilization of the waste heat from a gas turbine is improved by using an additional pressure level in a HRSG. The difference in T-Q diagrams is apparent from the comparison of Figure 7 and Appendix A. Adding one additional pressure level brings approx. 2 % higher efficiency of the whole CCPP. One more additional pressure level provides the

Quantity		**Single-press. system**	**Double-press. system**	**Triple-press. system**
Type of GT	[−]	GE 7241FA	GE 7241FA	GE 7241FA
Number of GTs	[−]	2	2	2
N_{el} of ST	[MW]	165.747	184.795	196.473
N_{el} overall	[MW]	499.284	516.794	526.956
Net heat rate	$\left[\frac{\mathrm{kJ}}{\mathrm{kWh}}\right]$	6750	6521	6399
Net efficiency	[%]	53.33	55.2	56.26

Table 1: Comparison of different arrangements of CCPPs

efficiency even by 1 % higher. These values obtained from thermodynamic calculations done in software GT PRO are consistent with the statements in [7] and [9].

The highest efficiencies are attained by using Heavy-Duty combustion turbines, especially representatives of H-class turbines. These types of turbines are optimized for operation in a combined cycle. Heat balance diagram of the triple-pressure cycle with two combustion turbines GE 7HA.02, which is the most powerful turbine available for 60 Hz market, is shown in Figure 14. The efficiency exceeds 60 %, which is almost the current achievable maximum.

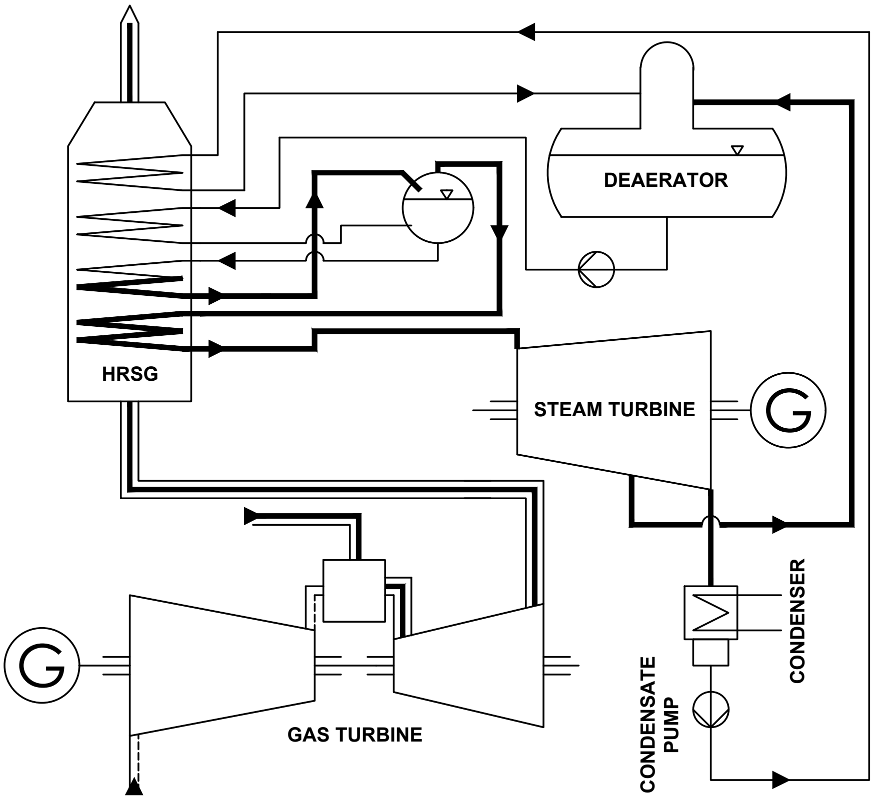

Figure 10: Example of a single-pressure system

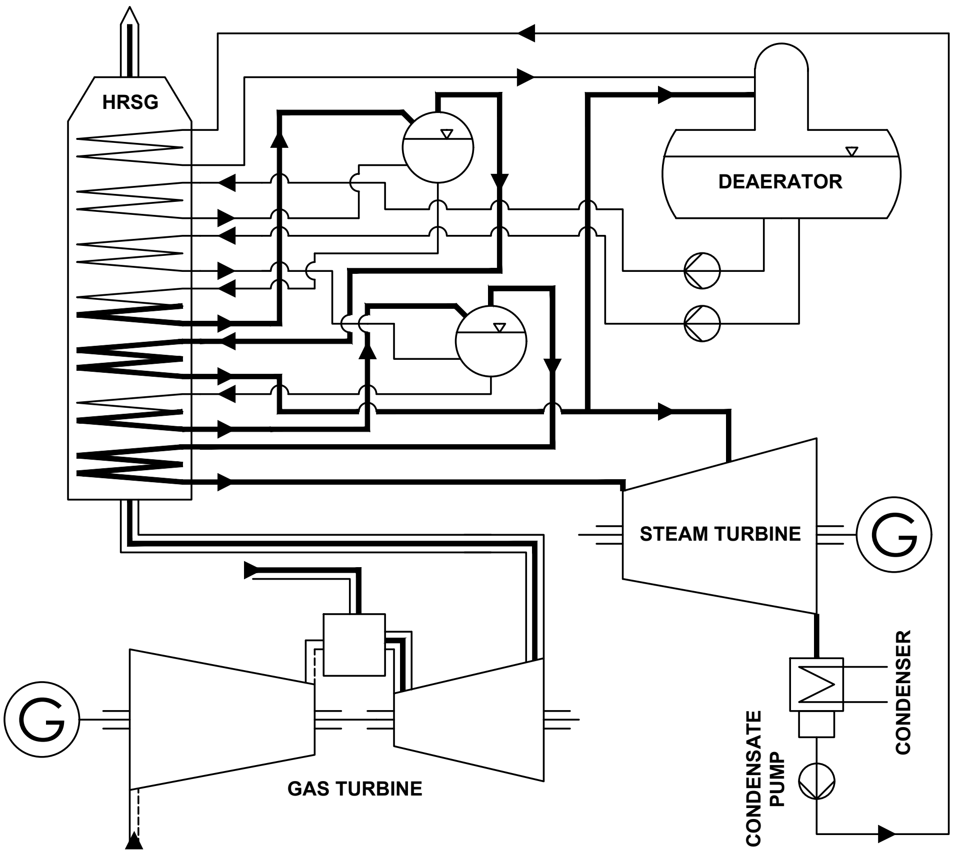

Figure 11: Example of a double-pressure system

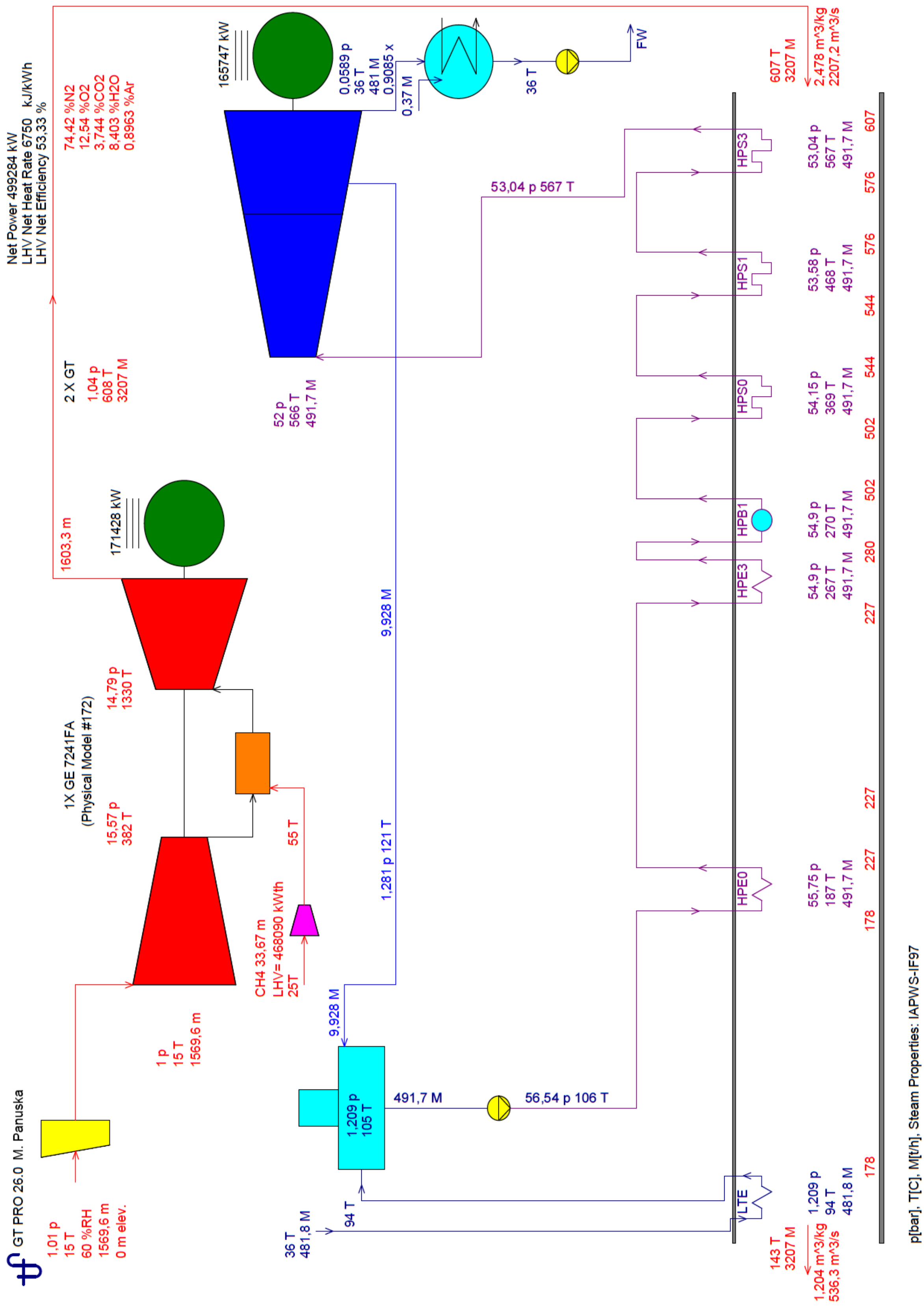

Figure 12: Heat balance of single-pressure cycle

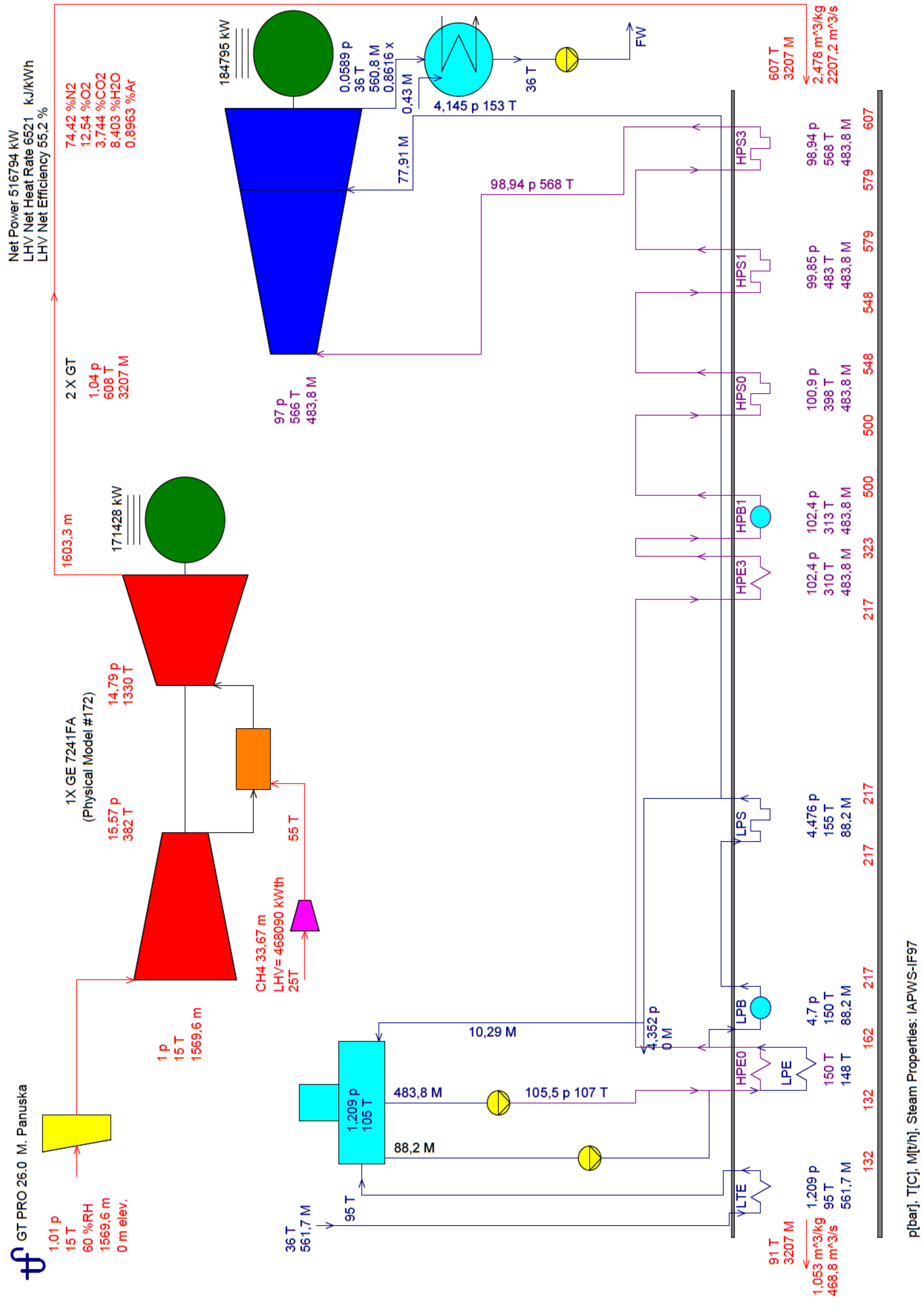

Figure 13: Heat balance of double-pressure cycle

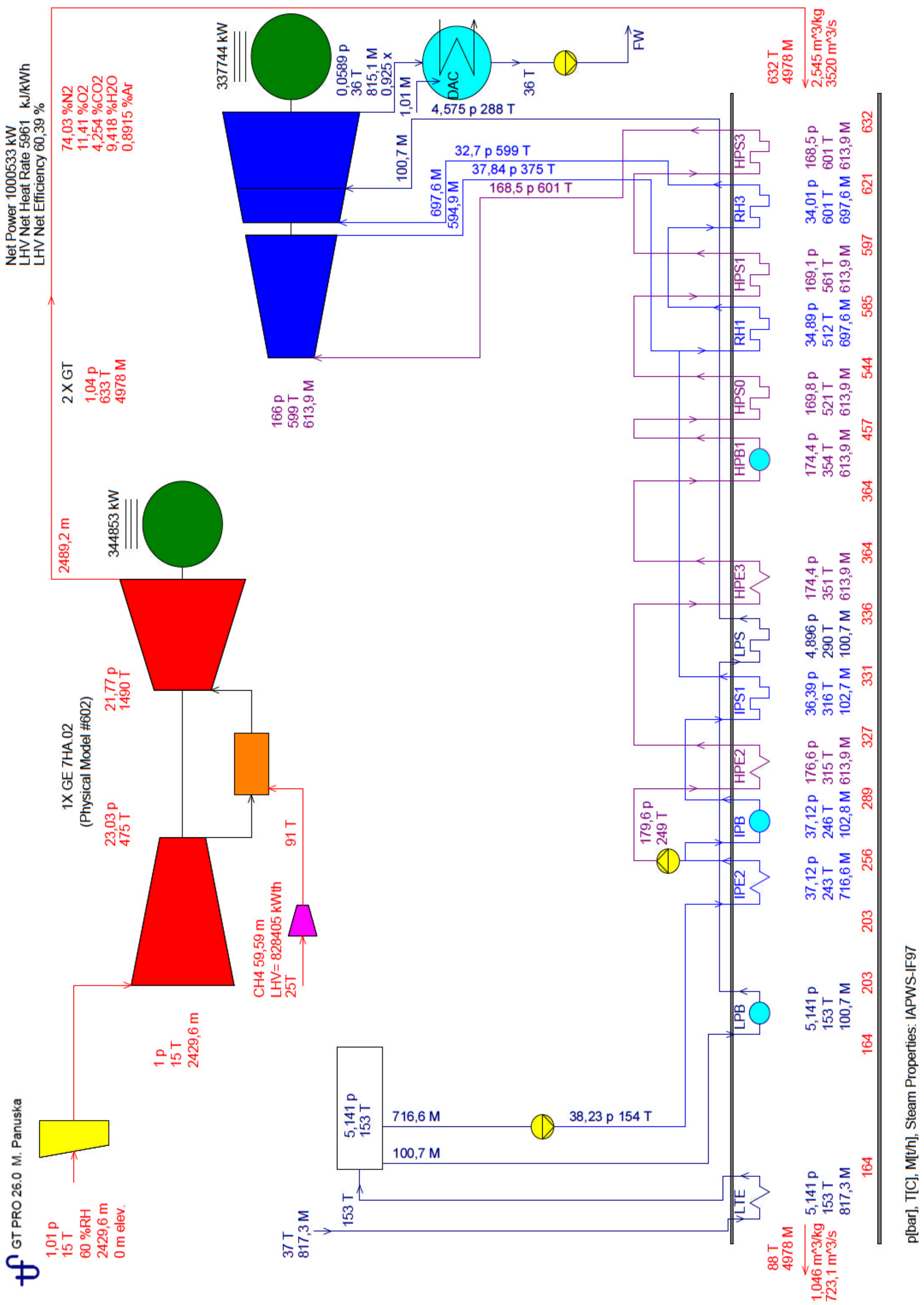

Figure 14: Heat balance of triple-pressure cycle using H-class gas turbine

3 Triple-pressure system to be optimized

The main purpose of this thesis is to optimize a reference 2+2+1 combined cycle in which a bottoming cycle can be offered by one of steam turbine manufacturers for existing F-class combustion turbines currently operating in single cycle in 60 Hz market. Actual arrangement which will be dealt with is a triple-pressure cycle with 2+2+1 configuration. There are two gas turbines General Electric PG7241FA with exhaust to the same number of HRSGs producing steam for one steam turbine. This system will be described in this chapter focusing on *steam tail*.

3.1 Overview of the system

As said above, this thesis deals with a triple-pressure HRSG with a triple pressure reheat bottoming cycle shown in Figure 15 (simplified for clarity with only one gas turbine and one HRSG). There are two identical combustion turbines GE PG7241FA (also known as 7FA.03) with generators producing approx. 170 MWe each. Every section in HRSGs (low pressure (LP), intermediate pressure (IP) and high pressure (HP)) has several heat exchangers. As we can see in Figure 15 there are economizers (E), evaporators (B) and superheaters (S). IP steam is merged with cold reheat steam and after that the whole amount of steam is reheated. Mass flow of IP steam is thus larger than mass flow of live steam. In case of LP steam is it very similar. LP steam from HRSG is added to crossover piping from IP part of the turbine to LP part. The steam turbine with the generator produces approx. 200 MWe (will be specified below).

3.2 Site conditions

Standardized ISO ambient conditions are considered (15°C, relative humidity 60 % and sea level elevation) since it is a reference power plant. Different projects may have different site conditions, optimization points and potential client needs.

3.3 Combustion turbine GE 7FA.03

There is an extensive fleet of General Electric (GE) 7F combustion turbines in 60 Hz market, with installations from the late 1990s until today. For this reference plant, the target units in this fleet are those that were installed in the early 2000s, which are mostly PG7241FA models (more commonly known as the 7FA.03). These units have a comparably smaller output and higher heat rate than the current 7F.05 model that is actively being supplied by GE. However, the reference plant is intended to cater to existing power plant owners who may want to get more out of their combustion turbines.

GE has offered packages for upgrading the existing combustion turbines. However, without knowing what package(s) have been implemented by a potential client, it is assumed that there have been no upgrades.

3.3.1 Specifications of GE 7FA.03

GE 7FA.03 is single shaft, heavy-duty combustion turbine. Ambient air is compressed in a 18-stage compressor. Then it is mixed with fuel, ignited and burned in a combus-

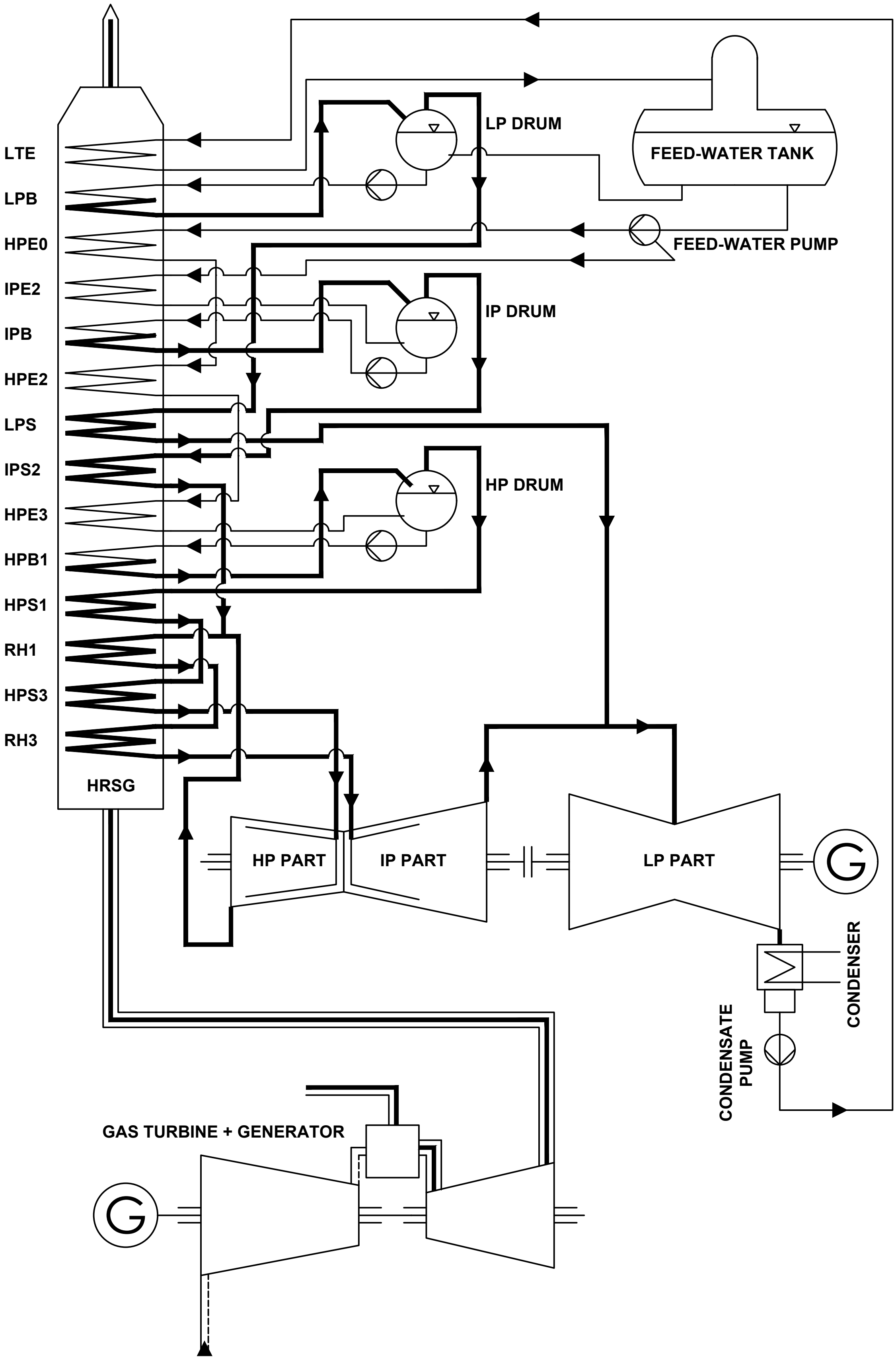

Figure 15: Flow diagram of 2+2+1 triple-pressure system

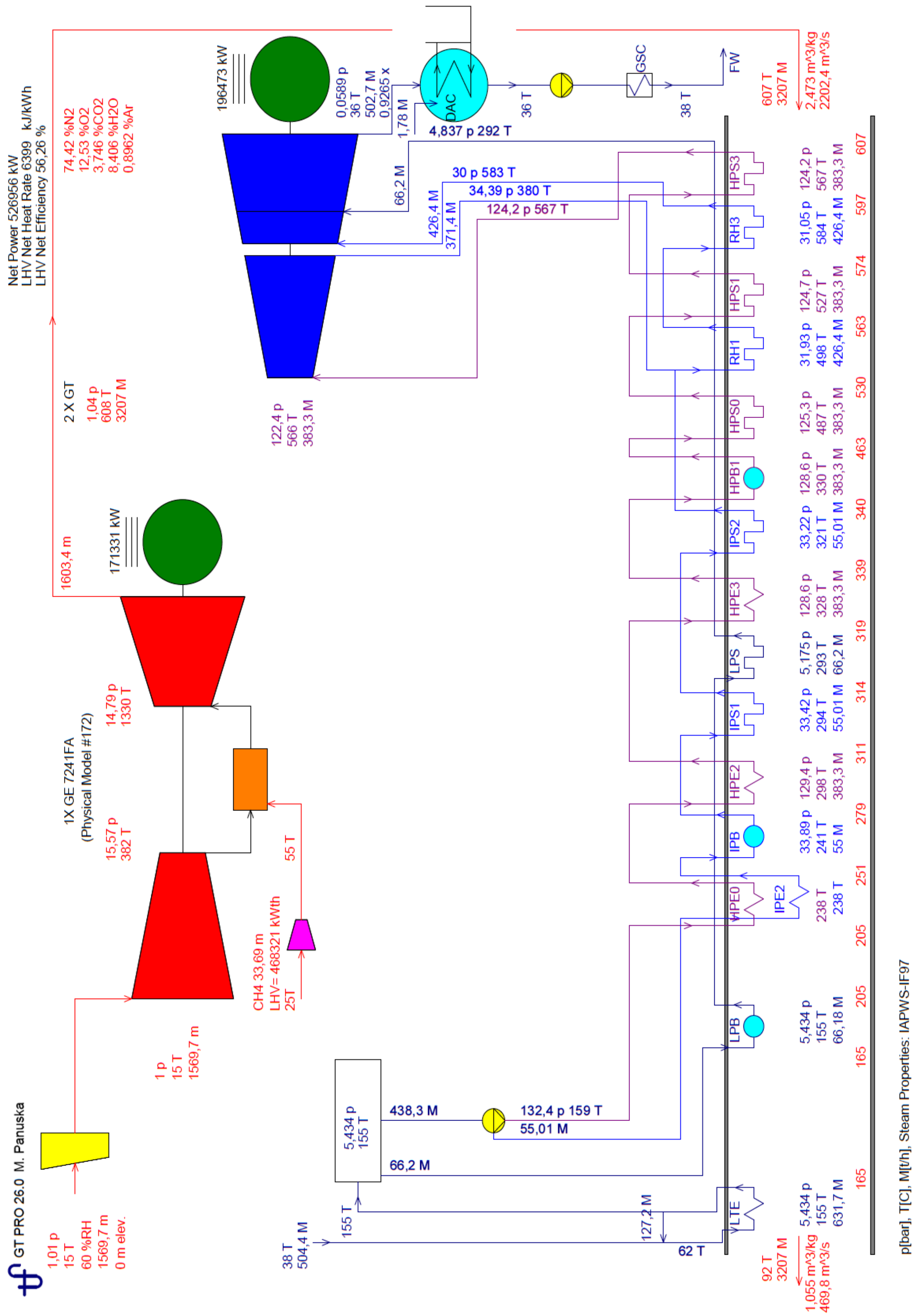

Figure 16: Heat balance of triple-pressure cycle

tion section, which consists of 14 can-annular combustion chambers. The gas turbine with 3 stages, where the hot gas mixture expands, drives both the compressor and the generator. Main parameters of this turbine, depending on ambient conditions, are:

- pressure ratio: 15.5,
- gas turbine inlet temperature: 1330 °C,
- gas turbine outlet temperature: 611 °C,
- gas mass flow: 888.6 kg/s,
- power output: 170.2 MWe.

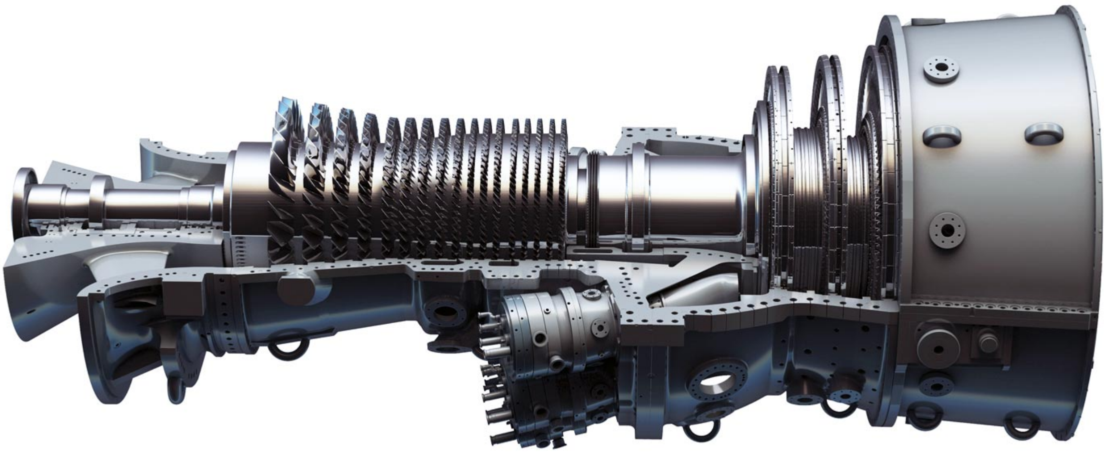

Figure 17: General Electric 7F.04 gas turbine (60 Hz) [11]

3.4 Bottoming cycle

A triple-pressure reheat cycle with a double casing design of a condensing steam turbine is assumed for this reference project.

3.5 Steam turbine

The steam turbine is of a double casing design with combined high pressure (HP) and intermediate pressure (IP) part and double flow low pressure (LP) part with downward exhaust to condenser. Cross-section through this turbine is shown in Figure 18.

3.5.1 Combined HP and IP part

Both HP and IP part combined in single casing are of a double shell design. The live (HP) steam enters the HP part via the main stop valve and control valve on the outer casing. Then it flows to a circular duct casted in the inner casing. The downstream direction of the steam is then towards the front bearing pedestal. There are 13 drum-stages in the HP part. Partly expanded steam flows back to the HRSG to reheat. The

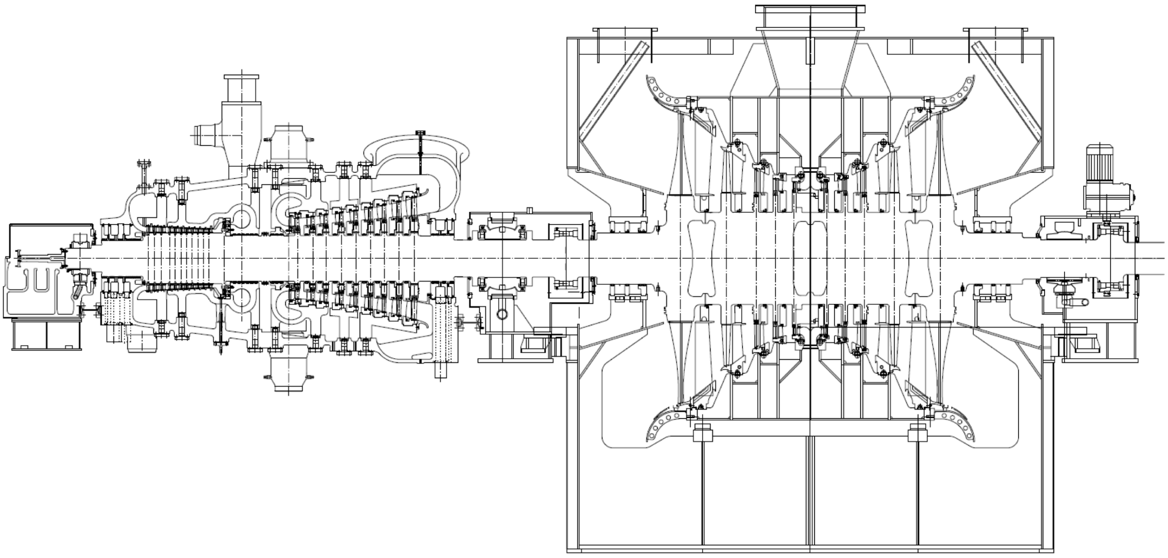

Figure 18: Cross-section through double casing steam turbine with combined HP-IP part (only for illustrative purpose - vary in number of stages) [12]

hot reheat steam is conducted through the combined stop and control valve to 10-stage IP part. The branches of the HP steam exhaust (cold reheat steam) are situated on the lower half of the HP-IP outer casing whilst the branches of crossover pipeline into the LP turbine are situated on the outer casing upper half.

3.5.2 LP part

The steam from IP part exhaust is merged with LP steam from the HRSG in the crossover pipeline. The mass flow of steam is split into the double flow design of LP part. Each flow has 6 stages. The exhaust neck to the condenser is of rectangular shape.

4 Ways to optimize the bottoming cycle

4.1 Condenser

One of possible ways to improve thermal efficiency and gain some performance of this cycle is to use a double-pressure condenser instead of a single-pressure one. For comparison and dimensioning of the heat transfer surface, the single pressure condenser has to be calculated first. Since the heat transfer surface is known, the influence of the double-pressure condenser can be calculated. There are two possible extreme variants with respect to the single pressure condenser which will be calculated:

- constant heat exchange surface (S),
- constant terminal temperature difference (TTD).

The flow of cooling water through these condensers is shown in Figure 19.

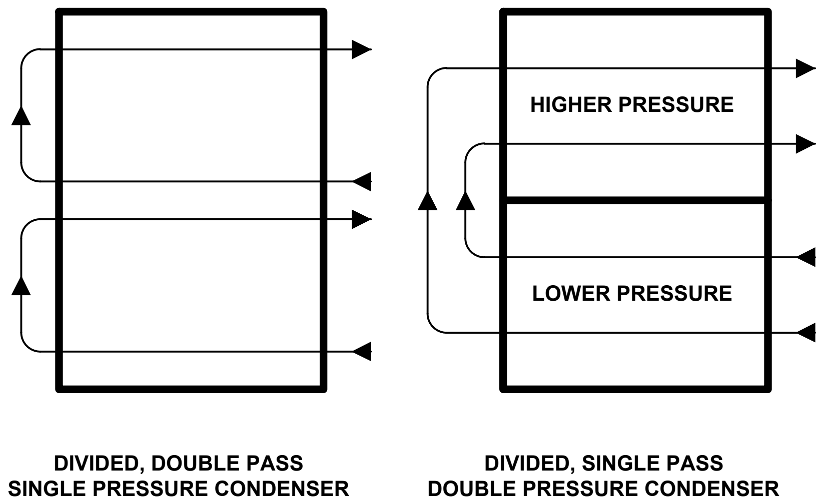

Figure 19: Single and double-pressure condenser

4.1.1 Single pressure condenser

Condenser is a heat exchanger where steam is condensed and cooling water is heated. Firstly, the design calculation of the condenser has to be done [10]. The heat balance equation of the condenser is:

$$\dot{Q} = \dot{m}_s \cdot (h_2 - h_c) = \dot{m}_w \cdot c_{pw} \cdot (t_{w2} - t_{w1}) = U \cdot A \cdot LMTD, \tag{9}$$

where

$$LMTD = \frac{TR}{\ln\left(\frac{ITD}{TTD}\right)} = \frac{t_{w2} - t_{w1}}{\ln\left(\frac{t_c - t_{w1}}{t_c - t_{w2}}\right)}. \tag{10}$$

The *logarithmic mean temperature difference (LMTD)* is a logarithmic average of the temperature difference between cold and hot side of the condenser. Process of the heat exchange is schematically shown in Figure 20. Analogy of the *pinch point* in an

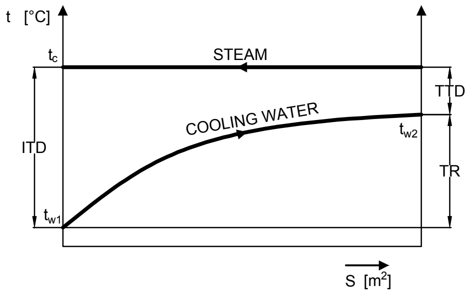

Figure 20: Heat transfer in condenser

HRSG is the *terminal temperature difference (TTD)* in the condenser. This is the main parameter which sets the size of heat exchange surface.

Given parameters from heat balance diagram calculated by the steam turbine manufacturer are:

$$\dot{m}_s = 140.99\ \frac{\text{kg}}{\text{s}},\ t_c = 35.82\ °\text{C},\ p_c = 0.0589\ \text{bar(a)},\ h_2 = 2393.38\ \frac{\text{kJ}}{\text{kg}}.$$

Other considered parameters are:

$$TTD = 3\ °\text{C},\ p_w = 3\ \text{bar(a)},\ TR = 10\ °\text{C},\ w = 2.1\ \frac{\text{m}}{\text{s}}.$$

Assuming that

$$t_c = t_{w1} + TR + TTD, \tag{11}$$

inlet temperature of the cooling water is

$$t_{w1} = t_c - TR - TTD = 35.82 - 10 - 3 = 22.82\ °\text{C} \tag{12}$$

and outlet temperature then is

$$t_{w2} = t_c - TTD = 35.82 - 3 = 32.82\ °\text{C}. \tag{13}$$

The specific heat of cooling water is determined from its mean temperature. This temperature is obtained from

$$t_{wm} = \frac{t_{w1} + t_{w2}}{2} = \frac{22.82 + 32.82}{2} = 27.82\ °\text{C}. \tag{14}$$

Now the specific heat of cooling water and enthalpy of the condensate (saturated water) can be determined. It is done using IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam (IF-97).

$$c_{pw}\,(3\ \text{bara}; 27.82\ °\text{C}) = 4.180\ \frac{\text{kJ}}{\text{kg} \cdot \text{K}} \tag{15}$$

$$h_c'\,(0.0589\ \text{bar(a)}) = 150.09\ \frac{\text{kJ}}{\text{kg}} \tag{16}$$

$$\rho\,(3\ \text{bara}; 27.82\ ^\circ\text{C}) = 996.38\ \frac{\text{kg}}{\text{m}^3} \tag{17}$$

Then the first part of heat balance equation in the condenser (9) can be solved, the needed mass flow of cooling water can be found out and the condenser performance (heat flow) as well.

$$\dot{m}_s \cdot (h_2 - h_c') = \dot{m}_w \cdot c_{pw} \cdot (t_{w2} - t_{w1}) \tag{18}$$

$$\dot{m}_w = \frac{\dot{m}_s \cdot (h_2 - h_c')}{c_{pw} \cdot (t_{w2} - t_{w1})} = \frac{140.99 \cdot (2393.38 - 150.09)}{4.181 \cdot 10^3 \cdot (32.83 - 22.83)} = 7563.8\ \frac{\text{kg}}{\text{s}} \tag{19}$$

$$\dot{Q} = 140.99 \cdot (2393.38 - 150.09) = 316282.66\ \text{kW} \tag{20}$$

In this calculation, the austenitic stainless steel tubes with the outside diameter of 25 mm and 1 mm wall thickness are considered. After that the number of tubes can be calculated from the equation of continuity:

$$\frac{\dot{m}_w}{\rho} = A_T \cdot w, \tag{21}$$

where

$$A_T = n_t \cdot \pi \cdot \frac{d_i^2}{4}. \tag{22}$$

By combining equations (21) and (22) is obtained

$$\frac{\dot{m}_w}{\rho} = \frac{n_t \cdot \pi \cdot d_i^2 \cdot w}{4} \tag{23}$$

and the number of tubes is

$$n_t = \frac{4 \cdot \dot{m}_w}{\rho \cdot \pi \cdot d_i^2 \cdot w} = \frac{4 \cdot 7563.8}{996.38 \cdot \pi \cdot 0,023^2 \cdot 2.1} \doteq 8701. \tag{24}$$

Before the calculation of the heat exchange surface is started, the heat transfer coefficient has to be determined. It will be done according to Heat Exchange Institute (HEI) Standard. This is an internationally recognized set of recommendations used in designing power plants equipment. In case of basic design of a condenser, the most widespread methodology is the one referred in [6]. The approach is based on an empirical formula.

The uncorrected heat transfer coefficient U_1 is multiplied by three correction factors to obtain the presumed heat transfer coefficient U.

$$U_1 = 2854 \cdot (1 - 2.23 \cdot d_o) \cdot \sqrt{w} \tag{25}$$

$$U = U_1 \cdot F_W \cdot F_M \cdot F_C \tag{26}$$

Above mentioned correction factors are:

- cooling water inlet temperature correction factor F_W,
- tube material and gauge correction factor F_M,
- tube cleanliness correction factor F_C.

The correction factor for inlet temperature of cooling water can be simply calculated from

$$F_W = 1.051 \cdot 10^{-2} \cdot t_{w1} - 1.506 \cdot 10^{-3} \cdot \left[\sqrt{(21.5 - t_{w1})^2}\right]^{1.39} + 0,7765. \tag{27}$$

The next factor is obtained from [6] and for this application (austenitic stainless steel tubes, wall thickness 1 mm) is considered

$$F_M = 0.79.$$

The tube cleanliness correction factor is dependent on tube material, operating and maintenance conditions and cooling water parameters. We will assume

$$F_C = 0.9$$

for given conditions and a continual cleaning system. As everything necessary is known, the heat exchange coefficient can be calculated.

$$U_1 = 2854 \cdot (1 - 2.23 \cdot 0.025) \cdot \sqrt{2.1} = 3905.26 \ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}} \tag{28}$$

$$F_W = 1.051 \cdot 10^{-2} \cdot 22.82 - 1.506 \cdot 10^{-3} \cdot \left[\sqrt{(21.5 - 22.82)^2}\right]^{1.39} + 0.7765 = 1.014123 \tag{29}$$

$$U = 3905.26 \cdot 1.014123 \cdot 0.79 \cdot 0.9 = 2815.86 \ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}} \tag{30}$$

The last step of this calculation is to determine the effective heat exchange surface area from the Fourier equation for heat transfer:

$$\dot{Q} = U \cdot A \cdot LMTD \rightarrow A = \frac{\dot{Q}}{k \cdot LMTD}, \tag{31}$$

where

$$LMTD = \frac{t_{w2} - t_{w1}}{\ln\left(\frac{t_c - t_{w1}}{t_c - t_{w2}}\right)} = \frac{32.82 - 22.82}{\ln\left(\frac{35.82 - 32.82}{35.82 - 22.82}\right)} = 6.82 \ \mathrm{K}. \tag{32}$$

The heat exchange surface area is

$$A = \frac{\dot{Q}}{U \cdot LMTD} = \frac{316282.66 \cdot 10^3}{2815.86 \cdot 6.82} = 16467.65 \ \mathrm{m^2}. \tag{33}$$

The length of tubes remains to be determined since we know the number of tubes.

$$A = \pi \cdot d_o \cdot L_t \cdot n_t \rightarrow L_t = \frac{A}{\pi \cdot d_o \cdot n_t} \tag{34}$$

$$L_t = \frac{A}{\pi \cdot d_o \cdot n_t} = \frac{16467.65}{\pi \cdot 0.025 \cdot 8701} = 24.1 \ \mathrm{m} \tag{35}$$

When a double pass condenser is assumed, it is half the length and number of tubes is doubled.

$$L_t = \frac{24.1}{2} = 12.05 \ \mathrm{m} \tag{36}$$

$$n_t = 8701 \cdot 2 = 17402 \tag{37}$$

Habitually some reserve for covered tubes in case of a leakage is considered. This calculation is done only for comparison and hence this reserve will be neglected. The obtained results will be used as input parameters for the following calculation.

4.1.2 Double-pressure condenser with preserved heat exchange surface

In a double-pressure condenser (or sometimes called serial configured condenser) cooling water flows in series through two condensers, which means that different condenser pressures are achieved. Anyway, this leads to an increase in price, but the power and efficiency gain usually pays off. If a double flow LP part of a steam turbine is used, the double-pressure condenser is in fact a single condenser divided into two parts.

A cheaper choice is with equal total heat exchange surface and mass flow of the cooling water as above. To remind, preserved parameters are:

$$\dot{m}_w = 7563.8\ \frac{\mathrm{kg}}{\mathrm{s}},\ A = 16467.65\ \mathrm{m}^2,\ L_t = 12.05\ \mathrm{m},\ n_t = 17402$$

The difference between the inlet and outlet temperature of the cooling water is the same (10 °C) but now it is equally divided into two stages. The temperature rise is thus 5 °C for each stage. The mean temperature for defining the specific heat of cooling water of the first stage is:

$$t_{wm1} = \frac{t_{w1} + t_{w,med}}{2} = \frac{22.82 + 27.82}{2} = 25.32\ ^\circ\mathrm{C}, \tag{38}$$

Specific heat for this value is:

$$c_{pw1}\,(3\ \mathrm{bara}; 25.32\ ^\circ\mathrm{C}) = 4.179\ \frac{\mathrm{kJ}}{\mathrm{kg}\cdot\mathrm{K}} \tag{39}$$

Saturation temperature in the condenser will be found out according to equation (9).

$$\dot{Q} = \dot{m}_w \cdot c_{pw} \cdot (t_{w2} - t_{w1}) = U \cdot A \cdot LMTD \tag{40}$$

After mathematical modification we obtain

$$\ln\left(\frac{t_c - t_{w1}}{t_c - t_{w2}}\right) = \frac{A \cdot U}{\dot{m}_w \cdot c_{pw1}}. \tag{41}$$

The right side of this equation will be substituted as X.

$$\ln\left(\frac{t_c - t_{w1}}{t_c - t_{w2}}\right) = X \tag{42}$$

Now let each side of the above equation be the exponent of the base e.

$$\frac{t_c - t_{w1}}{t_c - t_{w2}} = e^X \tag{43}$$

Then we can already calculate the saturation temperature of the first stage of the condenser.

$$t_c = \frac{e^X}{e^X - 1} \cdot t_{w2} - \frac{t_{w1}}{e^X - 1} \tag{44}$$

$$X = \frac{8233.82 \cdot 2815.86}{7563.8 \cdot 4.181} = 0.7333117 \tag{45}$$

$$t_c = \frac{e^{0.7333117}}{e^{0.7333117} - 1} \cdot 27.82 - \frac{22.82}{e^{0.7333117} - 1} = 32.44\ °\text{C} \tag{46}$$

Terminal temperature difference and saturation pressure in the condenser are

$$TTD = 32.44 - 27.82 = 4.62\ °\text{C}, \tag{47}$$

$$p_c\,(32.44\ °\text{C}) = 0.0488\ \text{bara}. \tag{48}$$

Before calculating the power gain, some considerations related to the impact of discussed changes on the turbine have to be done.

In general, a decrease of backpressure in the turbine affects only a few last stages (usually two). The impact on the penultimate stage is much smaller than on the ultimate one. Therefore, we will consider whole change of enthalpy and pressure drop on the last stage.

According to the initial heat balance calculation done by steam turbine manufacturer, the parameters of the steam are:

$$p_1 = 0.21477\ \text{bara},\ h_1 = 2536.7\ \frac{\text{kJ}}{\text{kg}},\ p_2 = 0.0589\ \text{bara},\ h_2 = 2393.38\ \frac{\text{kJ}}{\text{kg}},$$

where index 1 indicates the state before the last stage and index 2 behind it. Some other parameters such as entropy and vapour fraction can be determined from these values:

$$s\left(0.21477\ \text{bara}; 2536.7\ \frac{\text{kJ}}{\text{kg}}\right) = 7.65872\ \frac{\text{kJ}}{\text{kg}\cdot\text{K}} \tag{49}$$

$$x\left(0.21477\ \text{bara}; 2393.38\ \frac{\text{kJ}}{\text{kg}}\right) = 0.9285 \tag{50}$$

For determining the thermodynamic efficiency of the last stage (LS), both isoentropic and real enthalpy drop have to be known, but it is necessary to start with the enthalpy behind the LS for isoentropic expansion.

$$h_{2s}\left(0.05884\ \text{bara}; 7.65872\ \frac{\text{kJ}}{\text{kg}\cdot\text{K}}\right) = 2356.88\ \frac{\text{kJ}}{\text{kg}} \tag{51}$$

The power of the last stage can be determined as well:

$$P_{LS} = \dot{m}_s \cdot (h_1 - h_2) = 140.99 \cdot (2536.7 - 2393.38) = 20205.8\ \text{kW}. \tag{52}$$

The thermodynamic efficiency of the LS is

$$\eta_{td,LS} = \frac{h}{h_{ad}} = \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{2536.7 - 2393.38}{2536.7 - 2356.88} = 0.797. \tag{53}$$

It could be misleading that the value is too low. It is due to the fact that moisture loss and leaving-velocity loss are included. Analysis of this phenomenon is beyond the intent of this thesis.

The expansion of steam is slightly extended for the first stage, respectively shorter for the second stage when using a double-pressure condenser. In the first case it leads to an increase in moisture, leaving velocity and load of the LS. Therefore the thermodynamic efficiency decreases. In case of the second stage, the trend is opposite.

The considered thermodynamic efficiency of the LS for first stage (lower pressure) is

$$\eta_{td,LS1} = 0.77. \tag{54}$$

According to the condensing pressure calculated above, the enthalpy after isoentropic expansion on the last stage of LP part's first flow is

$$h_{2s}\left(0.0488\ \text{bara}; 7.65872\ \frac{\text{kJ}}{\text{kg}\cdot\text{K}}\right) = 2332.68\ \frac{\text{kJ}}{\text{kg}}. \tag{55}$$

From (54) an equation for determining the real enthalpy can be deduced:

$$h_2 = h_1 - (h_1 - h_{2s}) \cdot \eta_{td,LS1} = 2536.7 - (2536.7 - 2332.68) \cdot 0.77 = 2379.6\ \frac{\text{kJ}}{\text{kg}}. \tag{56}$$

Finally, the power of the turbine's last stage can be determined.

$$P_{LS1} = \dot{m}_s \cdot (h_1 - h_2) = 70.495 \cdot (2536.7 - 2379.6) = 11074.36\ \text{kW} \tag{57}$$

Considered efficiency of the generator is

$$\eta_G = 0.9871. \tag{58}$$

Electric power output of the last stage is obtained by

$$N_{el,LS1} = P_{LS1} \cdot \eta_G = 11074.36 \cdot 0.9871 = 10931.5\ \text{kW} \tag{59}$$

Now it is necessary to do the same for the second flow of LP part. It is an analogy of the first one, so the calculation will not be specified here. For results see Table 2.

4.1.3 Double-pressure condenser with preserved TTD

Another choice is a more powerful and of course more expensive double-pressure condenser with constant terminal temperature difference (enlarged heat exchange surface). The method of calculation is identical with a single-pressure condenser but it has to be done twice. The results of this calculation can be seen in Table 3.

The last thing, which has to be done for comparison of these various types of condensers, is the enumeration of the pressure drop of cooling water, which is done according to [10], [13] and [5]. It is necessary to introduce several assumptions. From Figure 19 can be concluded, that in all cases there is the same number of water boxes and water velocities as well. The pressure drop in these places (inlets and outlets of tubes, water boxes and necks) is thus the same. We will only focus only on pressure drop in tubes. The first two types of condensers have the same length of tubes, which means that the pressure drop is the same. The difference is in case of the condenser with extended tubes. This difference will be calculated and so will be an impact to

Quantity		1st stage	2nd stage
p_1	[bar(a)]	0.21477	0.21477
p_2	[bar(a)]	0.0488	0.0637
TTD	[°C]	4.62	4.4
h_1	$\left[\frac{kJ}{kg}\right]$	2536.7	2536.7
s	$\left[\frac{kJ}{kg \cdot K}\right]$	7.65872	7.65872
h_{2s}	$\left[\frac{kJ}{kg}\right]$	2332.68	2367.08
$\eta_{td,LS}$	[−]	0.77	0.81
h_2	$\left[\frac{kJ}{kg}\right]$	2379.6	2399.31
x	[−]	0.9256	0.9298
h_s	$\left[\frac{kJ}{kg}\right]$	204.02	169.61
h	$\left[\frac{kJ}{kg}\right]$	157.09	137.39
$\dot{m}_s$	$\left[\frac{kg}{s}\right]$	70.495	70.495
P_{LS}	[kW]	11074.36	9685.08

Table 2: Double-pressure condenser (constant S)

Quantity		1st stage	2nd stage
t_{w1}	[°C]	22.82	27.82
t_{w2}	[°C]	27.82	32.82
t_{wm}	[°C]	25.32	30.32
TTD	[°C]	3	3
$\dot{m}_w$	$\left[\frac{kg}{s}\right]$	7565.4	7567.7
t_c	[°C]	30.82	35.82
h'_c	$\left[\frac{kJ}{kg}\right]$	129.17	150.09
p_1	[bar(a)]	0.21477	0.21477
$p_2 = p_c$	[bar(a)]	0.0445	0.0589
h_1	$\left[\frac{kJ}{kg}\right]$	2536.7	2536.7
s	$\left[\frac{kJ}{kg \cdot K}\right]$	7.65872	7.65872
h_{2s}	$\left[\frac{kJ}{kg}\right]$	2320.99	2356.87
$\eta_{td,LS}$	[−]	0.76	0.797
h_2	$\left[\frac{kJ}{kg}\right]$	2372.76	2393.38
x	[−]	0.9241	0.9285
h_s	$\left[\frac{kJ}{kg}\right]$	215.71	179.83
h	$\left[\frac{kJ}{kg}\right]$	163.94	143.32
$\dot{m}_s$	$\left[\frac{kg}{s}\right]$	70.495	70.495
P_{LS}	[kW]	11557.06	10103.24
n_t	[−]	8697	8712
A	[m^2]	11018.26	10647.04
L_t	[m]	16.13	15.56

Table 3: Double-pressure condenser (constant TTD)

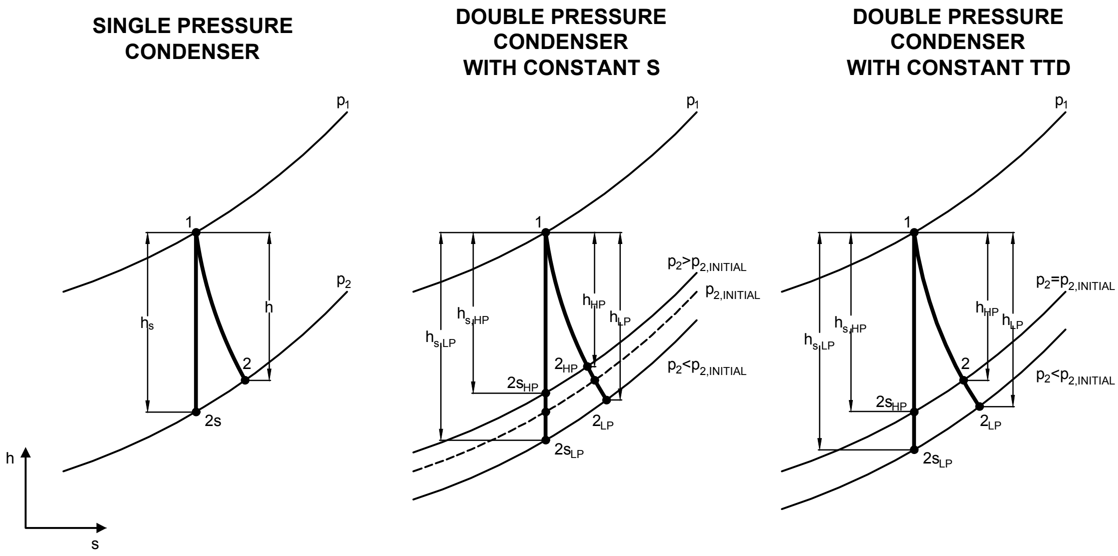

Figure 21: h-s diagram

power consumption of cooling water pumps. Numerical calculation will be done only for the original condenser, the rest is stated in Table 4. All needed thermodynamic properties of water are obtained again from IF-97.

Generally a formula of pressure drop is expressed:

$$\Delta p = \zeta \cdot \rho \cdot \frac{w^2}{2}. \tag{60}$$

ζ is coefficient of local resistance and is given by

$$\zeta = \lambda_{fr} \cdot \frac{L_t}{d_i}. \tag{61}$$

The friction coefficient λ_{fr} is dependent on:

- characteristic dimension (internal diameter of the pipe in this case),
- viscosity,
- flow velocity,
- internal roughness of the pipe.

The first three mentioned form a dimensionless quantity called *Reynolds number*:

$$\mathrm{Re} = \frac{w \cdot d_i}{\nu}, \tag{62}$$

where

$$\nu = \frac{\mu}{\rho}. \tag{63}$$

To remind, the cooling water velocity is assumed $w = 2.1$ m/s, $d_i = 0.023$ m. The dynamic viscosity is $\mu = 0.0008357 \ \frac{\mathrm{kg}}{\mathrm{s \cdot m}}$ and the density of cooling water is $\rho = 996.38 \ \frac{\mathrm{kg}}{\mathrm{m^3}}$. By substituting these values to (62) and (63) is obtained

$$\nu = \frac{0.0008357}{996.38} = 8.3874 \cdot 10^{-7} \tag{64}$$

and

$$\mathrm{Re} = \frac{2.1 \cdot 0.023}{8.3874 \cdot 10^{-7}} = 57586. \tag{65}$$

If *Reynolds number* is lower than 2320 the flow is laminar, otherwise it is presumably turbulent. The assumed tubes in these condensers are not smooth. The calculation must also take this fact into account. The roughness of the new austenitic stainless steel tubes is $K = 0,005$ mm (average height of all the projections). For the determination of the friction coefficient λ_{fr} it must be known whether the flow is completely governed by roughness or is in the transient range. *Reynolds number* which indicates this bound can be obtained from:

$$\mathrm{Re}_h = \frac{217.6 - 382.4 \cdot \log_{10} \frac{K}{d_i}}{\frac{K}{d_i}}. \tag{66}$$

If $\mathrm{Re} < \mathrm{Re}_h$, the equation for determination of the friction coefficient λ_{fr} is

$$\lambda_{fr} = \left(2 \cdot \log_{10} \left(\frac{2.51}{\mathrm{Re} \cdot \sqrt{\lambda_{fr}}} + \frac{K}{3.71 \cdot d_i} \right) \right)^{-2} \tag{67}$$

and otherwise it is

$$\lambda_{fr} = \left(2 \cdot \log_{10} \frac{d_i}{K} + 1.14 \right)^{-2}. \tag{68}$$

Numerically expressed

$$\mathrm{Re}_h = \frac{217.6 - 382.4 \cdot \log_{10} \frac{0.000005}{0.023}}{\frac{0.000005}{0.023}} = 7443898 \tag{69}$$

shows, that $\mathrm{Re} < \mathrm{Re}_h$. The determining of the friction coefficient λ_{fr} is done according to (67). The result obtained by iterative calculation is

$$\lambda_{fr} = 0.02106. \tag{70}$$

At this moment, the coefficient of local resistance can be calculated from (61)

$$\zeta = 0.02106 \cdot \frac{12.05}{0.023} = 11.035 \tag{71}$$

and the pressure drop from (72) as well

$$\Delta p = 11.035 \cdot 996.38 \cdot \frac{2.1^2}{2} = 48488 \text{ Pa}. \tag{72}$$

Once the pressure losses are known, it is possible to evaluate the power consumption of cooling pumps. The difference of the power consumption of cooling pumps is given by

$$\Delta P = \frac{Q_V \cdot \Delta p_{dif}}{\eta}, \tag{73}$$

Quantity		Single pressure condenser	Double press. condenser (1st stage)	Double press. condenser (2nd stage)
w	$\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$		2.1	
d_i	[m]		0.023	
p	[bar]	3	3	2.6
T	[°C]	27.82	25.32	30.32
μ	$\left[\frac{\mathrm{kg}}{\mathrm{s \cdot m}}\right]$	0.0008357	0.0008836	0.0007919
ρ	$\left[\frac{\mathrm{kg}}{\mathrm{m^3}}\right]$	996.38	997.05	995.63
ν	$\left[\frac{\mathrm{m^2}}{s}\right]$	8.387E-7	8.862E-7	7.954E-7
Re	[-]	57586	54502	60723
K	[mm]		0.005	
Re_h	[-]		7443898	
$\mathrm{Re} < \mathrm{Re}_h$	[-]	YES	YES	YES
λ_{fr}	[-]	0.02106	0.02129	0.02085
L_t	[m]	12.05	16.13	15.56
ζ_t	[-]	11.035	14.93	14.109
Δp	[Pa]	48488	32843	30973
Δp_{dif}	[Pa]		15310	
ΔP	[kW]		145.31	

Table 4: Pressure drop in condenser tubes

where Δp_{diff} is difference in the pressure loss between the single and double-pressure condenser and

$$Q_V = \frac{m}{\rho}. \tag{74}$$

Table 4 shows, that $\Delta p_{diff} = 15310$ Pa. Finally, the difference in auxiliary consumption can be estimated, considering $\eta = 0.8$:

$$\Delta P = \frac{\frac{7565.4}{996.38} \cdot 15310}{0.8} = 145307.7 \text{ W}. \tag{75}$$

This value has to be deduced from the power gain achieved by using the double pressure condenser with the extended heat exchange surface.

4.1.4 Design concept of double flow LP part

Advantages of the double-pressure condenser are without any controversy, but there comes a problem with a design solution. The exhaust space is common for both flows in LP part, which means that the pressure level is the same (Figure 22). It is clear that the exhaust space has to be divided into two separate exhaust sections.

The LP part should be partly redesigned with regard to fulfil all necessary properties:

- providing two different condensing pressures,

- tightness of all joints,
- keeping leaving velocity in the exhaust neck under 120 m/s.

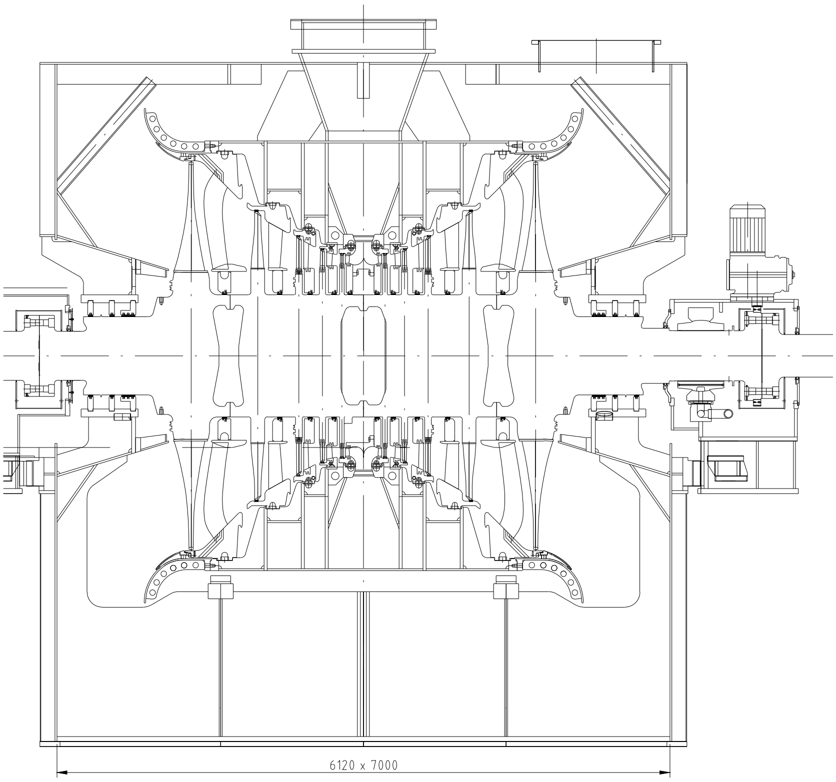

Figure 22: Original LP part cross-section (only for illustrative purpose - vary in number of stages) [12]

Figure 23 shows how it is done. The outer casing which is newly formed by two separate exhaust necks is flanged to the inner casing. The condenser is divided in the middle by a stainless steel plate to provide two different pressure levels possible.

As stated above, the leaving velocity of the steam in the exhaust neck should be kept below 120 m/s due to pressure loss. This velocity is obtained from equation

$$w = \frac{\dot{m}_s \cdot v}{A_E}. \tag{76}$$

In Figure 23 can bee seen that the exhaust area has the dimension 2560 x 7000 mm, which means that

$$A_E = 2.56 \cdot 7 = 17.92 \text{ m}^2. \tag{77}$$

According to the calculations above, it is known that the pressures at both ends of LP part are lower in case of using the double-pressure condenser with preserved TTD. If the pressure is lower, the specific volume is larger and the velocity is higher. The calculation will be done only for this type of the condenser, because the results will be "worse".

In case of the exhaust to lower pressure, the specific volume of steam leaving the

LP part is $v(0.0445 \text{ bara}; 2372.76 \frac{\text{kJ}}{\text{kg}}) = 29.07 \frac{\text{m}^3}{\text{kg}}$ and the mass flow is $\dot{m}_s = 70.495 \frac{\text{kg}}{\text{s}}$. The leaving velocity in the exhaust neck is

$$w = \frac{70.495 \cdot 29.07}{17.92} = 114.4 \frac{\text{m}}{\text{s}}. \tag{78}$$

The steam of higher pressure has the specific volume $v(0.0589 \text{ bara}; 2393.38 \frac{\text{kJ}}{\text{kg}}) = 22.43 \frac{\text{m}^3}{\text{kg}}$ and the mass flow remains the same. Thus the leaving velocity is

$$w = \frac{70.495 \cdot 22.43}{17.92} = 88.2 \frac{\text{m}}{\text{s}}. \tag{79}$$

These values of velocities in the exhaust necks are in the standard range. From this point of view the LP part is also properly designed.

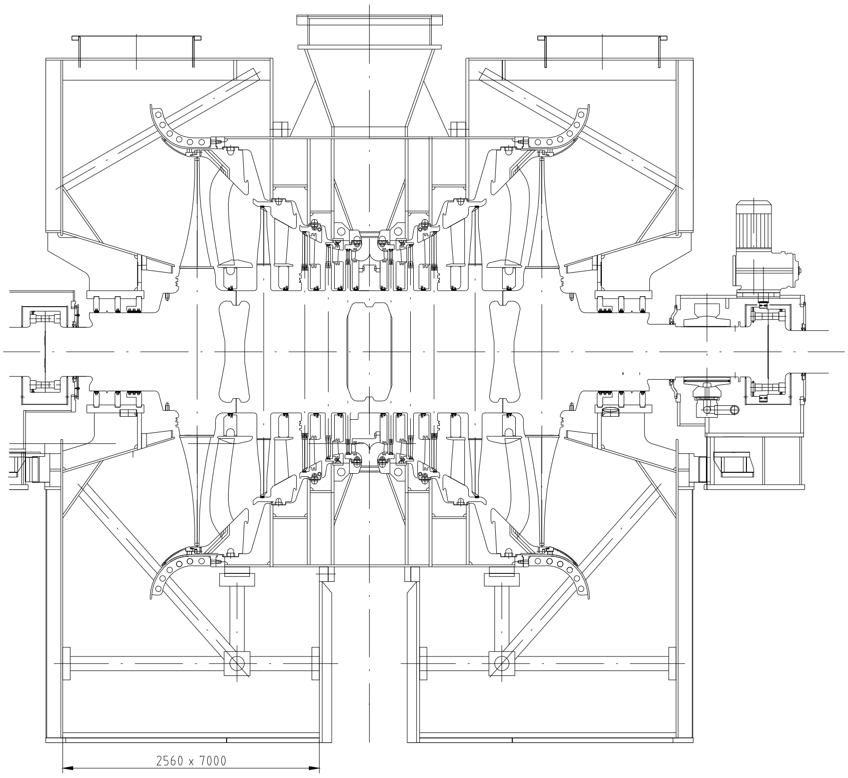

Figure 23: Redesigned LP part cross-section

4.1.5 Conclusion about condensers

Three types of the condenser were calculated and all available data are known to make evaluation and comparison.The first one, a double pass single-pressure condenser, is the initial solution designed by the steam turbine manufacturer. Splitting a condenser into two pressures results in a slight improvement of power output of the turbine. The heat exchange surface remains equal though. To achieve even more power output it is possible to extend the heat exchange surface for the price of higher investment costs. The gain of electrical power is 546.46 kW (0.28 %), respectively 1290.39 kW (0.66 %) in case of the last mentioned, which is reduced by the increase in auxiliary consumption

Quantity		**Single pressure condenser**	**Double press. condenser (const. S)**	**Double press. condenser (const. TTD)**
A	[m^2]	16467.65	16467.65	21665.3
A difference	[%]	-	0	32
p_c average	[bar(a)]	0.0589	0.0562	0.0517
N_{el} of last stage	[kW]	19945.18	20491.65	21380.88
N_{el} overall	[kW]	196312.9	196859.36	197748.6
N_{el} difference	[kW]	-	546.46	1290.39
N_{el} difference	[%]	-	0.28	0.66

Table 5: Comparison of the condensers

of cooling water pumps. If preservation of the terminal temperature difference in both pressure levels is required, the heat exchange surface has to be increased by 32 %. More detailed comparison is shown in Table. 5.

4.2 Layout

This part of the thesis focuses on the optimization of the bottoming cycle's layout. Using a suitable arrangement of a machine hall might save a big amount of investment costs.

4.2.1 Initial solution

The initial solution of layout designed by the steam turbine manufacturer can be seen in Figure 24. The turbo-generator set is at level +12.00 m. The steam turbine has a downward exhaust, which means that this height is given by the size of the condenser and a suction head of the condensate pumps. Further equipment which has to be below the level of the turbine due to the downgrade of piping are a main oil tank, an integrated oil system and all draining directed to an expansion tube (or a flash tank). It is not necessary to place this equipment below the level of condensate pumps.

The Piping and Instrumentation Diagram (P&ID) was drawn by the author for better understanding of the whole water and steam process (Appendix B). This drawing shows the piping and vessels, together with the control and instrumentation equipment which is in the presumed scope of the steam turbine manufacturer for offers to potential clients. The borders of this scope are marked too. The mentioned equipment is:

- steam turbine (combined HP-IP part, double flow LP part) with generator,
- condenser,
- flash tank,
- vent steam condenser,
- condensate pumps,

- continuous condenser cleaning system,
- water ring vacuum pumps.

4.2.2 Optimized solution

One possible way to lower the turbo-generator set level is to use a lateral exhaust from the LP part. The condenser is thus next to the LP part. Figure 25 shows that the turbo-generator set level was lowered to +5.5 m. The placement of the condenser is more apparent in Figure 26. Both drawing are also included in Appendix C. In this case, a critical point is the draining header which has to be at a low position to provide the downgrade of all draining. 3D view of the turbine with a lateral condenser is in Figure 27.

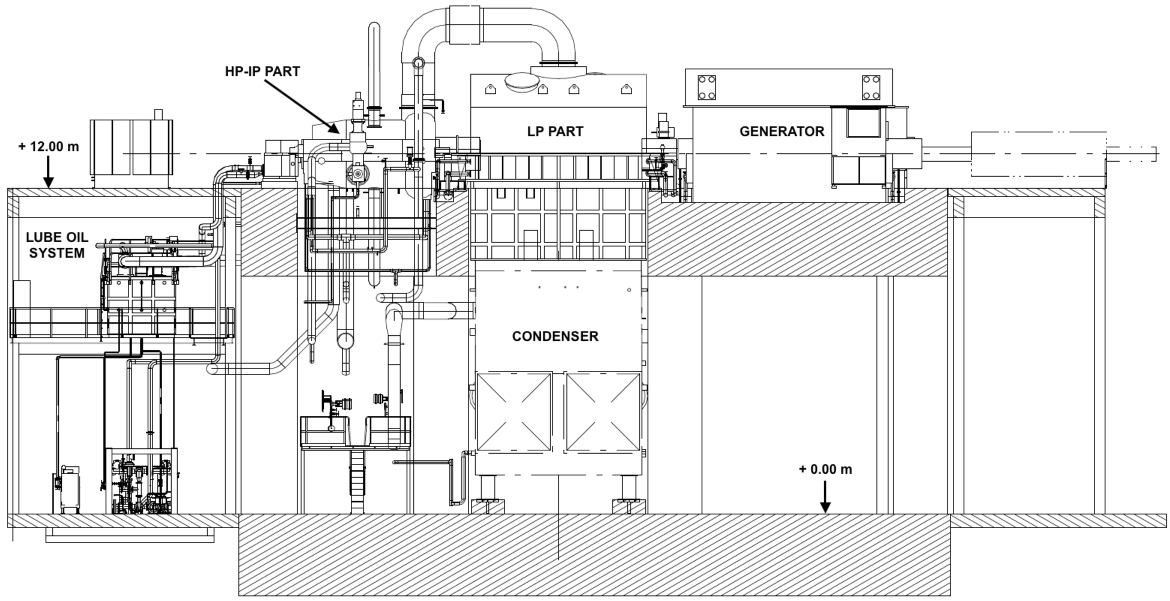

Figure 24: Original layout of the machine hall [12]

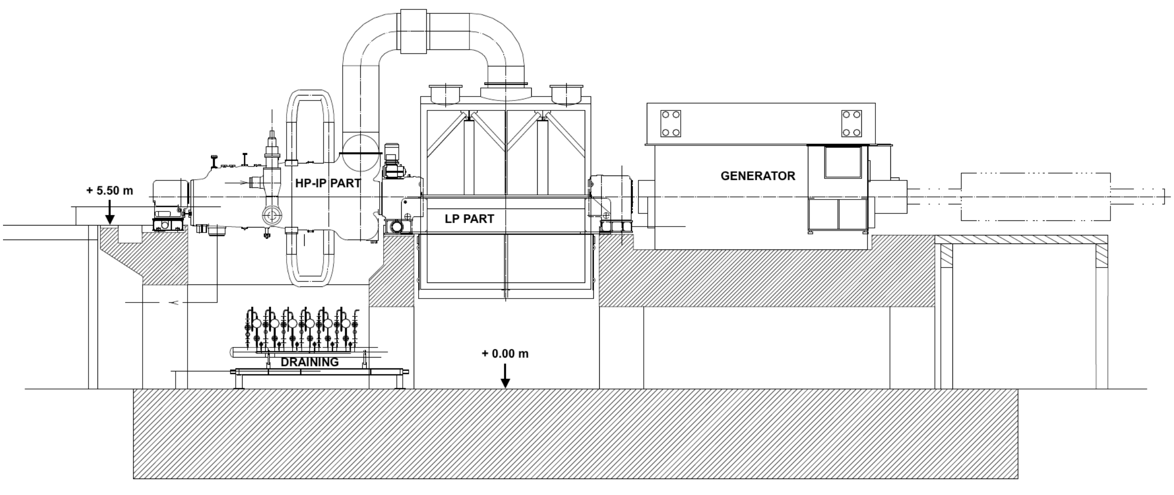

Figure 25: Longitudinal section through new layout of the machine hall

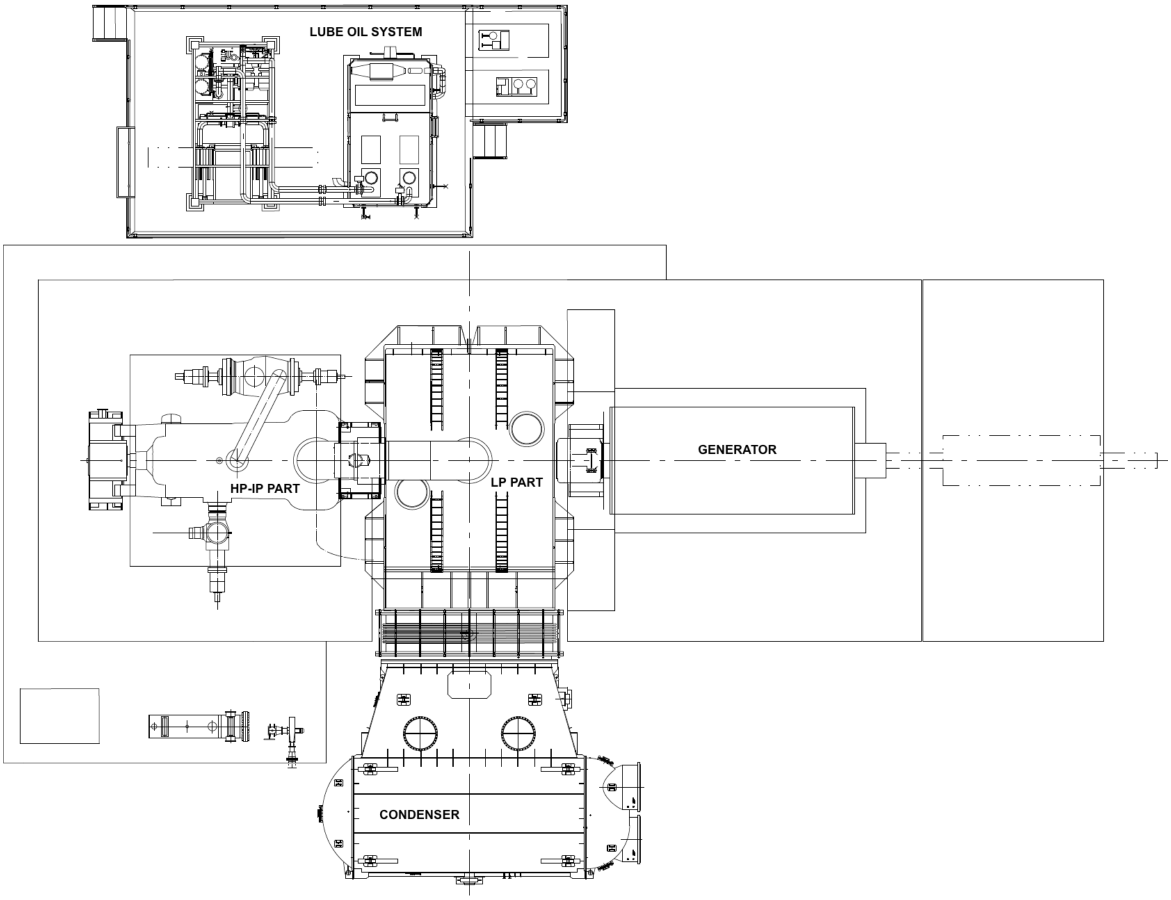

Figure 26: Level +5.5 m of the new layout

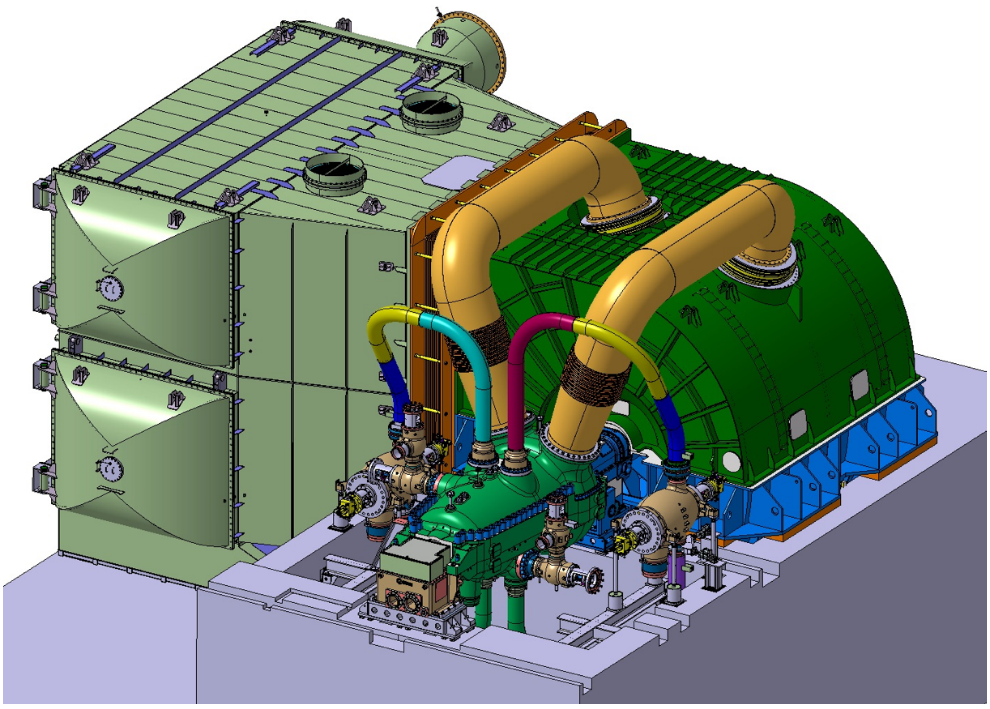

Figure 27: Steam turbine with lateral exhaust to condenser [12]

5 Technical and economic discussion

5.1 Condenser

The use of a double-pressure condenser brings additional investment costs. There are two calculated options – with constant heat exchange surface and with constant terminal temperature difference. It is obvious, that the variant where the heat exchange surface is extended is the more expensive one. The condenser with preserved heat exchange surface has not got the same price as the original condenser though. There is the necessity to split the inside area of the condenser into two sections to provide two different pressure levels. The price of the original single-pressure condenser is 2222925 USD. 1 kW of the electric power output earns approximately 4000 USD to a client per the whole lifetime of a power plant. Considered lifetime is 25 years and the turbo-generator set is in operation approximately 8000 h/year. In order to calculate a payback of the investment into a more expensive condenser, a profit for 1 kWh has to be known. It can be determined:

$$\pi_{1\ \mathrm{kWh}} = \frac{4000}{25 \cdot 8000} = 0.02\ \mathrm{USD/kWh} \tag{80}$$

5.1.1 Double-pressure condenser with constant S

Although this type of the condenser has the same heat exchange surface as the single-pressure one, there are additional costs. It is caused by:

- a steel plate for dividing the condenser,
- splitting the inlet area into two sections,
- more difficult assembly.

Therefore the price is increased by 74875 USD, which means that the total cost is 2297800 USD. The payback time of this additional investment is

$$t_{payback} = \frac{74875}{546 \cdot 0.02} = 0.86\ \text{years}. \tag{81}$$

The profit of this power gain for the whole lifetime of power plat is

$$\pi_{lifetime} = 546 \cdot 4000 - 74875 = 2109125\ \mathrm{USD}. \tag{82}$$

5.1.2 Double-pressure condenser with constant TTD

The second type of the condenser has extended heat exchange surface by 32 %. The cost is 2900997 USD (the difference is 678072 USD). The calculation of the payback time and the profit is an analogy of the previous one:

$$t_{payback} = \frac{678072}{1290 \cdot 0.02} = 3.29\ \text{years}, \tag{83}$$

$$\pi_{lifetime} = 1290 \cdot 4000 - 678072 = 4481928\ \mathrm{USD}. \tag{84}$$

5.2 Layout

Using the lateral exhaust of the turbine brings a big amount of saved concrete, reinforcement material, working hours and steel structures. At first, the volume of concrete of the original layout had to be calculated - volumes of all walls, columns and floors. Then the building was lowered by 6.5 meters and the calculation was done again. The actual values of volumes cannot be published. The difference between the volume of concrete in the original layout and the new solution is 14.3 %.

Buildings are usually supplied by local suppliers, therefore the cast cannot be quantified in a monetary value. The supplier is selected in a tender in a location of the new power plant.

Conclusion

This diploma thesis dealt with optimizing a reference steam tail package, which is being actively offered by the steam turbine manufacturer to potential clients. Two types of improvements were suggested and both the technical and economic point of view were taken into account.

At the beginning, the reader was introduced to the issue of thermodynamic cycles and their combining, which was necessary for understanding the following text. In the second part, a combined cycle power plant was described as well as its main components. A few different arrangements of a combined cycle power plant were mentioned and heat balances of these examples were calculated in GT PRO program (Thermoflow software package). The next part was focused on the description of actual triple-pressure system to be optimized. The fourth part presents the solution of the optimization and improvements, which is followed by the technical and economic evaluation.

The way to improve the power output of the power plant was to replace the single-pressure condenser by the double-pressure one. Two possible limit options were calculated: with preserved heat exchange surface and with preserved terminal temperature difference in relation to the original condenser. The first mentioned brings the the increase in electrical power output by 546 kW. Using this solution results in additional investment costs which make 74875 USD (approximately 1.8 million CZK). The payback time of this investment is 0.9 year. The profit for the whole lifetime of the power plant is 2109125 USD (51 million CZK).

In the second case, the terminal temperature difference was preserved and the power gain is 1 290 kW, the difference in the costs is 678072 USD (16.3 million CZK) and the payback time is 3.3 years. If a potential customer agrees with this more expensive solution, the profit for the whole lifetime of the power plant is 4481928 USD (108 million CZK).

The LP part had to be partly redesigned to provide two different condensing pressures. The common exhaust area was split into two separate exhaust necks. The double-pressure condenser is divided in the middle by a stainless steel plate for the same reason.

The next suggested improvement is aimed at optimizing the layout of the machine hall. The original layout where the condenser is under the steam turbine wass replaced by the new one, which has the condenser next to the turbine. The machine hall building was lowered by 6.5 m, and 14.3 % of concrete, steel reinforcements and working hours were saved.

Of course, both of these improvements can be combined together and they are not limited to being used only in combined cycle power plants. They can also be applied in fossil, nuclear, biomass power plants, etc.

The theme of the further development can be a detailed design of a double-pressure condenser as well as its design in lateral use. Another issue for follow-up work is for example a different solution of dividing the LP part also in detail design.

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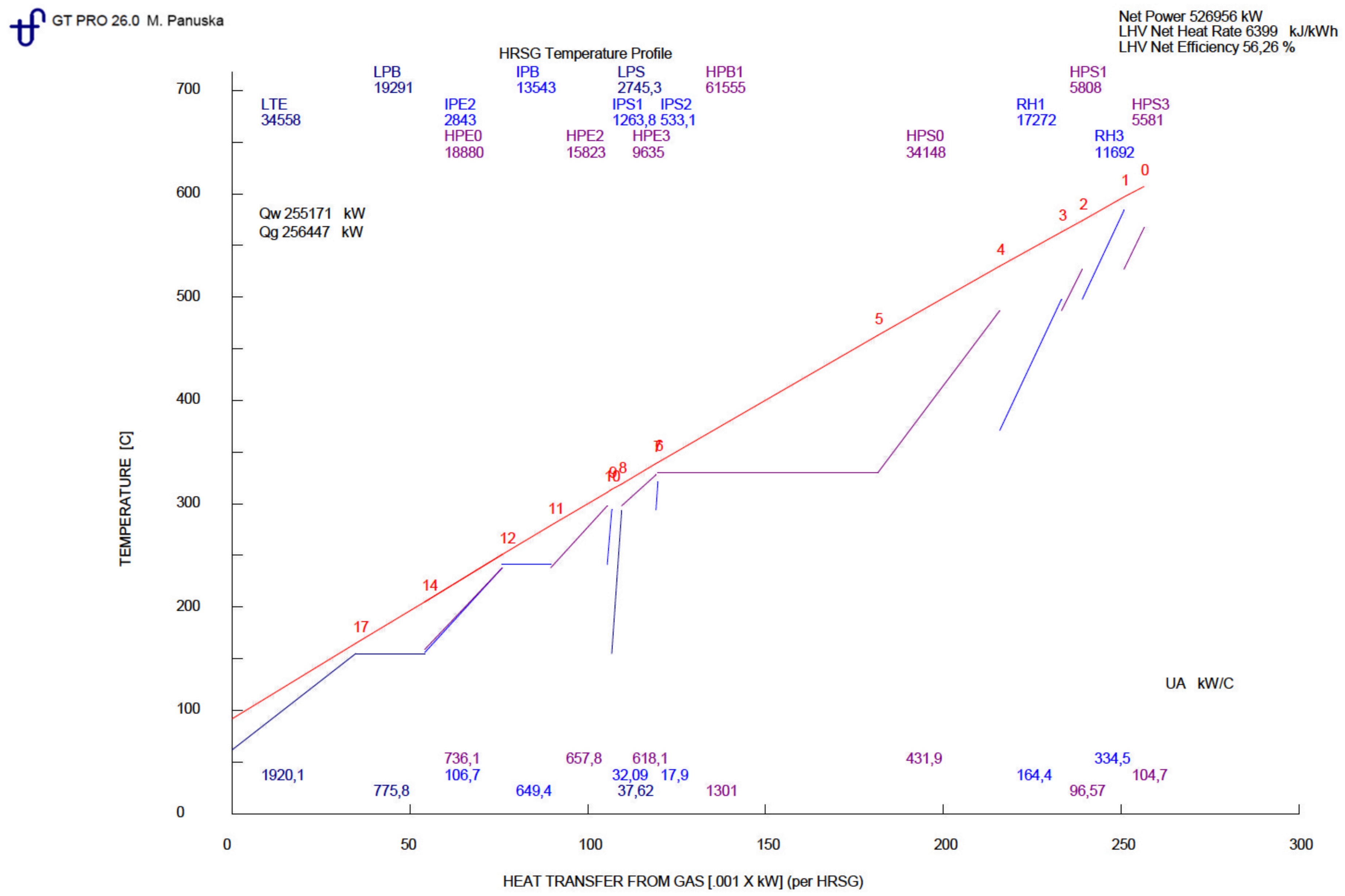
List of Figures

List of Tables

Appendix A
T-Q diagram of HRSG in triple-pressure cycle

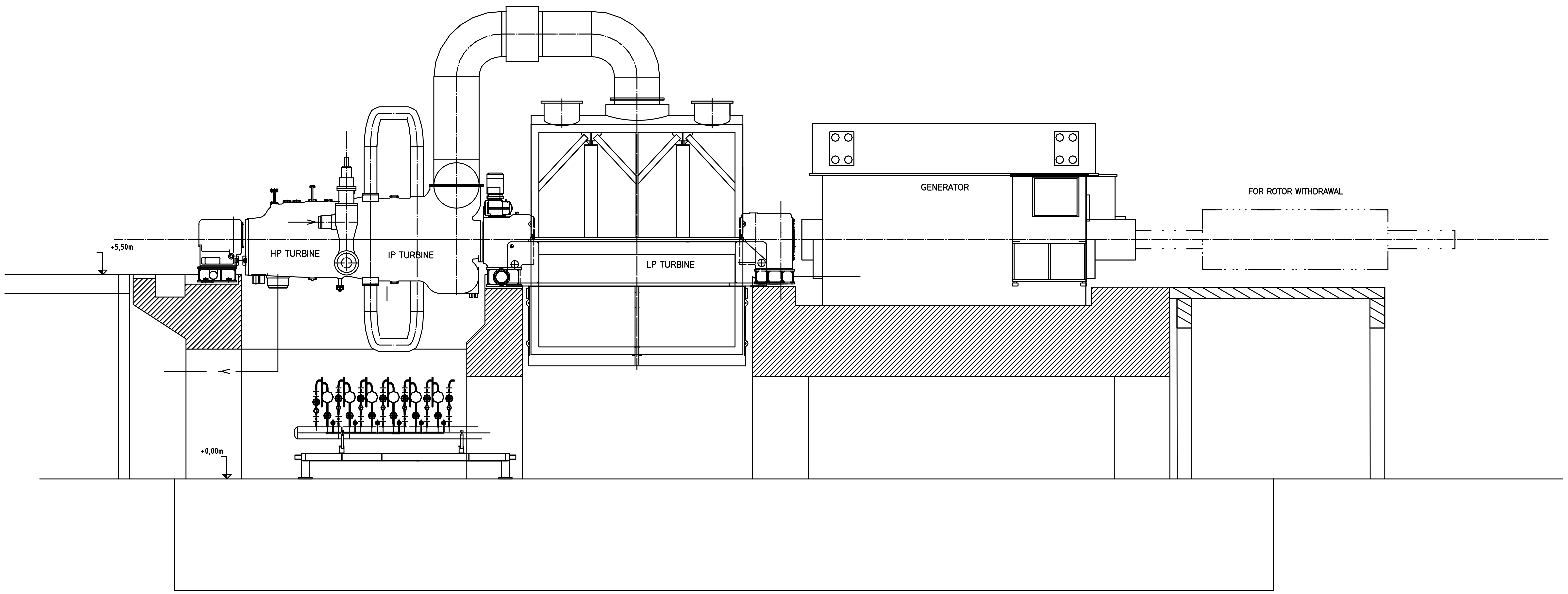

GT PRO 26.0 M. Panuska
Net Power 526956 kW
LHV Net Heat Rate 6399 kJ/kWh
LHV Net Efficiency 56,26 %
HRSG Temperature Profile
LTE 34558
LPB 19291
IPE2 2843
HPE0 18880
IPB 13543
HPE2 15823
LPS 2745,3
IPS1 1263,8
IPS2 533,1
HPE3 9635
HPB1 61555
HPS0 34148
RH1 17272
HPS1 5808
RH3 11692
HPS3 5581
Qw 255171 kW
Qg 256447 kW
UA kW/C
1920,1
775,8
736,1
106,7
649,4
657,8
618,1
32,09
37,62
17,9
1301
431,9
164,4
96,57
334,5
104,7
TEMPERATURE [C]
HEAT TRANSFER FROM GAS [.001 X kW] (per HRSG)

Appendix B
Piping and Instrumentation Diagram


BYPASS INLET
BYPASS WARMING
LIVE STEAM INLET
HOT REHEAT STEAM INLET
COLD REHEAT STEAM OUTLET
GLANDS FEEDING SYSTEM INLET
IP valves chamber
HP valves chamber
HP/IP part of the turbine
LP part of the turbine
Safety rupture disk
Exhaust neck
Generator
Generator air cooler
Lubricating oil cooler
HP hydraulic aggregate
Water ring vacuum pump No. 1
Water ring vacuum pump No. 2
Vacuum pump cooler
CCS - strainer
CCS - ball injector
Condenser Cleaning System
COOLING WATER
Condenser
Flash tank
Vent Steam Condenser Fan
VENT STEAM CONDENSER (VSC)
TO FEED WATER TANK
DEMI WATER FILLING
Condensate pump No.1
Condensate pump No.2
Condensate pump No.3
GRAFICKÉ SYMBOLY
GRAPHICAL SYMBOLS
All equipment which is not at this drawing will not be (according to usual requirements of clients) in scope of this turbine manufacturer.
DRAINING HEADER(S) MUST BE PLACED ABOVE MAX. CONDENSATE LEVEL IN THE FLASH TANK
Michal Panuška
2. 3. 2017
P&ID of Pre-engineered module

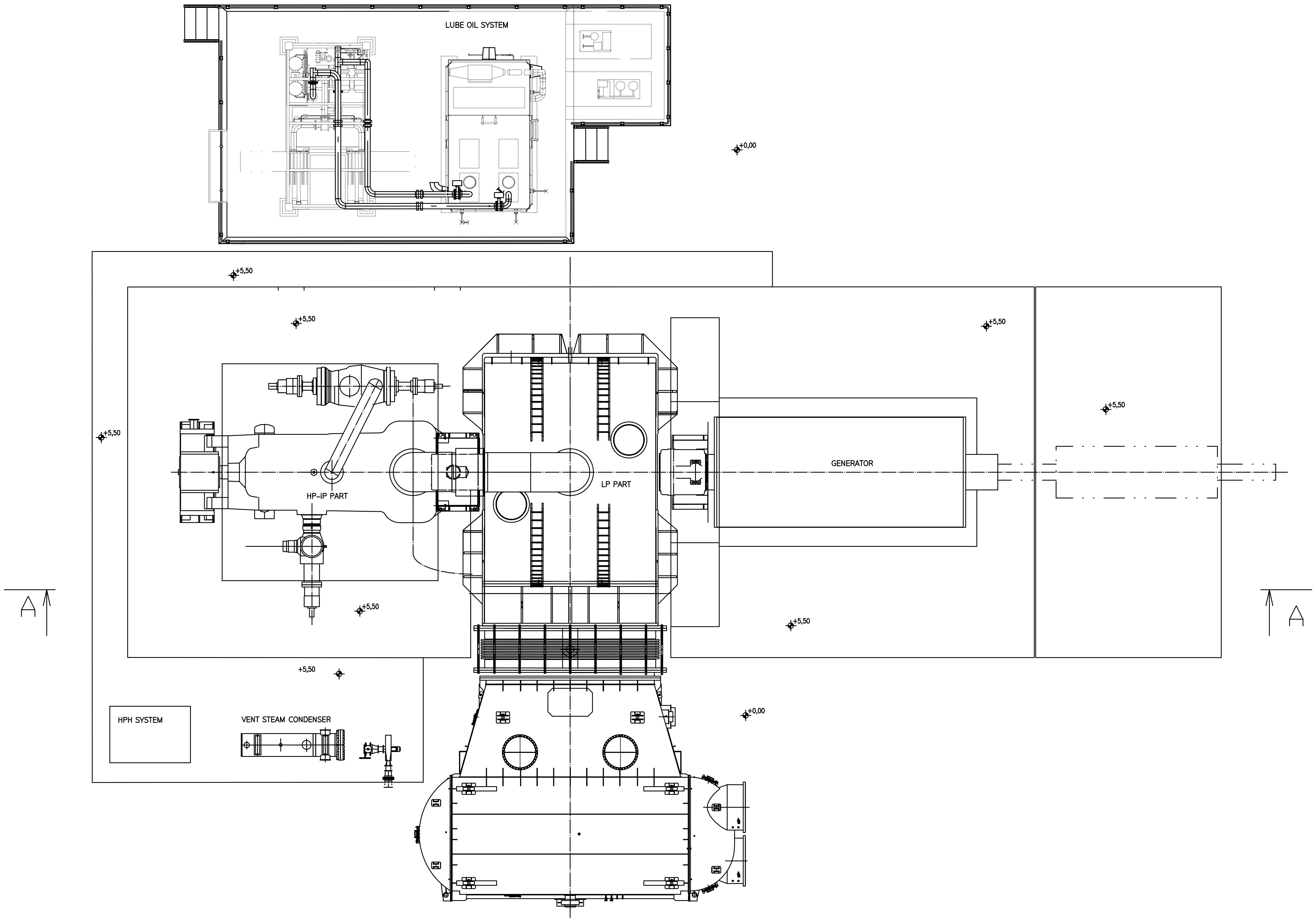
Appendix C
Layout of machine hall

LONGITUDINAL SECTION A-A



Author: Michal Panuška	Date: 30. 3. 2017
Project: LAYOUT - LONGITUDINAL SECTION	

LAYOUT LEVEL +5,50m



Author: Michal Panuška	Date: 30. 3. 2017
Project: LAYOUT - LEVEL +5.5 M	