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ENERGY INSTITUTE
ENERGETICKÝ ÚSTAV

THE DESIGN OF A DUAL PRESSURE HORIZONTAL HEAT RECOVERY STEAM GENERATOR (HRSG) UTILIZING FLUE GAS FROM A NATURAL GAS BURNING TURBINE

NAVRH DVOUTLAKÉHO HORIZONTÁLNÍHO KOTLE NA ODPADNÍ TEPLO (HRSG) ZA PLYNOVOU TURBÍNOU NA ZEMNÍ PLYN

BACHELOR'S THESIS
BAKALÁŘSKÁ PRÁCE

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Master's Thesis Assignment

Institut:	Energy Institute
Student:	**Bc. Jan Pauliny**
Degree programm:	Mechanical Engineering
Branch:	Power Engineering
Supervisor:	**Ing. Martin Lisý, Ph.D.**
Academic year:	2015/16

As provided for by the Act No. 111/98 Coll. on higher education institutions and the BUT Study and Examination Regulations, the director of the Institute hereby assigns the following topic of Master's Thesis:

The design of a duel pressure horizontal heat recovery steam generator (HRSG) utilizing flue gas from a natural gas burning turbine

Brief description:

The design of a duel pressure horizontal heat recovery steam generator (HRSG), utilizing flue gas from a natural gas burning turbine
The mass flow rate of the flue gas is 30kg/s, the flue gas temperature is 490°C
Superheated steam parameters: high pressure circuit 4.6MPa; 450°C
Low pressure circuit 0.46MPa; 170°C
Feed water temperature 53°C
Volumetric composition of the flue gas: O_2 = 13.9%; Ar = 0.9%; N_2 = 73%; CO_2 = 4.4%; H_2O = 7.8%

Master's Thesis goals:

To design of a duel pressure horizontal heat recovery steam generator (HRSG), utilizing flue gas from a natural gas burning turbine, including the sizing of the heat exchangers.

Bibliography:

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Students are required to submit the thesis within the deadlines stated in the schedule of the academic year 2015/16.

In Brno, 27. 11. 2015

L. S.

doc. Ing. Jiří Pospíšil, Ph.D.	doc. Ing. Jaroslav Katolický, Ph.D.
Director of the Institute	Dean

ABSTRACT

This master's thesis deals with the design of a dual pressure horizontal heat recovery steam generator (HRSG), utilizing the residual thermal energy of flue gases exhausted by a natural gas turbine. Included is: the design and sizing of individual heat exchangers and their general configuration according to the required outlet steam parameters, and the given inlet flue gas parameters. Furthermore, this thesis includes the design and associated sizing calculations of steam drums, downcomer tubes, and riser tubes. This thesis concludes by calculating and verifying pressure losses between the inlet and outlet of the HRSG. A significant part of this thesis is comprised of a technical drawing attached in the appendix.

KEYWORDS

Heat recovery steam generator, flue gas duct, downcomer tubers, riser tubes, heat exchanger

ABSTRAKT

Tato diplomová práce se zabývá návrhem dvojtlakého horizontálního kotle využívající teplo spalin za spalovací turbínou na zemní plyn. Zahrnuje návrh a výpočet jednotlivých výměníků, jejich základní uspořádání s ohledem na požadované parametry výstupní páry a dané vstupní a výstupní parametry spalin. Dále tato práce zahrnuje výpočet a konstrukční návrh parních bubnů, zavodňovacích trubek a převáděcích potrubí. Tato práce je zakončena výpočtem a prověřením tlakových ztrát mezi vstupem a výstupem kotle. Důležitou součástí této práce je přiložena výkresová dokumentace.

KLÍČOVÁ SLOVA

kotel na odpadní teplo, spalinový kanál, zavodňovací potrubí, převáděcí potrubí, tepelní výměník

BIBLIOGRAPHIC CITATION

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DECLARATION OF AUTHENTICITY

I, Jan Pauliny, declare that I prepared this master's thesis independently and disclosed all sources and literature.

Brno 27. May 2016 …………………………………………...

Jan Pauliny

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INTRODUCTION

A heat recovery steam generator (HRSG) is a boiler which takes advantage of waste or residual heat and uses it to produce steam. These boilers can be used in either a cogeneration power plant or a combined cycle power plant. In a cogeneration power plant, the heat form combustion is partly used to produce electricity and the rest of the heat is transferred into steam, through the HRSG, and used in other industrial applications. In combined cycle power plants (seen in *figure 0-1*) a combustion turbine is used to produce electricity and the exhaust gases (flue gas) then flows through a HRSG where steam is produced and then used to power a steam turbine to produce additional electricity, thus increasing the efficiency of the cycle.

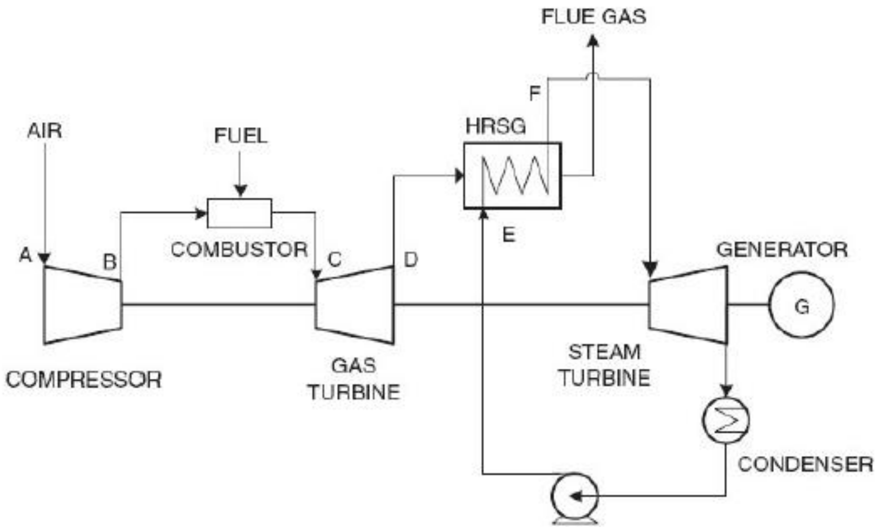

Figure 0-1 Combined cycle power plants [10]

In operation hot flue gas is contained within the inner liners of the HRSG casing (flue gas duct) and come in contact with heat transfer surfaces, as the pass through the HRSG, which transfer heat from the flue gas to the water/steam. Typically, the flue gas enters directly into a stack after passing through the HRSG. The heat transfer surfaces are arraigned inside the flue gas duct) and include; economizers, evaporators, and superheaters. An economizer is a gas to water heat exchanger which serves to preheat the water entering the evaporator. An evaporator is a water/wet steam heat exchanger where the phase change form liquid to gas takes place. A superheater is a gas to dry steam heat exchanger which serves to produce superheated steam.

HRSGs can be categorized into two main groups, depending on the direction of flue gas flow, they can be of either the vertical or horizontal configuration. Each of these configurations can be further categorized into three main types, according to their operating principle, these types include; natural circulation HRSGs, forced circulation HRSGs, and once through HRSGs. Natural circulation HRSGs are most commonly in a horizontal configuration, while forced circulation and once through HRSGs are typically in the vertical configuration. Another general way to categorize HRSGs is according to number of different outlet steam pressures they produce. In this manner a HRSG can be a single, dual, or even triple pressure HRSG. The use of multiple pressure circuits can enable the HRSG to recover more heat form the flue gas and thus increase it efficiency.

Some HRSGs are equipped with duct burners which are used to increase the steam production rate when the parameters of the flue gas form the combustion turbine are insufficient. HRSGs can also be equipped with built in systems for reducing emissions, such as selective catalytic reduction systems which reduce nitrogen oxides in the exhaust gasses.

1 GENERAL DESCRIPTION

The HRSG designed in this thesis is a dual pressure horizontal HRSG with natural circulation. It consists of 9 individual heat exchangers arranged inside the flue gas duct (the inner liners of the HRSG casing). The high pressure circuit includes the economizer, which has been split into 3 heat exchangers, the evaporator, and the superheater, which has been split into 2 heat exchangers. The feed water enters the first heat exchanger of the economizer and flows consecutively through all three exchangers where the water is preheated to a temperature slightly lower than the saturation temperature. This temperature difference is called the approach point. The water then enters the high pressure steam drum from which it flows through the downcomer tubes into the evaporator where the water becomes a water/steam mixture. This water/steam mixture is collected in headers and then flows through the riser tubes back into steam drum where the saturated steam is separated from the water (phase separation). The saturated steam travels from the top of the steam drum through the first superheater heat exchanger, where the steam becomes superheated. The steam then enters a header where feed water, at a temperature of 53 °C, is injected into the superheated steam in order to regulate the final outlet steam parameters. Afterwards the steam passes through the last superheater heat exchanger where the final outlet steam parameters are achieved. The low pressure circuit consists of an economizer, evaporator, and superheater. Each is made up of single heat exchangers. This circuit functions in the same manner as the high pressure circuit, with the omission of the regulation through water injection. In both the high and low pressure circuits the water and heat loss through boiler blowdown is negligible.

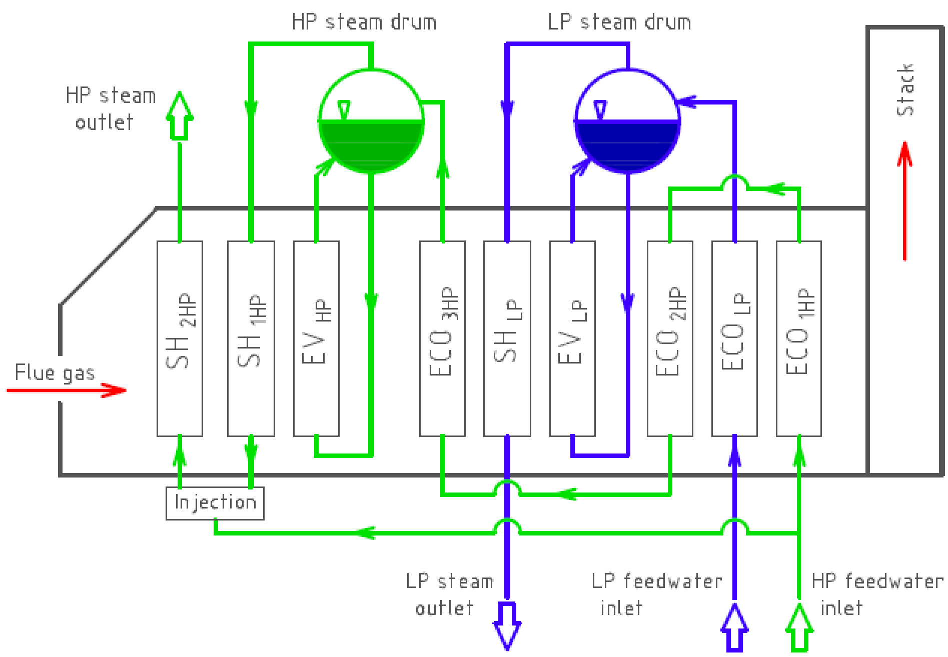

Figure 1-1 HRSG schematic

The design process begins with an outline of the HRSG temperature profile graph which is constructed according to the selected number of heat exchangers and the order in which they are arranged, seen in *figure 1-1*. According to the preliminary temperature profile graph the steam and flue gas parameters are calculated as they enter and exit each heat exchanger. The steam production rate and required heat transfer rate of each heat exchanger is determined. The geometry of the first heat exchanger to come in contact with the flue gas is selected and the inner dimensions of the HRSG casing (the flue gas duct dimensions) are defined accordingly. The rest of the heat exchangers are sized according to their heat transfer requirements and the set dimensions of the flue gas duct. Then the steam drums, downcomer tubes and riser tubes are sized. The final step of the design is to verify that the pressure losses in the flue gas through the HRSG are not excessive.

1.1.1 Given Boiler Parameters

Flue gas inlet parameters:

Temperature $t_{Flue} = 490°C$

Mass flow rate $\dot{M}_{Flue} = 30\, kg/s$

Feedwater inlet parameters:

Temperature $t_{FW} = 53°C$

High pressure steam output parameters:

Pressure $p_1^{HP} = 4.6 MPa$

Temperature $t_1^{HP} = 450°C$

Low pressure steam output parameters:

Pressure $p_1^{LP} = 0.46 MPa$

Temperature $t_1^{LP} = 170°C$

Flue gas volumetric composition:

The volumetric concentration of the individual components which makeup the flue gas are presented in *Table 1.1.*

Parameter	Symbol	Value	Units
O_2 concentration	x_{O2}	0.139	[-]
Ar concentration	x_{Ar}	0.009	[-]
N_2 concentration	x_{N2}	0.73	[-]
CO_2 concentration	x_{CO2}	0.044	[-]
H_2O concentration	x_{H2O}	0.078	[-]

Table 1.1 Volumetric composition, concentrations

Maximum acceptable flue gas pressure loss:

A maximum pressure loss value must be determined to insure proper flue gas exhaust from the boiler. This maximum acceptable pressure loss is dependent on the exhaust parameters of the gas turbine being used. The maximum acceptable pressure loss is assumed to be 1500Pa [7].

$$\Delta p_{Lmax} = 1500 Pa$$

2 THERMAL CALCULATIONS OF THE BOILER

2.1 Preliminary Temperature Profile Graph of the Boiler

The preliminary temperature profile graph of the boiler (seen in *figure 2-1*) is constructed according to the configuration and the number of heat exchangers the boiler is made up of. In both the high and low pressure circuit Δt_{Pi} (the pick point) is chosen to be 10 °C. The value of Δt_{Ap} (the approach point) is chosen to be 5 °C. The approach points insure that the water in high and low pressure the economizers does not begin to boil. The pinch points insure that there is an adequate temperature difference between the water entering the evaporators and the flue gas. This minimum temperature difference is what limits the heat transfer rate of the high pressure circuit. The purpose of splitting the high pressure economizer into three heat exchangers is to further cool the flue gas, and thus recover more heat.

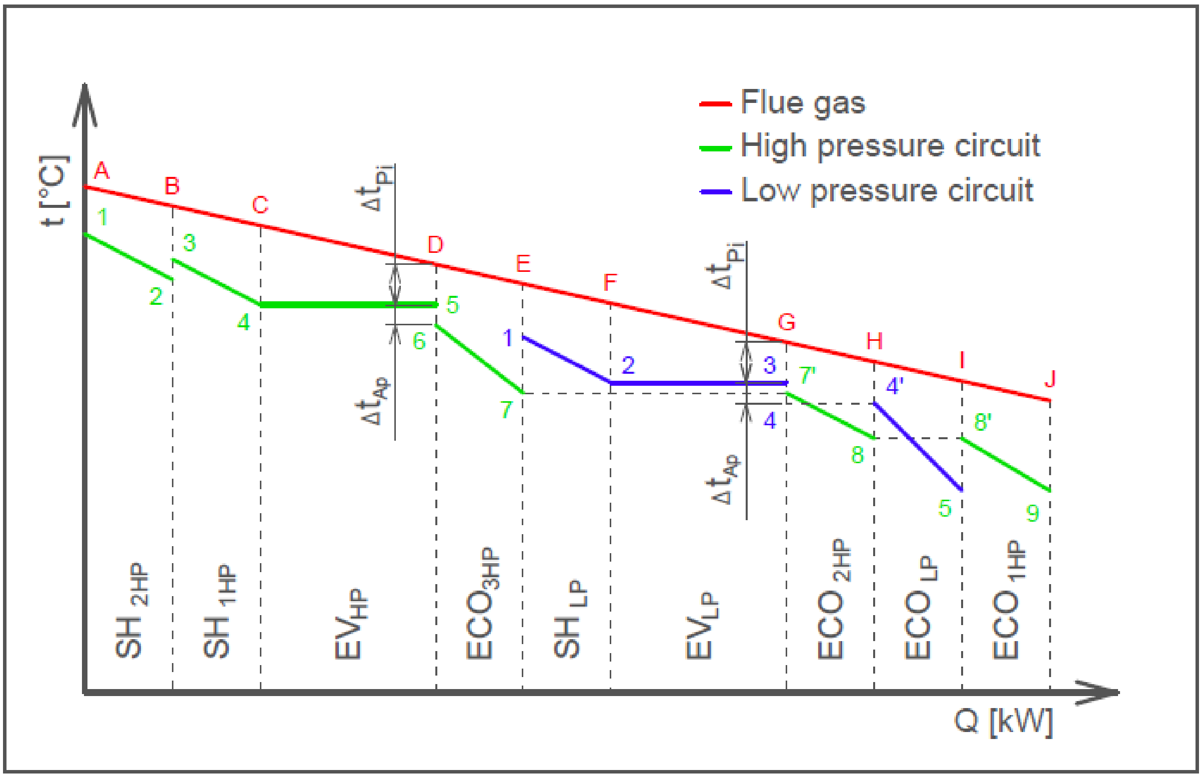

Figure 2-1 HRSG preliminary temperature profile graph

2.2 Selected Parameters Values

Parameters which have been selected are presented in the *Table 2.1*. These parameters include values which are necessary for the construction of the actual temperature profile graph of the boiler

Parameters	Symbol	Value	Units
High pressure superheater SH_{2HP} steam pressure loss	Δp_{SH2}^{HP}	0.1	MPa
High pressure superheater SH_{1HP} steam pressure loss	Δp_{SH1}^{HP}	0.1	MPa
High pressure economizer ECO_{3HP} water pressure loss	Δp_{ECO3}^{HP}	0.1	MPa
Low pressure superheater SH_{LP} steam pressure loss	Δp_{SH}^{LP}	0.1	MPa
High pressure economizer ECO_{2HP} water pressure loss	Δp_{ECO2}^{HP}	0.1	MPa
Low pressure economizer ECO_{LP} water pressure loss	Δp_{ECO}^{LP}	0.3	MPa
High pressure economizer ECO_{1HP} water pressure loss	Δp_{ECO1}^{HP}	0.1	MPa
High pressure superheater SH_{2HP} enthalpy difference	Δi_{SH2}^{HP}	250	kJ/kg
High pressure approach point	Δt_{Ap}^{HP}	5	℃
Low pressure approach point	Δt_{Ap}^{LP}	5	℃
High pressure pinch point	Δt_{Pi}^{HP}	10	℃
Low pressure pinch point	Δt_{Pi}^{HP}	10	℃
High pressure economizer ECO_{3HP} temperature gradient	Δt_{ECO3}^{HP}	105	℃
High pressure economizer ECO_{2HP} temperature gradient	Δt_{ECO2}^{HP}	12	℃
Amount of high pressure feedwater used to regulate outlet steam parameters (water injection)	$\dot{M}_{W\%}$	5	%

Table 2.1 Selected parameters values

2.3 Water/Steam Parameters

In this section the water/steam parameters are determined, for both the high and low pressure circuits, at each point in the preliminary temperature profile graph. The temperature, pressure, and enthalpy at each point is either calculated or determined using X-Steam. The first point of both the high and low pressure circuits are given as the required outlet steam parameters.

2.3.1 High pressure water/steam parameters

Point 1_{HP}:

$p_1^{HP} = 4.6\,MPa$ (given value)

$t_1^{HP} = 450℃$ (given value)

$i_1^{HP} = 3322.65\,kJ/kg$ (determined using X-Steam, f(p,t))

Point 2_{HP}:

$$p_2^{HP} = p_1^{HP} + \Delta p_{SH2}^{HP} = 4.6 + 0.1 = 4.7MPa$$

$$i_2^{HP} = i_1^{HP} - \Delta i_{SH2}^{HP} = 3322.65 - 250 = 3072.65\,kJ/kg$$

$t_2^{HP} = 348.48℃$ (determined using X-Steam, f(p,t))

Point 4HP:

$p_4^{HP} = p_2^{HP} + \Delta p_{SH1}^{HP} = 4.7 + 0.1 = 4.8 MPa$

$t_4^{HP} = 261.4℃$ (X-Steam, saturation temperature)

$i_4^{HP} = 2795.83\, kJ/kg$ (X-Steam, saturated steam)

Point 5HP:

$p_5^{HP} = p_4^{HP} = 4.8 MPa$

$t_5^{HP} = t_4^{HP} 261.4℃$ (saturation temperature)

$i_5^{HP} = 1141.81\, kJ/kg$ (X-Steam, saturated water)

Point 6HP:

$p_6^{HP} = p_5^{HP} = 4.8 MPa$

$t_6^{HP} = t_5^{HP} - \Delta t_{Ap}^{HP} = 261.4 - 5 = 256.4℃$

$i_6^{HP} = 1116.99\, kJ/kg$ (determined using X-Steam, f(p,t))

Point 7HP:

$p_7^{HP} = p_6^{HP} + p_{ECO3}^{HP} = 4.8 + 0.1 = 4.9 MPa$

$t_7^{HP} = t_6^{HP} - t_{ECO3}^{HP} = 256.4 - 105 = 151.4℃$

$i_7^{HP} = 641.03\, kJ/kg$ (determined using X-Steam, f(p,t))

Point 8HP:

$p_8^{HP} = p_7^{HP} + \Delta p_{ECO2}^{HP} = 4.9 + 0.1 = 5 MPa$

$t_8^{HP} = t_7^{HP} - \Delta t_{ECO2}^{HP} = 151.4 - 12 = 139.4℃$

$i_8^{HP} = 589.67\, kJ/kg$ (determined using X-Steam, f(p,t))

Point 9HP:

$p_9^{HP} = p_8^{HP} - \Delta P_{ECO1}^{HP} = 5 + 0.1 = 5.1 MPa$

$t_9^{HP} = t_{FW} = 53℃$ (given value)

$i_9^{HP} = 226.23\, kJ/kg$ (determined using X-Steam, f(p,t))

Point 3HP:

Water injection is used as a means of regulating the parameters of the high pressure steam outlet. Approximately 5% of the total high pressure feedwater is injected into the high pressure circuit between the two heat exchangers of the high pressure superheaters (SH_{2HP} and SH_{1HP}) [7]. The enthalpy at point 3_{HP} is calculated according to equation *(2.1)*.

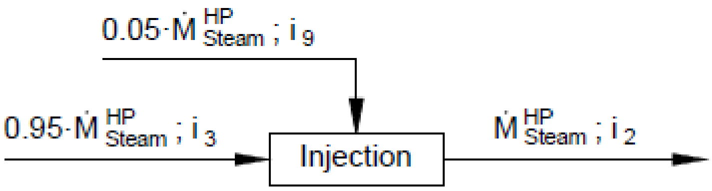

Figure 2-2 Feedwater injection

Conservation equation:

$$\dot{M}_{Steam}^{HP} \cdot i_2^{HP} = 0.95\dot{M}_{Steam}^{HP} \cdot i_3^{HP} + 0.05\dot{M}_{Steam}^{HP} \cdot i_9^{HP} \quad (2.1)$$

$$i_3^{HP} = \frac{i_2^{HP} - 0.05 \cdot i_9^{HP}}{0.95} = \frac{3072.65 - 0.05 \cdot 226.23}{0.95} = 3222.46\,kJ/kg$$

$$p_3^{HP} = p_2^{HP} = 4.7MPa$$

$t_3^{HP} = 408.46°\mathrm{C}$ (determined using X-Steam, f(p,t))

2.3.2 Low pressure water/steam parameters

Point 1LP:

$p_1^{LP} = 0.46MPa$ (given)

$t_1^{LP} = 170°\mathrm{C}$ (given)

$i_1^{LP} = 2792.98\,kJ/kg$ (determined using X-Steam, f(p,t))

Point 2LP:

$$p_2^{LP} = p_1^{LP} + \Delta p_{ECO}^{LP} = 0.46 + 0.1 = 0.56MPa$$

$t_2^{LP} = 156.15°\mathrm{C}$ (X-Steam, saturation temperature)

$i_2^{LP} = 2753.12\,kJ/kg$ (X-Steam, saturated steam)

Point 3LP:

$$p_3^{LP} = p_2^{LP} = 0.56MPa$$

$t_3^{LP} = 156.15°\mathrm{C}$ (X-Steam, saturation temperature)

$i_3^{LP} = 658.88\,kJ/kg$ (X-Steam, saturated water)

Point 4LP:

$$p_4^{LP} = p_3^{LP} = 0.56MPa$$

$$t_4^{LP} = t_3^{LP} - \Delta t_{Ap}^{LP} = 156.15 - 5 = 151.15°\text{C}$$

$i_4^{LP} = 637.28\, kJ/kg$ (determined using X-Steam, f(p,t))

Point 5_{LP}:

$$t_5^{LP} = t_{FW} = 53°\text{C}$$

$$p_5^{LP} = p_4^{LP} + \Delta p_{ECO}^{LP} = 0.56 + 0.3 = 0.86 MPa$$

$i_5^{LP} = 222.6\, kJ/kg$ (determined using X-Steam, f(p,t))

2.4 Flue Gas Parameters

2.4.1 Flue gas density

The flue gas density is calculated with the help of the volumetric concentration of the individual flue gas components (seen in *Table 1.1*) and their respective densities (seen in *Table 2.2*). This value is calculated according to equation *(2.2)*. The calculated density as well as the density of the individual components are under normal conditions, a temperature of 0°C and pressure of 0.101MPa.

Parameter	Symbol	Value	Units
O_2 density	ρ_{O2}	1.4289	$[kg/Nm^3]$
Ar density	ρ_{Ar}	1.7839	$[kg/Nm^3]$
N_2 density	ρ_{N2}	1.2505	$[kg/Nm^3]$
CO_2 density	ρ_{CO2}	1.9768	$[kg/Nm^3]$
H_2O density	ρ_{H2O}	0.804	$[kg/Nm^3]$

Table 2.2 Densities of individual flue gas components

$$\rho_{Flue} = x_{O2} \cdot \rho_{O2} + x_{Ar} \cdot \rho_{Ar} + x_{N2} \cdot \rho_{N2} + x_{CO2} \cdot \rho_{CO2} + x_{H2O} \cdot \rho_{H2O} \tag{2.2}$$

$$\rho_{Flue} = 0.139 \cdot 1.4289 + 0.009 \cdot 1.7839 + 0.73 \cdot 1.2505 + 0.044 \cdot 1.9768 + 0.07 \cdot 0.804$$

$$\rho_{Flue} = 1.2772\, kg/Nm^3$$

2.4.2 Flue gas volumetric flow rate, under normal conditions

The flue gas volumetric flow rate, under normal conditions, is required to determine the flue gas enthalpy and speed throughout the boiler. This value is calculated according to equation *(2.3)*.

$$\dot{M}_{VFlue} = \frac{\dot{M}_{Flue}}{\rho_{Flue}} \tag{2.3}$$

$$\dot{M}_{VFlue} = \frac{30}{1.2772} = 23.49\, Nm^3/s$$

2.4.3 Flue gas enthalpy calculations at various points

The flue gas enthalpy is determined as a function of the flue gas temperature. However, this correlation must be determined for the specific flue gas composition. Flue gas enthalpy at a given temperature is calculated with the help of the known enthalpy of each individual component of the flue at the given temperature, and its volumetric concentration (*Table 1.1*). This calculation is presented in equation *(2.4)*. The flue gas enthalpy at several temperatures is presented in *Table 2.3*.

$$I_{Flue} = x_{O2} \cdot I_{O2} + x_{Ar} \cdot I_{Ar} + x_{N2} \cdot I_{N2} + x_{CO2} \cdot I_{CO2} + x_{H2O} \cdot I_{H2O} \tag{2.4}$$

For example, the flue gas enthalpy is calculated for a temperature 100°C below. This calculation is repeated for all temperatures in *Table 2.3*.

$$I_{Flue:100} = 0.139 \cdot 132 + 0.009 \cdot 93 + 0.73 \cdot 130 + 0.044 \cdot 170 + 0.078 \cdot 150$$

$$I_{Flue:100} = 133.27\,kJ/Nm^3$$

Temperature [°C]	Enthalpy $[kJ/Nm^3]$					
	I_{O2}	I_{Ar}	I_{N2}	I_{CO2}	I_{H2O}	I_{Flue}
100	132	93	130	170	150	133.27
200	267	186	260	357	304	268.01
300	407	278	392	559	463	405.95
400	551	372	527	772	626	547.44
500	699	465	666	994	795	693.27
600	850	557	804	1225	969	839.57

Table 2.3 Enthalpy of the flue gas (calculated) and flue gas components at normal conditions, a temperature of 0°C and pressure of 0.101 MPa.

Point A:

The enthalpy of the flue gas at point A will be calculated according to the temperature at this point, which is also one of the given inlet parameters.

Temperature $t_A = t_{Flue} = 490℃$

The temperature at point A lies between 400°C and 500°C. Linear interpolation is used to calculate enthalpy of the flue gas at 490 °C, as seen below in equation *(2.5)*.

$$I_A = \frac{t_A - 400}{100} \cdot (I_{Flue:500} - I_{Flue:400}) + I_{Flue:400} \tag{2.5}$$

$$I_A = \frac{490 - 400}{100} \cdot (693.27 - 547.44) + 547.44 = 678.69\,kJ/Nm^3$$

Point D:

The enthalpy of the flue gas at point D will be calculated according to the temperature at this point. This temperature is calculated as the sum of the saturation temperature of the high pressure circuit and the high pressure pinch point, as seen below.

Temperature $t_D = t_5^{HP} + t_{Pi}^{HP} = 261.4 + 10 = 271.4°C$

The temperature at point D lies between 200°C and 300°C. Linear interpolation is used to calculate enthalpy of the flue gas at 271.4°C, as seen below in equation *(2.6)*.

$$I_D = \frac{t_D - 200}{100} \cdot (I_{Flue:300} - I_{Flue:200}) + I_{Flue:200} \tag{2.6}$$

$$I_A = \frac{271.40 - 200}{100} \cdot (405.95 - 268.01) + 268.01 = 366.5\, kJ/Nm^3$$

Point G:

The enthalpy of the flue gas at point G will be calculated according to the temperature at this point. This temperature is calculated as the sum of the saturation temperature of the low pressure circuit and the low pressure pinch point, as seen below.

Temperature $t_G = t_3^{LP} + t_{Pi}^{LP} = 156.15 + 10 = 166.15°C$

The temperature at point G lies between 100°C and 200°C. Linear interpolation is used to calculate enthalpy of the flue gas at 166.15°C, as seen below in equation *(2.7)*.

$$I_G = \frac{t_G - 100}{100} \cdot (I_{Flue:200} - I_{Flue:100}) + I_{Flue:100} \tag{2.7}$$

$$I_G = \frac{166.15 - 100}{100} \cdot (268.01 - 133.27) + 133.27 = 222.4\, kJ/Nm^3$$

2.5 Rate Heat Loss from the Boiler Due to Radiation, Calculation

The rate of heat loss from the boiler through radiation and convection is determined as a function of the boilers maximum usable thermal power and is also dependent on the type of fuel used, as seen in equation *(2.8)* according to standard ČSN EN 12952-15 [11]. This heat loss rate takes into account the heat escaping from the boiler through the boiler walls and into the surroundings. Losses can be reduced by increasing the quality or quantity of insulation in the boiler walls.

$$Q_{RC} = C \cdot {Q_N}^{0.7} \tag{2.8}$$

Where:

Q_{RC} is the rat of heat loss through radiation and convection [MW]

Q_N is the maximum usable thermal power (the rate of heat entering the boiler) [MW]

C is a constant (C = 0.0113, for boilers that use liquid fuels or natural gas)

Maximum usable thermal power:

The maximum usable thermal power is the rate of heat entering the boiler in the flue gas at the inlet, or point A. This value is calculated as the product of the flue gas enthalpy and volumetric flow rate at point A, as seen below in equation *(2.9)*.

$$Q_N = Q_A = I_A \cdot \dot{M}_{VFlue} \tag{2.9}$$

$$Q_N = Q_A = 678.69 \cdot 23.49 = 15941.29kW$$

Heat loss through radiation and convection:

The rate of heat loss through radiation and convection is calculated according to equation *(2.8)*, described above. The percentage of usable thermal power that is lost though the boiler walls by means of radiation and convection is calculated in equation *(2.10)*.

$$Q_{RC} = C \cdot {Q_N}^{0.7} = 0.0113 \cdot 15.94129^{0.7} = 0.07849MW$$

$$Q_{RC\%} = \frac{Q_{RC}}{Q_N} \cdot 100 = \frac{0.07849}{15.94129} \cdot 100 = 0.49\% \tag{2.10}$$

2.6 Steam Production Rates and Heat Transfer rates of Each Individual Heat Exchanger

The steam production rate is calculated for both the high pressure and the low pressure circuit. Based on these production rates, the rate of heat transferred through each individual heat exchanger is calculated of the high and low pressure circuits respectively.

2.6.1 High pressure steam production rate

The heat released from the flue gas will be transferred mainly to the steam and water respectively, and a small portion of the heat will escape the boiler by means of radiation and convection. There for the rate heat leaving the flue gas between points A and D must be equal to the rate heat transferred to the high pressure water/steam between points 1_{HP} and 6_{HP}, while taking into account the rate of heat loss through radiation and convection. This conservation of energy calculation is presented in equation *(2.12)*. The rate of heat leaving the flue gas between points A and D is calculated through equation *(2.11)*.

Rate of heat released between points A and D:

$$Q_{A-D} = \dot{M}_{VFlue} \cdot (I_A - I_D) = 23.49 \cdot (678.69 - 366.50) = 7332.81kW \tag{2.11}$$

Rate if heat absorbed between points 1_{HP} and 6_{HP}:

$$Q_{1-6}^{HP} = Q_{A-D} \cdot \left(1 - \frac{Q_{RC\%}}{100}\right) = 7332.81 \cdot \left(1 - \frac{0.49}{100}\right) = 7296.7kW \tag{2.12}$$

The conservation of energy enables equation *(2.13)* to be used to determine the high pressure steam production rate, according to the water and steam parameters calculated earlier.

Each heat exchanger will not have the same water/steam mass flow rate, because 5% of the total high pressure outlet mass flow rate is introduced into the high pressure circuit through water injection between superheaters SH_{2HP} and SH_{1HP}. This water is used to regulate the steam outlet parameters. The fact that water is injected into the high pressure must be taken into account.

$$Q_{6-1}^{HP} = \dot{M}_{Steam}^{HP} \cdot [(i_1^{HP} - i_2^{HP}) + 0.95 \cdot (i_1^{HP} - i_6^{HP}) + 0.05 \cdot (i_2^{HP} - i_9^{HP})] \tag{2.13}$$

$$\dot{M}_{Steam}^{HP} = \frac{Q_{6-1}^{HP}}{[(i_1^{HP} - i_2^{HP}) + 0.95 \cdot (i_1^{HP} - i_6^{HP}) + 0.05(i_2^{HP} - i_9^{HP})]}$$

$$\dot{M}_{Steam}^{HP} = \frac{7296.7}{[(3322.65 - 3072.65) + 0.95 \cdot (3322.65 - 1116.99) + 0.05(3072.65 - 226.23)]}$$

$$\dot{M}_{Steam}^{HP} = 3.24\, kg/s$$

The high pressure steam is produced at a rate of approximately 3.24 kg/s, or 11.66 t/h.

2.6.2 Heat transfer rates of each individual heat exchanger in the high pressure circuit of the boiler

Heat transfer rate of the high pressure superheater SH_{2HP}:

The total high pressure outlet steam mass flow rate of 3.24kg/s (as calculated above) flows through SH_{2HP}. This includes the 5% of mass flow rate added through water injection. The heat transfer rate is calculated between points 1_{HP} and 2_{HP}, according to equation *(2.14)*.

$$Q_{SH2}^{HP} = \dot{M}_{Steam}^{HP} \cdot (i_1^{HP} - i_2^{HP}) \tag{2.14}$$

$$Q_{SH2}^{HP} = 3.24 \cdot (3322.65 - 3072.65) = 810.67 kW$$

Heat transfer rate of the high pressure superheater SH_{1HP}:

From the total high pressure outlet steam mass flow rate of 3.24kg/s, only 95% flows through SH_{1HP}. This is because 5% of the total outlet mass flow rate is introduced into the high pressure circuit through water injection later on. The heat transfer rate is calculated between points 3_{HP} and 4_{HP}, according to equation *(2.15)*.

$$Q_{SH1}^{HP} = 0.95 \cdot \dot{M}_{Steam}^{HP} \cdot (i_3^{HP} - i_4^{HP}) \tag{2.15}$$

$$Q_{SH1}^{HP} = 0.95 \cdot 3.24 \cdot (3222.46 - 2795.83) = 1314.27 kW$$

Heat transfer rate of the high pressure evaporator EV_{HP}:

From the total high pressure outlet steam mass flow rate of 3.24kg/s, only 95% flows through EV_{HP}. The heat transfer rate is calculated between points 4_{HP} and 6_{HP}, according to equation *(2.16)*.

$$Q_{EV}^{HP} = 0.95 \cdot \dot{M}_{Steam}^{HP} \cdot (i_4^{HP} - i_6^{HP}) \tag{2.16}$$

$$Q_{EV}^{HP} = 0.95 \cdot 3.24 \cdot (2795.83 - 1116.99) = 5171.76 kW$$

Heat transfer rate of the high pressure economizer ECO_{3HP}:

From the total high pressure outlet steam mass flow rate of 3.24kg/s, only 95% flows through ECO_{3HP}. The heat transfer rate is calculated between points 6_{HP} and 7_{HP}, according to equation *(2.17)*.

$$Q_{ECO3}^{HP} = 0.95 \cdot \dot{M}_{Steam}^{HP} \cdot (i_6^{HP} - i_7^{HP}) \tag{2.17}$$

$$Q_{ECO3}^{HP} = 0.95 \cdot 3.24 \cdot (1116.99 - 641.03) = 1466.23kW$$

Heat transfer rate of the high pressure economizer ECO_{2HP}:

From the total high pressure outlet steam mass flow rate of 3.24kg/s, only 95% flows through ECO_{2HP}. The heat transfer rate is calculated between points 7_{HP} and 8_{HP}, according to equation *(2.18)*.

$$Q_{ECO2}^{HP} = 0.95 \cdot \dot{M}_{Steam}^{HP} \cdot (i_7^{HP} - i_8^{HP}) \tag{2.18}$$

$$Q_{ECO2}^{HP} = 0.95 \cdot 3.24 \cdot (641.03 - 589.67) = 158.21kW$$

Heat transfer rate of the high pressure economizer ECO_{1HP}:

From the total high pressure outlet steam mass flow rate of 3.24kg/s, only 95% flows through ECO_{1HP}. The heat transfer rate is calculated between points 8_{HP} and 9_{HP}, according to equation *(2.19)*.

$$Q_{ECO1}^{HP} = 0.95 \cdot \dot{M}_{Steam}^{HP} \cdot (i_8^{HP} - i_9^{HP}) \tag{2.19}$$

$$Q_{ECO1}^{HP} = 0.95 \cdot 3.24 \cdot (589.67 - 226.23) = 1119.61kW$$

2.6.3 Low pressure steam production rate

The rate heat leaving the flue gas between points E and G must be equal to the rate of heat transferred to the low pressure water/steam between points 1_{LP} and 4_{LP}, while taking into account the rate heat loss through radiation and convection. This conservation of energy calculation is presented in equation *(2.25)*. The rate of heat leaving the flue gas between points E and G is calculated through equation *(2.24)*, however this calculation requires that the enthalpy of the flue gas at point E be determined.

Flue gas parameter at point E:

The calculated heat transfer rates of the individual high pressure heat exchangers are used in conjunction with volumetric flow rate of the flue gas, and the heat loss percentage, calculated earlier, to determine the flue gas enthalpy at point E.

The rate of heat absorbed by the high pressure circuit between points 1_{HP} and 7_{HP} is presented in equation *(2.20)*.

$$Q_{1-7}^{HP} = Q_{SH2}^{HP} + Q_{SH1}^{HP} + Q_{EV}^{HP} + Q_{ECO3}^{HP} \tag{2.20}$$

$$Q_{1-7}^{HP} = 810.67 + 1314.27 + 5171.76 + 1466.23 = 8762.93kW$$

The conservation of energy calculation, equation (2.21), is used to determine in the rate of heat leaving the flue gas between points A and E.

$$Q_{1-7}^{HP} = Q_{A-E} \cdot \left(1 - \frac{Q_{RC\%}}{100}\right) \tag{2.21}$$

$$Q_{A-E} = \frac{Q_{1-7}^{HP}}{1 - \frac{Q_{RC\%}}{100}} = \frac{8762.93}{1 - \frac{0.49}{100}} = 8806.3kW$$

Flue gas enthalpy at point E:

$$Q_{A-E} = \dot{M}_{VFlue} \cdot (I_A - I_E) \tag{2.22}$$

$$I_E = I_A - \left(\frac{Q_{A-B}}{\dot{M}_{VFlue}}\right) = 678.69 - \frac{8806.3}{23.49} = 225.92\, kJ/Nm^3$$

Flue gas temperature at point E:

The flue gas temperature at point E is determined through liner interpolation between known values from *Table 2.3*.

$$t_E = \frac{I_E - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 \tag{2.23}$$

$$t_E = \frac{303.77 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 225.92℃$$

Rate of heat leaving the flue gas between points E and G:

$$Q_{E-G} = \dot{M}_{VFlue} \cdot (I_E - I_G) = 23.49 \cdot (225.92 - 222.4) = 1911.11kW \tag{2.24}$$

Rate of heat absorbed between points 1_{LP} and 4_{LP}:

$$Q_{1-4}^{LP} = Q_{E-G} \cdot \left(1 - \frac{Q_{RC\%}}{100}\right) = 1911.11 \cdot \left(1 - \frac{0.49}{100}\right) = 1901.7kW \tag{2.25}$$

The conservation of energy enables equation *(2.26)* to be used to determine the low pressure steam production rate, according to the water and steam parameters calculated earlier. Each heat exchanger will have the same water/steam mass flow rate, because water injection is not used to regulate the steam outlet parameters in the low pressure circuit.

$$Q_{1-4}^{LP} = \dot{M}_{Steam}^{LP} \cdot (i_1^{LP} - i_4^{LP}) \tag{2.26}$$

$$\dot{M}_{Steam}^{LP} = \frac{Q_{1-4}^{LP}}{i_1^{LP} - i_4^{LP}} = \frac{1901.7}{2792.98 - 637.28} = 0.88\, kg/s$$

The low pressure steam is produced at a rate of approximately 0.88 kg/s, or 3.18 t/h.

2.6.4 Heat transfer rates of each individual heat exchanger in the low pressure circuit of the boiler

Heat transfer rate of the low pressure superheater SH_{LP}:

The total low pressure outlet steam mass flow rate of 0.88kg/s (as calculated above) flows through this superheater. The heat transfer rate is calculated between points 1_{LP} and 2_{LP}, according to equation *(2.27).*

$$Q_{SH}^{LP} = \dot{M}_{Steam}^{LP} \cdot (i_1^{LP} - i_2^{LP}) = 0.88 \cdot (2792.98 - 2753.12) = 35.16kW \tag{2.27}$$

Heat transfer rate of the low pressure evaporator EV_{LP}:

The total low pressure outlet steam mass flow rate of 0.88kg/s flows through this evaporator. The heat transfer rate is calculated between points 2_{LP} and 4_{LP}, according to equation *(2.28).*

$$Q_{EV}^{LP} = \dot{M}_{Steam}^{LP} \cdot (i_2^{LP} - i_4^{LP}) = 0.88 \cdot (2753.12 - 637.28) = 1866.54kW \tag{2.28}$$

Heat transferred through the low pressure economizer ECO_{LP}:

The total low pressure outlet steam mass flow rate of 0.88kg/s flows through this economizer. The heat transfer rate is calculated between points 4_{LP} and 5_{LP}, according to equation *(2.29).*

$$Q_{ECO}^{LP} = \dot{M}_{Steam}^{LP} \cdot \left(i_4^{LP} - i_5^{LP}\right) = 0.88 \cdot (637.28 - 222.60) = 365.82kW \tag{2.29}$$

2.7 Calculating Flue Gas Parameters at Points B, C, F, H, I, and J

Point B:

Flue gas enthalpy at point B:

$$I_B = I_A - \frac{Q_{SH2}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 678.69 - \frac{810.67}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 644\, kJ/Nm^3 \tag{2.30}$$

Flue gas temperature at point B:

The flue gas temperature at point B is determined through liner interpolation between known values from *Table 2.3*.

$$t_B = \frac{I_B - I_{Flue:400}}{I_{Flue:500} - I_{Flue:400}} \cdot (500 - 400) + 400 \tag{2.31}$$

$$t_B = \frac{644 - 547.44}{693.27 - 547.44} \cdot 100 + 400 = 466.22℃$$

Point C:

Flue gas enthalpy at point C:

$$I_C = I_B - \frac{Q_{SH1}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 644 - \frac{1314.27}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 587.77\, kJ/Nm^3 \tag{2.32}$$

Flue gas temperature at point C:

The flue gas temperature at point C is determined through liner interpolation between known values from *Table 2.3*.

$$t_C = \frac{I_C - I_{Flue:400}}{I_{Flue:500} - I_{Flue:400}} \cdot (500 - 400) + 400 \tag{2.33}$$

$$t_C = \frac{587.77 - 547.44}{693.27 - 547.44} \cdot 100 + 400 = 427.66℃$$

Point F:

Flue gas enthalpy at point F:

$$I_F = I_E - \frac{Q_{SH}^{LP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 303.77 - \frac{35.16}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 302.26\, kJ/Nm^3 \tag{2.34}$$

Flue gas temperature at point F:

The flue gas temperature at point F is determined through liner interpolation between known values from *Table 2.3*.

$$t_F = \frac{I_F - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 \tag{2.35}$$

$$t_F = \frac{302.26 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 224.83℃$$

Point H:

Flue gas enthalpy at point H:

$$I_H = I_G - \frac{Q_{ECO2}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 222.4 - \frac{158.21}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 215.63\, kJ/Nm^3 \tag{2.36}$$

Flue gas temperature at point H:

The flue gas temperature at point H is determined through liner interpolation between known values from *Table 2.3*.

$$t_H = \frac{I_H - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 \tag{2.37}$$

$$t_H = \frac{215.63 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 161.13℃$$

Point I:

Flue gas enthalpy at point I:

$$I_I = I_H - \frac{Q_{ECO}^{LP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 215.63 - \frac{365.82}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 199.98\, kJ/Nm^3 \tag{2.38}$$

Flue gas temperature at point I:

The flue gas temperature at point I is determined through liner interpolation between known values from *Table 2.3*.

$$t_I = \frac{I_I - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 \tag{2.39}$$

$$t_I = \frac{199.98 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 149.52℃$$

Point J:

Flue gas enthalpy at point J:

$$I_J = I_J - \frac{Q_{ECO1}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 149.52 - \frac{1119.61}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 152.08\, kJ/Nm^3 \tag{2.40}$$

Flue gas temperature at point J:

The flue gas temperature at point J is determined through liner interpolation between known values from *Table 2.3*.

$$t_J = \frac{I_J - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 \tag{2.41}$$

$$t_J = \frac{152.08 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 113.96℃$$

2.8 An Overview of the Calculated Values and the HRSG temperature profile

The main values calculated in this section are depicted in various tables below. The water/steam parameters are organized in *Table 2.4*, the flue gas parameters are shown in *Table 2.5*, and the heat transfer rates of each individual heat exchanger in both the high and low pressure circuits are presented in *Table 2.6*. Some of these values are used to construct the temperature profile of the HRSG. Additional values are organized in *Table 2.7*.

	Point	Temperature [°C]		Enthalpy [kJ/kg]		Pressure [MPa]	
		symbol	value	symbol	value	symbol	value
High pressure	1_{HP}	t_1^{HP}	450	i_1^{HP}	3322.65	p_1^{HP}	4.6
	2_{HP}	t_2^{HP}	348.48	i_2^{HP}	3072.65	p_2^{HP}	4.7
	3_{HP}	t_3^{HP}	408.46	i_3^{HP}	3222.46	p_3^{HP}	4.7
	4_{HP}	t_4^{HP}	261.4	i_4^{HP}	2795.83	p_4^{HP}	4.8
	5_{HP}	t_5^{HP}	261.4	i_5^{HP}	1141.81	p_5^{HP}	4.8
	6_{HP}	t_6^{HP}	256.4	i_6^{HP}	1116.99	p_6^{HP}	4.8
	7_{HP}	t_7^{HP}	151.4	i_7^{HP}	641.03	p_7^{HP}	4.9
	8_{HP}	t_8^{HP}	139.4	i_8^{HP}	589.67	p_8^{HP}	5.0
	9_{HP}	t_9^{HP}	53	i_9^{HP}	226.26	p_9^{HP}	5.1
Low pressure	1_{LP}	t_1^{LP}	170	i_1^{LP}	2792.98	p_1^{LP}	0.46
	2_{LP}	t_2^{LP}	156.15	i_2^{LP}	2753.12	p_2^{LP}	0.56
	3_{LP}	t_3^{LP}	156.15	i_3^{LP}	658.88	p_3^{LP}	0.56
	4_{LP}	t_4^{LP}	151.15	i_4^{LP}	637.28	p_4^{LP}	0.56
	5_{LP}	t_5^{LP}	53	i_5^{LP}	222.60	p_5^{LP}	0.86

Table 2.4 Water/steam parameters at various points throughout each circuit

Point	Temperature [°C]		Enthalpy [kJ/Nm^3]	
	symbol	value	symbol	value
A	t_A	490	I_A	678.69
B	t_B	466.22	I_B	644
C	t_C	427.66	I_C	587.77
D	t_D	271.4	I_D	366.5
E	t_E	225.92	I_E	303.77
F	t_F	224.83	I_F	302.26
G	t_G	166.15	I_G	222.4
H	t_H	161.13	I_H	215.63
I	t_I	149.52	I_I	199.98
J	t_J	113.96	I_J	152.08

Table 2.5 Flue gas parameters at various points throughout the boiler

	Heat exchanger	Heat transferred [kW]	
		symbol	value
High pressure	superheater SH_{2HP}	Q_{SH2}^{HP}	810.67
	superheater SH_{1HP}	Q_{SH1}^{HP}	1314.27
	evaporator EV_{HP}	Q_{EV}^{HP}	5171.76
	economizer ECO_{3HP}	Q_{ECO3}^{HP}	1466.23
	economizer ECO_{2HP}	Q_{ECO2}^{HP}	158.21
	economizer ECO_{1HP}	Q_{ECO1}^{HP}	1119.61
	High pressure circuit total	-	10040.75
Low pressure	superheater SH_{LP}	Q_{SH}^{LP}	35.16
	evaporator EV_{LP}	Q_{EV}^{LP}	1866.54
	economizer ECO_{LP}	Q_{ECO}^{LP}	365.82
	Low pressure circuit total	-	2267.52
	Overall total	-	12308.27

Table 2.6 Heat transfer rates of individual heat exchangers in the high and low pressure circuits

Parameter	Symbol	Value	Units
Flue gas density	ρ_{Flue}	1.2772	$[kg/Nm^3]$
Volumetric flow rate of the flue gas, under normal conditions	$\dot{M}_{VFlue}$	23.49	$[Nm^3/s]$
High pressure steam production rate	$\dot{M}_{Steam}^{HP}$	3.24	$[kg/s]$
Low pressure steam production rate	$\dot{M}_{Steam}^{LP}$	0.88	$[kg/s]$
Percentage heat loss due to radiation and convection	$Q_{RC\%}$	0.49	[%]
Water injection (high pressure) percentage of high pressure mass flow	$\dot{M}_{W\%}$	5	[%]

Table 2.7 Other important values

HRSG Temperature profile:

The temperature profile of the boiler is depicted below in *figure 2-3*. This temperature profile graph was constructed according to the calculated parameters in the tables above, *Table 2.4*, *Table 2.5*, and *Table 2.6*.

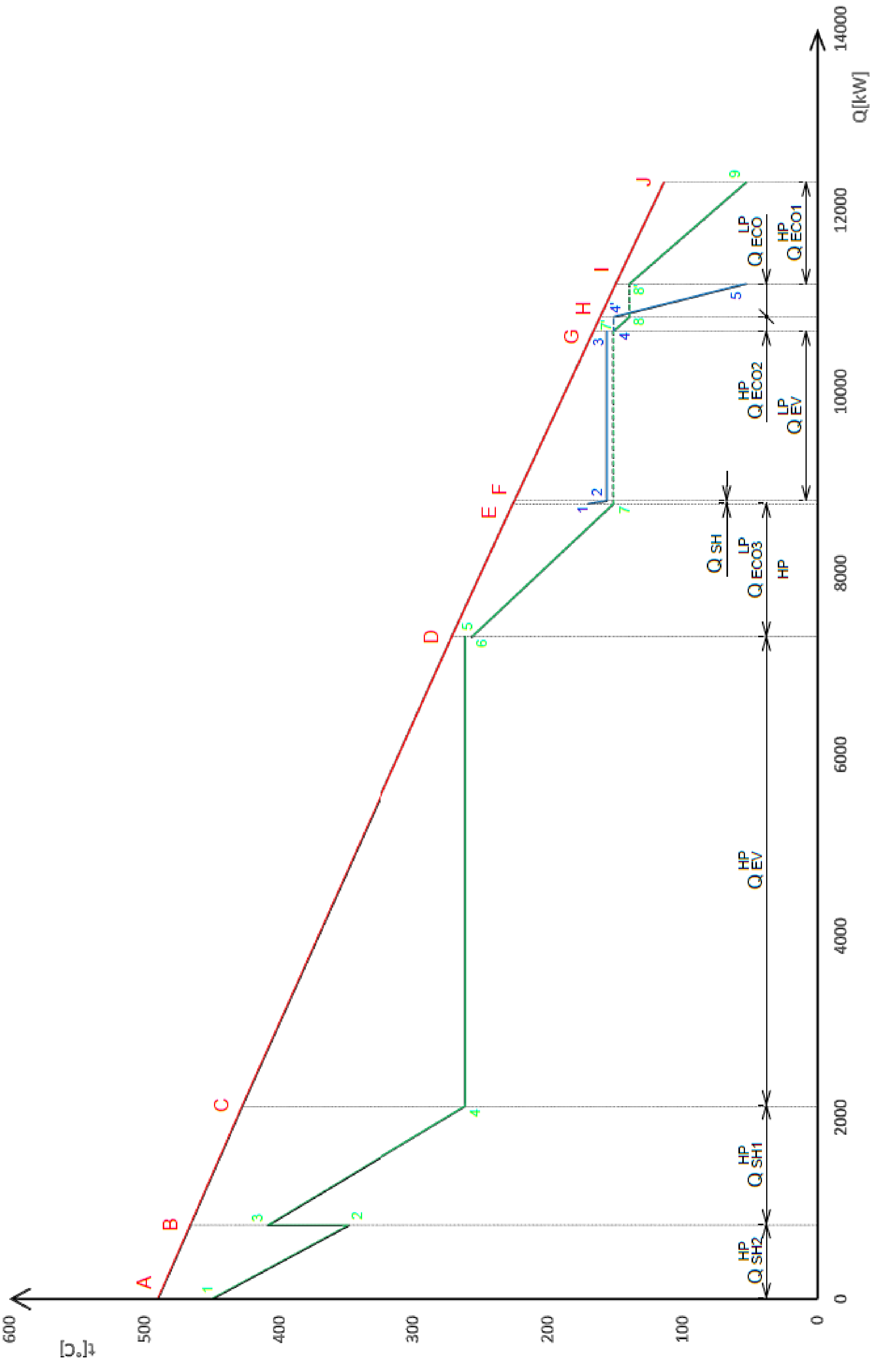

Figure 2-3 HRSG Temperature profile

3 FLUE GAS DUCT DESIGN

The flue gas duct size refers to the the dimensions of the inner casing liners of the HRSG, in which the flue gas is contained. The flue gas duct dimensions are determined according to the size of superheater SH_{2HP}, and suggested flow speeds. The calculations in this section are carried out according to source [2].

3.1 Finned Tube Geometry of the High Pressure Superheater (SH_{2HP})

The finned tubes of all the heat exchangers in the boiler are arranged in a staggered configuration, inside the flue gas duct, in order to optimize the heat transfer between the flue gas and the water/steam. The chosen dimensions of the finned tubes used in the high pressure superheater (SH_{2HP}) are shown in in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in are selected and later on adjusted in order to supply the superheater with an acceptable steam flow speed through its tubes, and these parameters are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger (in the direction of flue gas flow).

	Parameter	Symbol	Value	Units
tube	Outer tube diameter	D_{tube}^{SH2}	31.8	$[mm]$
	Tube wall thickness	th_{tube}^{SH2}	4	$[mm]$
	Inner tube diameter	d_{tube}^{SH2}	23.8	$[mm]$
fins	Fin thickness	th_{fin}^{SH2}	1	$[mm]$
	Number of fins per meter	n_{fin}^{SH2}	110	$[mm]$
	Fin spacing	s_{fin}^{SH2}	9.09	$[mm]$
	Fin height	h_{fin}^{SH2}	15	$[mm]$
	Outer fin diameter	D_{fin}^{SH2}	61.8	$[mm]$

Table 3.1 Parameters selected for the finned tubes used in the high pressure superheater (SH2HP)

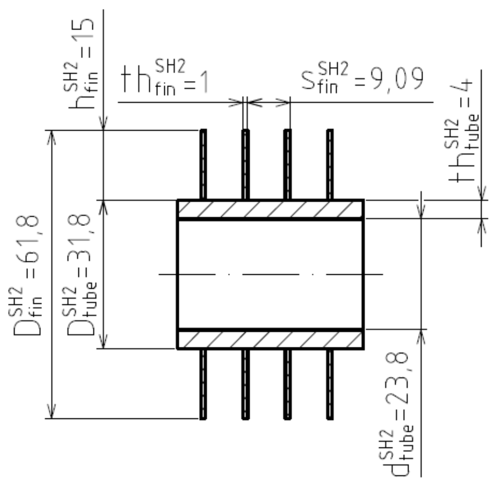

Figure 3-1 Tube geometry SH_{2HP}

The inner tube diameter and the outer fin diameter are calculated intuitively according to equations *(3.1)* and *(3.2)*. The approximate fin spacing is calculated through equation *(3.3)*.

Inner tube diameter:

$$d_{tube}^{SH2} = D_{tube}^{SH2} - 2 \cdot th_{tube}^{SH2} = 318 - 2 \cdot 4 = 23.8mm \tag{3.1}$$

Outer fin diameter:

$$D_{fin}^{SH2} = D_{tube}^{SH2} + 2 \cdot h_{fin}^{SH2} = 31.8 + 2 \cdot 15 = 61.8mm \tag{3.2}$$

Fin spacing:

$$s_{fin}^{SH2} = \frac{1000}{n_{fin}^{SH2}} = \frac{1000}{105} = 9.09mm \tag{3.3}$$

3.1.1 Determining the number of tubes in each lateral row of the high pressure superheater (SH_{2HP})

Average specific volume of steam in SH_{2HP}:

The average specific volume of the steam in SH_{2HP} is required later in the continuity equation *(3.5)*, this value is determined according to the average pressure and temperature of the steam inside SH_{2HP}.

$$\bar{t}_{SH2} = \frac{t_1^{HP} + t_2^{HP}}{2} = \frac{450 + 348.48}{2} = 399.24℃$$

$$\bar{p}_{SH2} = \frac{p_1^{HP} + p_2^{HP}}{2} = \frac{4.6 + 4.7}{2} = 4.65MPa$$

$\bar{v}_{SH2} = 0.062\, m^3/kg$ (determined using X-Steam, f(p,t))

Determining the number of tubes per lateral row required for SH_{2HP}:

The number of tubes in each lateral row of the high pressure superheater (SH_{2HP}) is calculated through equation *(3.6)*, which is a combination of two equations. These equations include intuitive cross sectional area equation *(3.4)* for all the tubes in one lateral row of the heat exchanger, according to the selected tube geometry. The second equation used is the continuity equation *(3.5)*. The speed of steam traveling through heat exchangers is suggested to be between 15 and 25 m/s, in the continuity equation *(3.5)* a preliminary value of 20 m/s is used [7]. The actual speed of the steam will be calculated later on.

$$A_{row}^{SH2} = \frac{\pi \cdot \left(d_{tube}^{SH2}\right)^2}{4} \cdot n_{tube/r}^{SH2} \tag{3.4}$$

$$A_{row}^{SH2} = \frac{\dot{M}_{Steam}^{HP} \cdot \bar{v}_{SH2}}{W_{Steam}} \tag{3.5}$$

$$n_{tube/r}^{SH2} = \frac{\dot{M}_{Steam}^{HP} \cdot \bar{v}_{SH2}}{\frac{\pi \cdot (d_{tube}^{SH2})^2}{4} \cdot W_{Steam}} = \frac{3.24 \cdot 0.062}{\frac{\pi \cdot 0.0238^2}{4} \cdot 20} = 22.76 \tag{3.6}$$

Evidently the number of tubes per lateral row must be a whole number, there for the value 22.76 is rounded to 23, the nearest whole number.

$$n_{tube/r}^{SH2} = 23$$

The actual speed of the steam in SH_{2HP} must be recalculated according to the actual number of tubes in each row.

$$W_{Steam}^{SH2} = \frac{\dot{M}_{Steam}^{HP} \cdot \bar{v}_{SH2}}{\frac{\pi \cdot (d_{tube}^{SH2})^2}{4} \cdot n_{tube/r}^{SH2}} = \frac{3.24 \cdot 0.062}{\frac{\pi \cdot 0.0238^2}{4} \cdot 23} = 19.79\, m/s \tag{3.7}$$

3.1.2 Volumetric follow rate of flue gas passing through the high pressure superheater (SH_{2HP})

The actual volumetric flow rate around superheater SH_{2HP} is calculated in equation *(3.9)*, according to the average flue gas temperature between points A and B, as seen in equation *(3.8)*.

$$\bar{t}_{(AB)} = \frac{t_A + t_B}{2} = \frac{490 + 466.22}{2} = 478.11℃ \tag{3.8}$$

$$\dot{M}_{VFlue}^{SH2} = \frac{273.15 + \bar{t}_{(AB)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 478.11}{273.15} \cdot 23.49 = 64.6\, m^3/s \tag{3.9}$$

The speed of the flue gas flowing around the tubes of the heat exchangers is suggested to be between 9 and 12 m/s, a preliminary value of 10 m/s will be used to determine the cross sectional area that the flue gas flows through [7], between the tubes of the heat exchanger and the duct walls.

$$A_{duct} = \frac{\dot{M}_{VFlue}^{SH2}}{W_{Flue}} = \frac{64.6}{10} = 6.46 m^2 \tag{3.10}$$

This value is used later on to determine the dimensions of the flue gas duct.

3.2 Flue Gas Duct Dimensions

Superheater SH_{2HP} lateral tube spacing inside the flue gas duct:

$$s_{1(SH2)} = D_{tube}^{SH2} + 2 \cdot h_{fin}^{SH2} + a_{SH2} = 0.0318 + 2 \cdot 0.015 + 0.013 = 0.0748 m \tag{3.11}$$

Where:

a_{SH2} Is the lateral gap between the finned tube of the heat exchanger (in the direction perpendicular to the flue gas flow), this value is selected for SH_{2HP} as 13 mm

Width of the flue gas duct:

$$L = 3 \cdot \frac{s_{1(SH2)}}{2} + \left(n_{tube/r}^{SH2} - 1\right) \cdot s_{1(SH2)} = 3 \cdot \frac{0.0748}{2} + (23 - 1) \cdot 0.0748 \approx 1.76 m \tag{3.12}$$

Height of the duct:

$$A_{duct}^{SH2} = H \cdot L - H \cdot D_{tube}^{SH2} \cdot n_{tube/r}^{SH2} - H \cdot 2 \cdot h_{fin}^{SH2} \cdot th_{fin}^{SH2} \cdot n_{fin}^{SH2} \cdot n_{tube/r}^{SH2} \tag{3.13}$$

$$H = \frac{A_{dust}^{SH2}}{L - D_{tube}^{SH2} \cdot n_{tube/r}^{SH2} - 2 \cdot h_{fin}^{SH2} \cdot th_{fin}^{SH2} \cdot n_{fin}^{SH2} \cdot n_{tube/r}^{SH2}}$$

$$H = \frac{6.46}{1.76 - 0.0318 \cdot 23 + 2 \cdot 0.015 \cdot 0.001 \cdot 110 \cdot 23} \approx 6.78m$$

The height of the gas duct is suggested to be 2 to 4 times the width of the duct [7]. This value is calculated below to be 3.85, which is within the suggested range.

$$\frac{H}{L} = \frac{6.76}{1.76} = 3.85 \tag{3.14}$$

The actual speed of the flue gas passing through SH_{2HP} must be recalculated according to the duct dimensions which are rounded to the nearest centimeter.

$$W_{Flue}^{SH2} = \frac{\dot{M}_{VFlue}^{SH2}}{L \cdot H - n_{tube/r}^{SH2} \cdot H \cdot \left(D_{tube}^{SH2} + 2 \cdot h_{fin}^{SH2} \cdot th_{fin}^{SH2} \cdot n_{fin}^{SH2}\right)} \tag{3.15}$$

$$W_{Flue}^{SH2} = \frac{64.6}{1.76 \cdot 6.78 - 23 \cdot 6.78 \cdot (0.0318 + 2 \cdot 0.015 \cdot 0.001 \cdot 110)} = 10\,m/s$$

3.2.1 Overview of the calculated values

An overview of the main values calculated in this section are presented in *Table 3.1*. They include values defining duct and superheater (SH_{2HP}) geometry as well as actual flow speeds.

Parameter	Symbol	Value	Units
Actual speed of steam in SH_{2HP}	W_{Steam}^{SH2}	19.79	$[m/s]$
Actual speed of flue gas around SH_{2HP}	W_{Flue}^{SH2}	10	$[m/s]$
Number of tubes in one lateral row of SH_{2HP}	$n_{tube/r}^{SH2}$	23	$[-]$
Width of the flue gas duct	L	1.76	$[m]$
Height of the flue gas duct	H	6.78	$[m]$

Table 3.1 Calculated values defining duct and superheater (SH_{2HP}) geometry, as well as actual flow speeds.

4 DESIGN AND SIZING OF HEAT TRANSFER SURFACES

The individual heat exchangers of the high and low pressure circuits in the boiler must be designed to transfer heat at the intended rate from the flue gas to the water/steam while respecting the already determined flue gas duct dimensions, and insuring that several parameters stay within their suggested ranges. All of the heat exchangers in the boiler will be tube type exchanger, made up of lateral row of tubes, in a staggered configuration to optimize heat transfer. The calculation procedure is very similar for most of the heat exchangers, there are slight differences depending on the purpose of the heat exchanger (superheater, evaporator, and economizer). There is a significant difference in the calculations used for a heat exchanger with smooth tubes rather than finned tubes. Differences in the calculation procedure or calculations themselves are described, where applicable, throughout the design process.

The first step is defining the geometry of the finned tubes used in the heat exchanger. The outer tube diameter D_{tube} and the wall thickness th_{tube} are chosen from the standard manufactured sizes, the inner tube diameter d_{tube} is then calculated intuitively. The height of the tube fins h_{fin} is chosen according to a suggested ranges, depending on the purpose of the particular heat exchanger. For superheaters and economizers the fin height is suggested to be within the range of 8-15mm, for evaporators the suggested range is 10-19mm. The outer diameter of the fins D_{fin} can then be intuitively calculated. The fin thickness th_{fin} is chosen to be 1mm for all of the heat exchangers in the boiler. The number of fins per meter n_{fin} is chosen to be within the suggested range of 100 to 250, the fin spacing s_{fin} is then calculated as the inverse of the number of fins per meter.

After the geometry of the finned tubes is defined, their general layout in the heat exchanger must be determined. The lateral gap between the tubes a is chosen, subsequently the lateral tube spacing s_1 can be calculated. From the duct width and the lateral tube spacing the number of tubes in one lateral row of the heat exchanger can be determined. According to the number of tubes in one lateral row, the speed of the water/steam W_{Steam} flowing through the tubes is calculated and verified to be within an acceptable range. After calculating the area between the heat exchanger tubes and the duct walls A_{duct} , which is the cross-sectional area the flue gas must flow through, it is possible to verify that the speed of the flue gas w_{Flue} flowing around the tubes of the heat exchanger is within an acceptable range. The longitudinal tube spacing (the spacing in the direction of the flue gas flow), or the spacing between lateral rows of tubes making up the heat exchanger s_2 , is chosen according to the purpose of the heat exchanger. For superheaters and evaporators the longitudinal tube spacing chosen to be 117mm, and for evaporators a value of 90mm is chosen.

To completely define the heat exchanger geometry it is necessary to determine the number of lateral rows of tubes n_{row} in the heat exchanger. This is determined as a quotient, when the total outer surface area needed for to meet the required heat transfer rate, or theoretical heat transfer rate of the heat exchanger S_{out} is divided by the outer surface area of one lateral row of tubes in the heat exchanger $S_{out/r}$. Naturally this value must be rounded to the nearest whole number, and thus there is a discrepancy between required heat transfer rate Q and the actual heat transfer rate of the heat exchanger Q^{real} . This discrepancy, or percent error between the theoretical and actual values, must be less than 2% for the heat exchanger configuration to be acceptable. In order to achieve this requirement changes may be made to the geometry of the finned tubes, or the number of tubes in each lateral row or the lateral tube spacing respectively. The actual heat transfer rate of each heat exchanger is calculated according to the heat transfer equation. The amount heat transfer rate is dependent on several values, including: the heat transfer coefficient, the outer surface area of the finned tubes of the heat exchanger S_{out} , and the logarithmic mean temperature difference Δt_{ln} . These values are dependent on the geometry of the heat exchanger, as well as the parameters of the water/steam and flue gas.

The calculations described in the paragraphs above must be completed for each heat exchanger consecutively, in the order in which they are arranged in the boiler, from the first heat exchanger to come in contact with the flue gas to the last. After the actual heat transfer rate of the heat exchanger is determined, the parameters of the flue gas leaving the heat exchanger (temperature, and enthalpy) must be recalculated and used as the inlet flue gas parameters for the subsequent heat exchanger.

Equations used in the design and sizing of heat exchangers:

The equations below are presented in an order that is clear and straight forward. This, however is not the chronological order in which the equations are calculated. Several values used in the heat transfer calculations are the same for all finned tube heat exchangers in the boiler, these values are presented in *Table 4.1*.

Collective values	Symbol	Value	Units
Width of the flue gas duct	L	1.76	$[m]$
Height of the flue gas duct	H	6.78	$[m]$
Flue gas volumetric flow rate under normal conditions	$\dot{M}_{VFlue}$	23.49	$[Nm^3/s]$
Percentage of useable thermal power lost through radiation and convection	$Q_{RC\%}$	0.49	[%]
Coefficient characterizing fin taper	μ	1	$[-]$
Coefficient characterizing the uneven distribution of the convective heat transfer coefficient (α_c) along the surface of the fins	Ψ_{fin}	0.85	$[-]$
Correction coefficient for fin fouling	ε	0.0043	$[m^2K/W]$
Thermal conductivity of the fins	λ_{fin}	38	$[W/(m \cdot K)]$
correction coefficient dependent on the temperature of the water/steam and tube wall temperature	c_t	1	$[-]$
correction coefficient dependent on the tube length relative to the tube diameter	c_l	1	$[-]$
correction coefficient for fluid flow between concentric tubes	c_m	1	$[-]$

Table 4.1 Collective values used in the heat transfer calculations

The number of lateral rows of tubers in the heat exchanger:

$$n_{row} = \frac{S_{out}}{S_{out/r}} \tag{4.1}$$

Where:

S_{out}	is the total outer surface area of all the tubes in the heat exchanger, or the total surface area of the heat exchanger that transfers heat from and is in contact with the flue gas [m^2]
$S_{out/r}$	is the outer surface area of all the tubes in one lateral row of the heat exchanger [m^2]

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r} = S_{out/1m} \cdot H \cdot n_{tube/r} \tag{4.2}$$

Where:

$S_{out/1m}$	is the outer surface area of one finned tube per meter length [m]

The outer surface area of one finned tube per meter length:

$$S_{out/1m} = \pi \cdot D_{tube} \cdot \left(1 - n_{fin} \cdot th_{fin}\right) + n_{fin} \cdot S_{1fin} \tag{4.3}$$

Where:

S_{1fin}	is the outer surface area of one fin [m^2]

The outer surface area of one fin:

$$S_{1fin} = 2 \cdot \pi \cdot \frac{\left(D_{fin}\right)^2 - (D_{tube})^2}{4} + \pi \cdot D_{fin} \cdot th_{fin} \tag{4.4}$$

The total outer surface area of all the tubes in the heat exchanger:

The heat transfer equation *(4.5)* is rearranged to solve for the total outer surface area of all the tubes in the heat exchanger instead of the total amount of heat transferred by the heat exchanger.

$$Q = k \cdot S_{out} \cdot \Delta t_{ln} \tag{4.5}$$

Where:

Q	is the heat transfer rate of the heat exchanger [W]
k	is the overall heat transfer coefficient [W/(m^{2}K)]
Δt_{ln}	is the logarithmic mean temperature difference [K]

The logarithmic mean temperature difference:

$$\Delta t_{ln} = \frac{\Delta t_2 - \Delta t_1}{ln\left(\frac{\Delta t_2}{\Delta t_1}\right)} \tag{4.6}$$

Where:

Δt_1 temperature difference between the flue gas entering and the water/steam exiting the heat exchanger [K]

Δt_2 temperature difference between the flue gas exiting and the water/steam entering the heat exchanger [K]

The overall heat transfer coefficient:

$$k = \frac{1}{\frac{1}{\alpha_{r:out}} + \frac{1}{\alpha_{r:in}} \cdot \frac{S_{out/1m}}{S_{in/1m}}} \tag{4.7}$$

Where:

$\alpha_{r:out}$ is the reduced heat transfer coefficient outside the tubes, between the flue gas and the finned heat exchanger tubes [W/(m^{2}K)]

$\alpha_{r:in}$ is the reduced heat transfer coefficient inside the tubes, between the finned heat exchanger tubes and the water/steam inside them [W/(m^{2}K)]

$S_{in/1m}$ is the inner surface area of one finned tube per meter length, or the area that is transferring heat to and in contact with the water/steam flowing through the tube per meter length [m]

The inner surface area of one finned tube per meter length:

$$S_{in/1m} = \pi \cdot d_{tube} \tag{4.8}$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out} = \left(\frac{S_{fin}}{S_{out}} \cdot E \cdot \mu + \frac{S_{out-fin}}{S_{out}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c} \tag{4.9}$$

Where:

E is a coefficient characterizing the effectiveness of the fins, this value is determined from a graph according to the product of $\beta \cdot h_{fin}$ and the quotient of D_{fin}/D_{tube}, where β is a coefficient [-]

μ is a coefficient characterizing fin taper [-]

Ψ_{fin} is a coefficient characterizing the uneven distribution of the convective heat transfer coefficient (α_c) along surface of the fins [-]

α_c The coefficient of heat transfer through convection outside of the heat exchanger tubes [W/(m^{2}K)]

ε	is a correction coefficient for fin fouling [m^{2}K/W]
$\frac{S_{fin}}{S_{out}}$	is the fraction of the total outer surface area of the finned tubes that is made up of the fins themselves [-]
$\frac{S_{out-fin}}{S_{out}}$	is the fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls [-]

Coefficient β:

$$\beta = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c}{th_{fin} \cdot \lambda_{fin} \cdot (1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c)}} \tag{4.10}$$

Where:

β	Coefficient used to calculate the coefficient characterizing the effectiveness of the fins [1/m]
λ_{fin}	is the thermal conductivity of the fins [W/(m·K)]

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}}{S_{out}} = \frac{\left(\frac{D_{fin}}{D_{tube}}\right)^2 - 1}{\left(\frac{D_{fin}}{D_{tube}}\right)^2 - 1 + 2 \cdot \left(\frac{s_{fin}}{D_{tube}} - \frac{th_{fin}}{D_{tube}}\right)} \tag{4.11}$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}}{S_{out}} = 1 - \frac{S_{fin}}{S_{out}} \tag{4.12}$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c = 0.23 \cdot c_z \cdot (\varphi_\sigma)^{0.2} \cdot \frac{\lambda_{Flue}}{s_{fin}} \cdot \left(\frac{D_{tube}}{s_{fin}}\right)^{-0.54} \cdot \left(\frac{h_{fin}}{s_{fin}}\right)^{-0.14} \cdot \left(\frac{W_{Flue} \cdot s_{fin}}{\nu_{Flue}}\right)^{0.65} \tag{4.13}$$

Where:

c_z	is the correction coefficient for the number of lateral row of tubes in the heat exchanger [-]
φ_σ	is a coefficient characterizing the relative tube spacing [-]
λ_{Flue}	is the thermal conductivity of the flue gas [W/(m·K)]
ν_{Flue}	is the kinematic viscosity of the flue gas [m^2/s]

Coefficient of the relative tube spacing:

$$\varphi_\sigma = \frac{\sigma_1 - 1}{\sigma_2' - 1} \tag{4.14}$$

Where:

σ_1 is the lateral tube spacing relative to the outer tube diameter [-]

σ_2' is the diagonal tube spacing relative to the outer tube diameter [-]

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1 = \frac{s_1}{D_{tube}} \tag{4.15}$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2' = \frac{s'}{D_{tube}} = \frac{\sqrt{\left(\frac{s_1}{2}\right)^2 + {s_2}^2}}{D_{tube}} \tag{4.16}$$

Where:

s' is the diagonal tube spacing [m]

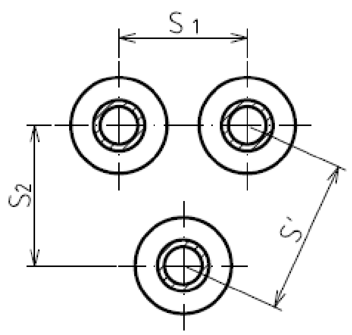

Figure 4-1 Tube spacing

The reduced heat transfer coefficient inside the tubes:

$$\alpha_{r:in} = 0.023 \cdot \frac{\lambda_{Steam}}{d_e} \cdot \left(\frac{W_{Steam} \cdot d_e}{\nu_{Steam}}\right)^{0.8} \cdot (Pr_{Steam})^{0.4} \cdot c_t \cdot c_l \cdot c_m \tag{4.17}$$

Where:

λ_{Steam} is the thermal conductivity of the water/steam [W/(m·K)]

d_e is the equivalent diameter, this value is equal to the inner tube diameter [m]

ν_{Steam} is the kinematic viscosity of the water/steam [m^2/s]

Pr_{Steam} is the Prandtl number of the water/steam [-]

c_t is the correction coefficient dependent on the temperature of the water/steam and tube wall temperature [-]

c_l is the correction coefficient dependent on the tube length relative to the tube diameter [-]

c_m is the correction coefficient for fluid flow between concentric tubes [-]

Flue gas parameters:

The flue gas parameters needed to complete the calculation of equation *(4.13)* are determined according to the moisture concentration contained in the flue gas (7.8%), and the average temperature of the flue gas passing through each individual heat exchanger. The thermal conductivity and the kinematic viscosity of the flue gas are determined from *Table 4.2* and *Table 4.3* respectively. Linear interpolation is used to interpolate between values obtained from source [2].

$\bar{t}_{Flue}$ [°C]	Moisture concentration X_{H2O} [%]		
	5	7.8	10
0	22.5	22.61	22.7
100	30.5	30.89	31.2
200	38.5	39.23	39.8
300	46.4	47.35	48.1
400	54.3	55.59	56.6
500	62.6	63.82	65.1

Table 4.2 Flue gas coefficient of thermal conductivity, $\lambda_{Flue} \cdot 10^3$ [W/(m·K)]

$\bar{t}_{Flue}$ [°C]	Moisture concentration X_{H2O} [%]		
	5	7.8	10
0	12.2	12.2	12.2
100	21.3	21.41	21.5
200	31.8	32.36	32.8
300	45	45.45	45.8
400	59.2	59.87	60.4
500	74.6	75.55	76.3

Table 4.3 Flue gas coefficient of kinematic viscosity, $\nu_{Flue} \cdot 10^6$ [m²/s]

Water/steam parameters:

The water/steam parameters needed to complete the calculation of equation *(4.17)* are determined according to the average temperature and pressure of the water/steam passing through each individual heat exchanger. The thermal conductivity and the Prandtl number of the water/steam are determined using X-Steam. The dynamic viscosity is also determined in this manner and is then used to calculate the kinematic viscosity of the water/steam.

Kinematic viscosity of the steam:

$$\nu_{Steam} = \frac{\mu_{Steam}}{\rho_{Steam}} \tag{4.18}$$

Where:

μ_{Steam} is dynamic viscosity of the water/steam passing through the heat exchanger [Pa·s]

ρ_{Steam} is the density of the water/steam passing through the heat exchanger [kg/m^3], determined using X-Steam according to average pressure and temperature

4.1 Design of the High Pressure Superheater SH_{2HP}

Several parameters are required in the calculations associated with the design of the high pressure superheater SH_{2HP}. Parameters, other than those describing the geometry of the superheater, that are necessary for the calculations regarding the design and sizing of superheater SH_{2HP}, some of which were determined in previous calculations, are organized in *Table 4.4* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of SH_{2HP} (theoretical value)	Q_{SH2}^{HP}	810.67	$[kW]$
Flue gas	Actual speed of flue gas around SH_{2HP}	W_{Flue}^{SH2}	10	$[m/s]$
Flue gas	Temperature of flue gas entering into SH_{2HP}	t_A	490	[℃]
Flue gas	Enthalpy of flue gas entering into SH_{2HP}	I_A	678.69	$[kJ/Nm^3]$
Flue gas	Temperature of flue gas exiting SH_{2HP} (projected value)	t_B	466.22	[℃]
Steam	Mass flow rate of steam through SH_{2HP}	$\dot{M}_{Steam}^{HP}$	3.24	$[kg/s]$
Steam	Actual speed of steam inside SH_{2HP}	W_{Steam}^{SH2}	19.79	$[m/s]$
Steam	Temperature of steam entering SH_{2HP}	t_2^{HP}	348.48	[℃]
Steam	Pressure of steam entering SH_{2HP}	p_2^{HP}	4.7	$[MPa]$
Steam	Temperature of steam exiting SH_{2HP}	t_1^{HP}	450	[℃]
Steam	Pressure of steam exiting SH_{2HP}	p_1^{HP}	4.6	$[MPa]$

Table 4.4 Parameters necessary for superheater SH_{2HP} design, not including parameters describing the geometry of SH_{2HP}

4.1.1 Geometry of the high pressure superheater SH_{2HP}

The geometry of the finned tubes, and the majority of the superheater layout itself have been determined earlier, as these values were required in previous calculations regarding the flue gas duct dimensions. Specific dimensions of the finned tubes used in superheater SH_{2HP} are shown in , and the values describing the heat exchanger layout are presented below in *Table 4.5*. Both and *Table 4.5* contain values in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units).

Dimension/parameter	Symbol	Value	Units
Lateral tube spacing of SH_{2HP}	$s_{1(SH2)}$	74.8	$[mm]$
Longitudinal tube spacing of SH_{2HP}	$s_{2(SH2)}$	117	$[mm]$
Number of tubes in each lateral row	$n_{tube/r}^{SH2}$	23	$[-]$

Table 4.5 Values describing the geometry of superheater SH_{2HP}

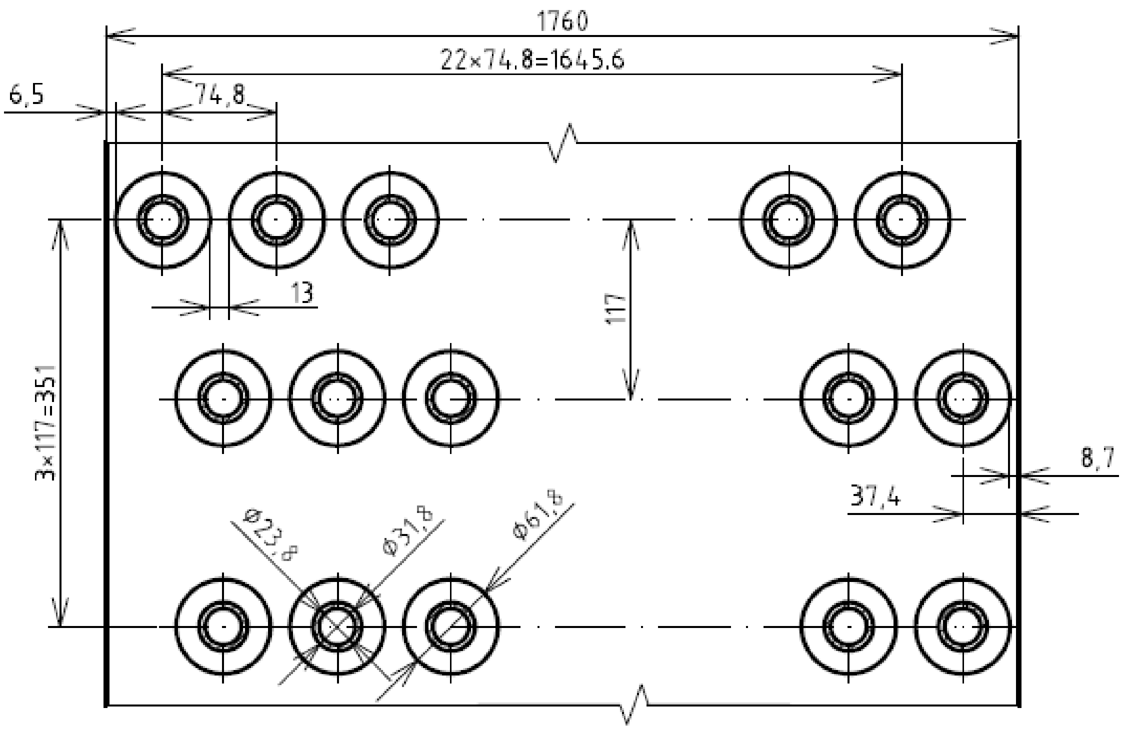

Figure 4-2 Tube layout SH_{2HP}

4.1.2 Reduced heat transfer coefficient outside the tubes of SH_{2HP}, through the flue gas

The coefficient of heat transfer through convection outside the finned tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{SH2} = \frac{s_{1(SH2)}}{D_{tube}^{SH2}} = \frac{0.0748}{0.0318} = 2.35$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2{'}^{SH2} = \frac{s'^{SH2}}{D_{tube}^{SH2}} = \frac{\sqrt{\left(\frac{s_{1(SH2)}}{2}\right)^2 + \left(s_{2(SH2)}\right)^2}}{D_{tube}^{SH2}} = \frac{\sqrt{\left(\frac{0.0748}{2}\right)^2 + 0.117^2}}{0.0318} = 3.86$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_\sigma^{SH2} = \frac{\sigma_1^{SH2} - 1}{\sigma_2{'}^{SH2} - 1} = \frac{2.35 - 1}{3.86 - 1} = 0.47$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through SH_{2HP} has been determined in previous calculations regarding the flue gas duct design.

Temperature $\bar{t}_{(AB)} = 478.11℃$

$$\lambda_{Flue}^{SH2} = \left[\frac{478.11 - 400}{100} \cdot (63.82 - 55.59) + 55.59\right] \cdot 10^{-3} = 6.202 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{SH2} = \left[\frac{478.11 - 400}{100} \cdot (75.55 - 59.87) + 59.87\right] \cdot 10^{-6} = 7.212 \cdot 10^{-5}\ m^2/s$$

The correction coefficient for the number of lateral row of tubes in the heat exchanger:

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 4 lateral rows of tubes in superheater SH_{2HP}.

$$c_z^{SH2} = 0.94$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c^{SH2} = 0.23 \cdot c_z^{SH2} \cdot (\varphi_\sigma^{SH2})^{0.2} \cdot \frac{\lambda_{Flue}^{SH2}}{s_{fin}^{SH2}} \cdot \left(\frac{D_{tube}^{SH2}}{s_{fin}^{SH2}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{SH2}}{s_{fin}^{SH2}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{SH2} \cdot s_{fin}^{SH2}}{\nu_{Flue}^{SH2}}\right)^{0.65}$$

$$\alpha_c^{SH2} = 0.23 \cdot 0.94 \cdot 0.47^{0.2} \cdot \frac{0.06202}{0.00909} \cdot \left(\frac{0.0318}{0.00909}\right)^{-0.54} \cdot \left(\frac{0.015}{0.00909}\right)^{-0.14} \cdot \left(\frac{10 \cdot 0.00909}{7.212 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{SH2} = 62.36\ W/(m^2K)$$

Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of $\beta^{SH2} \cdot h_{fin}^{SH2}$ and the quotient of $D_{fin}^{SH2}/D_{tube}^{SH2}$, where β^{SH2} is a coefficient.

Coefficient β^{SH2}:

$$\beta^{SH2} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{SH2}}{th_{fin}^{SH2} \cdot \lambda_{fin} \cdot \left(1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{SH2}\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 62.36}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 62.36)}}$$

$$\beta^{SH2} = 47.67\, m^{-1}$$

Values needed to determine the coefficient characterizing the effectiveness of the fins:

$$\beta^{SH2} \cdot h_{fin}^{SH2} = 47.67 \cdot 0.015 = 0.71$$

$$\frac{D_{fin}^{SH2}}{D_{tube}^{SH2}} = \frac{0.0618}{0.0318} = 1.94$$

Coefficient determined using the graph:

$$E^{SH2} = 0.81$$

Proportions of the tubes surface area made up of the fins, and tube wall:

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}^{SH2}}{S_{out}^{SH2}} = \frac{\left(\frac{D_{fin}^{SH2}}{D_{tube}^{SH2}}\right)^2 - 1}{\left(\frac{D_{fin}^{SH2}}{D_{tube}^{SH2}}\right)^2 - 1 + 2 \cdot \left(\frac{s_{fin}^{SH2}}{D_{tube}^{SH2}} - \frac{th_{fin}^{SH2}}{D_{tube}^{SH2}}\right)} = \frac{\left(\frac{0.0618}{0.0318}\right)^2 - 1}{\left(\frac{0.0618}{0.0318}\right)^2 - 1 + 2 \cdot \left(\frac{0.00909}{0.0318} - \frac{0.001}{0.0318}\right)}$$

$$\frac{S_{fin}^{SH2}}{S_{out}^{SH2}} = 0.85$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}^{SH2}}{S_{out}^{SH2}} = 1 - \frac{S_{fin}^{SH2}}{S_{out}^{SH2}} = 1 - 0.85 = 0.15$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out}^{SH2} = \left(\frac{S_{fin}^{SH2}}{S_{out}^{SH2}} \cdot E^{SH2} \cdot \mu + \frac{S_{out-fin}^{SH2}}{S_{out}^{SH2}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{SH2}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{SH2}}$$

$$\alpha_{r:out}^{SH2} = (0.85 \cdot 0.81 \cdot 1 + 0.15) \cdot \frac{0.85 \cdot 62.36}{1 + 0.0043 \cdot 0.85 \cdot 62.36} = 36.38\, W/(m^2 K)$$

4.1.3 Reduced heat transfer coefficient inside the tubes of SH_{2HP}, through the Steam

Steam parameters:

The kinematic viscosity, thermal conductivity, and Prandtl number of the steam passing through SH_{2HP} are determined through X-Steam according to the average temperature and pressure of the steam. The kinematic viscosity is calculated from the dynamic viscosity and the steam density. The average temperature and pressure of the steam have been calculated earlier in the flue gas duct design process.

Temperature $\bar{t}_{SH2} = 399.24\ ℃$

Pressure $\bar{p}_{SH2} = 4.65\ \mathrm{MPa}$

$$\rho_{Steam}^{SH2} = 16.0138\ kg/m^3$$

$$\mu_{Steam}^{SH2} = 2.4336 \cdot 10^{-5}\ Pa \cdot s$$

$$\lambda_{Steam}^{SH2} = 0.059\ W/(m \cdot K)$$

$$Pr_{Steam}^{SH2} = 1.0004$$

Kinematic viscosity of the steam:

$$\nu_{Steam}^{SH2} = \frac{\mu_{Steam}^{SH2}}{\rho_{Steam}^{SH2}} = \frac{2.4336 \cdot 10^{-5}}{16.0138} = 1.5197 \cdot 10^{-6}\ m^2/s$$

Reduced heat transfer coefficient inside the tubes of SH_{2HP}:

$$\alpha_{r:in}^{SH2} = 0.023 \cdot \frac{\lambda_{Steam}^{SH2}}{d_{tube}^{SH2}} \cdot \left(\frac{W_{Steam}^{SH2} \cdot d_{tube}^{SH2}}{\nu_{Steam}^{SH2}}\right)^{0.8} \cdot (Pr_{Steam}^{SH2})^{0.4} \cdot c_t \cdot c_l \cdot c_m$$

$$\alpha_{r:in}^{SH2} = 0.023 \cdot \frac{0.059}{0.0238} \cdot \left(\frac{19.79 \cdot 0.0238}{1.5197 \cdot 10^{-6}}\right)^{0.8} \cdot 1.0004^{0.4} \cdot 1 \cdot 1 \cdot 1 = 1410.24\ W/(m^2K)$$

4.1.4 Overall heat transfer coefficient for SH_{2HP}

The outer surface area of one fin:

$$S_{1fin}^{SH2} = 2 \cdot \pi \cdot \frac{\left(D_{fin}^{SH2}\right)^2 - \left(D_{tube}^{SH2}\right)^2}{4} + \pi \cdot D_{fin}^{SH2} \cdot th_{fin}^{SH2}$$

$$S_{1fin}^{SH2} = 2 \cdot \pi \cdot \frac{0.0618^2 - 0.0318^2}{4} + \pi \cdot 0.0618 \cdot 0.001 = 0.0046\ m^2$$

The outer surface area of one finned tube per meter length:

$$S_{out/1m}^{SH2} = \pi \cdot D_{tube}^{SH2} \cdot \left(1 - n_{fin}^{SH2} \cdot th_{fin}^{SH2}\right) + n_{fin}^{SH2} \cdot S_{1fin}^{SH2}$$

$$S_{out/1m}^{SH2} = \pi \cdot 0.0318 \cdot (1 - 110 \cdot 0.001) + 110 \cdot 0.0046 = 0.5955\,m$$

The inner surface area of one finned tube per meter length:

$$S_{in/1m}^{SH2} = \pi \cdot d_{tube}^{SH2} = \pi \cdot 0.0238 = 0.0748\,m$$

The overall heat transfer coefficient:

$$k^{SH2} = \frac{1}{\frac{1}{\alpha_{r:out}^{SH2}} + \frac{1}{\alpha_{r:in}^{SH2}} \cdot \frac{S_{out/1m}^{SH2}}{S_{in/1m}^{SH2}}} = \frac{1}{\frac{1}{36.38} + \frac{1}{1410.24} \cdot \frac{0.5955}{0.0748}} = 30.18\ W/(m^2K)$$

4.1.5 Logarithmic mean temperature difference across SH_{2HP}

$$\Delta t_1^{SH2} = t_A - t_1^{HP} = 490 - 450 = 40\,K$$

$$\Delta t_2^{SH2} = t_B - t_2^{HP} = 466.22 - 348.48 = 117.74\,K$$

$$\Delta t_{ln}^{SH2} = \frac{\Delta t_2^{SH2} - \Delta t_1^{SH2}}{ln\left(\frac{\Delta t_2^{SH2}}{\Delta t_1^{SH2}}\right)} = \frac{117.74 - 40}{ln\left(\frac{117.74}{40}\right)} = 72.01\,K$$

4.1.6 The number of lateral rows of tubes required for SH_{2HP}

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{SH2} = \frac{Q_{SH2}^{HP}}{k^{SH2} \cdot \Delta t_{ln}^{SH2}} = \frac{810670}{30.18 \cdot 72.01} = 373.01\,m^2$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{SH2} = S_{out/1m}^{SH2} \cdot H \cdot n_{tube/r}^{SH2} = 0.5955 \cdot 6.78 \cdot 23 = 92.86\,m^2$$

The number of lateral rows of tubers in SH_{2HP}:

$$n_{row}^{SH2} = \frac{S_{out}^{SH2}}{S_{out/r}^{SH2}} = \frac{373.01}{92.86} = 4.02$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{SH2} = 4$$

4.1.7 Calculating the actual heat transfer rate of SH_{2HP}

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{SH2:real} = S_{out/r}^{SH2} \cdot n_{row}^{SH2} = 92.86 \cdot 4 = 371.42\ m^2$$

The actual heat transfer rate of SH_{2HP}:

$$Q_{SH2}^{HP:real} = k^{SH2} \cdot S_{out}^{SH2:real} \cdot \Delta t_{ln}^{SH2} = 30.18 \cdot 371.42 \cdot 72.01 = 807.23\ kW$$

Error verification:

$$\%Error^{SH2} = \left|\left(\frac{807.23}{810.67} - 1\right) \cdot 100\right| = 0.43\%$$

The discrepancy between the theoretical and the actual heat transfer rate of superheater SH_{2HP} is less than 2%, thus the current configuration of SH_{2HP} is acceptable.

4.1.8 Determining the actual parameters of the flue gas exiting SH_{2HP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of superheater SH_{2HP} the parameters of the flue gas exiting SH_{2HP} at point B must be recalculated.

Actual enthalpy of the flue gas at point B:

$$I_B^{real} = I_A - \frac{Q_{SH2}^{HP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 678.69 - \frac{807.23}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 644.15\ kJ/Nm^3$$

Actual temperature of the flue gas at point B:

$$t_B^{real} = \frac{I_B^{real} - I_{Flue:400}}{I_{Flue:500} - I_{Flue:400}} \cdot (500 - 400) + 400 = \frac{644.15 - 547.44}{693.27 - 547.44} \cdot 100 + 400 = 466.32℃$$

4.2 Design of the High Pressure Superheater SH_{1HP}

Several parameters are required in the calculations associated with the design of the high pressure superheater SH_{1HP}. Parameters, other than those describing the geometry of the superheater, that are necessary for the calculations regarding the design and sizing of superheater SH_{1HP}, some of which were determined in previous calculations, are organized in *Table 4.6* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of SH_{1HP} (theoretical value)	Q_{SH1}^{HP}	1314.27	$[kW]$
Flue	Actual temperature of flue gas entering into SH_{1HP}	t_B^{real}	466.32	[°C]
	Actual enthalpy of flue gas entering into SH_{1HP}	I_B^{real}	644.15	$[kJ/Nm^3]$
Steam	Mass flow rate of steam in the high pressure circuit (only 95% passes through SH_{1HP})	$\dot{M}_{Steam}^{HP}$	3.24	$[kg/s]$
	Temperature of steam entering SH_{1HP}	t_4^{HP}	261.4	[°C]
	Pressure of steam entering SH_{1HP}	p_4^{HP}	4.8	$[MPa]$
	Temperature of steam exiting SH_{1HP}	t_3^{HP}	408.46	[°C]
	Pressure of steam exiting SH_{1HP}	p_3^{HP}	4.7	$[MPa]$

Table 4.6 Parameters necessary for superheater SH_{1HP} design, not including parameters describing the geometry of SH_{1HP}

4.2.1 Geometry of the high pressure superheater SH_{1HP}

The chosen dimensions of the finned tubes used in the high pressure superheater (SH_{1HP}) are shown in *Table 4.7* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.7* are selected and later on adjusted in order to supply the superheater with an acceptable steam flow rate through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

	Parameter	Symbol	Value	Units
tube	Outer tube diameter	D_{tube}^{SH1}	31.8	$[mm]$
	Tube wall thickness	th_{tube}^{SH1}	4	$[mm]$
	Inner tube diameter	d_{tube}^{SH1}	23.8	$[mm]$
fins	Fin thickness	th_{fin}^{SH1}	1	$[mm]$
	Number of fins per meter	n_{fin}^{SH1}	130	$[mm]$
	Fin spacing	s_{fin}^{SH1}	7.69	$[mm]$
	Fin height	h_{fin}^{SH1}	15	$[mm]$
	Outer fin diameter	D_{fin}^{SH1}	61.8	$[mm]$

Table 4.7 Parameters selected for the finned tubes used in the high pressure superheater SH_{1HP}

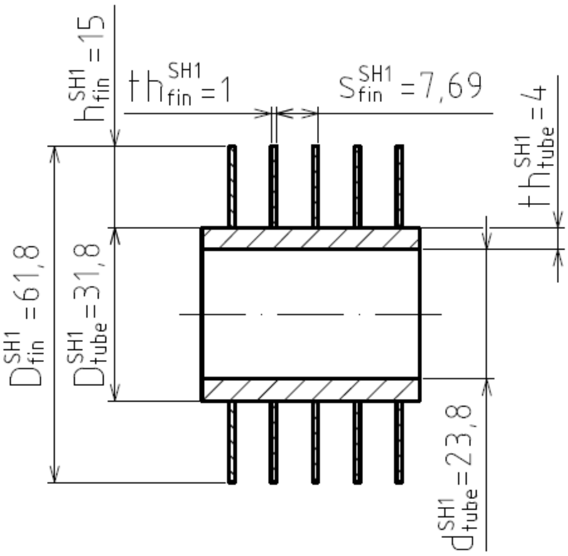

Figure 4-3 Tube geometry SH_{1HP}

The dimensions describing the layout of the finned tubes of superheater SH_{1HP} are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the superheater.

Chosen layout dimensions:

Lateral gap between tubes $a_{SH1} = 13$ mm

Longitudinal tube spacing $s_{2(SH1)} = 117$ mm

Lateral tube spacing in superheater SH_{1HP}:

$$s_{1(SH1)} = D_{tube}^{SH1} + 2 \cdot h_{fin}^{SH1} + a_{SH1} = 0.0318 + 2 \cdot 0.015 + 0.013 = 0.0748m$$

Number of tubes in each lateral row of tubes in SH_{1HP}:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{SH1} = \frac{L}{s_{1(SH1)}} - 0.5 = \frac{1.76}{0.0748} - 0.5 = 23.03$$

$$n_{tube/r}^{SH1} = 23$$

Gap verification:

There needs to be a gap between the wall of the flue gas duct and outer tubes of the heat exchanger, however it must not be excessive. This value should be no greater than 10 mm. The calculated gap is within the acceptable range.

$$gap = L - n_{tube/r}^{SH1} \cdot s_{1(SH1)} - \frac{D_{tube}^{SH1}}{2} - h_{fin}^{SH1}$$

$$gap = 1.76 - 23 \cdot 0.0748 - \frac{0.0318}{2} - 0.015 = 0.0087\ m$$

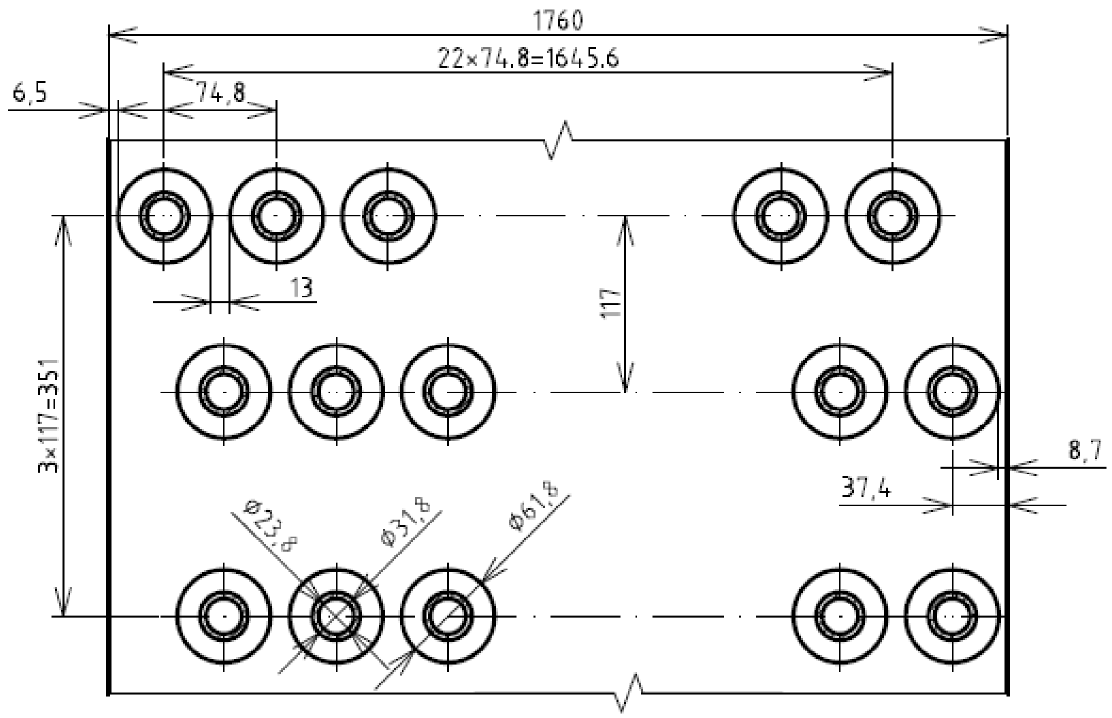

Figure 4-4 Tube layout SH_{1HP}

4.2.2 Verification of the speed of steam passing through superheater SH_{1HP}

The speed of the steam passing through superheater SH_{1HP} is calculated according to average specific volume of the steam. The specific volume of the steam is determined through X-Steam according to the average pressure and temperature of the steam passing through SH_{1HP}.

$$\bar{t}_{SH1} = \frac{t_3^{HP} + t_4^{HP}}{2} = \frac{408.46 + 261.4}{2} = 334.93\ ℃$$

$$\bar{p}_{SH1} = \frac{p_3^{HP} + p_4^{HP}}{2} = \frac{4.7 + 4.8}{2} = 4.75\ MPa$$

$\bar{v}_{SH1} = 0.0531\, m^3/kg$ (determined using X-Steam, f(p,t))

Speed of steam:

$$W_{Steam}^{SH1} = \frac{0.95 \cdot \dot{M}_{Steam}^{HP} \cdot \bar{v}_{SH1}}{\frac{\pi \cdot (d_{tube}^{SH1})^2}{4} \cdot n_{tube/r}^{SH1}} = \frac{0.95 \cdot 3.24 \cdot 0.0531}{\frac{\pi \cdot 0.0238^2}{4} \cdot 23} = 15.98\, m/s$$

The speed of steam traveling through the superheater is within the suggested range, between 15 and 25 m/s.

4.2.3 Verification of the speed of flue gas passing through superheater SH_{1HP}

Recalculated flue gas enthalpy and temperature at point C:

The parameters of the flue gas exiting the superheater at point C are recalculated according to the actual flue gas parameter entering SH_{1HP}.

$$I_C' = I_B^{real} - \frac{Q_{SH1}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 644.15 - \frac{1314.27}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 587.92\, kJ/Nm^3$$

$$t_C' = \frac{I_C' - I_{Flue:400}}{I_{Flue:500} - I_{Flue:400}} \cdot (500 - 400) + 400 = \frac{587.92 - 547.44}{693.27 - 547.44} \cdot 100 + 400 = 427.76℃$$

Average flue gas temperature:

$$\bar{t}_{(BC)} = \frac{t_B^{real} + t_C'}{2} = \frac{466.32 + 427.76}{2} = 447.04℃$$

The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{SH1} = \frac{273.15 + \bar{t}_{(BC)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 447.04}{273.15} \cdot 23.49 = 61.93\, m^3/s$$

The cross sectional area that the flue gas flows through:

$$A_{duct}^{SH1} = H \cdot L - H \cdot D_{tube}^{SH1} \cdot n_{tube/r}^{SH1} - H \cdot 2 \cdot h_{fin}^{SH1} \cdot th_{fin}^{SH1} \cdot n_{fin}^{SH1} \cdot n_{tube/r}^{SH1}$$

$$A_{duct}^{SH1} = 6.78 \cdot 1.76 - 6.78 \cdot 0.0318 \cdot 23 - 6.78 \cdot 2 \cdot 0.015 \cdot 0.001 \cdot 130 \cdot 23 = 6.37\, m^2$$

Speed of the flue gas passing through SH_{1HP}:

$$W_{Flue}^{SH1} = \frac{\dot{M}_{VFlue}^{SH1}}{A_{duct}^{SH1}} = \frac{61.93}{6.37} = 9.73\ m/s$$

The speed of the flue gas through the superheater is within the suggested range, between 9 and 12 m/s.

4.2.4 Reduced heat transfer coefficient outside the tubes of SH_{1HP}, through the flue gas

The coefficient of heat transfer through convection outside the finned tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{SH1} = \frac{s_{1(SH1)}}{D_{tube}^{SH1}} = \frac{0.0748}{0.0318} = 2.35$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2{'}^{SH1} = \frac{s'^{SH1}}{D_{tube}^{SH1}} = \frac{\sqrt{\left(\frac{s_{1(SH1)}}{2}\right)^2 + \left(s_{2(SH1)}\right)^2}}{D_{tube}^{SH1}} = \frac{\sqrt{\left(\frac{0.0748}{2}\right)^2 + 0.117^2}}{0.0318} = 3.86$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_{\sigma}^{SH1} = \frac{\sigma_1^{SH1} - 1}{{\sigma_2}'^{SH1} - 1} = \frac{2.35 - 1}{3.86 - 1} = 0.47$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through SH_{1HP} has been determined in previous calculations regarding the flue gas speed verification.

Temperature $\bar{t}_{(BC)} = 447.04\ ℃$

$$\lambda_{Flue}^{SH1} = \left[\frac{447.04 - 400}{100} \cdot (63.82 - 55.59) + 55.59\right] \cdot 10^{-3} = 5.946 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{SH1} = \left[\frac{447.04 - 400}{100} \cdot (75.55 - 59.87) + 59.87\right] \cdot 10^{-6} = 6.725 \cdot 10^{-5}\ m^2/s$$

The correction coefficient for the number of lateral row of tubes in the heat exchanger:

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 4 lateral rows of tubes in superheater SH_{1HP}.

$$c_z^{SH1} = 0.94$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c^{SH1} = 0.23 \cdot c_z^{SH1} \cdot (\varphi_{\sigma}^{SH1})^{0.2} \cdot \frac{\lambda_{Flue}^{SH1}}{s_{fin}^{SH1}} \cdot \left(\frac{D_{tube}^{SH1}}{s_{fin}^{SH1}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{SH1}}{s_{fin}^{SH1}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{SH1} \cdot s_{fin}^{SH1}}{\nu_{Flue}^{SH1}}\right)^{0.65}$$

$$\alpha_c^{SH1} = 0.23 \cdot 0.91 \cdot 0.47^{0.2} \cdot \frac{0.05946}{0.00769} \cdot \left(\frac{0.0318}{0.00769}\right)^{-0.54} \cdot \left(\frac{0.015}{0.00769}\right)^{-0.14} \cdot \left(\frac{9.73 \cdot 0.00769}{6.725 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{SH1} = 58.16\ W/(m^2K)$$

Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of $\beta^{SH1} \cdot h_{fin}^{SH1}$ and the quotient of $D_{fin}^{SH1}/D_{tube}^{SH1}$, where β^{SH1} is a coefficient.

Coefficient β^{SH1}:

$$\beta^{SH1} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{SH1}}{th_{fin}^{SH1} \cdot \lambda_{fin} \cdot \left(1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{SH1}\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 58.16}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 58.16)}}$$

$$\beta^{SH1} = 46.32\ m^{-1}$$

Values needed to determine the coefficient characterizing the effectiveness of the fins:

$$\beta^{SH1} \cdot h_{fin}^{SH1} = 46.32 \cdot 0.015 = 0.69$$

$$\frac{D_{fin}^{SH1}}{D_{tube}^{SH1}} = \frac{0.0618}{0.0318} = 1.94$$

Coefficient determined using the graph:

$$E^{SH1} = 0.84$$

Proportions of the tubes surface area made up of the fins, and tube wall:

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}^{SH1}}{S_{out}^{SH1}} = \frac{\left(\frac{D_{fin}^{SH1}}{D_{tube}^{SH1}}\right)^2 - 1}{\left(\frac{D_{fin}^{SH1}}{D_{tube}^{SH1}}\right)^2 - 1 + 2 \cdot \left(\frac{s_{fin}^{SH1}}{D_{tube}^{SH1}} - \frac{th_{fin}^{SH1}}{D_{tube}^{SH1}}\right)} = \frac{\left(\frac{0.0618}{0.0318}\right)^2 - 1}{\left(\frac{0.0618}{0.0318}\right)^2 - 1 + 2 \cdot \left(\frac{0.00769}{0.0318} - \frac{0.001}{0.0318}\right)}$$

$$\frac{S_{fin}^{SH1}}{S_{out}^{SH1}} = 0.87$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}^{SH1}}{S_{out}^{SH1}} = 1 - \frac{s_{fin}^{SH1}}{S_{out}^{SH1}} = 1 - 0.87 = 0.13$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out}^{SH1} = \left(\frac{S_{fin}^{SH1}}{S_{out}^{SH1}} \cdot E^{SH1} \cdot \mu + \frac{S_{out-fin}^{SH1}}{S_{out}^{SH1}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{SH1}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{SH1}}$$

$$\alpha_{r:out}^{SH1} = (0.87 \cdot 0.84 \cdot 1 + 0.13) \cdot \frac{0.85 \cdot 58.16}{1 + 0.0043 \cdot 0.85 \cdot 58.16} = 34.5\ W/(m^2K)$$

4.2.5 Reduced heat transfer coefficient inside the tubes of SH_{1HP}, through the Steam

Steam parameters:

The kinematic viscosity, thermal conductivity, and Prandtl number of the steam passing through SH_{1HP} are determined through X-Steam according to the average temperature and pressure of the steam. The kinematic viscosity is calculated from the dynamic viscosity and the steam density. The average temperature and pressure of the steam have been determined in earlier calculations regarding steam speed verification.

Temperature $\bar{t}_{SH1} = 334.93\ ℃$

Pressure $\bar{p}_{SH1} = 4.75$ MPa

$$\rho_{Steam}^{SH1} = 18.8405\ kg/m^3$$

$$\mu_{Steam}^{SH1} = 2.1446 \cdot 10^{-5}\ Pa \cdot s$$

$$\lambda_{Steam}^{SH1} = 0.0537\ W/(m \cdot K)$$

$$Pr_{Steam}^{SH1} = 2.245$$

Kinematic viscosity of the steam:

$$\nu_{Steam}^{SH1} = \frac{\mu_{Steam}^{SH1}}{\rho_{Steam}^{SH1}} = \frac{2.1446 \cdot 10^{-5}}{18.8405} = 1.1383 \cdot 10^{-6}\ m^2/s$$

Reduced heat transfer coefficient inside the tubes of SH_{2HP}:

$$\alpha_{r:in}^{SH1} = 0.023 \cdot \frac{\lambda_{Steam}^{SH1}}{d_{tube}^{SH1}} \cdot \left(\frac{W_{Steam}^{SH1} \cdot d_{tube}^{SH1}}{\nu_{Steam}^{SH1}}\right)^{0.8} \cdot (Pr_{Steam}^{SH1})^{0.4} \cdot c_t \cdot c_l \cdot c_m$$

$$\alpha_{r:in}^{SH1} = 0.023 \cdot \frac{0.0537}{0.0238} \cdot \left(\frac{15.98 \cdot 0.0238}{1.1383 \cdot 10^{-6}}\right)^{0.8} \cdot 2.245^{0.4} \cdot 1 \cdot 1 \cdot 1 = 1884.83\ W/(m^2K)$$

4.2.6 Overall heat transfer coefficient for SH_{1HP}

The outer surface area of one fin:

$$S_{1fin}^{SH1} = 2 \cdot \pi \cdot \frac{\left(D_{fin}^{SH1}\right)^2 - \left(D_{tube}^{SH1}\right)^2}{4} + \pi \cdot D_{fin}^{SH1} \cdot th_{fin}^{SH1}$$

$$S_{1fin}^{SH1} = 2 \cdot \pi \cdot \frac{0.0618^2 - 0.0318^2}{4} + \pi \cdot 0.0618 \cdot 0.001 = 0.0046\, m^2$$

The outer surface area of one finned tube per meter length:

$$S_{out/1m}^{SH1} = \pi \cdot D_{tube}^{SH1} \cdot \left(1 - n_{fin}^{SH1} \cdot th_{fin}^{SH1}\right) + n_{fin}^{SH1} \cdot S_{1fin}^{SH1}$$

$$S_{out/1m}^{SH1} = \pi \cdot 0.0318 \cdot (1 - 130 \cdot 0.001) + 130 \cdot 0.0046 = 0.6856\, m$$

The inner surface area of one finned tube per meter length:

$$S_{in/1m}^{SH1} = \pi \cdot d_{tube}^{SH1} = \pi \cdot 0.0238 = 0.0748\, m$$

The overall heat transfer coefficient:

$$k^{SH1} = \frac{1}{\frac{1}{\alpha_{r:out}^{SH1}} + \frac{1}{\alpha_{r:in}^{SH1}} \cdot \frac{S_{out/1m}^{SH1}}{S_{in/1m}^{SH1}}} = \frac{1}{\frac{1}{34.5} + \frac{1}{1884.83} \cdot \frac{0.6856}{0.0748}} = 29.54\ W/(m^2K)$$

4.2.7 Logarithmic mean temperature difference across SH_{1HP}

$$\Delta t_1^{SH1} = t_B^{real} - t_3^{HP} = 466.32 - 408.46 = 57.86\, K$$

$$\Delta t_2^{SH1} = t_C' - t_4^{HP} = 427.76 - 261.4 = 166.35\, K$$

$$\Delta t_{ln}^{SH1} = \frac{\Delta t_2^{SH1} - \Delta t_1^{SH1}}{ln\left(\frac{\Delta t_2^{SH1}}{\Delta t_1^{SH1}}\right)} = \frac{166.35 - 57.87}{ln\left(\frac{166.35}{57.87}\right)} = 102.73\, K$$

4.2.8 The number of lateral rows of tubes required for SH_{1HP}

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{SH1} = \frac{Q_{SH1}^{HP}}{k^{SH1} \cdot \Delta t_{ln}^{SH1}} = \frac{1314.27}{29.54 \cdot 102.73} = 433.03\, m^2$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{SH1} = S_{out/1m}^{SH1} \cdot H \cdot n_{tube/r}^{SH1} = 0.6856 \cdot 6.78 \cdot 23 = 106.9\, m^2$$

The number of lateral rows of tubers in SH_{2HP}:

$$n_{row}^{SH1} = \frac{S_{out}^{SH1}}{S_{out/r}^{SH1}} = \frac{433.03}{106.9} = 4.05$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{SH1} = 4$$

4.2.9 Calculating the actual heat transfer rate of SH_{1HP}

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{SH1:real} = S_{out/r}^{SH1} \cdot n_{row}^{SH1} = 106.9 \cdot 4 = 427.62\, m^2$$

The actual heat transfer rate of SH_{1HP}:

$$Q_{SH1}^{HP:real} = k^{SH1} \cdot S_{out}^{SH1:real} \cdot \Delta t_{ln}^{SH1} = 29.54 \cdot 427.62 \cdot 102.73 = 1297.87 kW$$

Error verification:

$$\%Error^{SH1} = \left| \left(\frac{1297.78}{1314.27} - 1 \right) \cdot 100 \right| = 1.25\%$$

The discrepancy between the theoretical and the actual heat transfer rate of superheater SH_{1HP} is less than 2% , thus the current configuration of SH_{1HP} is acceptable.

4.2.10 Determining the actual parameters of the flue gas exiting SH_{1HP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of superheater SH_{1HP} the parameters of the flue gas exiting SH_{1HP} at point C must be recalculated.

Actual enthalpy of the flue gas at point C:

$$I_C^{real} = I_B^{real} - \frac{Q_{SH1}^{HP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 644.32 - \frac{1297.87}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 588.62\, kJ/Nm^3$$

Actual temperature of the flue gas at point C:

$$t_C^{real} = \frac{I_C^{real} - I_{Flue:400}}{I_{Flue:500} - I_{Flue:400}} \cdot (500 - 400) + 400 = \frac{588.62 - 547.44}{693.27 - 547.44} \cdot 100 + 400 = 428.24\ ℃$$

4.3 Design of the High Pressure Evaporator EV_{HP}

Several parameters are required in the calculations associated with the design of the high pressure evaporator EV_{HP}. Parameters, other than those describing the geometry of the evaporator, that are necessary for the calculations regarding the design and sizing of evaporator EV_{HP}, some of which were determined in previous calculations, are organized in *Table 4.8* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of EV_{HP} (theoretical value)	Q_{EV}^{HP}	5171.76	$[kW]$
Flue	Actual temperature of flue gas entering into EV_{HP}	t_C^{real}	428.24	[℃]
	Actual enthalpy of flue gas entering into EV_{HP}	I_C^{real}	588.62	$[kJ/Nm^3]$
Water/steam	Temperature of water entering EV_{HP}	t_6^{HP}	256.4	[℃]
	Pressure of water entering EV_{HP}	p_6^{HP}	4.8	$[MPa]$
	Temperature of steam exiting EV_{HP}	t_4^{HP}	261.4	[℃]
	Pressure of steam exiting EV_{HP}	p_4^{HP}	4.8	$[MPa]$

Table 4.8 Parameters necessary for evaporator EV_{HP} design, not including parameters describing the geometry of EV_{HP}

4.3.1 Geometry of the high pressure evaporator EV_{HP}

The chosen dimensions of the finned tubes used in the high pressure evaporator (EV_{HP}) are shown in *Table 4.9* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.9* are selected and later on adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

	Parameter	Symbol	Value	Units
tube	Outer tube diameter	D_{tube}^{EV1}	57	$[mm]$
	Tube wall thickness	th_{tube}^{EV1}	4	$[mm]$
	Inner tube diameter	d_{tube}^{EV1}	49	$[mm]$
fins	Fin thickness	th_{fin}^{EV1}	1	$[mm]$
	Number of fins per meter	n_{fin}^{EV1}	150	$[mm]$
	Fin spacing	s_{fin}^{EV1}	6.67	$[mm]$
	Fin height	h_{fin}^{EV1}	19	$[mm]$
	Outer fin diameter	D_{fin}^{EV1}	95	$[mm]$

Table 4.9 Parameters selected for the finned tubes used in the high pressure evaporator EV_{HP}

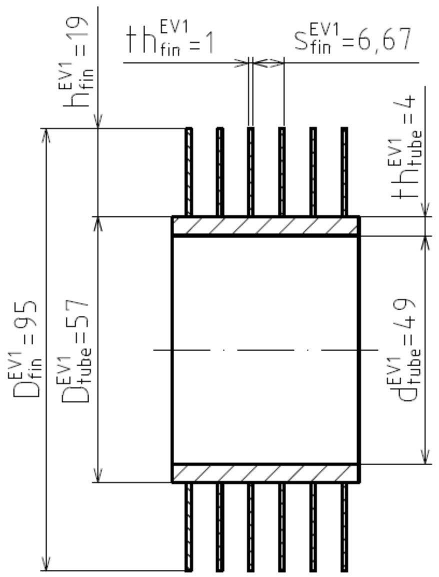

Figure 4-5 Tube geometry EV_{HP}

The dimensions describing the layout of the finned tubes of evaporator EV_{HP} are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the evaporator.

Chosen layout dimensions:

Lateral gap between tubes $a_{EV1} = 11.5$ mm

Longitudinal tube spacing $s_{2(EV1)} = 117$ mm

Lateral tube spacing in evaporator EV_{HP}:

$$s_{1(EV1)} = D_{tube}^{EV1} + 2 \cdot h_{fin}^{EV1} + a_{EV1} = 0.057 + 2 \cdot 0.019 + 0.0115 = 0.1065\ m$$

Number of tubes in each lateral row of tubes in EV_{HP}:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{EV1} = \frac{L}{s_{1(EV1)}} - 0.5 = \frac{1.76}{0.1065} - 0.5 = 16.03$$

$$n_{tube/r}^{EV1} = 16$$

Gap verification:

There needs to be a gap between the wall of the flue gas duct and outer tubes of the heat exchanger, however it must not be excessive. This value should be no greater than 10 mm. The calculated gap is within the acceptable range.

$$gap = L - n_{tube/r}^{EV1} \cdot s_{1(EV1)} - \frac{D_{tube}^{EV1}}{2} - h_{fin}^{EV1}$$

$$gap = 1.76 - 16 \cdot 0.1065 - \frac{0.057}{2} - 0.019 = 0.0085\ m$$

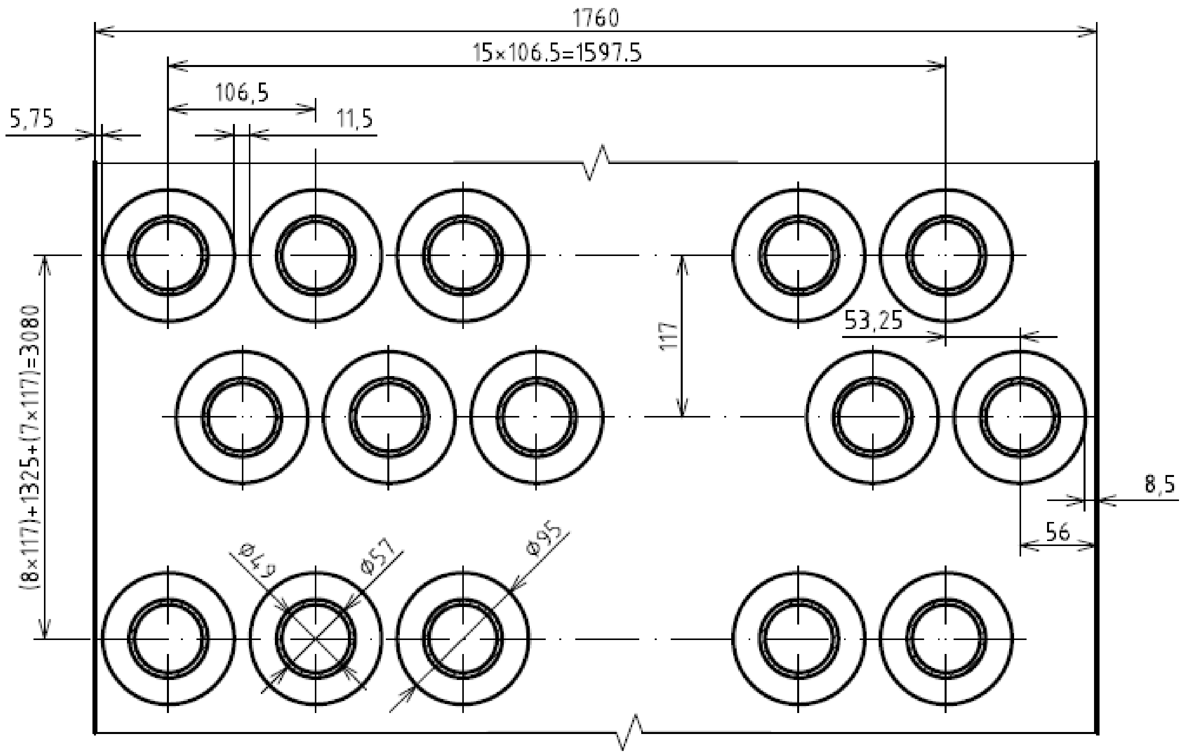

Figure 4-6 Tube layout EV_{HP}

4.3.2 Verification of the speed of flue gas passing through evaporator EV_{HP}

Recalculated flue gas enthalpy and temperature at point D:

The parameters of the flue gas exiting the evaporator at point C are recalculated according to the actual flue gas parameter entering EV_{HP}.

$$I'_D = I_C^{real} - \frac{Q_{EV}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 588.62 - \frac{5171.76}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 367.35\, kJ/Nm^3$$

$$t'_D = \frac{I'_D - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 = \frac{367.35 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 272.02℃$$

Average flue gas temperature:

$$\bar{t}_{(CD)} = \frac{t_C^{real} + t'_D}{2} = \frac{428.24 + 272.02}{2} = 350.13℃$$

The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{EV1} = \frac{273.15 + \bar{t}_{(CD)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 350.13}{273.15} \cdot 23.49 = 53.6\, m^3/s$$

The cross sectional area that the flue gas flows through:

$$A_{duct}^{EV1} = H \cdot L - H \cdot D_{tube}^{EV1} \cdot n_{tube/r}^{EV1} - H \cdot 2 \cdot h_{fin}^{EV1} \cdot th_{fin}^{EV1} \cdot n_{fin}^{EV1} \cdot n_{tube/r}^{EV1}$$

$$A_{duct}^{EV1} = 6.78 \cdot 1.76 - 6.78 \cdot 0.057 \cdot 16 - 6.78 \cdot 2 \cdot 0.019 \cdot 0.001 \cdot 150 \cdot 16 = 5.13\, m^2$$

Speed of the flue gas passing through EV_{HP}:

$$W_{Flue}^{EV1} = \frac{\dot{M}_{VFlue}^{EV1}}{A_{duct}^{EV1}} = \frac{53.6}{5.13} = 10.45\ m/s$$

The speed of the flue gas through the evaporator is within the suggested range, between 9 and 12 m/s.

4.3.3 Reduced heat transfer coefficient outside the tubes of EV_{HP}, through the flue gas

The coefficient of heat transfer through convection outside the finned tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{EV1} = \frac{s_{1(EV1)}}{D_{tube}^{EV1}} = \frac{0.1065}{0.057} = 1.87$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2{'}^{EV1} = \frac{s'^{EV1}}{D_{tube}^{EV1}} = \frac{\sqrt{\left(\frac{s_{1(EV1)}}{2}\right)^2 + \left(s_{2(EV1)}\right)^2}}{D_{tube}^{EV1}} = \frac{\sqrt{\left(\frac{0.1065}{2}\right)^2 + 0.117^2}}{0.057} = 2.26$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_\sigma^{EV1} = \frac{\sigma_1^{EV1} - 1}{\sigma_2{'}^{EV1} - 1} = \frac{1.87 - 1}{2.26 - 1} = 0.69$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through EV_{HP} has been determined in previous calculations regarding the flue gas speed verification.

Temperature $\bar{t}_{(CD)} = 350.13\ ℃$

$$\lambda_{Flue}^{EV1} = \left[\frac{350.13 - 300}{100} \cdot (55.59 - 47.35) + 47.35\right] \cdot 10^{-3} = 5.148 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{EV1} = \left[\frac{350.13 - 400}{100} \cdot (59.87 - 45.45) + 45.45\right] \cdot 10^{-6} = 5.268 \cdot 10^{-5}\ m^2/s$$

The correction coefficient for the number of lateral row of tubes in the heat exchanger:

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 17 lateral rows of tubes in evaporator EV_{HP}.

$$c_z^{EV1} = 1.01$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c^{EV1} = 0.23 \cdot c_z^{EV1} \cdot (\varphi_\sigma^{EV1})^{0.2} \cdot \frac{\lambda_{Flue}^{EV1}}{s_{fin}^{EV1}} \cdot \left(\frac{D_{tube}^{EV1}}{s_{fin}^{EV1}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{EV1}}{s_{fin}^{EV1}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{EV1} \cdot s_{fin}^{EV1}}{\nu_{Flue}^{EV1}}\right)^{0.65}$$

$$\alpha_c^{EV1} = 0.23 \cdot 1.01 \cdot 0.69^{0.2} \cdot \frac{0.05148}{0.00667} \cdot \left(\frac{0.057}{0.00667}\right)^{-0.54} \cdot \left(\frac{0.019}{0.00667}\right)^{-0.14} \cdot \left(\frac{10.45 \cdot 0.00667}{5.268 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{EV1} = 48.26\ W/(m^2K)$$

Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of $\beta^{EV1} \cdot h_{fin}^{EV1}$ and the quotient of $D_{fin}^{EV1}/D_{tube}^{EV1}$, where β^{EV1} is a coefficient.

Coefficient β^{EV1}:

$$\beta^{EV1} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{EV1}}{th_{fin}^{EV1} \cdot \lambda_{fin} \cdot \left(1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{EV1}\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 48.26}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 48.26)}}$$

$$\beta^{EV1} = 42.84\ m^{-1}$$

Values needed to determine the coefficient characterizing the effectiveness of the fins:

$$\beta^{EV1} \cdot h_{fin}^{EV1} = 42.84 \cdot 0.019 = 0.81$$

$$\frac{D_{fin}^{EV1}}{D_{tube}^{EV1}} = \frac{0.095}{0.057} = 1.67$$

Coefficient determined using the graph:

$$E^{EV1} = 0.79$$

Proportions of the tubes surface area made up of the fins, and tube wall:

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}^{EV1}}{S_{out}^{EV1}} = \frac{\left(\frac{D_{fin}^{EV1}}{D_{tube}^{EV1}}\right)^2 - 1}{\left(\frac{D_{fin}^{EV1}}{D_{tube}^{EV1}}\right)^2 - 1 + 2 \cdot \left(\frac{s_{fin}^{EV1}}{D_{tube}^{EV1}} - \frac{th_{fin}^{EV1}}{D_{tube}^{EV1}}\right)} = \frac{\left(\frac{0.095}{0.057}\right)^2 - 1}{\left(\frac{0.095}{0.057}\right)^2 - 1 + 2 \cdot \left(\frac{0.00667}{0.057} - \frac{0.001}{0.057}\right)}$$

$$\frac{S_{fin}^{EV1}}{S_{out}^{EV1}} = 0.9$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}^{EV1}}{S_{out}^{EV1}} = 1 - \frac{S_{fin}^{EV1}}{S_{out}^{EV1}} = 1 - 0.9 = 0.1$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out}^{EV1} = \left(\frac{S_{fin}^{EV1}}{S_{out}^{EV1}} \cdot E^{EV1} \cdot \mu + \frac{S_{out-fin}^{EV1}}{S_{out}^{EV1}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{EV1}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{EV1}}$$

$$\alpha_{r:out}^{EV1} = (0.9 \cdot 0.79 \cdot 1 + 0.1) \cdot \frac{0.85 \cdot 48.26}{1 + 0.0043 \cdot 0.85 \cdot 48.26} = 28.16\, W/(m^2K)$$

4.3.4 Overall heat transfer coefficient for EV$_{HP}$

For the design of evaporators and economizers the effect that the reduced coefficient of heat transfer inside the tubes $\alpha_{r:in}$ has on the overall heat transfer coefficient k is negligible. Thus the overall heat transfer coefficient of evaporators and economizers are calculated in the same manner, as seen in equation (4.19).

$$k^{EV1} = \alpha_{r:out}^{EV1} = 28.16\, W/(m^2K) \tag{4.19}$$

4.3.5 Logarithmic mean temperature difference across EV$_{HP}$

$$\Delta t_1^{EV1} = t_C^{real} - t_4^{HP} = 428.24 - 261.4 = 166.83\, K$$

$$\Delta t_2^{EV1} = t'_D - t_6^{HP} = 272.03 - 256.4 = 15.62\, K$$

$$\Delta t_{ln}^{EV1} = \frac{\Delta t_2^{EV1} - \Delta t_1^{EV1}}{ln\left(\frac{\Delta t_2^{EV1}}{\Delta t_1^{EV1}}\right)} = \frac{15.62 - 166.83}{ln\left(\frac{15.62}{166.83}\right)} = 63.84\, K$$

4.3.6 The number of lateral rows of tubes required for EV$_{HP}$

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{EV1} = \frac{Q_{EV}^{HP}}{k^{EVH} \cdot \Delta t_{ln}^{EV1}} = \frac{5171.76}{28.16 \cdot 63.84} = 2876.76\, m^2$$

The outer surface area of one fin:

$$S_{1fin}^{EV1} = 2 \cdot \pi \cdot \frac{\left(D_{fin}^{EV1}\right)^2 - \left(D_{tube}^{EV1}\right)^2}{4} + \pi \cdot D_{fin}^{EV1} \cdot th_{fin}^{EV1}$$

$$S_{1fin}^{EV1} = 2 \cdot \pi \cdot \frac{0.095^2 - 0.057^2}{4} + \pi \cdot 0.095 \cdot 0.001 = 0.0094\ m^2$$

The outer surface area of one finned tube per meter length:

$$S_{out/1m}^{EV1} = \pi \cdot D_{tube}^{EV1} \cdot \left(1 - n_{fin}^{EV1} \cdot th_{fin}^{EV1}\right) + n_{fin}^{EV1} \cdot S_{1fin}^{EV1}$$

$$S_{out/1m}^{EV1} = \pi \cdot 0.057 \cdot (1 - 150 \cdot 0.001) + 150 \cdot 0.0094 = 1.5579\ m$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{EV1} = S_{out/1m}^{EV1} \cdot H \cdot n_{tube/r}^{EV1} = 1.5579 \cdot 6.78 \cdot 16 = 169\ m^2$$

The number of lateral rows of tubers in EV_{HP}:

$$n_{row}^{EV1} = \frac{S_{out}^{EV1}}{S_{out/r}^{EV1}} = \frac{2876.76}{169} = 17.02$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{EV1} = 17$$

4.3.7 Calculating the actual heat transfer rate of EV_{HP}

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{EV1:real} = S_{out/r}^{EV1} \cdot n_{row}^{EV1} = 169 \cdot 17 = 2873.05\ m^2$$

The actual heat transfer arte of EV_{HP}:

$$Q_{EV}^{HP:real} = k^{EV1} \cdot S_{out}^{EV1:real} \cdot \Delta t_{ln}^{EV1} = 28.16 \cdot 2873.05 \cdot 63.84 = 5165.08\ kW$$

Error verification:

$$\%Error^{EV1} = \left|\left(\frac{5165.08}{5171.76} - 1\right) \cdot 100\right| = 0.13\%$$

The discrepancy between the theoretical and the actual heat transfer rate of evaporator EV_{HP} is less than 2% , thus the current configuration of the evaporator is acceptable.

4.3.8 Determining the actual parameters of the flue gas exiting EV_{HP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of evaporator EV_{HP} the parameters of the flue gas exiting EV_{HP} at point D must be recalculated.

Actual enthalpy of the flue gas at point D:

$$I_D^{real} = I_C^{real} - \frac{Q_{EV}^{HP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 588.62 - \frac{5165.08}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 367.64\, kJ/Nm^3$$

Actual temperature of the flue gas at point D:

$$t_D^{real} = \frac{I_D^{real} - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 = \frac{367.64 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 272.23\ ℃$$

4.4 Design of the High Pressure Economizer ECO_{3HP}

Several parameters are required in the calculations associated with the design of the high pressure economizer ECO_{3HP}. Parameters, other than those describing the geometry of the economizer, that are necessary for the calculations regarding the design and sizing of economizer ECO_{3HP}, some of which were determined in previous calculations, are organized in *Table 4.10* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of ECO_{3HP} (theoretical value)	Q_{ECO3}^{HP}	1466.23	$[kW]$
Flue	Actual temperature of flue gas entering into ECO_{3HP}	t_D^{real}	272.23	[℃]
Flue	Actual enthalpy of flue gas entering into ECO_{3HP}	I_D^{real}	367.64	$[kJ/Nm^3]$
water	Mass flow rate of water in the high pressure circuit (only 95% passes through ECO_{3HP})	$\dot{M}_{Steam}^{HP}$	3.24	$[kg/s]$
water	Temperature of water entering ECO_{3HP}	t_7^{HP}	151.4	[℃]
water	Pressure of water entering ECO_{3HP}	p_7^{HP}	4.9	$[MPa]$
water	Temperature of water exiting ECO_{3HP}	t_6^{HP}	256.4	[℃]
water	Pressure of water exiting ECO_{3HP}	p_6^{HP}	4.8	$[MPa]$

Table 4.10 Parameters necessary for economizer ECO_{3HP} design, not including parameters describing the geometry of ECO_{3HP}

4.4.1 Geometry of the high pressure economizer ECO_{3HP}

The chosen dimensions of the finned tubes used in the high pressure economizer (ECO_{3HP}) are shown in *Table 4.11* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.11* are selected and later on adjusted in order to supply the economizer with an acceptable water flow speed through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

	Parameter	Symbol	Value	Units
tube	Outer tube diameter	D_{tube}^{ECO3}	31.8	$[mm]$
	Tube wall thickness	th_{tube}^{ECO3}	4	$[mm]$
	Inner tube diameter	d_{tube}^{ECO3}	23.8	$[mm]$
fins	Fin thickness	th_{fin}^{ECO3}	1	$[mm]$
	Number of fins per meter	n_{fin}^{ECO3}	190	$[mm]$
	Fin spacing	s_{fin}^{ECO3}	5.26	$[mm]$
	Fin height	h_{fin}^{ECO3}	8	$[mm]$
	Outer fin diameter	D_{fin}^{ECO3}	47.8	$[mm]$

Table 4.11 Parameters selected for the finned tubes used in the high pressure economizer ECO$_{3HP}$

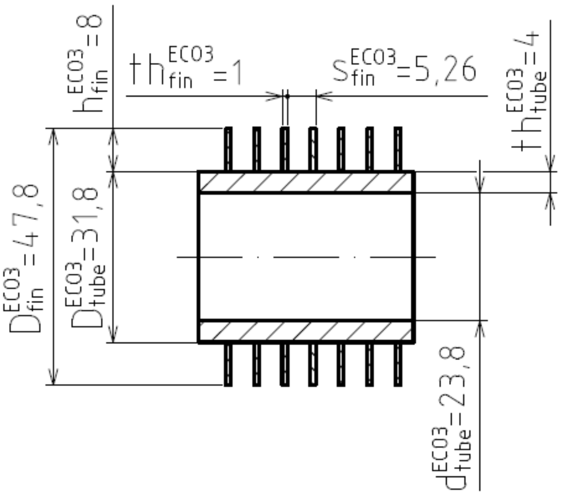

Figure 4-7 Tube geometry ECO$_{3HP}$

The dimensions describing the layout of the finned tubes of economizer ECO$_{3HP}$ are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the economizer.

Chosen layout dimensions:

Lateral gap between tubes $a_{ECO3} = 6.25$ mm

Longitudinal tube spacing $s_{2(ECO3)} = 90$ mm

Lateral tube spacing in economizer ECO3HP:

$$s_{1(ECO3)} = D_{tube}^{ECO3} + 2 \cdot h_{fin}^{ECO3} + a_{ECO3} = 0.0318 + 2 \cdot 0.008 + 0.00625 = 0.05405m$$

Number of tubes in each lateral row of tubes in ECO3HP:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{ECO3} = \frac{L}{s_{1(ECO3)}} - 0.5 = \frac{1.76}{0.05404} - 0.5 = 32.06$$

$$n_{tube/r}^{ECO3} = 32$$

Gap verification:

There needs to be a gap between the wall of the flue gas duct and outer tubes of the heat exchanger, however it must not be excessive. This value should be no greater than 10 mm. The calculated gap is within the acceptable range.

$$gap = L - n_{tube/r}^{ECO3} \cdot s_{1(ECO3)} - \frac{D_{tube}^{ECO3}}{2} - h_{fin}^{ECO3}$$

$$gap = 1.76 - 32 \cdot 0.05405 - \frac{0.0318}{2} - 0.008 = 0.0065\, m$$

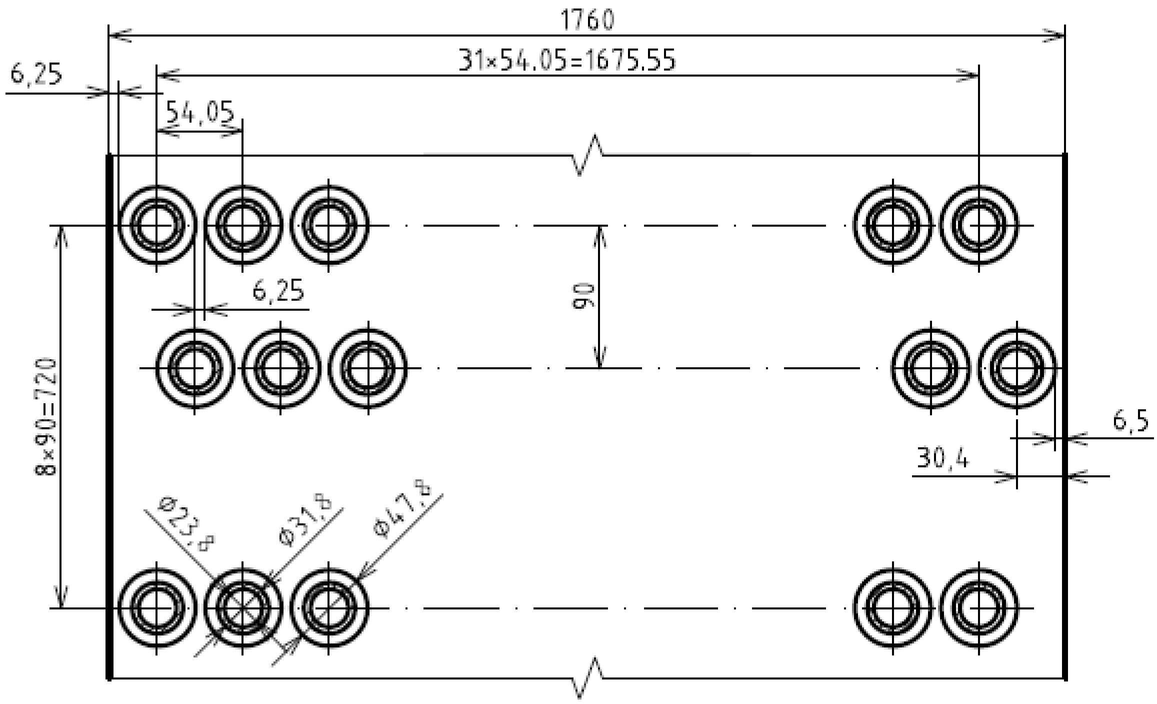

Figure 4-8 Tube layout ECO3HP

4.4.2 Verification of the speed of water passing through economizer ECO3HP

The speed of the water passing through economizer ECO3HP is calculated according to average specific volume of the water. The specific volume of the water is determined through X-Steam according to the average pressure and temperature of the water passing through ECO3HP.

$$\bar{t}_{ECO3} = \frac{t_6^{HP} + t_7^{HP}}{2} = \frac{265.4 + 151.4}{2} = 203.9\ ^\circ\mathrm{C}$$

$$\bar{p}_{ECO3} = \frac{p_6^{HP} + p_7^{HP}}{2} = \frac{4.8 + 4.9}{2} = 4.85\ MPa$$

$\bar{v}_{ECO3} = 0.0012\ m^3/kg$ (determined using X-Steam, f(p,t))

Speed of water, before splitting economizer into sections:

$$W_{Steam}^{ECO3\,\prime} = \frac{0.95 \cdot \dot{M}_{Steam}^{HP} \cdot \bar{v}_{ECO3}}{\frac{\pi \cdot (d_{tube}^{ECO3})^2}{4} \cdot n_{tube/r}^{ECO3}} = \frac{0.95 \cdot 3.24 \cdot 0.0012}{\frac{\pi \cdot 0.0238^2}{4} \cdot 32} = 0.25\ m/s$$

The speed of water flowing through the economizer must be within the suggested range of 0.8 to 1.3 m/s. This is accomplished by splitting each lateral row of tubes, of the economizer, into sections to increase the speed of water. However, it is recommended that the rows are not split into more than four sections. The actual speed of water flowing through the economizer is then calculated intuitively as seen in equation *(4.20)*.

Number of sections $n_{section}^{ECO3} = 4$

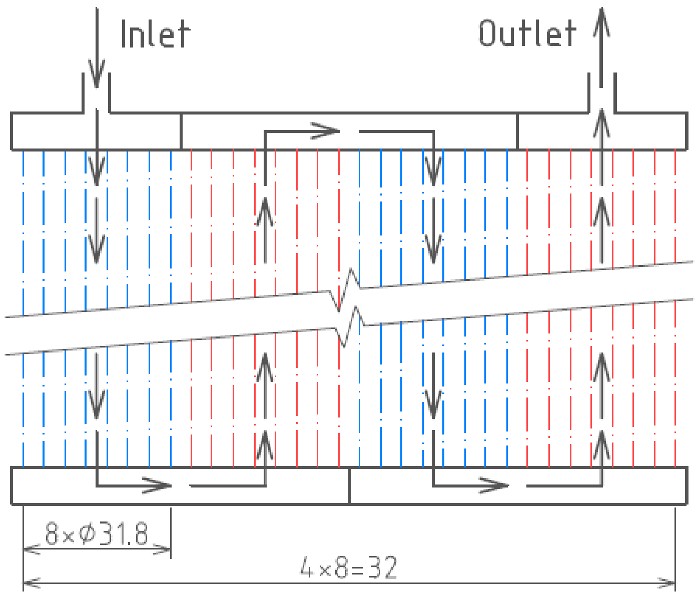

Figure 4-9 Splitting schematic ECO3HP

Actual speed of water flowing through ECO3HP:

$$W_{Steam}^{ECO3} = W_{Steam}^{ECO3\,\prime} \cdot n_{section}^{ECO3} = 0.25 \cdot 4 = 1\ m/s \qquad (4.20)$$

4.4.3 Verification of the speed of flue gas passing through economizer ECO_{3HP}

Recalculated flue gas enthalpy and temperature at point E:

The parameters of the flue gas exiting the economizer at point E are recalculated according to the actual flue gas parameter entering ECO_{3HP}.

$$I'_E = I_D^{real} - \frac{Q_{ECO3}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 367.64 - \frac{1466.23}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 304.9\, kJ/Nm^3$$

$$t'_E = \frac{I'_E - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 = \frac{304.9 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 226.75℃$$

Average flue gas temperature:

$$\bar{t}_{(DE)} = \frac{t_D^{real} + t'_E}{2} = \frac{272.23 + 226.75}{2} = 249.49℃$$

The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{ECO3} = \frac{273.15 + \bar{t}_{(DE)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 249.49}{273.15} \cdot 23.49 = 44.94\, m^3/s$$

The cross sectional area that the flue gas flows through:

$$A_{duct}^{ECO3} = H \cdot L - H \cdot D_{tube}^{ECO3} \cdot n_{tube/r}^{ECO3} - H \cdot 2 \cdot h_{fin}^{ECO3} \cdot th_{fin}^{ECO3} \cdot n_{fin}^{ECO3} \cdot n_{tube/r}^{ECO3}$$

$$A_{duct}^{ECO3} = 6.78 \cdot 1.76 - 6.78 \cdot 0.0318 \cdot 32 - 6.78 \cdot 2 \cdot 0.008 \cdot 0.001 \cdot 190 \cdot 23 = 4.37\, m^2$$

Speed of the flue gas passing through ECO_{3HP}:

$$W_{Flue}^{ECO3} = \frac{\dot{M}_{VFlue}^{ECO3}}{A_{duct}^{ECO3}} = \frac{44.94}{4.37} = 10.27\ m/s$$

The speed of the flue gas through the economizer is within the suggested range, between 9 and 12 m/s.

4.4.4 Reduced heat transfer coefficient outside the tubes of ECO_{3HP}, through the flue gas

The coefficient of heat transfer through convection outside the finned tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{ECO3} = \frac{s_{1(ECO3)}}{D_{tube}^{ECO3}} = \frac{0.05405}{0.0318} = 1.7$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{ECO3} = \frac{s'^{ECO3}}{D_{tube}^{ECO3}} = \frac{\sqrt{\left(\frac{s_{1(ECO3)}}{2}\right)^2 + \left(s_{2(ECO3)}\right)^2}}{D_{tube}^{ECO3}} = \frac{\sqrt{\left(\frac{0.05405}{2}\right)^2 + 0.09^2}}{0.0318} = 2.96$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_{\sigma}^{ECO3} = \frac{\sigma_1^{ECO3} - 1}{\sigma_2{}'^{ECO3} - 1} = \frac{1.7 - 1}{2.96 - 1} = 0.36$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through ECO_{3HP} has been determined in previous calculations regarding the flue gas speed verification.

Temperature $\bar{t}_{(DE)} = 249.49\ ℃$

$$\lambda_{Flue}^{ECO3} = \left[\frac{249.49 - 200}{100} \cdot (47.35 - 39.23) + 39.23\right] \cdot 10^{-3} = 4.325 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{ECO3} = \left[\frac{249.49 - 200}{100} \cdot (45.45 - 32.36) + 32.36\right] \cdot 10^{-6} = 3.884 \cdot 10^{-5}\ m^2/s$$

The correction coefficient for the number of lateral row of tubes in the heat exchanger:

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 9 lateral rows of tubes in economizer ECO_{3HP}.

$$c_z^{ECO3} = 1$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c^{ECO3} = 0.23 \cdot c_z^{ECO3} \cdot (\varphi_{\sigma}^{ECO3})^{0.2} \cdot \frac{\lambda_{Flue}^{ECO3}}{s_{fin}^{ECO3}} \cdot \left(\frac{D_{tube}^{ECO3}}{s_{fin}^{ECO3}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{ECO3}}{s_{fin}^{ECO3}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{ECO3} \cdot s_{fin}^{ECO3}}{\nu_{Flue}^{ECO3}}\right)^{0.65}$$

$$\alpha_c^{ECO3} = 0.23 \cdot 1 \cdot 0.36^{0.2} \cdot \frac{0.04325}{0.00526} \cdot \left(\frac{0.0318}{0.00526}\right)^{-0.54} \cdot \left(\frac{0.008}{0.00526}\right)^{-0.14} \cdot \left(\frac{10.27 \cdot 0.00526}{3.884 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{ECO3} = 60.72\ W/(m^2K)$$

Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of $\beta^{ECO3} \cdot h_{fin}^{ECO3}$ and the quotient of $D_{fin}^{ECO3}/D_{tube}^{ECO3}$, where β^{ECO3} is a coefficient.

Coefficient β^{ECO3}:

$$\beta^{ECO3} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{ECO3}}{th_{fin}^{ECO3} \cdot \lambda_{fin} \cdot \left(1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO3}\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 60.72}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 60.72)}}$$

$$\beta^{ECO3} = 47.15\, m^{-1}$$

Values needed to determine the coefficient characterizing the effectiveness of the fins:

$$\beta^{ECO3} \cdot h_{fin}^{ECO3} = 47.15 \cdot 0.008 = 0.38$$

$$\frac{D_{fin}^{ECO3}}{D_{tube}^{ECO3}} = \frac{0.0478}{0.0318} = 1.5$$

Coefficient determined using the graph:

$$E^{ECO3} = 0.94$$

Proportions of the tubes surface area made up of the fins, and tube wall:

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}^{ECO3}}{S_{out}^{ECO3}} = \frac{\left(\frac{D_{fin}^{ECO3}}{D_{tube}^{ECO3}}\right)^2 - 1}{\left(\frac{D_{fin}^{ECO3}}{D_{tube}^{ECO3}}\right)^2 - 1 + 2 \cdot \left(\frac{s_{fin}^{ECO3}}{D_{tube}^{ECO3}} - \frac{th_{fin}^{ECO3}}{D_{tube}^{ECO3}}\right)} = \frac{\left(\frac{0.0478}{0.0318}\right)^2 - 1}{\left(\frac{0.0478}{0.0318}\right)^2 - 1 + 2 \cdot \left(\frac{0.00526}{0.0318} - \frac{0.001}{0.0318}\right)}$$

$$\frac{S_{fin}^{ECO3}}{S_{out}^{ECO3}} = 0.82$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}^{ECO3}}{S_{out}^{ECO3}} = 1 - \frac{s_{fin}^{ECO3}}{S_{out}^{ECO3}} = 1 - 0.82 = 0.18$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out}^{ECO3} = \left(\frac{S_{fin}^{ECO3}}{S_{out}^{ECO3}} \cdot E^{ECO3} \cdot \mu + \frac{S_{out-fin}^{ECO3}}{S_{out}^{ECO3}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{ECO3}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO3}}$$

$$\alpha_{r:out}^{ECO3} = (0.82 \cdot 0.94 \cdot 1 + 0.18) \cdot \frac{0.85 \cdot 60.72}{1 + 0.0043 \cdot 0.85 \cdot 60.72} = 40.29\, W/(m^2 K)$$

4.4.5 Overall heat transfer coefficient for ECO3HP

For the design of evaporators and economizers the effect that the reduced coefficient of heat transfer inside the tubes $\alpha_{r:in}$ has on the overall heat transfer coefficient k is negligible. Thus the overall heat transfer coefficient of evaporators and economizers are calculated in the same manner, as seen in equation (4.19).

$$k^{ECO3} = \alpha_{r:out}^{ECO3} = 40.29\,W/(m^2K)$$

4.4.6 Logarithmic mean temperature difference across ECO3HP

$$\Delta t_1^{ECO3} = t_D^{real} - t_6^{HP} = 272.23 - 256.4 = 15.82\,K$$

$$\Delta t_2^{ECO3} = t_E' - t_7^{HP} = 226.75 - 151.4 = 75.34\,K$$

$$\Delta t_{ln}^{ECO3} = \frac{\Delta t_2^{ECO3} - \Delta t_1^{ECO3}}{ln\left(\frac{\Delta t_2^{ECO3}}{\Delta t_1^{ECO3}}\right)} = \frac{75.34 - 15.82}{ln\left(\frac{75.34}{15.82}\right)} = 38.14\,K$$

4.4.7 The number of lateral rows of tubes required for ECO3HP

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{ECO3} = \frac{Q_{ECO3}^{HP}}{k^{ECO3} \cdot \Delta t_{ln}^{ECO3}} = \frac{1466.23}{40.29 \cdot 38.14} = 954.16\,m^2$$

The outer surface area of one fin:

$$S_{1fin}^{ECO3} = 2 \cdot \pi \cdot \frac{\left(D_{fin}^{ECO3}\right)^2 - \left(D_{tube}^{ECO3}\right)^2}{4} + \pi \cdot D_{fin}^{ECO3} \cdot th_{fin}^{ECO3}$$

$$S_{1fin}^{ECO3} = 2 \cdot \pi \cdot \frac{0.0478^2 - 0.0318^2}{4} + \pi \cdot 0.0478 \cdot 0.001 = 0.0022\,m^2$$

The outer surface area of one finned tube per meter length:

$$S_{out/1m}^{ECO3} = \pi \cdot D_{tube}^{ECO3} \cdot \left(1 - n_{fin}^{ECO3} \cdot th_{fin}^{ECO3}\right) + n_{fin}^{ECO3} \cdot S_{1fin}^{ECO3}$$

$$S_{out/1m}^{ECO3} = \pi \cdot 0.0318 \cdot (1 - 190 \cdot 0.001) + 190 \cdot 0.0022 = 0.4896\,m$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{ECO3} = S_{out/1m}^{ECO3} \cdot H \cdot n_{tube/r}^{ECO3} = 0.4896 \cdot 6.78 \cdot 32 = 106.22\,m^2$$

The number of lateral rows of tubers in ECO_{3HP}:

$$n_{row}^{ECO3} = \frac{S_{out}^{ECO3}}{S_{out/r}^{ECO3}} = \frac{954.16}{106.22} = 8.98$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{ECO3} = 9$$

4.4.8 Calculating the actual heat transfer rate of ECO_{3HP}

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{ECO3:real} = S_{out/r}^{ECO3} \cdot n_{row}^{ECO3} = 106.22 \cdot 9 = 955.94\, m^2$$

The actual heat transfer rate of ECO_{3HP}:

$$Q_{ECO3}^{HP:real} = k^{ECO3} \cdot S_{out}^{ECO3:real} \cdot \Delta t_{ln}^{ECO3} = 40.29 \cdot 955.94 \cdot 38.14 = 1468.96 kW$$

Error verification:

$$\%Error^{ECO3} = \left| \left(\frac{1468.96}{1466.23} - 1 \right) \cdot 100 \right| = 0.19\%$$

The discrepancy between the theoretical and the actual heat transfer rate of economizer ECO_{3HP} is less than 2% , thus the current configuration of ECO_{3HP} is acceptable.

4.4.9 Determining the actual parameters of the flue gas exiting ECO_{3HP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of economizer ECO_{3HP} the parameters of the flue gas exiting ECO_{3HP} at point E must be recalculated.

Actual enthalpy of the flue gas at point E:

$$I_E^{real} = I_D^{real} - \frac{Q_{ECO3}^{HP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 367.64 - \frac{1468.96}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 304.79\, kJ/Nm^3$$

Actual temperature of the flue gas at point E:

$$t_E^{real} = \frac{I_E^{real} - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 = \frac{304.79 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 226.66\,℃$$

4.5 Design of the Low Pressure Superheater SH_{LP}

The low pressure superheater SH_{LP} will be designed with smooth tubes, because of the low rate of heat transfer required of this heat exchanger.

Equations used in the design and sizing of heat exchangers with smooth tubes:

Some of the equations parameters used in the design and sizing of heat exchangers with smooth tubes are the same as the those used for heat exchangers with finned tubes. However the equations regarding the coefficient of heat transfer for heat exchangers with smooth tube are different. These equations are presented below in an order that is clear and straight forward. This, however is not the chronological order in which the equations are calculated. Several values used in these calculations are presented in *Table 4.12*.

Values used in calculations (heat exchangers with smooth tubes)	Symbol	Value	Units
Coefficient of thermal effectiveness	Ψ	0.85	$[-]$
Coefficient of the degree to which the flue gas is utilized	ξ	1	$[-]$
Coefficient of emissivity of the tube walls	a_{wall}	0.8	$[-]$
Flue gas pressure (assuming atmospheric)	$p1$	0.1	$[MPa]$
Given temperature difference (for gaseous fuels)	Δt	25	[°C]

Table 4.12 Values used in the calculations for heat exchangers with smooth tubes

Overall coefficient of heat transfer:

$$k = \frac{\Psi \cdot \alpha_{out}}{1 + \frac{\alpha_{out}}{\alpha_{in}}} \tag{4.21}$$

Where:

Ψ is the coefficient of thermal effectiveness [-]

α_{out} is the coefficient of heat transfer outside the tubes, between the flue gas and the heat exchanger tubes [W/(m^{2}K)]

α_{in} is the coefficient of heat transfer inside the tubes, between the heat exchanger tubes and the water/steam inside them [W/(m^{2}K)]

The coefficient of heat transfer inside the tubes:

The coefficient of heat transfer inside the tubes is calculated in the same manner as the reduced heat transfer coefficient inside the tubes for heat exchangers with finned tubes.

$$\alpha_{in} = 0.023 \cdot \frac{\lambda_{Steam}}{d_e} \cdot \left(\frac{W_{Steam} \cdot d_e}{\nu_{Steam}}\right)^{0.8} \cdot Pr_{Steam}{}^{0.4} \cdot c_t \cdot c_l \cdot c_m \tag{4.22}$$

The coefficient of heat transfer outside the heat exchanger tubes:

$$\alpha_{out} = \xi \cdot (\alpha_{c1} + \alpha_r) \tag{4.23}$$

Where:

ξ is the coefficient characterizing the degree to which the flue gas is utilized [-]

α_{c1} is the coefficient of heat transfer through convection outside the smooth tubes of the heat exchanger [W/(m^{2}K)]

α_r is the coefficient of heat transfer through radiation outside the tubes of the heat exchanger [W/(m^{2}K)]

The coefficient of heat transfer through convection outside the smooth tubes:

$$\alpha_{c1} = c_s \cdot c_{z1} \cdot \frac{\lambda_{Flue}}{D_{tube}} \cdot \left(\frac{W_{Flue} \cdot D_{tube}}{\nu_{Flue}}\right)^{0.6} \cdot {Pr_{Flue}}^{0.33} \tag{4.24}$$

Where:

c_s is a correction coefficient characterizing the layout of the heat exchanger tubes [-]

c_{z1} is a correction coefficient depending on the number of lateral rows of tubes [-]

Pr_{Flue} is the Prandtl number of the flue gas [-]

The correction coefficient characterizing the layout of the heat exchanger tubes:

The flowing equation, equation *(4.25)*, is applicable in the case where, $0.1 < \varphi_\sigma \leq 1.7$.

$$c_s = 0.34 \cdot {\varphi_\sigma}^{0.1} \tag{4.25}$$

The correction coefficient depending on the number of lateral rows of tubes:

The flowing equation, equation *(4.26)*, is applicable in the case where, $n_{row} < 10$ and $\sigma_1 < 3$.

$$c_{z1} = 3.12 \cdot {n_{row}}^{0.05} - 2.5 \tag{4.26}$$

The coefficient of heat transfer through radiation outside the tubes:

$$\alpha_r = 5.7 \cdot 10^{-8} \cdot \frac{a_{wall} + 1}{2} \cdot a_{Flue} \cdot {T_{Flue}}^3 \cdot \frac{1 - \left(\frac{T_{wall}}{T_{Flue}}\right)^{3.6}}{1 - \frac{T_{wall}}{T_{Flue}}} \tag{4.27}$$

Where:

a_{wall} is the coefficient of emissivity of the outer surface of the tube walls [-]

a_{Flue} is the coefficient of emissivity of the flue gas [-]

T_{Flue} is the absolute flue gas temperature [K]

T_{wall} is the absolute temperature of the tarnished outer tube walls [K]

The absolute temperature of the outer tube wall:

$$T_{wall} = \bar{t}_{Steam} + \Delta t + 273.15 \tag{4.28}$$

Where:

$\bar{t}_{Steam}$ is the average temperature of the steam inside the tubes [°C]

Δt is a given temperature difference [°C]

The coefficient of emissivity of the flue gas:

$$a_{Flue} = 1 - e^{-b \cdot p \cdot s} \tag{4.29}$$

Where:

b is the coefficient of radiation decline [1/(m·MPa)]

s is the effective radiation layer thickness [m]

$p1$ is the flue gas pressure, which is assumed to be atmospheric [MPa]

The effective radiation layer thickness:

$$s = 0.9 \cdot D_{tube} \cdot \left(\frac{4}{\pi} \cdot \frac{s_1 \cdot s_2}{{D_{tube}}^2} - 1\right) \tag{4.30}$$

The coefficient of radiation decline:

$$b = b_{tri} \cdot x_{tri} \tag{4.31}$$

Where:

b_{tri} is the coefficient of radiation decline due to the presence of triatomic gases [1/(m·MPa)]

x_{tri} is the concentration of triatomic gases in the flue gas [-]

The coefficient of radiation decline due to the presence of triatomic gases:

$$b_{tri} = \left(\frac{7.8 + 16 \cdot x_{H2O}}{3.16 \cdot \sqrt{p_{p:tri} \cdot s}} - 1\right) \cdot \left(1 - 0.37 \cdot \frac{T_{Flue}}{1000}\right) \tag{4.32}$$

Where:

$p_{p:tri}$ is the partial pressure of triatomic gases in the flue gas [MPa]

The concentration of triatomic gases in the flue gas:

$$x_{tri} = x_{H2O} \cdot x_{CO2} \tag{4.33}$$

The partial pressure of triatomic gases in the flue gas:

$$p_{p:tri} = p1 \cdot x_{tri} \tag{4.34}$$

Flue gas parameters:

The flue gas parameters needed to complete the calculation of equation *(4.24)* are determined according to the moisture concentration contained in the flue gas (7.8%), and the average temperature of the flue gas passing through the heat exchanger. The thermal conductivity and the kinematic viscosity of the flue gas are determined from *Table 4.2* and *Table 4.3* respectively. The Prandtl number of the flue gas is determined from *Table 4.13*, seen below. Linear interpolation is used to interpolate between values obtained from source [2].

$\bar{t}_{Flue}$ [°C]	Moisture concentration X_{H2O} [%]		
	5	7.8	10
0	0.69	0.707	0.72
100	0.67	0.681	0.69
200	0.65	0.661	0.67
300	0.63	0.641	0.65
400	0.62	0.631	0.64
500	0.61	0.621	0.63

Table 4.13 Prandtl number of the flue gas, Pr_{Flue} [-]

Several parameters are required in the calculations associated with the design of the high pressure superheater SH_{LP}. Parameters, other than those describing the geometry of the superheater, that are necessary for the calculations regarding the design and sizing of superheater SH_{LP}, some of which were determined in previous calculations, are organized in *Table 4.14* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of SH_{LP} (theoretical value)	Q_{SH}^{LP}	35.16	$[kW]$
Flue	Actual temperature of flue gas entering into SH_{LP}	t_E^{real}	226.66	[°C]
	Actual enthalpy of flue gas entering into SH_{LP}	I_E^{real}	304.79	$[kJ/Nm^3]$
Steam	Mass flow rate of steam in the high pressure circuit (only 95% passes through SH_{LP})	$\dot{M}_{Steam}^{LP}$	0.88	$[kg/s]$
	Temperature of steam entering SH_{LP}	t_2^{LP}	156.15	[°C]
	Pressure of steam entering SH_{LP}	p_2^{LP}	0.56	$[MPa]$
	Temperature of steam exiting SH_{LP}	t_1^{LP}	170	[°C]
	Pressure of steam exiting SH_{LP}	p_1^{LP}	0.46	$[MPa]$

Table 4.14 Parameters necessary for superheater SH_{LP} design, not including parameters describing the geometry of SH_{LP}

4.5.1 Geometry of the high pressure superheater SH$_{LP}$

The chosen dimensions of the smooth tubes used in the high pressure superheater (SH$_{LP}$) are shown in *Table 4.15* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.7* are selected and later on adjusted in order to supply the superheater with an acceptable steam flow speed through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

Parameter	Symbol	Value	Units
Outer tube diameter	D_{tube}^{SH}	38	$[mm]$
Tube wall thickness	th_{tube}^{SH}	4	$[mm]$
Inner tube diameter	d_{tube}^{SH}	30	$[mm]$

Table 4.15 Parameters selected for the smooth tubes used in the high pressure superheater SH$_{LP}$

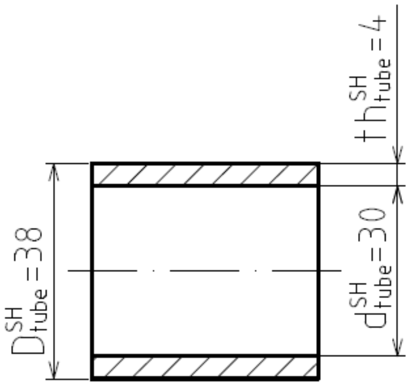

Figure 4-10 Tube geometry SH$_{LP}$

The dimensions describing the layout of the finned tubes of superheater SH$_{LP}$ are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the superheater.

Chosen layout dimensions:

Lateral gap between tubes $a_{SH} = 38$ mm

Longitudinal tube spacing $s_{2(SH)} = 117$ mm

Lateral tube spacing in superheater SH$_{LP}$:

$$s_{1(SH)} = D_{tube}^{SH} + a_{SH} = 0.038 + 0.037 = 0.075m$$

Number of tubes in each lateral row of tubes in SH$_{LP}$:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{SH} = \frac{L}{s_{1(SH)}} = \frac{1.76}{0.076} = 23.15$$

$$n_{tube/r}^{SH} = 23$$

Gap verification:

There needs to be a gap between the wall of the flue gas duct and outer tubes of the heat exchanger, however it must not be excessive. This value should be no greater than a_{SH}. The calculated gap is within the acceptable range.

$$gap = L - \left(n_{tube/r}^{SH} - 1\right) \cdot s_{1(SH)} - \frac{s_{1(SH)}}{2} - \frac{D_{tube}^{SH}}{2}$$

$$gap = 1.76 - (23 - 1) \cdot 0.076 - \frac{0.076}{2} - \frac{0.038}{2} = 0.031\ m$$

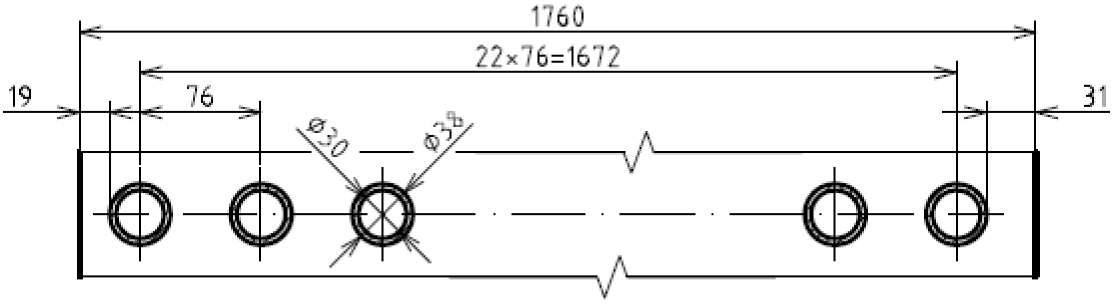

Figure 4-11 Tube layout SH$_{LP}$

4.5.2 Verification of the speed of steam passing through superheater SH$_{LP}$

The speed of the steam passing through superheater SH$_{LP}$ is calculated according to average specific volume of the steam. The specific volume of the steam is determined through X-Steam according to the average pressure and temperature of the steam passing through SH$_{LP}$.

$$\bar{t}_{SH} = \frac{t_1^{LP} + t_2^{LP}}{2} = \frac{170 + 156.15}{2} = 163.08\ ℃$$

$$\bar{p}_{SH} = \frac{p_1^{LP} + p_2^{LP}}{2} = \frac{0.46 + 0.56}{2} = 0.51\ MPa$$

$$\bar{v}_{SH} = 0.379\ m^3/kg$$ (determined using X-Steam, f(p,t))

Speed of steam:

$$W_{Steam}^{SH} = \frac{\dot{M}_{Steam}^{LP} \cdot \bar{v}_{SH}}{\frac{\pi \cdot (d_{tube}^{SH})^2}{4} \cdot n_{tube/r}^{SH}} = \frac{0.88 \cdot 0.379}{\frac{\pi \cdot 0.03^2}{4} \cdot 23} = 19.54\ m/s$$

The speed of steam traveling through the superheater is within the suggested range, between 15 and 25 m/s.

4.5.3 Verification of the speed of flue gas passing through superheater SH_{LP}

Recalculated flue gas enthalpy and temperature at point F:

The parameters of the flue gas exiting the superheater at point F are recalculated according to the actual flue gas parameter entering SH_{LP}.

$$I'_F = I_E^{real} - \frac{Q_{SH}^{LP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 226.66 - \frac{35.16}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 303.28\, kJ/Nm^3$$

$$t'_F = \frac{I'_E - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 = \frac{303.28 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 225.57℃$$

Average flue gas temperature:

$$\bar{t}_{(EF)} = \frac{t_E^{real} + t'_F}{2} = \frac{226.66 + 225.57}{2} = 226.12℃$$

The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{SH} = \frac{273.15 + \bar{t}_{(EF)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 226.12}{273.15} \cdot 23.49 = 42.93\, m^3/s$$

The cross sectional area that the flue gas flows through:

$$A_{duct}^{SH} = H \cdot L - H \cdot D_{tube}^{SH} \cdot n_{tube/r}^{SH} = 6.78 \cdot 1.76 - 6.78 \cdot 0.038 \cdot 23 = 6.01\, m^2$$

Speed of the flue gas passing through SH_{LP}:

$$W_{Flue}^{SH} = \frac{\dot{M}_{VFlue}^{SH}}{A_{duct}^{SH}} = \frac{42.93}{6.01} = 7.15\ m/s$$

4.5.4 Coefficient of heat transfer outside the tubes of SH_{LP}

The coefficient of heat transfer through convection outside the smooth tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{SH} = \frac{s_{1(SH)}}{D_{tube}^{SH}} = \frac{0.076}{0.038} = 2$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2{'}^{SH} = \frac{s'^{SH}}{D_{tube}^{SH}} = \frac{\sqrt{\left(\frac{s_{1(SH)}}{2}\right)^2 + \left(s_{2(SH)}\right)^2}}{D_{tube}^{SH}} = \frac{\sqrt{\left(\frac{0.076}{2}\right)^2 + 0.117^2}}{0.038} = 3.24$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_{\sigma}^{SH} = \frac{\sigma_1^{SH} - 1}{\sigma_2{'}^{SH} - 1} = \frac{1.97 - 1}{3.24 - 1} = 0.44$$

The correction coefficient characterizing the layout of the heat exchanger tubes:

The flowing equation is applicable, because $0.1 < \varphi_{\sigma}^{SH} \leq 1.7$.

$$c_s^{SH} = 0.34 \cdot (\varphi_{\sigma}^{SH})^{0.1} = 0.34 \cdot 0.44^{0.1} = 0.31$$

The correction coefficient depending on the number of lateral rows of tubes:

The flowing equation is applicable, because $n_{row}^{SH} < 10$ and $\sigma_1^{SH} < 3$. Assuming that there is only 1 lateral row of tubes in superheater SH_{LP}.

$$c_{z1}^{SH} = 3.12 \cdot (n_{row}^{SH})^{0.05} - 2.5 = 3.12 \cdot 1^{0.05} - 2.5 = 0.62$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The Prandtl number of the flue gas is determined from *Table 4.13*. The average temperature of the flue gas passing through SH_{LP} has been determined in previous calculations regarding flue gas speed verification.

Temperature $\bar{t}_{(EF)} = 226.12$℃

$$\lambda_{Flue}^{SH} = \left[\frac{226.12 - 200}{100} \cdot (47.35 - 39.23) + 39.23\right] \cdot 10^{-3} = 4.135 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{SH} = \left[\frac{226.12 - 200}{100} \cdot (45.45 - 32.36) + 32.36\right] \cdot 10^{-6} = 3.578 \cdot 10^{-5}\ m^2/s$$

$$Pr_{Flue}^{SH} = \left[\frac{226.12 - 200}{100} \cdot (0.641 - 0.661) + 0.661\right] \cdot 10^{-6} = 0.656\ m^2/s$$

The coefficient of heat transfer through convection outside the smooth tubes:

$$\alpha_{c1}^{SH} = c_s^{SH} \cdot c_{z1}^{SH} \cdot \frac{\lambda_{Flue}^{SH}}{D_{tube}^{SH}} \cdot \left(\frac{W_{Flue}^{SH} \cdot D_{tube}^{SH}}{\nu_{Flue}^{SH}}\right)^{0.6} \cdot (Pr_{Flue}^{SH})^{0.33}$$

$$\alpha_{c1}^{SH} = 0.31 \cdot 0.62 \cdot \frac{0.04135}{0.038} \cdot \left(\frac{7.15 \cdot 0.038}{3.578 \cdot 10^{-5}}\right)^{0.6} \cdot 0.656^{0.33} = 39.11\ W/(m^2K)$$

The coefficient of heat transfer through radiation outside the tubes of the heat exchanger:

The effective radiation layer thickness:

$$s = 0.9 \cdot D_{tube}^{SH} \cdot \left(\frac{4}{\pi} \cdot \frac{s_1^{SH} \cdot s_2^{SH}}{{D_{tube}^{SH}}^2} - 1\right) = 0.9 \cdot 0.038 \cdot \left(\frac{4}{\pi} \cdot \frac{0.076 \cdot 0.117}{0.038^2} - 1\right) = 0.23m$$

The partial pressure of triatomic gases in the flue gas:

$$p_{p:tri} = p1 \cdot x_{tri} = 0.1 \cdot 0.122 = 0.0122\ MPa$$

The absolute flue gas temperature:

$$T_{Flue}^{SH} = \bar{t}_{(EF)} + 273.15 = 226.12 + 273.15 = 499.27\ K$$

The coefficient of radiation decline due to the presence of triatomic gases:

$$b_{tri}^{SH} = \left(\frac{7.8 + 16 \cdot x_{H2O}}{3.16 \cdot \sqrt{p_{p:tri} \cdot s_{SH}}} - 1\right) \cdot \left(1 - 0.37 \cdot \frac{T_{Flue}^{SH}}{1000}\right)$$

$$b_{tri}^{SH} = \left(\frac{7.8 + 16 \cdot 0.078}{3.16 \cdot \sqrt{0.0122 \cdot 0.23}} - 1\right)\left(1 - 0.37 \cdot \frac{499.27}{1000}\right) = 37.14\ m^{-1} \cdot MPa^{-1}$$

The concentration of triatomic gases in the flue gas:

$$x_{tri} = x_{H2O} \cdot x_{CO2} = 0.078 + 0.044 = 0.122$$

The coefficient of radiation decline:

$$b = b_{tri}^{SH} \cdot x_{tri} = 37.14 \cdot 0.122 = 4.53\ m^{-1} \cdot MPa^{-1}$$

The coefficient of emissivity of the flue gas:

$$a_{Flue}^{SH} = 1 - e^{-b \cdot p1 \cdot s} = 1 - e^{-4.53 \cdot 0.1 \cdot 0.23} = 0.0991$$

The absolute temperature of the tarnished outer tube walls:

$$T_{wall}^{SH} = \bar{t}_{SH} + 25 + 273.15 = 461.23\ K$$

The coefficient of heat transfer through radiation outside the tubes:

$$\alpha_r^{SH} = 5.7 \cdot 10^{-8} \cdot \frac{a_{wall} + 1}{2} \cdot a_{Flue}^{SH} \cdot {T_{Flue}^{SH}}^3 \cdot \frac{1 - \left(\frac{T_{wall}^{SH}}{T_{Flue}^{SH}}\right)^{3.6}}{1 - \frac{T_{wall}^{SH}}{T_{Flue}^{SH}}}$$

$$\alpha_r^{SH} = 5.7 \cdot 10^{-8} \cdot \frac{0.8 + 1}{2} \cdot 0.0991 \cdot 461.23^3 \cdot \frac{1 - \left(\frac{461.23}{499.27}\right)^{3.6}}{1 - \frac{461.23}{499.27}} = 2.06\ W/(m^2K)$$

The coefficient of heat transfer outside the heat exchanger tubes:

$$\alpha_{out}^{SH} = \xi \cdot (\alpha_{c1}^{SH} + \alpha_r^{SH}) = 1 \cdot (39.11 + 2.06) = 41.17\ W/(m^2K)$$

4.5.5 Coefficient of heat transfer inside the tubes of SH_{LP}

Steam parameters:

The kinematic viscosity, thermal conductivity, and Prandtl number of the steam passing through SH_{LP} are determined through X-Steam according to the average temperature and pressure of the steam. The kinematic viscosity is calculated from the dynamic viscosity and the steam density. The average temperature and pressure of the steam have been determined in earlier calculations regarding steam speed verification.

Temperature	$\bar{t}_{SH} = 163.08\ ℃$
Pressure	$\bar{p}_{SH} = 0.51$ MPa

$$\rho_{Steam}^{SH} = 2.6384\ kg/m^3$$

$$\mu_{Steam}^{SH} = 1.4515 \cdot 10^{-5}\ Pa \cdot s$$

$$\lambda_{Steam}^{SH} = 0.0316\ W/(m \cdot K)$$

$$Pr_{Steam}^{SH} = 1.0567$$

Kinematic viscosity of the steam:

$$\nu_{Steam}^{SH} = \frac{\mu_{Steam}^{SH}}{\rho_{Steam}^{SH}} = \frac{1.4515 \cdot 10^{-5}}{2.6384} = 5.5015 \cdot 10^{-6}\ m^2/s$$

The heat transfer coefficient inside the tubes of SH_{2HP}:

$$\alpha_{in}^{SH} = 0.023 \cdot \frac{\lambda_{Steam}^{SH}}{d_{tube}^{SH}} \cdot \left(\frac{W_{Steam}^{SH} \cdot d_{tube}^{SH}}{\nu_{Steam}^{SH}}\right)^{0.8} \cdot (Pr_{Steam}^{SH})^{0.4} \cdot c_t \cdot c_l \cdot c_m$$

$$\alpha_{in}^{SH} = 0.023 \cdot \frac{0.0316}{0.03} \cdot \left(\frac{19.54 \cdot 0.03}{5.5015 \cdot 10^{-6}}\right)^{0.8} \cdot 1.0567^{0.4} \cdot 1 \cdot 1 \cdot 1 = 260.92\ W/(m^2K)$$

4.5.6 Overall heat transfer coefficient for SH_{LP}

The overall heat transfer coefficient is calculated according to equation *(4.21)*.

$$k^{SH} = \frac{\Psi \cdot \alpha_{out}^{SH}}{1 + \frac{\alpha_{out}^{SH}}{\alpha_{in}^{SH}}} = \frac{0.85 \cdot 41.17}{1 + \frac{41.17}{260.92}} = 30.22\ W/\left(m^2K\right)$$

4.5.7 Logarithmic mean temperature difference across SH_{LP}

$$\Delta t_1^{SH} = t_E^{real} - t_1^{LP} = 226.66 - 170 = 56.66\ K$$

$$\Delta t_2^{SH} = t_F' - t_2^{LP} = 225.57 - 156.15 = 69.42\ K$$

$$\Delta t_{ln}^{SH} = \frac{\Delta t_2^{SH} - \Delta t_1^{SH}}{ln\left(\frac{\Delta t_2^{SH}}{\Delta t_1^{SH}}\right)} = \frac{69.42 - 56.66}{ln\left(\frac{69.42}{56.66}\right)} = 62.83\ K$$

4.5.8 The number of lateral rows of tubes required for SH_{LP}

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{SH} = \frac{Q_{SH}^{HP}}{k^{SH} \cdot \Delta t_{ln}^{SH}} = \frac{35.16}{30.22 \cdot 62.83} = 18.52\ m^2$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{SH} = H \cdot \pi \cdot D_{tube}^{SH} \cdot n_{tube/r}^{SH} = 6.78 \cdot \pi \cdot 0.038 \cdot 23 = 18.62\ m^2$$

The number of lateral rows of tubers in SH_{LP}:

$$n_{row}^{SH} = \frac{S_{out}^{SH}}{S_{out/r}^{SH}} = \frac{18.52}{18.62} = 0.99$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{SH} = 1$$

4.5.9 Calculating the actual heat transfer rate of SH_{LP}

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{SH:real} = S_{out/r}^{SH} \cdot n_{row}^{SH} = 18.62 \cdot 1 = 18.62\ m^2$$

The actual heat transfer of SH_{LP}:

$$Q_{SH}^{LP:real} = k^{SH} \cdot S_{out}^{SH:real} \cdot \Delta t_{ln}^{SH} = 30.22 \cdot 18.62 \cdot 62.83 = 35.35 kW$$

Error verification:

$$\%Error^{SH} = \left|\left(\frac{35.35}{35.16} - 1\right) \cdot 100\right| = 0.53\%$$

The discrepancy between the theoretical and the actual heat transfer rate of superheater SH_{LP} is less than 2% , thus the current configuration of SH_{LP} is acceptable.

4.5.10 Determining the actual parameters of the flue gas exiting SH_{LP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of superheater SH_{LP} the parameters of the flue gas exiting SH_{LP} at point F must be recalculated.

Actual enthalpy of the flue gas at point F:

$$I_F^{real} = I_E^{real} - \frac{Q_{SH}^{LP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 367.64 - \frac{35.35}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 303.27\, kJ/Nm^3$$

Actual temperature of the flue gas at point F:

$$t_F^{real} = \frac{I_F^{real} - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 = \frac{303.27 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 225.57\,℃$$

4.6 Design of the Low Pressure Evaporator EV_{LP}

Several parameters are required in the calculations associated with the design of the low pressure evaporator EV_{LP}. Parameters, other than those describing the geometry of the evaporator, that are necessary for the calculations regarding the design and sizing of evaporator EV_{LP}, some of which were determined in previous calculations, are organized in *Table 4.16* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of EV_{LP} (theoretical value)	Q_{EV}^{LP}	1866.54	$[kW]$
Flue	Actual temperature of flue gas entering into EV_{LP}	t_f^{real}	225.57	[℃]
	Actual enthalpy of flue gas entering into EV_{LP}	I_f^{real}	303.27	$[kJ/Nm^3]$
Water/steam	Temperature of water entering EV_{LP}	t_4^{LP}	151.15	[℃]
	Pressure of water entering EV_{LP}	p_4^{LP}	0.56	$[MPa]$
	Temperature of steam exiting EV_{LP}	t_2^{LP}	156.15	[℃]
	Pressure of steam exiting EV_{LP}	p_2^{LP}	0.56	$[MPa]$

Table 4.16 Parameters necessary for evaporator EV_{LP} design, not including parameters describing the geometry of EV_{LP}

4.6.1 Geometry of the high pressure evaporator EV_{LP}

The chosen dimensions of the finned tubes used in the low pressure evaporator (EV_{LP}) are shown in *Table 4.17* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.17* are selected and later on adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

	Parameter	Symbol	Value	Units
tube	Outer tube diameter	D_{tube}^{EV}	57	$[mm]$
	Tube wall thickness	th_{tube}^{EV}	4	$[mm]$
	Inner tube diameter	d_{tube}^{EV}	49	$[mm]$
fins	Fin thickness	th_{fin}^{EV}	1	$[mm]$
	Number of fins per meter	n_{fin}^{EV}	230	$[mm]$
	Fin spacing	s_{fin}^{EV}	4.35	$[mm]$
	Fin height	h_{fin}^{EV}	19	$[mm]$
	Outer fin diameter	D_{fin}^{EV}	95	$[mm]$

Table 4.17 Parameters selected for the finned tubes used in the high pressure evaporator EV_{LP}

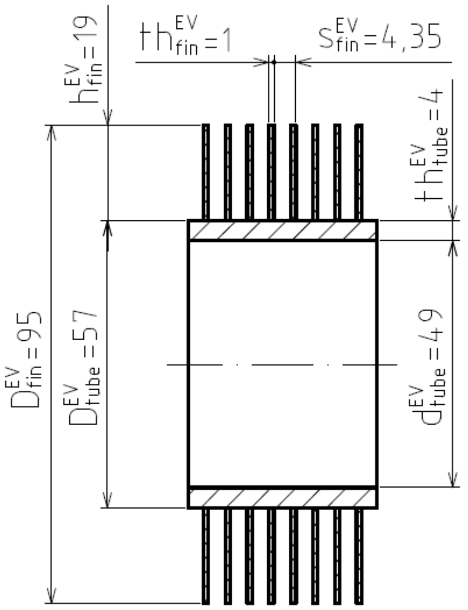

Figure 4-12 Tube geometry EV_{LP}

The dimensions describing the layout of the finned tubes of evaporator EV_{LP} are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the evaporator.

Chosen layout dimensions:

Lateral gap between tubes $a_{EV} = 5.5$ mm

Longitudinal tube spacing $s_{2(EV)} = 117$ mm

Lateral tube spacing in evaporator EV_{HP}:

$$s_{1(EV)} = D_{tube}^{EV} + 2 \cdot h_{fin}^{EV} + a_{EV} = 0.057 + 2 \cdot 0.019 + 0.0055 = 0.1005m$$

Number of tubes in each lateral row of tubes in EV$_{LP}$:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{EV} = \frac{L}{s_{1(EV)}} - 0.5 = \frac{1.76}{0.1005} - 0.5 = 17.01$$

$$n_{tube/r}^{EV} = 17$$

Gap verification:

There needs to be a gap between the wall of the flue gas duct and outer tubes of the heat exchanger, however it must not be excessive. This value should be no greater than 10 mm. The calculated gap is within the acceptable range.

$$gap = L - n_{tube/r}^{EV} \cdot s_{1(EVH)} - \frac{D_{tube}^{EV}}{2} - h_{fin}^{EV}$$

$$gap = 1.76 - 17 \cdot 0.1005 - \frac{0.057}{2} - 0.019 = 0.004\,m$$

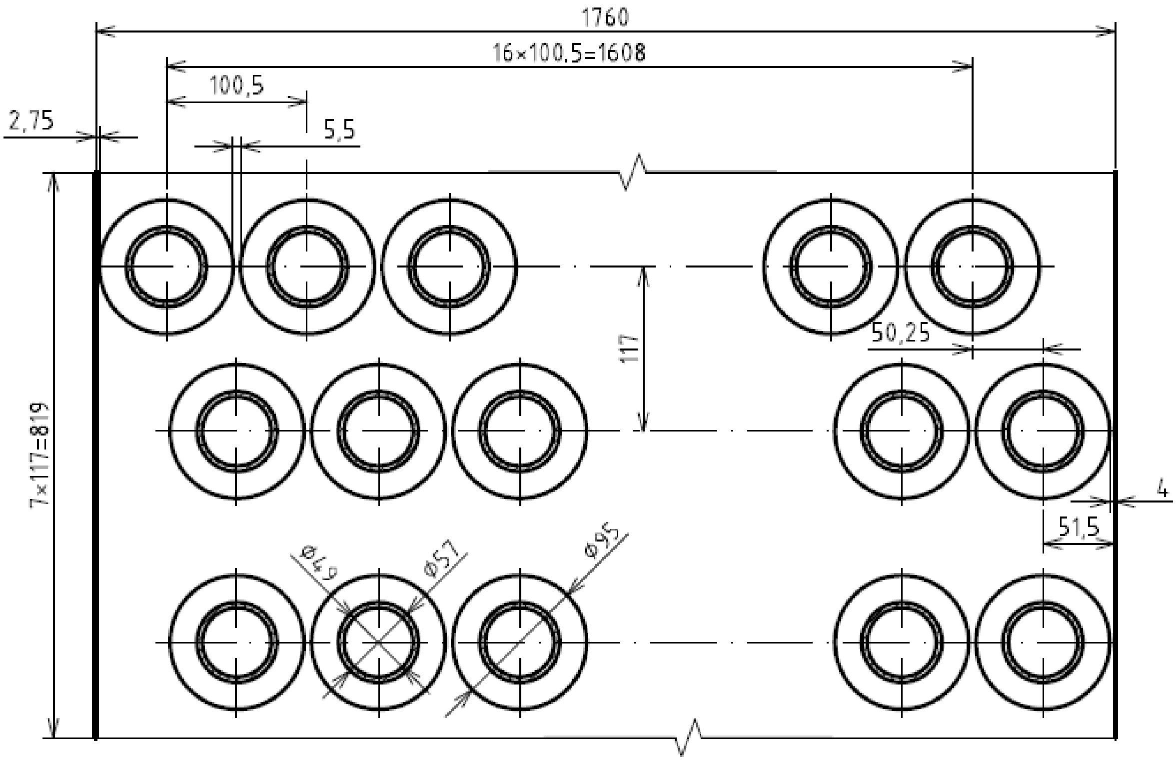

Figure 4-13 Tube layout EV$_{LP}$

4.6.2 Verification of the speed of flue gas passing through evaporator EV$_{LP}$

Recalculated flue gas enthalpy and temperature at point G:

The parameters of the flue gas exiting the evaporator at point G are recalculated according to the actual flue gas parameter entering EV$_{LP}$.

$$I_G' = I_F^{real} - \frac{Q_{EV}^{LP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 225.57 - \frac{1866.54}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 223.41\, kJ/Nm^3$$

$$t_G' = \frac{I_F' - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{223.41 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 166.9℃$$

Average flue gas temperature:

$$\bar{t}_{(FG)} = \frac{t_F^{real} + t_G'}{2} = \frac{225.57 + 166.9}{2} = 196.24℃$$

The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{EV} = \frac{273.15 + \bar{t}_{(FG)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 196.24}{273.15} \cdot 23.49 = 40.36\, m^3/s$$

The cross sectional area that the flue gas flows through:

$$A_{duct}^{EV} = H \cdot L - H \cdot D_{tube}^{EV} \cdot n_{tube/r}^{EV} - H \cdot 2 \cdot h_{fin}^{EV} \cdot th_{fin}^{EV} \cdot n_{fin}^{EV} \cdot n_{tube/r}^{EV}$$

$$A_{duct}^{EV} = 6.78 \cdot 1.76 - 6.78 \cdot 0.057 \cdot 17 - 6.78 \cdot 2 \cdot 0.019 \cdot 0.001 \cdot 230 \cdot 17 = 4.36 m^2$$

Speed of the flue gas passing through EV_{HP}:

$$W_{Flue}^{EV} = \frac{\dot{M}_{VFlue}^{EV}}{A_{duct}^{EV}} = \frac{40.36}{4.36} = 9.27\ m/s$$

The speed of the flue gas through the evaporator is within the suggested range, between 9 and 12 m/s.

4.6.3 Reduced heat transfer coefficient outside the tubes of EV_{LP}, through the flue gas

The coefficient of heat transfer through convection outside the finned tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{EV} = \frac{s_{1(EV)}}{D_{tube}^{EV}} = \frac{0.1005}{0.057} = 1.76$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{EV} = \frac{s'^{EV}}{D_{tube}^{EV}} = \frac{\sqrt{\left(\frac{s_{1(EV)}}{2}\right)^2 + \left(s_{2(EV)}\right)^2}}{D_{tube}^{EV}} = \frac{\sqrt{\left(\frac{0.1005}{2}\right)^2 + 0.117^2}}{0.057} = 2.23$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_{\sigma}^{EV} = \frac{\sigma_1^{EV} - 1}{\sigma_2{'}^{EV} - 1} = \frac{1.76 - 1}{2.23 - 1} = 0.62$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through EV_{LP} has been determined in previous calculations regarding the flue gas speed verification.

Temperature $\bar{t}_{(EF)} = 196.24\ ℃$

$$\lambda_{Flue}^{EV} = \left[\frac{196.24 - 100}{100} \cdot (39.23 - 30.89) + 30.89\right] \cdot 10^{-3} = 3.891 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{EV} = \left[\frac{196.24 - 100}{100} \cdot (32.36 - 21.41) + 21.41\right] \cdot 10^{-6} = 3.195 \cdot 10^{-5}\ m^2/s$$

The correction coefficient for the number of lateral row of tubes in the heat exchanger:

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 8 lateral rows of tubes in evaporator EV_{LP}.

$$c_z^{EV} = 0.99$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c^{EV} = 0.23 \cdot c_z^{EV} \cdot (\varphi_{\sigma}^{EV})^{0.2} \cdot \frac{\lambda_{Flue}^{EV}}{s_{fin}^{EV}} \cdot \left(\frac{D_{tube}^{EV}}{s_{fin}^{EV}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{EV}}{s_{fin}^{EV}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{EV} \cdot s_{fin}^{EV}}{\nu_{Flue}^{EV}}\right)^{0.65}$$

$$\alpha_c^{EV} = 0.23 \cdot 0.99 \cdot 0.62^{0.2} \cdot \frac{0.03891}{0.00435} \cdot \left(\frac{0.057}{0.00435}\right)^{-0.54} \cdot \left(\frac{0.019}{0.00435}\right)^{-0.14} \cdot \left(\frac{9.27 \cdot 0.00435}{3.195 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{EV} = 38.89\ W/(m^2K)$$

Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of $\beta^{EV} \cdot h_{fin}^{EV}$ and the quotient of $D_{fin}^{EV}/D_{tube}^{EV}$, where β^{EV} is a coefficient.

Coefficient β^{EV}:

$$\beta^{EV} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{EV}}{th_{fin}^{EV} \cdot \lambda_{fin} \cdot \left(1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{EV}\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 38.89}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 38.89)}}$$

$$\beta^{EV} = 39.03\ m^{-1}$$

Values needed to determine the coefficient characterizing the effectiveness of the fins:

$$\beta^{EV} \cdot h_{fin}^{EV} = 39.03 \cdot 0.019 = 0.74$$

$$\frac{D_{fin}^{EV}}{D_{tube}^{EV}} = \frac{0.095}{0.057} = 1.67$$

Coefficient determined using the graph:

$$E^{EV} = 0.82$$

Proportions of the tubes surface area made up of the fins, and tube wall:

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}^{EV}}{S_{out}^{EV}} = \frac{\left(\frac{D_{fin}^{EV}}{D_{tube}^{EV}}\right)^2 - 1}{\left(\frac{D_{fin}^{EV}}{D_{tube}^{EV}}\right)^2 - 1 + 2 \cdot \left(\frac{s_{fin}^{EV}}{D_{tube}^{EV}} - \frac{th_{fin}^{EV}}{D_{tube}^{EV}}\right)} = \frac{\left(\frac{0.095}{0.057}\right)^2 - 1}{\left(\frac{0.095}{0.057}\right)^2 - 1 + 2 \cdot \left(\frac{0.00435}{0.057} - \frac{0.001}{0.057}\right)}$$

$$\frac{S_{fin}^{EV}}{S_{out}^{EV}} = 0.94$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}^{EV}}{S_{out}^{EV}} = 1 - \frac{s_{fin}^{EV}}{S_{out}^{EV}} = 1 - 0.94 = 0.06$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out}^{EV} = \left(\frac{S_{fin}^{EV}}{S_{out}^{EV}} \cdot E^{EV} \cdot \mu + \frac{S_{out-fin}^{EV}}{S_{out}^{EV}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{EV}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{EV}}$$

$$\alpha_{r:out}^{EV} = (0.94 \cdot 0.82 \cdot 1 + 0.06) \cdot \frac{0.85 \cdot 38.89}{1 + 0.0043 \cdot 0.85 \cdot 38.89} = 23.97\, W/(m^2K)$$

4.6.4 Overall heat transfer coefficient for EV_{LP}

For the design of evaporators and economizers the effect that the reduced coefficient of heat transfer inside the tubes $\alpha_{r:in}$ has on the overall heat transfer coefficient k is negligible. Thus the overall heat transfer coefficient of evaporators and economizers are calculated in the same manner, as seen in equation (4.19).

$$k^{EV} = \alpha_{r:out}^{EV} = 23.97\, W/(m^2K)$$

4.6.5 Logarithmic mean temperature difference across EVLP

$$\Delta t_1^{EV} = t_F^{real} - t_2^{LP} = 225.57 - 156.15 = 69.41\ K$$

$$\Delta t_2^{EV} = t'_G - t_4^{LP} = 166.9 - 151.15 = 15.75\ K$$

$$\Delta t_{ln}^{EV} = \frac{\Delta t_2^{EV} - \Delta t_1^{EV}}{ln\left(\frac{\Delta t_2^{EV}}{\Delta t_1^{EV}}\right)} = \frac{15.75 - 69.41}{ln\left(\frac{15.75}{69.41}\right)} = 36.18\ K$$

4.6.6 The number of lateral rows of tubes required for EVLP

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{EV} = \frac{Q_{EV}^{LP}}{k^{EV} \cdot \Delta t_{ln}^{EV}} = \frac{1866.54}{23.97 \cdot 36.18} = 2152.14\ m^2$$

The outer surface area of one fin:

$$S_{1fin}^{EV} = 2 \cdot \pi \cdot \frac{\left(D_{fin}^{EV}\right)^2 - \left(D_{tube}^{EV}\right)^2}{4} + \pi \cdot D_{fin}^{EV} \cdot th_{fin}^{EV}$$

$$S_{1fin}^{EV} = 2 \cdot \pi \cdot \frac{0.095^2 - 0.057^2}{4} + \pi \cdot 0.095 \cdot 0.001 = 0.0094\ m^2$$

The outer surface area of one finned tube per meter length:

$$S_{out/1m}^{EV} = \pi \cdot D_{tube}^{EV} \cdot \left(1 - n_{fin}^{EV} \cdot th_{fin}^{EV}\right) + n_{fin}^{EV} \cdot S_{1fin}^{EV}$$

$$S_{out/1m}^{EV} = \pi \cdot 0.057 \cdot (1 - 230 \cdot 0.001) + 230 \cdot 0.0094 = 2.29\ m$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{EV} = S_{out/1m}^{EV} \cdot H \cdot n_{tube/r}^{EV} = 2.29 \cdot 6.78 \cdot 17 = 264.3\ m^2$$

The number of lateral rows of tubers in EVLP:

$$n_{row}^{EV} = \frac{S_{out}^{EV}}{S_{out/r}^{EV}} = \frac{2152.14}{264.33} = 8.14$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{EVH} = 8$$

4.6.7 Calculating the actual heat transfer rate of EV_{LP}

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{EV:real} = S_{out/r}^{EV} \cdot n_{row}^{EV} = 264.33 \cdot 17 = 2114.61\ m^2$$

The actual heat transfer rate of EV_{LP}:

$$Q_{EV}^{LP:real} = k^{EV} \cdot S_{out}^{EV:real} \cdot \Delta t_{ln}^{EV} = 23.97 \cdot 2114.61 \cdot 36.18 = 1833.98\ kW$$

Error verification:

$$\%Error^{EVH} = \left|\left(\frac{1833.98}{1866.54} - 1\right) \cdot 100\right| = 1.74\%$$

The discrepancy between the theoretical and the actual heat transfer rate of evaporator EV_{LP} is less than 2% , thus the current configuration of EV_{LP} is acceptable.

4.6.8 Determining the actual parameters of the flue gas exiting EV_{LP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of evaporator EV_{LP} the parameters of the flue gas exiting EV_{LP} at point G must be recalculated.

Actual enthalpy of the flue gas at point G:

$$I_G^{real} = I_F^{real} - \frac{Q_{EV}^{LP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 303.27 - \frac{1833.98}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 224.81\ kJ/Nm^3$$

Actual temperature of the flue gas at point G:

$$t_G^{real} = \frac{I_F^{real} - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{224.81 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 167.94\ ℃$$

4.7 Design of the High Pressure Economizer ECO_{2HP}

Several parameters are required in the calculations associated with the design of the high pressure economizer ECO_{2HP}. Parameters, other than those describing the geometry of the economizer, that are necessary for the calculations regarding the design and sizing of economizer ECO_{2HP}, some of which were determined in previous calculations, are organized in *Table 4.18* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of ECO_{2HP} (theoretical value)	Q_{ECO2}^{HP}	158.21	$[kW]$
Flue	Actual temperature of flue gas entering into ECO_{2HP}	t_G^{real}	167.94	[°C]
	Actual enthalpy of flue gas entering into ECO_{2HP}	I_G^{real}	224.81	$[kJ/Nm^3]$
water	Mass flow rate of water in the high pressure circuit (only 95% passes through ECO_{2HP})	$\dot{M}_{Steam}^{HP}$	3.24	$[kg/s]$
	Temperature of water entering ECO_{2HP}	t_8^{HP}	139.4	[°C]
	Pressure of water entering ECO_{2HP}	p_8^{HP}	5	$[MPa]$
	Temperature of water exiting ECO_{2HP}	t_7^{HP}	151.4	[°C]
	Pressure of water exiting ECO_{2HP}	p_7^{HP}	4.9	$[MPa]$

Table 4.18 Parameters necessary for economizer ECO_{2HP} design, not including parameters describing the geometry of ECO_{2HP}

4.7.1 Geometry of the high pressure economizer ECO_{2HP}

The chosen dimensions of the finned tubes used in the high pressure economizer (ECO_{2HP}) are shown in *Table 4.19* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.19* are selected and later on adjusted in order to supply the economizer with an acceptable water flow speed through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

	Parameter	Symbol	Value	Units
tube	Outer tube diameter	D_{tube}^{ECO2}	33.7	$[mm]$
	Tube wall thickness	th_{tube}^{ECO2}	4	$[mm]$
	Inner tube diameter	d_{tube}^{ECO2}	25.7	$[mm]$
fins	Fin thickness	th_{fin}^{ECO2}	1	$[mm]$
	Number of fins per meter	n_{fin}^{ECO2}	200	$[mm]$
	Fin spacing	s_{fin}^{ECO2}	5	$[mm]$
	Fin height	h_{fin}^{ECO2}	8	$[mm]$
	Outer fin diameter	D_{fin}^{ECO2}	49.7	$[mm]$

Table 4.19 Parameters selected for the finned tubes used in the high pressure economizer ECO_{2HP}

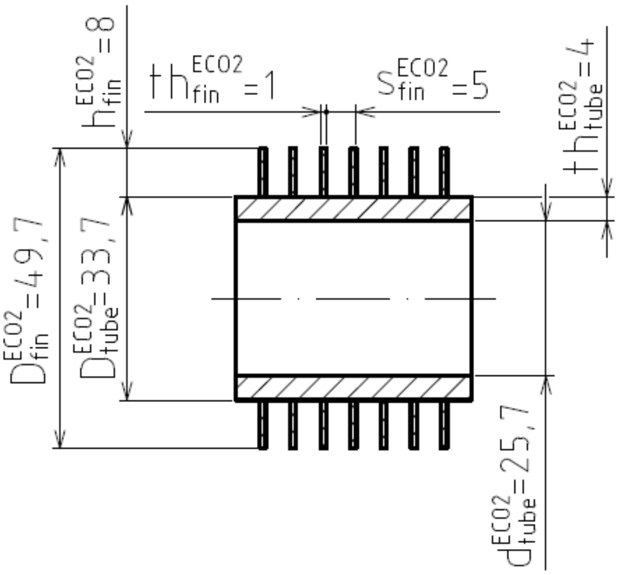

Figure 4-14 Tube geometry ECO_{2HP}

The dimensions describing the layout of the finned tubes of economizer ECO_{2HP} are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the economizer.

Chosen layout dimensions:

Lateral gap between tubes $a_{ECO2} = 4.45$ mm

Longitudinal tube spacing $s_{2(ECO2)} = 90$ mm

Lateral tube spacing in economizer ECO_{2HP}:

$$s_{1(ECO2)} = D_{tube}^{ECO2} + 2 \cdot h_{fin}^{ECO2} + a_{ECO2} = 0.0337 + 2 \cdot 0.008 + 0.00445 = 0.05415m$$

Number of tubes in each lateral row of tubes in ECO_{2HP}:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{ECO2} = \frac{L}{s_{1(ECO2)}} - 0.5 = \frac{1.76}{0.05415} - 0.5 = 32.002$$

$$n_{tube/r}^{ECO2} = 32$$

Gap verification:

There needs to be a gap between the wall of the flue gas duct and outer tubes of the heat exchanger, however it must not be excessive. This value should be no greater than 10 mm. The calculated gap is within the acceptable range.

$$gap = L - n_{tube/r}^{ECO2} \cdot s_{1(ECO2)} - \frac{D_{tube}^{ECO2}}{2} - h_{fin}^{ECO2}$$

$$gap = 1.76 - 32 \cdot 0.05415 - \frac{0.0337}{2} - 0.008 = 0.00235\,m$$

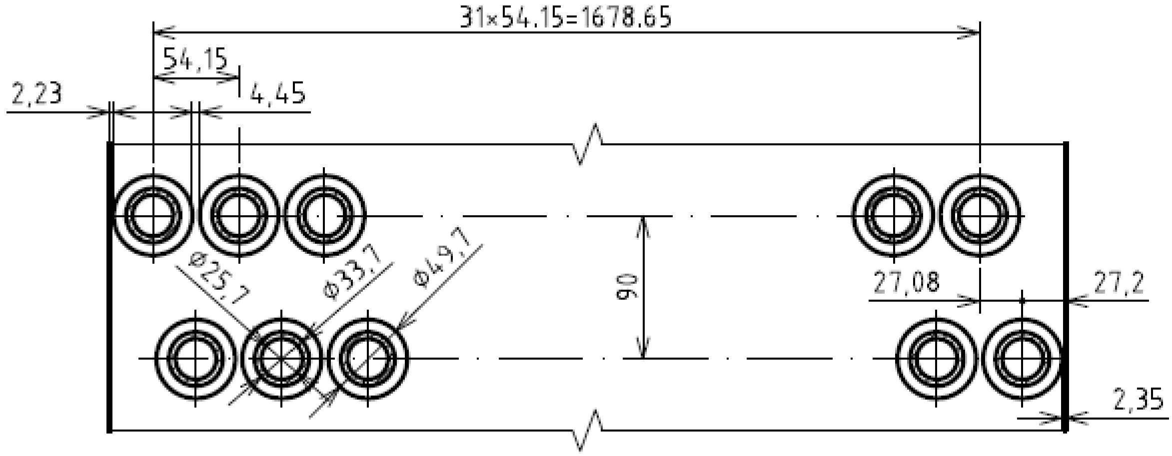

Figure 4-15 Tube layout ECO_{2HP}

4.7.2 Verification of the speed of water passing through economizer ECO_{2HP}

The speed of the water passing through economizer ECO_{2HP} is calculated according to average specific volume of the water. The specific volume of the water is determined through X-Steam according to the average pressure and temperature of the water passing through ECO_{2HP}.

$$\bar{t}_{ECO2} = \frac{t_7^{HP} + t_8^{HP}}{2} = \frac{151.4 + 139.4}{2} = 145.4\,℃$$

$$\bar{p}_{ECO2} = \frac{p_7^{HP} + p_8^{HP}}{2} = \frac{4.9 + 5}{2} = 4.95\,MPa$$

$\bar{v}_{ECO2} = 0.0011\,m^3/kg$ (determined using X-Steam, f(p,t))

Speed of water, before splitting economizer into sections:

$$W_{Steam}^{ECO2\,\prime} = \frac{0.95 \cdot \dot{M}_{Steam}^{HP} \cdot \bar{v}_{ECO2}}{\frac{\pi \cdot (d_{tube}^{ECO2})^2}{4} \cdot n_{tube/r}^{ECO2}} = \frac{0.95 \cdot 3.24 \cdot 0.0011}{\frac{\pi \cdot 0.0257^2}{4} \cdot 32} = 0.20\,m/s$$

The speed of water flowing through the economizer must be within the suggested range of 0.8 to 1.3 m/s. This is accomplished by splitting each lateral row of tubes, of the economizer, into sections to increase the speed of water. However it is recommended that the rows are not split into more than four sections. The actual speed of water flowing through the economizer is then calculated intuitively.

Number of sections $n_{section}^{ECO2} = 4$

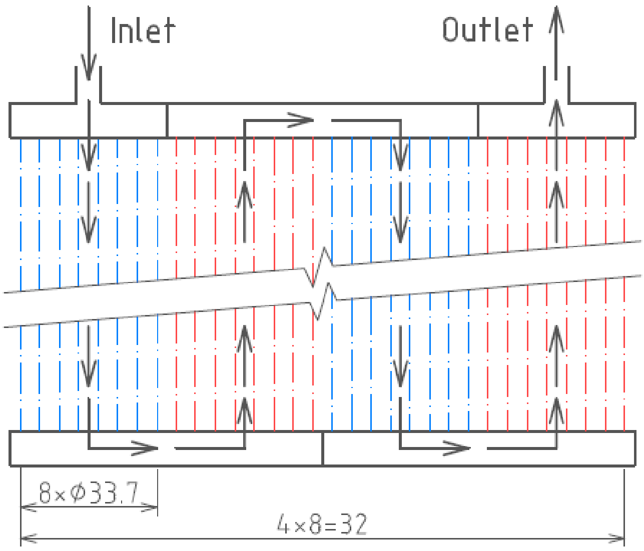

Figure 4-16 Splitting schematic ECO2HP

Actual speed of water flowing through ECO2HP:

$$W_{Steam}^{ECO2} = W_{Steam}^{ECO2'} \cdot n_{section}^{ECO2} = 0.20 \cdot 4 = 0.80 \ m/s$$

4.7.3 Verification of the speed of flue gas passing through economizer ECO2HP

Recalculated flue gas enthalpy and temperature at point H:

The parameters of the flue gas exiting the economizer at point H are recalculated according to the actual flue gas parameter entering ECO2HP.

$$I_H' = I_G^{real} - \frac{Q_{ECO2}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 224.8 - \frac{158.21}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 218.04 \, kJ/Nm^3$$

$$t_H' = \frac{I_H' - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{218.04 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 162.91℃$$

Average flue gas temperature:

$$\bar{t}_{(GH)} = \frac{t_G^{real} + t_H'}{2} = \frac{167.94 + 162.91}{2} = 165.43℃$$

The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{ECO2} = \frac{273.15 + \bar{t}_{(GH)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 165.43}{273.15} \cdot 23.49 = 37.71 \, m^3/s$$

The cross sectional area that the flue gas flows through:

$$A_{duct}^{ECO2} = H \cdot L - H \cdot D_{tube}^{ECO2} \cdot n_{tube/r}^{ECO2} - H \cdot 2 \cdot h_{fin}^{ECO2} \cdot th_{fin}^{ECO2} \cdot n_{fin}^{ECO2} \cdot n_{tube/r}^{ECO2}$$

$$A_{duct}^{ECO2} = 6.78 \cdot 1.76 - 6.78 \cdot 0.0337 \cdot 32 - 6.78 \cdot 2 \cdot 0.008 \cdot 0.001 \cdot 200 \cdot 32 = 3.93\ m^2$$

Speed of the flue gas passing through ECO_{2HP}:

$$W_{Flue}^{ECO2} = \frac{\dot{M}_{VFlue}^{ECO2}}{A_{duct}^{ECO2}} = \frac{37.71}{3.93} = 9.6\ m/s$$

The speed of the flue gas through the economizer is within the suggested range, between 9 and 12 m/s.

4.7.4 Reduced heat transfer coefficient outside the tubes of ECO_{2HP}, through the flue gas

The coefficient of heat transfer through convection outside the finned tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{ECO2} = \frac{s_{1(ECO2)}}{D_{tube}^{ECO2}} = \frac{0.05415}{0.0337} = 1.6$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2{'}^{ECO2} = \frac{s'^{ECO2}}{D_{tube}^{ECO2}} = \frac{\sqrt{\left(\frac{s_{1(ECO2)}}{2}\right)^2 + \left(s_{2(ECO2)}\right)^2}}{D_{tube}^{ECO2}} = \frac{\sqrt{\left(\frac{0.05415}{2}\right)^2 + 0.09^2}}{0.0337} = 2.79$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_\sigma^{ECO2} = \frac{\sigma_1^{ECO2} - 1}{\sigma_2{'}^{ECO2} - 1} = \frac{1.6 - 1}{2.79 - 1} = 0.34$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through ECO_{2HP} has been determined in previous calculations regarding the flue gas speed verification.

Temperature $\quad \bar{t}_{(GH)} = 165.43\ ℃$

$$\lambda_{Flue}^{ECO2} = \left[\frac{165.43 - 100}{100} \cdot (39.228 - 30.89) + 30.89\right] \cdot 10^{-3} = 3.635 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{ECO2} = \left[\frac{165.43 - 100}{100} \cdot (32.36 - 21.41) + 21.41\right] \cdot 10^{-6} = 2.857 \cdot 10^{-5}\ m^2/s$$

The correction coefficient for the number of lateral row of tubes in the heat exchanger:

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 2 lateral rows of tubes in economizer ECO_{2HP}.

$$c_z^{ECO2} = 0.87$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c^{ECO2} = 0.23 \cdot c_z^{ECO2} \cdot (\varphi_\sigma^{ECO2})^{0.2} \cdot \frac{\lambda_{Flue}^{ECO2}}{s_{fin}^{ECO2}} \cdot \left(\frac{D_{tube}^{ECO2}}{s_{fin}^{ECO2}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{ECO2}}{s_{fin}^{ECO2}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{ECO2} \cdot s_{fin}^{ECO2}}{\nu_{Flue}^{ECO2}}\right)^{0.65}$$

$$\alpha_c^{ECO2} = 0.23 \cdot 0.87 \cdot 0.34^{0.2} \cdot \frac{0.03635}{0.005} \cdot \left(\frac{0.0337}{0.005}\right)^{-0.54} \cdot \left(\frac{0.008}{0.005}\right)^{-0.14} \cdot \left(\frac{9.6 \cdot 0.005}{2.857 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{ECO2} = 48.81\ W/(m^2K)$$

Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of $\beta^{ECO2} \cdot h_{fin}^{ECO2}$ and the quotient of $D_{fin}^{ECO2}/D_{tube}^{ECO2}$, where β^{ECO2} is a coefficient.

Coefficient β^{ECO2}:

$$\beta^{ECO2} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{ECO2}}{th_{fin}^{ECO2} \cdot \lambda_{fin} \cdot \left(1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO2}\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 48.81}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 48.81)}}$$

$$\beta^{ECO2} = 43.05\ m^{-1}$$

Values needed to determine the coefficient characterizing the effectiveness of the fins:

$$\beta^{ECO2} \cdot h_{fin}^{ECO2} = 43.05 \cdot 0.008 = 0.34$$

$$\frac{D_{fin}^{ECO2}}{D_{tube}^{ECO2}} = \frac{0.0497}{0.0337} = 1.47$$

Coefficient determined using the graph:

$$E^{ECO2} = 0.95$$

Proportions of the tubes surface area made up of the fins, and tube wall:

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}^{ECO2}}{S_{out}^{ECO2}}=\frac{\left(\frac{D_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)^2-1}{\left(\frac{D_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)^2-1+2\cdot\left(\frac{s_{fin}^{ECO2}}{D_{tube}^{ECO2}}-\frac{th_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)}=\frac{\left(\frac{0.0497}{0.0337}\right)^2-1}{\left(\frac{0.0497}{0.0337}\right)^2-1+2\cdot\left(\frac{0.005}{0.0337}-\frac{0.001}{0.0337}\right)}$$

$$\frac{S_{fin}^{ECO2}}{S_{out}^{ECO2}}=0.83$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}^{ECO2}}{S_{out}^{ECO2}}=1-\frac{s_{fin}^{ECO2}}{S_{out}^{ECO2}}=1-0.83=0.17$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out}^{ECO2}=\left(\frac{S_{fin}^{ECO2}}{S_{out}^{ECO2}}\cdot E^{ECO2}\cdot\mu+\frac{S_{out-fin}^{ECO2}}{S_{out}^{ECO2}}\right)\cdot\frac{\Psi_{fin}\cdot\alpha_c^{ECO2}}{1+\varepsilon\cdot\Psi_{fin}\cdot\alpha_c^{ECO2}}$$

$$\alpha_{r:out}^{ECO2}=(0.83\cdot 0.95\cdot 1+0.17)\cdot\frac{0.85\cdot 48.81}{1+0.0043\cdot 0.85\cdot 48.81}=33.8\,W/(m^2K)$$

4.7.5 Overall heat transfer coefficient for ECO_{2HP}

For the design of evaporators and economizers the effect that the reduced coefficient of heat transfer inside the tubes $\alpha_{r:in}$ has on the overall heat transfer coefficient k is negligible. Thus the overall heat transfer coefficient of evaporators and economizers are calculated in the same manner, as seen in equation (4.19).

$$k^{ECO2}=\alpha_{r:out}^{ECO2}=33.8\,W/(m^2K)$$

4.7.6 Logarithmic mean temperature difference across ECO_{2HP}

$$\Delta t_1^{ECO2}=t_G^{real}-t_7^{HP}=167.94-151.4=16.53\,K$$

$$\Delta t_2^{ECO2}=t_H'-t_8^{HP}=162.91-139.4=23.51\,K$$

$$\Delta t_{ln}^{ECO2}=\frac{\Delta t_2^{ECO2}-\Delta t_1^{ECO2}}{ln\left(\frac{\Delta t_2^{ECO2}}{\Delta t_1^{ECO2}}\right)}=\frac{23.51-16.53}{ln\left(\frac{23.51}{16.53}\right)}=19.82\,K$$

4.7.7 The number of lateral rows of tubes required for ECO2HP

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{ECO2} = \frac{Q_{ECO2}^{HP}}{k^{ECO2} \cdot \Delta t_{ln}^{ECO2}} = \frac{158.21}{33.8 \cdot 19.82} = 236.18\ m^2$$

The outer surface area of one fin:

$$S_{1fin}^{ECO2} = 2 \cdot \pi \cdot \frac{\left(D_{fin}^{ECO2}\right)^2 - \left(D_{tube}^{ECO2}\right)^2}{4} + \pi \cdot D_{fin}^{ECO2} \cdot th_{fin}^{ECO2}$$

$$S_{1fin}^{ECO2} = 2 \cdot \pi \cdot \frac{0.0497^2 - 0.0337^2}{4} + \pi \cdot 0.0497 \cdot 0.001 = 0.0023\ m^2$$

The outer surface area of one finned tube per meter length:

$$S_{out/1m}^{ECO2} = \pi \cdot D_{tube}^{ECO2} \cdot \left(1 - n_{fin}^{ECO2} \cdot th_{fin}^{ECO2}\right) + n_{fin}^{ECO2} \cdot S_{1fin}^{ECO2}$$

$$S_{out/1m}^{ECO2} = \pi \cdot 0.0337 \cdot (1 - 200 \cdot 0.001) + 200 \cdot 0.0023 = 0.5351\ m$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{ECO2} = S_{out/1m}^{ECO2} \cdot H \cdot n_{tube/r}^{ECO2} = 0.5351 \cdot 6.78 \cdot 32 = 116.1\ m^2$$

The number of lateral rows of tubers in ECO2HP:

$$n_{row}^{ECO2} = \frac{S_{out}^{ECO2}}{S_{out/r}^{ECO2}} = \frac{236.18}{116.1} = 2.03$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{ECO2} = 2$$

4.7.8 Calculating the actual heat transfer rate of ECO2HP

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{ECO2:real} = S_{out/r}^{ECO2} \cdot n_{row}^{ECO2} = 116.1 \cdot 2 = 232.21\ m^2$$

The actual heat transfer rate of ECO2HP:

$$Q_{ECO2}^{HP:real} = k^{ECO2} \cdot S_{out}^{ECO2:real} \cdot \Delta t_{ln}^{ECO2} = 33.8 \cdot 232.21 \cdot 19.82 = 155.55\ kW$$

Error verification:

$$\%Error^{ECO2} = \left|\left(\frac{155.55}{158.21} - 1\right) \cdot 100\right| = 1.68\%$$

The discrepancy between the theoretical and the actual heat transfer rate of economizer ECO_{2HP} is less than 2%, thus the current configuration of ECO_{2HP} is acceptable.

4.7.9 Determining the actual parameters of the flue gas exiting ECO_{2HP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of economizer ECO_{2HP} the parameters of the flue gas exiting ECO_{2HP} at point H must be recalculated.

Actual enthalpy of the flue gas at point H:

$$I_H^{real} = I_G^{real} - \frac{Q_{ECO2}^{HP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 224.81 - \frac{155.55}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 218.15\, kJ/Nm^3$$

Actual temperature of the flue gas at point H:

$$t_H^{real} = \frac{I_H^{real} - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{218.15 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 163\ ℃$$

4.8 Design of the Low Pressure Economizer ECO_{LP}

Several parameters are required in the calculations associated with the design of the low pressure economizer ECO_{LP}. Parameters, other than those describing the geometry of the economizer, that are necessary for the calculations regarding the design and sizing of economizer ECO_{LP}, some of which were determined in previous calculations, are organized in *Table 4.20* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of ECO_{LP} (theoretical value)	Q_{ECO}^{LP}	365.82	$[kW]$
Flue	Actual temperature of flue gas entering into ECO_{LP}	t_H^{real}	163	[℃]
Flue	Actual enthalpy of flue gas entering into ECO_{LP}	I_H^{real}	218.15	$[kJ/Nm^3]$
water	Mass flow rate of water in the high pressure circuit	$\dot{M}_{Steam}^{LP}$	0.88	$[kg/s]$
water	Temperature of water entering ECO_{LP}	t_5^{LP}	53	[℃]
water	Pressure of water entering ECO_{LP}	p_5^{LP}	0.86	$[MPa]$
water	Temperature of water exiting ECO_{LP}	t_4^{LP}	151.15	[℃]
water	Pressure of water exiting ECO_{LP}	p_4^{LP}	0.56	$[MPa]$

Table 4.20 Parameters necessary for economizer ECO_{LP} design, not including parameters describing the geometry of ECO_{LP}

4.8.1 Geometry of the low pressure economizer ECO_{LP}

The chosen dimensions of the finned tubes used in the low pressure economizer (ECO_{LP}) are shown in *Table 4.21* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.21* are selected and later on adjusted in order to supply the economizer with an acceptable water flow speed through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

	Parameter	Symbol	Value	Units
tube	Outer tube diameter	D_{tube}^{ECO}	22	$[mm]$
	Tube wall thickness	th_{tube}^{ECO}	4	$[mm]$
	Inner tube diameter	d_{tube}^{ECO}	14	$[mm]$
fins	Fin thickness	th_{fin}^{ECO}	1	$[mm]$
	Number of fins per meter	n_{fin}^{ECO}	230	$[mm]$
	Fin spacing	s_{fin}^{ECO}	4.35	$[mm]$
	Fin height	h_{fin}^{ECO}	15	$[mm]$
	Outer fin diameter	D_{fin}^{ECO}	52	$[mm]$

Table 4.21 Parameters selected for the finned tubes used in the low pressure economizer ECO_{LP}

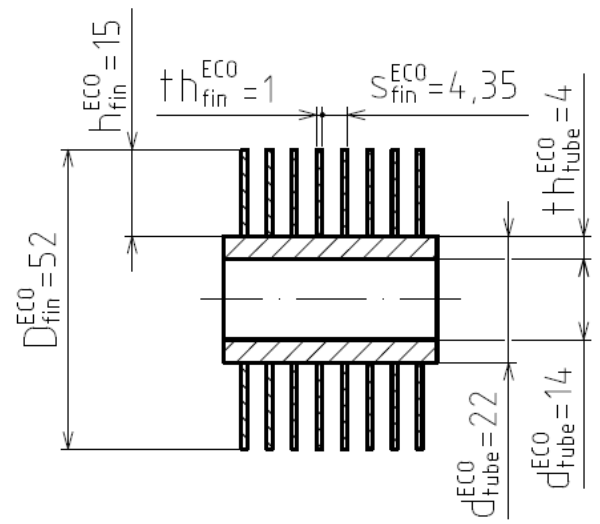

Figure 4-17 Tube geometry ECO_{LP}

The dimensions describing the layout of the finned tubes of economizer ECO_{LP} are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the economizer.

Chosen layout dimensions:

Lateral gap between tubes $a_{ECO} = 9.75$ mm

Longitudinal tube spacing $s_{2(ECO)} = 90$ mm

Lateral tube spacing in economizer ECO$_{LP}$:

$$s_{1(ECO)} = D_{tube}^{ECO} + 2 \cdot h_{fin}^{ECO} + a_{ECO} = 0.022 + 2 \cdot 0.015 + 0.00975 = 0.06175m$$

Number of tubes in each lateral row of tubes in ECO$_{LP}$:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{ECO} = \frac{L}{s_{1(ECO)}} - 0.5 = \frac{1.76}{0.06175} - 0.5 = 28.002$$

$$n_{tube/r}^{ECO} = 28$$

Gap verification:

There needs to be a gap between the wall of the flue gas duct and outer tubes of the heat exchanger, however it must not be excessive. This value should be no greater than 10 mm. The calculated gap is within the acceptable range.

$$gap = L - n_{tube/r}^{ECO} \cdot s_{1(ECO)} - \frac{D_{tube}^{ECO}}{2} - h_{fin}^{ECO}$$

$$gap = 1.76 - 28 \cdot 0.06175 - \frac{0.022}{2} - 0.015 = 0.005\ m$$

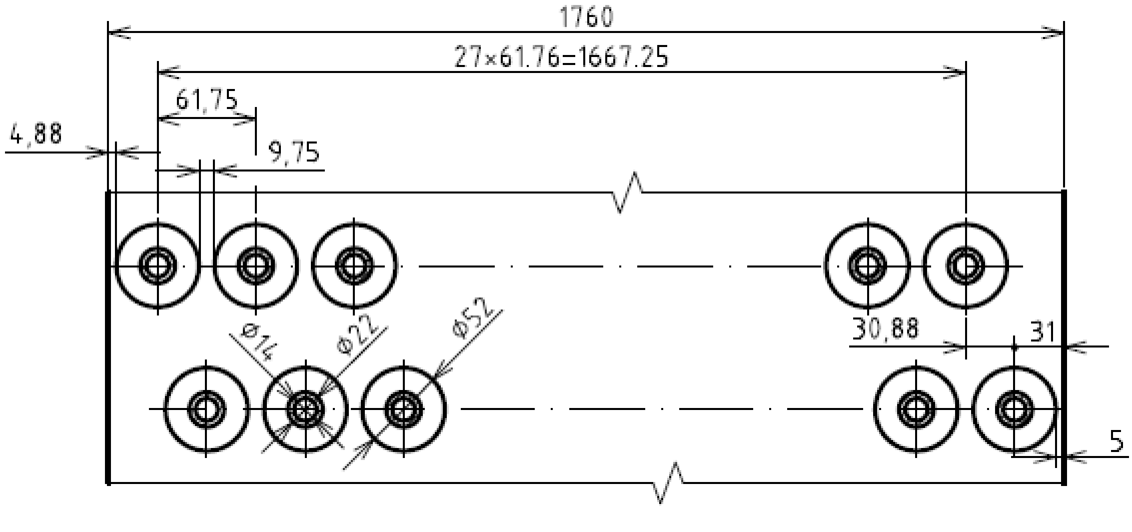

Figure 4-18 Tube layout ECO$_{LP}$

4.8.2 Verification of the speed of water passing through economizer ECO$_{LP}$

The speed of the water passing through economizer ECO$_{LP}$ is calculated according to average specific volume of the water. The specific volume of the water is determined through X-Steam according to the average pressure and temperature of the water passing through ECO$_{LP}$.

$$\bar{t}_{ECO} = \frac{t_4^{LP} + t_5^{LP}}{2} = \frac{151.15 + 53}{2} = 102.08\ ℃$$

$$\bar{p}_{ECO} = \frac{p_4^{LP} + p_5^{LP}}{2} = \frac{0.56 + 0.86}{2} = 0.71\,MPa$$

$\bar{v}_{ECO} = 0.001\,m^3/kg$ (determined using X-Steam, f(p,t))

Speed of water, before splitting economizer into sections:

$$W_{Steam}^{ECO}{}' = \frac{\dot{M}_{Steam}^{LP} \cdot \bar{v}_{ECO}}{\frac{\pi \cdot (d_{tube}^{ECO})^2}{4} \cdot n_{tube/r}^{ECO}} = \frac{0.88 \cdot 0.001}{\frac{\pi \cdot 0.014^2}{4} \cdot 28} = 0.203\,m/s$$

The speed of water flowing through the economizer must be within the suggested range of 0.8 to 1.3 m/s. This is accomplished by splitting each lateral row of tubes, of the economizer, into sections to increase the speed of water. However, it is recommended that the rows are not split into more than four sections. The actual speed of water flowing through the economizer is then calculated intuitively.

Number of sections $n_{section}^{ECO} = 4$

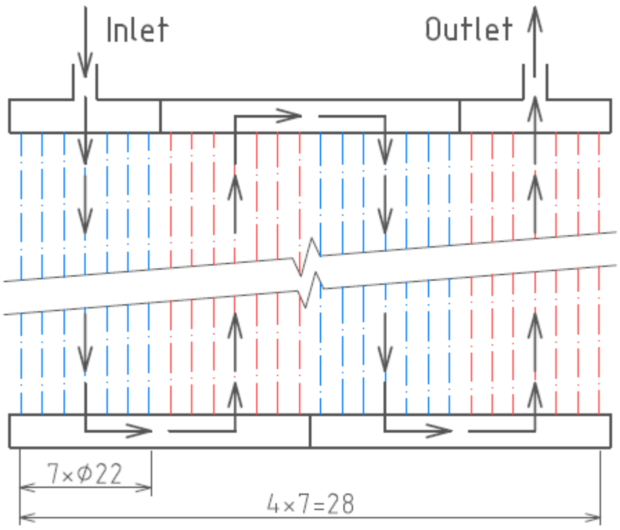

Figure 4-19 Splitting schematic ECO$_{LP}$

Actual speed of water flowing through ECO$_{LP}$:

$$W_{Steam}^{ECO} = W_{Steam}^{ECO}{}' \cdot n_{section}^{ECO} = 0.203 \cdot 4 = 0.81\ m/s$$

4.8.3 Verification of the speed of flue gas passing through economizer ECO$_{LP}$

Recalculated flue gas enthalpy and temperature at point I:

The parameters of the flue gas exiting the economizer at point I are recalculated according to the actual flue gas parameter entering ECO$_{LP}$.

$$I_I' = I_H^{real} - \frac{Q_{ECO}^{LP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 218.15 - \frac{365.82}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 202.5\, kJ/Nm^3$$

$$t_I' = \frac{I_I' - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{202.5 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 151.38℃$$

Average flue gas temperature:

$$\bar{t}_{(HI)} = \frac{t_H^{real} + t_I'}{2} = \frac{163 + 151.38}{2} = 157.19℃$$

The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{ECO} = \frac{273.15 + \bar{t}_{(HI)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 157.19}{273.15} \cdot 23.49 = 37.01\, m^3/s$$

The cross sectional area that the flue gas flows through:

$$A_{duct}^{ECO} = H \cdot L - H \cdot D_{tube}^{ECO} \cdot n_{tube/r}^{ECO} - H \cdot 2 \cdot h_{fin}^{ECO} \cdot th_{fin}^{ECO} \cdot n_{fin}^{ECO} \cdot n_{tube/r}^{ECO}$$

$$A_{duct}^{ECO} = 6.78 \cdot 1.76 - 6.78 \cdot 0.022 \cdot 28 - 6.78 \cdot 2 \cdot 0.015 \cdot 0.001 \cdot 230 \cdot 28 = 6.45\, m^2$$

Speed of the flue gas passing through ECO_{LP}:

$$W_{Flue}^{ECO} = \frac{\dot{M}_{VFlue}^{ECO}}{A_{duct}^{ECO}} = \frac{37.01}{6.45} = 5.74\ m/s$$

4.8.4 Reduced heat transfer coefficient outside the tubes of ECO_{LP}, through the flue gas

The coefficient of heat transfer through convection outside the finned tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{ECO} = \frac{s_{1(ECO)}}{D_{tube}^{ECO}} = \frac{0.06175}{0.022} = 2.81$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{ECO} = \frac{s'^{ECO}}{D_{tube}^{ECO}} = \frac{\sqrt{\left(\frac{s_{1(ECO)}}{2}\right)^2 + \left(s_{2(ECO)}\right)^2}}{D_{tube}^{ECO}} = \frac{\sqrt{\left(\frac{0.06175}{2}\right)^2 + 0.09^2}}{0.022} = 4.32$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_\sigma^{ECO} = \frac{\sigma_1^{ECO} - 1}{\sigma_2'^{ECO} - 1} = \frac{2.81 - 1}{4.32 - 1} = 0.54$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through ECO_{LP} has been determined in previous calculations regarding the flue gas speed verification.

Temperature $\bar{t}_{(HI)} = 157.19\ ℃$

$$\lambda_{Flue}^{ECO} = \left[\frac{157.19 - 100}{100} \cdot (39.228 - 30.89) + 30.89\right] \cdot 10^{-3} = 3.566 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{ECO} = \left[\frac{157.19 - 100}{100} \cdot (32.36 - 21.41) + 21.41\right] \cdot 10^{-6} = 2.767 \cdot 10^{-5}\ m^2/s$$

The correction coefficient for the number of lateral row of tubes in the heat exchanger:

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 2 lateral rows of tubes in economizer ECO_{LP}.

$$c_z^{ECO} = 0.87$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c^{ECO} = 0.23 \cdot c_z^{ECO} \cdot (\varphi_\sigma^{ECO})^{0.2} \cdot \frac{\lambda_{Flue}^{ECO}}{s_{fin}^{ECO}} \cdot \left(\frac{D_{tube}^{ECO}}{s_{fin}^{ECO}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{ECO}}{s_{fin}^{ECO}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{ECO} \cdot s_{fin}^{ECO}}{\nu_{Flue}^{ECO}}\right)^{0.65}$$

$$\alpha_c^{ECO} = 0.23 \cdot 0.87 \cdot 0.54^{0.2} \cdot \frac{0.03566}{0.00435} \cdot \left(\frac{0.022}{0.00435}\right)^{-0.54} \cdot \left(\frac{0.0015}{0.00435}\right)^{-0.14} \cdot \left(\frac{5.74 \cdot 0.00435}{2.767 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{ECO} = 42.41\ W/(m^2K)$$

Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of $\beta^{ECO} \cdot h_{fin}^{ECO}$ and the quotient of $D_{fin}^{ECO}/D_{tube}^{ECO}$, where β^{ECO} is a coefficient.

Coefficient β^{ECO}:

$$\beta^{ECO} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{ECO}}{th_{fin}^{ECO} \cdot \lambda_{fin} \cdot \left(1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO}\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 42.41}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 42.41)}}$$

$$\beta^{ECO} = 40.53\ m^{-1}$$

Values needed to determine the coefficient characterizing the effectiveness of the fins:

$$\beta^{ECO} \cdot h_{fin}^{ECO} = 40.53 \cdot 0.015 = 0.61$$

$$\frac{D_{fin}^{ECO}}{D_{tube}^{ECO}} = \frac{0.052}{0.022} = 2.36$$

Coefficient determined using the graph:

$$E^{ECO} = 0.85$$

Proportions of the tubes surface area made up of the fins, and tube wall:

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}^{ECO}}{S_{out}^{ECO}} = \frac{\left(\frac{D_{fin}^{ECO}}{D_{tube}^{ECO}}\right)^2 - 1}{\left(\frac{D_{fin}^{ECO2}}{D_{tube}^{ECO}}\right)^2 - 1 + 2 \cdot \left(\frac{s_{fin}^{ECO}}{D_{tube}^{ECO}} - \frac{th_{fin}^{ECO}}{D_{tube}^{ECO}}\right)} = \frac{\left(\frac{0.052}{0.022}\right)^2 - 1}{\left(\frac{0.052}{0.022}\right)^2 - 1 + 2 \cdot \left(\frac{0.00435}{0.022} - \frac{0.001}{0.022}\right)}$$

$$\frac{S_{fin}^{ECO}}{S_{out}^{ECO}} = 0.94$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}^{ECO}}{S_{out}^{ECO}} = 1 - \frac{s_{fin}^{ECO}}{S_{out}^{ECO}} = 1 - 0.94 = 0.06$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out}^{ECO} = \left(\frac{S_{fin}^{ECO}}{S_{out}^{ECO}} \cdot E^{ECO} \cdot \mu + \frac{S_{out-fin}^{ECO}}{S_{out}^{ECO}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{ECO}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO}}$$

$$\alpha_{r:out}^{ECO} = (0.94 \cdot 0.85 \cdot 1 + 0.06) \cdot \frac{0.85 \cdot 42.41}{1 + 0.0043 \cdot 0.85 \cdot 42.41} = 26.68\, W/(m^2K)$$

4.8.5 Overall heat transfer coefficient for ECO_{LP}

For the design of evaporators and economizers the effect that the reduced coefficient of heat transfer inside the tubes $\alpha_{r:in}$ has on the overall heat transfer coefficient k is negligible. Thus the overall heat transfer coefficient of evaporators and economizers are calculated in the same manner, as seen in equation (4.19).

$$k^{ECO} = \alpha_{r:out}^{ECO} = 26.68\, W/(m^2K)$$

4.8.6 Logarithmic mean temperature difference across ECO_{LP}

$$\Delta t_1^{ECO} = t_H^{real} - t_4^{LP} = 163 - 151.15 = 11.83\ K$$

$$\Delta t_2^{ECO} = t_I' - t_5^{LP} = 151.38 - 53 = 98.38\ K$$

$$\Delta t_{ln}^{ECO} = \frac{\Delta t_2^{ECO} - \Delta t_1^{ECO}}{ln\left(\frac{\Delta t_2^{ECO}}{\Delta t_1^{ECO}}\right)} = \frac{98.38 - 11.83}{ln\left(\frac{98.38}{11.83}\right)} = 40.88\ K$$

4.8.7 The number of lateral rows of tubes required for ECO_{LP}

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{ECO} = \frac{Q_{ECO}^{HP}}{k^{ECO} \cdot \Delta t_{ln}^{ECO}} = \frac{365.82}{26.68 \cdot 40.88} = 335.48\ m^2$$

The outer surface area of one fin:

$$S_{1fin}^{ECO} = 2 \cdot \pi \cdot \frac{\left(D_{fin}^{ECO}\right)^2 - \left(D_{tube}^{ECO}\right)^2}{4} + \pi \cdot D_{fin}^{ECO} \cdot th_{fin}^{ECO}$$

$$S_{1fin}^{ECO} = 2 \cdot \pi \cdot \frac{0.052^2 - 0.022^2}{4} + \pi \cdot 0.052 \cdot 0.001 = 0.0037\ m^2$$

The outer surface area of one finned tube per meter length:

$$S_{out/1m}^{ECO} = \pi \cdot D_{tube}^{ECO} \cdot \left(1 - n_{fin}^{ECO} \cdot th_{fin}^{ECO}\right) + n_{fin}^{ECO} \cdot S_{1fin}^{ECO}$$

$$S_{out/1m}^{ECO} = \pi \cdot 0.022 \cdot (1 - 230 \cdot 0.001) + 230 \cdot 0.0037 = 0.8928\ m$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{ECO} = S_{out/1m}^{ECO} \cdot H \cdot n_{tube/r}^{ECO} = 0.8928 \cdot 6.78 \cdot 28 = 169.5\ m^2$$

The number of lateral rows of tubers in ECO$_{3HP}$:

$$n_{row}^{ECO} = \frac{S_{out}^{ECO2}}{S_{out/r}^{ECO2}} = \frac{335.48}{169.5} = 1.98$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{ECO} = 2$$

4.8.8 Calculating the actual heat transfer rate of ECO_{LP}

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{ECO:real} = S_{out/r}^{ECO} \cdot n_{row}^{ECO} = 169.5 \cdot 2 = 338.99\ m^2$$

The actual heat transfer rate of ECO_{2HP}:

$$Q_{ECO}^{LP:real} = k^{ECO} \cdot S_{out}^{ECO:real} \cdot \Delta t_{ln}^{ECO} = 26.68 \cdot 338.99 \cdot 40.88 = 369.65\ kW$$

Error verification:

$$\%Error^{ECO} = \left| \left(\frac{369.65}{365.82} - 1 \right) \cdot 100 \right| = 1.05\%$$

The discrepancy between the theoretical and the actual heat transfer rate of economizer ECO_{LP} is less than 2%, thus the current configuration of ECO_{LP} is acceptable.

4.8.9 Determining the actual parameters of the flue gas exiting ECO_{LP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of economizer ECO_{LP} the parameters of the flue gas exiting ECO_{LP} at point I must be recalculated.

Actual enthalpy of the flue gas at point I:

$$I_I^{real} = I_H^{real} - \frac{Q_{ECO}^{LP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 218.15 - \frac{369.65}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 202.34\ kJ/Nm^3$$

Actual temperature of the flue gas at point I:

$$t_I^{real} = \frac{I_I^{real} - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{202.34 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 151.26℃$$

4.9 Design of the High Pressure Economizer ECO_{1HP}

Several parameters are required in the calculations associated with the design of the high pressure economizer ECO_{1HP}. Parameters, other than those describing the geometry of the economizer, that are necessary for the calculations regarding the design and sizing of economizer ECO_{1HP}, some of which were determined in previous calculations, are organized in *Table 4.22* below.

	Parameters	Symbol	Value	Units
	Heat transfer rate required of ECO_{1HP} (theoretical value)	Q_{ECO1}^{HP}	1119.61	$[kW]$
Flue	Actual temperature of flue gas entering into ECO_{1HP}	t_I^{real}	151.26	[°C]
	Actual enthalpy of flue gas entering into ECO_{1HP}	I_I^{real}	202.34	$[kJ/Nm^3]$
water	Mass flow rate of water in the high pressure circuit (only 95% passes through ECO_{1HP})	$\dot{M}_{Steam}^{HP}$	3.24	$[kg/s]$
	Temperature of water entering ECO_{1HP}	t_9^{HP}	53	[°C]
	Pressure of water entering ECO_{1HP}	p_9^{HP}	5.1	$[MPa]$
	Temperature of water exiting ECO_{1HP}	t_8^{HP}	139.4	[°C]
	Pressure of water exiting ECO_{1HP}	p_8^{HP}	5	$[MPa]$

Table 4.22 Parameters necessary for economizer ECO_{1HP} design, not including parameters describing the geometry of ECO_{1HP}

4.9.1 Geometry of the high pressure economizer ECO_{1HP}

The chosen dimensions of the finned tubes used in the high pressure economizer (ECO_{1HP}) are shown in *Table 4.23* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.23* are selected and later on adjusted in order to supply the economizer with an acceptable water flow speed through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

	Parameter	Symbol	Value	Units
tube	Outer tube diameter	D_{tube}^{ECO1}	33.7	$[mm]$
	Tube wall thickness	th_{tube}^{ECO1}	4	$[mm]$
	Inner tube diameter	d_{tube}^{ECO1}	25.7	$[mm]$
fins	Fin thickness	th_{fin}^{ECO1}	1	$[mm]$
	Number of fins per meter	n_{fin}^{ECO1}	220	$[mm]$
	Fin spacing	s_{fin}^{ECO1}	4.55	$[mm]$
	Fin height	h_{fin}^{ECO1}	12	$[mm]$
	Outer fin diameter	D_{fin}^{ECO1}	57.7	$[mm]$

Table 4.23 Parameters selected for the finned tubes used in the high pressure economizer ECO_{1HP}

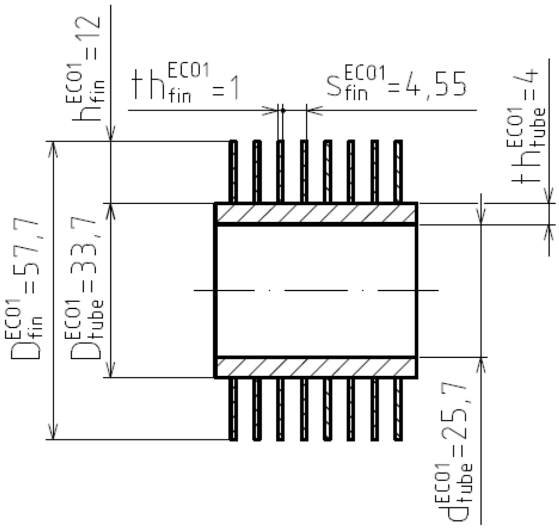

Figure 4-20 Tube geometry ECO_{1HP}

The dimensions describing the layout of the finned tubes of economizer ECO_{1HP} are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the economizer.

Chosen layout dimensions:

Lateral gap between tubes	$a_{ECO1} = 4$ mm
Longitudinal tube spacing	$s_{2(ECO1)} = 90$ mm

Lateral tube spacing in economizer ECO_{2HP}:

$$s_{1(ECO1)} = D_{tube}^{ECO1} + 2 \cdot h_{fin}^{ECO1} + a_{ECO1} = 0.0337 + 2 \cdot 0.012 + 0.004 = 0.0617m$$

Number of tubes in each lateral row of tubes in ECO_{1HP}:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{ECO1} = \frac{L}{s_{1(ECO1)}} - 0.5 = \frac{1.76}{0.0617} - 0.5 = 28.03$$

$$n_{tube/r}^{ECO1} = 28$$

Gap verification:

There needs to be a gap between the wall of the flue gas duct and outer tubes of the heat exchanger, however it must not be excessive. This value should be no greater than 10 mm. The calculated gap is within the acceptable range.

$$gap = L - n_{tube/r}^{ECO1} \cdot s_{1(ECO1)} - \frac{D_{tube}^{ECO1}}{2} - h_{fin}^{ECO1}$$

$$gap = 1.76 - 28 \cdot 0.0617 - \frac{0.0337}{2} - 0.012 = 0.00355\ m$$

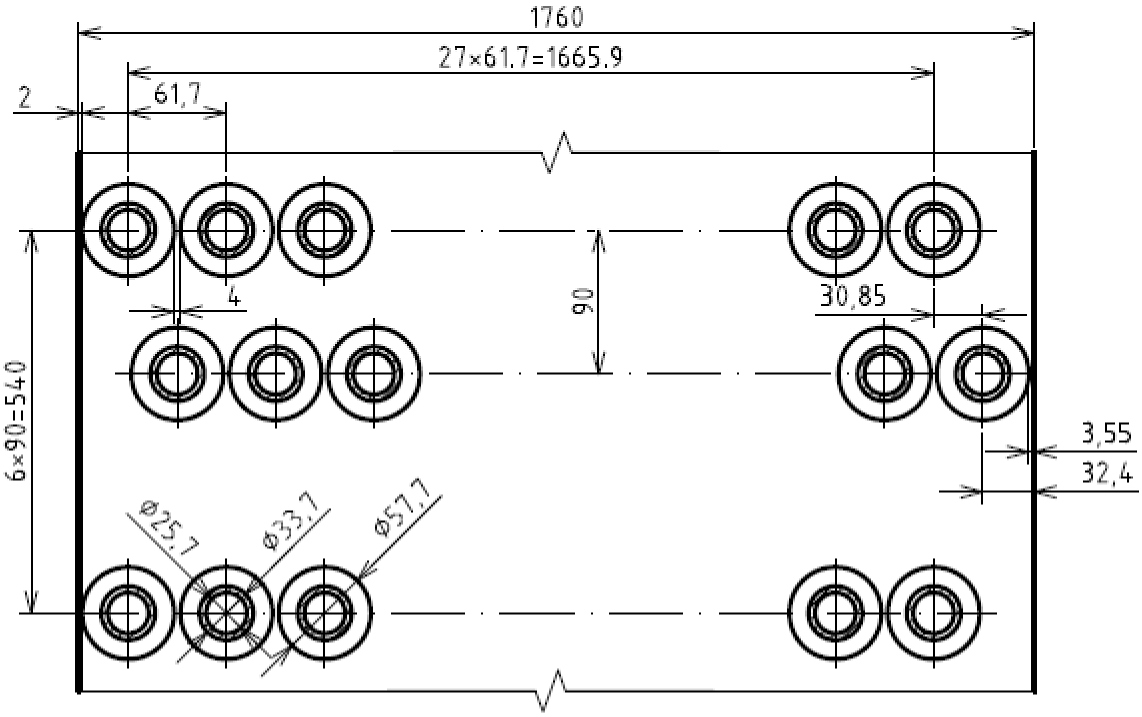

Figure 4-21 Tube layout ECO_{1HP}

4.9.2 Verification of the speed of water passing through economizer ECO_{1HP}

The speed of the water passing through economizer ECO_{1HP} is calculated according to average specific volume of the water. The specific volume of the water is determined through X-Steam according to the average pressure and temperature of the water passing through ECO_{1HP}.

$$\bar{t}_{ECO1} = \frac{t_8^{HP} + t_9^{HP}}{2} = \frac{139.4 + 53}{2} = 96.2\ ℃$$

$$\bar{p}_{ECO1} = \frac{p_8^{HP} + p_9^{HP}}{2} = \frac{5 + 5.1}{2} = 5.05\ MPa$$

$\bar{v}_{ECO1} = 0.001\, m^3/kg$ (determined using X-Steam, f(p,t))

Speed of water, before splitting economizer into sections:

$$W_{Steam}^{ECO1}{}' = \frac{0.95 \cdot \dot{M}_{Steam}^{HP} \cdot \bar{v}_{ECO1}}{\frac{\pi \cdot (d_{tube}^{ECO1})^2}{4} \cdot n_{tube/r}^{ECO1}} = \frac{0.95 \cdot 3.24 \cdot 0.001}{\frac{\pi \cdot 0.0257^2}{4} \cdot 28} = 0.22\, m/s$$

The speed of water flowing through the economizer must be within the suggested range of 0.8 to 1.3 m/s. This is accomplished by splitting each lateral row of tubes, of the economizer, into sections to increase the speed of water. However, it is recommended that the rows are not split into more than four sections. The actual speed of water flowing through the economizer is then calculated intuitively.

Number of sections $n_{section}^{ECO1} = 4$

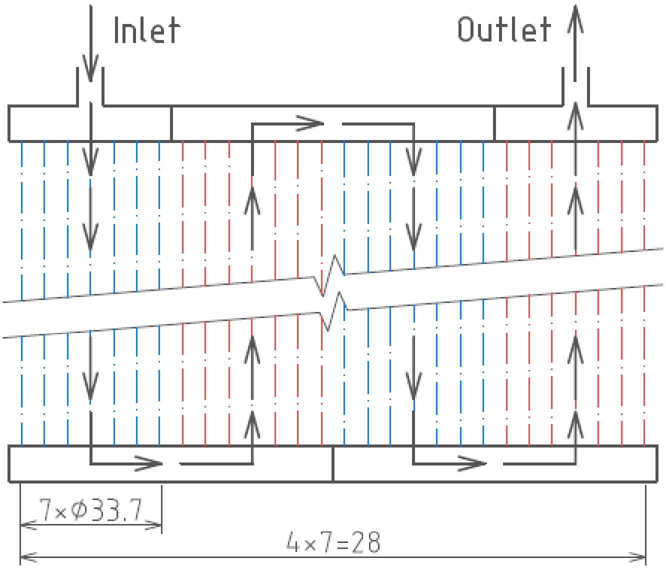

Figure 4-22 Splitting schematic ECO1HP

Actual speed of water flowing through ECO1HP:

$$W_{Steam}^{ECO1} = W_{Steam}^{ECO1\,'} \cdot n_{section}^{ECO1} = 0.22 \cdot 4 = 0.88\ m/s$$

4.9.3 Verification of the speed of the gas passing through economizer ECO1HP

Recalculated flue gas enthalpy and temperature at point H:

The parameters of the flue gas exiting the economizer at point H are recalculated according to the actual flue gas parameter entering ECO1HP.

$$I_J' = I_I^{real} - \frac{Q_{ECO1}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 202.34 - \frac{1119.61}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 154.43\, kJ/Nm^3$$

$$t_J' = \frac{I_J' - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{154.43 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 115.71℃$$

Average flue gas temperature:

$$\bar{t}_{(IJ)} = \frac{t_I^{real} + t_J'}{2} = \frac{151.26 + 115.71}{2} = 133.49℃$$

The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{ECO1} = \frac{273.15 + \bar{t}_{(IJ)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 133.49}{273.15} \cdot 23.49 = 34.97\, m^3/s$$

The cross sectional area that the flue gas flows through:

$$A_{duct}^{ECO1} = H \cdot L - H \cdot D_{tube}^{ECO1} \cdot n_{tube/r}^{ECO1} - H \cdot 2 \cdot h_{fin}^{ECO1} \cdot th_{fin}^{ECO1} \cdot n_{fin}^{ECO1} \cdot n_{tube/r}^{ECO1}$$

$$A_{duct}^{ECO1} = 6.78 \cdot 1.76 - 6.78 \cdot 0.0337 \cdot 28 - 6.78 \cdot 2 \cdot 0.012 \cdot 0.001 \cdot 220 \cdot 28 = 4.53\ m^2$$

Speed of the flue gas passing through ECO$_{1HP}$:

$$W_{Flue}^{ECO1} = \frac{\dot{M}_{VFlue}^{ECO1}}{A_{duct}^{ECO1}} = \frac{34.97}{4.53} = 7.71\ m/s$$

4.9.4 Reduced heat transfer coefficient outside the tubes of ECO$_{1HP}$, through the flue gas

The coefficient of heat transfer through convection outside the finned tubes:

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{ECO1} = \frac{s_{1(ECO1)}}{D_{tube}^{ECO1}} = \frac{0.0617}{0.0337} = 1.83$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2{}'^{ECO1} = \frac{s'^{ECO1}}{D_{tube}^{ECO1}} = \frac{\sqrt{\left(\frac{s_{1(ECO1)}}{2}\right)^2 + \left(s_{2(ECO1)}\right)^2}}{D_{tube}^{ECO1}} = \frac{\sqrt{\left(\frac{0.0617}{2}\right)^2 + 0.09^2}}{0.0337} = 2.82$$

Coefficient of the relative tube spacing:

Both the calculated values of lateral and diagonal relative tube spacing are substituted into equation *(4.14)* to determine the coefficient of the relative tube spacing.

$$\varphi_\sigma^{ECO1} = \frac{\sigma_1^{ECO1} - 1}{\sigma_2{}'^{ECO1} - 1} = \frac{1.83 - 1}{2.82 - 1} = 0.46$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through ECO$_{1HP}$ has been determined in previous calculations regarding the flue gas speed verification.

Temperature $\bar{t}_{(IJ)} = 133.49\ ℃$

$$\lambda_{Flue}^{ECO1} = \left[\frac{133.49 - 100}{100} \cdot (39.228 - 30.89) + 30.89\right] \cdot 10^{-3} = 3.368 \cdot 10^{-2}\ W/(m \cdot K)$$

$$\nu_{Flue}^{ECO1} = \left[\frac{133.49 - 100}{100} \cdot (32.36 - 21.41) + 21.41\right] \cdot 10^{-6} = 2.508 \cdot 10^{-5}\ m^2/s$$

The correction coefficient for the number of lateral row of tubes in the heat exchanger:

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 7 lateral rows of tubes in economizer ECO_{1HP}.

$$c_z^{ECO1} = 0.98$$

The coefficient of heat transfer through convection outside the finned tubes:

$$\alpha_c^{ECO1} = 0.23 \cdot c_z^{ECO1} \cdot (\varphi_\sigma^{ECO1})^{0.2} \cdot \frac{\lambda_{Flue}^{ECO1}}{s_{fin}^{ECO1}} \cdot \left(\frac{D_{tube}^{ECO1}}{s_{fin}^{ECO1}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{ECO1}}{s_{fin}^{ECO1}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{ECO1} \cdot s_{fin}^{ECO1}}{\nu_{Flue}^{ECO1}}\right)^{0.65}$$

$$\alpha_c^{ECO1} = 0.23 \cdot 0.98 \cdot 0.46^{0.2} \cdot \frac{0.03368}{0.00455} \cdot \left(\frac{0.0337}{0.00455}\right)^{-0.54} \cdot \left(\frac{0.0012}{0.00455}\right)^{-0.14} \cdot \left(\frac{7.71 \cdot 0.00455}{2.508 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{ECO1} = 46.81\ W/(m^2K)$$

Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of $\beta^{ECO1} \cdot h_{fin}^{ECO1}$ and the quotient of $D_{fin}^{ECO1}/D_{tube}^{ECO1}$, where β^{ECO1} is a coefficient.

Coefficient β^{ECO1}:

$$\beta^{ECO1} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{ECO1}}{th_{fin}^{ECO1} \cdot \lambda_{fin} \cdot \left(1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO1}\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 46.81}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 46.81)}}$$

$$\beta^{ECO1} = 42.29\, m^{-1}$$

Values needed to determine the coefficient characterizing the effectiveness of the fins:

$$\beta^{ECO1} \cdot h_{fin}^{ECO1} = 42.29 \cdot 0.012 = 0.51$$

$$\frac{D_{fin}^{ECO1}}{D_{tube}^{ECO1}} = \frac{0.0577}{0.0337} = 1.71$$

Coefficient determined using the graph:

$$E^{ECO1} = 0.9$$

Proportions of the tubes surface area made up of the fins, and tube wall:

The fraction of the total outer surface area of the finned tubes that is made up of the fins themselves:

$$\frac{S_{fin}^{ECO1}}{S_{out}^{ECO1}} = \frac{\left(\frac{D_{fin}^{ECO1}}{D_{tube}^{ECO1}}\right)^2 - 1}{\left(\frac{D_{fin}^{ECO1}}{D_{tube}^{ECO1}}\right)^2 - 1 + 2 \cdot \left(\frac{s_{fin}^{ECO1}}{D_{tube}^{ECO1}} - \frac{th_{fin}^{ECO1}}{D_{tube}^{ECO1}}\right)} = \frac{\left(\frac{0.0577}{0.0337}\right)^2 - 1}{\left(\frac{0.0577}{0.0337}\right)^2 - 1 + 2 \cdot \left(\frac{0.00455}{0.0337} - \frac{0.001}{0.0337}\right)}$$

$$\frac{S_{fin}^{ECO1}}{S_{out}^{ECO1}} = 0.9$$

The fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls:

$$\frac{S_{out-fin}^{ECO1}}{S_{out}^{ECO1}} = 1 - \frac{s_{fin}^{ECO1}}{S_{out}^{ECO1}} = 1 - 0.9 = 0.1$$

The reduced heat transfer coefficient outside the tubes:

$$\alpha_{r:out}^{ECO1} = \left(\frac{S_{fin}^{ECO1}}{S_{out}^{ECO1}} \cdot E^{ECO1} \cdot \mu + \frac{S_{out-fin}^{ECO1}}{S_{out}^{ECO1}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{ECO1}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO1}}$$

$$\alpha_{r:out}^{ECO1} = (0.9 \cdot 0.9 \cdot 1 + 0.1) \cdot \frac{0.85 \cdot 46.81}{1 + 0.0043 \cdot 0.85 \cdot 46.81} = 30.94\, W/(m^2K)$$

4.9.5 Overall heat transfer coefficient for ECO1HP

For the design of evaporators and economizers the effect that the reduced coefficient of heat transfer inside the tubes $\alpha_{r:in}$ has on the overall heat transfer coefficient k is negligible. Thus the overall heat transfer coefficient of evaporators and economizers are calculated in the same manner, as seen in equation (4.19).

$$k^{ECO1} = \alpha_{r:out}^{ECO1} = 30.94\, W/(m^2K)$$

4.9.6 Logarithmic mean temperature difference across ECO1HP

$$\Delta t_1^{ECO1} = t_I^{real} - t_8^{HP} = 151.26 - 139.4 = 11.86\, K$$

$$\Delta t_2^{ECO1} = t_J' - t_9^{HP} = 115.71 - 53 = 62.71\, K$$

$$\Delta t_{ln}^{ECO1} = \frac{\Delta t_2^{ECO1} - \Delta t_1^{ECO1}}{ln\left(\frac{\Delta t_2^{ECO1}}{\Delta t_1^{ECO1}}\right)} = \frac{62.71 - 11.86}{ln\left(\frac{62.71}{11.86}\right)} = 30.53\, K$$

4.9.7 The number of lateral rows of tubes required for ECO1HP

The total theoretical outer surface area of the heat exchanger tubes needed for the required heat transfer rate:

$$S_{out}^{ECO1} = \frac{Q_{ECO1}^{HP}}{k^{ECO1} \cdot \Delta t_{ln}^{ECO1}} = \frac{1119.61}{30.94 \cdot 30.53} = 1185.18\, m^2$$

The outer surface area of one fin:

$$S_{1fin}^{ECO1} = 2 \cdot \pi \cdot \frac{\left(D_{fin}^{ECO1}\right)^2 - \left(D_{tube}^{ECO1}\right)^2}{4} + \pi \cdot D_{fin}^{ECO1} \cdot th_{fin}^{ECO1}$$

$$S_{1fin}^{ECO1} = 2 \cdot \pi \cdot \frac{0.0577^2 - 0.0337^2}{4} + \pi \cdot 0.0577 \cdot 0.001 = 0.0036\, m^2$$

The outer surface area of one finned tube per meter length:

$$S_{out/1m}^{ECO1} = \pi \cdot D_{tube}^{ECO1} \cdot \left(1 - n_{fin}^{ECO1} \cdot th_{fin}^{ECO1}\right) + n_{fin}^{ECO1} \cdot S_{1fin}^{ECO1}$$

$$S_{out/1m}^{ECO1} = \pi \cdot 0.0337 \cdot (1 - 220 \cdot 0.001) + 220 \cdot 0.0036 = 0.8805\, m$$

The outer surface area of all the tubes in one lateral row of the heat exchanger:

$$S_{out/r}^{ECO1} = S_{out/1m}^{ECO1} \cdot H \cdot n_{tube/r}^{ECO1} = 0.8805 \cdot 6.78 \cdot 28 = 167.16\, m^2$$

The number of lateral rows of tubers in ECO1HP:

$$n_{row}^{ECO1} = \frac{S_{out}^{ECO1}}{S_{out/r}^{ECO1}} = \frac{1185.18}{167.16} = 7.09$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{ECO1} = 7$$

4.9.8 Calculating the actual heat transfer rate of ECO1HP

The actual total outer surface area of the heat exchanger tubes:

$$S_{out}^{ECO1:real} = S_{out/r}^{ECO1} \cdot n_{row}^{ECO1} = 167.16 \cdot 7 = 1170.1\, m^2$$

The actual heat transfer rate of ECO1HP:

$$Q_{ECO1}^{HP:real} = k^{ECO1} \cdot S_{out}^{ECO1:real} \cdot \Delta t_{ln}^{ECO1} = 30.94 \cdot 1170.1 \cdot 30.53 = 1105.36\, kW$$

Error verification:

$$\%Error^{ECO1} = \left| \left(\frac{1105.36}{1119.61} - 1 \right) \cdot 100 \right| = 1.27\%$$

The discrepancy between the theoretical and the actual heat transfer rate of economizer ECO_{1HP} is less than 2%, thus the current configuration of ECO_{1HP} is acceptable.

4.9.9 Determining the actual parameters of the flue gas exiting ECO_{1HP}

Due to the discrepancy between the theoretical and the actual heat transfer rate of economizer ECO_{1HP} the parameters of the flue gas exiting ECO_{1HP} at point J must be recalculated.

Actual enthalpy of the flue gas at point J:

$$I_J^{real} = I_I^{real} - \frac{Q_{ECO1}^{HP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 202.34 - \frac{1105.36}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 155.04\, kJ/Nm^3$$

Actual temperature of the flue gas at point J:

$$t_J^{real} = \frac{I_J^{real} - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{155.04 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 116.16℃$$

4.10 Overview of the Calculated Values

An overview of the main values calculated in this section are presented in *Table 4.24* and *Table 4.25*. They include values regarding individual heat exchanger as well as actual flue gas parameters.

parameters		Flue gas speed	Water/steam speed	Actual heat transfer rate	Longitudinal tube spacing	Lateral tube spacing	Number of tubes per lateral row	Number of lateral rows
Symbol		W_{Flue}	W_{Steam}	Q^{real}	s_2	s_1	$n_{tube/r}$	n_{row}
Units		$[m/s]$	$[m/s]$	$[kW]$	$[mm]$	$[mm]$	$[-]$	$[-]$
High pressure	SH_{2HP}	10	19.79	807.23	117	74.8	23	4
	SH_{1HP}	9.73	15.98	1297.87	117	74.8	23	4
	EV_{HP}	10.45	-	5165.08	117	106.5	16	17
	ECO_{3HP}	10.27	1	1468.96	90	54.05	32	9
	ECO_{2HP}	9.6	0.8	155.55	90	54.15	32	2
	ECO_{1HP}	7.71	0.88	1105.36	90	61.7	28	7
Low pressure	SH_{LP}	7.15	19.54	35.35	117	76	23	1
	EV_{LP}	9.27	-	1833.98	117	100.5	17	8
	ECO_{LP}	5.74	0.81	369.65	90	61.75	28	2

Table 4.24 Parameters regarding individual heat exchangers

Point	Temperature [°C]		Enthalpy [kJ/Nm^3]	
	symbol	value	symbol	value
A	t_A	490	I_A	678.69
B	t_B	466.32	I_B	644.15
C	t_C	428.24	I_C	588.62
D	t_D	272.23	I_D	367.64
E	t_E	226.66	I_E	304.79
F	t_F	225.57	I_F	303.27
G	t_G	167.94	I_G	224.81
H	t_H	163	I_H	218.15
I	t_I	151.26	I_I	202.34
J	t_J	116.16	I_J	155.04

Table 4.25 Actual flue gas parameters at various points throughout the boiler

5 MATERIAL SELECTION

The material that the heat exchanger tubes are made of is selected according standers (ČSN EN 12952-3) which outline the maximum temperature that each material can be used for. For finned tubes the reference temperature, used to determine which materials can be used, is the temperature of the water/steam exiting the heat exchanger with and additional safety margin. For smooth tube the reference temperature is simply the temperature of the water/steam exiting the heat exchanger. The Material chosen for the fins on all the heat exchanger tubes is CS, as the flue gas temperature does not exceed 500 °C [7]. The materials chosen for the tubes of each heat exchanger are shown in *Table 5.1*.

<u>Calculations used to determine the reference temperature (finned tubes):</u>

Reference temperature:

$$t_{ref} = t_{Steam:out} + \Delta t_{safe} \tag{5.1}$$

Where:

$t_{Steam:out}$ is temperature water/steam exiting the heat exchanger [°C]

Δt_{safe} is additional safety margin [°C]

Safety margin (for superheaters):

$$\Delta t_{safe} = 35 \text{ °C} \tag{5.2}$$

Safety margin (for all other heat exchangers):

$$\Delta t_{safe} = 15 + 2 \cdot th_{tube(mm)} \tag{5.3}$$

Where:

$th_{tube(mm)}$ is the thickness of the tube wall [mm]

	Parameter	Exiting steam temperature	Safety margin	Reference temperature	Selected tube material (ČSN EN 12952-3)
	Symbol	$t_{Steam:in}$	Δt_{safe}	t_{ref}	-
	Units	[°C]	[°C]	[°C]	-
High pressure	$SH2_{HP}$	450	35	485	16Mo3
	SH_{1HP}	408.46	35	443.46	16Mo3
	EV_{HP}	261.4	23	284.4	P235GH
	ECO_{3HP}	256.4	23	279.4	P235GH
	ECO_{2HP}	151.4	23	174.4	P235GH
	ECO_{1HP}	139.4	23	162.4	P235GH
Low	SH_{LP}	170	0	170	P235GH
	EV_{LP}	156.15	23	179.15	P235GH
	ECO_{LP}	151.15	23	174.15	P235GH

Table 5.1 Materials selected for heat exchanger tubes

6 STEAM DRUM DESIGN

Both the high and low pressure steam drums serve the same purpose. They physically separate the saturated steam from the water, with the help of various separators housed inside. This phase separation insured that only steam enters into the superheaters. The design of a steam drum entails defining the geometry, and then verifying that the drum load is acceptable. The outer diameter of the drum D_{drum} is chosen according to the steam production rate $\dot{M}_{Steam}$ of the respective circuit, for steam production rates of less than 15 t/h the outer drum diameter is suggested to be 1.2 m. The length if the drum L_{drum} is chosen to be the same as the width of the flue gas duct L. The wall thickness of the steam drum th_{drum} is selected from a suggested range of between 40 and 120 mm [7].

Equations used in the design of steam drums

Load verification:

$$z_{drum} < z_{max} \tag{6.1}$$

Where:

z_{drum}	is the steam drum load [kg/(s·m^3)]
z_{max}	is the acceptable steam drum load [kg/(s·m^3)]

Steam drum load:

$$z_{drum} = \frac{\dot{M}_{Steam}}{V_{drum}} \tag{6.2}$$

Where:

$\dot{M}_{Steam}$	is the steam production rate of the respective circuit [kg/s]
V_{drum}	is half of the steam drum volume [m^3]

Half of the steam drum volume:

$$V_{drum} = \frac{\pi \cdot (d_{drum})^2 \cdot L_{drum}}{8} \tag{6.3}$$

Where:

d_{drum}	is the inner steam drum diameter [m]
L_{drum}	is steam drum length [m]

6.1 High Pressure Drum Design

The steam drum geometry is chosen according to the steam production rate of the high pressure circuit, which is 3.24 kg/s or approximately 11.67 t/h:

$$D_{drum}^{HP} = 1.2\ m$$

$$th_{drum}^{HP} = 0.04\ m$$

$$d_{drum}^{HP} = D_{drum}^{HP} - 2 \cdot th_{drum}^{HP} = 1.2 - 2 \cdot 0.04 = 1.12\ m$$

$$L_{drum}^{HP} = L = 1.76\ m$$

Half of the steam drum volume:

$$V_{drum}^{HP} = \frac{\pi \cdot \left(d_{drum}^{HP}\right)^2 \cdot L_{drum}^{HP}}{8} = \frac{\pi \cdot 1.12^2 \cdot 1.76}{8} = 0.87 m^2$$

Steam drum load:

$$z_{drum}^{HP} = \frac{\dot{M}_{Steam}^{HP}}{V_{drum}^{HP}} = \frac{3.24}{0.87} = 3.74\ kg/(s \cdot m^3)$$

Load verification:

The acceptable steam drum load is determined from *Table 6.1*, through linear interpolation, according to the pressure in the steam drum. This pressure has been determined in previous calculations.

Pressure $p_{drum}^{HP} = p_4^{HP} = 4.8\ MPa$

Parameter	Symbol	Units					
Steam drum pressure	p_{drum}	$[MPa]$	0.932	1.47	2.65	4.22	6.8
Acceptable load	z_{max}	$[kg/(s \cdot m^3)]$	2.25	2.8	4.25	5.4	6.9

Table 6.1 Acceptable steam drum load according to pressure

$$z_{max}^{HP} = \frac{4.8 - 4.22}{6.8 - 4.22} \cdot (6.9 - 5.4) + 5.4 = 5.74\ kg/(s \cdot m^3)$$

The value of the steam drum load is lower than the acceptable load, thus the current geometry of the high pressure steam drum is acceptable.

6.2 Low Pressure Drum Design

The steam drum geometry is chosen according to the steam production rate of the high pressure circuit, which is 0.88 kg/s or approximately 3.18 t/h:

$$D_{drum}^{LP} = 1.2\ m$$

$$th_{drum}^{LP} = 0.04\ m$$

$$d_{drum}^{LP} = D_{drum}^{LP} - 2 \cdot th_{drum}^{LP} = 1.2 - 2 \cdot 0.04 = 1.12\ m$$

$$L_{drum}^{LP} = L = 1.76\ m$$

Half of the steam drum volume:

$$V_{drum}^{LP} = \frac{\pi \cdot \left(d_{drum}^{LP}\right)^2 \cdot L_{drum}^{LP}}{8} = \frac{\pi \cdot 1.12^2 \cdot 1.76}{8} = 0.87 m^2$$

Steam drum load:

$$z_{drum}^{LP} = \frac{\dot{M}_{Steam}^{LP}}{V_{drum}^{LP}} = \frac{0.88}{0.87} = 1.02\ kg/(s \cdot m^3)$$

Load verification:

The acceptable steam drum load is determined from *Table 6.1*, through linear interpolation, according to the pressure in the steam drum. This pressure has been determined in previous calculations.

Pressure $\qquad p_{drum}^{LP} = p_4^{LP} = 0.56\ MPa$

$$z_{max}^{LP} = \frac{0.56 - 0.932}{1.47 - 0.932} \cdot (2.8 - 2.25) + 2.25 = 1.87\ kg/(s \cdot m^3)$$

The value of the steam drum load is lower than the acceptable load, thus the current geometry of the high pressure steam drum is acceptable.

6.3 Overview of the Calculated Values

An overview of the main values calculated in this section are presented in *Table 4.24*. They include values regarding individual heat exchanger as well as actual flue gas parameters.

Parameter	Symbol	Units	High pressure steam drum	Low pressure steam drum
Steam drum pressure	p_{drum}	$[MPa]$	4.8	0.56
Acceptable steam drum load	z_{drum}	$[kg/(s \cdot m^2)]$	3.74	1.02
Steam drum load	z_{max}	$[kg/(s \cdot m^2)]$	5.74	1.87
Outer diameter	D_{drum}	$[m]$	1.2	1.2
Wall thickness	th_{drum}	$[m]$	0.04	0.04
Length	L_{drum}	$[m]$	1.76	1.76

Table 6.2 Overview of calculated values

7 DESIGN AND SIZING OF THE DOWNCOMER AND RISER TUBES

Both the high and low pressure circuits in this boiler depend on natural circulation for the production of saturated steam. The driving force of this circulation is the deference in density between the water in the downcomer tubes and the water/steam mixture rising through the evaporator tubes. Downcomer tubes supply water to the evaporator tubes, and riser tubes carry the water/steam mixture from the evaporator tubes back into the steam drum. The downcomer and riser tubes are sized according to their cross sectional area in relation to the cross sectional area of the respective evaporator tubes. These sizes are also dependent on the approximate height of the steam drum, or the height of the evaporator circulation loop. The calculations regarding sizing of the downcomer and riser tubes are carried out according to source [5].

Equations used in the sizing of downcomer and riser tubes:

Cross sectional area of the downcomer tubes [m²]:

$$A_{Down} = A_{EV} \cdot (0.06 + 0.016 \cdot p_{drum} + 0.005 \cdot h_{drum}) \quad (7.1)$$

Where:

A_{EV} is the cross sectional area of the evaporator tubes [m²]

h_{drum} is the approximate steam drum height [kg/(s·m³)]

Steam drum height:

The approximate height steam drum, or the height of the evaporator circulation loop is calculated according to equation *(7.2)*, approximately 3 meters higher than the flue gas duct height [7].

$$h_{drum} = H + 3 \quad (7.2)$$

Cross sectional area of the evaporator tubes:

$$A_{EV} = \frac{\pi \cdot (d_{tube})^2}{4} \cdot n_{tube/r} \cdot n_{row} \quad (7.3)$$

Cross sectional area of the riser tubes [m²]:

$$A_{Riser} = A_{EV} \cdot (0.1 + 0.01 \cdot p_{drum} + 0.01 \cdot h_{drum}) \quad (7.4)$$

7.1 Downcomer Tube Sizing

Approximate high and low pressure Steam drum height:

$$h_{drum}^{HP} = h_{drum}^{LP} = H + 3 = 6.78 + 3 = 9.78\ m$$

7.1.1 High pressure downcomer tube sizing

Cross sectional area of the evaporator tubes:

$$A_{EV}^{HP} = \frac{\pi \cdot \left(d_{tube}^{EV1}\right)^2}{4} \cdot n_{tube/r}^{EV1} \cdot n_{row}^{EV1} = \frac{\pi \cdot 0.049^2}{4} \cdot 16 \cdot 17 = 0.513\, m^2$$

Cross sectional area of the downcomer tubes:

$$A_{Down}^{HP} = A_{EV}^{HP} \cdot \left(0.06 + 0.016 \cdot p_{drum}^{HP} + 0.005 \cdot h_{drum}^{HP}\right)$$

$$A_{Down}^{HP} = 0.51 \cdot (0.06 + 0.016 \cdot 4.8 + 0.005 \cdot 9.78) = 0.095\, m^2$$

The geometry of the downcomer tubes is chosen and then used to determine the number of downcomer tubes required.

Parameter	Symbol	Value	Units
Outer tube diameter	D_{Down}^{HP}	244.5	$[mm]$
Wall thickness	th_{Down}^{HP}	10	$[mm]$
Inner tube diameter	d_{Down}^{HP}	224.5	$[mm]$

Table 7.1 High pressure downcomer tube dimensions

Number of downcomer tubes required:

The number of downcomer tubes is rounded up to a whole number

$$n_{Down}^{HP} = \frac{A_{Down}^{HP}}{\frac{\pi \cdot (d_{Down}^{HP})^2}{4}} = \frac{0.513}{\frac{\pi \cdot 0.2245^2}{4}} = 2.4$$

$$n_{Down}^{HP} = 3$$

7.1.2 Low pressure downcomer tube sizing

Cross sectional area of the evaporator tubes:

$$A_{EV}^{LP} = \frac{\pi \cdot \left(d_{tube}^{EV}\right)^2}{4} \cdot n_{tube/r}^{EV} \cdot n_{row}^{EV} = \frac{\pi \cdot 0.049^2}{4} \cdot 17 \cdot 8 = 0.256\, m^2$$

Cross sectional area of the downcomer tubes:

$$A_{Down}^{LP} = A_{EV}^{LP} \cdot \left(0.06 + 0.016 \cdot p_{drum}^{LP} + 0.005 \cdot h_{drum}^{LP}\right)$$

$$A_{Down}^{LP} = 0.51 \cdot (0.06 + 0.016 \cdot 0.56 + 0.005 \cdot 9.78) = 0.03\, m^2$$

The geometry of the downcomer tubes is chosen and then used to determine the number of downcomer tubes required.

Parameter	Symbol	Value	Units
Outer tube diameter	D_{Down}^{LP}	244.5	$[mm]$
Wall thickness	th_{Down}^{LP}	10	$[mm]$
Inner tube diameter	d_{Down}^{LP}	224.5	$[mm]$

Table 7.2 Low pressure downcomer tube dimensions

Number of downcomer tubes required:

The number of downcomer tubes is rounded up to a whole number

$$n_{Down}^{LP} = \frac{A_{Down}^{LP}}{\frac{\pi \cdot (d_{Down}^{LP})^2}{4}} = \frac{0.256}{\frac{\pi \cdot 0.2245^2}{4}} = 0.76$$

$$n_{Down}^{LP} = 1$$

7.2 Riser Tube Sizing

7.2.1 Height pressure riser tube sizing

Cross sectional area of the riser tubes:

$$A_{Riser}^{HP} = A_{EV}^{HP} \cdot \left(0.1 + 0.01 \cdot p_{drum}^{HP} + 0.01 \cdot h_{drum}^{HP}\right)$$

$$A_{Riser}^{HP} = 0.513 \cdot (0.1 + 0.01 \cdot 4.8 + 0.01 \cdot 9.78) = 0.126\, m^2$$

The geometry of the riser tubes is chosen and then used to determine the number of riser tubes required.

Parameter	Symbol	Value	Units
Outer tube diameter	D_{Riser}^{HP}	133	$[mm]$
Wall thickness	th_{Riser}^{HP}	8	$[mm]$
Inner tube diameter	d_{Riser}^{HP}	117	$[mm]$

Table 7.3 High pressure riser tube dimensions

Number of riser tubes required:

The number of riser tubes is rounded up to a whole number

$$n_{Riser}^{HP} = \frac{A_{Riser}^{HP}}{\frac{\pi \cdot \left(d_{Riser}^{HP}\right)^2}{4}} = \frac{0.126}{\frac{\pi \cdot 0.117^2}{4}} = 11.73$$

$$n_{Riser}^{HP} = 12$$

7.2.2 Low pressure riser tube sizing

Cross sectional area of the riser tubes:

$$A_{Riser}^{LP} = A_{EV}^{LP} \cdot \left(0.1 + 0.01 \cdot p_{drum}^{LP} + 0.01 \cdot h_{drum}^{LP}\right)$$

$$A_{Riser}^{LP} = 0.513 \cdot (0.1 + 0.01 \cdot 0.56 + 0.01 \cdot 9.78) = 0.052\, m^2$$

The geometry of the riser tubes is chosen and then used to determine the number of riser tubes required.

Parameter	Symbol	Value	Units
Outer tube diameter	D_{Riser}^{HP}	168.3	$[mm]$
Wall thickness	th_{Riser}^{HP}	8	$[mm]$
Inner tube diameter	d_{Riser}^{HP}	152.3	$[mm]$

Table 7.4 Low pressure riser tube dimensions

Number of riser tubes required:

The number of riser tubes is rounded up to a whole number

$$n_{Riser}^{HP} = \frac{A_{Riser}^{HP}}{\frac{\pi \cdot \left(d_{Riser}^{HP}\right)^2}{4}} = \frac{0.052}{\frac{\pi \cdot 0.1523^2}{4}} = 2.86$$

$$n_{Riser}^{HP} = 3$$

8 FLUE GAS PRESSURE LOSSES THROUGH THE BOILER

The total flue gas pressure loss through the boiler $\Delta p_{L:total}$ must be lower than the maximum acceptable pressure loss $\Delta p_{Lmax} = 1500Pa$. The total flue gas pressure loss is calculated as the sum of the pressure losses across each heat exchanger and the pressure loss across the stack.

8.1 Flue Gas Pressure Losses across Individual Heat Exchangers

Calculation regarding pressure losses across the individual heat exchangers are carried out according to source [3], calculation regarding heat exchangers with smooth walls are calculated according to source [5].

Equations used to determine flue gas pressure losses:

Pressure loss across a heat exchanger (with finned or smooth tubes):

$$\Delta p_L = \varsigma \cdot \frac{(W_{Flue})^2}{2} \cdot \rho_{Flue} \tag{8.1}$$

Where:

Δp_L is flue gas pressure loss across individual heat exchangers [Pa]

ς is the pressure loss coefficient, determined from the geometry of the heat exchanger [-]

The pressure loss coefficient (for heat exchangers with finned tubes):

$$\varsigma = k_4 \cdot n_{row} \cdot \left(\frac{h_{fin}}{D_{tube}}\right)^{k_1} \cdot \left(\frac{s_{fin}}{D_{tube}}\right)^{-k_2} \cdot (Re)^{-k_3} \tag{8.2}$$

Where:

Re is the Reynolds number of the flue gas [-]

$k_{1,2,3,4}$ are constants, which depend on the configuration of the tubes in the heat exchangers [-]

For heat exchangers with a staggered tube configuration the values of the constants used are presented in *Table 8.1*.

Constant	k_1	k_2	k_3	k_4
Value [-]	0	0.72	0.24	2

Table 8.1 Pressure loss coefficient constants

The pressure loss coefficient (for heat exchangers with smooth tubes):

This equation is applicable for cases where $s_1/D_{tube} < s_2/D_{tube}$

$$\varsigma = (4 + 6.6 \cdot n_{row}) \cdot (Re)^{-0.28} \tag{8.3}$$

Reynolds number:

$$Re = \frac{(W_{Flue} \cdot d_e)^2}{\nu_{Flue}} \tag{8.4}$$

Where:

d_e	is equivalent diameter [m]
ν_{Flue}	is the kinematic viscosity of the flue gas, according to the average temperature of the tarnished outer tube walls [m^2/s]

Equivalent diameter of the flue gas duct:

$$d_{e:duct} = \frac{4 \cdot A_{E:duct}}{PER_{duct}} = \frac{4 \cdot L \cdot H}{2 \cdot (L + H)} = \frac{4 \cdot 1.76 \cdot 6.78}{2 \cdot (1.76 + 6.78)} = 2.79\ m \tag{8.5}$$

Where:

$A_{E:duct}$	is the cross sectional area of the flue gas duct [m^2]
PER_{duct}	Is the perimeter of the flue gas duct form the view of the cross section [m]

Average temperature of the tarnished outer tube walls:

$$t_{wall} = \bar{t}_{Steam} + \Delta t \tag{8.6}$$

Where:

$\bar{t}_{Steam}$	is the average temperature of the water/steam inside the tubes [Pa]
Δt	is a given temperature difference, for gashouse fuels (Δt = 25°C) [°C]

Values that will be used to calculate the flue gas pressure drop across the individual heat exchangers are organized in *Table 8.2* below. This table also includes the Reynolds number and outer tube wall surface temperature, which have been calculated according to the equations above.

	Parameters	Outer tube diameter	Fin height	Fin spacing	Flue gas speed	Number of lateral rows	Average wall temperature	kinematic viscosity	Reynolds number
	Symbol	D_{tube}	h_{fin}	s_{fin}	W_{Flue}	n_{row}	t_{wall}	ν_{Flue}	Re
	Units	$[mm]$	$[mm]$	$[mm]$	$[m/s]$	$[-]$	[℃]	$[m^2/s]$	$[-]$
High pressure	SH_{1HP}	31.8	15	9.09	10	4	424.23	$6.36 \cdot 10^{-5}$	$1.23 \cdot 10^{7}$
	SH_{2HP}	31.8	15	7.69	9.73	4	359.93	$5.41 \cdot 10^{-5}$	$1.37 \cdot 10^{7}$
	EV_{HP}	57	19	6.67	10.45	17	238.9	$4.33 \cdot 10^{-5}$	$1.97 \cdot 10^{7}$
	ECO_{3HP}	31.8	8	5.26	10.27	9	228.9	$3.61 \cdot 10^{-5}$	$2.28 \cdot 10^{7}$
	ECO_{2HP}	33.7	8	5	9.6	2	170.4	$2.91 \cdot 10^{-5}$	$2.47 \cdot 10^{7}$
	ECO_{1HP}	33.7	12	4.55	7.71	7	121.2	$2.37 \cdot 10^{-5}$	$1.96 \cdot 10^{7}$
Low	SH_{LP}	38	-	-	7.15	1	188.08	$3.11 \cdot 10^{-5}$	$1.28 \cdot 10^{7}$
	EV_{LP}	57	19	4.35	9.27	8	178.65	$3 \cdot 10^{-5}$	$2.23 \cdot 10^{7}$
	ECO_{LP}	22	15	4.35	5.74	2	127.08	$2.44 \cdot 10^{-5}$	$1.06 \cdot 10^{7}$

Table 8.2 Values for calculations regarding flue gas pressure losses

8.1.1 Pressure losses across superheater SH_{2HP}

The pressure loss coefficient for SH_{2HP}:

$$\varsigma_{SH2} = k_4 \cdot n_{row}^{SH2} \cdot \left(\frac{h_{fin}^{SH2}}{D_{tube}^{SH2}}\right)^{k_1} \cdot \left(\frac{s_{fin}^{SH2}}{D_{tube}^{SH2}}\right)^{-k_2} \cdot \left(Re_{(SH2)}\right)^{-k_3}$$

$$\varsigma_{SH2} = 2 \cdot 4 \cdot \left(\frac{15}{31.8}\right)^{0} \cdot \left(\frac{9.09}{31.8}\right)^{-0.72} \cdot (1.23 \cdot 10^7)^{-0.24} = 0.39$$

Pressure loss across SH_{2HP}:

$$\Delta p_L^{SH2} = \varsigma_{SH2} \cdot \frac{\left(W_{Flue}^{SH2}\right)^2}{2} \cdot \rho_{Flue} = 0.39 \cdot \frac{10^2}{2} \cdot 1.2772 = 25.04\ Pa$$

8.1.2 Pressure losses across superheater SH_{1HP}

The pressure loss coefficient for SH_{1HP}:

$$\varsigma_{SH1} = k_4 \cdot n_{row}^{SH1} \cdot \left(\frac{h_{fin}^{SH1}}{D_{tube}^{SH1}}\right)^{k_1} \cdot \left(\frac{s_{fin}^{SH1}}{D_{tube}^{SH1}}\right)^{-k_2} \cdot \left(Re_{(SH1)}\right)^{-k_3}$$

$$\varsigma_{SH1} = 2 \cdot 4 \cdot \left(\frac{15}{31.8}\right)^{0} \cdot \left(\frac{7.69}{31.8}\right)^{-0.72} \cdot (1.37 \cdot 10^7)^{-0.24} = 0.43$$

Pressure loss across SH_{1HP}:

$$\Delta p_L^{SH1} = \varsigma_{SH1} \cdot \frac{\left(W_{Flue}^{SH1}\right)^2}{2} \cdot \rho_{Flue} = 0.43 \cdot \frac{9.73^2}{2} \cdot 1.2772 = 26.04\ Pa$$

8.1.3 Pressure losses across evaporator EV_{HP}

The pressure loss coefficient for EV_{HP}:

$$\varsigma_{EV1} = k_4 \cdot n_{row}^{EV1} \cdot \left(\frac{h_{fin}^{EV1}}{D_{tube}^{EV1}}\right)^{k_1} \cdot \left(\frac{s_{fin}^{EV1}}{D_{tube}^{EV1}}\right)^{-k_2} \cdot \left(Re_{(EV1)}\right)^{-k_3}$$

$$\varsigma_{EV1} = 2 \cdot 17 \cdot \left(\frac{19}{57}\right)^0 \cdot \left(\frac{6.67}{57}\right)^{-0.72} \cdot (1.97 \cdot 10^7)^{-0.24} = 2.83$$

Pressure loss across EV_{HP}:

$$\Delta p_L^{EV1} = \varsigma_{EV1} \cdot \frac{\left(W_{Flue}^{EV1}\right)^2}{2} \cdot \rho_{Flue} = 2.83 \cdot \frac{10.45^2}{2} \cdot 1.2772 = 197.3\ Pa$$

8.1.4 Pressure losses across economizer ECO_{3HP}

The pressure loss coefficient for ECO_{3HP}:

$$\varsigma_{ECO3} = k_4 \cdot n_{row}^{ECO3} \cdot \left(\frac{h_{fin}^{ECO3}}{D_{tube}^{ECO3}}\right)^{k_1} \cdot \left(\frac{s_{fin}^{ECO3}}{D_{tube}^{ECO3}}\right)^{-k_2} \cdot \left(Re_{(ECO3)}\right)^{-k_3}$$

$$\varsigma_{ECO3} = 2 \cdot 9 \cdot \left(\frac{8}{31.8}\right)^0 \cdot \left(\frac{5.26}{31.8}\right)^{-0.72} \cdot (2.28 \cdot 10^7)^{-0.24} = 1.13$$

Pressure loss across ECO_{3HP}:

$$\Delta p_L^{ECO3} = \varsigma_{ECO3} \cdot \frac{\left(W_{Flue}^{ECO3}\right)^2}{2} \cdot \rho_{Flue} = 1.13 \cdot \frac{10.27^2}{2} \cdot 1.2772 = 75.96\ Pa$$

8.1.5 Pressure losses across economizer ECO_{2HP}

The pressure loss coefficient for ECO_{2HP}:

$$\varsigma_{ECO2} = k_4 \cdot n_{row}^{ECO2} \cdot \left(\frac{h_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)^{k_1} \cdot \left(\frac{s_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)^{-k_2} \cdot \left(Re_{(ECO2)}\right)^{-k_3}$$

$$\varsigma_{ECO2} = 2 \cdot 2 \cdot \left(\frac{8}{33.7}\right)^0 \cdot \left(\frac{5}{33.7}\right)^{-0.72} \cdot (2.47 \cdot 10^7)^{-0.24} = 0.27$$

Pressure loss across ECO_{2HP}:

$$\Delta p_L^{ECO2} = \varsigma_{ECO2} \cdot \frac{\left(W_{Flue}^{ECO2}\right)^2}{2} \cdot \rho_{Flue} = 0.27 \cdot \frac{9.6^2}{2} \cdot 1.2772 = 15.65\ Pa$$

8.1.6 Pressure losses across economizer ECO_{1HP}

The pressure loss coefficient for ECO_{1HP}:

$$\varsigma_{ECO1} = k_4 \cdot n_{row}^{ECO1} \cdot \left(\frac{h_{fin}^{ECO1}}{D_{tube}^{ECO1}}\right)^{k_1} \cdot \left(\frac{s_{fin}^{ECO1}}{D_{tube}^{ECO1}}\right)^{-k_2} \cdot \left(Re_{(ECO1)}\right)^{-k_3}$$

$$\varsigma_{ECO1} = 2 \cdot 7 \cdot \left(\frac{12}{33.7}\right)^{0} \cdot \left(\frac{4.55}{33.7}\right)^{-0.72} \cdot (1.96 \cdot 10^7)^{-0.24} = 1.05$$

Pressure loss across ECO_{1HP}:

$$\Delta p_L^{ECO1} = \varsigma_{ECO1} \cdot \frac{\left(W_{Flue}^{ECO1}\right)^2}{2} \cdot \rho_{Flue} = 1.05 \cdot \frac{7.71^2}{2} \cdot 1.2772 = 40.03\ Pa$$

8.1.7 Pressure losses across superheater SH_{LP}

The pressure loss coefficient for SH_{LP}:

$$\varsigma_{SH} = (4 + 6.6 \cdot n_{row}^{SH}) \cdot \left(Re_{(SH)}\right)^{-0.28} = (4 + 6.6 \cdot 1) \cdot (1.28 \cdot 10^7)^{-.028} = 0.11$$

Pressure loss across SH_{LP}:

$$\Delta p_L^{SH} = \varsigma_{SH} \cdot \frac{\left(W_{Flue}^{SH}\right)^2}{2} \cdot \rho_{Flue} = 0.11 \cdot \frac{10^2}{2} \cdot 1.2772 = 3.53\ Pa$$

8.1.8 Pressure losses across evaporator EV_{LP}

The pressure loss coefficient for EV_{LP}:

$$\varsigma_{EV} = k_4 \cdot n_{row}^{EV} \cdot \left(\frac{h_{fin}^{EV}}{D_{tube}^{EV}}\right)^{k_1} \cdot \left(\frac{s_{fin}^{EV}}{D_{tube}^{EV}}\right)^{-k_2} \cdot \left(Re_{(EV)}\right)^{-k_3}$$

$$\varsigma_{EV} = 2 \cdot 8 \cdot \left(\frac{19}{57}\right)^{0} \cdot \left(\frac{4.35}{57}\right)^{-0.72} \cdot (2.23 \cdot 10^7)^{-0.24} = 1.76$$

Pressure loss across EV_{LP}:

$$\Delta p_L^{EV} = \varsigma_{EVH} \cdot \frac{\left(W_{Flue}^{EV}\right)^2}{2} \cdot \rho_{Flue} = 1.76 \cdot \frac{9.27^2}{2} \cdot 1.2772 = 96.41\ Pa$$

8.1.9 Pressure losses across economizer ECO_{LP}

The pressure loss coefficient for ECO_{LP}:

$$\varsigma_{ECO} = k_4 \cdot n_{row}^{ECO} \cdot \left(\frac{h_{fin}^{ECO}}{D_{tube}^{ECO}}\right)^{k_1} \cdot \left(\frac{s_{fin}^{ECO}}{D_{tube}^{ECO}}\right)^{-k_2} \cdot \left(Re_{(ECO)}\right)^{-k_3}$$

$$\varsigma_{ECO} = 2 \cdot 2 \cdot \left(\frac{15}{22}\right)^{0} \cdot \left(\frac{4.35}{22}\right)^{-0.72} \cdot (1.06 \cdot 10^7)^{-0.24} = 0.27$$

Pressure loss across ECO$_{LP}$:

$$\Delta p_L^{ECO} = \varsigma_{ECO} \cdot \frac{\left(W_{Flue}^{ECO}\right)^2}{2} \cdot \rho_{Flue} = 0.27 \cdot \frac{5.74^2}{2} \cdot 1.2772 = 5.58\ Pa$$

8.1.10 Flue gas pressure losses across all the heat exchangers in the boiler

The pressure losses across all the heat exchangers in the boiler are calculated intuitively as the sum of the pressure losses across the individual heat exchangers, as seen in equation *(8.7)*

$$\Delta p_{L1} = \Delta p_L^{SH2} + \Delta p_L^{SH1} + \Delta p_L^{EV1} + \Delta p_L^{ECO3} + \Delta p_L^{ECO2} + \Delta p_L^{ECO1} + \Delta p_L^{SH} + \Delta p_L^{EV} + \Delta p_L^{ECO} \quad (8.7)$$

$$\Delta p_{L1} = 25.04 + 26.04 + 197.3 + 75.96 + 15.65 + 40.03 + 3.53 + 96.41 + 5.58 = 485.53\ Pa$$

8.2 Flue Gas Pressure Losses across the Stack

The total pressure e loss across the stack is the sum of the local pressure losses at the stack inlet, outlet, across the stack silencer, and the pressure loss due to the friction of the inner surface of the stack. It is also important to take into account the stack effect (draft effect) which is counteracting the pressure losses. Chosen values that are used in the calculations regarding pressure losses are organized in *Table 8.3*.

Parameter	Symbol	Value	Units
Functional stack height	H_{stack}	16	$[m]$
Inner stack diameter	d_{stack}	1.05	$[m]$
Coefficient of friction inside the stack (sheet metal)	λ_{stack}	0.03	$[-]$
Inlet pressure loss coefficient	ς_{stack}^{in}	1	$[-]$
Outlet pressure loss coefficient	ς_{stack}^{out}	1	$[-]$
Pressure loss across the stack silencer	Δp_L^{sil}	200	$[Pa]$
Stack inlet temperature	t_J	116.16	[°C]
Density of air	ρ_{air}	1.275	$[kg/Nm^3]$
Acceleration due to gravity	g	9.81	$[m/s^2]$

Table 8.3 Values for stack pressure loss calculations

8.2.1 Pressure loss across the stack due to friction

The pressure loss across the stack due to friction is calculated according to equation *(8.8)*.

$$\Delta p_L^{fric} = \lambda_{stack} \cdot \frac{H_{stack}}{d_{stack}} \cdot \frac{(W_{Flue})^2}{2} \cdot \rho_{Flue} \quad (8.8)$$

Where:

λ_{stack} is coefficient of friction inside the stack [-]

H_{stack} is functional stack height [m]

d_{stack} is the inner stack diameter [m]

The speed of the flue gas in the stack is calculated intuitively according to the cross sectional area of the stack and the actual volumetric flow rate of the flue gas.

Cross sectional area of the stack:

$$A_{stack} = \frac{\pi \cdot (d_{stack})^2}{2} = \frac{\pi \cdot 1.05^2}{2} = 1.73\ m^2$$

Actual volumetric flow rate of the flue gas in the stack:

$$\dot{M}_{VFlue}^{stack} = \dot{M}_{VFlue} \cdot \frac{t_J + 273.15}{273.15} = 23.49 \cdot \frac{116.16 + 273.15}{273.15} = 33.48\ m^3/s$$

Speed of the flue gas in the stack:

$$W_{Flue}^{stack} = \frac{\dot{M}_{VFlue}^{stack}}{A_{stack}} = \frac{33.48}{1.73} = 19.33\ m/s$$

Pressure loss due to friction:

$$\Delta p_L^{fric} = \lambda_{stack} \cdot \frac{H_{stack}}{d_{stack}} \cdot \frac{(W_{Flue})^2}{2} \cdot \rho_{Flue} = 0.03 \cdot \frac{16}{1.05} \cdot \frac{19.33^2}{2} \cdot 1.2772 = 109.09\ Pa$$

8.2.2 Local pressure losses at the inlet and outlet of the stake

The pressure losses at the inlet and outlet of the stack are calculated in the same manner as the calculation of pressure losses across the individual heat exchangers, where equation *(8.1)* was used.

$$\Delta p_L^{in:out} = \left(\varsigma_{stack}^{in} + \varsigma_{stack}^{out}\right) \cdot \frac{\left(W_{Flue}^{stack}\right)^2}{2} \cdot \rho_{Flue} = (1+1) \cdot \frac{19.33^2}{2} \cdot 1.2772 = 477.28\ Pa$$

8.2.3 Stack effect

The stack effect acts against the pressure losses and aids in the expulsion of cooled flue gas from the boiler. The stack effect is calculated according to equation *(8.9)*.

$$\Delta p_{eff} = H_{stack} \cdot \left(\rho_{air} - \rho_{Flue} \cdot \frac{273.15}{273.15 + t_J}\right) \cdot g \tag{8.9}$$

$$\Delta p_{eff} = 16 \cdot \left(1.275 - 1.2772 \cdot \frac{273.15}{273.15 + 116.16}\right) \cdot 9.81 = 59.53\ Pa$$

8.2.4 Overall pressure loss across the stack

The overall stack pressure loss is calculated, as seen in equation *(8.10)*, as the sum of the losses at the inlet and outlet, the loss across the stack silencer, the losses due to friction, and the stack effect opposing these losses.

$$\Delta p_L^{stack} = \Delta p_L^{fric} + \Delta p_L^{in:out} + \Delta p_L^{sil} - \Delta p_{eff} \tag{8.10}$$

$$\Delta p_L^{stack} = 109.09 + 477.28 + 200 - 59.53 = 726.84\ Pa$$

8.3 Overall Flue Gas Pressure Losses through the HRSG

The overall flue gas pressure drop is calculated according to equation *(8.11)*, as the sum of the overall pressure loss across the stack and the pressure loss across all the heat exchangers in the boiler.

$$\Delta p_{L:total} = \Delta p_L^{stack} + \Delta p_{L1} = 726.84 + 485.53 = 1212.38\ Pa \tag{8.11}$$

The total pressure loss is lower than the maximum acceptable flue gas pressure loss of 1500Pa. Thus the stack dimensions and the configuration of the heat exchangers in the boiler are acceptable.

CONCLUSION

The aim of this thesis was to design a dual pressure horizontal heat recovery steam generator (HRSG), utilizing flue gas from a natural gas burning turbine, including the sizing of the heat exchangers. The initial step in the design of this boiler was choosing the general configuration of the heat exchangers in the boiler, the number of heat exchangers and the order in which they are arraigned. The steam production rates of both the high and low pressure circuits, along with the theoretical heat transfer rates required of each heat exchanger where calculated according to the given flue gas parameters and the required steam outlet parameters.

The geometry of the first heat exchanger to come in contact with the flue gas (superheater SH_{2HP}) was selected according to the required heat transfer rate and the suggested steam and flue gas flow speeds. The inner dimensions of the HRSG casing (flue gas duct dimensions) where then determined according to the geometry of superheater SH_{2HP}. The geometry of the remaining heat heat exchangers, along with the specific geometry of the tubes and tube layout, were determined according to the dimensions of the flue gas duct and the heat transfer of requirements of each individual heat exchanger. The discrepancy between the required heat transfer rates and the actual heat transfer rates, resulting from the geometry of each heat exchanger, was kept below 2%. The parameters of the flue gas exiting each heat exchanger where accurately recalculated according to the actual heat transfer rate of the heat exchanger. Materials for the heat exchanger tubes where then selected according to the temperature of the water/steam exiting each heat exchanger.

Both the high and low pressure steam drums where sized according to the respective high and low pressure steam production rates. The steam drum loads where then calculated and verified to be lower than the limiting load value for the given steam drum pressure. The downcomer and riser tubes where also sized for both circuits to insure natural circulation of the water from the steam drum into the evaporator, through the evaporator tubes, where it becomes a water/steam mixture and back into the steam drum. The final step in the design of this dual pressure HRSG was to calculate the flue gas pressure losses across the heat exchangers and the stack, and to verify that the total pressure loss does not exceed the maximum allowable pressure loss of 1500 Pa.

The assembly drawing of the dual pressure HRSG (seen in appendix O-DP-HRSG/01) shows the main dimensions of the HRSG and its components, including the general layout of the individual heat exchangers inside the HRSG casing, which has been split into six modules for transportation purposes.

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LIST OF SYMBOLS

Abbreviations:	**Meaning:**
HRSG	heat recovery steam generator
HP	high pressure circuit
LP	low pressure circuit
ECO_{LP}	low pressure economizer
ECO_{1HP}	first heat exchanger of the high pressure economizer
ECO_{2HP}	second heat exchanger of the high pressure economizer
ECO_{3HP}	third heat exchanger of the high pressure economizer
EV_{LP}	low pressure evaporator
EV_{HP}	high pressure evaporator
SH_{LP}	low pressure superheater
SH_{1HP}	first heat exchanger of the high pressure superheater
SH_{2HP}	second heat exchanger of the high pressure superheater

Index:	**Meaning:**
$1,2,\dots,9$	points on the temperature profile graph (water/steam)
$A,B,\dots,J$	points on the temperature profile graph (flue gas)
HP	high pressure circuit
LP	low pressure circuit
ECO	low pressure economizer
$ECO1$	first heat exchanger of the high pressure economizer
$ECO2$	second heat exchanger of the high pressure economizer
$ECO3$	third heat exchanger of the high pressure economizer
EV	low pressure evaporator
$EV1$	high pressure evaporator
SH	low pressure superheater
$SH1$	first heat exchanger of the high pressure superheater
$SH2$	second heat exchanger of the high pressure superheater
Ar	argon
$CO2$	carbon dioxide
$H2O$	water
$N2$	nitrogen
$O2$	oxygen

$drum$	steam drum
$duct$	flue gas duct
$Down$	downcomer tubes
fin	tube fins
$Flue$	flue gas
$real$	actual value
$Riser$	riser tubes
$stack$	HRSG stack
$Steam$	water/steam
$tube$	heat exchanger tubes

Symbol:	**Meaning:**	**Units:**
α_c	convective heat transfer coefficient outside the heat exchanger tubes (find tubes)	$[W/(m^2K)]$
α_{c1}	convective heat transfer coefficient outside the heat exchanger tubes (smooth tubes)	$[W/(m^2K)]$
α_{in}	heat transfer coefficient inside the heat exchanger tubes (smooth tubes)	$[W/(m^2K)]$
α_{out}	heat transfer coefficient outside the heat exchanger tubes (smooth tubes)	$[W/(m^2K)]$
α_r	coefficient of heat transfer through radiation outside the tubes of the heat exchanger (smooth tubes)	$[W/(m^2K)]$
$\alpha_{r:in}$	reduced heat transfer coefficient inside heat exchanger tubes (find tubes)	$[W/(m^2K)]$
$\alpha_{r:out}$	reduced heat transfer coefficient outside heat exchanger tubes (find tubes)	$[W/(m^2K)]$
β	coefficient	$[1/m]$
ε	correction coefficient for fin fouling	$[m^2K/W]$
λ_{fin}	thermal conductivity of the fins	$[W/(m\cdot K)]$
λ_{Flue}	thermal conductivity of the flue gas	$[W/(m\cdot K)]$
λ_{stack}	coefficient of friction inside the stack	$[-]$
λ_{Steam}	thermal conductivity of the water/steam	$[W/(m\cdot K)]$
μ	coefficient characterizing fin taper	$[-]$
μ_{Steam}^{SH}	dynamic viscosity of the water/steam	$[Pa\cdot s]$
ν_{Flue}	kinematic viscosity of the flue gas	$[m^2/s]$
ν_{Steam}^{SH2}	kinematic viscosity of the water/steam	$[m^2/s]$
ξ	coefficient of the degree to which the flue gas is utilized	$[-]$
ρ_{air}	density of air	$[kg/Nm^3]$
ρ_{Flue}	densities of individual flue gas	$[kg/Nm^3]$
$\rho_{O2,\dots H2O}$	densities of individual flue gas components	$[kg/Nm^3]$

ρ_{Steam}	density of the water/steam	$[kg/m^3]$
ς	pressure loss coefficient	$[-]$
ς_{stack}^{in}	stack inlet pressure loss coefficient	$[-]$
ς_{stack}^{out}	stack outlet pressure loss coefficient	$[-]$
σ_1	lateral tube spacing relative to the outer tube diameter	$[-]$
$\sigma_2{}'$	diagonal tube spacing relative to the outer tube diameter	$[-]$
φ_σ	coefficient characterizing the relative tube spacing	$[-]$
Ψ	coefficient of thermal effectiveness (smooth tubes)	$[-]$
Ψ_{fin}	coefficient characterizing the uneven distribution of the convective heat transfer coefficient (α_c) along the surface of the fins	$[-]$
a	lateral gap between tubes, or fined tubes where applicable	$[m]$
a_{wall}	coefficient of emissivity of the tube walls	$[-]$
a_{Flue}	coefficient of emissivity of the flue gas	$[-]$
A	cross sectional area	$[m^2]$
A_{duct}	cross sectional area between the heat exchanger tubes and the walls of the flue gas duct	$[m^2]$
$A_{E:duct}$	cross sectional area of the empty flue gas duct	$[m^2]$
A_{EV}	cross sectional area of the evaporator tubes	$[m^2]$
b	coefficient of radiation decline	$[1/(m \cdot MPa)]$
b_{tri}	coefficient of radiation decline caused by triatomic gases	$[1/(m \cdot MPa)]$
c_l	correction coefficient dependent on the tube length relative to the tube diameter	$[-]$
c_m	correction coefficient for fluid flow between concentric tubes	$[-]$
c_s	correction coefficient characterizing the layout of the heat exchanger tubes (smooth tubes)	$[-]$
c_t	correction coefficient dependent on the temperature of the water/steam and tube wall temperature	$[-]$
c_z	correction coefficient for depending on the number of lateral rows	$[-]$
c_{z1}	correction coefficient depending on the number of lateral rows (smooth tubes)	$[-]$
C	constant for calculating the rate of heat loss through radiation and convection	$[-]$
d	inner diameter	$[m]$
d_e	equivalent diameter	$[m]$
D	outer diameter	$[m]$
E	coefficient characterizing the effectiveness of the fins	$[-]$
$\%Error$	discrepancy between the theoretical and actual heat transfer rates	$[\%]$
g	acceleration due to gravity	$[m/s^2]$

gap	gap between the duct wall and the outer tubes	$[m]$
h_{drum}	is the approximate steam drum height	$[m]$
h_{fin}	fin height	$[m]$
H	height of the flue gas duct	$[m]$
H_{stack}	functional stack height	$[m]$
i	enthalpy of the steam/water	$[kJ/kg]$
Δi	enthalpy difference	$[kJ/kg]$
I	enthalpy of the flue gas	$[kJ/Nm^3]$
I'	recalculated enthalpy of the flue gas	$[kJ/Nm^3]$
k	is the overall heat transfer coefficient	$[W/(m^2K)]$
$k_{1,2,3,4}$	constants depending on tube configuration	$[-]$
L	width of the flue gas duct	$[m]$
L_{drum}	is steam drum length	$[m]$
$\dot{M}_{Flue}$	mass flow rate of flue gas	$[kg/s]$
$\dot{M}_{Steam}$	mass flow rate of steam (steam production rate)	$[kg/s]$
$\dot{M}_{VFlue}$	volumetric flow rate of flue gas	$[Nm^3/s]$
$\dot{M}_{W\%}$	percentage of high pressure feedwater used to regulate outlet steam parameters (water injection)	[%]
n	number tubes	$[-]$
n_{fin}	number of fins per meter tube length	$[-]$
n_{row}	number of lateral row of tubes	$[-]$
$n_{tube/r}$	number of lateral tubes per lateral row	$[-]$
p	pressure	$[MPa]$
$p1$	flue gas pressure (assuming atmospheric)	$[MPa]$
$p_{p:tri}$	partial pressure of triatomic gases in the flue gas	$[MPa]$
$\bar{p}$	average pressure	$[MPa]$
Δp	water/steam pressure losses through the hrsg	$[MPa]$
Δp_{eff}	flue gas pressure difference due to the stack effect	$[Pa]$
Δp_L	flue gas pressure loses across individual heat exchangers	$[Pa]$
Δp_{L1}	flue gas pressure loss across all of the heat exchangers	$[Pa]$
Δp_L^{fric}	flue gas pressure loss across the stack due to friction	$[Pa]$
Δp_{Lmax}	maximum acceptable flue gas pressure loss	$[Pa]$
$\Delta p_L^{in:out}$	local flue gas pressure losses at the stack inlet and outlet	$[Pa]$
Δp_L^{sil}	flue gas pressure loss across the stack silencer	$[Pa]$
Δp_L^{stack}	the overall flue gas pressure loss across the stack	$[Pa]$
$\Delta p_{L:total}$	overall flue gas pressure losses through the hrsg	$[Pa]$

PER_{duct}	perimeter of the flue gas duct cross section	$[m]$
Pr	prandtl number	$[-]$
Q	rate of heat transfer	$[W]$
Q_N	maximum usable thermal power	$[W]$
Q_{RC}	rate of heat loss through radiation and convection	$[W]$
$Q_{RC\%}$	percentage of usable thermal power that is lost through radiation and convection	$[W]$
Re	reynolds number	$[-]$
s	effective radiation layer thickness	$[m]$
s_{fin}	fin spacing	$[m]$
s_1	lateral tube spacing	$[m]$
s_2	longitudinal tube spacing	$[m]$
s'	diagonal tube spacing	$[m]$
S_{1fin}	outer surface area of one fin	$[m^2]$
S_{out}	theoretical outer surface area of all the heat exchanger tubes needed to meet the heat transfer requirements	$[m^2]$
S_{out}^{real}	actual outer surface area of all the heat exchanger tubes	$[m^2]$
$S_{out/1m}$	the outer surface area of one tube per meter length	$[m^2]$
$S_{out/r}$	the outer surface area of all the tubes in one lateral row	$[m^2]$
$S_{in/1m}$	the inner surface area of one tube per meter length	$[m^2]$
$\frac{S_{fin}}{S_{out}}$	the fraction of the total outer surface area of the finned tubes that is made up of the fins themselves	$[-]$
$\frac{S_{out-fin}}{S_{out}}$	the fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls	$[-]$
t	temperature	$[°C]$
t_{Flue}	inlet flue gas temperature	$[°C]$
t_{FW}	feedwater temperature	$[°C]$
t_{ref}	reference temperature for material selection	$[°C]$
$t_{Steam:out}$	temperature of water/steam exiting the heat exchanger	$[°C]$
t_{wall}	average temperature of the tarnished outer tube walls	$[°C]$
th	thickness or wall thickness	$[m]$
$th_{tube(mm)}$	tube wall thickness	$[mm]$
t'	recalculated temperature	$[°C]$
$\bar{t}$	average temperature	$[°C]$
Δt	temperature difference	$[°C]$
Δt_{Ap}	approach point	$[°C]$
Δt_{ln}	logarithmic mean temperature difference	$[K]$

Δt_{Pi}	pinch point	[°C]
Δt_{safe}	safety margin	[°C]
Δt_1	temperature difference between the flue gas entering and the water/steam exiting the heat exchanger	$[K]$
Δt_2	temperature difference between the flue gas exiting and the water/steam entering the heat exchanger	$[K]$
T	the absolute temperature	$[K]$
$\bar{v}$	average specific volume	
V_{duct}	half of the steam drum volume	$[m^3]$
W	speed of flow	$[m/s]$
$W_{Steam}{}'$	water/steam speed before splitting lateral rows into sections	$[m/s]$
x	volumetric concentration	$[-]$
x_{tri}	volumetric concentration of triatomic gases in the flue gas	$[-]$
z_{drum}	steam drum load	$[kg/(s \cdot m^3)]$
z_{max}	acceptable steam drum load	$[kg/(s \cdot m^3)]$

LIST OF FIGURES

LIST OF TABLES

APPENDIX

0-DP-HRGS/01 STEAM BOILER (HRSG)