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BRNO UNIVERSITY OF TECHNOLOGY

FAKULTA STROJNÍHO INŽENÝRSTVÍ
FACULTY OF MECHANICAL ENGINEERING

ENERGETICKÝ ÚSTAV
ENERGY INSTITUTE

NÁVRH DVOUTLAKÉHO HORIZONTÁLNÍHO KOTLE NA ODPADNÍ TEPLO
PROPOSAL OF HORIZONTAL DUAL-PRESSURE HEAT RECOVERY STEAM GENERATOR

DIPLOMOVÁ PRÁCE
MASTER'S THESIS

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Specification Master's Thesis

Department:	Energy Institute
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Pursuant to Act no. 111/1998 concerning universities and the BUT study and examination rules, you have been assigned the following topic by the institute director Master's Thesis:

Proposal of horizontal dual-pressure heat recovery steam generator

Concise characteristic of the task:

The goal is propose horizontal dual-pressure heat recovery steam generator for the gas turbine on natural gas. Amount of flue gas is 50 kg/s, the flue gas. temperature is 520 °C.
Superheated steam parameters:
High pressure circuit 6 MPa: 490 °C
Low pressure circuit 0,46MPa: 180 °C
Feed water temperature: 62 °C
The volumetric exhaust gas composition: O_2 = 13.9 %; Ar = 0.9 %; N_2 = 73 %; CO_2 = 4.4 %; H_2O = 7.8 %

Goals Master's Thesis:

The goal is propose horizontal dual-pressure heat recovery steam generator for the gas turbine on natural gas.

List of literature:

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VILIMEC, L.: Stavba kotlů II. Skripta VŠB-TU Ostrava, 2008. ISBN 978-80-248-1716-3.

Deadline for submission Master's Thesis is given by the Schedule of the Academic year 2015/16

In Brno,

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

ABSTRAKT

Těžištěm této diplomové práce je zpracování návrhu dvojtlakého horizontálního kotle na odpadní teplo. Úvodní část zahrnuje tepelný výpočet, jakož i návrh uspořádání teplosměnných ploch a provedení a rozložení kotle. Jednotlivé kapitoly práce pak dle svého členění nastiňují návrh uspořádání výhřevných ploch, a to dle zadaných parametrů spalin a páry. Práce obsahuje zobrazení skutečného pilového diagramu a věnuje se i výpočtu zavodňovacích a převáděcích potrubí a bubnů. Závěrečná část pak popisuje výpočet tahové ztráty kotle. Hlavní tematickou myšlenku práce dokresluje technická dokumentace výkresu kotle.

KLÍČOVÁ SLOVA

Kotel na odpadní teplo, dvoutlaký, návrh spalinového kanálu, návrh výhřevných ploch, tlaková ztráta, tahová ztráta kotle, návrh bubnů.

ABSTRACT

The focus of this thesis is a proposal of a horizontal dual-pressure heat recovery steam generator. The introductory part includes thermal calculation, as well as a design of the layout and a design of the heat transfer surfaces and the layout of the boiler. Individual chapters are broken down according to the outline of the proposal for the arrangement of the heating surfaces, according to the parameters of the flue gas and steam. The master thesis contains a scheme of a real heat transfer temperature diagram and it also includes the calculation of connecting and downcomer pipes and drums. The final part describes the calculation of the boiler draft loss. The main idea of the thesis is accompanied by the technical documentation of the drawing of the boiler.

KEY WORDS

Heat recovery steam generator, dual pressure levels, design of flue-gas duct, design of heating surfaces, pressure loss, boiler draft loss, drums design.

BIBLIOGRAPHIC CITACION

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AFFIDAVIT

I declare that I wrote this thesis on my own without using any other sources and aids as I state in the list. I had worked indenpendently under the direction of Ing. Marek Baláš Ph.D.

Brno, date 26 th May 2016 …………………………………

Karel Slíva

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CONTENTS

INTRODUCTION

The goal of this thesis is to propose a horizontal dual-pressure level Heat Recovery Steam Generator (HRSG). HRGS can be found in virtually every chemical processing plants or metallurgical plants. It produces steam which can be used in operation such as combined-cycle mode or the cogeneration mode shown in the *Figure 1*. In each mode,the steam is used in different way. Steam in a combined-cycle mode is used to generate electricity via a steam turbine with efficiency of up to 60%. The cogeneration mode uses the steam for process applications.[4]

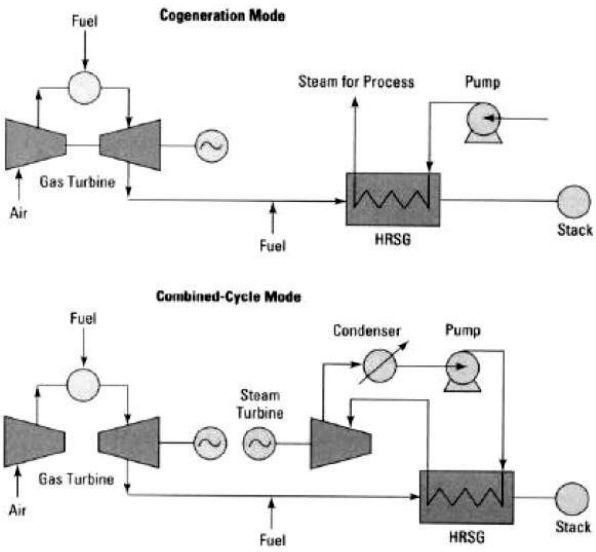

Figure 1 HRSG - combined-cycle mode and cogeneration mod[4]

HRSG are divided into several types – vertical, horizontal and once through. In the vertical HRSG flue gases flow vertically through horizontal arrangement of the heat exchanger surfaces. The advantage of the vertical arrangement is a smaller footprint but the disadvantage is the higher own energy consumption because of the use of a circulation pump. Horizontal design works on the opposite principle. The last type of HRSG is the once-through steam generator which does not use boiler drums. The arrangement of heat transfer surfaces and the direction of the flue gas stream can be horizontal or vertical. The main advantage is that phase change from water to steam can move without restriction throughout the bundle.[5]

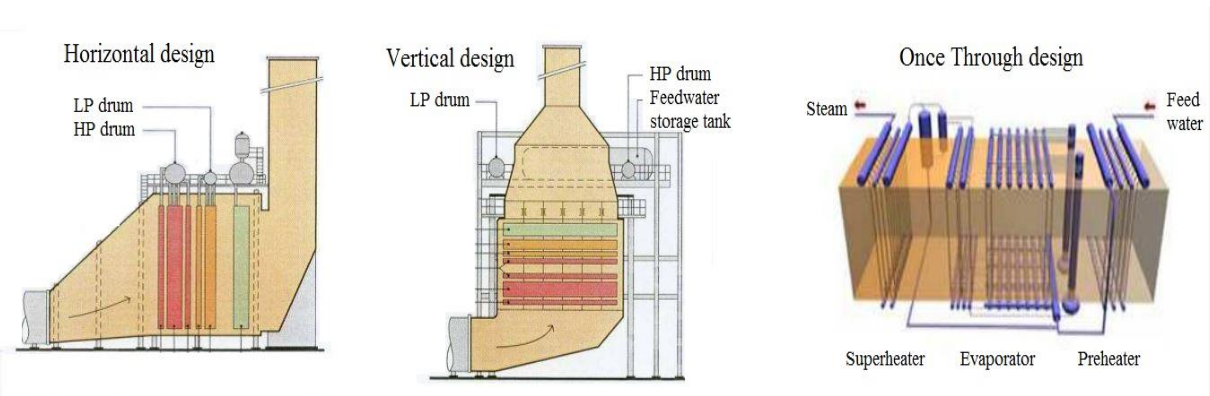

Figure 2 Horizontal, vertical and Once Through design of HRSG[5]

1. BASIC DESCRIPTION OF HORIZONTAL HRSG

Horizontal HRSG to which this work relates can be connected into a combined cycle mode with a gas turbine. HGSR is designed as a dual-pressure natural-circulation boiler. The boiler is due to better transportability divided along its length into several parts with inspection holes. As is evident from *Figure1.1*, the boiler is composed of nine heating surfaces which form the low and high pressure circuit.

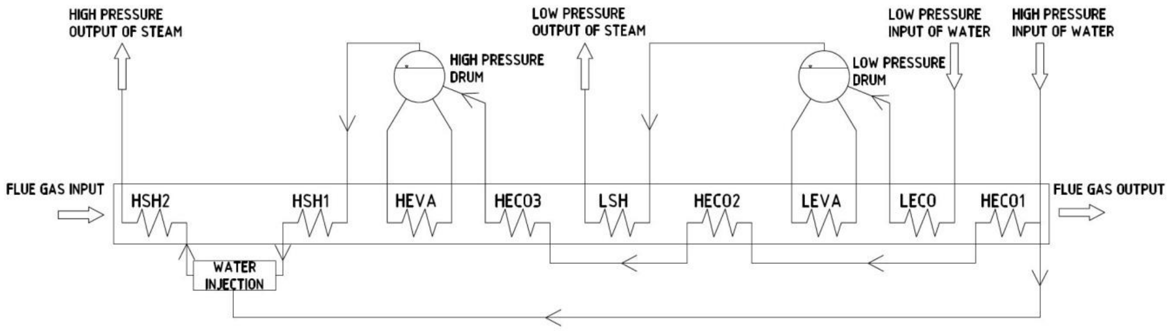

Figure 1. 1 Scheme of heat transfer surfaces in HRSG

Feed water of a temperature 62 ° C flows into the economizer, and then further into the drum subsequent by the downcomer and into an evaporator and then by cross-pipe loop into the drum, where is separated from the saturated steam and saturated liquid. At the highest part of the drum there is a valve for saturated steam offtake which continues into the superheater, where it produce superheated steam.

High pressure circuit includes economizer which is divided into 3 levels and superheater which is divided into 2 levels. For better regulation of the output temperature of the superheated steam from the high pressure circuit uses the feedwater injection valve before the second superheater.

1.1 Specified boiler parameters

<u>High – pressure circuit parameters:</u>

Output temperature $t_{HP} = 490\ °C$

Output pressure $p_{HP} = 6\ MPa$

<u>Low – pressure circuit parameters:</u>

Output temperature $t_{LP} = 180\ °C$

Output pressure $p_{LP} = 0{,}46\ MPa$

Feed watter temperature $t_{FW} = 62\ °C$

<u>Flue gas parameters:</u>

Flue gas mass flow $\dot{m} = 50\ kg/s$

Inlet flue gas temperature $t_{FG} = 520\ °C$

Volumetric flue gas composition:

O_2 volume	$x_{O_2} = 13.9\ \%$
Ar volume	$x_{Ar} = 0.9\ \%$
N_2 volume	$x_{N_2} = 73\ \%$
CO_2 volume	$x_{CO_2} = 4.4\ \%$
H_2O volume	$x_{H_2O} = 7.8\ \%$

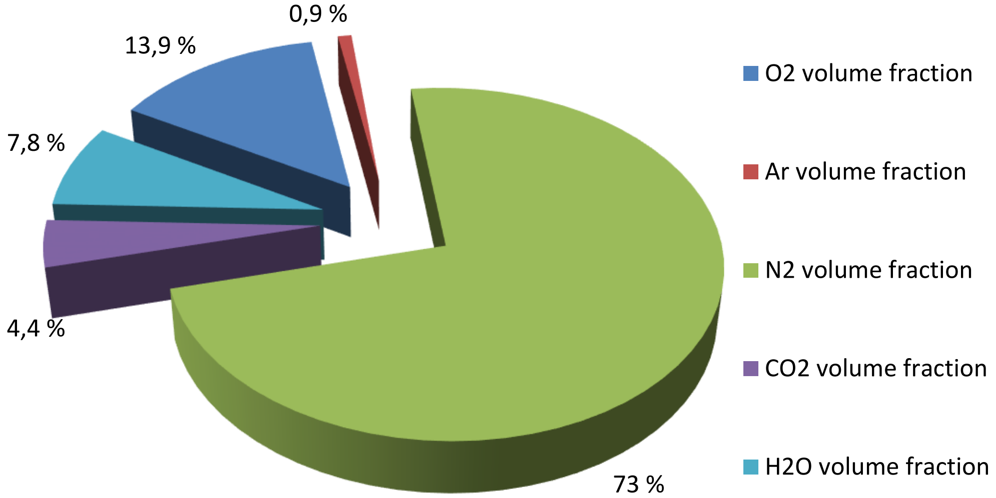

Figure 1. 2 Chart of the volumetric flue gas composition

Density of flue gas components:

O_2 density	$\rho_{O_2} = 1.4289\ kg/m^3$
Ar density	$\rho_{Ar} = 1.7839\ kg/m^3$
N_2 density	$\rho_{N_2} = 1.2505\ kg/m^3$
CO_2 density	$\rho_{CO_2} = 1.9768\ kg/m^3$
H_2O density	$\rho_{H_2O} = 0.804\ kg/m^3$

2. THERMAL CALCULATION OF BOILER

2.1 Temperature heat transfer diagram

In the first part of the thermal calculation of boiler, it is necessary to create a temperature heat transfer diagram and the layout of the heating surfaces. It is necessary to choose the value of the pinch point (Δt_{pp}) and the temperature difference between the evaporator and the economizer (Δt_n). Heating surfaces are selected in order to obtain maximum thermal energy that is contained in the flue gases therefore the superheater in the high pressure circuit is divided into two parts (QHPsh1, QHPsh2) and an economizer is devided into three parts (QHPeco1, QHPeco2, QHPeco3). The preliminary proposal of the heating surfaces is displayed in temperature heat transfer diagram in the *Figure 2.1*.

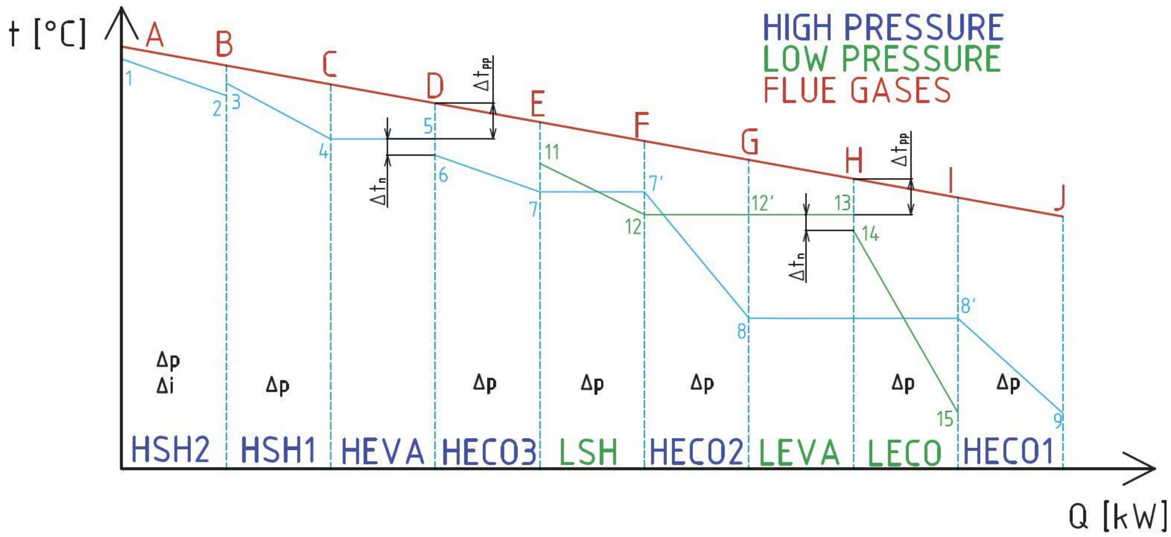

Figure 2. 1Temperature heat transfer diagram

Not all values are given and therefore it is necessary to calculate them by software XSTEAM or choose them after consulting with the supervisor. The given, selected and calculated values are displayed in *Table 2.-1* below.

Table 2. 1 The chosen amount for thermal calculation

The chosen amount for thermal calculations	Indication	Amount
The pressure loss in the high pressure superheaters (HPsh1, HPsh2)	Δpsh	$0.1\ MPa$
The pressure loss in the high pressure economizers (HPeco1, HPeco2, HPeco3)	$\Delta peco$	$0.1\ MPa$
Enthalpy drop in high pressure superheater HPsh1	Δi	$250\ kJ/kg$
The temperature difference between the flue gases and the evaporator – pinchpoint	Δt_{pp}	$10\ °C$
The temperature difference between the economizer and the evaporator – approache point	Δt_N	$5\ °C$
The temperature at point 7 of the heat transfer temperaturediagram	t_7	$220\ °C$
The temperature at point 8 of the heat transfer temperaturediagram	t_8	$145\ °C$

2.2 Parameters of water and steam in the temperature heat transfer diagram

For the calculation of these magnitudes the softwares as Microsoft EXCEL and XSTEAM was used.

2.2.1 Parameters of water and steam in the high pressure circle

Parameters in point 1:

$t_1 = t_{HP} = 490\ °C$ (specified value)

$p_1 = p_{HP} = 6\ MPa$ (specified value)

$h_1 = 3399.12\ kJ/kg$ (XSTEAM)

Parameters in point 2:

$t_2 = 389.53\ °C$ (XSTEAM)

$p_2 = p_1 + \Delta psh = 6 + 0{,}1 = 6{,}1\ MPa$

$h_2 = h_1 - \Delta h = 3399.12 - 250 = 3149.12\ kJ/kg$

Parameters in point 3:

Between the high pressure superheaters is the steam temperature controlled by injection of feed water. The value of injection was chosen to be 5% of the total amount in the high pressure circuit. In calculating the enthalpy is utilized *Figure 2-2*.

Mass balance:

$M_{PV} \cdot h_2 = 0.95 \cdot M_{PV} \cdot h_3 + 0.05 \cdot M_{PV} \cdot h_9$

$$h_3 = \frac{h_2 - 0.05 \cdot h_9}{0.95}$$

$$h_3 = \frac{3149.12 - 0.05 \cdot 264.93}{0.95}$$

$h_3 = 3300.92\ kJ/kg$

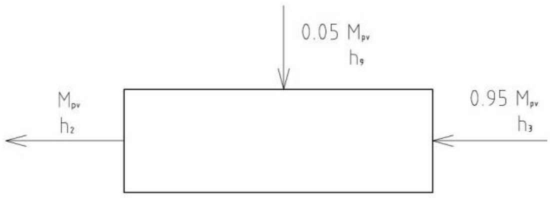

Figure 2. 2 Mass balance

$t_3 = 449.8\ °C$ (XSTEAM)

$p_3 = p_2 = 6.1\ MPa$

$h_3 = 3300.92\ kJ/kg$

Parameters in point 4:

$t_4 = 277.73\ °C$ (XSTEAM)

$p_4 = p_3 + \Delta p = 6.1 + 0.1 = 6.2\ MPa$

$i_4 = 2782.33\ kJ/kg$ (XSTEAM)

Parameters in point 5:

$t_5 = t_4 = 277.73\ °C$

$p_5 = p_4 = 6.2\ MPa$

$h_5 = 1224.86\ kJ/kg$ (XSTEAM)

Parameters in point 6:

$t_6 = t_5 - \Delta t_N = 277.73 - 5 = 272.73\ °C$

$p_6 = p_5 = 6.2\ MPa$

$h_6 = 1198.85\ kJ/kg$ (XSTEAM)

Parameters in point 7:

$t_7 = 220\ °C$ (specified value)

$p_7 = p_6 + \Delta p = 6.2 + 0.1 = 6.3\ MPa$

$h_7 = 944.76\ kJ/kg$ (XSTEAM)

Parameters in point 8:

$t_8 = 145\ °C$ (specified value)

$p_8 = p_7 + \Delta p = 6.3 + 0.1 = 6.4\ MPa$

$h_8 = 614.50\ kJ/kg$ (XSTEAM)

Parameters in point 9:

$t_9 = t_{NV} = 62\ °C$ (specified value)

$p_9 = p_8 + \Delta p = 6.4 + 0.1 = 6.5\ MPa$

$h_9 = 264.93\ kJ/kg$ (XSTEAM)

2.2.2 Parameters of water and steam in the low pressure circle

Parameters in point 11:

$t_{11} = t_{LP} = 180\ °C$ (specified value)

$p_{11} = p_{LP} = 0.46\ MPa$ (specified value)

$h_{11} = 2814.95\ kJ/kg$ (XSTEAM)

Parameters in point 12:

$t_{12} = 156.15\ °C$ (XSTEAM)

$p_{12} = p_{11} + \Delta p = 0.46 + 0.1 = 0.56\ MPa$

$h_{12} = 2753.12\ kJ/kg$ (XSTEAM)

Parameters in point 13:

$t_{13} = t_{12} = 156.15\ °C$

$p_{13} = p_{12} = 0.56\ MPa$

$h_{13} = 658.88\ kJ/kg$ (XSTEAM)

Parameters in point 14:

$t_{14} = t_{13} - \Delta t_N = 156.15 - 5 = 151.15\ °C$

$p_{14} = p_{13} = 0.56\ MPa$

$h_{14} = 637.26\ kJ/kg$ (XSTEAM)

Parameters in point 15:

$t_{15} = t_{FW} = 62\ °C$ (specified value)

$p_{15} = p_{14} + \Delta p = 0.56 + 0.1 = 0.66\ MPa$

$h_{15} = 260.06\ kJ/kg$ (XSTEAM)

2.3 Flue gas parameters

2.3.1 Flue gas volume flow

Using the values from the *Table 2. 2* and volumetric flue gases composition is possible to calculate density of the flue gases $\rho_{FG}[\ kg/m^3]$.

Table 2. 2 Specific density of flue gas for 1 m3 at 0 ° C and 0.101 MPa[1]

[kg/Nm³]	CO_2	H_2O	N_2	O_2	Ar
ρ	1.9768	0.804	1.2505	1.4289	1.7839

$$\rho_{FG} = x_{CO_2} \cdot \rho_{CO_2} + x_{H_2O} \cdot \rho_{H_2O} + x_{O_2} \cdot \rho_{O_2} + x_{N_2} \cdot \rho_{N_2} + x_{Ar} \cdot \rho_{Ar}$$

$$\rho_{FG} = 0.044 . 1.9768 + 0.078 \cdot 0.804 + 0.139 \cdot 1.4289 + 0.73 \cdot 1.2505 + 0.009 \cdot 1.7839$$

$$\rho_{FG} = 1.2772\ kg/m^3$$

From the calculated density of of flue gases $\rho_{FG}[\ kg/m^3]$ and specified mass flow $\dot{m}\ [\ kg/s\ ]$ it is possible to determine the flue gases volume flow $M_{FG}\ [\ m^3/s\ ]$.

$$M_{FG} = \frac{\dot{m}}{\rho_{FG}} = \frac{50}{1.2772} = 39.15\ m^3/s$$

2.3.2 Flue gas enthalpy

At first it is necessary to determine the flue gas enthalpy for each temperature. Example of calculation of enthalpy for the 100 ° C are shown bellow. *Table 2. 3* shows the calculated enthalpy values for the other temperatures.

The general formula for calculating the flue gas enthalpy:

$$H_{FG}^t = x_{CO_2} \cdot H_{CO_2}^t + x_{H_2O} \cdot H_{H_2O}^t + x_{O_2} \cdot H_{O_2}^t + x_{N_2} \cdot H_{N_2}^t + x_{Ar} \cdot H_{Ar}^t\ [kJ/m^3]$$

Calculation of enthalpy for 100 °C:

$$H_{FG}^{100} = x_{CO_2} \cdot H_{CO_2}^{100} + x_{H_2O} \cdot H_{H_2O}^{100} + x_{O_2} \cdot H_{O_2}^{100} + x_{N_2} \cdot H_{N_2}^{100} + x_{Ar} \cdot H_{Ar}^{100}$$

$$H_{FG}^{100} = 0.044 \cdot 170 + 0.078 \cdot 150 + 0.139 \cdot 132 + 0.73 \cdot 130 + 0.009 \cdot 93$$

$$H_{FG}^{100} = 133.265\ kJ/Nm^3$$

Table 2. 3 Specific enthalpy of the flue gas for 1 m3 at 0 ° C and 0.101 MPa[1]

t	CO_2	H_2O	O_2	N_2	Ar	H_{FG}
[°C]	[kJ/Nm^3]	[kJ/Nm^3]	[kJ/Nm^3]	[kJ/Nm^3]	[kJ/Nm^3]	[kJ/Nm^3]
100	170	150	132	130	93	133.27
200	357	304	267	260	186	268.01
300	559	463	407	392	278	405.95
400	772	626	551	527	372	547.44
500	994	795	699	666	465	693.27
600	1225	969	850	804	557	839.57

2.3.2.1 Flue gas enthalpy in point A

The temperature at point A is equal to the inlet flue gas temperature $t_{FG} = 520\,°C$. The enthalpy at point A is determined by interpolation from *Table 2. 3.*

$$H_A = H_{FG}^{500} + (T_A - 500) \cdot \frac{(H_{FG}^{600} - H_{FG}^{500})}{(600 - 500)} = 693{,}27 + 30 \cdot \frac{(839.57 - 693.27)}{100}$$

$$H_A = 722.53\ kJ/m^3$$

2.3.2.2 Flue gas enthalpy in point D

The temperature at the point D is determined by pinchpoint Δt_{pp} $[°C]$ and the evaporation temperature in the evaporator Δt_5 $[°C]$.

$$T_D = t_5 + \Delta t_{PP} = 277.73 + 10 = 287.73\ °C$$

The enthalpy at point D is determined by interpolation from *Table 2. 3.*

$$H_D = H_{FG}^{200} + (T_D - 200) \cdot \frac{(H_{FG}^{300} - H_{FG}^{200})}{(300 - 200)} = 268.01 + 87.73 \cdot \frac{(405.95 - 268.01)}{100}$$

$$H_D = 389.02\ kJ/m^3$$

2.3.2.3 Flue gas enthalpy in point E

Enthalpy at point E depends on the heat transferred by flue gases betwen points D and E.

$$Q_{HPeco3} = Q_{6-7} = Q_{D-E} \cdot (1 - L_R)$$

Heat transferred by flue gases betwen points D and E

$$Q_{D-E} = M_{FG} \cdot (H_D - H_E)$$

From these two equations we can express the enthalpy in point E:

$$Q_{HPeco3} = M_{FG} \cdot (H_D - H_E) \cdot (1 - L_R) \Rightarrow$$

$$H_E = H_D - \frac{Q_{HPeco3}}{M_{FG} \cdot (1 - L_R)} \qquad (2\text{–}3)$$

$$H_E = 389.02 - \frac{1397.62}{39.15(1 - 0.004146)}$$

$$H_E = 353.17\ kJ/m^3$$

The temperature at point E is determined by interpolation from *Table 2. 3.*

$$T_E = 200 + (300 - 200) \cdot \frac{(H_E - H_{FG}^{200})}{(H_{FG}^{300} - H_{FG}^{200})} = 200 + 100 \cdot \frac{(353.17 - 268.01)}{(405.95 - 268.01)}$$

$$T_E = 261.74\ °C$$

2.3.2.4 Flue gas enthalpy in point H

The temperature at the point H is determined by pinchpoint Δt_{pp} $[°C]$ and the temperature t_{13} $[°C]$.

$$T_H = t_{13} + \Delta t_{PP} = 156.15 + 10 = 166.15\ °C$$

The enthalpy at point H is determined by interpolation from *Table 2. 3.*

$$H_H = H_{FG}^{100} + (T_H - 100) \cdot \frac{(H_{FG}^{200} - H_{FG}^{100})}{(200 - 100)} = 133.27 + 66.15 \cdot \frac{(268.01 - 133.27)}{100}$$

$$H_H = 222.4\ kJ/m^3$$

2.3.2.5 Flue gas enthalpy in point B

The enthalpy at point B is determined by formula *(2–3)*

$$H_B = H_A - \frac{Q_{HPsp2}}{M_{FG}(1 - L_R)} = 722.53 - \frac{1447.5}{39.15(1 - 0.004146)} = 685.4\ kJ/m^3.$$

The temperature at point B is determined by interpolation from *Table 2. 3*

$$T_B = 400 + (500 - 400) \cdot \frac{(H_B - H_{FG}^{400})}{(H_{FG}^{500} - H_{FG}^{400})} = 400 + 100 \cdot \frac{(685.54 - 547.44)}{(693.27 - 547.44)} = 494.6\ °C.$$

2.3.2.6 Flue gas enthalpy in point C

The enthalpy at point C is determined by formula *(2–3)*

$$H_C = H_B - \frac{Q_{HPsh1}}{M_{FG}(1 - L_R)} = 685.54 - \frac{2852.5}{39.15(1 - 0.004146)} = 612.29\ kJ/m^3.$$

The temperature at point C is determined by interpolation from *Table 2. 3*

$$T_C = 400 + (500 - 400) \cdot \frac{(H_C - H_{FG}^{400})}{(H_{FG}^{500} - H_{FG}^{400})} = 400 + 100 \cdot \frac{(612.29 - 547.44)}{(693.27 - 547.44)} = 444.47\ °C.$$

2.3.2.7 Flue gas enthalpy in point D

The enthalpy at point D is determined by formula *(2–3)*

$$H_D = H_C^{REAL} - \frac{Q_{HPeva}}{M_{FG}(1 - L_R)} = 612.878 - \frac{8709.93}{39.15(1 - 0.004146)} = 389.48\ kJ/m^3.$$

The temperature at point D is determined by interpolation from *Table 2. 3*

$$T_D = 200 + (300 - 200) \cdot \frac{(H_D - H_{FG}^{200})}{(H_{FG}^{300} - H_{FG}^{200})} = 200 + 100 \cdot \frac{(389.48 - 268.01)}{(405.95 - 268.01)} = 288.06\ °C.$$

2.3.2.8 Flue gas enthalpy in point F

The enthalpy at point F is determined by formula *(2–3)*

$$H_F = H_E^{REAL} - \frac{Q_{LPsh}}{M_{FG}(1-L_R)} = 353.802 - \frac{93.36}{39.15(1-0.004146)} = 351.41\,\frac{kJ}{m^3}$$

The temperature at point F is determined by interpolation from *Table 2. 3*

$$T_F = 200 + (300-200)\cdot\frac{(H_F - H_{SP}^{200})}{(H_{SP}^{300} - H_{SP}^{200})} = 200 + 100\cdot\frac{(353.80 - 268.01)}{(405.95 - 268.01)} = 260.46\,°C$$

2.3.2.9 Flue gas enthalpy in point G

The enthalpy at point G is determined by formula *(2–3)*

$$H_G = H_F^{REAL} - \frac{Q_{HPeco2}}{M_{FG}(1-L_R)} = 351.423 - \frac{1816.6}{39.15(1-0.004146)} = 304.829\,kJ/m^3$$

The temperature at point G is determined by interpolation from *Table 2. 3*

$$T_G = 200 + (300-200)\frac{(H_G - H_{FG}^{200})}{(H_{FG}^{300} - H_{FG}^{200})} = 200 + 100\cdot\frac{(304.829 - 268.01)}{(405.95 - 268.01)}$$

$$T_G = 226.695\,°C$$

2.3.2.10 Flue gas enthalpy in point I

The enthalpy at point I is determined by formula *(2–3)*

$$H_I = H_H^{REAL} - \frac{Q_{LPeco}}{M_{FG}(1-L_R)} = 222.17 - \frac{569.57}{39.15(1-0.004146)} = 207.56\,kJ/m^3$$

The temperature at point I is determined by interpolation from *Table 2. 3*

$$T_I = 100 + (200-100)\frac{(H_I - H_{FG}^{100})}{(H_{FG}^{200} - H_{FG}^{100})} = 100 + 100\cdot\frac{(207.56 - 133.27)}{(268.01 - 133.27)} = 155.14\,°C$$

2.3.2.11 Flue gas enthalpy in point J

The enthalpy at point J is determined by formula *(2–3)*

$$H_J = H_I^{REAL} - \frac{Q_{HPeco1}}{M_{FG}(1-L_R)} = 207.48 - \frac{1922.81}{39.15(1-0.004146)} = 158.17\,kJ/m^3$$

The temperature at point J is determined by interpolation from *Table 2. 3*

$$T_J = 100 + (200-100)\frac{\left(H_J - H_{FG}^{100}\right)}{(H_{FG}^{200} - H_{FG}^{100})} = 100 + 100\cdot\frac{(158.17 - 133.27)}{(268.01 - 133.27)} = 118.48\,°C$$

2.4 Heat radiation loss

Radiation loss represents heat lost to the surroundings due to the warm surfaces of a boiler. This loss depends on the size of the boiler its insulation properties and maximal output (small boiler has a proportionately larger percentage loss than large boiler). [3] [6]

Maximal heat output:

$$Q_A = H_A \cdot M_{FG} = 722.53 \cdot 39.15 = 28\,287\ kW = 28.29\ MW$$

Radiation loss:

$$Q_{RS} = {Q_A}^{0,7} \cdot C_{gas} = 28.29^{0,7} \cdot 0.0113 = 0.117272\ MW = 117.27\ kW,$$

$C_{gas} = 0.0113$ (constant used for boilers for natural gas and liquid fuels)

Relative radiation loss:

$$L_R = \frac{Q_{RS}}{Q_A} = \frac{0.117272}{39.15} = 0.004146 = 0.4146\ \%$$

2.5 High pressure performance of the boiler and heat transferred in the heating surfaces

2.5.1 High pressure performance of the boiler

For the calculation will use the calculated enthalpy and temperature at point D and A in chapters 2.3.2.1, 2.3.2.2

$$Q_{A-D} = M_{FG} \cdot (H_A - H_D) = 39.15 \cdot (722.53 - 389.02) = 13\,056.92\ kW$$

Heat transferred between the points 1 and 6 in the temperature heat transfer diagram:

$$Q_{1-6} = Q_{A-D} \cdot (1 - L_R) = 13\,056.92 \cdot (1 - 0.004146) = 13\,002.79\ kW$$

High pressure performance of the boiler:

$$Q_{1-6} = M_{HP}[(h_1 - h_2) + 0.95 \cdot (h_2 - h_6) + 0.05 \cdot (h_2 - h_9)]$$

$$M_{HP} = \frac{Q_{1-6}}{(h_1 - h_2) + 0.95 \cdot (h_2 - h_6) + 0.05 \cdot (h_2 - h_9)} =$$

$$M_{HP} = \frac{13\,002.79}{(3399.12 - 3149.12) + 0.95 \cdot (3149.12 - 1198.85) + 0.05 \cdot (3149.12 - 264.93)}$$

$$M_{HP} = 5.79\ kg/s$$

2.5.2 Heat transferred in the high pressure heating surfaces

Thermal output in high-pressure superheater HPsh2:

$$Q_{\mathrm{HPsh2}} = M_{VP} \cdot (h_1 - h_2) = 5.79 \cdot (3399.12 - 3149.12) = 1447.5\ kW$$

Thermal output in high-pressure superheater HPsh1:

$$Q_{HPsh1} = 0.95 \cdot M_{VP} \cdot (h_3 - h_4) = 0.95 \cdot 5.79 \cdot (3300.92 - 2782.33) = 2\,852.5\ kW$$

Thermal output in high-pressure evaporater HPeva:

$$Q_{HPeva} = 0,95 \cdot M_{VP} \cdot (h_4 - h_6) = 0.95 \cdot 5.79 \cdot (2782.33 - 1198.85) = 8\,709.93\ kW$$

Thermal output in high-pressure econoizer HPeco3:

$$Q_{HPeco3} = 0.95 \cdot M_{VP} \cdot (h_6 - h_7) = 0.95 \cdot 5.79 \cdot (1198.85 - 944.76) = 1\,397.62\ kW$$

Thermal output in high-pressure econoizer HPeco2:

$$Q_{HPeco2} = 0.95 \cdot M_{VP} \cdot (h_7 - h_8) = 0.95 \cdot 5.79 \cdot (944.76 - 614.5) = 1\,816.6\ kW$$

Thermal output in high-pressure econoizer HPeco1:

$$Q_{HPeco1} = 0.95 \cdot M_{VP} \cdot (h_8 - h_9) = 0.95 \cdot 5.79 \cdot (614.5 - 264.93) = 1\,922.81\ kW$$

2.5.3 Checking of the boiler thermal balance in high pressure circle

The tolerance for heat transferred is 0.5%.

Heat difference trasfered to steam:

$$\Delta Q_{HP} = Q_{1-6} - (Q_{Hsh2} + Q_{Hsh1} + Q_{eva}) = 13002.79 - (1447.5 + 28525 + 8709.93) =$$

$$\Delta Q_{HP} = 7.14\ kW$$

Checking of the heat transferred:

$$\Delta_{HQ} = \frac{\Delta Q_{HP}}{Q_{1-6}} = \frac{7.14}{13002.79} = 0.0549\ \% \ < \ 0{,}5\ \%$$

As shown the tolerance condition was met and the calculation is correct.

2.6 Low pressure performance of the boiler and heat transferred in the heating surfaces

2.6.1 Low pressure performance of the boiler

The heat supplied by the flue gas between the points E and H in the temperature heat transfer diagram:

$$Q_{E-H} = M_{FG} \cdot (H_E - H_H) = 39.15 \cdot (353.17 - 222.4) = 5\,119.65\ kW$$

Heat transferred between the points 11 and 14 in the temperature heat transfer diagram:

In the calculation we have to take into account the location of high pressure economizer (HPeco2) in the low-pressure circuit. It is therefore necessary to subtract the amount of heat in the high pressure economizer.

$$Q_{11-14} = Q_{E-H} \cdot (1 - L_R) - Q_{HPeco2} = 5119.65 \cdot (1 - 0.004146) - 1816.6 = 3281.82\ kW$$

Low pressure performance of the boiler:

$$Q_{11-14} = M_{LP} \cdot (h_{11} - h_{14}) \Rightarrow$$

$$M_{LP} = \frac{Q_{11-14}}{(h_{11} - h_{14})} = \frac{3281.82}{(2814.95 - 637.26)}$$

$$M_{LP} = 1.51\ kg/s$$

2.6.2 Heat transferred in the low pressure heating surfaces

Thermal output in low-pressure superheater LPsh:

$$Q_{LPsh} = M_{LP} \cdot (h_{11} - h_{12}) = 1.51 \cdot (2814.82 - 2753.12) = 93.36\ kW$$

Thermal output in low-pressure evaporater LPeva:

$$Q_{LPeva} = M_{LP} \cdot (h_{12} - h_{14}) = 1.51 \cdot (2753.12 - 637.26) = 3\ 194.95\ kW$$

Thermal output in low-pressure economizer LPeco:

$$Q_{LPeco} = M_{LP} \cdot (h_{14} - h_{15}) = 1.51 \cdot (637.26 - 260.06) = 569.57\ kW$$

2.6.3 Checking of the boiler thermal balance in low presssure circle

The tolerance for heat transferred is 0.5%.

Heat difference trasfered to steam:

$$\Delta Q_{LP} = Q_{11-14} - (Q_{Lsh} + Q_{Leva}) = 3281.82 - (93.36 + 3194.95)$$

$$\Delta Q_{LP} = 6.49\ kW$$

Checking of the heat transferred:

$$\Delta_{LQ} = \frac{\Delta Q_{LP}}{Q_{11-14}} = \frac{6.49}{3281.82} = 0.197\ \% \ < \ 0.5\ \%$$

As shown the tolerance condition was met and the calculation is correct.

2.7 List of calculated values

In this chapter the tables of importatant calculated values, for example values of heat transferred in heat transfer surfaces, are shown.

Table 2. 4 The values o heat transferred in heat transfer surfaces

Heat transfer surface	Indication	Amount	Unit
Heat transferred in HPsh2	Q_{HPsh2}	1447.5	[kW]
Heat transferred in HPsh1	Q_{HPsh1}	2852.5	[kW]
Heat transferred in HPeva	Q_{HPeva}	8709.93	[kW]
Heat transferred in HPeco3	Q_{HPeco3}	1397.62	[kW]
Heat transferred in HPeco2	Q_{HPeco2}	1816.6	[kW]
Heat transferred in HPeco1	Q_{HPeco1}	1922.81	[kW]
Heat transferred in LPsh	Q_{LPsh}	93.36	[kW]
Heat transferred in LPeva	Q_{LPeva}	3194.95	[kW]
Heat transferred in LPeco	Q_{LPeco}	569.57	[kW]

Table 2. 5 Important calculated values

Calculated values	Indication	Amount	Unit
Relative radiation loss	L_R	0.004146	[-]
High pressure boiler output	M_{HP}	5.79	$[kg/s]$
Low pressure boiler output	M_{LP}	1.51	$[kg/s]$
Flue gas density	ρ_{FG}	1.2772	$[kg/m^3]$
Flue gas volume flow	M_{FG}	39.15	$[m^3/s]$

3. DESIGN OF FLUE GAS DUCT

The design of flue gas duct is necessary choose the dimensions one of heat transfer surface. Choosen heat transfer surface is first high pressure superheater - HPsh2.

In first step of flue gas duct design velocity of superheated stean and dimensions of fin tubes is detrmined. Next step contains calculation of tube pitch, amount of fin tubes in one longitudial line and control calculation of superheated stean velocity inside the fin tubes. From tube pinch and amount of fin tube is possible to calculate a width of flue gas duct. Next value which is necessary to determine is the height of flue gas duct. For calculation of flue gas duct height it is necessary to calculate real flue gas volume flow for mean temperature value and the flow velocity of flue gas.

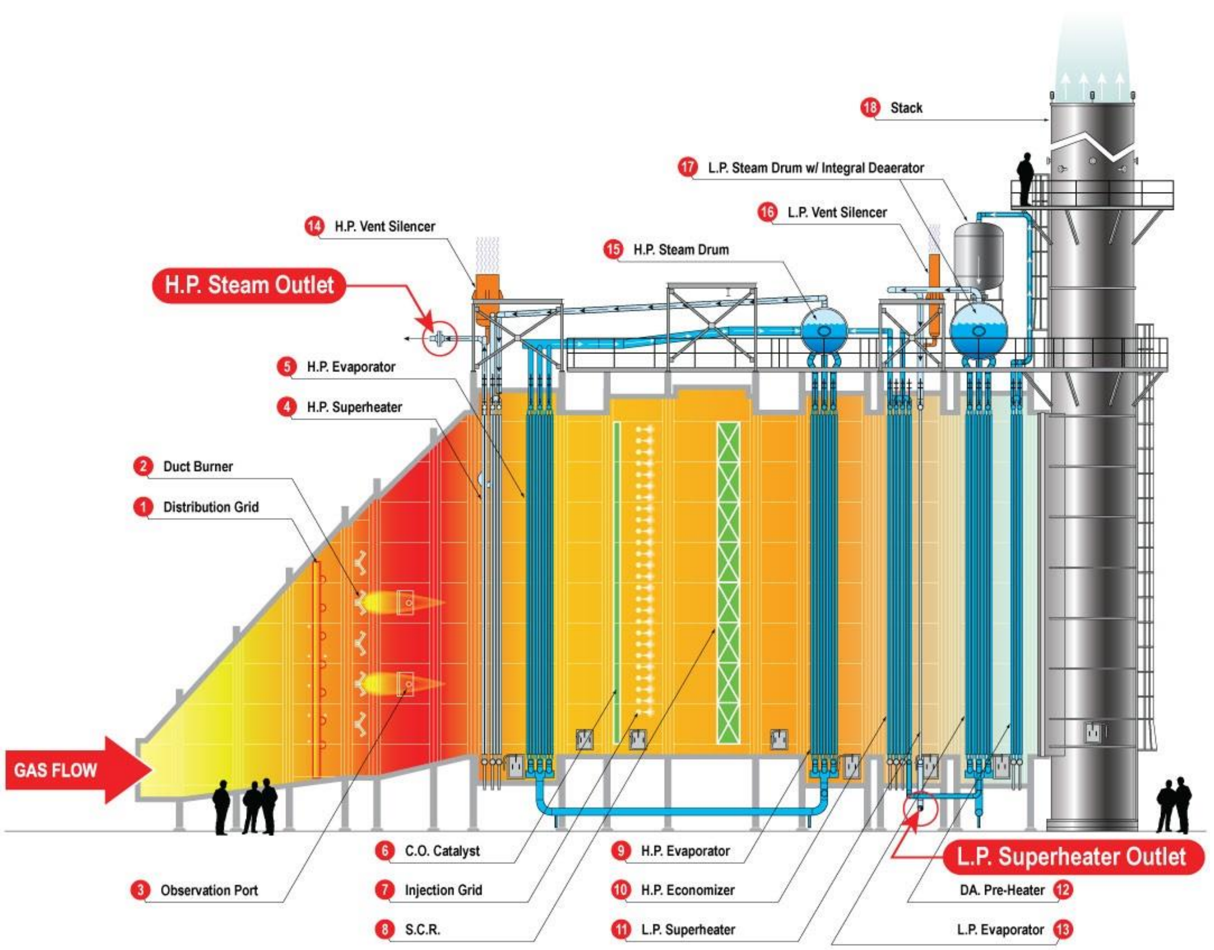

Figure 3. 1 Conception of flue gas duct[7]

3.1 Fin tube design of high pressure superheater HPsh2

The fin tube dimensions are shown in *Table 3. 1* and the drawing of fin tube is shown in *Figure 3. 2.*

Table 3. 1 Fin tube dimensions of HPsh2

Tube dimensions	Indications	Amount	Unit
Outer diameter	D	31.8	[mm]
Wall thickness	t	4.5	[mm]
Inner diameter	d	22.8	[mm]
Fins height	h_f	15	[mm]
Fins thickness	t_f	1	[mm]
Fins per meter	n_f	215	[1/m]
Fins pitch	p_f	4.65	[mm]
Outer fin diameter	D_f	61.8	[mm]

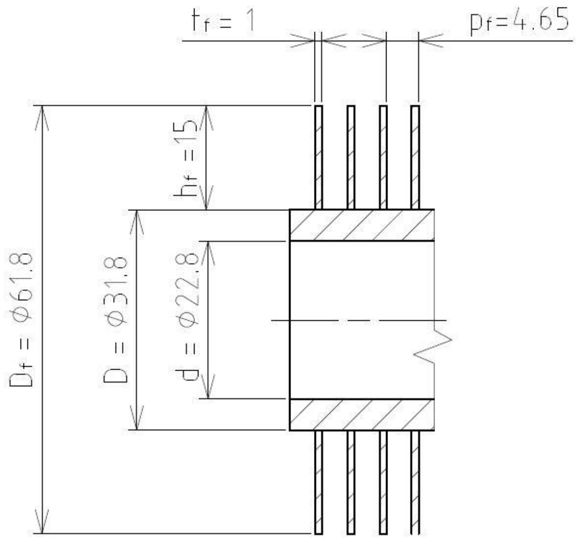

Figure 3. 2 Fin tube dimensions of HPsh2

3.2 Amount of fin tubes in one line of HPsh2

Firstly the velocity of steam in the tube was selected. The speed should be in the range 15-25m/s. Choosen value is: $v_s = 20\ m/s$.

In the next step is important to determine the mean specific volume of steam which is given by average temperature and pressure in HPsh2. The mean specific volume of steam is then calculated by the softwar XSTEAM.

$$t_{1-2} = \frac{t_1 + t_2}{2} = \frac{490 + 389.53}{2} = 439.765\ °C$$

$$p_{1-2} = \frac{p_1 + p_2}{2} = \frac{6 + 6.1}{2} = 6.05\ MPa$$

$v_{1-2} = f(t_{1-2}, p_{1-2}) = 0.050766\ m^3/kg$

The total cross section of the tubes is determined by the formula:

$$v_s = \frac{M_s \cdot v_{sp}}{S_s}\ [m/s] \tag{3–1}$$

M_s $[kg/s]$ – steam mass flow in the heating surface

v_{sp} $[m^3/kg]$ – mean specific volume of steam in the heating surface

S_s $[m^2]$ – total cross section of tubes

$S_s = \frac{M_{HP} \cdot v_{1-2}}{v_s} = \frac{5.79 \cdot 0.050766}{15} = 0.0147\ m^2.$

Amount of fin tubes in one longitudial line is determined by the formula:

$$S_s = \frac{\pi \cdot d^2}{4} \cdot n_{TU}\ [m^2], \tag{3–2}$$

d $[m]$ – inner diameter

n_{TU} $[-]$ – amount of tubes in one longitudinal line

$n_{TU} = \frac{4 \cdot S_s}{\pi \cdot d^2} = \frac{4 \cdot 0.0147}{\pi \cdot 0.0228^2} = 35.99\ [-]$

The real tubes amount in a single longitudinal line: $n_{TU} = 35.99\ => \ 36.$

The real flow velocity of steam – formulas (3-1;3-2)

$v_s^{REAL} = \frac{M_s \cdot v_s}{S_s} = \frac{M_{HP} \cdot v_{1-2}}{\frac{\pi \cdot d^2}{4} \cdot n_{TU}} = \frac{4 \cdot M_{VP} \cdot v_{1-2}}{\pi \cdot d^2 \cdot n_{TU}} = \frac{4 \cdot 5.79 \cdot 0{,}050766}{\pi \cdot 0.0228^2 \cdot 36} = 20\ m/s$

3.3 Volumetric flow of flue gases in HPsh2

Selected flue gas velocity $v_{FG} = 10\ m/s$.

For the calculation we will use already calculated enthalpy and temperature of the flue gases at point B.

The average temperature of the flue gas stream:

$T_{A-B} = \frac{T_A + T_B}{2} = \frac{520 + 494.6}{2} = 507.3\ °C.$

A real volumetric flow takes into account the flue gas temperature:

$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{A-B} + 273.15}{27.,15} = 39.15 \cdot \frac{507.3 + 273.15}{273.15} = 111.86\ m^3/s.$

The flow area of the flue gas duct is determined by the formula:

$$S_{DUCT} = \frac{M_{FG}^{REAL}}{v_{FG}}\ [m^2] \tag{3–3}$$

M_{FG}^{REAL} $[m^3/s]$ – real flue gases volumetric flow

v_{FG} $[m/s]$ – flue gas velocity

$$S_{DUCT} = \frac{M_{FG}^{REAL}}{v_{FG}} = \frac{111.86}{10} = 11.186\ m^2$$

3.4 Dimensions of flue gas duct

For Calculation width of flue gas duct is important to calculate a lateral pitch. Latera pitch consist of outer rib diameter and the distance betwen the tube $a = 10mm$.

$p_1 = D_f + a = 61.8 + 10 = 71.8\ mm.$

Width of flue gas duct is determined by the formula:

$$w = \frac{p_1}{2} + (n_{TU} - 1) \cdot p_1 + \frac{p_1}{2} + \frac{p_1}{2} = \left(n_{TU} + \frac{1}{2}\right) \cdot p_1\ [m] \qquad (3\text{–}4)$$

p_1 $[m]$ – lateral pitch

$w = \left(n_{TU} + \frac{1}{2}\right) \cdot p_1 = \left(36 + \frac{1}{2}\right) \cdot 71.8 = 2.62\ mm.$

$\boldsymbol{w \cong 2.7\ m}$

The height of the flue gas duct is determined by the formula:

$$S_{DUCT} = w \cdot h - h \cdot D \cdot n_{TU} - 2 \cdot h_f \cdot t_f \cdot h \cdot n_f \cdot n_{TU}\ [m^2] \qquad (3\text{–}5)$$

h $[m]$ – height of the flue gas duct

D $[m]$ – outer diameter

h_f $[m]$ – fins height

t_f $[m]$ – fins thickness

n_f $[1/m]$ – fins per meter

$$h = \frac{S_{DUCT}}{w - n_{TU} \cdot (D + 2 \cdot h_f \cdot t_f \cdot n_f)} = \frac{11.186}{2.7 - 36 \cdot (0{,}0318 + 2 \cdot 0.015 \cdot 0.001 \cdot 215)}$$

$h = 8.45\ m.$

$\boldsymbol{h \cong 8.5\ m}$

A real cross section of the the flue gas duct :

$S_{DUCT}^{REAL} = H \cdot \left(w - n_{TU} \cdot (D + 2 \cdot h_f \cdot t_f \cdot n_f)\right)$

$S_{DUCT}^{REAL} = 8..5 \cdot (2.7 - 36 \cdot (0.0318 + 2 \cdot 0.015 \cdot 0.001 \cdot 215)) = 11.246\ m^2$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{111.86}{11.246} = 9.95\ m/s$$

3.5 List of calculated values

Table 3. 2 Calculated values and main dimensions of flue-gas duct

Calculated values	Indication	Amount	Unit
Amount of tubes in one line	n_{TU}	36	[-]
Real steam velocity	v_s^{REAL}	20	$[m/s]$
Real flue gases volumetric flow	M_{FG}^{REAL}	111.86	$[m^3/s]$
Width of flue gas duct	w	2.7	$[m]$
Height of the flue gas duct	h	8.5	$[m]$
Real flue gas velocity	v_{FG}^{REAL}	9.95	$[m/s]$

4. DESIGN OF HEAT TRANSFER SURFACES

Design of heat transfer surfaces must comply with the dimensions of flue gas duct which was designed in the previous chapter. The calculation procedure for all heat transfer surfaces are nearly identical and differ only in some parts. The exception is a low-pressure superheater, which is composed of smooth tubes

Specific tube dimensions are chosen according to a table of produced dimensions (standard ČSN EN 10216-2). I was the outer diameter D, wall tube thickness t and the inner diameter d will be calculated as $d = D - 2 \cdot t$. Fin height h_f is for evaporators between 10-19 mm and for economizers and superheater between 10-15 mm. Outlet fin diameter D_f will be calculated as $D_f = D + 2 \cdot h_f$. Amount of fins per meter n_f between 150-250 was used. Fin pitch p_f is the inverse of amount of fins per meter n_f.

According to the dimension of the finned or smooth tube and the distance between the tubes a is determined the lateral pitch p_1 and the number of tubes n_{TU} in a one longitudinal line. Subsequently the steam velocity v_s in the tubes was controled. Further is necessary to determine the flue gas velocity v_{FG} for heat transfer serface and the size of the flow area S.

In the next step the number of longitudinal line n_{LL} will be determined. The number of longitudinal line is determined by calculating of the heat transfer coefficient on the flue gas side α_{1r} and the heat transfer coefficient on the steam side α_{2r}. Further the logarithmic temperature drop Δt_{LN} and overal coefficien of heat transfer k will be defined.

For the chosen amount of longitudinal line the real flue gas heat transfer surface S_{FG}^{REAL} and the real heat transferred in the heat transfer surface Q^{REAL} is determined. The value of the real heat transferred and the value of heat input should be as close as possible. And therefore determines the condition that the heat difference will not be more than 5%. For more accurate values is possible to change the dimensions of the tubes, the number of tubes or the lateral spacing.

The calculation is based on information from the literature [1].

Table 4. 1The values for the calculation of heat transfer surfaces

Quantity	Indications	Amount	Unit
Height of flue gas duct	h	8.5	$[m]$
Width of flue gas guct	w	2.7	$[m]$
Flow gas volume flow	M_{FG}	39.15	$[m^3/s]$
Relative radiation lost	L_R	0.4146	[%]
Coefficient of fin expansion	μ	1	$[-]$
Coefficient of fin thermal conductivity	λ_f	40	$[W/m \cdot K]$
Correction annulus coefficient	c_m	1	$[-]$
Correction length coefficient	c_l	1	$[-]$
Correcting coefficient dependent on current and wall temperature	c_t	1	$[-]$
The coefficient of unevenness distribution $\boldsymbol{\alpha_C}$ across the surface of the fin	ψ_f	0.85	$[-]$

Used formulas

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C} \; [W/m^2/K] \qquad (4–1)$$

μ	$[-]$	– coefficient of fin expansion
E	$[-]$	– coefficient of fins efficiency
$\frac{S_f}{S}$	$[-]$	– ratio of fin heat transfer surfaces and the total area of the flue gas side
$\frac{S_h}{S}$	$[-]$	– ratio of not finned tube surface and the total area
ψ_f	$[-]$	– coefficient of unevenness distribution α_k across the surface of the fin
α_C	$[W/m^2/K]$	– convection heat transfer coefficient
ε	$[-]$	– coefficient of fins fouling $\varepsilon = 0.0045$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} \; [-] \qquad (4–2)$$

D_f	$[m]$	– outer fin diameter
p_f	$[m]$	– fin pitch

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} \; [-] \qquad (4–3)$$

Coefficient β:

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot (1 + \varepsilon \cdot \psi_f \cdot \alpha_C)}} \; [-] \qquad (4–4)$$

λ_f	$[W/m/K]$	– coefficient of fin thermal conductivity
α_C	$[W/m^2/K]$	– coefficient of heat transfer convection

Coefficient of heat transfer convection α_C:

$$\alpha_C = 0{,}23 \cdot c_Z \cdot {\varphi_\sigma}^{0,2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0,54} \cdot \left(\frac{h_f}{p_f}\right)^{-0,14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0,65} \; [W/m^2/K] \qquad (4–5)$$

c_Z	$[-]$	–coefficient of the number of longitudinal line
φ_σ	$[-]$	– coefficient of relative pitch
λ_{FG}	$[W/m/K]$	– cofficinet of flue gas thermal conductivity

ν_{FG} $[m^2/s]$ – coefficionet of flue gas kinematic viscosity

v_{FG} $[m/s]$ – flue gas velocity

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{\sigma_1 - 1}{\sigma_2' - 1} = \frac{\frac{p_1}{D} - 1}{\frac{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2}}{D} - 1} = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2} - D}\ [-] \tag{4–6}$$

The relative lateral spacing σ_1:

$$\sigma_1 = \frac{p_1}{D}\ [-] \tag{4–7}$$

The relative diagonal pitch σ_2':

$$\sigma_2' = \frac{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2}}{D} [-] \tag{4–8}$$

Heat transfer coefficient on the steam side α_{2r}:

$$\alpha_{2r} = 0{,}023 \cdot \frac{\lambda_S}{d_e} \cdot \left(\frac{v_S \cdot d_e}{\nu_S}\right)^{0{,}8} \cdot Pr^{0{,}4} \cdot c_t \cdot c_l \cdot c_m\ [W/m^2/K] \tag{4–9}$$

λ_S $[W/m/K]$ – coefficient of steam thermal conductivity

d_e $[m]$ – equivalent diameter, $d_e = d$

ν_S $[m^2/s]$ – coefficionet of steam kinematic viscosity

Pr $[-]$ – Prandtl number

c_t $[-]$ – correcting coefficient dependent on current and wall temperature

c_l $[-]$ – correction length coefficient

c_m $[-]$ – correction annulus coefficient

v_S $[m/s]$ – steam velocity

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot \left({D_f}^2 - D^2\right)}{4} + \pi \cdot D_f \cdot t_f\ [m^2] \tag{4–10}$$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot \left(1 - n_f \cdot t_f\right) + n_f \cdot S_{1f}\ [m^2] \tag{4–11}$$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d\ [m^2] \tag{4–12}$$

The overall coefficient of heat transfer k:

$$k = \frac{1}{\frac{1}{\alpha_{1r}} + \frac{1}{\alpha_{2r}} \cdot \frac{S_{1m}}{S_{2m}}}\ [W/m^2/K] \tag{4–13}$$

Logarithmic temperature drop Δt_{LN}:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)}\ [K] \tag{4–14}$$

Δt_1 $[°C]$ – temperature difference between flue gas inlet and outlet of steam/ water

Δt_2 $[°C]$ – temperature difference between flue gas outlet and intlet of steam/ water

External heat transfer surface S_{EX}:

$Q_{HTS} = k \cdot S_{EX} \cdot \Delta t_{LN}$ =>

$$S_{EX} = \frac{Q_{TP}}{k \cdot \Delta t_{LN}}\ [m^2] \tag{4–15}$$

Q_{HTS} $[W]$ – heat transferred in heat transfer surface

The number of longitudinal lines n_{LI}:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{l \cdot S_{1m} \cdot n_{TU}}\ [-] \tag{4–16}$$

S_{LI} $[m^2]$ – heat transfer surface in one longitudinal lines

The real external heat transfer surface S_{EX}^{REAL}:

$$S_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m}\ [m^2] \tag{4–17}$$

The real heat transferred in heat transfer surface:

$$Q^{REAL} = k \cdot S_{EX} \cdot \Delta t_{LN}\ [\mathrm{kW}] \tag{4–18}$$

4.1 Proposal of second high pressure superheater HPsh2

Proposal of dimension and arrangement of finned tubes in flue-gas duct in chapter 3 was executed. The dimensions of fin tube are shown in *Table 3.1* and calculated values and main dimensions of flue-gas duct in *Table 3.2*.

4.1.1 Heat transfer coefficient of high pressure superheater HPsh2

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + p_2{}^2} - D} = \frac{71.8 - 31.8}{\sqrt{\left(\frac{71.8}{2}\right)^2 + 117^2} - 31.8} = 0.44158\ [-]$$

The cofficinet of flue gas thermal conductivity λ_{FG} and coefficient of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{A-B} = 507.3\ °C$ and for volume fraction of H_2O $x_{H_2O} = 7.8\ \%$ were defined:

$\lambda_{FG} = 0.06442\ W/m/K$

$\nu_{FG} = 7.68 \cdot 10^{-5}\ m^2/s$

Coefficient of heat transfer convection α_C:

Coefficient of the number of longitudinal line $c_Z = 0.95$ after consultation with supervissor was determined.

$$\alpha_C = 0.23 \cdot c_Z \cdot {\varphi_\sigma}^{0,2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0,54} \cdot \left(\frac{h_f}{p_f}\right)^{-0,14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0,65}$$

$$\alpha_C = 0.23 \cdot 0.95 \cdot 0.44158^{0,2} \cdot \frac{0.06442}{0.00465} \cdot \left(\frac{0.038}{0.00465}\right)^{-0,54} \cdot \left(\frac{0.015}{0.00465}\right)^{-0,14} \cdot \left(\frac{9.95 \cdot 0.00465}{7.68 \cdot 10^{-5}}\right)^{0,65}$$

$$\alpha_C = 49.524\, W/m^2/K$$

Coefficient β :

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot (1 + \varepsilon \cdot \psi_f \cdot \alpha_C)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 49.524}{0.001 \cdot 40 \cdot (1 + 0.0045 \cdot 0.85 \cdot 49.524)}} = 42.066$$

Product of $\beta \cdot h_r$:

$$\beta \cdot h_f = 42.066 \cdot 0.015 = 0.631$$

Ratio $\frac{D_f}{D}$:

$$\frac{D_f}{D} = \frac{61.8}{31.8} = 1.94$$

Now is possible to determine E from nomogram in literature [1]

$$E = 0.87.$$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} = \frac{\left(\frac{61.8}{31.8}\right)^2 - 1}{\left(\frac{61.8}{31.8}\right)^2 - 1 + 2 \cdot \left(\frac{4.65}{31.8} - \frac{1}{31.8}\right)} = 0.92362$$

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} = 1 - 0.92362 = 0.07638$$

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C}$$

$$\alpha_{1r} = [0.92362 \cdot 0,87 \cdot 1 + 0.07638] \cdot \frac{0.85 \cdot 49.524}{1 + 0.0045 \cdot 0.85 \cdot 49.524} = 31.142\, W/m^2/K$$

Heat transfer coefficient on the steam side α_{2r}:

Using the software XSTEAM is for mean temperature and pressure determined coefficient of steam thermal conductivity λ_s, Prandtl number Pr and coefficient of dynamic steam viscosity μ_s.

$$t_{1-2} = 439.765\ °C$$

$$p_{1-2} = 6.05\ MPa$$

$$\lambda_s = f(t_{1-2}, p_{1-2}) = 0.06479\ W/m/K$$

$$\mu_s = f(t_{1-2}, p_{1-2}) = 2.61 \cdot 10^{-5}\ Pa \cdot s$$

$$Pr = f(t_{1-2}, p_{1-2}) = 0.99162$$

Coefficient of steam viscosity ν_s :

$$\nu_s = \mu_s \cdot v_{1-2} = 2.61 \cdot 10^{-5} \cdot 0.050766 = 1.33 \cdot 10^{-6}\ m^2/s$$

Heat transfer coefficient on the steam side α_{2r}:

$$\alpha_{2r} = 0.023 \cdot \frac{\lambda_S}{d_e} \cdot \left(\frac{v_S \cdot d_e}{\nu_S}\right)^{0,8} \cdot Pr^{0,4} \cdot c_t \cdot c_l \cdot c_m$$

$$\alpha_{2r} = 0.023 \cdot \frac{0.06479}{0.0228} \cdot \left(\frac{9.95 \cdot 0.0228}{1.33 \cdot 10^{-6}}\right)^{0,8} \cdot 0.99162^{0,4} \cdot 1 \cdot 1 \cdot 1 = 1749.03\ W/m^2/K$$

4.1.2 Overall coeficient of heat transfer for HPsh2

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot \left({D_f}^2 - D^2\right)}{4} + \pi \cdot D_f \cdot t_f = \frac{2 \cdot \pi \cdot (0.0618^2 - 0.0318^2)}{4} + \pi \cdot 0.0618 \cdot 0.001$$

$$S_{1f} = 0.0046\ m^2$$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot \left(1 - n_f \cdot t_f\right) + n_f \cdot S_{1f} = \pi \cdot 0.0318 \cdot (1 - 215 \cdot 0.001) + 215 \cdot 0.0046$$

$$S_{1m} = 1.006849\ m^2$$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d = \pi \cdot 0.0228 = 0.07163\ m^2$$

The overall coefficient of heat transfer k:

$$k = \frac{1}{\frac{1}{\alpha_{1r}} + \frac{1}{\alpha_{2r}} \cdot \frac{S_{1m}}{S_{2m}}} = \frac{1}{\frac{1}{31.142} + \frac{1}{1749.03} \cdot \frac{1.006849}{0.07163}} = 24.606\ W/m^2/K$$

4.1.3 The number of longitudinal lines for HPsh2

Temperature diferences for calculation of logarithmic temperature drop:

$$\Delta t_1 = T_A - t_1 = 520 - 490 = 30\ °C,$$

$\Delta t_2 = T_B - t_2 = 494.6 - 389.53 = 105.07\ °C.$

Logarithmic temperature drop Δt_{LN}:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{30 - 105.07}{ln\left(\frac{30}{105.07}\right)} = 59.892\ K$$

External heat transfer surface:

$$S_{EX} = \frac{Q_{HPsh2}}{k \cdot \Delta t_{LN}} = \frac{1447500}{24.606 \cdot 59.892} = 982.207\ m^2$$

The number of longitudinal lines n_{LI}:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{982.207}{8.5 \cdot 1.06849 \cdot 36} = 3.004$$

I choose the number of lines $\boldsymbol{n_{LI} = 3}$.

4.1.4 Scheme of arrangement of tubes in HPsh2

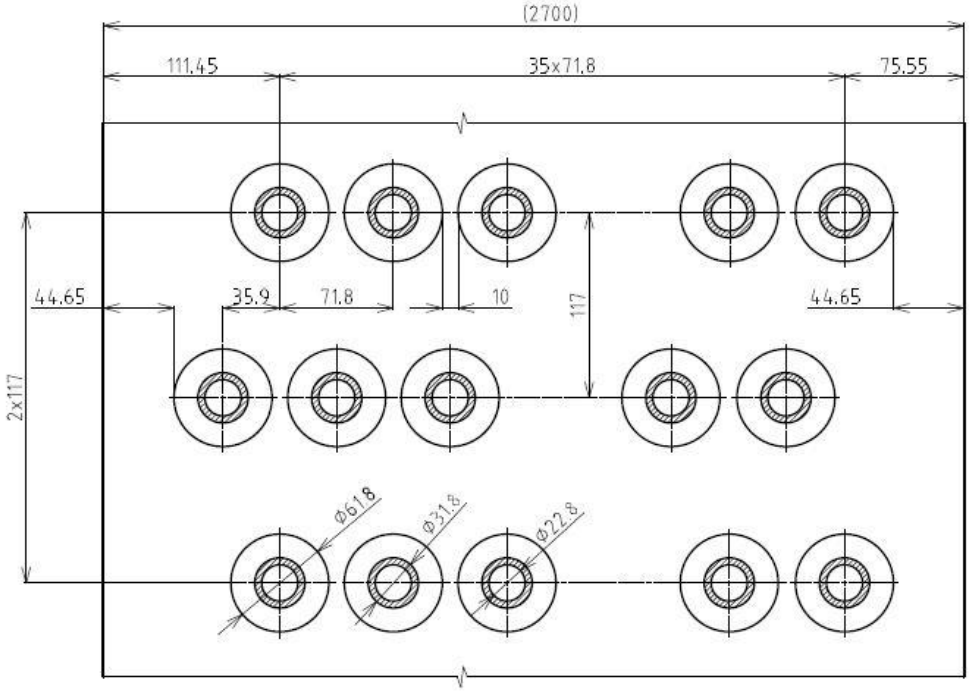

Figure 4. 1Tube arrangement of HPsh2

4.1.5 The real heat transferred in HPsh2

The real external heat transfer surface S_{EX}^{REAL}:

$$S_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 3 \cdot 36 \cdot 8.5 \cdot 1.06849 = 980.871\ m^2$$

The real heat transferred in HPsh2:

$$Q_{HPsh2}^{REAL} = k \cdot S_{EX}^{REAL} \cdot \Delta t_{LN} = 24.606 \cdot 980.871 \cdot 59.892 = 1445.53\ \text{kW}$$

Check of the real heat transferred

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left| \frac{Q_{HPsh2}^{REAL} - Q_{HPsh2}}{Q_{HPsh2}^{REAL}} \right| = \left| \frac{1445.53 - 1447.5}{1445.53} \right| = 0.136\ \%$$

$\Delta Q = 0.136\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.1.6 Real flue gas temperature in point B

The real enthalpy at point B:

$$H_B^{REAL} = H_A - \frac{Q_{HPsh2}^{REAL}}{M_{FG}(1 - L_R)} = 722{,}53 - \frac{1445.53}{39.15(1 - 0.004146)} = 685.453\ kJ/m^3$$

The real temperature at point B is determined by interpolation from *Table 2. 3*

$$T_B^{REAL} = 400 + (500 - 400) \cdot \frac{(H_B^{REAL} - H_{FG}^{400})}{(H_{FG}^{500} - H_{FG}^{400})} = 400 + 100 \cdot \frac{(685{,}274 - 547{,}44)}{(693{,}27 - 547{,}44)}$$

$$T_B^{REAL} = 494.656\ °C$$

Check of the real temperature:

The temeration difference may not be greater than 3°C:

$$\Delta T = |T_B - T_B^{REAL}| = |494.6 - 494.656| = 0.0555\ °C$$

$\Delta T = 0.0555\ °C < 3\ °C$ => Selected number of longitudinal lines is correct.

4.1.7 List of calculated values in HPsh2

Table 4. 2 Calculated values of HPsh2

Calculated values	Indication	Amount	Unit
Logarithmic temperature drop	Δt_{LN}	59.892	$[K]$
Number of longitudinal lines	n_{LI}	3	[-]
Overall coefficient of heat transfer	k	24.606	$[W/m^2/K]$
Real external heat transfer surface	S_{EX}^{REAL}	980.871	$[m^2]$
Real temperature at point B	T_B^{REAL}	494.656	$[°C]$
Real heat transferred	Q_{HPsh2}^{REAL}	1445.53	[kW]

4.2 Proposal of first high pressure superheater HPsh1

4.2.1 Fin tube design of high pressure superheater HPsh1

The fin tube dimensions are shown in *Table 4. 3* and the drawing of fin tube is shown at *Figure 4. 2.*

Table 4. 3 Fin tube dimensions of HPsh1

Tube dimensions	Indications	Amount	Unit
Outer diameter	D	31.8	[mm]
Wall thickness	t	4.5	[mm]
Inner diameter	d	22.8	[mm]
Fins height	h_f	15	[mm]
Fins thickness	t_f	1	[mm]
Fins per meter	n_f	200	[1/m]
Fins pitch	p_f	5	[mm]
Outer fin diameter	D_f	61.8	[mm]

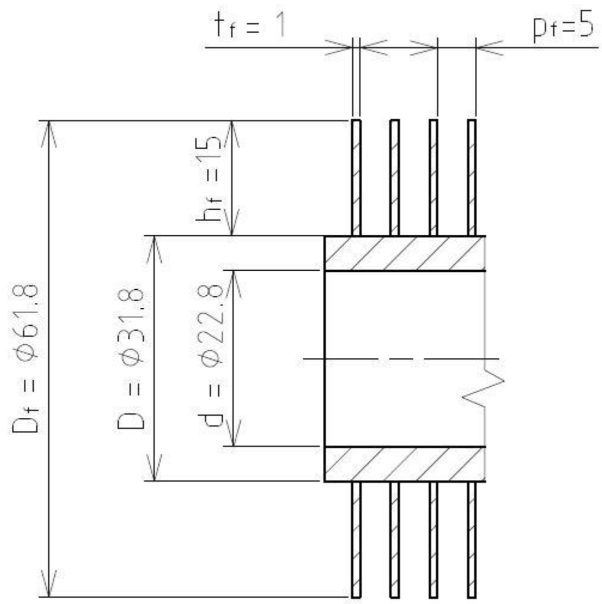

Figure 4. 2 Fin tube dimensions of HPsh1

4.2.2 Amount of fin tubes in one line HPsh1

<u>Lateral pitch:</u>

Lateral pitch consist of outer rib diameter and the distance betwen the tube $a = 15\ mm$. $p_1 = D_f + a = 61.8 + 12 = 73.8\ mm.$

<u>Longitudinal pitch after consultation with supervissor was choosen:</u>

$p_2 = 117\ mm.$

Amount of tubes in one longitudinal line:

$$n_{TU} = \frac{w}{p_1} - \frac{1}{2} = \frac{2.7}{0.0738} - \frac{1}{2} = 36.08$$

The real tubes amount in a single longitudinal line: $n_{TU} = 36.08 \; => \; \mathbf{36}$

Flow velocity of steam:

In the next step it is important to determine the mean specific volume of steam which is given by average temperature and pressure in HPsh1. The mean specific volume of steam is then calculated by the softwar XSTEAM.

$$t_{3-4} = \frac{t_3 + t_4}{2} = \frac{449.84 + 277.73}{2} = 363.758\,°C$$

$$p_{3-4} = \frac{p_3 + p_4}{2} = \frac{6.1 + 6.2}{2} = 6.15MPa$$

$$v_{3-4} = f(t_{3-4}, p_{3-4}) = 0.042527\; m^3/kg$$

The flow velocity of steam – formulas (3-1;3-2) :

$$v_s = \frac{M_s \cdot v_s}{S_s} = \frac{4 \cdot 0.95 \cdot M_{HP} \cdot v_{3-4}}{\pi \cdot d^2 \cdot n_{TU}} = \frac{4 \cdot 0.95 \cdot 5.79 \cdot 0.042527}{\pi \cdot 0.0228^2 \cdot 36} = 15.915\; m/s.$$

Real flue gas volume flow:

For the calculation of the real flue gas volume flow the calculated enthalpy and temperature at point C in Chapter 2.3.2.6 will be used.

The average temperature of the flue gas stream:

$$T_{B-C} = \frac{T_B^{REAL} + T_C}{2} = \frac{494.656 + 444.47}{2} = 469.563\,°C.$$

A real volumetric flow takes into account the flue gas temperature:

$$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{B-C} + 273.15}{273.15} = 39.15 \cdot \frac{469.563 + 273.15}{273.15} = 106.451\; m^3/s.$$

A real cross section of the the flue gas duct :

$$S_{DUCT}^{REAL} = h \cdot \left(w - n_{TU} \cdot \left(D + 2 \cdot h_f \cdot t_f \cdot n_f\right)\right)$$

$$S_{DUCT}^{REAL} = 8.5 \cdot \left(2.7 - 36 \cdot (0.0318 + 2 \cdot 0.015 \cdot 0.001 \cdot 200)\right) = 11.383 m^2$$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{106.451}{11.383} = 9.352\; m/s.$$

4.2.3 Heat transfer coefficient of high pressure superheater HPsh1

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2} - D} = \frac{73.8 - 31.8}{\sqrt{\left(\frac{73.8}{2}\right)^2 + 117^2} - 31.8} = 0.46214[-]$$

The cofficinet of flue gas thermal conductivity λ_{FG} and coefficionet of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{B-C} = 469.563\ °C$ and for volume of H_2O $x_{H_2O} = 7.8$ % were defined:

$\lambda_{FG} = 0.06131 W/m/K$

$\nu_{FG} = 7.08 \cdot 10^{-5}\ m^2/s$

Coefficient of heat transfer convection α_C:

Coefficient of the number of longitudinal line $c_Z = 0.95$ after consultation with supervissor was determined.

$$\alpha_C = 0.23 \cdot c_Z \cdot \varphi_\sigma{}^{0,2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0,54} \cdot \left(\frac{h_f}{p_f}\right)^{-0,14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0,65}$$

$$\alpha_C = 0.23 \cdot 0.95 \cdot 0.46214^{0,2} \frac{0.06131}{0.005} \left(\frac{0.0318}{0.005}\right)^{-0,54} \left(\frac{0.015}{0.005}\right)^{-0,14} \left(\frac{9.352 \cdot 0.005}{7.08 \cdot 10^{-5}}\right)^{0,65}$$

$\alpha_C = 49.355\ W/m^2/K$

Coefficient β :

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot (1 + \varepsilon \cdot \psi_f \cdot \alpha_C)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 49.355}{0.001 \cdot 40 \cdot (1 + 0.0045 \cdot 0.85 \cdot 49.355)}} = 42.006$$

Product of $\beta \cdot h_r$:

$\beta \cdot h_f = 42.006 \cdot 0.015 = 0.630$

Ratio $\frac{D_f}{D}$:

$$\frac{D_f}{D} = \frac{61.8}{31.8} = 1.94$$

Now is possible to determine E from nomogram in literature [1]

$E = 0.87.$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} = \frac{\left(\frac{61.8}{31.8}\right)^2 - 1}{\left(\frac{61.8}{31.8}\right)^2 - 1 + 2 \cdot \left(\frac{5}{31.8} - \frac{1}{31.8}\right)} = 0.91693$$

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} = 1 - 0.9135 = 0.08307$$

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C}$$

$$\alpha_{1r} = [0.91693 \cdot 0.87 \cdot 1 + 0.08307] \cdot \frac{0.85 \cdot 49.355}{1 + 0.0045 \cdot 0.85 \cdot 49.355} = 31.083\ W/m^2/K$$

Heat transfer coefficient on the steam side α_{2r}:

Using the software XSTEAM is for mean temperature and pressure determined coefficient of steam thermal conductivity λ_s, Prandtl number Pr and coefficient of dynamic steam viscosity μ_s .

$$t_{3-4} = 363.785\ °C$$

$$p_{3-4} = 6.15\ MPa$$

$$\lambda_s = f(t_{3-4}, p_{3-4}) = 0.05818\ W/m/K$$

$$\mu_s = f(t_{3-4}, p_{3-4}) = 2.27 \cdot 10^{-5}\ Pa \cdot s$$

$$Pr = f(t_{3-4}, p_{3-4}) = 1.08267$$

Coefficient of steam viscosity ν_s :

$$\nu_s = \mu_s \cdot v_{3-4} = 2.27 \cdot 10^{-5} \cdot 0.042527 = 9.669 \cdot 10^{-7}\ m^2/s$$

Heat transfer coefficient on the steam side α_{2r}:

$$\alpha_{2r} = 0.023 \cdot \frac{\lambda_S}{d_e} \cdot \left(\frac{v_S \cdot d_e}{\nu_S}\right)^{0,8} \cdot Pr^{0,4} \cdot c_t \cdot c_l \cdot c_m$$

$$\alpha_{2r} = 0.023 \cdot \frac{0.05818}{0.0228} \cdot \left(\frac{15.915 \cdot 0.0228}{9.669 \cdot 10^{-7}}\right)^{0,8} \cdot 1.08267^{0,4} \cdot 1 \cdot 1 \cdot 1 = 1745.25\ W/m^2/K$$

4.2.4 Overall coeffcient of heat transfer for HPsh1

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot \left(D_f{}^2 - D^2\right)}{4} + \pi \cdot D_f \cdot t_f = \frac{2 \cdot \pi \cdot (0.0618^2 - 0.0318^2)}{4} + \pi \cdot 0.0618 \cdot 0.001$$

$$S_{1f} = 0.0046\ m^2$$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot \left(1 - n_f \cdot t_f\right) + n_f \cdot S_{1f} = \pi \cdot 0.0318 \cdot (1 - 200 \cdot 0.001) + 200 \cdot 0.0046$$

$$S_{1m} = 1.0009\ m^2$$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d = \pi \cdot 0.0228 = 0.07163\ m^2$$

The overall coefficient of heat transfer k:

$$k = \frac{1}{\frac{1}{\alpha_{1r}} + \frac{1}{\alpha_{2r}} \cdot \frac{S_{1m}}{S_{2m}}} = \frac{1}{\frac{1}{31.083} + \frac{1}{1745.25} \cdot \frac{1.0009}{0.07163}} = 24.889\ W/m^2/K$$

4.2.5 The number of longitudinal lines for HPsh1

Temperature diferences for calculation of logarithmic temperature drop:

$\Delta t_1 = T_B^{REAL} - t_3 = 494.656 - 449.84 = 44.82°C$

$\Delta t_2 = T_C - t_4 = 444.47 - 277.73 = 166.74\ °C$

Logarithmic temperature drop Δt_{LN}:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{44.82 - 166.74}{ln\left(\frac{44.82}{166.74}\right)} = 92.797\ °C$$

External heat transfer surface:

$$S_{EX} = \frac{Q_{HPsh1}}{k \cdot \Delta t_{LN}} = \frac{2852500}{24.889 \cdot 92.797} = 1235.0468\ m^2$$

The number of longitudinal lines n_{LI}:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{1235.0468}{8.5 \cdot 1.0009 \cdot 36} = 4.03$$

I choose the number of lines $\boldsymbol{n_{LI} = 4}$**.**

4.2.6 Scheme of arrangement of tubes in HPsh1

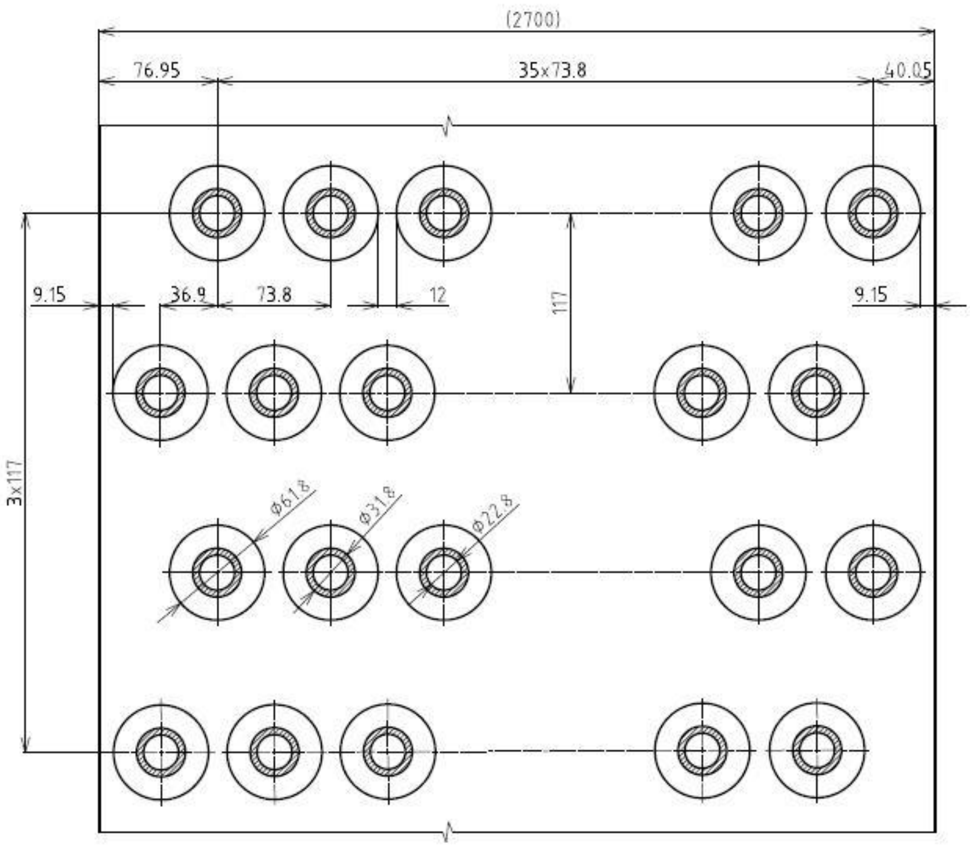

Figure 4. 3 Tube arrangement in Hsh1

4.2.7 The real heat transferred in HPsh1

The real external heat transfer surface S_{EX}^{REAL}:

$$S_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 4 \cdot 36 \cdot 8.5 \cdot 1.0009 = 1225.1156\ m^2$$

The real heat transferred in HPsh1:

$$Q_{HPsh1}^{REAL} = k \cdot S_{EX}^{REAL} \cdot \Delta t_{LN} = 24.889 \cdot 1225.1156 \cdot 92.797 = 2829.56\ \text{kW}$$

Check of the real heat transferred

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left| \frac{Q_{HPsh1}^{REAL} - Q_{HPsh1}}{Q_{HPsh1}^{REAL}} \right| = \left| \frac{2829.56 - 2852.5}{2829.56} \right| = 0.811\%$$

$\Delta Q = 0.811\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.2.8 Real flue gas temperature in point C

The real enthalpy at point C:

$$H_C^{REAL} = H_B^{REAL} - \frac{Q_{HPsh1}^{REAL}}{M_{FG}(1 - L_R)} = 685.453 - \frac{2829.56}{39.15(1 - 0.004146)} = 612.878\ kJ/m^3$$

The real temperature at point B is determined by interpolation from *Table 2. 3*

$$T_C^{REAL} = 400 + (500 - 400) \cdot \frac{(H_C^{REAL} - H_{FG}^{400})}{(H_{FG}^{500} - H_{FG}^{400})} = 400 + 100 \cdot \frac{(612.878 - 547.44)}{(693.27 - 547.44)}$$

$$T_C^{REAL} = 444.871\ °C$$

Check of the real temperature:

The temerature difference may not be greater than 3°C:

$$\Delta T = |T_C - T_C^{REAL}| = |444.47 - 444.871| = 0.401\ °C$$

$\Delta T = 0.401\ °C < 3\ °C$ => Selected number of longitudinal lines is correct.

4.2.9 List of calculated values in HPsh1

Table 4. 4 Calculated values of HPsh1

Calculated values	Indication	Amount	Unit
Number of tubes in one longitudial line	n_{TU}	36	$[-]$
Real volumetric flow	M_{FG}^{REAL}	106.451	$[m^3/s]$
Real flue gas velocity	v_{FG}^{REAL}	9.352	$[m/s]$
Velocity of steam	v_S	15.915	$[m/s]$
Logarithmic temperature drop	Δt_{LN}	92.797	$[K]$
Number of longitudinal lines	n_{LI}	4	$[-]$
Overall coefficient of heat transfer	k	24.889	$[W/m^2$
Real external heat transfer surface	S_{EX}^{REAL}	1225.1156	$[m^2]$

Real heat transferred	Q_{HPsh1}^{REAL}	2829.56	[kW]
Real temperature at point C	T_C^{REAL}	444.871	$[°C]$

4.3 Proposal of high pressure evaporater HPeva

4.3.1 Fin tube design of high pressure evaporater HPeva

The fin tube dimensions are shown in *Table 4. 5* and the drawind of fin tube is shown at *Figure 4. 4*

Table 4. 5 Fin tube dimensions of HPeva

Tube dimensions	Indications	Amount	Unit
Outer diameter	D	57	[mm]
Wall thickness	t	4.5	[mm]
Inner diameter	d	48	[mm]
Fins height	h_f	19	[mm]
Fins thickness	t_f	1	[mm]
Fins per meter	n_f	230	[1/m]
Fins pitch	p_f	4.35	[mm]
Outer fin diameter	D_f	95	[mm]

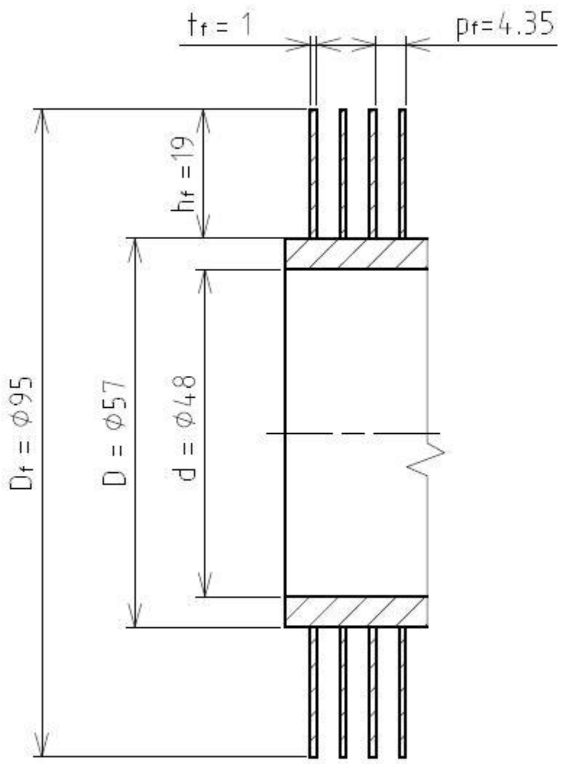

Figure 4. 4 Fin tube dimensions of HPeva

4.3.2 Amount of fin tubes in one line HPeva

Lateral pitch:

Latera pitch consist of outer rib diameter and the distance betwen the tube $a = 7\ mm$. $p_1 = D_f + a = 95 + 6 = 101\ mm$.

Longitudinal pitch after consultation with supervissor was choosen:

$p_2 = 117\ mm$.

Amount of tubes in one longitudinal line:

$$n_{TU} = \frac{w}{p_1} - \frac{1}{2} = \frac{2.7}{0.101} - \frac{1}{2} = 26.23$$

The real tubes amount in a single longitudinal line: $n_{TU} = 26.23\ => \mathbf{26}$

Real flue gas volume flow :

For the calculation of the real flue gas volume flow will use the calculated enthalpy and temperature at point D in Chapter 2.3.2.7

The average temperature of the flue gas stream:

$$T_{C-D} = \frac{T_C^{REAL} + T_D}{2} = \frac{444.871 + 288.06}{2} = 366.466\ °C.$$

A real volumetric flow takes into account the flue gas temperature:

$$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{C-D} + 273.15}{273.15} = 39.15 \cdot \frac{366.466 + 273.15}{273.15} = 91.675\ m^3/s.$$

A real cross section of the the flue gas duct :

$$S_{DUCT}^{REAL} = h \cdot \left(w - n_{TU} \cdot (D + 2 \cdot h_r \cdot t_r \cdot n_r)\right)$$

$$S_{DUCT}^{REAL} = 8.5 \cdot \left(2.7 - 26 \cdot (0.057 + 2 \cdot 0.019 \cdot 0.001 \cdot 230)\right) = 8.422 m^2$$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{91.675}{8.422} = 10.886\ m/s.$$

4.3.3 Heat transfer coefficient of high pressure evaporater HPeva

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2} - D} = \frac{101 - 57}{\sqrt{\left(\frac{101}{2}\right)^2 + 117^2} - 57} = 0.624704$$

The cofficinet of flue gas thermal conductivity λ_{FG} and coefficionet of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{C-D} = 366.466\ °C$ and for volume fraction of H_2O $x_{H_2O} = 7.8$ % were defined:

$$\lambda_{FG} = 0.052826\ W/m/K$$

$$\nu_{FG} = 5.50 \cdot 10^{-5}\ m^2/s$$

Coefficient of heat transfer convection α_C:

Coefficient of the number of longitudinal line $c_Z = 1$ after consultation with supervissor was determined.

$$\alpha_C = 0.23 \cdot c_Z \cdot {\varphi_\sigma}^{0.2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0.54} \cdot \left(\frac{h_f}{p_f}\right)^{-0.14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0.65}$$

$$\alpha_C = 0.23 \cdot 0.95 \cdot 0.624704^{0.2} \frac{0.052826}{0.00435} \left(\frac{0.057}{0.00435}\right)^{-0.54} \left(\frac{0.019}{0.00435}\right)^{-0.14} \left(\frac{10.886 \cdot 0.00435}{5.50 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_C = 41.657\ W/m^2/K$$

Coefficient β :

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot (1 + \varepsilon \cdot \psi_f \cdot \alpha_C)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 41.657}{0.001 \cdot 40 \cdot (1 + 0.0045 \cdot 0.85 \cdot 41.657)}} = 39.078$$

Product of $\beta \cdot h_r$:

$$\beta \cdot h_f = 39.078 \cdot 0.019 = 0.742$$

Ratio $\frac{D_f}{D}$:

$$\frac{D_f}{D} = \frac{95}{57} = 1.67$$

Now is possible to determine E from nomogram in literature [1]

$$E = 0.82$$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} = \frac{\left(\frac{95}{57}\right)^2 - 1}{\left(\frac{95}{57}\right)^2 - 1 + 2 \cdot \left(\frac{4.35}{57} - \frac{1}{57}\right)} = 0.93802$$

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} = 1 - 0.93802 = 0.06198$$

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C}$$

$$\alpha_{1r} = [0.93802 \cdot 0.82 \cdot 1 + 0.06198] \cdot \frac{0.85 \cdot 41.657}{1 + 0.0045 \cdot 0.85 \cdot 41.657} = 25.385\ W/m^2/K$$

4.3.4 Overall coefficient of heat transfer for HPeva

Heat transfer coefficient on the steam side is in evaporator neglected due to heat transfer coefficient on the steam side is many times smaller than heat transfer coefficient on the flue gas side ($\alpha_{2r} \ll \alpha_{1r}$). Only the heat transfer on the flue gas side is taken into account.

Overall coefficient of heat transfer will calculated as $k = \alpha_{1r} = 25.385\ W/m^2/K$

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot \left({D_f}^2 - D^2\right)}{4} + \pi \cdot D_f \cdot t_f = \frac{2 \cdot \pi \cdot (0.095^2 - 0.057^2)}{4} + \pi \cdot 0.095 \cdot 0.001$$

$$S_{1f} = 0.009371\ m^2$$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot \left(1 - n_f \cdot t_f\right) + n_f \cdot S_{1f} = \pi \cdot 0.057 \cdot (1 - 230 \cdot 0.001) + 230 \cdot 0.009371$$

$$S_{1m} = 2.293215\ m^2$$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d = \pi \cdot 0.048 = 0.105796\ m^2$$

4.3.5 The number of longitudinal lines for HPeva

Temperature diferences for calculation of logarithmic temperature drop:

$$\Delta t_1 = T_C^{REAL} - t_4 = 444.871 - 277.73 = 167.14\ °C$$

$$\Delta t_2 = T_{\mathrm{D}} - t_5 = 288.06 - 277.73 = 10.33\ °C$$

Logarithmic temperature drop Δt_{LN}:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{167.14 - 10.33}{ln\left(\frac{167.14}{10.33}\right)} = 56.33\ K$$

External heat transfer surface:

$$S_{EX} = \frac{Q_{HPeva}}{k \cdot \Delta t_{LN}} = \frac{8709930}{25.385 \cdot 56.33} = 6091.13\ m^2$$

The number of longitudinal lines n_{LI}:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{6091.13}{8.5 \cdot 2.293215 \cdot 26} = 12.01$$

I choose the number of lines $\boldsymbol{n_{LI} = 12}$.

4.3.6 Scheme of arrangement of tubes in HPeva

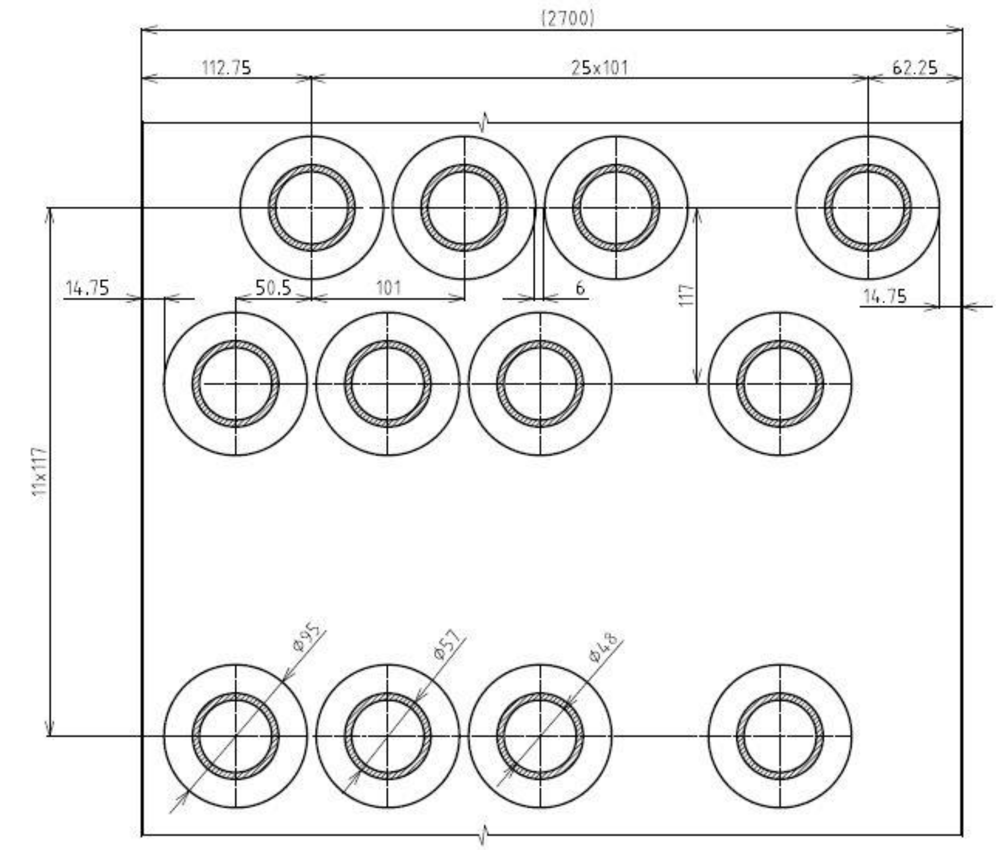

Figure 4. 5 Tube arrengement in HPeva

4.3.7 The real heat transferred in HPeva

The real external heat transfer surface S_{EX}^{REAL}:

$S_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 12 \cdot 26 \cdot 8.5 \cdot 2.293215 = 6081.606\, m^2$

The real heat transferred in HPeva:

$Q_{HPeva}^{REAL} = k \cdot S_{EX}^{REAL} \cdot \Delta t_{LN} = 25.385 \cdot 6081.606 \cdot 56.33 = 8696.31\text{kW}$

Check of the real heat transferred

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left|\frac{Q_{HPeva}^{REAL} - Q_{HPeva}}{Q_{HPeva}^{REAL}}\right| = \left|\frac{8696.31 - 8709.93}{8696.31}\right| = 0.157\ \%$$

$\Delta Q = 0.157\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.3.8 Real flue gas temperature in point D

The real enthalpy at point D:

$$H_D^{REAL} = H_C - \frac{Q_{HPeva}^{REAL}}{M_{FG}(1 - L_R)} = 612.878 - \frac{8696.31}{39.15(1 - 0.004146)} = 389.825\ kJ/m^3$$

The real temperature at point D is determined by interpolation from *Table 2. 3*

$$T_D^{REAL} = 200 + (300 - 200)\frac{(H_D^{REAL} - H_{FG}^{200})}{(H_{FG}^{300} - H_{FG}^{200})} = 200 + 100 \cdot \frac{(389.825 - 268.01)}{(405.95 - 268.01)}$$

$$T_D^{REAL} = 288.314\,°C$$

Check of the real temperature:

The temeratute difference may not be greater than 3°C:

$$\Delta T = |T_D - T_D^{REAL}| = |288.06 - 288.314| = 0.254\,°C$$

$\Delta T = 0.254\,°C < 3\,°C$ => Selected number of longitudinal lines is correct

4.3.9 List of calculated values in HPeva

Table 4. 6 Calculated values of HPeva

Calculated values	Indication	Amount	Unit
Number of tubes in one longitudial line	n_{TU}	26	$[-]$
Real volumetric flow	M_{FG}^{REAL}	91.675	$[m^3/s]$
Real flue gas velocity	v_{FG}^{REAL}	10.886	$[m/s]$
Logarithmic temperature drop	Δt_{LN}	56.33	$[K]$
Number of longitudinal lines	n_{LI}	12	$[-]$
Overall coefficient of heat transfer	k	25.385	$[W/m^2$
Real external heat transfer surface	S_{EX}^{REAL}	6081.606	$[m^2]$
Real heat transferred	Q_{HPeva}^{REAL}	8696.31	[kW]
Real temperature at point D	T_D^{REAL}	288.314	$[°C]$

4.4 Proposal of high pressure econimizer HPeco3

The calculation of high pressure economizer is executed as high pressure evaporater calculation. The value of heat transfer coefficient is expected high and can be neglected.

4.4.1 Fin tube design of high pressure economizer HPeco3

The fin tube dimensions are given in *Table 4. 7* and the drawind of fin tube is shown at *Figure 4.7.*

Table 4. 7 Fin tube dimensions of HPeco3

Tube dimensions	Indications	Amount	Unit
Outer diameter	D	31.8	[mm]
Wall thickness	t	4.5	[mm]
Inner diameter	d	22.8	[mm]
Fins height	h_f	15	[mm]
Fins thickness	t_f	1	[mm]
Fins per meter	n_f	200	[1/m]
Fins pitch	p_f	5	[mm]
Outer fin diameter	D_f	61.8	[mm]

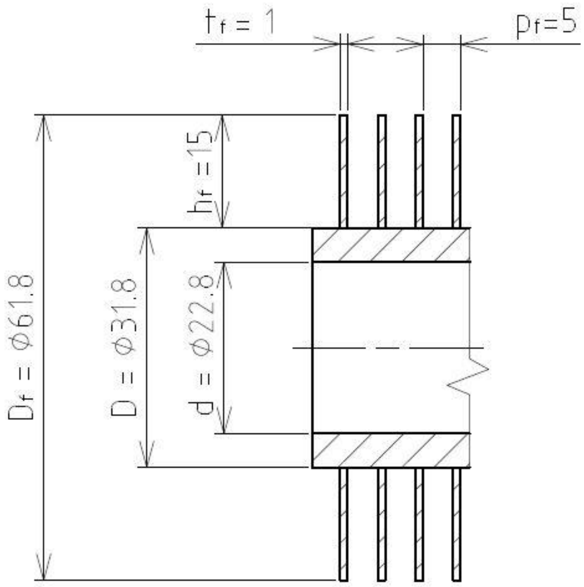

Figure 4. 6 Fin tube dimensions of HPeco3

4.4.2 Amount of fin tubes in one line HPeco3

<u>Lateral pitch:</u>

Latera pitch consist of outer fin diameter and the distance between the tube $a = 12\,mm$. $p_1 = D_f + a = 61.8 + 12 = 73.8\,mm.$

<u>Longitudinal pitch after consultation with supervisor was choosen:</u>

$p_2 = 92\,mm.$

Amount of tubes in one longitudinal line:

$$n_{TU} = \frac{w}{p_1} - \frac{1}{2} = \frac{2.7}{0.0738} - \frac{1}{2} = 36.08$$

The real tubes amount in a single longitudinal line: $n_{TU} = 36.08 \ => \mathbf{36}$

Real flue gas volume flow:

For the calculation of the real flue gas volume flow the calculated enthalpy and temperature at point E in Chapter 2.3.2.3 will be used.

The average temperature of the flue gas stream:

$$T_{D-E} = \frac{T_D^{REAL} + T_E}{2} = \frac{288.314 + 261.74}{2} = 275.027\ °C.$$

A real volumetric flow takes into account the flue gas temperature:

$$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{D-E} + 273.15}{273.15} = 39.15 \cdot \frac{275.027 + 273.15}{273.15} = 78.569\ m^3/s.$$

A real cross section of the the flue gas duct :

$$S_{DUCT}^{REAL} = h \cdot \left(w - n_{TU} \cdot \left(D + 2 \cdot h_f \cdot t_f \cdot n_f\right)\right)$$

$$S_{DUCT}^{REAL} = 8.5 \cdot \left(2.7 - 36 \cdot (0.0318 + 2 \cdot 0.015 \cdot 0.001 \cdot 200)\right) = 11.383 m^2$$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{78.569}{11.383} = 6.902\ m/s$$

The feed water flow velocity in HPeco3:

In the next step is important to determine the mean specific volume of steam which is given by average temperature and pressure in HPeco3. The mean specific volume of steam is then calculated by the softwar XSTEAM.

$$t_{6-7} = \frac{t_6 + t_7}{2} = \frac{272.73 + 220}{2} = 246.365\ °C$$

$$p_{6-7} = \frac{p_6 + p_7}{2} = \frac{6.2 + 6.3}{2} = 6.25\ MPa$$

$$v_{6-7} = f(t_{6-7}, p_{6-7}) = 0.001239\ m^3/kg$$

$$v_s = \frac{M_s \cdot v_s}{S_s} = \frac{4 \cdot 0.95 \cdot M_{HP} \cdot v_{6-7}}{\pi \cdot d^2 \cdot n_{TU}} = \frac{4 \cdot 0.95 \cdot 5.79 \cdot 0.001239}{\pi \cdot 0228^2 \cdot 36} = 0.464\ m/s.$$

Due to slow feed water flow velocity in economizer it is necessary to split heat transfer surfect into two parts as you can see in the *Figure 4. 7.* By this division it is possible to increase feed water flow velocity. In our case two times.

$$v_s = 2 \cdot 0.464 = 0.928\ m/s$$

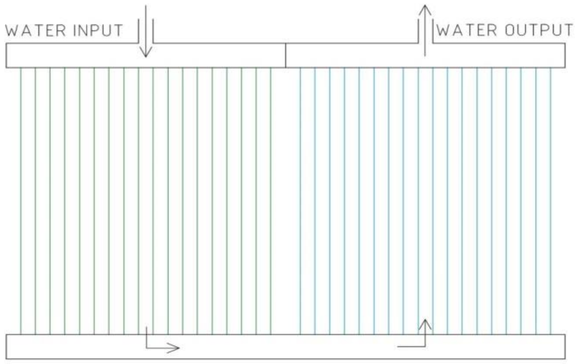

Figure 4. 7 Scheme of devided heat transfer surface in HPeco3.

4.4.3 Heat transfer coefficient of high pressure ecomonizer HPeco3

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2} - D} = \frac{73.8 - 31.8}{\sqrt{\left(\frac{73.8}{2}\right)^2 + 92^2} - 31.8} = 0.62385$$

The cofficinet of flue gas thermal conductivity λ_{FG} and coefficionet of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{D-E} = 275.027\ °C$ and for volume fraction of H_2O $x_{H_2O} = 7.8$ % were defined:

$\lambda_{FG} = 0.045345 W/m/K$

$\nu_{FG} = 4.22 \cdot 10^{-5}\ m^2/s$

Coefficient of heat transfer convection α_C:

Coefficient of the number of longitudinal line $c_Z = 0.95$ after consultation with supervisor was determined.

$$\alpha_C = 0.23 \cdot c_Z \cdot {\varphi_\sigma}^{0,2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0,54} \cdot \left(\frac{h_f}{p_f}\right)^{-0,14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0,65}$$

$$\alpha_C = 0.23 \cdot 0.95 \cdot 0.62385^{0,2} \frac{0.045345}{0.005} \left(\frac{0.0318}{0.005}\right)^{-0,54} \left(\frac{0.015}{0.005}\right)^{-0,14} \left(\frac{6.902 \cdot 0.005}{4.22 \cdot 10^{-5}}\right)^{0,65}$$

$\alpha_C = 44.537\ W/m^2/K$

Coefficient β :

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot (1 + \varepsilon \cdot \psi_f \cdot \alpha_C)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 44.537}{0.001 \cdot 40 \cdot (1 + 0.0045 \cdot 0.85 \cdot 44.537)}} = 40.216$$

Product of $\beta \cdot h_r$:

$\beta \cdot h_f = 42.216 \cdot 0.015 = 0.603$

Ratio $\frac{D_f}{D}$:

$$\frac{D_f}{D} = \frac{61.8}{31.8} = 1.94$$

Now is possible to determine E from nomogram in literature [1]

$E = 0.88.$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} = \frac{\left(\frac{61.8}{31.8}\right)^2 - 1}{\left(\frac{61.8}{31.8}\right)^2 - 1 + 2 \cdot \left(\frac{5}{31.8} - \frac{1}{31.8}\right)} = 0.91693$$

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} = 1 - 0{,}9135 = 0.08307$$

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C}$$

$$\alpha_{1r} = [0.91693 \cdot 0.88 \cdot 1 + 0.08307] \cdot \frac{0.85 \cdot 44.537}{1 + 0.0045 \cdot 0.85 \cdot 44.537} = 28.787\ W/m^2/K$$

4.4.4 Overall coefficient of heat transfer for HPeco3

Heat transfer coefficient on the steam side is in economizer neglected due to heat transfer coefficient on the steam side is many times smaller than heat transfer coefficient on the flue gas side ($\alpha_{2r} \ll \alpha_{1r}$). Only the heat transfer on the flue gas side is taken into account.

Overall coefficient of heat transfer will calculated as $k = \alpha_{1r} = 28.787\ W/m^2/K$.

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot \left({D_f}^2 - D^2\right)}{4} + \pi \cdot D_f \cdot t_f = \frac{2 \cdot \pi \cdot (0.0618^2 - 0.0318^2)}{4} + \pi \cdot 0.0618 \cdot 0.001$$

$$S_{1f} = 0.0046\ m^2$$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot \left(1 - n_f \cdot t_f\right) + n_f \cdot S_{1f} = \pi \cdot 0.0318 \cdot (1 - 200 \cdot 0.001) + 200 \cdot 0.0046$$

$$S_{1m} = 1.0009\ m^2$$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d = \pi \cdot 0.0228 = 0.07163\ m^2$$

4.4.5 The number of longitudinal lines in HPeco3

Temperature diferences for calculation of logarithmic temperature drop:

$\Delta t_1 = T_D^{REAL} - t_6 = 288.314 - 272.73 = 15.584\ °C$

$\Delta t_2 = T_{\mathrm{E}} - t_7 = 261.74 - 220 = 41.74\ °C$

Logarithmic temperature drop Δt_{LN}:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{15.584\ - 41.74}{ln\left(\frac{15.584}{41.74}\right)} = 26.549K$$

External heat transfer surface:

$$S_{EX} = \frac{Q_{HPeco3}}{k \cdot \Delta t_{LN}} = \frac{1397620}{28.787 \cdot 26.549} = 1828.709\ m^2$$

The number of longitudinal lines n_{LI}:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{1828.709}{8.5 \cdot 1.0009 \cdot 36} = 5.97$$

I choose the number of lines $\boldsymbol{n_{LI} = 6}$.

4.4.6 Scheme of arrangement of tubes in HPeco3

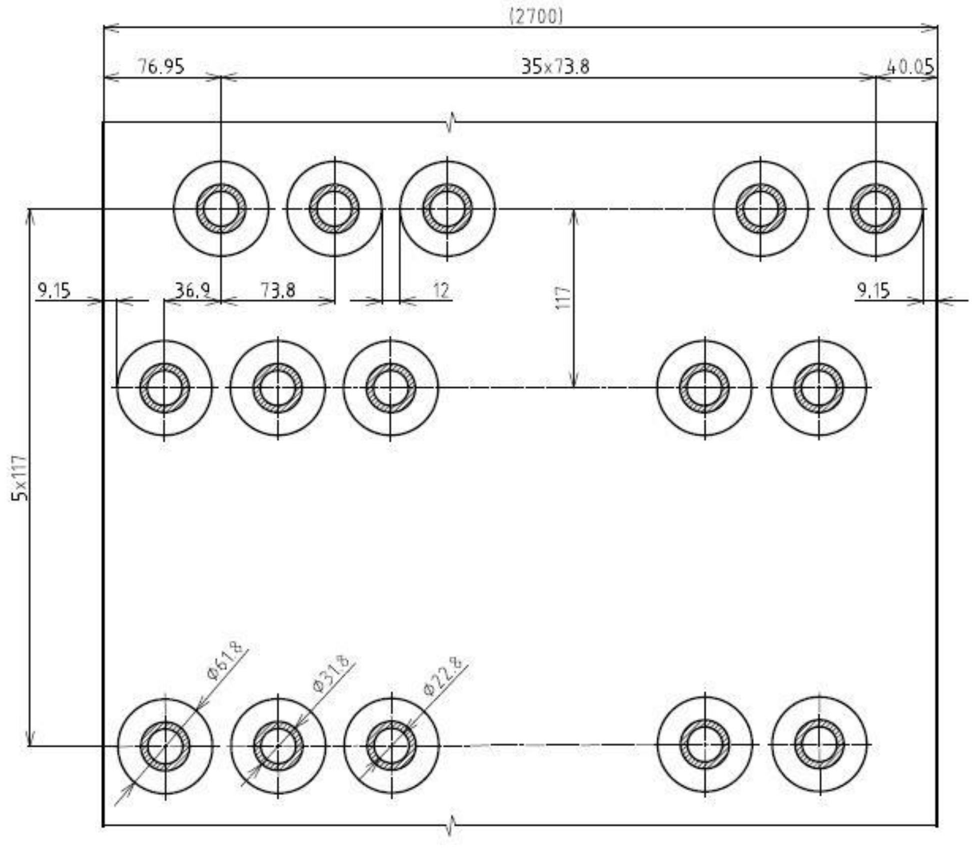

Figure 4. 8 Tube arrangement in HPeco3

4.4.7 The real heat transferred in HPeco3

The real external heat transfer surface S_{EX}^{REAL}:

$$S_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 6 \cdot 36 \cdot 8.5 \cdot 1.0009 = 1837.673\ m^2$$

The real heat transferred in HPeco3:

$$Q_{HPeco3}^{REAL} = k \cdot S_{EX}^{REAL} \cdot \Delta t_{LN} = 28.787 \cdot 1837.673 \cdot 26.549 = 1404.471\ \text{kW}$$

Check of the real heat transferred

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left| \frac{Q_{HPeco3}^{REAL} - Q_{HPeco3}}{Q_{HPeco3}^{REAL}} \right| = \left| \frac{1404.471 - 1397.62}{1404.471} \right| = 0.488\ \%$$

$\Delta Q = 0.488\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.4.8 Real flue gas temperature in point E

The real enthalpy at point E:

$$H_E^{REAL} = H_D^{REAL} - \frac{Q_{HPeco3}^{REAL}}{M_{FG}(1 - L_R)} = 387.634 - \frac{1404.471}{39.15(1 - 0.004146)} = 353.802\ kJ/m^3$$

The real temperature at point E is determined by interpolation from *Table 2. 3*

$$T_E^{REAL} = 200 + (300 - 200)\frac{(H_E^{REAL} - H_{FG}^{200})}{(H_{FG}^{300} - H_{FG}^{200})} = 200 + 100 \cdot \frac{(353.802 - 268.01)}{(405.95 - 268.01)}$$

$$T_E^{REAL} = 262.198\ °C$$

Check of the real temperature:

The temerature difference may not be greater than 3°C:

$$\Delta T = |T_E - T_E^{REAL}| = |261.74 - 262.198| = 0.458\ °C$$

$\Delta T = 0.458\ °C < 3\ °C$ => Selected number of longitudinal lines is correct.

4.4.9 List of calculated values in HPeco3

Table 4. 8 Calculated values HPeco3

Calculated values	Indication	Amount	Unit
Number of tubes in one longitudial line	n_{TU}	36	$[-]$
Real volumetric flow	M_{FG}^{REAL}	78.759	$[m^3/s]$
Real flue gas velocity	v_{FG}^{REAL}	6.902	$[m/s]$
The feed water flow velocity	v_S	0.928	$[m/s]$
Logarithmic temperature drop	Δt_{LN}	26.549	$[K]$
Number of longitudinal lines	n_{LI}	5	$[-]$
Overall coefficient of heat transfer	k	28.787	$[W/m^2$
Real external heat transfer surface	S_{EX}^{REAL}	1837.673	$[m^2]$

Real heat transferred	Q_{HPeco3}^{REAL}	1404.471	[kW]
Real temperature at point E	T_E^{REAL}	262.198	$[°C]$

4.5 Proposal of low pressure superheater LPsh

Because the amount of heat transferred is not high it is not necessary to use fin tubes. In calculations will use the smooth tubes and for the calculation procedure will use the literature [1] [8]. Because of the smooth tubes it is impossible to use the same formulas as in calculation with finned tubes.

Use formulas:

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \xi\,(\alpha_C + \alpha_R)[W/m^2/K] \qquad (4\text{–}19)$$

α_C	$[W/m^2/K]$	– convection heat transfer coefficient
α_R	$[W/m^2/K]$	– radiation heat transfer coefficient
ξ	$[-]$	– utilization factor

Coefficient of heat transfer convection α_C:

$$\alpha_C = c_S \cdot c_Z \cdot \frac{\lambda_{FG}}{D}\left(\frac{v_{FG} \cdot D}{\nu_{FG}}\right) \cdot Pr\,[W/m^2/K] \qquad (4\text{–}20)$$

α_C	$[W/m^2/K]$	– convection heat transfer coefficient
α_R	$[W/m^2/K]$	– radiation heat transfer coefficient
ξ	$[-]$	– utilization factor
λ_{FG}	$[W/m/K]$	– cofficinet of flue gas thermal conductivity
ν_{FG}	$[m^2/s]$	– coefficionet of flue gas kinematic viscosity
Pr	$[-]$	– Prandtl number

Radiation heat transfer coefficient α_R:

$$\alpha_R = 5{,}7 \cdot 10^{-8} \cdot \frac{a_c + 1}{2} \cdot a_{tr} \cdot (T_{FG})^3 \cdot \frac{\left(\frac{T_F}{T_{FG}}\right)^2}{1 - \frac{T_F}{T_{FG}}} \cdot Pr\,[W/m^2/K] \qquad (4\text{–}20)$$

T_{FG}	$[K]$	– mean flue gas temperature
T_{FO}	$[K]$	– fouled wall surface temperature
a_{tr}	$[-]$	– emisivity of tri-atomic gases
a_c	$[-]$	– cofficinet of flue gas thermal conductivity
$\nu_{FG}c$	$[m^2/s]$	– coefficionet of flue gas kinematic viscosity
Pr	$[-]$	– Prandtl number

The emissivity of tri-atomic gases a_{tr} :

$$a_{tr} = 1 - e^{-k \cdot p \cdot s}\ [-] \qquad (4–22)$$

k_y	$[m/MPa]$	– coefficient of radiant absorption due to tri-atomic gases
p	$[MPa]$	– atmospheric pressure
s	$[mm]$	– effective thickness of the radiation layer

The coefficient of radiant absorption due to tri-atomic gases k_y :

$$k_y = \left(\frac{7{,}8 + 16 \cdot x_{H2O}}{3{,}16 \cdot \sqrt{p_{par} \cdot s}}\right) \cdot \left(1 - 0{,}37 \cdot \frac{t_{sp} + 273{,}15}{1000}\right)\ [m/Mpa] \qquad (4–23)$$

p_{par}	$[MPa]$	– partial pressure of triatomic flue gases in Lsh
p	$[MPa]$	– atmospheric pressure
s	$[mm]$	– effective thickness of the radiation layer

The thickness of the effective radiation layer s :

$$s = 0{,}9 \cdot d \cdot \left(\frac{4}{\pi} \cdot \frac{p_1 \cdot p_2}{d^2} - 1\right)\ [mm] \qquad (4–24)$$

p_1	$[mm]$	– lateral pitch
p_2	$[mm]$	– longitudinal pitch
d	$[mm]$	– outer diametr of the tube

The overall heat transfer coefficient k :

$$k = \frac{\psi \cdot \alpha_{1r}}{1 + \frac{\alpha_{1r}}{\alpha_{2r}}}\ [-] \qquad (4–25)$$

ψ	$[-]$	– coefficient of thermal efficiency,(ψ = 0,85)
p_2	$[mm]$	– longitudinal pitch

4.5.1 Tube design of low pressure superheater LPsh

The tube dimensions are given in *Table 4. 9* and the drawind of fin tube is shown at *Figure 4. 9.*

Table 4. 9 Smooth tube dimensions of LPsh

Tube dimensions	Indication	Amount	Unit
Outer diameter	*D*	48.3	[mm]
Wall thickness	*t*	3.2	[mm]
Inner diameter	*d*	41.9	[mm]

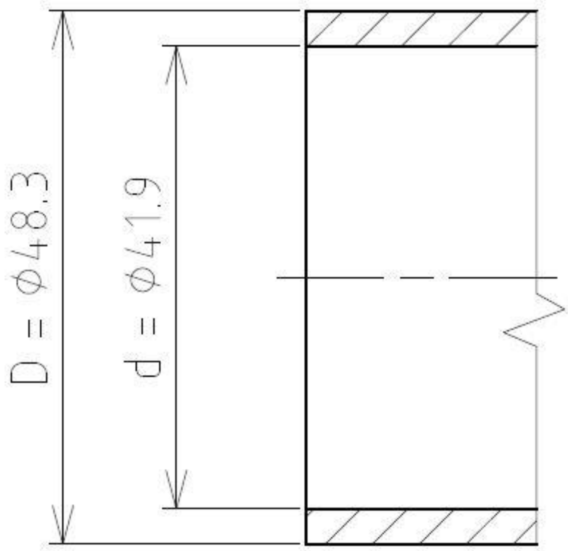

Figure 4. 9 Smooth tube dimensions of LPsh

4.5.2 Amount of tubes in one line LPsh

Lateral pitch:

Lateral pitch was chosen based on the allowable velocity of steam

$p_1 = 131\ mm.$

Longitudinal pitch after consultation with supervissor was choosen:

$p_2 = 117\ mm.$

Amount of tubes in one longitudinal line:

$$n_{TU} = \frac{w}{p_1} - \frac{1}{2} = \frac{2.7}{0.131} - \frac{1}{2} = 20.11$$

The real tubes amount in a single longitudinal line: $n_{TU} = 20.11\ => \mathbf{20}$

Flow velocity of steam:

In the next step it is important to determine the mean specific volume of steam which is given by average temperature and pressure in LPsh. The mean specific volume of steam is then calculated by the softwar XSTEAM.

$$t_{11-12} = \frac{t_{11} + t_{12}}{2} = \frac{180 + 156.15}{2} = 168.075\ °C$$

$$p_{11-12} = \frac{p_{11} + p_{12}}{2} = \frac{0.46 + 0.56}{2} = 0.51\ MPa$$

$$v_{11-12} = f(t_{11-12}, p_{11-12}) = 0.38422\ m^3/kg$$

The flow velocity of steam – formulas (3-1;3-2) :

The flow velocity of steam should be in range 12-25 m/s.

$$v_s = \frac{M_s \cdot v_s}{S_s} = \frac{4 \cdot 0.95 \cdot M_{LP} \cdot v_{11-12}}{\pi \cdot d^2 \cdot n_{TU}} = \frac{4 \cdot 1.51 \cdot 0.38422}{\pi \cdot 0.0419^2 \cdot 20} = 21.04\ m/s.$$

Real flue gas volume flow:

For the calculation of the real flue gas volume flow the calculated enthalpy and temperature at point F in Chapter 2.3.2.8 will be used.

The average temperature of the flue gas stream:

$$T_{E-F} = \frac{T_E^{REAL} + T_F}{2} = \frac{262.198 + 260.46}{2} = 261.329\ °C.$$

A real volumetric flow takes into account the flue gas temperature:

$$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{E-F} + 273.15}{273.15} = 39.15 \cdot \frac{261.329 + 273.15}{273.15} = 76.606\ m^3/s.$$

A real cross section of the the flue gas duct:

$$S_{DUCT}^{REAL} = h \cdot w - h \cdot D \cdot n_{TU}$$

$$S_{DUCT}^{REAL} = 8.5 \cdot 2.7 - 8.5 \cdot 0.0419 \cdot 20 = 14.739\ m^2$$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{76.606}{14.739} = 5.197\ m/s.$$

4.5.3 Heat transfer coefficient of low pressure superheater LPsh

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + p_2{}^2} - D} = \frac{131 - 117}{\sqrt{\left(\frac{131}{2}\right)^2 + 117^2} - 48.3} = 0.964019[-]$$

Relative longitudinal pitch σ_1:

$$\sigma_1 = \frac{p_1}{D} = \frac{131}{48.3} = 2.71$$

From literature [1] for coefficient of relative pitch $\varphi_\sigma = 0.1 \div 1.7$ will be determined corection factor that takes into account the geometric arrangement of tube bank and its spacing of tube bundles c_s.[8]

$$c_s = 0.34 \cdot (\varphi_\sigma)^2 = 0.34 \cdot (1.01)^2 = 0.31$$

In next step from literature [1] for expected number of longitudial lines $n_{LI\_exp} < 10$ and for relative transversep pitch $\sigma_1 < 3$ the correction factor taking account of the number of rows of tube c_z will be determined.[8]

$$c_z = 3.12 \cdot \left(n_{LI\_exp}\right)^{0,05} - 2.5 = 3.12 \cdot (1)^{0,05} - 2.5 = 0.62$$

The cofficinet of flue gas thermal conductivity λ_{FG}, Prandtl number Pr and coefficionet of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{E-F} = 261.329\ °C$ and for volume fraction of H_2O $x_{H_2O} = 7.8\ \%$ were defined:

$\lambda_{FG} = 0.04424\ W/m/K$

$\nu_{FG} = 4.04 \cdot 10^{-5}\ m^2/s$

$Pr\ \ = 0.649$

Coefficient of heat transfer convection α_C:

$$\alpha_C = c_S \cdot c_Z \cdot \frac{\lambda_{FG}}{D}\left(\frac{v_{FG} \cdot D}{\nu_{FG}}\right) \cdot \mathrm{Pr}[W/m^2/K]$$

$$\alpha_C = 0.31 \cdot 0.62 \cdot \frac{0.04424}{0.0483}\left(\frac{5.197 \cdot 0.0483}{4.04 \cdot 10^{-5}}\right) \cdot 0.649\ [W/m^2/K]$$

$$\alpha_C = 21.58\ W/m^2/K$$

The thickness of the effective radiation layer s :

$$s = 0.9 \cdot d \cdot \left(\frac{4}{\pi} \cdot \frac{p_1 \cdot p_2}{d^2} - 1\right) = 0.9 \cdot 41.9 \cdot \left(\frac{4}{\pi} \cdot \frac{131 \cdot 117}{41.9^2} - 1\right) = 381.47\ [mm]$$

Volume of triatomic gases x:

$$x = x_{H2O} + x_{CO2} = 0.078 + 0.044 = 0.122$$

Partial pressure of triatomic flue gases p_{par}:

$$p_{par} = p \cdot x = 0.1 \cdot 0.122 = 0.0123617\ \ [MPa]$$

$$p_{par} = 12361.7\ [Pa]$$

The coefficient of radiant absorption due to tri-atomic gases k_y :

$$k_y = \left(\frac{7.8 + 16 \cdot x_{H2O}}{3.16 \cdot \sqrt{p_{par} \cdot s}}\right) \cdot \left(1 - 0.37 \cdot \frac{T_{E-F} + 273.15}{1000}\right)$$

$$k_y = \left(\frac{7.8 + 16 \cdot 0.078}{3.16 \cdot \sqrt{0.0123617 \cdot 0.38147}}\right) \cdot \left(1 - 0.37 \cdot \frac{261.329\ + 273.15}{1000}\right)$$

$$k_y = 32.65\ \ [m/Mpa]$$

Coefficient k:

$$k = k_y \cdot x = 32.65 \cdot 0.122 = 3.983\ [m/Mpa]$$

The emisivity of tri-atomic gases:

$$a_{tr} = 1 - e^{-k \cdot p \cdot s} = 1 - e^{-3.983 \cdot 0.1 \cdot 0.38147} = 0.143$$

The cofficinet of flue gas thermal conductivity in the calculation was contemplated as: $a = 0.8$

The mean steam temperature:

$$T_{1-2} = \frac{T_{11} + T_{12}}{2} = \frac{180 + 156.15}{2} = 168.075\ [°C]$$

During the gas combustion $\Delta t = 25°C$ was choosen. The fouled wall surface temperature T_{FO} will be calculated.

$$T_{FO} = T_{1-2} + \Delta t = 168.075 + 25 = 193.075\ °C$$

Radiation heat transfer coefficient α_R:

$$\alpha_R = 5.7 \cdot 10^{-8} \cdot \frac{a_c + 1}{2} \cdot a_{tr} \cdot (T_{FG})^3 \cdot \frac{\left(\frac{T_{FO}}{T_{FG}}\right)^{3,6}}{1 - \frac{T_{FO}}{T_{FG}}} \ [W/m^2/K]$$

$$\alpha_R = 5.7 \cdot 10^{-8} \cdot \frac{0.8 + 1}{2} \cdot 0.143 \cdot (261.329 + 273.15)^3 \cdot \frac{\left(\frac{193.075}{261.329 + 273.15}\right)^{3,6}}{1 - \frac{193.075}{261.329 + 273.15}}$$

$$\alpha_R = 3.408 \ [W/m^2/K]$$

Heat transfer coefficient on the flue gas side α_{1r}:

Utilization factor for this calculation is $\xi = 1$

$$\alpha_{1r} = \xi \cdot \ (\alpha_C + \alpha_R) = 1 \cdot \ (21.58 + 3.408) = 24.984 \ [W/m^2/K]$$

Heat transfer coefficient on the steam side α_{2r}:

Using the software XSTEAM is for calculation of steam thermal conductivity λ_s, Prandtl number Pr and coefficient of dynamic steam viscosity μ_s .

$$t_{1-2} = 168.075\ °C$$

$$p_{1-2} = 0.51\ MPa$$

$$\lambda_s = f(t_{1-2}, p_{1-2}) = 0.031934\ W/m/K$$

$$\mu_s = f(t_{1-2}, p_{1-2}) = 1.4722 \cdot 10^{-5}\ Pa \cdot s$$

$$Pr = f(t_{1-2}, p_{1-2}) = 1.04637$$

Coefficient of steam viscosity ν_s :

$$\nu_s = \mu_s \cdot v_{1-2} = 1.4722 \cdot 10^{-5} \cdot 0.38422 = 5.66 \cdot 10^{-6}\ m^2/s$$

Heat transfer coefficient on the steam side α_{2r}:

$$\alpha_{2r} = 0.023 \cdot \frac{\lambda_S}{d_e} \cdot \left(\frac{v_S \cdot d_e}{\nu_S}\right)^{0,8} \cdot Pr^{0,4} \cdot c_t \cdot c_l \cdot c_m$$

$$\alpha_{2r} = 0.023 \cdot \frac{0.031934}{0.0419} \cdot \left(\frac{21.04 \cdot 0.0419}{5.66 \cdot 10^{-6}}\right)^{0,8} \cdot 1.04637^{0,4} \cdot 1 \cdot 1 \cdot 1$$

$$\alpha_{2r} = 254.022\ [W/m^2/K]$$

4.5.4 Overall coefficient of heat transfer in LPsh

$$k = \frac{\psi \cdot \alpha_{1r}}{1 + \frac{\alpha_{1r}}{\alpha_{2r}}} = \frac{0.85 \cdot 24.984}{1 + \frac{24.984}{254.022}} = 19.335$$

4.5.5 The number of longitudial lines in LPsh

Temperature diferences for calculation of logarithmic temperature drop:

$\Delta t_1 = T_E^{SKUT} - t_{11} = 262.198 - 180 = 82.198\ °C,$

$\Delta t_2 = T_\mathrm{F} - t_{12} = 260.46 - 156.15 = 104.31\ °C.$

Logarithmic temperature drop $\Delta \mathrm{t_{LN}}$:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{82.198\ - 104.31}{ln\left(\frac{82.198}{104.31}\right)} = 92.815\ °C$$

The total outer surface of the one meter long smooth tube $\mathrm{S_{1m}}$:

$S_{1m} =\ \pi \cdot D = \pi \cdot 0.0483 = 0.152\ [m^2]$

External heat transfer surface:

$$S_{EX} = \frac{\mathrm{Q}_{LPsh}}{k \cdot \Delta t_{LN}} = \frac{93360}{19.335 \cdot 92.815} = 52.023\ m^2$$

The number of longitudinal lines $\mathrm{n_{LI}}$:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{52.023}{8.5 \cdot 0.152 \cdot 20} = 2{,}013$$

I choose the number of lines $\boldsymbol{n_{LI} = 2}$.

4.5.6 Scheme of arrangement of tubes in LPsh

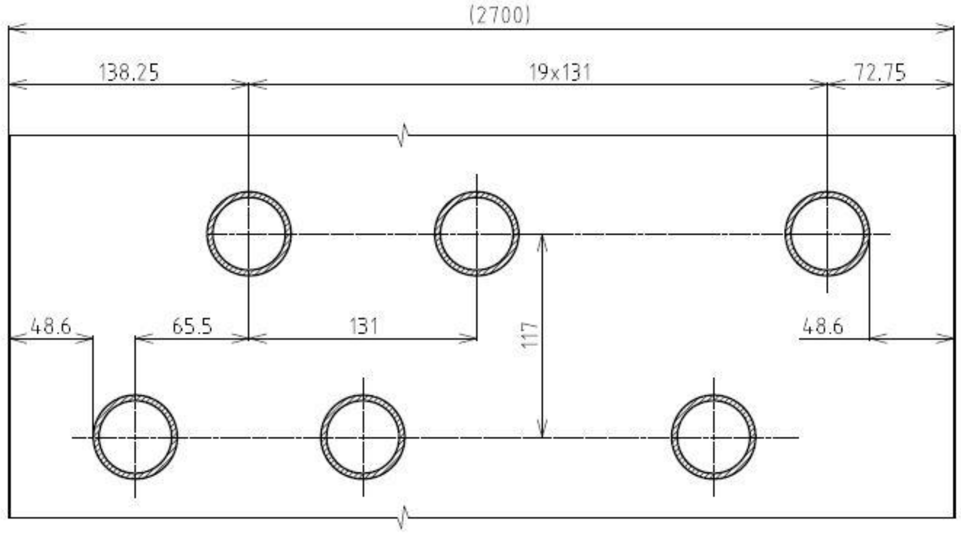

Figure 4. 10 Tube arrangement in LPsh

4.5.7 The real heat transferred in LPsh

The real external heat transfer surface S_{EX}^{REAL}:

$\mathrm{S}_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 2 \cdot 20 \cdot 8.5 \cdot 0.152 = 51.68\ m^2$

The real heat transferred in LPsh:

$$Q_{LPsh}^{REAL} = k \cdot S_{EX}^{REAL} \cdot \Delta t_{LN} = 19.335 \cdot 51.68 \cdot 92.815 = 92.744 \text{ kW}$$

Check of the real heat transferred

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left| \frac{Q_{LPsh}^{REAL} - Q_{LPsh}}{Q_{LPsh}^{REAL}} \right| = \left| \frac{92.744 - 93.36}{92.744} \right| = 0.664\ \%$$

$\Delta Q = 0.664\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.5.8 Real flue gas temperature in point F

The real enthalpy at point F:

$$H_F^{REAL} = H_E - \frac{Q_{LPsh}^{REAL}}{M_{FG}(1 - L_R)} = 353.802 - \frac{92.744}{39.15(1 - 0.004146)} = 351.423\ kJ/m^3$$

The real temperature at point F is determined by interpolation from *Table 2. 3*

$$T_F^{REAL} = 200 + (300 - 200)\frac{(H_F^{REAL} - H_{FG}^{200})}{(H_{FG}^{300} - H_{FG}^{200})} = 200 + 100 \cdot \frac{(351.423 - 268.01)}{(405.95 - 268.01)}$$

$$T_F^{REAL} = 260.474\ °C$$

Check of the real temperature:

The temerature difference may not be greater than 3°C:

$$\Delta T = |T_F^{REAL} - T_F| = |260.474 - 260.46| = 0.014\ °C$$

$\Delta T = 0.014°C < 3\ °C$ => Selected number of longitudinal lines is correct.

4.5.9 List of calculated values in LPsh

Table 4. 10 Calculated values of LPsh

Calculated values	Indication	Amount	Unit
Number of tubes in one longitudial line	n_{TU}	20	$[-]$
Real volumetric flow	M_{FG}^{REAL}	76.606	$[m^3/s]$
Real flue gas velocity	v_{FG}^{REAL}	5.197	$[m/s]$
Velocity of steam	v_s	21.04	$[m/s]$
Logarithmic temperature drop	Δt_{LN}	92.815	$[K]$
Number of longitudinal lines	n_{LI}	2	$[-]$
Overall coefficient of heat transfer	k	19.335	$[W/m^2$
Real external heat transfer surface	S_{EX}^{REAL}	51.68	$[m^2]$
Real heat transferred	Q_{LPsh}^{REAL}	92.744	[kW]
Real temperature at point F	T_F^{REAL}	260.474	$[°C]$

4.6 Proposal of high pressure economizer HPeco2

The calculation of high pressure economizer 2 is executed as high pressure economizer 3 calculation. The value of heat transfer coefficient is expected high and can be neglected.

4.6.1 Fin tube design of high pressure economizer HPeco2

The fin tube dimensions are given in *Table 4. 11* and the drawind of fin tube is shown at *Figure 4. 11*.

Table 4. 11 Fin tube dimensions of HPeco2

Tube dimensions	Indications	Amount	Unit
Outer diameter	D	31.8	[mm]
Wall thickness	t	4	[mm]
Inner diameter	d	22.8	[mm]
Fins height	h_f	13	[mm]
Fins thickness	t_f	1	[mm]
Fins per meter	n_f	230	[1/m]
Fins pitch	p_f	4.35	[mm]
Outer fin diameter	D_f	57.8	[mm]

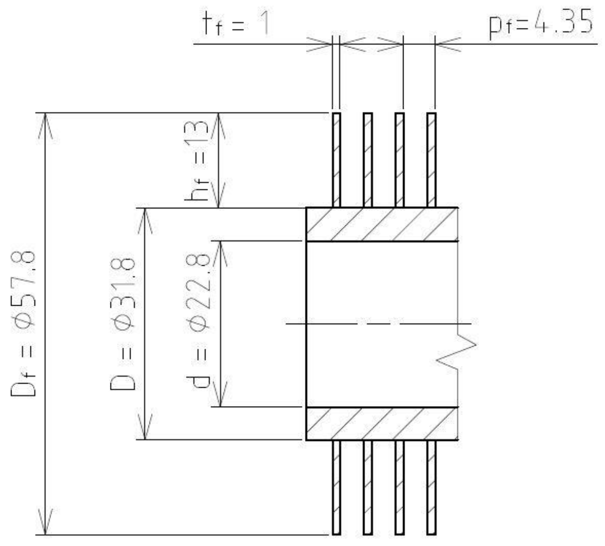

Figure 4. 11 Fin tube dimensions of HPeco2

4.6.2 Amount of fin tubes in one line HPeco2

Lateral pitch:

Latera pitch consist of outer fin diameter and the distance betwen the tube $a = 5.5\ mm$. $p_1 = D_f + a = 57.8 + 5.5 = 63.3\ mm$.

Longitudinal pitch after consultation with supervisor was choosen:

$p_2 = 92\ mm$.

Amount of tubes in one longitudinal line:

$$n_{TU} = \frac{w}{p_1} - \frac{1}{2} = \frac{2.7}{0.0633} - \frac{1}{2} = 42.15$$

The real tubes amount in a single longitudinal line: $n_{TU} = 42.15\ \Rightarrow\ \mathbf{42}$

Real flue gas volume flow:

For the calculation of the real flue gas volume flow the calculated enthalpy and temperature at point G in Chapter 2.3.2.9 will be used.

The average temperature of the flue gas stream:

$$T_{F-G} = \frac{T_F^{SKUT} + T_G}{2} = \frac{260.474 + 226.695}{2} = 243.585\ °C.$$

A real volumetric flow takes into account the flue gas temperature:

$$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{F-G} + 273.15}{273.15} = 39.15 \cdot \frac{243.585\ + 273.15}{273.15} = 74.063\ m^3/s.$$

A real cross section of the the flue gas duct :

$$S_{DUCT}^{REAL} = h \cdot \left(w - n_{TU} \cdot \left(D + 2 \cdot h_f \cdot t_f \cdot n_f\right)\right)$$

$$S_{DUCT}^{REAL} = 8.5 \cdot \left(2.7 - 42 \cdot (0.0318 + 2 \cdot 0.013 \cdot 0.001 \cdot 230)\right) = 9.463 m^2$$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{74.063}{9.463} = 7.827\ m/s.$$

The feed water flow velocity in HPeco2:

In the next step is important to determine the mean specific volume of steam which is given by average temperature and pressure in HPeco2. The mean specific volume of steam is then calculated by the softwar XSTEAM.

$$t_{7-8} = \frac{t_7 + t_8}{2} = \frac{220 + 145}{2} = 182{,}5\ °C$$

$$p_{7-8} = \frac{p_7 + p_8}{2} = \frac{6{,}3 + 6{,}4}{2} = 6{,}35\ MPa$$

$$v_{7-8} = f(t_{7-8}, p_{7-8}) = 0{,}001126\ m^3/kg$$

$$v_s = \frac{M_s \cdot v_s}{S_s} = \frac{4 \cdot 0.95 \cdot M_{HP} \cdot v_{7-8}}{\pi \cdot d^2 \cdot n_{TU}} = \frac{4 \cdot 0.95 \cdot 5.79 \cdot 0.001126}{\pi \cdot 0.0238^2 \cdot 42} = 0.3315\ m/s.$$

Due to slow feed water flow velocity in economizer it is necessary to split heat transfer surfect into three parts as you can see in te *Figure 4. 12*. By this division it is possible to increase feed water flow velocity. In our case three times.

$$v_s = 3 \cdot 0.3315 = 0.995\ m/s$$

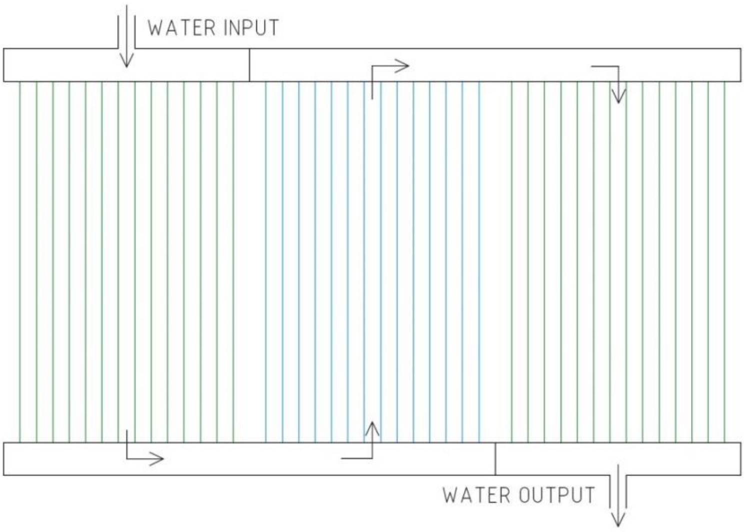

Figure 4. 12 Scheme of devided heat transfer surface.in HPeco2

4.6.3 Heat transfer coefficient of high pressure ecomonizer HPeco2

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2} - D} = \frac{63.3 - 31.8}{\sqrt{\left(\frac{63.3}{2}\right)^2 + 92^2} - 31.8} = 0.48098$$

The cofficinet of flue gas thermal conductivity λ_{FG} and coefficionet of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{F-G} = 243.585°C$ and for volume fraction of H_2O $x_{H_2O} = 7.8$ % were defined:

$$\lambda_{FG} = 0.04282 W/m/K$$

$$\nu_{FG} = 3.806 \cdot 10^{-5}\ m^2/s$$

Coefficient of heat transfer convection α_C:

Coefficient of the number of longitudinal line $c_Z = 0.95$ after consultation with supervisor was determined.

$$\alpha_C = 0.23 \cdot c_Z \cdot {\varphi_\sigma}^{0,2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0,54} \cdot \left(\frac{h_f}{p_f}\right)^{-0,14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0,65}$$

$$\alpha_C = 0.23 \cdot 0.95 \cdot 0.48098^{0,2} \cdot \frac{0.04282}{0.00435} \cdot \left(\frac{0.0318}{0.00435}\right)^{-0,54} \cdot \left(\frac{0.013}{0.00435}\right)^{-0,14} \cdot \left(\frac{7.827 \cdot 0.00435}{3.806 \cdot 10^{-5}}\right)^{0,65}$$

$$\alpha_C = 45.119\ W/m^2/K$$

Coefficient β:

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot (1 + \varepsilon \cdot \psi_f \cdot \alpha_C)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 45.119}{0.001 \cdot 40 \cdot (1 + 0.0045 \cdot 0.85 \cdot 45.119)}} = 40.439$$

Product of $\beta \cdot h_r$:

$$\beta \cdot h_f = 40.439 \cdot 0.013 = 0.526$$

Ratio $\frac{D_f}{D}$:

$$\frac{D_f}{D} = \frac{57.8}{31.8} = 1.82$$

Now is possible to determine E from nomogram in literature [1]

$E = 0.92.$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} = \frac{\left(\frac{57.8}{31.8}\right)^2 - 1}{\left(\frac{57.8}{31.8}\right)^2 - 1 + 2 \cdot \left(\frac{4.35}{31.8} - \frac{1}{31.8}\right)} = 0.91626$$

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} = 1 - 0.91626 = 0.083744$$

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C}$$

$$\alpha_{1r} = [0.91626 \cdot 0.92 \cdot 1 + 0.083744] \cdot \frac{0.85 \cdot 45.119}{1 + 0.0045 \cdot 0.85 \cdot 45.119} = 30.309 W/m^2/K$$

4.6.4 Overall heat transfer coefficient for HPeco2

Heat transfer coefficient on the steam side is in economizer neglected due to heat transfer coefficient on the steam side is many times smaller than heat transfer coefficient on the flue gas side ($\alpha_{2r} \ll \alpha_{1r}$). Only the heat transfer on the flue gas side is taken into account.

Overall coefficient of heat transfer will calculated as $k = \alpha_{1r} = 30.309\ W/m^2/K$.

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot (D_f{}^2 - D^2)}{4} + \pi \cdot D_f \cdot t_f = \frac{2 \cdot \pi \cdot (0.0578^2 - 0.0318^2)}{4} + \pi \cdot 0.0578 \cdot 0.001$$

$$S_{1f} = 0.003841\ m^2$$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot (1 - n_f \cdot t_f) + n_f \cdot S_{1f} = \pi \cdot 0.0318 \cdot (1 - 230 \cdot 0.001) + 230 \cdot 0.003841$$

$$S_{1m} = 0.96036\ m^2$$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d = \pi \cdot 0.0238 = 0.07477\ m^2$$

4.6.5 The number of longitudinal lines in HPeco2

Temperature diferences for calculation of logarithmic temperature drop:

$$\Delta t_1 = T_F^{REAL} - t_7 = 260.474 - 220 = 40.474\ °C$$

$$\Delta t_2 = T_G - t_8 = 226.695 - 145 = 81.695\ °C$$

Logarithmic temperature drop Δt_{LN}:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{40.474 - 81.695}{ln\left(\frac{40.474}{81.695}\right)} = 58.629\ K$$

External heat transfer surface:

$$S_{EX} = \frac{Q_{HPeco2}}{k \cdot \Delta t_{LN}} = \frac{1816600}{30.309 \cdot 58.629} = 1021.195\ m^2$$

The number of longitudinal lines n_{LI}

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{1021.195}{8.5 \cdot 0.96036 \cdot 42} = 2.98$$

I choose the number of lines $\boldsymbol{n_{LI} = 3}$.

4.6.6 Scheme of arrangement of tubes in HPeco2

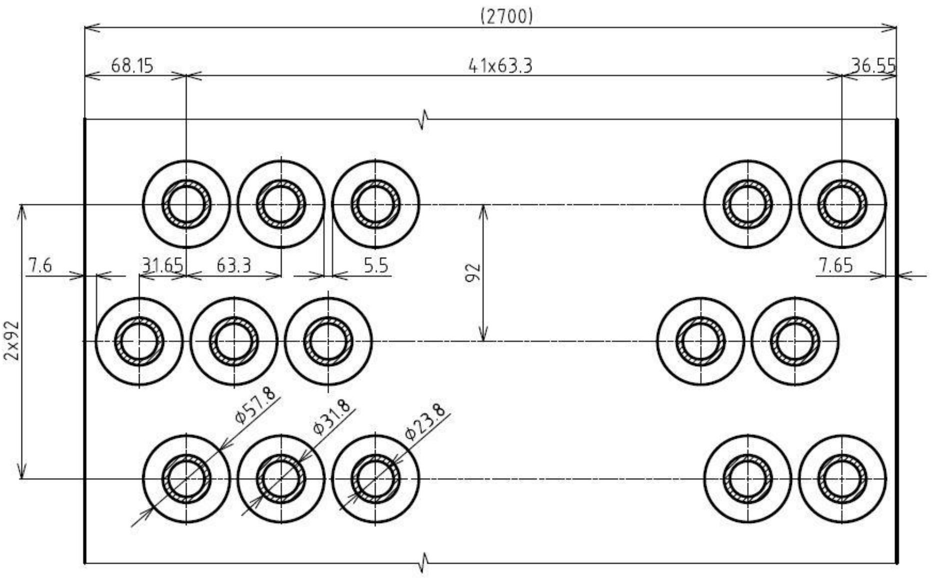

Figure 4. 13 Tube arrangement in HPeco2

4.6.7 The real heat transferred in HPeco2

The real external heat transfer surface S_{EX}^{REAL}:

$$S_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 3 \cdot 42 \cdot 8.5 \cdot 0.96036 = 1028.54\ m^2$$

The real heat transferred in HPeco2:

$$Q_{HPeco2}^{REAL} = k \cdot S_{EX}^{REAL} \cdot \Delta t_{LN} = 30.309 \cdot 1028.54 \cdot 58.692 = 1829.666\ \mathrm{kW}$$

Check of the real heat transferred:

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left|\frac{Q_{HPeco2}^{REAL} - Q_{HPeco2}}{Q_{HPeco2}^{REAL}}\right| = \left|\frac{1829.666 - 1816.6}{1829.666}\right| = 0.714\ \%$$

$\Delta Q = 0.714\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.6.8 Real flue gas temperature in point G

The real enthalpy at point G:

$$H_G^{REAL} = H_F^{REAL} - \frac{Q_{HPeco2}^{REAL}}{M_{FG}(1 - L_R)} = 351.423 - \frac{1829.666}{39.15(1 - 0.004146)} = 304.49 kJ/m^3$$

The real temperature at point B is determined by interpolation from *Table 2. 3*

$$T_G^{REAL} = 200 + (300 - 200)\frac{(H_G^{REAL} - H_{FG}^{200})}{(H_{FG}^{300} - H_{FG}^{200})} = 200 + 100 \cdot \frac{(304.49 - 268.01)}{(405.95 - 268.01)}$$

$$T_G^{REAL} = 226.452\ °C$$

Check of the real temperature:

The temerature difference may not be greater than 3°C:

$$\Delta T = |T_G - T_G^{REAL}| = |226.657 - 226.452\ | = 0.243\ °C$$

$\Delta T = 0.243\ °C < 3\ °C$ => Selected number of longitudinal lines is correct.

4.6.9 List of calculated values in HPeco2

Table 4. 12 Calculated values of HPeco2

Calculated values	Indication	Amount	Unit
Number of tubes in one longitudinal line	n_{TU}	42	$[-]$
Real volumetric flow	M_{FG}^{REAL}	74.063	$[m^3/s]$
Real flue gas velocity	v_{FG}^{REAL}	7.827	$[m/s]$
The feed water flow velocity	v_s	0.995	$[m/s]$
Logarithmic temperature drop	Δt_{LN}	58.692	$[K]$
Number of longitudinal lines	n_{LI}	3	$[-]$
Overall coefficient of heat transfer	k	30.309	$[W/m^2$
Real external heat transfer surface	S_{EX}^{REAL}	1028.54	$[m^2]$

Real heat transferred	Q_{HPeco2}^{REAL}	1829.666	[kW]
Real temperature at point G	T_G^{REAL}	226.452	[°C]

4.7 Proposal of Low pressure evaporator LPeva

Calculation method of low pressure evaporator is similar to the one for high pressure evaporator.

4.7.1 Fin tube design of low pressure evaporater LPeva

The fin tube dimensions are shown in *Table 4. 13* and the drawing of fin tube is shown at *Figure 4. 14*

Table 4. 13 Fin tube dimensions of LPeva

Tube dimensions	Indications	Amount	Unit
Outer diameter	D	57	[mm]
Wall thickness	t	3.6	[mm]
Inner diameter	d	49.8	[mm]
Fins height	h_f	19	[mm]
Fins thickness	t_f	1	[mm]
Fins per meter	n_f	230	[1/m]
Fins pitch	p_f	4.35	[mm]
Outer fin diameter	D_f	95	[mm]

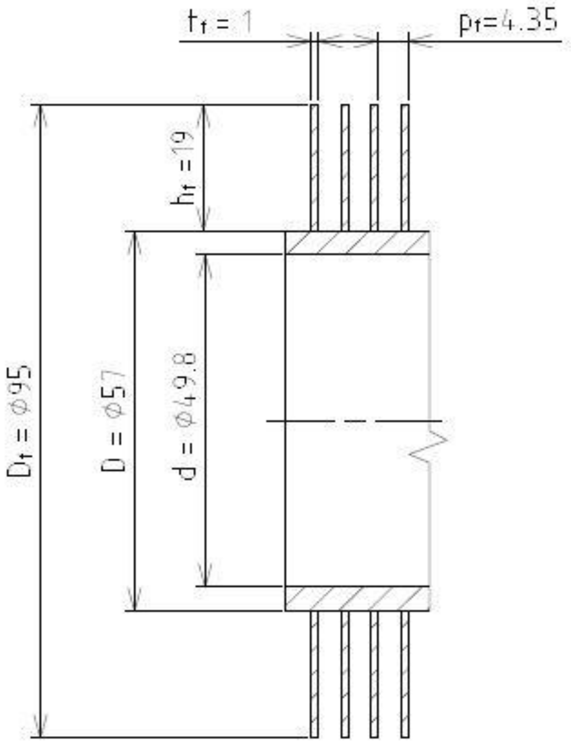

Figure 4. 14 Fin tube dimensions of LPeva

4.7.2 Amount of fin tubes in one line LPeva

Lateral pitch:

Lateral pitch consist of outer rib diameter and the distance betwen the tube $a = 6\ mm$. $p_1 = D_f + a = 95 + 6 = 101\ mm$.

Longitudinal pitch after consultation with supervissor was choosen:

$p_2 = 117\ mm$.

Amount of tubes in one longitudinal line:

$$n_{TU} = \frac{w}{p_1} - \frac{1}{2} = \frac{2.7}{0.101} - \frac{1}{2} = 26.23$$

The real tubes amount in a single longitudinal line: $n_{TU} = 26.23\ \Rightarrow\ \mathbf{26}$

Real flue gas volume flow:

For the calculation of the real flue gas volume flow will use the calculated enthalpy and temperature at point H in Chapter 2.3.2.4

The average temperature of the flue gas stream:

$$T_{G-H} = \frac{T_G^{REAL} + T_H}{2} = \frac{226.452 + 166.15}{2} = 196.301\ °C.$$

A real volumetric flow takes into account the flue gas temperature:

$$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{G-H} + 273.15}{273.15} = 39.15 \cdot \frac{196.301 + 273.15}{273.15} = 67.285\ m^3/s.$$

A real cross section of the the flue gas duct:

$$S_{DUCT}^{REAL} = h \cdot \left(w - n_{TU} \cdot (D + 2 \cdot h_r \cdot t_r \cdot n_r)\right)$$

$$S_{DUCT}^{REAL} = 8.5 \cdot \left(2.7 - 26 \cdot (0.057 + 2 \cdot 0.019 \cdot 0.001 \cdot 230)\right) = 8.422 m^2$$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{67.285}{8.422} = 7.99\ m/s.$$

4.7.3 Heat transfer coefficient of low pressure evaporater LPeva

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2} - D} = \frac{101 - 57}{\sqrt{\left(\frac{101}{2}\right)^2 + 117^2} - 57} = 0.6247$$

The cofficinet of flue gas thermal conductivity λ_{FG} and coefficionet of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{G-H} = 196.301\ °C$ and for volume fraction of H_2O $x_{H_2O} = 7.8\ \%$ were defined:

$\lambda_{FG} = 0.03900\ W/m/K$

$\nu_{FG} = 3.195 \cdot 10^{-5}\ m^2/s$

Coefficient of heat transfer convection α_C:

$$\alpha_C = 0.23 \cdot c_Z \cdot {\varphi_\sigma}^{0,2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0,54} \cdot \left(\frac{h_f}{p_f}\right)^{-0,14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0,65}$$

$$\alpha_C = 0.23 \cdot 0.95 \cdot 0.6247^{0,2} \frac{0.03900}{0.00435}\left(\frac{0.057}{0.00435}\right)^{-0,54} \left(\frac{0.019}{0.00435}\right)^{-0,14} \left(\frac{7.99 \cdot 0.00435}{3.195 \cdot 10^{-5}}\right)^{0,65}$$

$$\alpha_C = 35.818\ W/m^2/K$$

Coefficient β :

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot \left(1 + \varepsilon \cdot \psi_f \cdot \alpha_C\right)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 35.818}{0.001 \cdot 40 \cdot (1 + 0.0045 \cdot 0.85 \cdot 35.818)}} = 36.59$$

Product of $\beta \cdot h_r$:

$$\beta \cdot h_f = 36.59 \cdot 0.019 = 0.695$$

Ratio $\frac{D_f}{D}$:

$$\frac{D_f}{D} = \frac{95}{57} = 1.67$$

Now is possible to determine E from nomogram in literature [1]

$$E = 0.84$$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} = \frac{\left(\frac{95}{57}\right)^2 - 1}{\left(\frac{95}{57}\right)^2 - 1 + 2 \cdot \left(\frac{4.35}{57} - \frac{1}{57}\right)} = 0.93802$$

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} = 1 - 0.93802 = 0.06198$$

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C}$$

$$\alpha_{1r} = [0.93802 \cdot 0.84 \cdot 1 + 0.06198] \cdot \frac{0.85 \cdot 35.818}{1 + 0.0045 \cdot 0.85 \cdot 35.818} = 22.758\ W/m^2/K$$

4.7.4 Overall coefficient of heat transfer for LPeva

Heat transfer coefficient on the steam side is in evaporator neglected due to heat transfer coefficient on the steam side is many times smaller than heat transfer coefficient on the flue gas side ($\alpha_{2r} \ll \alpha_{1r}$). Only the heat transfer on the flue gas side is taken into account.

Overall coefficient of heat transfer will calculated as $k = \alpha_{1r} = 22.758\ W/m^2/K$

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot \left({D_f}^2 - D^2\right)}{4} + \pi \cdot D_f \cdot t_f = \frac{2 \cdot \pi \cdot (0.095^2 - 0.057^2)}{4} + \pi \cdot 0.095 \cdot 0.001$$

$$S_{1f} = 0.009371\ m^2$$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot \left(1 - n_f \cdot t_f\right) + n_f \cdot S_{1f} = \pi \cdot 0.057 \cdot (1 - 230 \cdot 0.001) + 230 \cdot 0.009371$$

$$S_{1m} = 2.293215\ m^2$$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d = \pi \cdot 0.0498 = 0.156451\ m^2$$

4.7.5 The number of longitudinal lines for LPeva

Temperature diferences for calculation of logarithmic temperature drop:

$$\Delta t_1 = T_G^{REAL} - t_{12} = 226.452 - 156.15 = 70.302\ °C$$

$$\Delta t_2 = T_{\mathrm{G}} - t_{13} = 166.15 - 156.15 = 10\ °C$$

Logarithmic temperature drop Δt_{LN}:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{70.302 - 10}{ln\left(\frac{70.302}{10}\right)} = 30.921\ K$$

External heat transfer surface:

$$S_{EX} = \frac{Q_{LPeva}}{k \cdot \Delta t_{LN}} = \frac{3194950}{22.758 \cdot 30.921} = 4540.215\ m^2$$

The number of longitudinal lines n_{LI}:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{4540.215}{8.5 \cdot 2.293215 \cdot 26} = 8.96$$

I choose the number of lines $\boldsymbol{n_{LI} = 9}$.

4.7.6 Scheme of arrangement of tubes in Leva

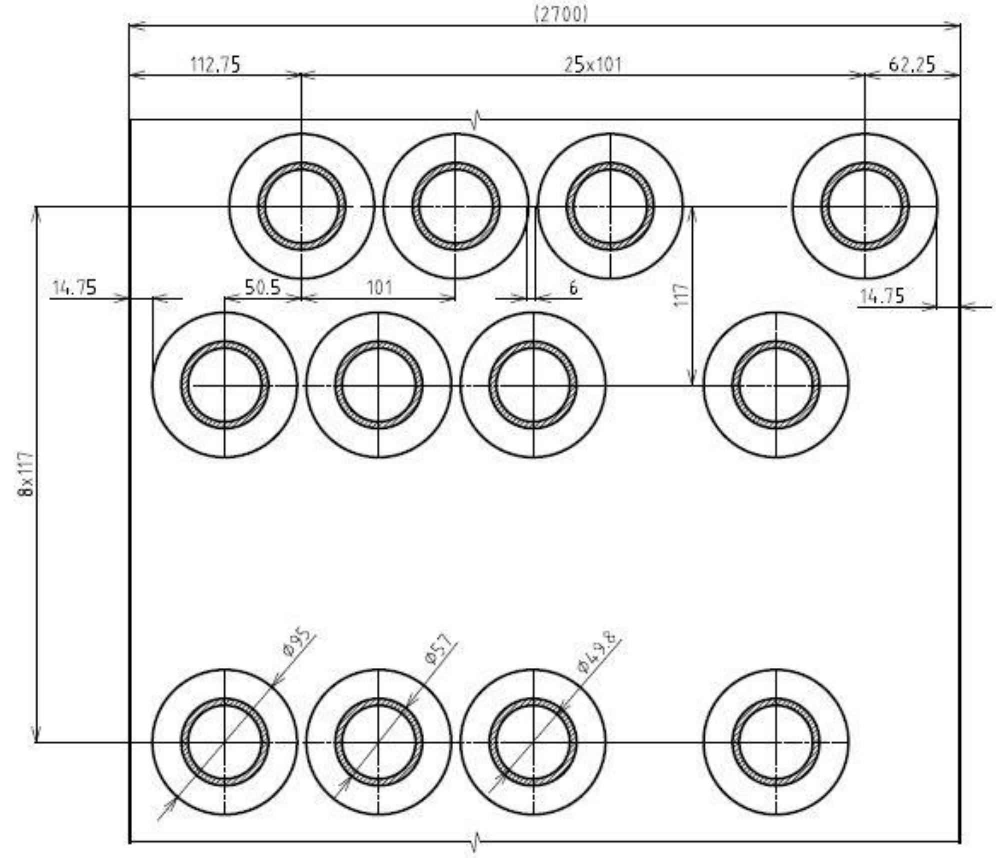

Figure 4. 15 Tube arrangement in LPeva

4.7.7 The real heat transferred in LPeva

The real external heat transfer surface S_{EX}^{REAL}:

$$S_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 9 \cdot 26 \cdot 8.5 \cdot 2.293215 = 4561.205\, m^2$$

The real heat transferred in LPeva:

$$Q_{LPeva}^{REAL} = k \cdot S_{EX}^{REAL} \cdot \Delta t_{LN} = 22.758 \cdot 4561.205 \cdot 30.921 = 3209.72 \text{ kW}$$

Check of the real heat transferred

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left| \frac{Q_{LPeva}^{REAL} - Q_{LPeva}}{Q_{LPeva}^{REAL}} \right| = \left| \frac{3209.72 - 3194.95}{3209.72} \right| = 0.460\ \%$$

$\Delta Q = 0.460\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.7.8 Real flue gas temperature in point H

The real enthalpy at point H:

$$H_H^{REAL} = H_G^{REAL} - \frac{Q_{LPeva}^{REAL}}{M_{FG}(1 - L_R)} = 304.49 - \frac{3209.72}{39.15(1 - 0.004146)} = 222.17\ kJ/m^3$$

The real temperature at point H is determined by interpolation from *Table 2. 3*

$$T_H^{REAL} = 100 + (200 - 100)\frac{(H_H^{REAL} - H_{FG}^{100})}{(H_{FG}^{200} - H_{FG}^{100})} = 200 + 100 \cdot \frac{(222.17 - 133.27)}{(268.01 - 133.27)}$$

$$T_H^{REAL} = 165.98\ °C$$

Check of the real temperature:

The temerature difference may not be greater than 3°C:

$$\Delta T = |T_H - T_H^{REAL}| = |166.15 - 165.98| = 0.171\ °C$$

$\Delta T = 0.171\ °C < 3\ °C$ => Selected number of longitudinal lines is correct

4.7.9 List of calculated values in LPeva

Table 4. 14 Calculated values of LPeva

Calculated values	Indication	Amount	Unit
Number of tubes in one longitudial line	n_{TU}	26	$[-]$
Real volumetric flow	M_{FG}^{REAL}	67.285	$[m^3/s]$
Real flue gas velocity	v_{FG}^{REAL}	7.99	$[m/s]$
Logarithmic temperature drop	Δt_{LN}	30.921	$[K]$
Number of longitudinal lines	n_{LI}	9	$[-]$
Overall coefficient of heat transfer	k	22758	$[W/m^2$
Real external heat transfer surface	S_{EX}^{REAL}	4561.205	$[m^2]$
Real heat transferred	Q_{LPeva}^{REAL}	3209.72	[kW]
Real temperature at point H	T_H^{REAL}	165.98	$[°C]$

4.8 Proposal of low pressure economizer LPeco

The calculation of low pressure economizer is executed as high pressure economizer calculation. The value of heat transfer coefficient is expected high and can be neglected.

4.8.1 Fin tube design of low pressure economizer LPeco

The fin tube dimensions are given in *Table 4. 15* and the drawing of fin tube is shown at *Figure 4. 16*

Table 4. 15 Fin tube dimensions of LPeco

Tube dimensions	Indications	Amount	Unit
Outer diameter	D	31.8	[mm]
Wall thickness	t	3.2	[mm]
Inner diameter	d	25.4	[mm]
Fins height	h_f	15	[mm]
Fins thickness	t_f	1	[mm]
Fins per meter	n_f	160	[1/m]
Fins pitch	p_f	6.25	[mm]
Outer fin diameter	D_f	61.8	[mm]

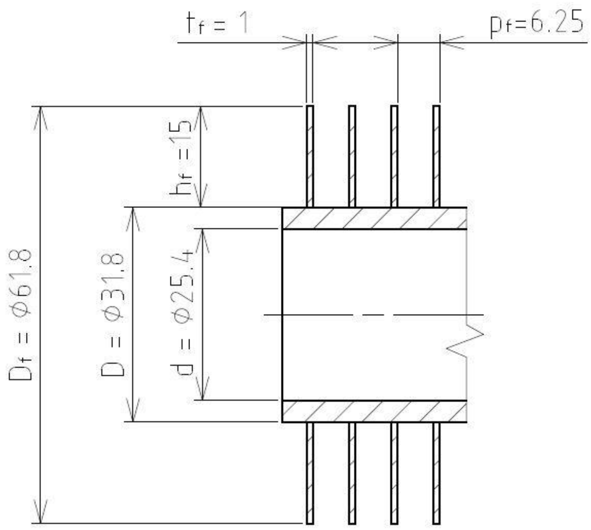

Figure 4. 16 Fin tube dimensions of LPeva

4.8.2 Amount of fin tubes in one line LPeco

<u>Lateral pitch:</u>

Latera pitch consist of outer fin diameter and the distance betwen the tube $a = 12\ mm$. $p_1 = D_f + a = 61.8 + 12 = 73.8\ mm.$

<u>Longitudinal pitch after consultation with supervisor was chosen:</u>

$p_2 = 92\ mm.$

<u>Amount of tubes in one longitudinal line:</u>

$$n_{TU} = \frac{w}{p_1} - \frac{1}{2} = \frac{2.7}{0.0738} - \frac{1}{2} = 36.08$$

The real tubes amount in a single longitudinal line: $n_{TU} = 36.08 \ \ => \ \mathbf{36}$

Real flue gas volume flow:

For the calculation of the real flue gas volume flow will use the calculated enthalpy and temperature at point I in Chapter 2.3.2.10

The average temperature of the flue gas stream:

$$T_{H-I} = \frac{T_H^{REAL} + T_I}{2} = \frac{165.98 + 155.14}{2} = 160.56\ °C.$$

A real volumetric flow takes into account the flue gas temperature:

$$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{H-I} + 273.15}{273.15} = 39.15 \cdot \frac{160.56 + 273.15}{273.15} = 62.163\ m^3/s.$$

A real cross section of the the flue gas duct:

$$S_{DUCT}^{REAL} = h \cdot \left(w - n_{TU} \cdot \left(D + 2 \cdot h_f \cdot t_f \cdot n_f\right)\right)$$

$$S_{DUCT}^{REAL} = 8.5 \cdot \left(2.7 - 36 \cdot (0.0318 + 2 \cdot 0.015 \cdot 0.001 \cdot 160)\right) = 11.75\ m^2$$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{62.163}{11.75} = 5.29\ m/s$$

The feed water flow velocity in LPeco:

In the next step is important to determine the mean specific volume of steam which is given by average temperature and pressure in LPeco. The mean specific volume of steam is then calculated by the softwar XSTEAM.

$$t_{14-15} = \frac{t_{14} + t_{15}}{2} = \frac{151.15 + 62}{2} = 106.575\ °C$$

$$p_{14-15} = \frac{p_{14} + p_{15}}{2} = \frac{0.56 + 0.66}{2} = 0.61\ MPa$$

$$v_{14-15} = f(t_{14-15}, p_{14-15}) = 0.00104847\ m^3/kg$$

$$v_s = \frac{M_s \cdot v_s}{S_s} = \frac{4 \cdot M_{LP} \cdot v_{14-15}}{\pi \cdot d^2 \cdot n_{TU}} = \frac{4 \cdot 0.95 \cdot 1.51 \cdot 0.00104847}{\pi \cdot 0.0254^2 \cdot 36} = 0.0825\ m/s.$$

Due to slow feed water flow velocity in economizer it is necessary to split heat transfer surfect into twelve parts as is shown in te *Figure 4. 17*. By this division it is possible to increase feed water flow velocity. In our case twelve times.

$$v_s = 12 \cdot 0.0825 = 0.99\ m/s$$

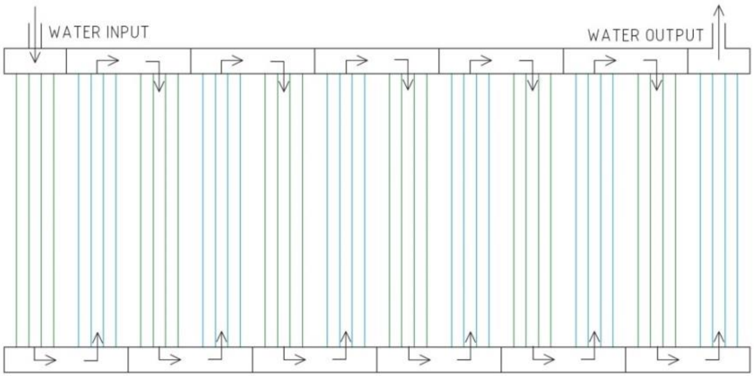

Figure 4. 17 Scheme of devided heat transfer surface in LPeco

4.8.3 Heat transfer coefficient of low pressure ecomonizer LPeco

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + {p_2}^2} - D} = \frac{73.8 - 31.8}{\sqrt{\left(\frac{73.8}{2}\right)^2 + 92^2} - 31.8} = 0.62385$$

The cofficinet of flue gas thermal conductivity λ_{FG} and coefficionet of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{H-I} = 160.56\ °C$ and for volume of H_2O $x_{H_2O} = 7.8\ \%$ were defined:

$$\lambda_{FG} = 0.035994\ W/m/K$$

$$\nu_{FG} = 2.8 \cdot 10^{-5}\ m^2/s$$

Coefficient of heat transfer convection α_C:

Coefficient of the number of longitudinal line $c_Z = 0.95$ after consultation with supervisor was determined.

$$\alpha_C = 0.23 \cdot c_Z \cdot {\varphi_\sigma}^{0,2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0,54} \cdot \left(\frac{h_f}{p_f}\right)^{-0,14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0,65}$$

$$\alpha_C = 0.23 \cdot 0.95 \cdot 0.62385^{0,2} \frac{0.035994}{0.00625} \left(\frac{0.0318}{0.00625}\right)^{-0,54} \left(\frac{0.015}{0.00625}\right)^{-0,14} \left(\frac{5.29 \cdot 0.00625}{2.8 \cdot 10^{-5}}\right)^{0,65}$$

$$\alpha_C = 41.742\ W/m^2/K$$

Coefficient β :

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot (1 + \varepsilon \cdot \psi_f \cdot \alpha_C)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 41.742}{0.001 \cdot 40 \cdot (1 + 0.0045 \cdot 0.85 \cdot 41.742)}} = 39.112$$

Product of $\beta \cdot h_r$:

$$\beta \cdot h_f = 39.112 \cdot 0.015 = 0.587$$

Ratio $\frac{D_f}{D}$:

$$\frac{D_f}{D} = \frac{61.8}{31.8} = 1.94$$

Now is possible to determine E from nomogram in literature [1]

$E = 0.86.$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} = \frac{\left(\frac{61.8}{31.8}\right)^2 - 1}{\left(\frac{61.8}{31.8}\right)^2 - 1 + 2 \cdot \left(\frac{6.25}{31.8} - \frac{1}{31.8}\right)} = 0.893727$$

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} = 1 - 0.893727 = 0.106273$$

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C}$$

$$\alpha_{1r} = [0.893727 \cdot 0.86 \cdot 1 + 0.106273] \cdot \frac{0.85 \cdot 41.742}{1 + 0.0045 \cdot 0.85 \cdot 41.742} = 26.768\ W/m^2/K$$

4.8.4 Overall heat transfer coefficient for LPeco

Heat transfer coefficient on the steam side is in economizer neglected due to heat transfer coefficient on the steam side is many times smaller than heat transfer coefficient on the flue gas side ($\alpha_{2r} \ll \alpha_{1r}$). Only the heat transfer on the flue gas side is taken into account.

Overall coefficient of heat transfer will calculated as $k = \alpha_{1r} = 26.768\ 7\ W/m^2/K$.

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot \left({D_f}^2 - D^2\right)}{4} + \pi \cdot D_f \cdot t_f = \frac{2 \cdot \pi \cdot (0.0618^2 - 0.0318^2)}{4} + \pi \cdot 0.0618 \cdot 0.001$$

$$S_{1f} = 0.0046\ m^2$$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot \left(1 - n_f \cdot t_f\right) + n_f \cdot S_{1f} = \pi \cdot 0.0318 \cdot (1 - 160 \cdot 0.001) + 200 \cdot 0.0046$$

$$S_{1m} = 0.81992\ m^2$$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d = \pi \cdot 0{,}0254 = 0.0798\ m^2$$

4.8.5 The number of longitudinal lines in LPeco

Temperature diferences for calculation of logarithmic temperature drop:

$\Delta t_1 = T_H^{REAL} - t_{14} = 165.98 - 151.15 = 14.83\ °C$

$\Delta t_2 = T_\mathrm{I} - t_{15} = 155.14 - 62 = 93.14\ °C$

Logarithmic temperature drop $\Delta \mathrm{t_{LN}}$:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{14.83\ - 93.14}{ln\left(\frac{14.83}{93.14}\right)} = 42.618\ K$$

External heat transfer surface:

$$S_{EX} = \frac{\mathrm{Q}_{LPeco}}{k \cdot \Delta t_{LN}} = \frac{774990}{26.786 \cdot 42.618} = 499.273\ m^2$$

The number of longitudinal lines $\mathrm{n_{LI}}$:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{499.273}{8.5 \cdot 0.81992 \cdot 36} = 1.99$$

I choose the number of lines $\boldsymbol{n_{LI} = 2}$.

4.8.6 SCHEM of arrangement of tubes in LPeco

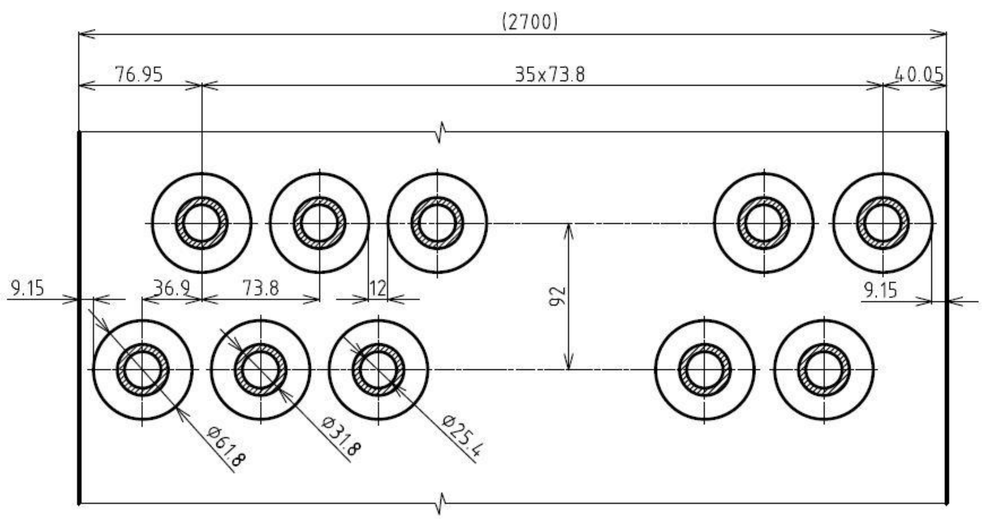

Figure 4. 18 Tube arrangement in LPeco

4.8.7 The real heat transferred in LPeco

The real external heat transfer surface S_{EX}^{REAL}:

$\mathrm{S}_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 2 \cdot 36 \cdot 8.5 \cdot 0.81992 = 501.791\ m^2$

The real heat transferred in LPeco:

$Q_{LPeco}^{REAL} = k \cdot \mathrm{S}_{EX}^{REAL} \cdot \Delta t_{LN}\ = 26.768 \cdot 501.791 \cdot 42.618 = 572.442\ \mathrm{kW}$

Check of the real heat transferred

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left| \frac{Q_{LPeco}^{REAL} - Q_{LPeco}}{Q_{LPeco}^{REAL}} \right| = \left| \frac{572.442 - 569{,}57}{572.442} \right| = 0.502\ \%$$

$\Delta Q = 0.502\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.8.8 Real flue gas temperature in point I

The real enthalpy at point I:

$$H_I^{REAL} = H_H^{REAL} - \frac{Q_{LPeco}^{REAL}}{M_{FG}(1 - L_R)} = 222.17 - \frac{572.442}{39.15(1 - 0.004146)} = 207.48\ kJ/m^3$$

The real temperature at point B is determined by interpolation from *Table 2. 3*

$$T_I^{REAL} = 100 + (200 - 100)\frac{(H_I^{REAL} - H_{FG}^{100})}{(H_{FG}^{200} - H_{FG}^{100})} = 200 + 100 \cdot \frac{(207.48 - 133.27)}{(268.01 - 133.27)}$$

$$T_I^{REAL} = 155.082\ °C$$

Check of the real temperature:

The temerature difference may not be greater than 3°C:

$$\Delta T = |T_I - T_I^{REAL}| = |155.14 - 155.082\ | = 0.058\ °C$$

$\Delta T = 0.058\ °C < 3\ °C$ => Selected number of longitudinal lines is correct.

4.8.9 List of calculated values in LPeco

Table 4. 16 Calculated values of LPeco

Calculated values	Indication	Amount	Unit
Number of tubes in one longitudial line	n_{TU}	36	$[-]$
Real volumetric flow	M_{FG}^{REAL}	62.163	$[\ m^3/s]$
Real flue gas velocity	v_{FG}^{REAL}	5.29	$[m/s]$
Velocity of steam	v_s	0.99	$[m/s]$
Logarithmic temperature drop	Δt_{LN}	42.618	$[K]$
Number of longitudinal lines	n_{LI}	2	$[-]$
Overall coefficient of heat transfer	k	26.768	$[W/m^2$
Real external heat transfer surface	S_{EX}^{REAL}	501.791	$[m^2]$
Real heat transferred	Q_{LPeco}^{REAL}	572.442	[kW]
Real temperature at point I	T_I^{REAL}	155.082	$[°C]$

4.9 Proposal of high pressure econimizer HPeco1

As in the high pressure economizer 3 and 2, in the high pressure economizer 1 the value of heat transfer coefficient is expected to be high and therefore can be neglected.

4.9.1 Fin tube design of high pressure economizer HPeco1

The fin tube dimensions are shown in *Table 4. 17* and the drawing of fin tube is shown at *Figue 4. 19.*

Table 4. 17 Fin tube dimensions of HPeco1

Tube dimensions	Indications	Amount	Unit
Outer diameter	D	31.8	[mm]
Wall thickness	t	4	[mm]
Inner diameter	d	23.8	[mm]
Fins height	h_f	15	[mm]
Fins thickness	t_f	1	[mm]
Fins per meter	n_f	250	[1/m]
Fins pitch	p_f	4	[mm]
Outer fin diameter	D_f	61.8	[mm]

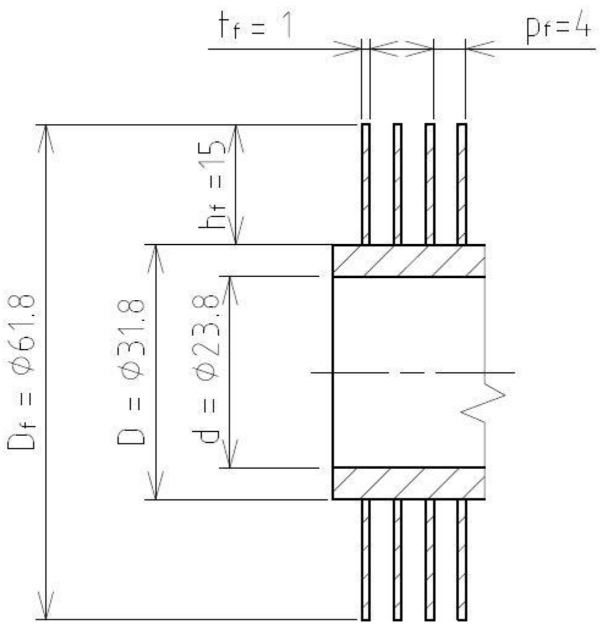

Figure 4. 19 Fin tube dimensions of HPeco1

4.9.2 Amount of fin tubes in one line HPeco1

<u>Lateral pitch:</u>

Latera pitch consist of outer fin diameter and the distance betwen the tube $a = 6\ mm$. $p_1 = D_f + a = 61.8 + 6 = 67.8\ mm$.

<u>Longitudinal pitch after consultation with supervissor was choosen:</u>

$p_2 = 92\ mm$.

Amount of tubes in one longitudinal line:

$$n_{TU} = \frac{w}{p_1} - \frac{1}{2} = \frac{2.7}{0.0678} - \frac{1}{2} = 39.3$$

The real tubes amount in a single longitudinal line: $n_{TU} = 39.3 \;=>\; \mathbf{39}$

Real flue gas volume flow:

For the calculation of the real flue gas volume flow will use the calculated enthalpy and temperature at point J in Chapter 2.3.2.11

The average temperature of the flue gas stream:

$$T_{I-J} = \frac{T_I^{REAL} + T_J}{2} = \frac{155.082 + 118.48}{2} = 136.78\ °C.$$

A real volumetric flow takes into account the flue gas temperature:

$$M_{FG}^{REAL} = M_{FG} \cdot \frac{T_{I-J} + 273.15}{273.15} = 39.15 \cdot \frac{136.78 + 273.15}{273.15} = 58.754\ m^3/s.$$

A real cross section of the the flue gas duct :

$$S_{DUCT}^{REAL} = h \cdot \left(w - n_{TU} \cdot \left(D + 2 \cdot h_f \cdot t_f \cdot n_f\right)\right)$$

$$S_{DUCT}^{REAL} = 8.5 \cdot \left(2.7 - 39 \cdot (0.0318 + 2 \cdot 0.015 \cdot 0.001 \cdot 250)\right) = 9.922 m^2$$

Real flue gas velocity:

$$v_{FG}^{REAL} = \frac{M_{FG}^{REAL}}{S_{DUCT}^{REAL}} = \frac{58.754}{9.922} = 5.92\ m/s$$

The feed water flow velocity in HPeco1:

In the next step is important to determine the mean specific volume of steam which is given by average temperature and pressure in HPeco1. The mean specific volume of steam is then calculated by the softwar XSTEAM.

$$t_{8-9} = \frac{t_8 + t_9}{2} = \frac{145 + 62}{2} = 103.5\ °C$$

$$p_{8-9} = \frac{p_8 + p_9}{2} = \frac{6.4 + 6.5}{2} = 6.45\ MPa$$

$$v_{8-9} = f(t_{8-9}, p_{8-9}) = 0.001043\ m^3/kg$$

$$v_s = \frac{M_s \cdot v_s}{S_s} = \frac{4 \cdot 0.95 \cdot M_{HP} \cdot v_{8-9}}{\pi \cdot d^2 \cdot n_{TU}} = \frac{4 \cdot 0.95 \cdot 5.79 \cdot 0.001043}{\pi \cdot 0.0238^2 \cdot 39} = 0.3306\ m/s$$

Due to slow feed water flow velocity in economizer it is necessary to split heat transfer surfect into three parts as you can see in te *Figure 4. 20*. By this division it is possible to increase feed water flow velocity. In our case three times.

$$v_s = 3 \cdot 0.3306 = 0.992\ m/s$$

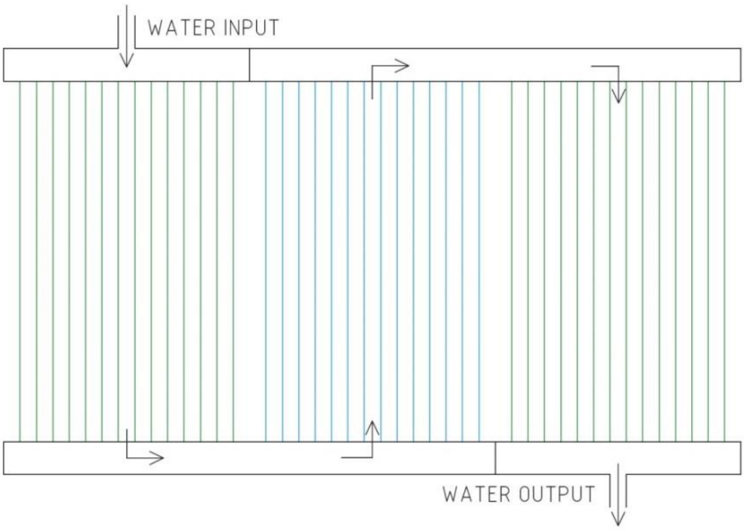

Figure 4. 20 Scheme of divided heat transfer surface in HPeco1

4.9.3 Heat transfer coefficient of high pressure ecomonizer HPeco1

Coefficient of relative pitch φ_σ:

$$\varphi_\sigma = \frac{p_1 - D}{\sqrt{\left(\frac{p_1}{2}\right)^2 + p_2{}^2} - D} = \frac{67.8 - 31.8}{\sqrt{\left(\frac{67.8}{2}\right)^2 + 92^2} - 31.8} = 0.54342$$

The cofficinet of flue gas thermal conductivity λ_{FG} and coefficionet of flue gas kinematic viscosity ν_{FG} is defined by interpolation from [1]. Temperature for interpolation $T_{I-J} = 136.78\ °C$ and for volume fraction of H_2O $x_{H_2O} = 7.8\ \%$ were defined:

$\lambda_{FG} = 0.033953 W/m/K$

$\nu_{FG} = 2.54 \cdot 10^{-5}\ m^2/s$

Coefficient of heat transfer convection α_C:

Coefficient of the number of longitudinal line $c_Z = 0.95$ after consultation with supervisor was determined.

$$\alpha_C = 0.23 \cdot c_Z \cdot \varphi_\sigma{}^{0,2} \cdot \frac{\lambda_{FG}}{p_f} \cdot \left(\frac{D}{p_f}\right)^{-0,54} \cdot \left(\frac{h_f}{p_f}\right)^{-0,14} \cdot \left(\frac{v_{FG} \cdot p_f}{\nu_{FG}}\right)^{0,65}$$

$$\alpha_C = 0.23 \cdot 0.95 \cdot 0.54342^{0,2} \frac{0.033953}{0.004} \left(\frac{0.0318}{0.004}\right)^{-0,54} \left(\frac{0.015}{0.004}\right)^{-0,14} \left(\frac{5.92 \cdot 0.004}{2.54 \cdot 10^{-5}}\right)^{0,65}$$

$\alpha_C = 37.917\ \ W/m^2/K$

Coefficient β :

$$\beta = \sqrt{\frac{2 \cdot \psi_f \cdot \alpha_C}{t_f \cdot \lambda_f \cdot (1 + \varepsilon \cdot \psi_f \cdot \alpha_C)}} = \sqrt{\frac{2 \cdot 0.85 \cdot 37.917}{0.001 \cdot 40 \cdot (1 + 0.0045 \cdot 0.85 \cdot 37.917)}} = 37.515$$

Product of $\beta \cdot h_r$:

$\beta \cdot h_f = 37.515 \cdot 0.015 = 0.563$

Ratio $\frac{D_f}{D}$:

$$\frac{D_f}{D} = \frac{61.8}{31.8} = 1.94$$

Now is possible to determine E from nomogram in literature [1]

$E = 0.88.$

Ratio of fin heat transfer surfaces and the total area of the flue gas side $\frac{S_f}{S}$:

$$\frac{S_f}{S} = \frac{\left(\frac{D_f}{D}\right)^2 - 1}{\left(\frac{D_f}{D}\right)^2 - 1 + 2 \cdot \left(\frac{p_f}{D} - \frac{t_f}{D}\right)} = \frac{\left(\frac{61.8}{31.8}\right)^2 - 1}{\left(\frac{61.8}{31.8}\right)^2 - 1 + 2 \cdot \left(\frac{5}{31.8} - \frac{1}{31.8}\right)} = 0.93637$$

Ratio of not finned tube surface and the total area $\frac{S_h}{S}$:

$$\frac{S_h}{S} = 1 - \frac{S_f}{S} = 1 - 0.93637 = 0.06363$$

Heat transfer coefficient on the flue gas side α_{1r}:

$$\alpha_{1r} = \left[\frac{S_f}{S} \cdot E \cdot \mu + \frac{S_h}{S}\right] \cdot \frac{\psi_f \cdot \alpha_C}{1 + \varepsilon \cdot \psi_f \cdot \alpha_C}$$

$$\alpha_{1r} = [0.93637 \cdot 0.88 \cdot 1 + 0.06363] \cdot \frac{0.85 \cdot 37.917}{1 + 0.0045 \cdot 0.85 \cdot 37.917} = 24.984\ W/m^2/K$$

4.9.4 Overall heat transfer coefficient for HPeco1

Heat transfer coefficient on the steam side α_{2r}:

Heat transfer coefficient on the steam side is in economizer neglected due to heat transfer coefficient on the steam side is many times smaller than heat transfer coefficient on the flue gas side ($\alpha_{2r} \ll \alpha_{1r}$). Only the heat transfer on the flue gas side is taken into account.

Overall coefficient of heat transfer will calculated as $k = \alpha_{1r} = 24.984\ W/m^2/K.$

The surface of one fin S_{1f}:

$$S_{1f} = \frac{2 \cdot \pi \cdot \left({D_f}^2 - D^2\right)}{4} + \pi \cdot D_f \cdot t_f = \frac{2 \cdot \pi \cdot (0.0618^2 - 0.0318^2)}{4} + \pi \cdot 0.0618 \cdot 0.001$$

$S_{1f} = 0.0046\ m^2$

The total outer surface of the one meter long tube with fins S_{1m}:

$$S_{1m} = \pi \cdot D \cdot \left(1 - n_f \cdot t_f\right) + n_f \cdot S_{1f} = \pi \cdot 0.0318 \cdot (1 - 250 \cdot 0.001) + 250 \cdot 0.0046$$

$S_{1m} = 1.226177\ m^2$

The inner surface of the one meter long tube S_{2m}:

$$S_{2m} = \pi \cdot d = \pi \cdot 0.0238 = 0.07477\ m^2$$

4.9.5 The number of longitudinal lines in HPeco1

Temperature diferences for calculation of logarithmic temperature drop:

$$\Delta t_1 = T_I^{REAL} - t_8 = 155.082 - 145 = 10.082\ °C$$

$$\Delta t_2 = T_J - t_9 = 118.48 - 62 = 56.48\ °C$$

Logarithmic temperature drop Δt_{LN}:

$$\Delta t_{LN} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} = \frac{10.082 - 56.48}{ln\left(\frac{10.082}{56.48}\right)} = 26.927\ K$$

External heat transfer surface:

$$S_{EX} = \frac{Q_{HPeco1}}{k \cdot \Delta t_{LN}} = \frac{1922810}{24.984 \cdot 26.927} = 2858.212\ m^2$$

The number of longitudinal lines n_{LI}:

$$n_{LI} = \frac{S_{EX}}{S_{LI}} = \frac{S_{EX}}{h \cdot S_{1m} \cdot n_{TU}} = \frac{2858.212}{8.5 \cdot 1.226177 \cdot 39} = 7.03$$

I choose the number of lines $\boldsymbol{n_{LI} = 7}$.

4.9.6 Scheme of arrangement of tubes in HPeco1

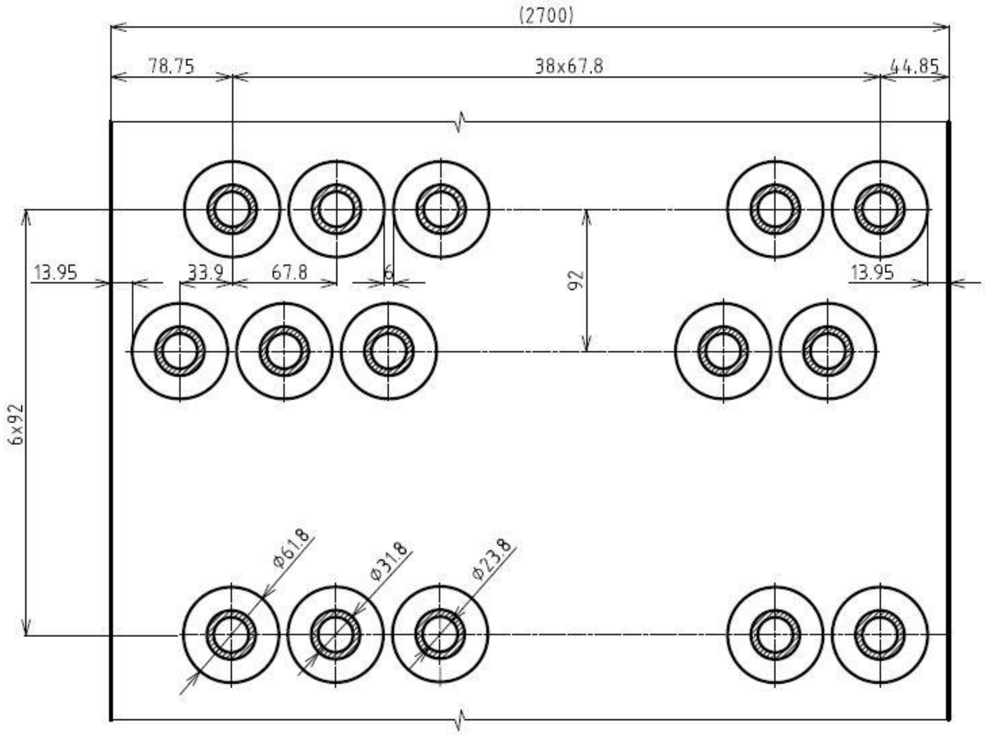

Figure 4. 21 Tube arrangement in HPeco1

4.9.7 The real heat transferred in HPeco1

The real external heat transfer surface S_{EX}^{REAL}:

$$S_{EX}^{REAL} = n_{LI} \cdot n_{TU} \cdot h \cdot S_{1m} = 7 \cdot 39 \cdot 8.5 \cdot 1.226177 = 2845.344\ m^2$$

The real heat transferred in HPeco1:

$$Q_{HPeco1}^{REAL} = k \cdot S_{EX}^{REAL} \cdot \Delta t_{LN} = 24.984 \cdot 2845.344 \cdot 26.927 = 1914.153\ \text{kW}$$

Check of the real heat transferred:

The heat transferred difference may not be greater than 5 %:

$$\Delta Q = \left| \frac{Q_{HPeco1}^{REAL} - Q_{HPeco1}}{Q_{HPeco1}^{REAL}} \right| = \left| \frac{1914.153 - 1922.81}{1914.153} \right| = 0.452\ \%$$

$\Delta Q = 0.452\ \% < 5\ \%$ => Selected number of longitudinal lines is correct.

4.9.8 Real flue gas temperature in point J

The real enthalpy at point J:

$$H_J^{REAL} = H_I^{REAL} - \frac{Q_{HPeco1}^{REAL}}{M_{FG}(1 - L_R)} = 207.48 - \frac{1914.153}{39.15(1 - 0.004146)} = 158.388\ kJ/m^3$$

The real temperature at point J is determined by interpolation from *Table 2. 3*

$$T_J^{REAL} = 100 + (200 - 100) \frac{\left(H_J^{REAL} - H_{FG}^{100}\right)}{\left(H_{FG}^{200} - H_{FG}^{100}\right)} = 100 + 100 \cdot \frac{(158.388 - 133.27)}{(268.01 - 133.27)}$$

$$T_J^{REAL} = 118.645\ °C$$

Check of the real temperature:

The temerature difference may not be greater than 3°C:

$$\Delta T = \left| T_J - T_J^{REAL} \right| = |118.48 - 118.645| = 0.165\ °C$$

$\Delta T = 0.165\ °C < 3\ °C$ => Selected number of longitudinal lines is correct.

4.9.9 List of calculated values in HPeco1

Table 4. 18 Calculated values of HPeco1

Calculated values	Indication	Amount	Unit
Number of tubes in one longitudial line	n_{TU}	39	$[-]$
Real volumetric flow	M_{FG}^{REAL}	58.754	$[m^3/s]$
Real flue gas velocity	v_{FG}^{REAL}	5.92	$[m/s]$
The velocity of steam	v_s	0.992	$[m/s]$
Logarithmic temperature drop	Δt_{LN}	26.927	$[K]$
Number of longitudinal lines	n_{LI}	7	$[-]$
Overall coefficient of heat transfer	k	24.984	$[W/m^2$
Real external heat transfer surface	S_{EX}^{REAL}	2845.344	$[m^2]$
Real heat transferred	Q_{HPeco1}^{REAL}	1914.153	[kW]
Real temperature at point J	T_J^{REAL}	118.645	$[°C]$

5. REAL HEAT TRANSFER TEMPERATURE DIAGRAM

The real heat transfer temperature diagram at the *Figure 5. 1* is shown.

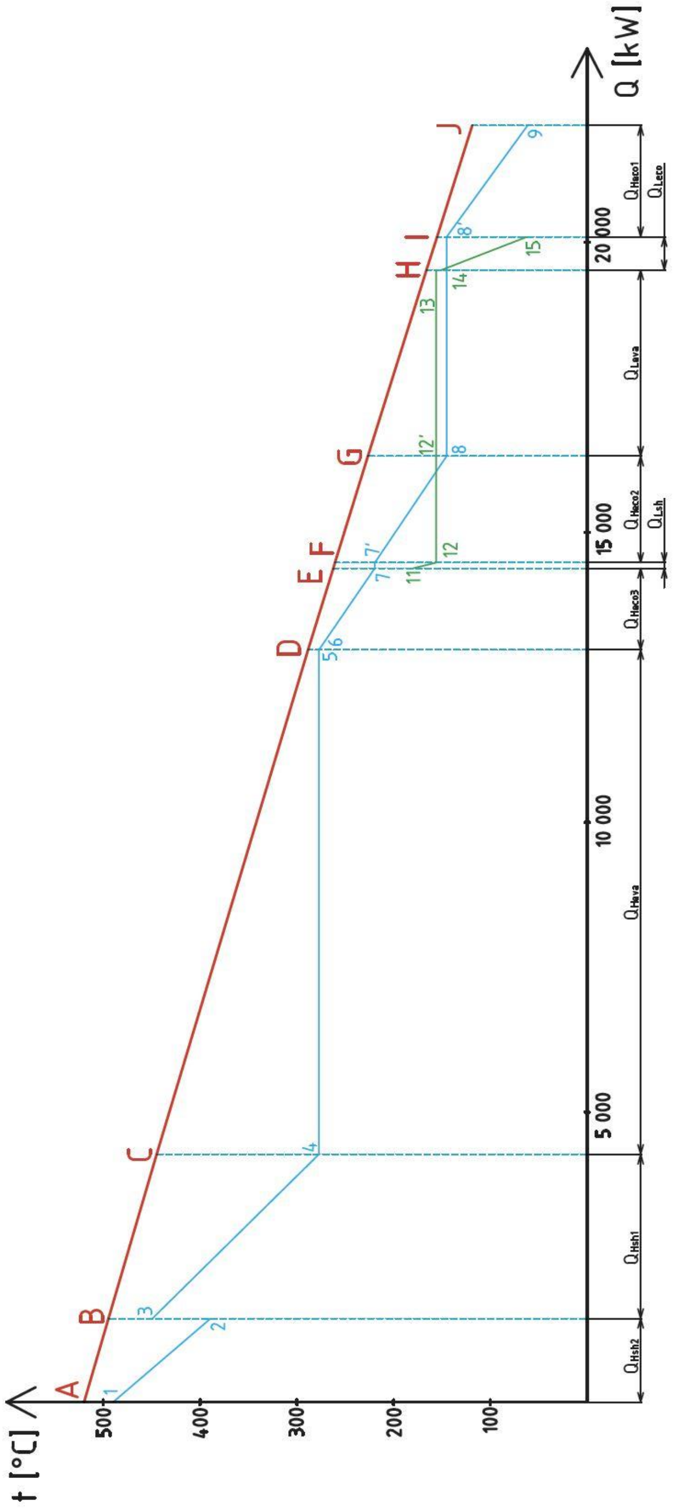

Figure 5. 1 Real heat transfer temperature diagram

6. DESIGN OF THE DRUMS DIMENSIONS AND THEIR CONTROL

The outer drum diameter D_D is chosen according to the boiler mass flow. The dependence of the outer drum diameter and the mass flow rate is given in *Table. 6. 1.* The wall thickness of the drum t_D is selected in the range 40 to 120 mm and the length of the drum is equal to the flue gas duct $l_D = w = 2.7\ m$.

Table 6. 1Dependence of the outer drum diameter ant the mass flow rate

Boilet mass flow rate $M\ [t/h]$	Outer drum diameter
≤ 15	1.2
$15 - 60$	1.4
≥ 60	1.6

Half of the drum volume:

$$V_D = \frac{\pi \cdot {d_D}^2}{4} \cdot \frac{l_D}{2}\ [m^3] \tag{6–1}$$

Drum load:

$$L_D = \frac{M}{V_D}\ [kg/s/m^3] \tag{6–2}$$

M $[kg/s/m^3]$ – boiler mass flow rate

6.1 Design of high pressure drum

In the case of high pressure drum with mass flow rate $M_{HP} = 5.79\ kg/s = 20.84\ t/h$ those values are chosen:

$l_D = w = 2.7\ m$

$D_D = 1.4\ m$

$t_D = 0.04\ m$

$d_D = D_D - 2 \cdot t_D = 1.4 - 2 \cdot 0.04 = 1.32\ m$

Half of the drum volume:

$$V_D = \frac{\pi \cdot {d_D}^2}{4} \cdot \frac{l_D}{2} = \frac{\pi \cdot 1.32^2}{4} \cdot \frac{2.7}{2} = 1.847\ m^3$$

Drum load:

$$L_D = \frac{M_{HP}}{V_D} = \frac{5.79}{1.847} = 3.135\ kg/s/m^3$$

The limit high pressure drum load is given by interpolation from table for the pressure in the high pressure drum $p_{HD} = p_5 = 6.2$ Mpa

$L_D = 6.56\ kg/s/m^3$

Control:

$\mathrm{L_D} = 3.135\ \mathrm{kg/s/m^3} < \mathrm{L_{LD}} = 6.56\ \mathrm{kg/s/m^3}$

Calculated load is less than the limit load and therefore the proposed high-pressure drum suits.

6.2 Design of low pressure drum

In the case of high pressure drum with mass flow rate $M_{LP} = 1.51\ kg/s = 5.44\ t/h$ those values are chosen:

$l_D = w = 2.7\ m$

$D_D = 1.2\ m$

$t_D = 0.04\ m$

$d_D = D_D - 2 \cdot t_D = 1.2 - 2 \cdot 0.04 = 1.12\ m$

Half of the drum volume:

$$V_D = \frac{\pi \cdot {d_D}^2}{4} \cdot \frac{l_D}{2} = \frac{\pi \cdot 1.12^2}{4} \cdot \frac{2.7}{2} = 1.33\ m^3$$

Drum load:

$$L_D = \frac{M_{LP}}{V_D} = \frac{1.51}{1.33} = 1.135\ kg/s/m^3$$

The limit high pressure drum load is given by interpolation from table for the pressure in the high pressure drum $p_D = p_{12} = 0.56\ MPa$

$L_{LD} = 1.87\ kg/s/m^3$

Control:

$\mathrm{L_D} = 1.33\ \mathrm{kg/s/m^3} < \mathrm{L_{LD}} = 1.87\ \mathrm{kg/s/m^3}$

Calculated load is less than the limit load and therefore the proposed high-pressure drum suits.

7. PROPOSAL OF DOWNCOMER PIPING AND CONNECTION PIPING

For the following calculations the literature [2] was used. First of all it is necessary to determine the total cross section of all the tubes in the evaporator S_{ET} and the total cross section of downcomer piping S_{DP}. Furthermore it is necessary to determine at what height h_D the high and low pressure drum will be positioned. Height of the water level in the high and low pressure drum is 3 meters higher than the height of flue gas duct.

The total cross section of all the tubes in the evaporator:

$$S_{ET} = \frac{\pi \cdot d^2}{4} \cdot n_{TU} \cdot n_{LI} \; [m^2] \tag{7–1}$$

Ratio of the total cross section of downcomer piping to all tubes in evaporator:

$$\frac{S_{DP}}{S_{ET}} = 0.06 + 0.016 \cdot p_D + 0.005 \cdot h_D \; [-] \tag{7–2}$$

p_D $[MPa]$ – drum pressure

h_D $[m]$ – drum position

Drum position:

$$h_D = h + 3 = 8.5 + 3 = 11.5[m] \tag{7–3}$$

The calculation of connection piping is detrmine as ratio of the total cross section of all the tubes in the evaporator S_{ET} to the totall cross section of connecting piping S_{CP}.

Ratio of the total cross section of connecting piping to all tubes in evaporator:

$$\frac{S_{CP}}{S_{ET}} = 0.1 + 0.01 \cdot p_D + 0.01 \cdot h_D \; [-] \tag{7–4}$$

7.1 Design of dimensions and number of downcomer piping

7.1.1 High pressure downcomer piping

The total cross section of all the tubes in the evaporator:

$$S_{ET} = \frac{\pi \cdot d^2}{4} \cdot n_{TU} \cdot n_{LI} = \frac{\pi \cdot 0.048^2}{4} \cdot 26 \cdot 12 = 0.5646[m^2]$$

The total cross section of downcomer piping:

$$S_{DP} = S_{ET} \cdot (0.06 + 0.016 \cdot p_D + 0.005 \cdot h_D) = 0.5646 \cdot (0.06 + 0.016 \cdot 6.2 + 0.005 \cdot 11.5)$$

$$S_{DP} = 0.1223[m^2]$$

Chosen dimensions of downcomer piping:

$$D_{DP} = 323.9 \, mm$$

$$t_{DP} = 20mm$$

$$d_{DP} = D_{DP} - 2 \cdot t_{DP} = 323.9 - 2 \cdot 20 = 283.9mm$$

Number of downcomer piping:

$$n_{DP} = \frac{S_{DP}}{\frac{\pi \cdot (d_{DP})^2}{4}} = \frac{0.1223}{\frac{\pi \cdot (0.2839)^2}{4}} = 1.93$$

Is necessary to chose whole number => $\boldsymbol{n_{DP} = 2}$.

7.1.2 Low pressure downcomer piping

<u>The total cross section of all the tubes in the evaporator:</u>

$$S_{ET} = \frac{\pi \cdot d^2}{4} \cdot n_{TU} \cdot n_{LI} = \frac{\pi \cdot 0.0498^2}{4} \cdot 26 \cdot 9 = 0.45579\ [m^2]$$

<u>The total cross section of downcomer piping:</u>

$$S_{DP} = S_{ET} \cdot (0.06 + 0.016 \cdot p_D + 0.005 \cdot h_D) = 0.45579 \cdot (0.06 + 0.016 \cdot 0.56 + 0.005 \cdot 11.5)$$

$$S_{DP} = 0.05764 [m^2]$$

<u>Chosen dimensions of downcomer piping:</u>

$$D_{DP} = 219.1\ mm$$

$$t_{DP} = 14.2\ mm$$

$$d_{DP} = D_{DP} - 2 \cdot t_{DP} = 219.1 - 2 \cdot 14.2 = 190.7\ mm$$

<u>Number of downcomer piping:</u>

$$n_{DP} = \frac{S_{DP}}{\frac{\pi \cdot (d_{DP})^2}{4}} = \frac{0.05764}{\frac{\pi \cdot (0.1907)^2}{4}} = 2.02$$

Is necessary to chose whole number => $\boldsymbol{n_{DP} = 2}$.

7.2 Design of dimensions and number of connecting piping

7.2.1 High pressure connecting piping

<u>The total cross section of connecting piping:</u>

$$S_{CP} = S_{ET} \cdot (0.1 + 0.01 \cdot p_D + 0.01 \cdot h_D) = 0{,}65058 \cdot (0.1 + 0.01 \cdot 6.2 + 0.01 \cdot 11.5)$$

$$S_{CP} = 0.15639 [m^2]$$

<u>Chosen dimensions of connecting piping:</u>

$$D_{CP} = 139.7\ mm$$

$$t_{CP} = 5.4\ mm$$

$$d_{CP} = D_{CP} - 2 \cdot t_{CP} = 139.7 - 2 \cdot 5.4 = 128.9\ mm$$

<u>Number of connecting piping:</u>

$$n_{CP} = \frac{S_{CP}}{\frac{\pi \cdot (d_{CP})^2}{4}} = \frac{0.15639}{\frac{\pi \cdot (0.1289)^2}{4}} = 11.98$$

Is necessary to chose whole number => $\boldsymbol{n_{CP} = 12}$.

7.2.2 Low pressure connecting piping

The total cross section of connecting piping:

$$S_{CP} = S_{ET} \cdot (0.1 + 0.01 \cdot p_D + 0.01 \cdot h_D) = 0575 \cdot (0.1 + 0.01 \cdot 0.56 + 0.01 \cdot 11.5)$$

$$S_{CP} = 0.10055[m^2]$$

Chosen dimensions of connecting piping:

$$D_{CP} = 127\ mm$$

$$t_{CP} = 7.1\ mm$$

$$d_{CP} = D_{CP} - 2 \cdot t_{CP} = 127 - 2 \cdot 7.1 = 112.8\ mm$$

Number of connecting piping:

$$n_{CP} = \frac{S_{CP}}{\frac{\pi \cdot (d_{CP})^2}{4}} = \frac{0.10055}{\frac{\pi \cdot (0.1128)^2}{4}} = 10.06$$

Is necessary to chose whole number => $\boldsymbol{n_{CP} = 10}$

8. MATERIALS

This chapter deals with materials used for pipes and plates, which are selected according to ČSN EN 12 952-3. It also gives a temperature range for given materials. According to the regulation is relative temperature determined according to the equation (8-1). According to the table 6.1.1 in the regulation 12 925-3 are given temperature allowances of the heat transfer surfaces. By heat transfer surfaces, in which predominant heat transfer by convection (superheaters), is determined a temperature allowance $\Delta t_a = 35\ °C$.

Temperature allowance at the other heat exchange surfaces (economizer, evaporators) is determined according to the formula (8-2), which depends on the thickness of the pipe.

As the material for the fin pipes to 800 ° C was chosen X10Cr13.

The materials, which are selected for the heating surfaces, are shown in the tab. 8.1.

$$t_{REL} = T_{EFG} + \Delta t_a\ [°C] \tag{8–1}$$

T_{EFG} $[°C]$ - entrance flue gas temperature of the heat transfer surface

Δt_a $[°C]$ - temperature allowance for the heat transfer surface

$$\Delta t_a = 15 + 2 \cdot t_w\ [°C] \tag{8–2}$$

t_w $[mm]$ – tube wall thickness of the pressure part of the boiler.

Table 8. 1 Chosen materials for heat transfer surfaces

	Entrance flue gas temperature $T_{EFG}[°C]$	Tube wall thickness $t_w\ [mm]$	Temperature allowance $\Delta t_a\ [°C]$	Relative temperature $t_{TP}^{RELATIV}$ $[°C]$	Chosen material
HPsh2	520.00 $°C$	4.5	35 $°C$	555.000 $°C$	13CrMo4-5
HPsh1	494.656$°C$	4.5	35 $°C$	529.656$°C$	16Mo3
HPeva	444.871 $°C$	4.5	24 $°C$	468.871 $°C$	16Mo3
HPeco3	288.314 $°C$	4.5	24$°C$	312.314 $°C$	P265GH
LPsh	262.198 $°C$	3.2	35 $°C$	297.198 $°C$	P235GH
HPeco2	260.474$°C$	4	23 $°C$	283.474 $°C$	P265GH
LPeva	226.452 $°C$	3.6	22.2 $°C$	248.652$°C$	P235GH
LPeco	165.98 $°C$	3.2	21.4 $°C$	187.38 $°C$	P235GH
HPeco1	155.082$°C$	4	23 $°C$	178.082$°C$	P265GH

9. BOILER DRAFT LOSS

In this chapter is calculated the total draft loss Δp_t which is determined by sum of all pressure loss of heat transfer surfaces. The total draft loss have to be less than maximum allowed value of draft loss $\boldsymbol{\Delta p_t^{ALLOWED} = 1500\ Pa}$.

The calculations are performed according to the literature [2].

9.1 Pressure loss of heat transfer surfaces

<u>Pressure loss of smooth tube bundle:</u>

$$\Delta p_L = \Delta p_1 \cdot x_3 \cdot x_4\ [Pa] \tag{9–1}$$

Δp_1 $[Pa]$ – Pressure difference according to the flow flue gas velocity

x_3 $[-]$ – Correction coefficient which depends on the tube pitch

x_4 $[-]$ – Correction coefficient which depends on the flue gas flow rate and the wall temperature

<u>Pressure loss of fin tube bundle:</u>

$$\Delta p_L = \xi \cdot \frac{{v_{FG}}^2}{2} \cdot \rho_{FG}\ [Pa] \tag{9–2}$$

ξ $[-]$ – pressure loss coefficient of fin tube bundle at a perpendicular flow

v_{FG} $[m/s]$ – flue gas velocity in heat transfer surface

ρ_{FG} $[kg/m^3]$ – flue gas density $\rho_{FG} = 1.277228\ kg/m^3$.

<u>Pressure loss coefficient of fin tube bundle:</u>

Coefficients $k_0[-]$, $k_1[-]$, $k_2[-]$, $k_3[-]$ are determined by the Tab 8.6 in the literatur [2] ($k_0 = 2.7$, $k_1 = 0.45$, $k_2 = 0.72$, $k_3 = 0.24$).

$$\xi = k_0 \cdot n_{LI} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot Re^{-k_3}\ [-] \tag{9–3}$$

Re $[-]$ – Reynolds number relative to the mean wall temperature

<u>Reynolds number:</u>

$$Re = \frac{v_{FG} \cdot d_e}{\nu_{FG}}\ [-] \tag{9–4}$$

ν_S $[m^2/s]$ – coefficient of kinematic viscosity of the flue gas relative to the mean wall temperature

d_e $[m]$ – equivalent diameter

<u>Equivalent diameter:</u>

$$d_e = \frac{4 \cdot S_B}{O_B}\ [m] \tag{9–5}$$

S_B $[m^2]$ – flow area of the boiler S_B

O_B $[m]$ – flow area circuit of the boiler

$$d_e = \frac{4 \cdot S_B}{O_B} = \frac{4 \cdot (w \cdot h)}{2 \cdot (w + h)} = \frac{4 \cdot (2.7 \cdot 8.5)}{2 \cdot (2.7 + 8.5)} = 4.098\ m$$

<u>Mean wall temperature:</u>

For gas combustion is determined temperature increase $\Delta t_z = 25\ °C$ by literatur [1].

$$t_{MWT} = t_{MST} + \Delta t_Z\ [°C] \quad (9\text{–}6)$$

t_{MST} $[°C]$ – mean steam temperature for heat transfer surface

Values for calculation of pressure loss are given in *Table 9.1*.

Table 9. 1 Necessary values for determination of pressure loss in heat transfer surfaces

	Mean steam temperature t_{MST} $[°C]$	Coefficient of flue gas kinematic viscosity ν_S $[m^2/s]$	Reynolds number Re relative to t_{MST}
HPsh2	464.765	7.003E-05	5.952E+05
HPsh1	388.785	5.825E-05	6.309E+05
HPeva	302.730	4.584E-05	9.333E+05
HPeco3	271.365	4.170E-05	6.075E+05
LPsh	193.075	3.145E-05	6.064E+05
HPeco2	207.500	3.334E-05	9.225E+05
LPeva	181.150	3.030E-05	1.036E+06
LPeco	131.575	2.487E-05	8.360E+05
HPeco1	128.500	2.453E-05	9.486E+05

9.1.1 Pressure loss in HPsh2

<u>Pressure loss coefficient of fin tube bundle:</u>

$$\xi = k_0 \cdot n_{TU} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot {Re_S}^{-k_3} = 2.7 \cdot 2 \cdot \left(\frac{15}{31.8}\right)^{0.45} \cdot \left(\frac{4.65}{31.8}\right)^{-0.72} \cdot (5.952 \cdot 10^5)^{-0.24}$$

$$\xi = 0.948$$

<u>Pressure loss in HPsh2</u>

$$\Delta p_{HPsh2} = \xi \cdot \frac{{v_{FG}}^2}{2} \cdot \rho_{FG} = 0.948 \cdot \frac{9.95^2}{2} \cdot 1.277228 = 59.904\ Pa$$

9.1.2 Pressure loss in HPsh1

Pressure loss coefficient of fin tube bundle:

$$\xi = k_0 \cdot n_{TU} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot {Re_S}^{-k_3} = 2.7 \cdot 2 \cdot \left(\frac{15}{31.8}\right)^{0.45} \cdot \left(\frac{5}{31.8}\right)^{-0.72} \cdot (6.309 \cdot 10^5)^{-0.24}$$

$\xi = 1.183$

Pressure loss in HPsh1

$$\Delta p_{HPsh1} = \xi \cdot \frac{{v_{FG}}^2}{2} \cdot \rho_{FG} = 1.183 \cdot \frac{9.352^2}{2} \cdot 1.277228 = 66.085\ Pa$$

9.1.3 Pressure loss in HPeva

Pressure loss coefficient of fin tube bundle:

$$\xi = k_0 \cdot n_{TU} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot {Re_S}^{-k_3} = 2.7 \cdot 2 \cdot \left(\frac{19}{57}\right)^{0.45} \cdot \left(\frac{4.35}{57}\right)^{-0.72} \cdot (9.333 \cdot 10^5)^{-0.24}$$

$\xi = 4.653$

Pressure loss in HPeva

$$\Delta p_{HPeva} = \xi \cdot \frac{{v_{FG}}^2}{2} \cdot \rho_{FG} = 4.653 \cdot \frac{10.886^2}{2} \cdot 1.277228 = 352.119\ Pa$$

9.1.4 Pressure loss in HPeco3

Pressure loss coefficient of fin tube bundle:

$$\xi = k_0 \cdot n_{TU} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot {Re_S}^{-k_3} = 2.7 \cdot 2 \cdot \left(\frac{15}{31.8}\right)^{0.45} \cdot \left(\frac{5}{31.8}\right)^{-0.72} \cdot (6.075 \cdot 10^5)^{-0.24}$$

$\xi = 1.791$

Pressure loss in HPeco3

$$\Delta p_{HPeco3} = \xi \cdot \frac{{v_{FG}}^2}{2} \cdot \rho_{FG} = 1.791 \cdot \frac{6.902^2}{2} \cdot 1.277228 = 54.491\ Pa$$

9.1.5 Pressure loss in LPsh

Correction coefficient which depends on the tube pitch is determined as ratio of lateral pitch $p_1 = 131\ mm$ to outer diameter $D = 48.3\ mm$ and ratio of longitudinal pitch $p_2 = 117\ mm$ to outer diameter $D = 48.3\ mm$

$$\frac{p_1}{D} = \frac{131}{48.3} = 2.712$$

$$\frac{p_2}{D} = \frac{117}{48.3} = 2.422$$

For this values is possible to deduct the coefficient value of the chart 8.12 in the literature [2]:

$x_3 = 0.88$

Correction coefficient which depends on the temperature of the flue gas flow rate $T_{E-F} = 261.329\ °C$ and the wall temperature $T_{FO} = 193.075$ is determine of the chart 8.12 in the literature [2]:

$x_4 = 1.57$

Pressure difference according to the flow flue gas velocity $v_{FG}^{REAL} = 5.197\ m/s$ from the chart 8.12 in the literature [2] is determine:

$\Delta p_1 = 1.32\ Pa$

<u>Pressure loss of smooth tube bundle:</u>

$$\Delta p_{LPsh} = \Delta p_1 \cdot x_3 \cdot x_4 = 1.32 \cdot 0.88 \cdot 1.57 = 1.824 Pa$$

9.1.6 Pressure loss in HPeco2

<u>Pressure loss coefficient of fin tube bundle:</u>

$$\xi = k_0 \cdot n_{TU} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot Re_S{}^{-k_3} = 2.7 \cdot 2 \cdot \left(\frac{13}{31.8}\right)^{0.45} \cdot \left(\frac{4.35}{31.8}\right)^{-0.72} \cdot (9.225 \cdot 10^5)^{-0.24}$$

$\xi = 0.840$

<u>Pressure loss in HPeco2:</u>

$$\Delta p_{HPeco2} = \xi \cdot \frac{v_{FG}{}^2}{2} \cdot \rho_{FG} = 0.840 \cdot \frac{7.827^2}{2} \cdot 1.277228 = 32.858\ Pa$$

9.1.7 Pressure loss in LPeva

<u>Pressure loss coefficient of fin tube bundle:</u>

$$\xi = k_0 \cdot n_{TU} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot Re_S{}^{-k_3} = 2.7 \cdot 2 \cdot \left(\frac{19}{57}\right)^{0.45} \cdot \left(\frac{5}{57}\right)^{-0.72} \cdot (1.036 \cdot 10^6)^{-0.24}$$

$\xi = 3.403$

<u>Pressure loss in LPeva:</u>

$$\Delta p_{LPeva} = \xi \cdot \frac{v_{FG}{}^2}{2} \cdot \rho_{FG} = 3.403 \cdot \frac{7.99^2}{2} \cdot 1.277228 = 132.723\ Pa$$

9.1.8 Pressure loss in LPeco

<u>Pressure loss coefficient of fin tube bundle:</u>

$$\xi = k_0 \cdot n_{TU} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot Re_S{}^{-k_3} = 2.7 \cdot 2 \cdot \left(\frac{15}{31.8}\right)^{0.45} \cdot \left(\frac{6.25}{31.8}\right)^{-0.72} \cdot (8.360 \cdot 10^5)^{-0.24}$$

$\xi = 0.471$

Pressure loss in LPeco:

$$\Delta p_{LPeco} = \xi \cdot \frac{{v_{FG}}^2}{2} \cdot \rho_{FG} = 0.471 \cdot \frac{5.29^2}{2} \cdot 1.277228 = 8.417\ Pa$$

9.1.9 Pressure loss in HPeco1

Pressure loss coefficient of fin tube bundle:

$$\xi = k_0 \cdot n_{TU} \cdot \left(\frac{h_f}{D}\right)^{k_1} \cdot \left(\frac{p_f}{D}\right)^{-k_2} \cdot {Re_S}^{-k_3} = 2.7 \cdot 2 \cdot \left(\frac{15}{31.8}\right)^{0.45} \cdot \left(\frac{4}{31.8}\right)^{-0.72} \cdot (9.486 \cdot 10^5)^{-0.24}$$

$$\xi = 2.205$$

Pressure loss in HPeco1:

$$\Delta p_{HPeco1} = \xi \cdot \frac{{v_{FG}}^2}{2} \cdot \rho_{FG} = 2.205 \cdot \frac{5.92^2}{2} \cdot 1.277228 = 49.373\ Pa$$

9.2 Pressure loss in stack

HRSG stack loss is determine as sum of local loss Δp_{LL}, frictional loss Δp_{FL} buoyancy loss Δp_{BL} and loss caused by silencer in stack $\Delta p_{SL} = 250\ Pa$.

Pressure loss in stack:

$$\Delta p_{ST} = \Delta p_{FL} + \Delta p_{LL} - \Delta p_{BL} + \Delta p_{SL}\ [Pa], \qquad (9\text{–}7)$$

Δp_{FL} $[Pa]$ – frictional loss

Δp_{LL} $[Pa]$ – local loss

Δp_{BL} $[Pa]$ – buoyancy loss

Δp_{SL} $[Pa]$ – silencer loss

Frictional loss in stack:

$$\Delta p_{FL} = \lambda_{STACK} \cdot \frac{H_{STACK}}{d_{STACK}} \cdot \frac{{v_{STACK}}^2}{2} \cdot \rho_{FG}\ [Pa] \qquad (9\text{–}8)$$

$\lambda_{STACK}\ [-]$ – frictional coefficient in stack for brick stack $\lambda_{STACK} = 0.04\ W/m/K$

$H_{STACK}[m]$ – heigh of stack was choosen $H_{STACK} = 25\ m$

$d_{STACK}\ [m]$ – stack diameter

$v_{STACK}\ [m/s]$ – flue gas velocity in stack

Local loss in stack:

$$\Delta p_{LL} = (\xi_{IN} + \xi_{OUT}) \cdot \frac{{v_{STACK}}^2}{2} \cdot \rho_{FG}\ [Pa], \qquad (9\text{–}9)$$

$\xi_{IN}\ [-]$, $\xi_{OUT}\ [-]$ – loss of local input and the output from the stack, $\xi_{IN} = \xi_{OUT} = 1$.

Buoyancy loss:

$$\Delta p_{BL} = H_{STACK} \cdot \left(\rho_A - \rho_{FG} \cdot \frac{273.15}{273.15 + \Delta t_{MFGT}}\right) \cdot g \; [Pa], \qquad (9–10)$$

ρ_A $[kg/m^3]$ – density of atmospheric air, elected $\rho_A = 1.279\ kg/m^3$

$\Delta t_{MFGT} [°C]$ – mean flue gas temperature, elected $\Delta t_{MFGT} = T_J^{REAL} = 118.645\ °C$

g $[m^2/s]$ – gravitational acceleration, $g = 9.81\ m^2/s$.

9.3 Stack loss

9.3.1 Frictional loss is stack

Frictional loss is calculated from the formula (9-8). It is necessary to determine real flue gas flow in stack M_{STACK}^{REAL} and mean flue gas velocity in stack v_{STACK}. Furthermore is elected the inner diameter of the stack $d_{STACK} = 3.5m$.

$$M_{STACK}^{REAL} = M_{FG} \cdot \frac{T_J^{REAL} + 273.15}{273.15} = 39.15 \cdot \frac{118.645 + 273.15}{273.15} = 56.155\ m^3/s.$$

Mean flue gas velocity in stack:

$$v_{STACK} = \frac{4 \cdot M_{STACK}^{REAL}}{\pi \cdot d_{STACK}{}^2} = \frac{4 \cdot 56.155}{\pi \cdot 3.5^2} = 5.84\ m/s$$

$$\Delta p_{FL} = \lambda_{STACK} \cdot \frac{H_{STACK}}{d_{STACK}} \cdot \frac{v_{STACK}{}^2}{2} \cdot \rho_{FG} = 0.04 \cdot \frac{25}{3.5} \cdot \frac{5.86^2}{2} \cdot 1.277228 = 6.27\ Pa$$

9.3.2 Local loss in stack

$$\Delta p_{LL} = (\xi_{IN} + \xi_{OUT}) \cdot \frac{v_{STACK}{}^2}{2} \cdot \rho_{FG} = (1+1) \cdot \frac{5.84^2}{2} \cdot 1.277228 = 43.56\ Pa$$

9.3.3 Buoyancy loss

$$\Delta p_{BL} = H_{STACK} \cdot \left(\rho_A - \rho_{FG} \cdot \frac{273.15}{273.15 + \Delta t_{MFGT}}\right) \cdot g$$

$$\Delta p_{BL} = 25 \cdot \left(1.279 - 1.277228 \cdot \frac{273.15}{273.15 + 118.645}\right) \cdot 9.81$$

$$\Delta p_{BL} = 95.29\ Pa$$

9.3.4 Pressure loss in stack

$$\Delta p_{ST} = \Delta p_{FL} + \Delta p_{LL} - \Delta p_{BL} + \Delta p_{SL} = 6.72 + 43.56 - 95.29 + 250 = 204.99\ Pa$$

9.4 Total stack loss

Total stack loss Δp_{STACK} is determine as sum of all pressure loss in heat transfer surfaces and a total pressure loss in stack.

$$\Delta p_{STACK} = \Delta p_{Hsh2} + \Delta p_{Hsh1} + \Delta p_{Heva} + \Delta p_{Heco3} + \Delta p_{Lsh} + \Delta p_{Heco2} + \Delta p_{Leva} + +\Delta p_{Leco} + \Delta p_{Heco1} + \Delta p_{STACK}$$

$$\Delta p_{STACK} = 59.904 + 66.085 + 352.119 + 54.491 + 1.824 + 32.858 + 138.723 + 8.417 + 49.373 + 204.99$$

$$\Delta p_{STACK} = \mathbf{968.783\ Pa}$$

Total stack loss Δp_{STACK} is smaller than the maximum allowable loss

$$\Delta p_{STACK} = 968.783\ Pa < \Delta \boldsymbol{p}_{\boldsymbol{STACK}}^{\boldsymbol{ALLOWED}} = 1500\ Pa$$

CONCLUSION

This thesis has been prepared with the intention to make a proposal of a horizontal dual-pressure heat recovery steam generator for gas turbine according to the specified and then selected parameters. The thesis is focused, in a certain level, on the design of the heating surfaces.

In the first part of the thesis, the thermal calculations were carried, besides other things, from which emerged the core values of the relative radiation loss $L_R = 0.4146\%$, high pressure performance of the boiler $M_{HP} = 5.79\ kg/s$ and low pressure performance of the boiler $M_{LP} = 1.51\ kg/s$.

With regard to the given parameters a preliminary proposal of the temperature heat transfer diagram was subsequently created and at the same time the arrangement of the individual heating surfaces has been selected. The 5% injection of the feedwater was chosen to regulate the outlet temperature of the high-pressure steam, which was placed between the HPsh2 and HPsh1. To make a better use of the temperature gradient, the high-pressure economizer and superheater were divided into 3 and 2 parts. The total number of the heat transfer surfaces was 9.

In the following part, the dimensions of the flue gas duct were designed so, that the heat transfer surface HPsh2 was used for the dimensioning of flue gas duct (width $\mathrm{w} = 2.7\ \mathrm{m}$, height $\mathrm{h} = 8.5\ \mathrm{m}$). Based on these dimensions, the other heat transfer surfaces were determined and by the calculation flue gas temperature at the boiler outlet was also determined, with a value of $T_J^{REAL} = 118.645\ °C$ °

After a series of introductory calculations, the real heat transfer temperature diagram was designed as well as the materials, from which the boiler will be designed. Downcomer and connection piping was designed with a diameter that was at the high-pressure evaporator $D_{DP} = 323.9\ mm,\ D_{CP} = 139.7\ mm,$ and at the low-pressure evaporator $D_{DP} = 219.1\ mm,\ D_{CP} = 127\ mm$. The drums were also designed; a high-pressure drum having an outer diameter $D_D = 1.4\ m$ and a low pressure drum with a diameter $D_D = 1.2\ m$, and their load was inspected. For high- and low-pressure drum, the load was calculated to be lower than the normal load. The drums were therefore identified as compliant.

The final part of the thesis handles the calculations of the boiler draft loss from the pressure loss of the heating surfaces and from the losses incurred in the stack. The maximum allowed boiler draft loss, that had to be followed, was $\Delta p_t^{ALLOWED} = 1500\ Pa$. Further a total control of boiler draft loss was performed, the value of which should not exceed $\Delta p_t^{ALLOWED}$. From the executed inspection, the value $\Delta p_{STACK} = 968.783\ Pa$ was gained, which determined, that the calculation was correct. A part of this thesis is a drawing, which, according to the set or calculated values, demonstrates a possible way of the realization of the boiler (see Attachment no. 1).

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α_{1r}	[W/m^2/K]	heat transfer coefficient on the flue gas side
α_{2r}	[W/m^2/K]	heat transfer coefficient on the steam side
α_C	[W/m^2/K]	coefficient of heat transfer convection
α_R	$[W/m^2/K]$	radiation heat transfer coefficient
β	[-]	coefficient to determine the cefficient of fins efficiency E
μ	[-]	coefficient of fin expansion
ρ_X	[kg/m^3]	ensity of flue gas components
σ_1	[-]	relative lateral spacing
σ_2'	[-]	relative diagonal pitch
φ_σ	[-]	coefficient of relative pitch
ψ_f	[-]	coefficient of unevenness distribution α_C across the surface of the fin
ξ	[-]	utilization factor
ε	[-]	coefficient of fins fouling
λ_S	$[W/m/K]$	coefficient of steam thermal conductivity
λ_{FG}	$[W/m/K]$	cofficinet of flue gas thermal conductivity
λ_f	[W/m/K]	coefficient of fin thermal conductivity
t_{HP}	[°C]	high pressure circuit output temperature
t_{LP}	[°C]	low pressure circuit output temperature
p_{HP}	[MPa]	high pressure circuit output pressure
p_{LP}	[MPa]	low pressure circuit output pressure
m	[kg/s]	flue gas mass flow
t_{FG}	[°C]	intlet flue gas temperature
x_X	[%]	volumetric flue gas composition
ΔQ	[%]	heat transferred difference
Δpsh	[Pa]	pressure loss in the high pressure superheaters
$\Delta peco$	[Pa]	pressure loss in the high pressure economizers
Δh	[kJ/kg]	enthalpy drop in high pressure superheater
Δt_{pp}	[°C]	pinchpoint
ΔT	[°C]	temerature difference
Δt_{LN}	[K]	logarithmic temperature drop
ΔQ_{HP}	[kW]	heat difference trasfered to steam in high pressure circle
ΔQ_{LP}	[kW]	heat difference trasfered to steam in low pressure circle
Δ_{LQ}	[%]	checking of the heat transferred in low pressure circle
Δ_{HQ}	[%]	checking of the heat transferred in high pressure circle
Δt_{LN}	[K]	logarithmic temperature drop
Δt_N	[°C]	approache point
M_{FG}	[m^3/s]	flue gases volume flow
M_{HP}	[kg/s]	high pressure performance ot the boiler

M_{LP}	[kg/s]	low pressure performance of the boiler
M_s	[kg/s]	steam mass flow in the heating surface
v_{sp}	$[m^3/kg]$	mean specific volume of steam in the heating surface
S_s	$[m^2]$	total cross section of tubes
n_{TU}	[-]	amount of tubes in one longitudinal line
v_s^{REAL}	[m/s]	real flow velocity of steam
M_{FG}^{REAL}	$[m^3/s]$	real volumetric flow takes into account the flue gas temperature
h_f	[m]	fins height
t_f	[m]	fins thickness
S_{DUCT}^{REAL}	[m^2]	real cross section of the the flue gas duct
v_{FG}^{REAL}	[m/s]	real flue gas velocity
c_m	[-]	correction annulus coefficient
c_l	[-]	correction length coefficient
c_t	[-]	correcting coefficient dependent on current and wall temperature
c_Z	[-]	coefficient of the number of longitudinal line
E	[-]	coefficient of fins efficiency
ν_{FG}	$[m^2/s]$	coefficionet of flue gas kinematic viscosity
ν_S	$[m^2/s]$	coefficionet of steam kinematic viscosity
k	$[W/m^2/K]$	overall coefficient of heat transfer
S_{EX}	$[m^2]$	external heat transfer surface
S_{LI}	$[m^2]$	heat transfer surface in one longitudinal lines
n_{LI}	[-]	number of longitudinal lines
d	[m]	inner tube diameter
d_D	[m]	inner drum diameter
d_e	[m]	equivalent diameter
S_{EX}^{REAL}	$[m^2]$	real external heat transfer surface
D	[m]	outer tube diameter
D_D	[m]	outer drum diametr
D_f	[m]	outer fin diameter
Q_X^{REAL}	[kW]	real heat transferred in heat transfer surface
H_X^{REAL}	$[kJ/m^3]$	real enthaply at point of heat transfer temperature diagram
T_X^{REAL}	[°C]	real temperature at point of heat transfer temperature diagram
H_{FG}^t	[kJ/m^3]	flue gas enthalpy for scpecific temperature
h_X	[kJ/kg]	enthalphy at point of heat transfer temperature diagram
H_X	[kJ/m^3]	flue gas enthalpy in poit of the heat transfer temperature diagram
k_0, k_1, k_2, k_3	[-]	coefficient depending on the tube arrangement
h	[m]	height of the flue gas duct
T_{FO}	[K]	fouled wall surface temperature
a_{tr}	[-]	emisivity of tri-atomic gases
a_c	[-]	cofficinet of flue gas thermal conductivity

k_y	$[m/MPa]$	coefficient of radiant absorption due to tri-atomic gases
s	[mm]	effective thickness of the radiation layer
p_{par}	[MPa]	partial pressure of triatomic flue gases
ψ	[-]	coefficient of thermal efficiency
C_{gas}	[-]	constant used for boilers for natural gas and liquid fuels
n_f	[-]	fins per meter
p_D	[MPa]	drum pressure
Pr	[-]	prandtel number
p_X	[MPa]	pressure of point of heat transfer temperature diagram on the media side
$p_{X\text{-}Y}$	[MPa]	mean pressure value
Re	[-]	reynolds number relative to the mean wall temperature
Q_A	[MW]	maximal heat output
Q_{RS}	[MW]	radiation loss
Q_X	[kW]	heat transfered in heat transfer surface
$Q_{X\text{-}Y}$	[kW]	heat transferred between specific points of heat transfer temperature diagram
p_1	[m]	lateral pitch
p_2	[m]	longitudinal pitch
S_{DUCT}	[m^2]	cross section of the flue gas duct
M	$[kg/s/m^3]$	boiler mass flow rate
S_{ET}	[m^2]	total cross section of all the tubes in the evaporator
S_{DP}	[m^2]	total cross section of downcomer piping
D_{DP}	[mm]	outer diameter of downcomer piping
d_{DP}	[mm]	inner diameter of downcomer piping
t_{DP}	[mm]	thickness of downcomer piping
n_{DP}	[-]	number of downcomer piping
S_{CP}	[m^2]	total cross section of connecting piping
D_{CP}	[mm]	outer diameter of connecting piping
d_{CP}	[mm]	inner diameter of connecting piping
t_{CP}	[mm]	thickness of connecting piping
n_{CP}	[-]	number of connecting piping
S_{1m}	[m^2]	total outer surface of the one meter long tube
S_{2m}	[m^2]	inner surface of the one meter long tube
S_{1f}	[m^2]	surface of one fin
P_f	[m]	fin pitch
S_f/S	[-]	ratio of fin heat transfer surfaces and the total area of the flue gas side
S_h/S	[-]	ratio of not finned tube surface and the total area
w	[m]	width of flue gas duct
t	[m]	thickness of tube wall

t_D	[m]	thickess of drum wall
t_{FW}	[°C]	feed water temperature
t_X	[°C]	temperature at point of the heat transfer temperature diagram on media side
$t_{X\text{-}Y}, T_{X\text{-}Y}$	[°C]	mean value temperature
T_X	[°C]	temperature at point of the heat transfer temperature diagram on the flue gas side
V_D	[m^3]	half of the drum volume
v_S	[m/s]	flow velocity of steam
v_{FG}	[m/s]	flue gas velocity
x_X	[%]	objemový podíl určité sloučeniny ve spalinách
L_D	[kg/s/m^3]	drum load
L_{LD}	[kg/s/m^3]	limit load of drum
L_R	[-]	relative Radiation loss
T_{EFG}	$[°C]$	entrance flue gas temperature of the heat transfer surface
Δt_a	$[°C]$	temperature allowance for the heat transfer surface
t_w	$[mm]$	tube wall thickness of the pressure part of the boiler
x_3	$[-]$	correction coefficient which depends on the tube pitch
x_4	$[-]$	correction coefficient which depends on the flue gas flow rate
S_B	$[m^2]$	flow area of the boiler
O_B	$[m]$	flow area circuit of the boiler
t_{MWT}	$[°C]$	mean wall temperature
t_{MST}	$[°C]$	mean steam temperature for heat transfer surface
Δp_X	$[Pa]$	pressure loss in heat transfer surfaces
Δp_{ST}	$[Pa]$	stack loss
Δp_{FL}	$[Pa]$	frictional loss
Δp_{LL}	$[Pa]$	local loss
Δp_{BL}	$[Pa]$	buoyancy loss
Δp_{SL}	$[Pa]$	silencer loss
λ_{STACK}	$[-]$	frictional coefficient in stack for brick stack
H_{STACK}	$[m]$	heigh of stack
d_{STACK}	$[m]$	stack diameter
v_{STACK}	$[m/s]$	stack diameter
ρ_A	$[kg/m^3]$	density of atmospheric air
g	$[m^2/s]$	gravitational acceleration
$\Delta p_{STACK}^{ALLOWED}$	$[Pa]$	allowed pressure loss in stack

SHORTCUTS

1-9 ....points of heat transfer temperature diagramu – coolant of high-pressure part of the boiler

11-15 ....points of heat transfer temperature diagramu – coolant of low-pressure part of the boiler

A-J......................points of heat transfer temperature diagramu on the flue gas side

LPeco ....low-pressure economizer

LPsh ....low-pressure superheater

LPeva ....low-pressure evaporator

HPeco1...............high-pressure economizer 1

HPeco 2.............high-pressure economizer 2

HPeco 3.............high-pressure economizer 3

HPsh1.................high-pressure superheater 1

HPsh2.................high-pressure superheater 2

HPeva.................high-pressure evaporator

HRSG.................heat recovery steam generator

LIST OF ATTACHMENT

Attachment no. 1	Drawing of dual-pressure heat recovery steam generator-A1-DP-2016/1
Attachment no. 2	CD