



# BRNO UNIVERSITY OF TECHNOLOGY

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## ENERGY INSTITUTE

ENERGETICKÝ ÚSTAV

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

NAVRH DVOU TLAKÉHO HORIZONTÁLNÍHO KOTLE NA ODPADNÍ TEPLU (HRSG) ZA PLYNOVOU TURBÍNOU NA ZEMNÍ PLYN

## BACHELOR'S THESIS

BAKALÁŘSKÁ PRÁCE

### AUTHOR

AUTOR PRÁCE

Bc. Jan Pauliny

### SUPERVISOR

VEDOUCÍ PRÁCE

Ing. Martin Lisý, Ph.D.

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# Master's Thesis Assignment

Institut: Energy Institute  
Student: **Bc. Jan Pauliny**  
Degree program: Mechanical Engineering  
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Supervisor: **Ing. Martin Lisý, Ph.D.**  
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As provided for by the Act No. 111/98 Coll. on higher education institutions and the BUT Study and Examination Regulations, the director of the Institute hereby assigns the following topic of Master's Thesis:

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

### **Brief description:**

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

The mass flow rate of the flue gas is 30kg/s, the flue gas temperature is 490°C

Superheated steam parameters: high pressure circuit 4.6MPa; 450°C

Low pressure circuit 0.46MPa; 170°C

Feed water temperature 53°C

Volumetric composition of the flue gas: O<sub>2</sub> = 13.9%; Ar = 0.9%; N<sub>2</sub> = 73%; CO<sub>2</sub> = 4.4%; H<sub>2</sub>O = 7.8%

### **Master's Thesis goals:**

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

### **Bibliography:**

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Dlouhý, T.: Výpočty kotlů a spalínových výměníků, ČVUT v Praze, 2007, ISBN 978-80-01-03757-7

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

doc. Ing. Jaroslav Katolický, Ph.D.  
Dean

## ABSTRACT

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

## KEYWORDS

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

## ABSTRAKT

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

## KLÍČOVÁ SLOVA

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



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**DECLARATION OF AUTHENTICITY**

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

Brno                      27. May 2016

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Jan Pauliny



## TABLE OF CONTENTS

|   |    |
|---|----|
| INTRODUCTION .....  | 17 |
| 1 GENERAL DESCRIPTION.....  | 19 |
| 1.1.1 Given Boiler Parameters.....  | 20 |
| 2 THERMAL CALCULATIONS OF THE BOILER.....   | 21 |
| 2.1 Preliminary Temperature Profile Graph of the Boiler .....   | 21 |
| 2.2 Selected Parameters Values .....  | 21 |
| 2.3 Water/Steam Parameters .....  | 22 |
| 2.3.1 High pressure water/steam parameters .....  | 22 |
| 2.3.2 Low pressure water/steam parameters.....  | 24 |
| 2.4 Flue Gas Parameters .....   | 25 |
| 2.4.1 Flue gas density .....  | 25 |
| 2.4.2 Flue gas volumetric flow rate, under normal conditions .....                                    | 25 |
| 2.4.3 Flue gas enthalpy calculations at various points .....  | 26 |
| 2.5 Rate Heat Loss from the Boiler Due to Radiation, Calculation.....                                 | 27 |
| 2.6 Steam Production Rates and Heat Transfer rates of Each Individual Heat Exchanger                  | 28 |
| 2.6.1 High pressure steam production rate .....   | 28 |
| 2.6.2 Heat transfer rates of each individual heat exchanger in the high pressure circuit              | 29 |
| of the boiler.....  | 29 |
| 2.6.3 Low pressure steam production rate .....  | 30 |
| 2.6.4 Heat transfer rates of each individual heat exchanger in the low pressure circuit               | 32 |
| of the boiler.....  | 32 |
| 2.7 Calculating Flue Gas Parameters at Points B, C, F, H, I, and J .....                              | 32 |
| 2.8 An Overview of the Calculated Values and the HRSG temperature profile .....                       | 34 |
| 3 FLUE GAS DUCT DESIGN.....   | 39 |
| 3.1 Finned Tube Geometry of the High Pressure Superheater (SH <sub>2HP</sub> ).....                   | 39 |
| 3.1.1 Determining the number of tubes in each lateral row of the high pressure                        | 40 |
| superheater (SH <sub>2HP</sub> ) .....  | 40 |
| 3.1.2 Volumetric flow rate of flue gas passing through the high pressure superheater                  | 41 |
| (SH <sub>2HP</sub> ) .....  | 41 |
| 3.2 Flue Gas Duct Dimensions .....  | 41 |
| 3.2.1 Overview of the calculated values.....  | 42 |
| 4 DESIGN AND SIZING OF HEAT TRANSFER SURFACES .....   | 43 |
| 4.1 Design of the High Pressure Superheater SH <sub>2HP</sub> .....                                   | 50 |
| 4.1.1 Geometry of the high pressure superheater SH <sub>2HP</sub> .....                               | 51 |
| 4.1.2 Reduced heat transfer coefficient outside the tubes of SH <sub>2HP</sub> , through the flue gas | 51 |
| .....   | 51 |

|        |  |    |
|--------|--|----|
| 4.1.3  | Reduced heat transfer coefficient inside the tubes of SH <sub>2HP</sub> , through the Steam      | 54 |
| 4.1.4  | Overall heat transfer coefficient for SH <sub>2HP</sub> .....                                    | 54 |
| 4.1.5  | Logarithmic mean temperature difference across SH <sub>2HP</sub> .....                           | 55 |
| 4.1.6  | The number of lateral rows of tubes required for SH <sub>2HP</sub> .....                         | 55 |
| 4.1.7  | Calculating the actual heat transfer rate of SH <sub>2HP</sub> .....                             | 56 |
| 4.1.8  | Determining the actual parameters of the flue gas exiting SH <sub>2HP</sub> .....                | 56 |
| 4.2    | Design of the High Pressure Superheater SH <sub>1HP</sub> .....                                  | 56 |
| 4.2.1  | Geometry of the high pressure superheater SH <sub>1HP</sub> .....                                | 57 |
| 4.2.2  | Verification of the speed of steam passing through superheater SH <sub>1HP</sub> .....           | 59 |
| 4.2.3  | Verification of the speed of flue gas passing through superheater SH <sub>1HP</sub> .....        | 60 |
| 4.2.4  | Reduced heat transfer coefficient outside the tubes of SH <sub>1HP</sub> , through the flue gas  | 60 |
| 4.2.5  | Reduced heat transfer coefficient inside the tubes of SH <sub>1HP</sub> , through the Steam      | 63 |
| 4.2.6  | Overall heat transfer coefficient for SH <sub>1HP</sub> .....                                    | 63 |
| 4.2.7  | Logarithmic mean temperature difference across SH <sub>1HP</sub> .....                           | 64 |
| 4.2.8  | The number of lateral rows of tubes required for SH <sub>1HP</sub> .....                         | 64 |
| 4.2.9  | Calculating the actual heat transfer rate of SH <sub>1HP</sub> .....                             | 64 |
| 4.2.10 | Determining the actual parameters of the flue gas exiting SH <sub>1HP</sub> .....                | 65 |
| 4.3    | Design of the High Pressure Evaporator EV <sub>HP</sub> .....                                    | 65 |
| 4.3.1  | Geometry of the high pressure evaporator EV <sub>HP</sub> .....                                  | 66 |
| 4.3.2  | Verification of the speed of flue gas passing through evaporator EV <sub>HP</sub> .....          | 68 |
| 4.3.3  | Reduced heat transfer coefficient outside the tubes of EV <sub>HP</sub> , through the flue gas   | 69 |
| 4.3.4  | Overall heat transfer coefficient for EV <sub>HP</sub> .....                                     | 71 |
| 4.3.5  | Logarithmic mean temperature difference across EV <sub>HP</sub> .....                            | 71 |
| 4.3.6  | The number of lateral rows of tubes required for EV <sub>HP</sub> .....                          | 71 |
| 4.3.7  | Calculating the actual heat transfer rate of EV <sub>HP</sub> .....                              | 72 |
| 4.3.8  | Determining the actual parameters of the flue gas exiting EV <sub>HP</sub> .....                 | 72 |
| 4.4    | Design of the High Pressure Economizer ECO <sub>3HP</sub> .....                                  | 73 |
| 4.4.1  | Geometry of the high pressure economizer ECO <sub>3HP</sub> .....                                | 73 |
| 4.4.2  | Verification of the speed of water passing through economizer ECO <sub>3HP</sub> .....           | 75 |
| 4.4.3  | Verification of the speed of flue gas passing through economizer ECO <sub>3HP</sub> .....        | 77 |
| 4.4.4  | Reduced heat transfer coefficient outside the tubes of ECO <sub>3HP</sub> , through the flue gas | 77 |
| 4.4.5  | Overall heat transfer coefficient for ECO <sub>3HP</sub> .....                                   | 80 |
| 4.4.6  | Logarithmic mean temperature difference across ECO <sub>3HP</sub> .....                          | 80 |
| 4.4.7  | The number of lateral rows of tubes required for ECO <sub>3HP</sub> .....                        | 80 |

|        |   |     |
|--------|---|-----|
| 4.4.8  | Calculating the actual heat transfer rate of $ECO_{3HP}$ .....                                      | 81  |
| 4.4.9  | Determining the actual parameters of the flue gas exiting $ECO_{3HP}$ .....                         | 81  |
| 4.5    | Design of the Low Pressure Superheater $SH_{LP}$ .....  | 81  |
| 4.5.1  | Geometry of the high pressure superheater $SH_{LP}$ .....   | 86  |
| 4.5.2  | Verification of the speed of steam passing through superheater $SH_{LP}$ .....                      | 87  |
| 4.5.3  | Verification of the speed of flue gas passing through superheater $SH_{LP}$ .....                   | 88  |
| 4.5.4  | Coefficient of heat transfer outside the tubes of $SH_{LP}$ .....                                   | 88  |
| 4.5.5  | Coefficient of heat transfer inside the tubes of $SH_{LP}$ .....                                    | 91  |
| 4.5.6  | Overall heat transfer coefficient for $SH_{LP}$ .....   | 91  |
| 4.5.7  | Logarithmic mean temperature difference across $SH_{LP}$ .....                                      | 91  |
| 4.5.8  | The number of lateral rows of tubes required for $SH_{LP}$ .....                                    | 92  |
| 4.5.9  | Calculating the actual heat transfer rate of $SH_{LP}$ .....  | 92  |
| 4.5.10 | Determining the actual parameters of the flue gas exiting $SH_{LP}$ .....                           | 92  |
| 4.6    | Design of the Low Pressure Evaporator $EV_{LP}$ .....   | 93  |
| 4.6.1  | Geometry of the high pressure evaporator $EV_{LP}$ .....  | 93  |
| 4.6.2  | Verification of the speed of flue gas passing through evaporator $EV_{LP}$ .....                    | 95  |
| 4.6.3  | Reduced heat transfer coefficient outside the tubes of $EV_{LP}$ , through the flue gas<br>96       |     |
| 4.6.4  | Overall heat transfer coefficient for $EV_{LP}$ .....   | 98  |
| 4.6.5  | Logarithmic mean temperature difference across $EV_{LP}$ .....                                      | 99  |
| 4.6.6  | The number of lateral rows of tubes required for $EV_{LP}$ .....                                    | 99  |
| 4.6.7  | Calculating the actual heat transfer rate of $EV_{LP}$ .....  | 100 |
| 4.6.8  | Determining the actual parameters of the flue gas exiting $EV_{LP}$ .....                           | 100 |
| 4.7    | Design of the High Pressure Economizer $ECO_{2HP}$ .....  | 100 |
| 4.7.1  | Geometry of the high pressure economizer $ECO_{2HP}$ .....  | 101 |
| 4.7.2  | Verification of the speed of water passing through economizer $ECO_{2HP}$ .....                     | 103 |
| 4.7.3  | Verification of the speed of flue gas passing through economizer $ECO_{2HP}$ .....                  | 104 |
| 4.7.4  | Reduced heat transfer coefficient outside the tubes of $ECO_{2HP}$ , through the flue<br>gas<br>105 |     |
| 4.7.5  | Overall heat transfer coefficient for $ECO_{2HP}$ .....   | 107 |
| 4.7.6  | Logarithmic mean temperature difference across $ECO_{2HP}$ .....                                    | 107 |
| 4.7.7  | The number of lateral rows of tubes required for $ECO_{2HP}$ .....                                  | 108 |
| 4.7.8  | Calculating the actual heat transfer rate of $ECO_{2HP}$ .....                                      | 108 |
| 4.7.9  | Determining the actual parameters of the flue gas exiting $ECO_{2HP}$ .....                         | 109 |
| 4.8    | Design of the Low Pressure Economizer $ECO_{LP}$ .....  | 109 |
| 4.8.1  | Geometry of the low pressure economizer $ECO_{LP}$ .....  | 110 |
| 4.8.2  | Verification of the speed of water passing through economizer $ECO_{LP}$ .....                      | 111 |
| 4.8.3  | Verification of the speed of flue gas passing through economizer $ECO_{LP}$ .....                   | 112 |

|       |   |     |
|-------|---|-----|
| 4.8.4 | Reduced heat transfer coefficient outside the tubes of $ECO_{LP}$ , through the flue gas  | 113 |
| 4.8.5 | Overall heat transfer coefficient for $ECO_{LP}$ .....                                    | 115 |
| 4.8.6 | Logarithmic mean temperature difference across $ECO_{LP}$ .....                           | 116 |
| 4.8.7 | The number of lateral rows of tubes required for $ECO_{LP}$ .....                         | 116 |
| 4.8.8 | Calculating the actual heat transfer rate of $ECO_{LP}$ .....                             | 117 |
| 4.8.9 | Determining the actual parameters of the flue gas exiting $ECO_{LP}$ .....                | 117 |
| 4.9   | Design of the High Pressure Economizer $ECO_{1HP}$ .....                                  | 117 |
| 4.9.1 | Geometry of the high pressure economizer $ECO_{1HP}$ .....                                | 118 |
| 4.9.2 | Verification of the speed of water passing through economizer $ECO_{1HP}$ .....           | 120 |
| 4.9.3 | Verification of the speed of the gas passing through economizer $ECO_{1HP}$ .....         | 121 |
| 4.9.4 | Reduced heat transfer coefficient outside the tubes of $ECO_{1HP}$ , through the flue gas | 122 |
| 4.9.5 | Overall heat transfer coefficient for $ECO_{1HP}$ .....                                   | 124 |
| 4.9.6 | Logarithmic mean temperature difference across $ECO_{1HP}$ .....                          | 124 |
| 4.9.7 | The number of lateral rows of tubes required for $ECO_{1HP}$ .....                        | 125 |
| 4.9.8 | Calculating the actual heat transfer rate of $ECO_{1HP}$ .....                            | 125 |
| 4.9.9 | Determining the actual parameters of the flue gas exiting $ECO_{1HP}$ .....               | 126 |
| 4.10  | Overview of the Calculated Values .....   | 126 |
| 5     | MATERIAL SELECTION .....  | 129 |
| 6     | STEAM DRUM DESIGN .....   | 131 |
| 6.1   | High Pressure Drum Design.....  | 131 |
| 6.2   | Low Pressure Drum Design .....  | 132 |
| 6.3   | Overview of the Calculated Values.....  | 133 |
| 7     | DESIGN AND SIZING OF THE DoWNCOMER AND RISER TUBES.....                                   | 135 |
| 7.1   | Downcomer Tube Sizing.....  | 135 |
| 7.1.1 | High pressure downcomer tube sizing.....  | 136 |
| 7.1.2 | Low pressure downcomer tube sizing.....   | 136 |
| 7.2   | Riser Tube Sizing.....  | 137 |
| 7.2.1 | Height pressure riser tube sizing.....  | 137 |
| 7.2.2 | Low pressure riser tube sizing .....  | 138 |
| 8     | FLUE GAS PRESSURE LOSSES THROUGH THE BOILER.....  | 139 |
| 8.1   | Flue Gas Pressure Losses across Individual Heat Exchangers .....                          | 139 |
| 8.1.1 | Pressure losses across superheater $SH_{2HP}$ .....                                       | 141 |
| 8.1.2 | Pressure losses across superheater $SH_{1HP}$ .....                                       | 141 |
| 8.1.3 | Pressure losses across evaporator $EV_{HP}$ .....   | 142 |
| 8.1.4 | Pressure losses across economizer $ECO_{3HP}$ .....                                       | 142 |
| 8.1.5 | Pressure losses across economizer $ECO_{2HP}$ .....                                       | 142 |

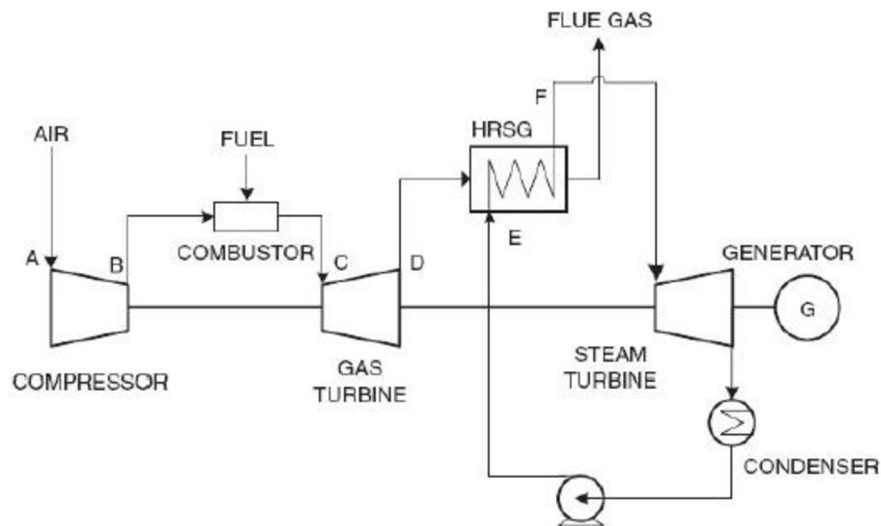
|        |   |     |
|--------|---|-----|
| 8.1.6  | Pressure losses across economizer $ECO_{IHP}$ .....                         | 143 |
| 8.1.7  | Pressure losses across superheater $SH_{LP}$ .....                          | 143 |
| 8.1.8  | Pressure losses across evaporator $EV_{LP}$ .....                           | 143 |
| 8.1.9  | Pressure losses across economizer $ECO_{LP}$ .....                          | 143 |
| 8.1.10 | Flue gas pressure losses across all the heat exchangers in the boiler ..... | 144 |
| 8.2    | Flue Gas Pressure Losses across the Stack .....                             | 144 |
| 8.2.1  | Pressure loss across the stack due to friction.....                         | 144 |
| 8.2.2  | Local pressure losses at the inlet and outlet of the stake .....            | 145 |
| 8.2.3  | Stack effect .....  | 145 |
| 8.2.4  | Overall pressure loss across the stack.....                                 | 146 |
| 8.3    | Overall Flue Gas Pressure Losses through the HRSG.....                      | 146 |
|        | Conclusion .....  | 147 |
|        | References.....   | 149 |
|        | List of symbols .....   | 151 |
|        | List of figures.....  | 157 |
|        | List of Tables .....  | 158 |
|        | appendix.....   | 161 |





## INTRODUCTION

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



*Figure 0-1 Combined cycle power plants [10]*

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

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

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



## 1 GENERAL DESCRIPTION

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

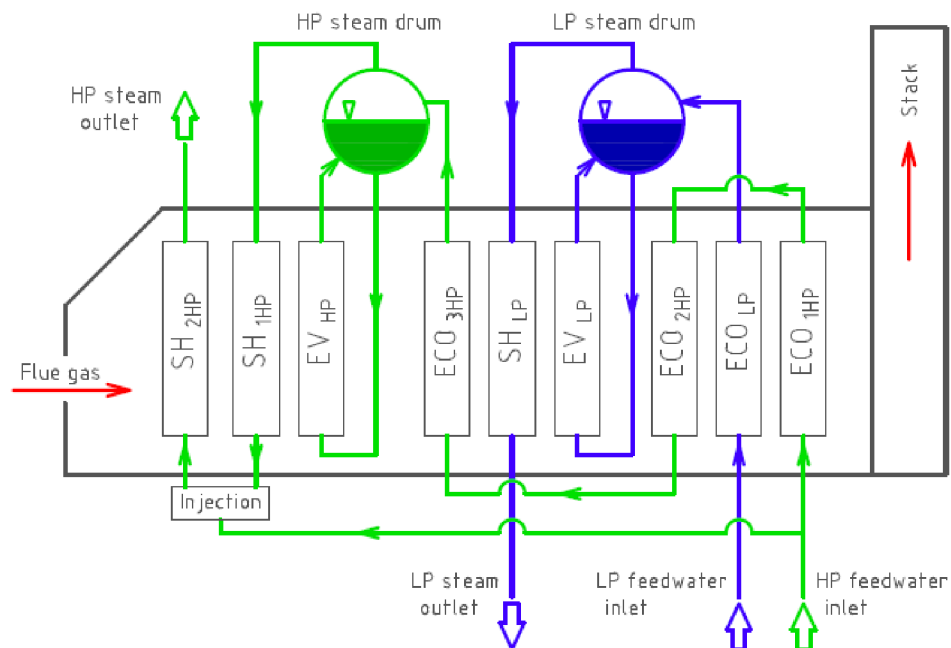


Figure 1-1 HRSG schematic

The design process begins with an outline of the HRSG temperature profile graph which is constructed according to the selected number of heat exchangers and the order in which they are arranged, seen in *figure 1-1*. According to the preliminary temperature profile graph the steam and flue gas parameters are calculated as they enter and exit each heat exchanger. The steam production rate and required heat transfer rate of each heat exchanger is determined. The geometry of the first heat exchanger to come in contact with the flue gas is selected and the inner dimensions of the HRSG casing (the flue gas duct dimensions) are defined accordingly. The rest of the heat exchangers are sized according to their heat transfer requirements and the

set dimensions of the flue gas duct. Then the steam drums, downcomer tubes and riser tubes are sized. The final step of the design is to verify that the pressure losses in the flue gas through the HRSG are not excessive.

### 1.1.1 Given Boiler Parameters

#### Flue gas inlet parameters:

|                |                                    |
|----------------|------------------------------------|
| Temperature    | $t_{Flue} = 490^{\circ}\text{C}$   |
| Mass flow rate | $\dot{M}_{Flue} = 30 \text{ kg/s}$ |

#### Feedwater inlet parameters:

|             |                               |
|-------------|-------------------------------|
| Temperature | $t_{FW} = 53^{\circ}\text{C}$ |
|-------------|-------------------------------|

#### High pressure steam output parameters:

|             |                                  |
|-------------|----------------------------------|
| Pressure    | $p_1^{HP} = 4.6 \text{ MPa}$     |
| Temperature | $t_1^{HP} = 450^{\circ}\text{C}$ |

#### Low pressure steam output parameters:

|             |                                  |
|-------------|----------------------------------|
| Pressure    | $p_1^{LP} = 0.46 \text{ MPa}$    |
| Temperature | $t_1^{LP} = 170^{\circ}\text{C}$ |

#### Flue gas volumetric composition:

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

| Parameter                      | Symbol     | Value | Units |
|--------------------------------|------------|-------|-------|
| O <sub>2</sub> concentration   | $x_{O_2}$  | 0.139 | [ - ] |
| Ar concentration               | $x_{Ar}$   | 0.009 | [ - ] |
| N <sub>2</sub> concentration   | $x_{N_2}$  | 0.73  | [ - ] |
| CO <sub>2</sub> concentration  | $x_{CO_2}$ | 0.044 | [ - ] |
| H <sub>2</sub> O concentration | $x_{H_2O}$ | 0.078 | [ - ] |

*Table 1.1 Volumetric composition, concentrations*

#### Maximum acceptable flue gas pressure loss:

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

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

## 2 THERMAL CALCULATIONS OF THE BOILER

### 2.1 Preliminary Temperature Profile Graph of the Boiler

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

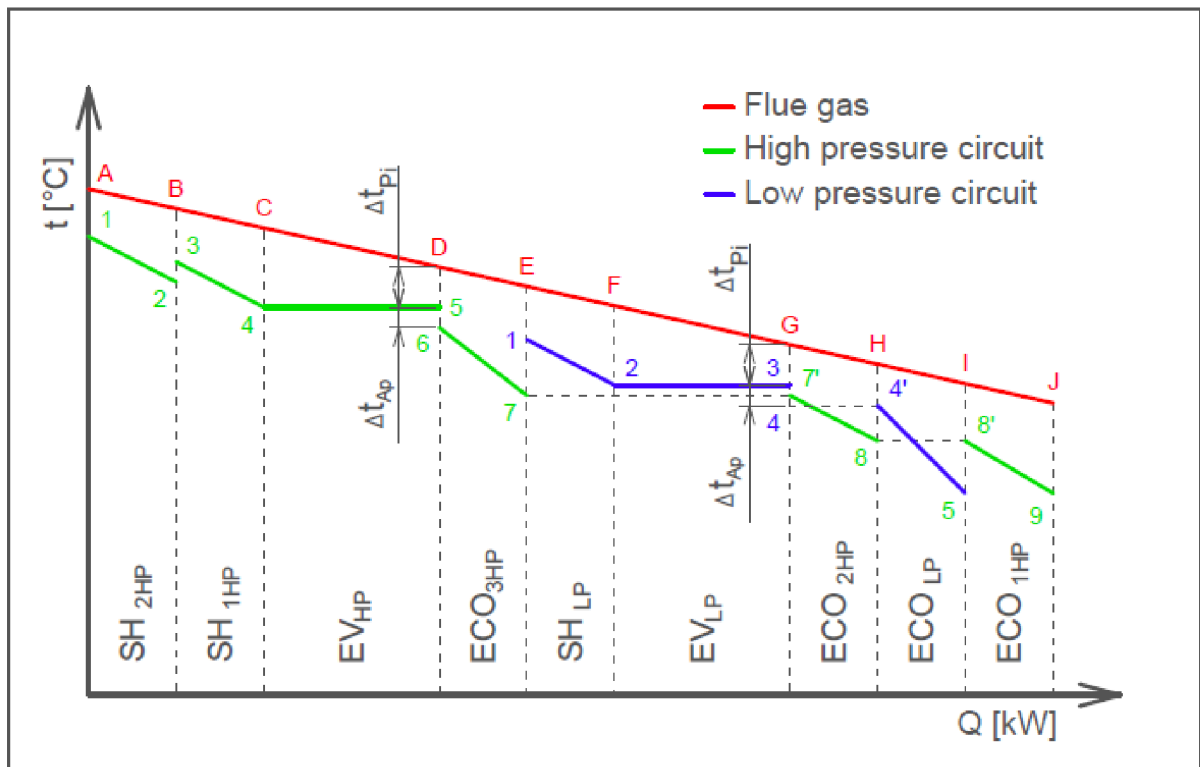


Figure 2-1 HRSG preliminary temperature profile graph

### 2.2 Selected Parameters Values

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

| Parameters   | Symbol                 | Value | Units        |
|--|------------------------|-------|--------------|
| High pressure superheater SH <sub>2HP</sub> steam pressure loss                              | $\Delta p_{SH2}^{HP}$  | 0.1   | MPa          |
| High pressure superheater SH <sub>1HP</sub> steam pressure loss                              | $\Delta p_{SH1}^{HP}$  | 0.1   | MPa          |
| High pressure economizer ECO <sub>3HP</sub> water pressure loss                              | $\Delta p_{ECO3}^{HP}$ | 0.1   | MPa          |
| Low pressure superheater SH <sub>LP</sub> steam pressure loss                                | $\Delta p_{SH}^{LP}$   | 0.1   | MPa          |
| High pressure economizer ECO <sub>2HP</sub> water pressure loss                              | $\Delta p_{ECO2}^{HP}$ | 0.1   | MPa          |
| Low pressure economizer ECO <sub>LP</sub> water pressure loss                                | $\Delta p_{ECO}^{LP}$  | 0.3   | MPa          |
| High pressure economizer ECO <sub>1HP</sub> water pressure loss                              | $\Delta p_{ECO1}^{HP}$ | 0.1   | MPa          |
| High pressure superheater SH <sub>2HP</sub> enthalpy difference                              | $\Delta i_{SH2}^{HP}$  | 250   | <i>kJ/kg</i> |
| High pressure approach point   | $\Delta t_{Ap}^{HP}$   | 5     | °C           |
| Low pressure approach point  | $\Delta t_{Ap}^{LP}$   | 5     | °C           |
| High pressure pinch point  | $\Delta t_{Pi}^{HP}$   | 10    | °C           |
| Low pressure pinch point   | $\Delta t_{Pi}^{HP}$   | 10    | °C           |
| High pressure economizer ECO <sub>3HP</sub> temperature gradient                             | $\Delta t_{ECO3}^{HP}$ | 105   | °C           |
| High pressure economizer ECO <sub>2HP</sub> temperature gradient                             | $\Delta t_{ECO2}^{HP}$ | 12    | °C           |
| Amount of high pressure feedwater used to regulate outlet steam parameters (water injection) | $\dot{M}_{W\%}$        | 5     | %            |

Table 2.1 Selected parameters values

## 2.3 Water/Steam Parameters

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

### 2.3.1 High pressure water/steam parameters

#### Point 1<sub>HP</sub>:

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

$$t_1^{HP} = 450^\circ\text{C} \quad (\text{given value})$$

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

#### Point 2<sub>HP</sub>:

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

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

$$t_2^{HP} = 348.48^\circ\text{C} \quad (\text{determined using X-Steam, } f(p,t))$$

**Point 4<sub>HP</sub>:**

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

$$t_4^{HP} = 261.4^\circ\text{C} \quad (\text{X-Steam, saturation temperature})$$

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

**Point 5<sub>HP</sub>:**

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

$$t_5^{HP} = t_4^{HP} = 261.4^\circ\text{C} \quad (\text{saturation temperature})$$

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

**Point 6<sub>HP</sub>:**

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

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

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

**Point 7<sub>HP</sub>:**

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

$$t_7^{HP} = t_6^{HP} - t_{ECO3}^{HP} = 256.4 - 105 = 151.4^\circ\text{C}$$

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

**Point 8<sub>HP</sub>:**

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

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

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

**Point 9<sub>HP</sub>:**

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

$$t_9^{HP} = t_{FW} = 53^\circ\text{C} \quad (\text{given value})$$

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

**Point 3<sub>HP</sub>:**

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

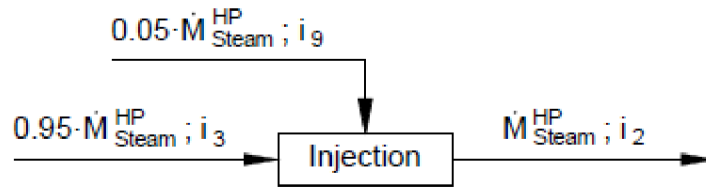


Figure 2-2 Feedwater injection

Conservation equation:

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

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

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

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

### 2.3.2 Low pressure water/steam parameters

#### Point 1<sub>LP</sub>:

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

$$t_1^{LP} = 170^\circ\text{C} \quad (\text{given})$$

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

#### Point 2<sub>LP</sub>:

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

$$t_2^{LP} = 156.15^\circ\text{C} \quad (\text{X-Steam, saturation temperature})$$

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

#### Point 3<sub>LP</sub>:

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

$$t_3^{LP} = 156.15^\circ\text{C} \quad (\text{X-Steam, saturation temperature})$$

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

#### Point 4<sub>LP</sub>:

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



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

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

### Point 5<sub>LP</sub>:

$$t_5^{LP} = t_{FW} = 53^\circ\text{C}$$

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

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

## 2.4 Flue Gas Parameters

### 2.4.1 Flue gas density

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

| Parameter                | Symbol        | Value  | Units                 |
|--------------------------|---------------|--------|-----------------------|
| O <sub>2</sub> density   | $\rho_{O_2}$  | 1.4289 | [kg/Nm <sup>3</sup> ] |
| Ar density               | $\rho_{Ar}$   | 1.7839 | [kg/Nm <sup>3</sup> ] |
| N <sub>2</sub> density   | $\rho_{N_2}$  | 1.2505 | [kg/Nm <sup>3</sup> ] |
| CO <sub>2</sub> density  | $\rho_{CO_2}$ | 1.9768 | [kg/Nm <sup>3</sup> ] |
| H <sub>2</sub> O density | $\rho_{H_2O}$ | 0.804  | [kg/Nm <sup>3</sup> ] |

Table 2.2 Densities of individual flue gas components

$$\rho_{Flue} = x_{O_2} \cdot \rho_{O_2} + x_{Ar} \cdot \rho_{Ar} + x_{N_2} \cdot \rho_{N_2} + x_{CO_2} \cdot \rho_{CO_2} + x_{H_2O} \cdot \rho_{H_2O} \quad (2.2)$$

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

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

### 2.4.2 Flue gas volumetric flow rate, under normal conditions

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

$$\dot{M}_{VFlue} = \frac{\dot{M}_{Flue}}{\rho_{Flue}} \quad (2.3)$$

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

### 2.4.3 Flue gas enthalpy calculations at various points

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

$$I_{Flue} = x_{O_2} \cdot I_{O_2} + x_{Ar} \cdot I_{Ar} + x_{N_2} \cdot I_{N_2} + x_{CO_2} \cdot I_{CO_2} + x_{H_2O} \cdot I_{H_2O} \quad (2.4)$$

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

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

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

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

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

#### Point A:

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

$$\text{Temperature} \quad t_A = t_{Flue} = 490^\circ\text{C}$$

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

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

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

**Point D:**

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

$$\text{Temperature} \quad t_D = t_5^{HP} + t_{pi}^{HP} = 261.4 + 10 = 271.4^\circ\text{C}$$

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

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

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

**Point G:**

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

$$\text{Temperature} \quad t_G = t_3^{LP} + t_{pi}^{LP} = 156.15 + 10 = 166.15^\circ\text{C}$$

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

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

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

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

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

$$Q_{RC} = C \cdot Q_N^{0.7} \quad (2.8)$$

Where:

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

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

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

### Maximum usable thermal power:

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

$$Q_N = Q_A = I_A \cdot \dot{M}_{VFlue} \quad (2.9)$$

$$Q_N = Q_A = 678.69 \cdot 23.49 = 15941.29 \text{ kW}$$

### Heat loss through radiation and convection:

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

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

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

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

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

### 2.6.1 High pressure steam production rate

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

#### Rate of heat released between points A and D:

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

#### Rate of heat absorbed between points 1<sub>HP</sub> and 6<sub>HP</sub>:

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

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

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

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

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

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

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

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

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

### Heat transfer rate of the high pressure superheater SH<sub>2HP</sub>:

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

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

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

### Heat transfer rate of the high pressure superheater SH<sub>1HP</sub>:

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

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

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

### Heat transfer rate of the high pressure evaporator EV<sub>HP</sub>:

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

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

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

**Heat transfer rate of the high pressure economizer ECO<sub>3HP</sub>:**

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

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

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

**Heat transfer rate of the high pressure economizer ECO<sub>2HP</sub>:**

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

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

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

**Heat transfer rate of the high pressure economizer ECO<sub>1HP</sub>:**

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

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

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

**2.6.3 Low pressure steam production rate**

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

**Flue gas parameter at point E:**

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

The rate of heat absorbed by the high pressure circuit between points 1<sub>HP</sub> and 7<sub>HP</sub> is presented in equation (2.20).

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

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

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

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

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

Flue gas enthalpy at point E:

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

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

Flue gas temperature at point E:

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

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

$$t_E = \frac{303.77 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 225.92^\circ\text{C}$$

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

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

**Rate of heat absorbed between points 1<sub>LP</sub> and 4<sub>LP</sub>:**

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

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

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

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

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

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

#### Heat transfer rate of the low pressure superheater SH<sub>LP</sub>:

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

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

#### Heat transfer rate of the low pressure evaporator EV<sub>LP</sub>:

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

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

#### Heat transferred through the low pressure economizer ECO<sub>LP</sub>:

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

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

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

### Point B:

Flue gas enthalpy at point B:

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

Flue gas temperature at point B:

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

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

$$t_B = \frac{644 - 547.44}{693.27 - 547.44} \cdot 100 + 400 = 466.22^\circ\text{C}$$



**Point C:**

Flue gas enthalpy at point C:

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

Flue gas temperature at point C:

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

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

$$t_C = \frac{587.77 - 547.44}{693.27 - 547.44} \cdot 100 + 400 = 427.66^\circ\text{C}$$

**Point F:**

Flue gas enthalpy at point F:

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

Flue gas temperature at point F:

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

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

$$t_F = \frac{302.26 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 224.83^\circ\text{C}$$

**Point H:**

Flue gas enthalpy at point H:

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

Flue gas temperature at point H:

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

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

$$t_H = \frac{215.63 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 161.13^\circ\text{C}$$

**Point I:**

Flue gas enthalpy at point I:

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

Flue gas temperature at point I:

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

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

$$t_I = \frac{199.98 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 149.52^\circ\text{C}$$

**Point J:**

Flue gas enthalpy at point J:

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

Flue gas temperature at point J:

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

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

$$t_J = \frac{152.08 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 113.96^\circ\text{C}$$

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

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

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

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

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

Table 2.5 Flue gas parameters at various points throughout the boiler

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

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

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

Table 2.7 Other important values

### HRSG Temperature profile:

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

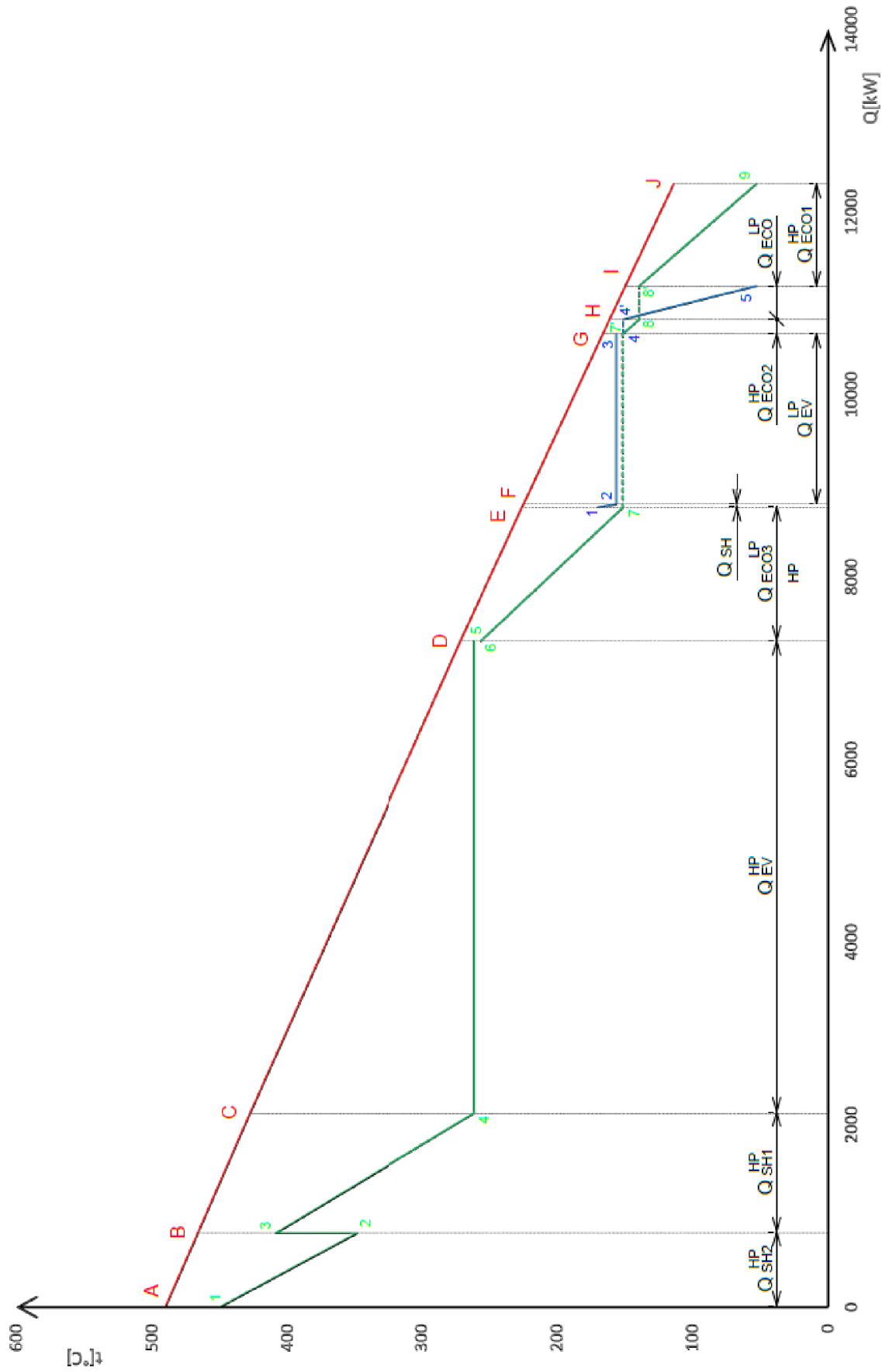


Figure 2-3 HRSG Temperature profile



### 3 FLUE GAS DUCT DESIGN

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

#### 3.1 Finned Tube Geometry of the High Pressure Superheater (SH<sub>2HP</sub>)

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

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

Table 3.1 Parameters selected for the finned tubes used in the high pressure superheater (SH<sub>2HP</sub>)

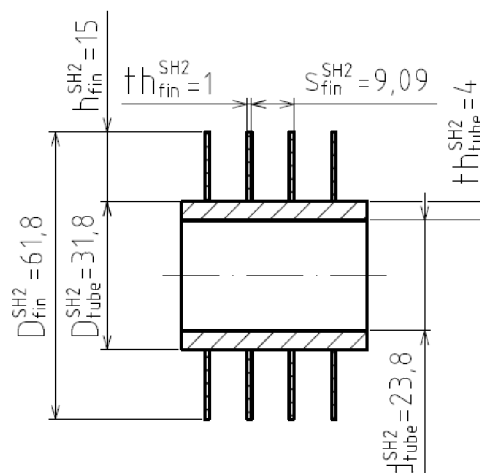


Figure 3-1 Tube geometry SH<sub>2HP</sub>

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

**Inner tube diameter:**

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

**Outer fin diameter:**

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

**Fin spacing:**

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

### 3.1.1 Determining the number of tubes in each lateral row of the high pressure superheater (SH<sub>2HP</sub>)

**Average specific volume of steam in SH<sub>2HP</sub>:**

The average specific volume of the steam in SH<sub>2HP</sub> is required later in the continuity equation (3.5), this value is determined according to the average pressure and temperature of the steam inside SH<sub>2HP</sub>.

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

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

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

**Determining the number of tubes per lateral row required for SH<sub>2HP</sub>:**

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

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

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

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



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

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

The actual speed of the steam in SH<sub>2HP</sub> must be recalculated according to the actual number of tubes in each row.

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

### 3.1.2 Volumetric follow rate of flue gas passing through the high pressure superheater (SH<sub>2HP</sub>)

The actual volumetric flow rate around superheater SH<sub>2HP</sub> is calculated in equation (3.9), according to the average flue gas temperature between points A and B, as seen in equation (3.8).

$$\bar{t}_{(AB)} = \frac{t_A + t_B}{2} = \frac{490 + 466.22}{2} = 478.11^\circ\text{C} \quad (3.8)$$

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

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

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

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

## 3.2 Flue Gas Duct Dimensions

### Superheater SH<sub>2HP</sub> lateral tube spacing inside the flue gas duct:

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

Where:

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

### Width of the flue gas duct:

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

**Height of the duct:**

$$A_{duct}^{SH_2} = H \cdot L - H \cdot D_{tube}^{SH_2} \cdot n_{tube/r}^{SH_2} - H \cdot 2 \cdot h_{fin}^{SH_2} \cdot t h_{fin}^{SH_2} \cdot n_{fin}^{SH_2} \cdot n_{tube/r}^{SH_2} \quad (3.13)$$

$$H = \frac{A_{dust}^{SH_2}}{L - D_{tube}^{SH_2} \cdot n_{tube/r}^{SH_2} - 2 \cdot h_{fin}^{SH_2} \cdot t h_{fin}^{SH_2} \cdot n_{fin}^{SH_2} \cdot n_{tube/r}^{SH_2}}$$

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

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

$$\frac{H}{L} = \frac{6.76}{1.76} = 3.85 \quad (3.14)$$

The actual speed of the flue gas passing through SH<sub>2HP</sub> must be recalculated according to the duct dimensions which are rounded to the nearest centimeter.

$$W_{Flue}^{SH_2} = \frac{\dot{M}_{VFlue}^{SH_2}}{L \cdot H - n_{tube/r}^{SH_2} \cdot H \cdot (D_{tube}^{SH_2} + 2 \cdot h_{fin}^{SH_2} \cdot t h_{fin}^{SH_2} \cdot n_{fin}^{SH_2})} \quad (3.15)$$

$$W_{Flue}^{SH_2} = \frac{64.6}{1.76 \cdot 6.78 - 23 \cdot 6.78 \cdot (0.0318 + 2 \cdot 0.015 \cdot 0.001 \cdot 110)} = 10 \text{ m/s}$$

**3.2.1 Overview of the calculated values**

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

| Parameter   | Symbol              | Value | Units |
|---|---------------------|-------|-------|
| Actual speed of steam in SH <sub>2HP</sub>              | $W_{Steam}^{SH_2}$  | 19.79 | [m/s] |
| Actual speed of flue gas around SH <sub>2HP</sub>       | $W_{Flue}^{SH_2}$   | 10    | [m/s] |
| Number of tubes in one lateral row of SH <sub>2HP</sub> | $n_{tube/r}^{SH_2}$ | 23    | [–]   |
| Width of the flue gas duct                              | $L$                 | 1.76  | [m]   |
| Height of the flue gas duct                             | $H$                 | 6.78  | [m]   |

*Table 3.1* Calculated values defining duct and superheater (SH<sub>2HP</sub>) geometry, as well as actual flow speeds.

## 4 DESIGN AND SIZING OF HEAT TRANSFER SURFACES

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

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

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

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

the logarithmic mean temperature difference  $\Delta t_{ln}$ . These values are dependent on the geometry of the heat exchanger, as well as the parameters of the water/steam and flue gas.

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

**Equations used in the design and sizing of heat exchangers:**

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

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

*Table 4.1 Collective values used in the heat transfer calculations*

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

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

Where:

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

$S_{out/r}$  is the outer surface area of all the tubes in one lateral row of the heat exchanger [m<sup>2</sup>]

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

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

Where:

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

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

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

Where:

$S_{1fin}$  is the outer surface area of one fin [m<sup>2</sup>]

**The outer surface area of one fin:**

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

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

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

$$Q = k \cdot S_{out} \cdot \Delta t_{ln} \quad (4.5)$$

Where:

$Q$  is the heat transfer rate of the heat exchanger [W]

$k$  is the overall heat transfer coefficient [W/(m<sup>2</sup>K)]

$\Delta t_{ln}$  is the logarithmic mean temperature difference [K]

**The logarithmic mean temperature difference:**

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

Where:

- $\Delta t_1$  temperature difference between the flue gas entering and the water/steam exiting the heat exchanger [K]
- $\Delta t_2$  temperature difference between the flue gas exiting and the water/steam entering the heat exchanger [K]

**The overall heat transfer coefficient:**

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

Where:

- $\alpha_{r:out}$  is the reduced heat transfer coefficient outside the tubes, between the flue gas and the finned heat exchanger tubes [W/(m<sup>2</sup>K)]
- $\alpha_{r:in}$  is the reduced heat transfer coefficient inside the tubes, between the finned heat exchanger tubes and the water/steam inside them [W/(m<sup>2</sup>K)]
- $S_{in/1m}$  is the inner surface area of one finned tube per meter length, or the area that is transferring heat to and in contact with the water/steam flowing through the tube per meter length [m]

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

$$S_{in/1m} = \pi \cdot d_{tube} \quad (4.8)$$

**The reduced heat transfer coefficient outside the tubes:**

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

Where:

- $E$  is a coefficient characterizing the effectiveness of the fins, this value is determined from a graph according to the product of  $\beta \cdot h_{fin}$  and the quotient of  $D_{fin}/D_{tube}$ , where  $\beta$  is a coefficient [-]
- $\mu$  is a coefficient characterizing fin taper [-]
- $\Psi_{fin}$  is a coefficient characterizing the uneven distribution of the convective heat transfer coefficient ( $\alpha_c$ ) along surface of the fins [-]
- $\alpha_c$  The coefficient of heat transfer through convection outside of the heat exchanger tubes [W/(m<sup>2</sup>K)]

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

**Coefficient  $\beta$ :**

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

Where:

|                 |   |
|-----------------|---|
| $\beta$         | Coefficient used to calculate the coefficient characterizing the effectiveness of the fins [ $1/\text{m}$ ] |
| $\lambda_{fin}$ | is the thermal conductivity of the fins [ $\text{W}/(\text{m} \cdot \text{K})$ ]                            |

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

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

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

$$\frac{S_{out-fin}}{S_{out}} = 1 - \frac{S_{fin}}{S_{out}} \quad (4.12)$$

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

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

Where:

|                  |  |
|------------------|--|
| $c_z$            | is the correction coefficient for the number of lateral row of tubes in the heat exchanger [-] |
| $\varphi_\sigma$ | is a coefficient characterizing the relative tube spacing [-]                                  |
| $\lambda_{Flue}$ | is the thermal conductivity of the flue gas [ $\text{W}/(\text{m} \cdot \text{K})$ ]           |
| $\nu_{Flue}$     | is the kinematic viscosity of the flue gas [ $\text{m}^2/\text{s}$ ]                           |

**Coefficient of the relative tube spacing:**

$$\varphi_{\sigma} = \frac{\sigma_1 - 1}{\sigma_2' - 1} \quad (4.14)$$

Where:

$\sigma_1$  is the lateral tube spacing relative to the outer tube diameter [-]

$\sigma_2'$  is the diagonal tube spacing relative to the outer tube diameter [-]

**Lateral tube spacing relative to the outer tube diameter:**

$$\sigma_1 = \frac{s_1}{D_{tube}} \quad (4.15)$$

**Diagonal tube spacing relative to the outer tube diameter:**

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

Where:

$s'$  is the diagonal tube spacing [m]

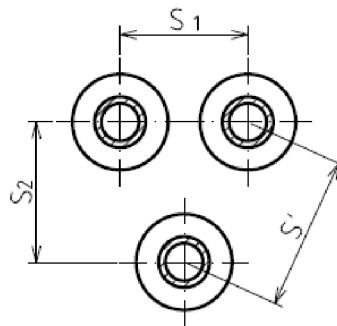


Figure 4-1 Tube spacing

**The reduced heat transfer coefficient inside the tubes:**

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

Where:

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

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

$\nu_{Steam}$  is the kinematic viscosity of the water/steam [m<sup>2</sup>/s]



- $Pr_{Steam}$  is the Prandtl number of the water/steam [-]
- $c_t$  is the correction coefficient dependent on the temperature of the water/steam and tube wall temperature [-]
- $c_l$  is the correction coefficient dependent on the tube length relative to the tube diameter [-]
- $c_m$  is the correction coefficient for fluid flow between concentric tubes [-]

### Flue gas parameters:

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

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

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

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

Table 4.3 Flue gas coefficient of kinematic viscosity,  $\nu_{Flue} \cdot 10^6$  [m<sup>2</sup>/s]

### Water/steam parameters:

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

Kinematic viscosity of the steam:

$$v_{Steam} = \frac{\mu_{Steam}}{\rho_{Steam}} \quad (4.18)$$

Where:

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

$\rho_{Steam}$  is the density of the water/steam passing through the heat exchanger [kg/m<sup>3</sup>], determined using X-Steam according to average pressure and temperature

## 4.1 Design of the High Pressure Superheater SH<sub>2HP</sub>

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

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

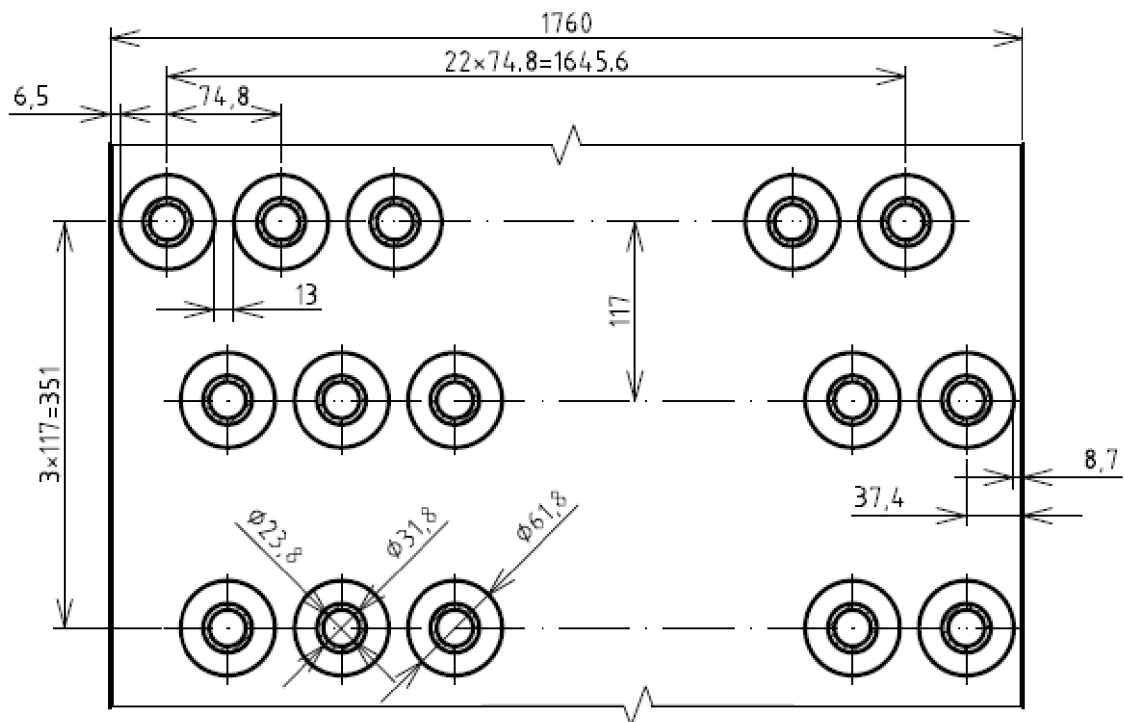
Table 4.4 Parameters necessary for superheater SH<sub>2HP</sub> design, not including parameters describing the geometry of SH<sub>2HP</sub>

### 4.1.1 Geometry of the high pressure superheater SH<sub>2HP</sub>

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

| Dimension/parameter                            | Symbol              | Value | Units |
|--|---------------------|-------|-------|
| Lateral tube spacing of SH <sub>2HP</sub>      | $s_1(SH_2)$         | 74.8  | [mm]  |
| Longitudinal tube spacing of SH <sub>2HP</sub> | $s_2(SH_2)$         | 117   | [mm]  |
| Number of tubes in each lateral row            | $n_{tube/r}^{SH_2}$ | 23    | [-]   |

*Table 4.5 Values describing the geometry of superheater SH<sub>2HP</sub>*



*Figure 4-2 Tube layout SH<sub>2HP</sub>*

### 4.1.2 Reduced heat transfer coefficient outside the tubes of SH<sub>2HP</sub>, through the flue gas

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

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{SH_2} = \frac{s_1(SH_2)}{D_{tube}^{SH_2}} = \frac{0.0748}{0.0318} = 2.35$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{SH2} = \frac{s'^{SH2}}{D_{tube}^{SH2}} = \frac{\sqrt{\left(\frac{s_1(SH2)}{2}\right)^2 + (s_2(SH2))^2}}{D_{tube}^{SH2}} = \frac{\sqrt{\left(\frac{0.0748}{2}\right)^2 + 0.117^2}}{0.0318} = 3.86$$

Coefficient of the relative tube spacing:

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

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

Flue gas Parameters:

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

$$\text{Temperature} \quad \bar{t}_{(AB)} = 478.11^\circ\text{C}$$

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

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

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

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 4 lateral rows of tubes in superheater SH<sub>2HP</sub>.

$$c_z^{SH2} = 0.94$$

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

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

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

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

**Coefficient characterizing the effectiveness of the fins:**

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

Coefficient  $\beta^{SH2}$ :

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

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

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

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

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

Coefficient determined using the graph:

$$E^{SH2} = 0.81$$

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

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

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

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

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

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

### The reduced heat transfer coefficient outside the tubes:

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

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

### 4.1.3 Reduced heat transfer coefficient inside the tubes of SH<sub>2HP</sub>, through the Steam

#### Steam parameters:

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

$$\text{Temperature} \quad \bar{t}_{SH2} = 399.24 \text{ }^\circ\text{C}$$

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

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

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

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

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

#### Kinematic viscosity of the steam:

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

#### Reduced heat transfer coefficient inside the tubes of SH<sub>2HP</sub>:

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

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

### 4.1.4 Overall heat transfer coefficient for SH<sub>2HP</sub>

#### The outer surface area of one fin:

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

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

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

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

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

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

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

**The overall heat transfer coefficient:**

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

#### 4.1.5 Logarithmic mean temperature difference across SH<sub>2HP</sub>

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

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

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

#### 4.1.6 The number of lateral rows of tubes required for SH<sub>2HP</sub>

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

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

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

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

**The number of lateral rows of tubers in SH<sub>2HP</sub>:**

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

Naturally tis value must be rounded to the nearest whole number

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

#### 4.1.7 Calculating the actual heat transfer rate of SH<sub>2HP</sub>

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

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

The actual heat transfer rate of SH<sub>2HP</sub>:

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

Error verification:

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

The discrepancy between the theoretical and the actual heat transfer rate of superheater SH<sub>2HP</sub> is less than 2%, thus the current configuration of SH<sub>2HP</sub> is acceptable.

#### 4.1.8 Determining the actual parameters of the flue gas exiting SH<sub>2HP</sub>

Due to the discrepancy between the theoretical and the actual heat transfer rate of superheater SH<sub>2HP</sub> the parameters of the flue gas exiting SH<sub>2HP</sub> at point B must be recalculated.

Actual enthalpy of the flue gas at point B:

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

Actual temperature of the flue gas at point B:

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

## 4.2 Design of the High Pressure Superheater SH<sub>1HP</sub>

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



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

Table 4.6 Parameters necessary for superheater SH<sub>IHP</sub> design, not including parameters describing the geometry of SH<sub>IHP</sub>

#### 4.2.1 Geometry of the high pressure superheater SH<sub>IHP</sub>

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

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

Table 4.7 Parameters selected for the finned tubes used in the high pressure superheater SH<sub>IHP</sub>

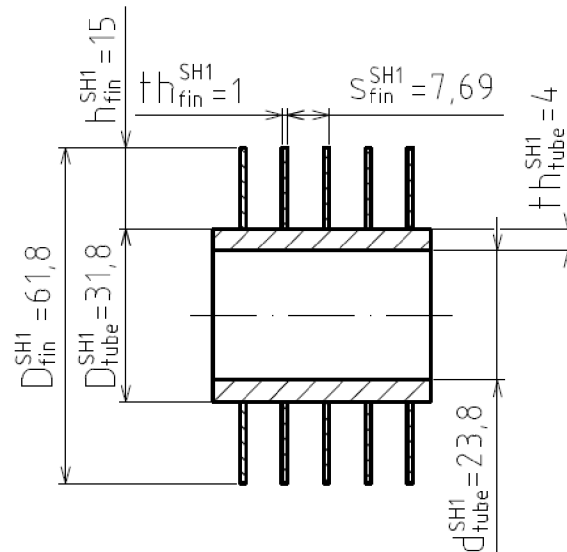


Figure 4-3 Tube geometry  $SH_{IHP}$

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

#### Chosen layout dimensions:

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

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

#### Lateral tube spacing in superheater $SH_{IHP}$ :

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

#### Number of tubes in each lateral row of tubes in $SH_{IHP}$ :

Naturally this value must be rounded to the nearest whole number

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

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

#### Gap verification:

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

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

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

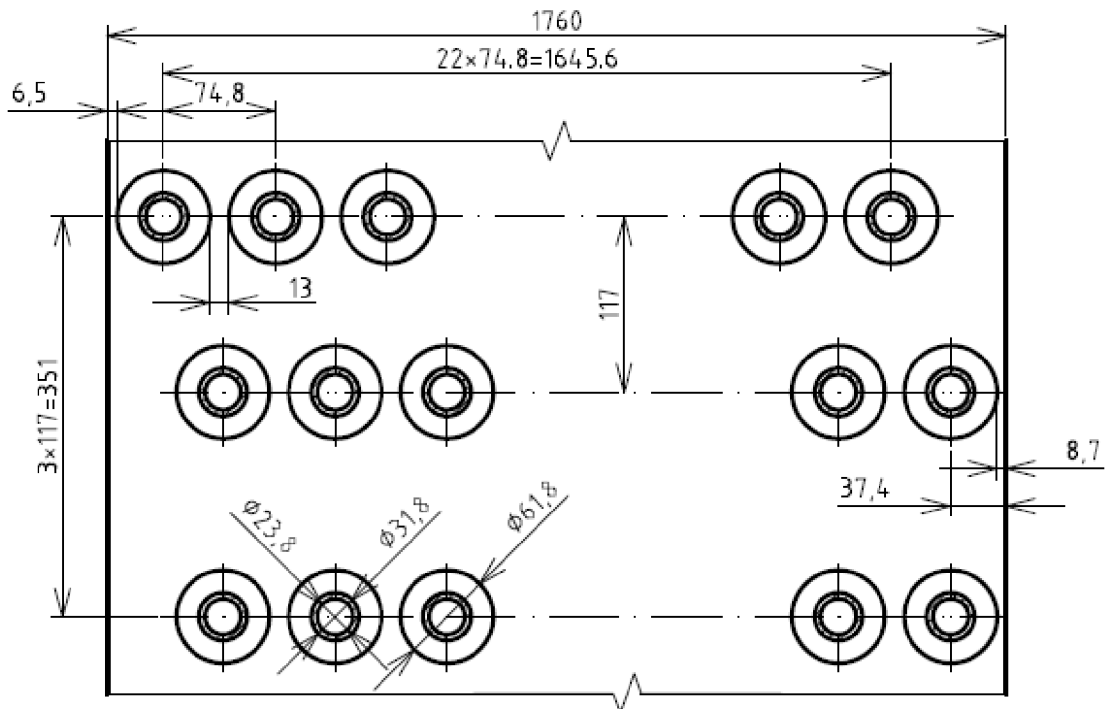


Figure 4-4 Tube layout SH<sub>1HP</sub>

#### 4.2.2 Verification of the speed of steam passing through superheater SH<sub>1HP</sub>

The speed of the steam passing through superheater SH<sub>1HP</sub> is calculated according to average specific volume of the steam. The specific volume of the steam is determined through X-Steam according to the average pressure and temperature of the steam passing through SH<sub>1HP</sub>.

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

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

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

#### Speed of steam:

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

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

### 4.2.3 Verification of the speed of flue gas passing through superheater SH<sub>1HP</sub>

#### Recalculated flue gas enthalpy and temperature at point C:

The parameters of the flue gas exiting the superheater at point C are recalculated according to the actual flue gas parameter entering SH<sub>1HP</sub>.

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

$$t'_C = \frac{I'_C - I_{Flue:400}}{I_{Flue:500} - I_{Flue:400}} \cdot (500 - 400) + 400 = \frac{587.92 - 547.44}{693.27 - 547.44} \cdot 100 + 400 = 427.76^\circ\text{C}$$

#### Average flue gas temperature:

$$\bar{t}_{(BC)} = \frac{t_B^{real} + t'_C}{2} = \frac{466.32 + 427.76}{2} = 447.04^\circ\text{C}$$

#### The actual volumetric flow rate of the flue gas:

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

#### The cross sectional area that the flue gas flows through:

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

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

#### Speed of the flue gas passing through SH<sub>1HP</sub>:

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

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

### 4.2.4 Reduced heat transfer coefficient outside the tubes of SH<sub>1HP</sub>, through the flue gas

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

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{SH1} = \frac{S_1(SH1)}{D_{tube}^{SH1}} = \frac{0.0748}{0.0318} = 2.35$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{SH1} = \frac{S'^{SH1}}{D_{tube}^{SH1}} = \frac{\sqrt{\left(\frac{S_1(SH1)}{2}\right)^2 + (S_2(SH1))^2}}{D_{tube}^{SH1}} = \frac{\sqrt{\left(\frac{0.0748}{2}\right)^2 + 0.117^2}}{0.0318} = 3.86$$

Coefficient of the relative tube spacing:

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

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

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are determined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through SH<sub>IHP</sub> has been determined in previous calculations regarding the flue gas speed verification.

$$\text{Temperature} \quad \bar{t}_{(BC)} = 447.04 \text{ }^{\circ}\text{C}$$

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

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

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

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 4 lateral rows of tubes in superheater SH<sub>IHP</sub>.

$$c_z^{SH1} = 0.94$$

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

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

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

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

**Coefficient characterizing the effectiveness of the fins:**

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

Coefficient  $\beta^{SH1}$ :

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

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

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

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

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

Coefficient determined using the graph:

$$E^{SH1} = 0.84$$

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

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

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

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

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

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

**The reduced heat transfer coefficient outside the tubes:**

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

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

#### 4.2.5 Reduced heat transfer coefficient inside the tubes of SH<sub>1HP</sub>, through the Steam

##### Steam parameters:

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

$$\text{Temperature} \quad \bar{t}_{SH1} = 334.93 \text{ }^{\circ}\text{C}$$

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

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

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

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

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

##### Kinematic viscosity of the steam:

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

##### Reduced heat transfer coefficient inside the tubes of SH<sub>2HP</sub>:

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

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

#### 4.2.6 Overall heat transfer coefficient for SH<sub>1HP</sub>

##### The outer surface area of one fin:

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

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

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

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

$$S_{out/1m}^{SH1} = \pi \cdot 0.0318 \cdot (1 - 130 \cdot 0.001) + 130 \cdot 0.0046 = 0.6856 \text{ m}^2$$

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

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

**The overall heat transfer coefficient:**

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

#### 4.2.7 Logarithmic mean temperature difference across SH<sub>1HP</sub>

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

$$\Delta t_2^{SH1} = t'_C - t_4^{HP} = 427.76 - 261.4 = 166.35 \text{ K}$$

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

#### 4.2.8 The number of lateral rows of tubes required for SH<sub>1HP</sub>

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

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

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

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

**The number of lateral rows of tubers in SH<sub>2HP</sub>:**

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

Naturally tis value must be rounded to the nearest whole number

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

#### 4.2.9 Calculating the actual heat transfer rate of SH<sub>1HP</sub>

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

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



**The actual heat transfer rate of SH<sub>IHP</sub>:**

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

**Error verification:**

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

The discrepancy between the theoretical and the actual heat transfer rate of superheater SH<sub>IHP</sub> is less than 2% , thus the current configuration of SH<sub>IHP</sub> is acceptable.

**4.2.10 Determining the actual parameters of the flue gas exiting SH<sub>IHP</sub>**

Due to the discrepancy between the theoretical and the actual heat transfer rate of superheater SH<sub>IHP</sub> the parameters of the flue gas exiting SH<sub>IHP</sub> at point C must be recalculated.

**Actual enthalpy of the flue gas at point C:**

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

**Actual temperature of the flue gas at point C:**

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

**4.3 Design of the High Pressure Evaporator EV<sub>HP</sub>**

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

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

Table 4.8 Parameters necessary for evaporator EV<sub>HP</sub> design, not including parameters describing the geometry of EV<sub>HP</sub>

### 4.3.1 Geometry of the high pressure evaporator EV<sub>HP</sub>

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

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

Table 4.9 Parameters selected for the finned tubes used in the high pressure evaporator EV<sub>HP</sub>

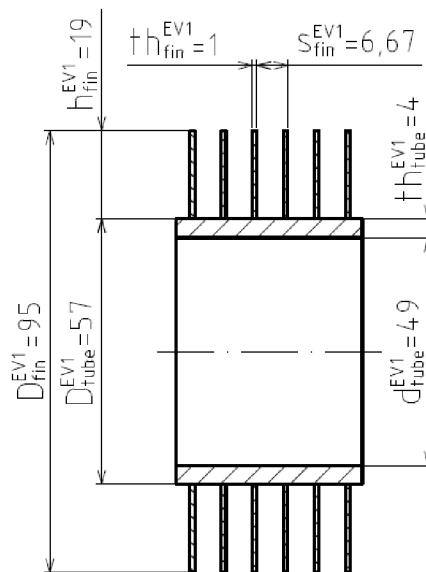


Figure 4-5 Tube geometry  $EV_{HP}$

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

**Chosen layout dimensions:**

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

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

**Lateral tube spacing in evaporator  $EV_{HP}$ :**

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

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

Naturally this value must be rounded to the nearest whole number

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

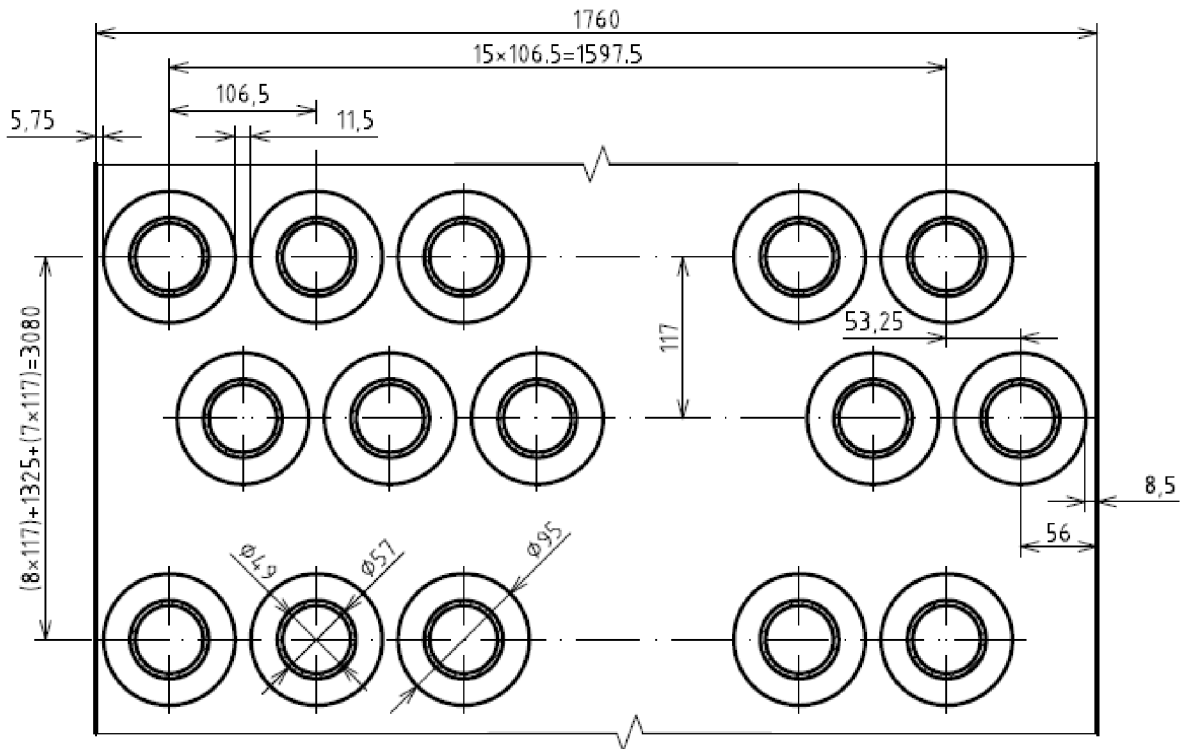
$$n_{tube/r}^{EV1} = 16$$

**Gap verification:**

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

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

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


 Figure 4-6 Tube layout  $EV_{HP}$ 

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

#### Recalculated flue gas enthalpy and temperature at point D:

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

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

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

#### Average flue gas temperature:

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

#### The actual volumetric flow rate of the flue gas:

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

#### The cross sectional area that the flue gas flows through:

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

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

**Speed of the flue gas passing through EV<sub>HP</sub>:**

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

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

**4.3.3 Reduced heat transfer coefficient outside the tubes of EV<sub>HP</sub>, through the flue gas****The coefficient of heat transfer through convection outside the finned tubes:**

Lateral tube spacing relative to the outer tube diameter:

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

Diagonal tube spacing relative to the outer tube diameter:

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

Coefficient of the relative tube spacing:

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

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

Flue gas Parameters:

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

$$\text{Temperature} \quad \bar{t}_{(CD)} = 350.13 \text{ }^{\circ}\text{C}$$

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

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

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

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

$$c_z^{EV1} = 1.01$$

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

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

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

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

### Coefficient characterizing the effectiveness of the fins:

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

Coefficient  $\beta^{EV1}$ :

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

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

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

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

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

Coefficient determined using the graph:

$$E^{EV1} = 0.79$$

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

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

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

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

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

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

**The reduced heat transfer coefficient outside the tubes:**

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

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

#### 4.3.4 Overall heat transfer coefficient for EV<sub>HP</sub>

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

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

#### 4.3.5 Logarithmic mean temperature difference across EV<sub>HP</sub>

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

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

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

#### 4.3.6 The number of lateral rows of tubes required for EV<sub>HP</sub>

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

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

**The outer surface area of one fin:**

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

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

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

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

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

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

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

**The number of lateral rows of tubers in EV<sub>HP</sub>:**

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

Naturally tis value must be rounded to the nearest whole number

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

**4.3.7 Calculating the actual heat transfer rate of EV<sub>HP</sub>****The actual total outer surface area of the heat exchanger tubes:**

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

**The actual heat transfer arte of EV<sub>HP</sub>:**

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

**Error verification:**

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

The discrepancy between the theoretical and the actual heat transfer rate of evaporator EV<sub>HP</sub> is less than 2%, thus the current configuration of the evaporator is acceptable.

**4.3.8 Determining the actual parameters of the flue gas exiting EV<sub>HP</sub>**

Due to the discrepancy between the theoretical and the actual heat transfer rate of evaporator EV<sub>HP</sub> the parameters of the flue gas exiting EV<sub>HP</sub> at point D must be recalculated.



**Actual enthalpy of the flue gas at point D:**

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

**Actual temperature of the flue gas at point D:**

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

**4.4 Design of the High Pressure Economizer ECO<sub>3HP</sub>**

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

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

*Table 4.10 Parameters necessary for economizer ECO<sub>3HP</sub> design, not including parameters describing the geometry of ECO<sub>3HP</sub>*

**4.4.1 Geometry of the high pressure economizer ECO<sub>3HP</sub>**

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

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

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

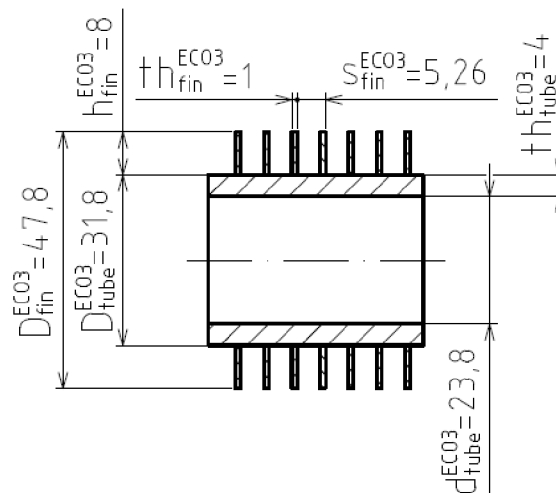


Figure 4-7 Tube geometry  $ECO_{3HP}$

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

#### Chosen layout dimensions:

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

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

**Lateral tube spacing in economizer ECO<sub>3HP</sub>:**

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

**Number of tubes in each lateral row of tubes in ECO<sub>3HP</sub>:**

Naturally this value must be rounded to the nearest whole number

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

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

**Gap verification:**

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

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

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

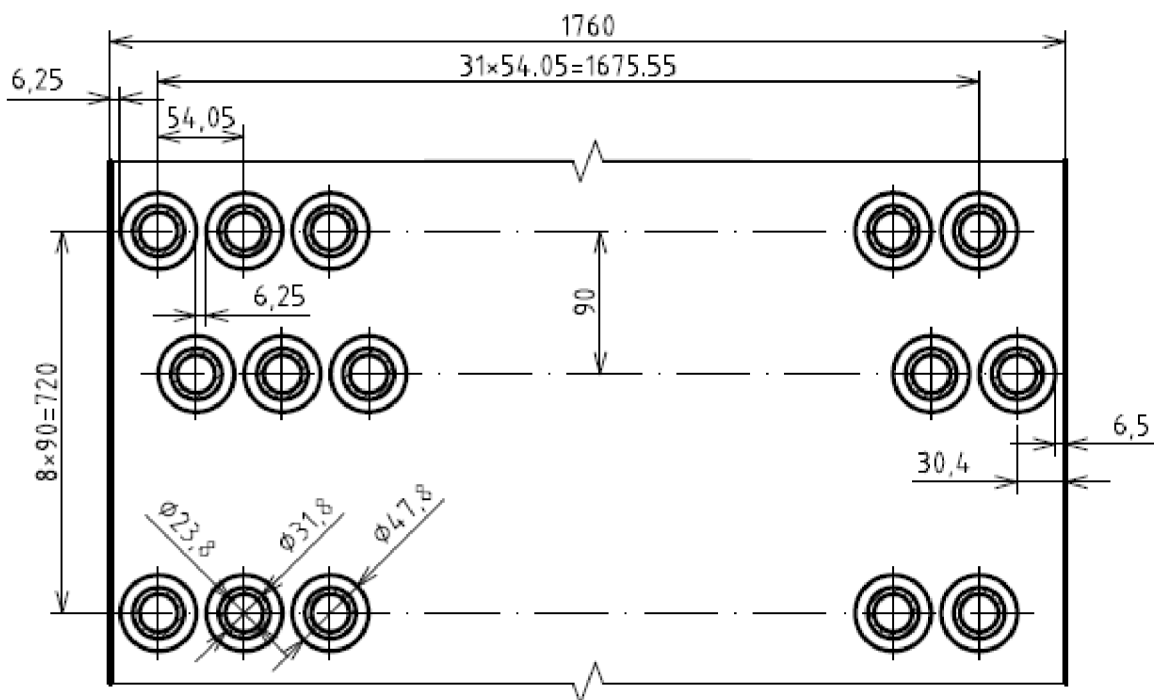


Figure 4-8 Tube layout ECO<sub>3HP</sub>

**4.4.2 Verification of the speed of water passing through economizer ECO<sub>3HP</sub>**

The speed of the water passing through economizer ECO<sub>3HP</sub> is calculated according to average specific volume of the water. The specific volume of the water is determined through X-Steam according to the average pressure and temperature of the water passing through ECO<sub>3HP</sub>.

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

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

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

**Speed of water, before splitting economizer into sections:**

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

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

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

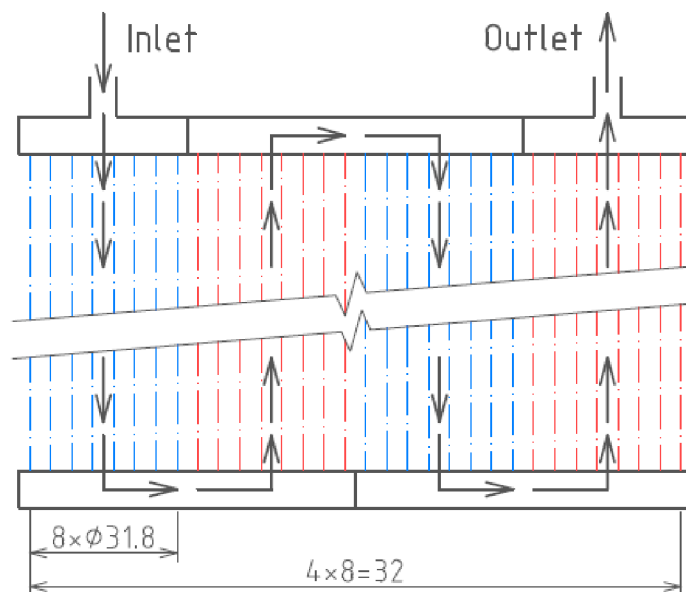


Figure 4-9 Splitting schematic  $ECO_{3HP}$

**Actual speed of water flowing through  $ECO_{3HP}$ :**

$$W_{Steam}^{ECO3} = W_{Steam}^{ECO3'} \cdot n_{section}^{ECO3} = 0.25 \cdot 4 = 1 \text{ m/s} \quad (4.20)$$

#### 4.4.3 Verification of the speed of flue gas passing through economizer ECO<sub>3HP</sub>

##### Recalculated flue gas enthalpy and temperature at point E:

The parameters of the flue gas exiting the economizer at point E are recalculated according to the actual flue gas parameter entering ECO<sub>3HP</sub>.

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

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

##### Average flue gas temperature:

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

##### The actual volumetric flow rate of the flue gas:

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

##### The cross sectional area that the flue gas flows through:

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

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

##### Speed of the flue gas passing through ECO<sub>3HP</sub>:

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

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

#### 4.4.4 Reduced heat transfer coefficient outside the tubes of ECO<sub>3HP</sub>, through the flue gas

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

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{ECO3} = \frac{S_1(ECO3)}{D_{tube}^{ECO3}} = \frac{0.05405}{0.0318} = 1.7$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2^{ECO3} = \frac{S_1^{ECO3}}{D_{tube}^{ECO3}} = \frac{\sqrt{\left(\frac{S_1(ECO3)}{2}\right)^2 + (S_2(ECO3))^2}}{D_{tube}^{ECO3}} = \frac{\sqrt{\left(\frac{0.05405}{2}\right)^2 + 0.09^2}}{0.0318} = 2.96$$

Coefficient of the relative tube spacing:

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

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

Flue gas Parameters:

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

$$\text{Temperature} \quad \bar{t}_{(DE)} = 249.49 \text{ }^{\circ}\text{C}$$

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

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

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

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

$$c_z^{ECO3} = 1$$

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

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

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

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

**Coefficient characterizing the effectiveness of the fins:**

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

Coefficient  $\beta^{ECO3}$ :

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

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

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

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

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

Coefficient determined using the graph:

$$E^{ECO3} = 0.94$$

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

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

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

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

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

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

### The reduced heat transfer coefficient outside the tubes:

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

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

#### 4.4.5 Overall heat transfer coefficient for ECO<sub>3HP</sub>

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

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

#### 4.4.6 Logarithmic mean temperature difference across ECO<sub>3HP</sub>

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

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

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

#### 4.4.7 The number of lateral rows of tubes required for ECO<sub>3HP</sub>

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

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

The outer surface area of one fin:

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

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

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

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

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

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

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



**The number of lateral rows of tubers in ECO<sub>3HP</sub>:**

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

Naturally tis value must be rounded to the nearest whole number

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

**4.4.8 Calculating the actual heat transfer rate of ECO<sub>3HP</sub>****The actual total outer surface area of the heat exchanger tubes:**

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

**The actual heat transfer rate of ECO<sub>3HP</sub>:**

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

**Error verification:**

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

The discrepancy between the theoretical and the actual heat transfer rate of economizer ECO<sub>3HP</sub> is less than 2% , thus the current configuration of ECO<sub>3HP</sub> is acceptable.

**4.4.9 Determining the actual parameters of the flue gas exiting ECO<sub>3HP</sub>**

Due to the discrepancy between the theoretical and the actual heat transfer rate of economizer ECO<sub>3HP</sub> the parameters of the flue gas exiting ECO<sub>3HP</sub> at point E must be recalculated.

**Actual enthalpy of the flue gas at point E:**

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

**Actual temperature of the flue gas at point E:**

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

**4.5 Design of the Low Pressure Superheater SH<sub>LP</sub>**

The low pressure superheater SH<sub>LP</sub> will be designed with smooth tubes, because of the low rate of heat transfer required of this heat exchanger.

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

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

| Values used in calculations<br>(heat exchangers with smooth tubes) | Symbol     | Value | Units |
|--|------------|-------|-------|
| Coefficient of thermal effectiveness                               | $\Psi$     | 0.85  | [-]   |
| Coefficient of the degree to which the flue gas is utilized        | $\xi$      | 1     | [-]   |
| Coefficient of emissivity of the tube walls                        | $a_{wall}$ | 0.8   | [-]   |
| Flue gas pressure (assuming atmospheric)                           | $p_1$      | 0.1   | [MPa] |
| Given temperature difference (for gaseous fuels)                   | $\Delta t$ | 25    | [°C]  |

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

**Overall coefficient of heat transfer:**

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

Where:

- $\Psi$  is the coefficient of thermal effectiveness [-]
- $\alpha_{out}$  is the coefficient of heat transfer outside the tubes, between the flue gas and the heat exchanger tubes [W/(m<sup>2</sup>K)]
- $\alpha_{in}$  is the coefficient of heat transfer inside the tubes, between the heat exchanger tubes and the water/steam inside them [W/(m<sup>2</sup>K)]

The coefficient of heat transfer inside the tubes:

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

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

**The coefficient of heat transfer outside the heat exchanger tubes:**

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

Where:

- $\xi$  is the coefficient characterizing the degree to which the flue gas is utilized [-]
- $\alpha_{c1}$  is the coefficient of heat transfer through convection outside the smooth tubes of the heat exchanger [W/(m<sup>2</sup>K)]
- $\alpha_r$  is the coefficient of heat transfer through radiation outside the tubes of the heat exchanger [W/(m<sup>2</sup>K)]

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

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

Where:

- $c_s$  is a correction coefficient characterizing the layout of the heat exchanger tubes [-]
- $c_{z1}$  is a correction coefficient depending on the number of lateral rows of tubes [-]
- $Pr_{Flue}$  is the Prandtl number of the flue gas [-]

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

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

$$c_s = 0.34 \cdot \varphi_\sigma^{0.1} \quad (4.25)$$

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

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

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

**The coefficient of heat transfer through radiation outside the tubes:**

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

Where:

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

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

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

**The absolute temperature of the outer tube wall:**

$$T_{wall} = \bar{t}_{Steam} + \Delta t + 273.15 \quad (4.28)$$

Where:

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

$\Delta t$  is a given temperature difference [°C]

**The coefficient of emissivity of the flue gas:**

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

Where:

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

$s$  is the effective radiation layer thickness [m]

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

**The effective radiation layer thickness:**

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

**The coefficient of radiation decline:**

$$b = b_{tri} \cdot x_{tri} \quad (4.31)$$

Where:

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

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

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

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

Where:

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

**The concentration of triatomic gases in the flue gas:**

$$x_{tri} = x_{H2O} \cdot x_{CO2} \quad (4.33)$$

**The partial pressure of triatomic gases in the flue gas:**

$$p_{p:tri} = p_1 \cdot x_{tri} \quad (4.34)$$

**Flue gas parameters:**

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

| $\bar{t}_{Flue}$ [°C] | Moisture concentration $X_{H_2O}$ [%] |       |      |
|-----------------------|---------------------------------------|-------|------|
|                       | 5                                     | 7.8   | 10   |
| 0                     | 0.69                                  | 0.707 | 0.72 |
| 100                   | 0.67                                  | 0.681 | 0.69 |
| 200                   | 0.65                                  | 0.661 | 0.67 |
| 300                   | 0.63                                  | 0.641 | 0.65 |
| 400                   | 0.62                                  | 0.631 | 0.64 |
| 500                   | 0.61                                  | 0.621 | 0.63 |

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

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

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

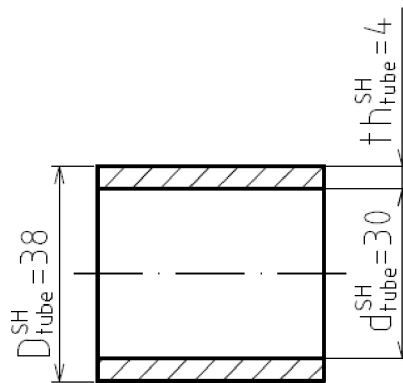
Table 4.14 Parameters necessary for superheater  $SH_{LP}$  design, not including parameters describing the geometry of  $SH_{LP}$

### 4.5.1 Geometry of the high pressure superheater SH<sub>LP</sub>

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

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

*Table 4.15 Parameters selected for the smooth tubes used in the high pressure superheater SH<sub>LP</sub>*



*Figure 4-10 Tube geometry SH<sub>LP</sub>*

The dimensions describing the layout of the finned tubes of superheater SH<sub>LP</sub> are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the superheater.

**Chosen layout dimensions:**

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

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

**Lateral tube spacing in superheater SH<sub>LP</sub>:**

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

### Number of tubes in each lateral row of tubes in SH<sub>LP</sub>:

Naturally this value must be rounded to the nearest whole number

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

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

### Gap verification:

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

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

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

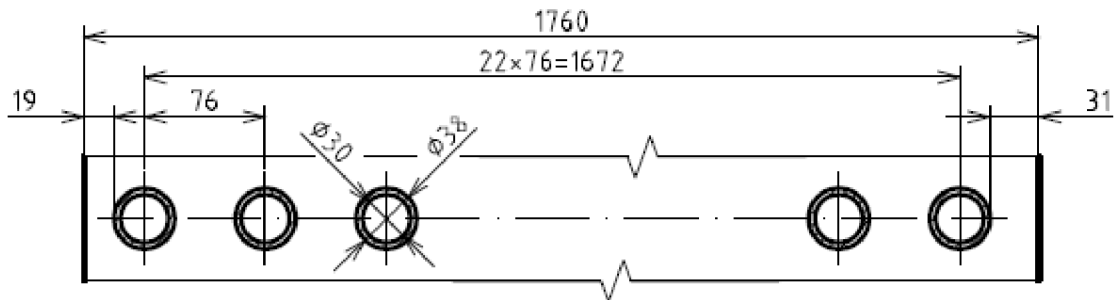


Figure 4-11 Tube layout SH<sub>LP</sub>

### 4.5.2 Verification of the speed of steam passing through superheater SH<sub>LP</sub>

The speed of the steam passing through superheater SH<sub>LP</sub> is calculated according to average specific volume of the steam. The specific volume of the steam is determined through X-Steam according to the average pressure and temperature of the steam passing through SH<sub>LP</sub>.

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

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

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

### Speed of steam:

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

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

### 4.5.3 Verification of the speed of flue gas passing through superheater SH<sub>LP</sub>

#### Recalculated flue gas enthalpy and temperature at point F:

The parameters of the flue gas exiting the superheater at point F are recalculated according to the actual flue gas parameter entering SH<sub>LP</sub>.

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

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

#### Average flue gas temperature:

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

#### The actual volumetric flow rate of the flue gas:

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

#### The cross sectional area that the flue gas flows through:

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

#### Speed of the flue gas passing through SH<sub>LP</sub>:

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

### 4.5.4 Coefficient of heat transfer outside the tubes of SH<sub>LP</sub>

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

Lateral tube spacing relative to the outer tube diameter:

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

Diagonal tube spacing relative to the outer tube diameter:

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

Coefficient of the relative tube spacing:

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



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

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

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

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

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

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

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

Flue gas Parameters:

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

$$\text{Temperature} \quad \bar{t}_{(EF)} = 226.12^{\circ}\text{C}$$

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

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

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

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

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

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

**The coefficient of heat transfer through radiation outside the tubes of the heat exchanger:**

The effective radiation layer thickness:

$$s = 0.9 \cdot D_{tube}^{SH} \cdot \left( \frac{4}{\pi} \cdot \frac{s_1^{SH} \cdot s_2^{SH}}{D_{tube}^{SH}{}^2} - 1 \right) = 0.9 \cdot 0.038 \cdot \left( \frac{4}{\pi} \cdot \frac{0.076 \cdot 0.117}{0.038^2} - 1 \right) = 0.23\text{m}$$

The partial pressure of triatomic gases in the flue gas:

$$p_{p:tri} = p_1 \cdot x_{tri} = 0.1 \cdot 0.122 = 0.0122 \text{ MPa}$$

The absolute flue gas temperature:

$$T_{Flue}^{SH} = \bar{t}_{(EF)} + 273.15 = 226.12 + 273.15 = 499.27 \text{ K}$$

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

$$b_{tri}^{SH} = \left( \frac{7.8 + 16 \cdot x_{H_2O}}{3.16 \cdot \sqrt{p_{p:tri} \cdot s_{SH}}} - 1 \right) \cdot \left( 1 - 0.37 \cdot \frac{T_{Flue}^{SH}}{1000} \right)$$

$$b_{tri}^{SH} = \left( \frac{7.8 + 16 \cdot 0.078}{3.16 \cdot \sqrt{0.0122 \cdot 0.23}} - 1 \right) \left( 1 - 0.37 \cdot \frac{499.27}{1000} \right) = 37.14 \text{ m}^{-1} \cdot \text{MPa}^{-1}$$

The concentration of triatomic gases in the flue gas:

$$x_{tri} = x_{H_2O} \cdot x_{CO_2} = 0.078 + 0.044 = 0.122$$

The coefficient of radiation decline:

$$b = b_{tri}^{SH} \cdot x_{tri} = 37.14 \cdot 0.122 = 4.53 \text{ m}^{-1} \cdot \text{MPa}^{-1}$$

The coefficient of emissivity of the flue gas:

$$a_{Flue}^{SH} = 1 - e^{-b \cdot p_1 \cdot s} = 1 - e^{-4.53 \cdot 0.1 \cdot 0.23} = 0.0991$$

The absolute temperature of the tarnished outer tube walls:

$$T_{wall}^{SH} = \bar{t}_{SH} + 25 + 273.15 = 461.23 \text{ K}$$

The coefficient of heat transfer through radiation outside the tubes:

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

$$\alpha_r^{SH} = 5.7 \cdot 10^{-8} \cdot \frac{0.8 + 1}{2} \cdot 0.0991 \cdot 461.23^3 \cdot \frac{1 - \left( \frac{461.23}{499.27} \right)^{3.6}}{1 - \frac{461.23}{499.27}} = 2.06 \text{ W}/(\text{m}^2\text{K})$$

**The coefficient of heat transfer outside the heat exchanger tubes:**

$$\alpha_{out}^{SH} = \xi \cdot (\alpha_{c1}^{SH} + \alpha_r^{SH}) = 1 \cdot (39.11 + 2.06) = 41.17 \text{ W}/(\text{m}^2\text{K})$$

#### 4.5.5 Coefficient of heat transfer inside the tubes of SH<sub>LP</sub>

##### Steam parameters:

The kinematic viscosity, thermal conductivity, and Prandtl number of the steam passing through SH<sub>LP</sub> are determined through X-Steam according to the average temperature and pressure of the steam. The kinematic viscosity is calculated from the dynamic viscosity and the steam density. The average temperature and pressure of the steam have been determined in earlier calculations regarding steam speed verification.

$$\text{Temperature} \quad \bar{t}_{SH} = 163.08 \text{ }^\circ\text{C}$$

$$\text{Pressure} \quad \bar{p}_{SH} = 0.51 \text{ MPa}$$

$$\rho_{Steam}^{SH} = 2.6384 \text{ kg/m}^3$$

$$\mu_{Steam}^{SH} = 1.4515 \cdot 10^{-5} \text{ Pa} \cdot \text{s}$$

$$\lambda_{Steam}^{SH} = 0.0316 \text{ W/(m} \cdot \text{K)}$$

$$Pr_{Steam}^{SH} = 1.0567$$

##### Kinematic viscosity of the steam:

$$v_{Steam}^{SH} = \frac{\mu_{Steam}^{SH}}{\rho_{Steam}^{SH}} = \frac{1.4515 \cdot 10^{-5}}{2.6384} = 5.5015 \cdot 10^{-6} \text{ m}^2/\text{s}$$

##### The heat transfer coefficient inside the tubes of SH<sub>2HP</sub>:

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

$$\alpha_{in}^{SH} = 0.023 \cdot \frac{0.0316}{0.03} \cdot \left( \frac{19.54 \cdot 0.03}{5.5015 \cdot 10^{-6}} \right)^{0.8} \cdot 1.0567^{0.4} \cdot 1 \cdot 1 \cdot 1 = 260.92 \text{ W/(m}^2\text{K)}$$

#### 4.5.6 Overall heat transfer coefficient for SH<sub>LP</sub>

The overall heat transfer coefficient is calculated according to equation (4.21).

$$k^{SH} = \frac{\Psi \cdot \alpha_{out}^{SH}}{1 + \frac{\alpha_{out}^{SH}}{\alpha_{in}^{SH}}} = \frac{0.85 \cdot 41.17}{1 + \frac{41.17}{260.92}} = 30.22 \text{ W/(m}^2\text{K)}$$

#### 4.5.7 Logarithmic mean temperature difference across SH<sub>LP</sub>

$$\Delta t_1^{SH} = t_E^{real} - t_1^{LP} = 226.66 - 170 = 56.66 \text{ K}$$

$$\Delta t_2^{SH} = t_F' - t_2^{LP} = 225.57 - 156.15 = 69.42 \text{ K}$$

$$\Delta t_{ln}^{SH} = \frac{\Delta t_2^{SH} - \Delta t_1^{SH}}{\ln\left(\frac{\Delta t_2^{SH}}{\Delta t_1^{SH}}\right)} = \frac{69.42 - 56.66}{\ln\left(\frac{69.42}{56.66}\right)} = 62.83 \text{ K}$$

#### 4.5.8 The number of lateral rows of tubes required for SH<sub>LP</sub>

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

$$S_{out}^{SH} = \frac{Q_{SH}^{HP}}{k^{SH} \cdot \Delta t_{ln}^{SH}} = \frac{35.16}{30.22 \cdot 62.83} = 18.52 \text{ m}^2$$

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

$$S_{out/r}^{SH} = H \cdot \pi \cdot D_{tube}^{SH} \cdot n_{tube/r}^{SH} = 6.78 \cdot \pi \cdot 0.038 \cdot 23 = 18.62 \text{ m}^2$$

The number of lateral rows of tubers in SH<sub>LP</sub>:

$$n_{row}^{SH} = \frac{S_{out}^{SH}}{S_{out/r}^{SH}} = \frac{18.52}{18.62} = 0.99$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{SH} = 1$$

#### 4.5.9 Calculating the actual heat transfer rate of SH<sub>LP</sub>

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

$$S_{out}^{SH:real} = S_{out/r}^{SH} \cdot n_{row}^{SH} = 18.62 \cdot 1 = 18.62 \text{ m}^2$$

The actual heat transfer of SH<sub>LP</sub>:

$$Q_{SH}^{LP:real} = k^{SH} \cdot S_{out}^{SH:real} \cdot \Delta t_{ln}^{SH} = 30.22 \cdot 18.62 \cdot 62.83 = 35.35 \text{ kW}$$

**Error verification:**

$$\%Error^{SH} = \left| \left( \frac{35.35}{35.16} - 1 \right) \cdot 100 \right| = 0.53\%$$

The discrepancy between the theoretical and the actual heat transfer rate of superheater SH<sub>LP</sub> is less than 2% , thus the current configuration of SH<sub>LP</sub> is acceptable.

#### 4.5.10 Determining the actual parameters of the flue gas exiting SH<sub>LP</sub>

Due to the discrepancy between the theoretical and the actual heat transfer rate of superheater SH<sub>LP</sub> the parameters of the flue gas exiting SH<sub>LP</sub> at point F must be recalculated.

**Actual enthalpy of the flue gas at point F:**

$$I_F^{real} = I_E^{real} - \frac{Q_{SH}^{LP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 367.64 - \frac{35.35}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 303.27 \text{ kJ/Nm}^3$$

**Actual temperature of the flue gas at point F:**

$$t_F^{real} = \frac{I_F^{real} - I_{Flue:200}}{I_{Flue:300} - I_{Flue:200}} \cdot (300 - 200) + 200 = \frac{303.27 - 268.01}{405.95 - 268.01} \cdot 100 + 200 = 225.57 \text{ }^\circ\text{C}$$

**4.6 Design of the Low Pressure Evaporator EV<sub>LP</sub>**

Several parameters are required in the calculations associated with the design of the low pressure evaporator EV<sub>LP</sub>. Parameters, other than those describing the geometry of the evaporator, that are necessary for the calculations regarding the design and sizing of evaporator EV<sub>LP</sub>, some of which were determined in previous calculations, are organized in *Table 4.16* below.

|             | Parameters  | Symbol        | Value   | Units                 |
|-------------|---|---------------|---------|-----------------------|
|             | Heat transfer rate required of EV <sub>LP</sub> (theoretical value) | $Q_{EV}^{LP}$ | 1866.54 | [kW]                  |
| Flue        | Actual temperature of flue gas entering into EV <sub>LP</sub>       | $t_f^{real}$  | 225.57  | [°C]                  |
|             | Actual enthalpy of flue gas entering into EV <sub>LP</sub>          | $I_f^{real}$  | 303.27  | [kJ/Nm <sup>3</sup> ] |
| Water/steam | Temperature of water entering EV <sub>LP</sub>                      | $t_4^{LP}$    | 151.15  | [°C]                  |
|             | Pressure of water entering EV <sub>LP</sub>                         | $p_4^{LP}$    | 0.56    | [MPa]                 |
|             | Temperature of steam exiting EV <sub>LP</sub>                       | $t_2^{LP}$    | 156.15  | [°C]                  |
|             | Pressure of steam exiting EV <sub>LP</sub>                          | $p_2^{LP}$    | 0.56    | [MPa]                 |

*Table 4.16 Parameters necessary for evaporator EV<sub>LP</sub> design, not including parameters describing the geometry of EV<sub>LP</sub>*

**4.6.1 Geometry of the high pressure evaporator EV<sub>LP</sub>**

The chosen dimensions of the finned tubes used in the low pressure evaporator (EV<sub>LP</sub>) are shown in *Table 4.17* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.17* are selected and later on adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

|      | Parameter                | Symbol           | Value | Units |
|------|--------------------------|------------------|-------|-------|
| tube | Outer tube diameter      | $D_{tube}^{EV}$  | 57    | [mm]  |
|      | Tube wall thickness      | $th_{tube}^{EV}$ | 4     | [mm]  |
|      | Inner tube diameter      | $d_{tube}^{EV}$  | 49    | [mm]  |
| fins | Fin thickness            | $th_{fin}^{EV}$  | 1     | [mm]  |
|      | Number of fins per meter | $n_{fin}^{EV}$   | 230   | [mm]  |
|      | Fin spacing              | $s_{fin}^{EV}$   | 4.35  | [mm]  |
|      | Fin height               | $h_{fin}^{EV}$   | 19    | [mm]  |
|      | Outer fin diameter       | $D_{fin}^{EV}$   | 95    | [mm]  |

Table 4.17 Parameters selected for the finned tubes used in the high pressure evaporator  $EV_{LP}$

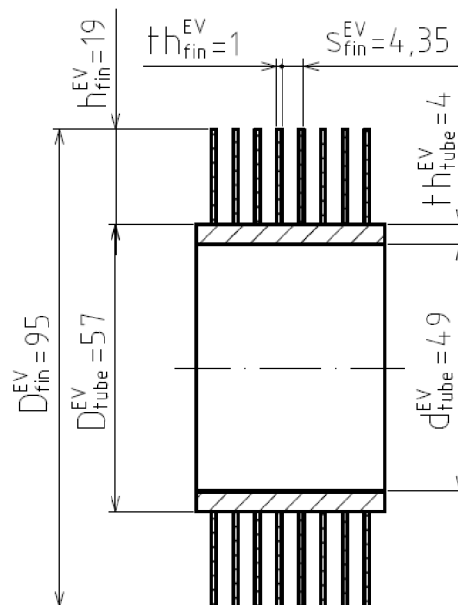


Figure 4-12 Tube geometry  $EV_{LP}$

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

#### Chosen layout dimensions:

$$\text{Lateral gap between tubes} \quad a_{EV} = 5.5 \text{ mm}$$

$$\text{Longitudinal tube spacing} \quad s_{2(EV)} = 117 \text{ mm}$$

#### Lateral tube spacing in evaporator $EV_{HP}$ :

$$s_{1(EV)} = D_{tube}^{EV} + 2 \cdot h_{fin}^{EV} + a_{EV} = 0.057 + 2 \cdot 0.019 + 0.0055 = 0.1005 \text{ m}$$

### Number of tubes in each lateral row of tubes in $EV_{LP}$ :

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{EV} = \frac{L}{s_1(EV)} - 0.5 = \frac{1.76}{0.1005} - 0.5 = 17.01$$

$$n_{tube/r}^{EV} = 17$$

### Gap verification:

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

$$gap = L - n_{tube/r}^{EV} \cdot s_1(EVH) - \frac{D_{tube}^{EV}}{2} - h_{fin}^{EV}$$

$$gap = 1.76 - 17 \cdot 0.1005 - \frac{0.057}{2} - 0.019 = 0.004 \text{ m}$$

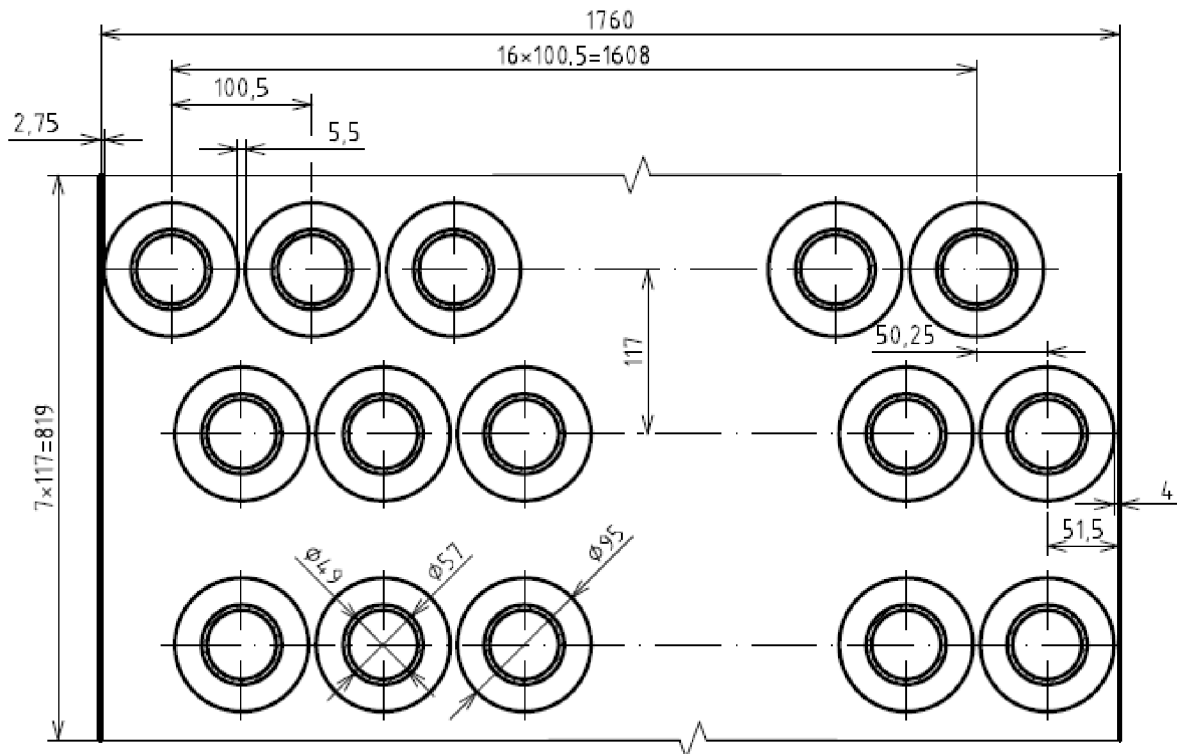


Figure 4-13 Tube layout  $EV_{LP}$

### 4.6.2 Verification of the speed of flue gas passing through evaporator $EV_{LP}$

#### Recalculated flue gas enthalpy and temperature at point G:

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

$$I'_G = I_F^{real} - \frac{Q_{EV}^{LP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 225.57 - \frac{1866.54}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 223.41 \text{ kJ/Nm}^3$$

$$t'_G = \frac{I'_F - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{223.41 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 166.9^\circ\text{C}$$

**Average flue gas temperature:**

$$\bar{t}_{(FG)} = \frac{t_F^{real} + t'_G}{2} = \frac{225.57 + 166.9}{2} = 196.24^\circ\text{C}$$

**The actual volumetric flow rate of the flue gas:**

$$\dot{M}_{VFlue}^{EV} = \frac{273.15 + \bar{t}_{(FG)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 196.24}{273.15} \cdot 23.49 = 40.36 \text{ m}^3/\text{s}$$

**The cross sectional area that the flue gas flows through:**

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

$$A_{duct}^{EV} = 6.78 \cdot 1.76 - 6.78 \cdot 0.057 \cdot 17 - 6.78 \cdot 2 \cdot 0.019 \cdot 0.001 \cdot 230 \cdot 17 = 4.36 \text{ m}^2$$

**Speed of the flue gas passing through EV<sub>HP</sub>:**

$$W_{Flue}^{EV} = \frac{\dot{M}_{VFlue}^{EV}}{A_{duct}^{EV}} = \frac{40.36}{4.36} = 9.27 \text{ m/s}$$

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

#### 4.6.3 Reduced heat transfer coefficient outside the tubes of EV<sub>LP</sub>, through the flue gas

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

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{EV} = \frac{s_{1(EV)}}{D_{tube}^{EV}} = \frac{0.1005}{0.057} = 1.76$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{EV} = \frac{s'^{EV}}{D_{tube}^{EV}} = \frac{\sqrt{\left(\frac{s_{1(EV)}}{2}\right)^2 + (s_{2(EV)})^2}}{D_{tube}^{EV}} = \frac{\sqrt{\left(\frac{0.1005}{2}\right)^2 + 0.117^2}}{0.057} = 2.23$$

Coefficient of the relative tube spacing:

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



$$\varphi_{\sigma}^{EV} = \frac{\sigma_1^{EV} - 1}{\sigma_2'^{EV} - 1} = \frac{1.76 - 1}{2.23 - 1} = 0.62$$

Flue gas Parameters:

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

$$\text{Temperature} \quad \bar{t}_{(EF)} = 196.24 \text{ }^{\circ}\text{C}$$

$$\lambda_{Flue}^{EV} = \left[ \frac{196.24 - 100}{100} \cdot (39.23 - 30.89) + 30.89 \right] \cdot 10^{-3} = 3.891 \cdot 10^{-2} \text{ W/(m} \cdot \text{K)}$$

$$\nu_{Flue}^{EV} = \left[ \frac{196.24 - 100}{100} \cdot (32.36 - 21.41) + 21.41 \right] \cdot 10^{-6} = 3.195 \cdot 10^{-5} \text{ m}^2/\text{s}$$

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

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 8 lateral rows of tubes in evaporator  $EV_{LP}$ .

$$c_z^{EV} = 0.99$$

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

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

$$\alpha_c^{EV} = 0.23 \cdot 0.99 \cdot 0.62^{0.2} \cdot \frac{0.03891}{0.00435} \cdot \left( \frac{0.057}{0.00435} \right)^{-0.54} \cdot \left( \frac{0.019}{0.00435} \right)^{-0.14} \cdot \left( \frac{9.27 \cdot 0.00435}{3.195 \cdot 10^{-5}} \right)^{0.65}$$

$$\alpha_c^{EV} = 38.89 \text{ W/(m}^2\text{K)}$$

**Coefficient characterizing the effectiveness of the fins:**

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

Coefficient  $\beta^{EV}$ :

$$\beta^{EV} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{EV}}{th_{fin}^{EV} \cdot \lambda_{fin} \cdot (1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{EV})}} = \sqrt{\frac{2 \cdot 0.85 \cdot 38.89}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 38.89)}}$$

$$\beta^{EV} = 39.03 \text{ m}^{-1}$$

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

$$\beta^{EV} \cdot h_{fin}^{EV} = 39.03 \cdot 0.019 = 0.74$$

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

Coefficient determined using the graph:

$$E^{EV} = 0.82$$

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

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

$$\frac{S_{fin}^{EV}}{S_{out}^{EV}} = \frac{\left(\frac{D_{fin}^{EV}}{D_{tube}^{EV}}\right)^2 - 1}{\left(\frac{D_{fin}^{EV}}{D_{tube}^{EV}}\right)^2 - 1 + 2 \cdot \left(\frac{S_{fin}^{EV}}{D_{tube}^{EV}} - \frac{th_{fin}^{EV}}{D_{tube}^{EV}}\right)} = \frac{\left(\frac{0.095}{0.057}\right)^2 - 1}{\left(\frac{0.095}{0.057}\right)^2 - 1 + 2 \cdot \left(\frac{0.00435}{0.057} - \frac{0.001}{0.057}\right)}$$

$$\frac{S_{fin}^{EV}}{S_{out}^{EV}} = 0.94$$

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

$$\frac{S_{out-fin}^{EV}}{S_{out}^{EV}} = 1 - \frac{S_{fin}^{EV}}{S_{out}^{EV}} = 1 - 0.94 = 0.06$$

### The reduced heat transfer coefficient outside the tubes:

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

$$\alpha_{r:out}^{EV} = (0.94 \cdot 0.82 \cdot 1 + 0.06) \cdot \frac{0.85 \cdot 38.89}{1 + 0.0043 \cdot 0.85 \cdot 38.89} = 23.97 \text{ W}/(\text{m}^2\text{K})$$

#### 4.6.4 Overall heat transfer coefficient for EV<sub>LP</sub>

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

$$k^{EV} = \alpha_{r:out}^{EV} = 23.97 \text{ W}/(\text{m}^2\text{K})$$

#### 4.6.5 Logarithmic mean temperature difference across $EV_{LP}$

$$\Delta t_1^{EV} = t_F^{real} - t_2^{LP} = 225.57 - 156.15 = 69.41 \text{ K}$$

$$\Delta t_2^{EV} = t_G' - t_4^{LP} = 166.9 - 151.15 = 15.75 \text{ K}$$

$$\Delta t_{ln}^{EV} = \frac{\Delta t_2^{EV} - \Delta t_1^{EV}}{\ln\left(\frac{\Delta t_2^{EV}}{\Delta t_1^{EV}}\right)} = \frac{15.75 - 69.41}{\ln\left(\frac{15.75}{69.41}\right)} = 36.18 \text{ K}$$

#### 4.6.6 The number of lateral rows of tubes required for $EV_{LP}$

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

$$S_{out}^{EV} = \frac{Q_{EV}^{LP}}{k^{EV} \cdot \Delta t_{ln}^{EV}} = \frac{1866.54}{23.97 \cdot 36.18} = 2152.14 \text{ m}^2$$

The outer surface area of one fin:

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

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

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

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

$$S_{out/1m}^{EV} = \pi \cdot 0.057 \cdot (1 - 230 \cdot 0.001) + 230 \cdot 0.0094 = 2.29 \text{ m}$$

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

$$S_{out/r}^{EV} = S_{out/1m}^{EV} \cdot H \cdot n_{tube/r}^{EV} = 2.29 \cdot 6.78 \cdot 17 = 264.3 \text{ m}^2$$

The number of lateral rows of tubers in  $EV_{LP}$ :

$$n_{row}^{EV} = \frac{S_{out}^{EV}}{S_{out/r}^{EV}} = \frac{2152.14}{264.33} = 8.14$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{EVH} = 8$$

#### 4.6.7 Calculating the actual heat transfer rate of $EV_{LP}$

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

$$S_{out}^{EV:real} = S_{out/r}^{EV} \cdot n_{row}^{EV} = 264.33 \cdot 17 = 2114.61 \text{ m}^2$$

The actual heat transfer rate of  $EV_{LP}$ :

$$Q_{EV}^{LP:real} = k^{EV} \cdot S_{out}^{EV:real} \cdot \Delta t_{ln}^{EV} = 23.97 \cdot 2114.61 \cdot 36.18 = 1833.98 \text{ kW}$$

**Error verification:**

$$\%Error^{EVH} = \left| \left( \frac{1833.98}{1866.54} - 1 \right) \cdot 100 \right| = 1.74\%$$

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

#### 4.6.8 Determining the actual parameters of the flue gas exiting $EV_{LP}$

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

**Actual enthalpy of the flue gas at point G:**

$$I_G^{real} = I_F^{real} - \frac{Q_{EV}^{LP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 303.27 - \frac{1833.98}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 224.81 \text{ kJ/Nm}^3$$

**Actual temperature of the flue gas at point G:**

$$t_G^{real} = \frac{I_F^{real} - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{224.81 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 167.94 \text{ }^\circ\text{C}$$

### 4.7 Design of the High Pressure Economizer $ECO_{2HP}$

Several parameters are required in the calculations associated with the design of the high pressure economizer  $ECO_{2HP}$ . Parameters, other than those describing the geometry of the economizer, that are necessary for the calculations regarding the design and sizing of economizer  $ECO_{2HP}$ , some of which were determined in previous calculations, are organized in *Table 4.18* below.

|       | Parameters  | Symbol                 | Value  | Units                 |
|-------|---|------------------------|--------|-----------------------|
|       | Heat transfer rate required of ECO <sub>2HP</sub><br>(theoretical value)                              | $Q_{ECO_2}^{HP}$       | 158.21 | [kW]                  |
| Flue  | Actual temperature of flue gas entering into<br>ECO <sub>2HP</sub>                                    | $t_G^{real}$           | 167.94 | [°C]                  |
|       | Actual enthalpy of flue gas entering into ECO <sub>2HP</sub>  | $I_G^{real}$           | 224.81 | [kJ/Nm <sup>3</sup> ] |
| water | Mass flow rate of water in the high pressure circuit<br>(only 95% passes through ECO <sub>2HP</sub> ) | $\dot{M}_{Steam}^{HP}$ | 3.24   | [kg/s]                |
|       | Temperature of water entering ECO <sub>2HP</sub>  | $t_8^{HP}$             | 139.4  | [°C]                  |
|       | Pressure of water entering ECO <sub>2HP</sub>   | $p_8^{HP}$             | 5      | [MPa]                 |
|       | Temperature of water exiting ECO <sub>2HP</sub>   | $t_7^{HP}$             | 151.4  | [°C]                  |
|       | Pressure of water exiting ECO <sub>2HP</sub>  | $p_7^{HP}$             | 4.9    | [MPa]                 |

Table 4.18 Parameters necessary for economizer ECO<sub>2HP</sub> design, not including parameters describing the geometry of ECO<sub>2HP</sub>

#### 4.7.1 Geometry of the high pressure economizer ECO<sub>2HP</sub>

The chosen dimensions of the finned tubes used in the high pressure economizer (ECO<sub>2HP</sub>) are shown in Table 4.19 in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in Table 4.19 are selected and later on adjusted in order to supply the economizer with an acceptable water flow speed through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

|      | Parameter                | Symbol              | Value | Units |
|------|--------------------------|---------------------|-------|-------|
| tube | Outer tube diameter      | $D_{tube}^{ECO_2}$  | 33.7  | [mm]  |
|      | Tube wall thickness      | $th_{tube}^{ECO_2}$ | 4     | [mm]  |
|      | Inner tube diameter      | $d_{tube}^{ECO_2}$  | 25.7  | [mm]  |
| fins | Fin thickness            | $th_{fin}^{ECO_2}$  | 1     | [mm]  |
|      | Number of fins per meter | $n_{fin}^{ECO_2}$   | 200   | [mm]  |
|      | Fin spacing              | $s_{fin}^{ECO_2}$   | 5     | [mm]  |
|      | Fin height               | $h_{fin}^{ECO_2}$   | 8     | [mm]  |
|      | Outer fin diameter       | $D_{fin}^{ECO_2}$   | 49.7  | [mm]  |

Table 4.19 Parameters selected for the finned tubes used in the high pressure economizer ECO<sub>2HP</sub>

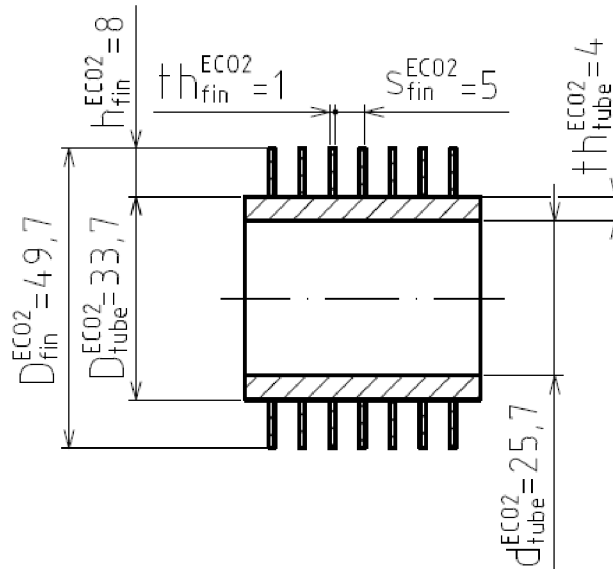


Figure 4-14 Tube geometry  $ECO_{2HP}$

The dimensions describing the layout of the finned tubes of economizer  $ECO_{2HP}$  are chosen and then used together with the flue gas duct dimensions to determine the number of finned tubes in each lateral row of tubes in the economizer.

#### Chosen layout dimensions:

Lateral gap between tubes  $a_{ECO_2} = 4.45 \text{ mm}$

Longitudinal tube spacing  $S_2(ECO_2) = 90 \text{ mm}$

#### Lateral tube spacing in economizer $ECO_{2HP}$ :

$$S_1(ECO_2) = D_{tube}^{ECO_2} + 2 \cdot h_{fin}^{ECO_2} + a_{ECO_2} = 0.0337 + 2 \cdot 0.008 + 0.00445 = 0.05415 \text{ m}$$

#### Number of tubes in each lateral row of tubes in $ECO_{2HP}$ :

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{ECO_2} = \frac{L}{S_1(ECO_2)} - 0.5 = \frac{1.76}{0.05415} - 0.5 = 32.002$$

$$n_{tube/r}^{ECO_2} = 32$$

#### Gap verification:

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

$$gap = L - n_{tube/row}^{ECO2} \cdot s_1^{(ECO2)} - \frac{D_{tube}^{ECO2}}{2} - h_{fin}^{ECO2}$$

$$gap = 1.76 - 32 \cdot 0.05415 - \frac{0.0337}{2} - 0.008 = 0.00235 \text{ m}$$

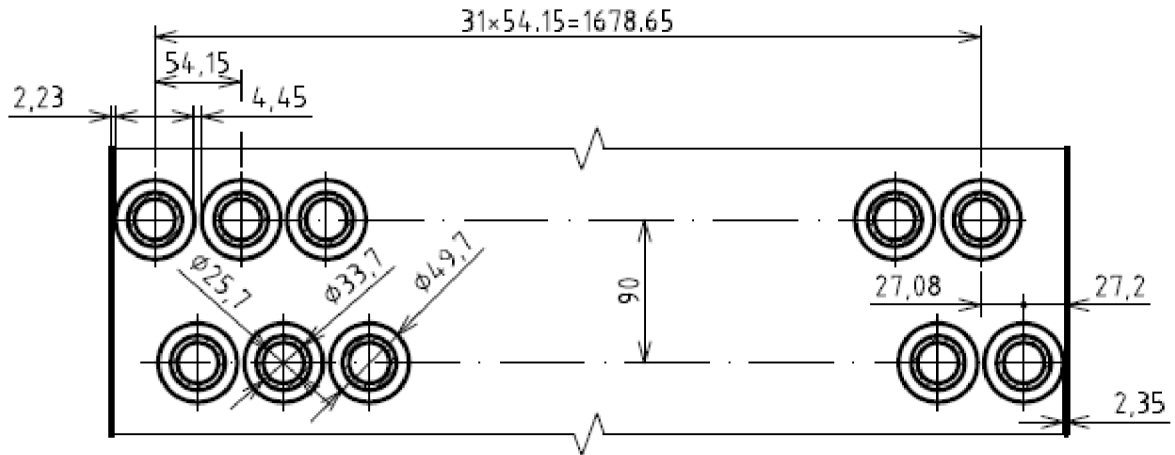


Figure 4-15 Tube layout  $ECO_{2HP}$

#### 4.7.2 Verification of the speed of water passing through economizer $ECO_{2HP}$

The speed of the water passing through economizer  $ECO_{2HP}$  is calculated according to average specific volume of the water. The specific volume of the water is determined through X-Steam according to the average pressure and temperature of the water passing through  $ECO_{2HP}$ .

$$\bar{t}_{ECO2} = \frac{t_7^{HP} + t_8^{HP}}{2} = \frac{151.4 + 139.4}{2} = 145.4 \text{ } ^\circ\text{C}$$

$$\bar{p}_{ECO2} = \frac{p_7^{HP} + p_8^{HP}}{2} = \frac{4.9 + 5}{2} = 4.95 \text{ MPa}$$

$$\bar{v}_{ECO2} = 0.0011 \text{ m}^3/\text{kg} \quad (\text{determined using X-Steam, } f(p,t))$$

**Speed of water, before splitting economizer into sections:**

$$W_{Steam}^{ECO2'} = \frac{0.95 \cdot \dot{M}_{Steam}^{HP} \cdot \bar{v}_{ECO2}}{\frac{\pi \cdot (d_{tube}^{ECO2})^2}{4} \cdot n_{tube/row}^{ECO2}} = \frac{0.95 \cdot 3.24 \cdot 0.0011}{\frac{\pi \cdot 0.0257^2}{4} \cdot 32} = 0.20 \text{ m/s}$$

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

$$\text{Number of sections} \quad n_{section}^{ECO2} = 4$$

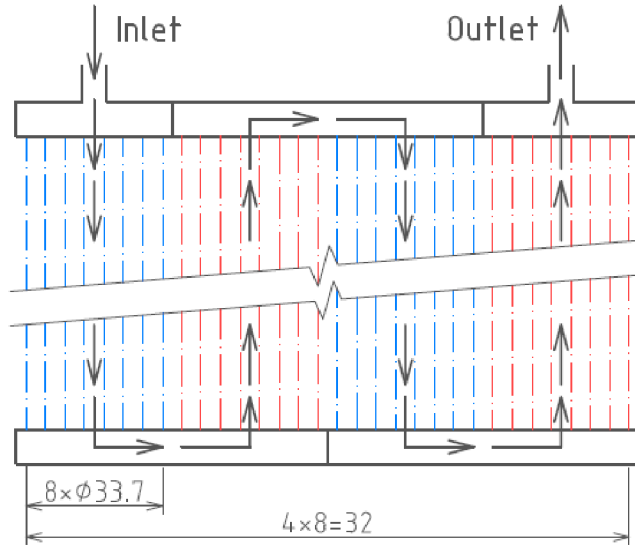


Figure 4-16 Splitting schematic  $ECO_{2HP}$

#### Actual speed of water flowing through $ECO_{2HP}$ :

$$W_{Steam}^{ECO2} = W_{Steam}^{ECO2'} \cdot n_{section}^{ECO2} = 0.20 \cdot 4 = 0.80 \text{ m/s}$$

#### 4.7.3 Verification of the speed of flue gas passing through economizer $ECO_{2HP}$

##### Recalculated flue gas enthalpy and temperature at point H:

The parameters of the flue gas exiting the economizer at point H are recalculated according to the actual flue gas parameter entering  $ECO_{2HP}$ .

$$I'_H = I_G^{real} - \frac{Q_{ECO2}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 224.8 - \frac{158.21}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 218.04 \text{ kJ/Nm}^3$$

$$t'_H = \frac{I'_H - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{218.04 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 162.91^\circ\text{C}$$

##### Average flue gas temperature:

$$\bar{t}_{(GH)} = \frac{t_G^{real} + t'_H}{2} = \frac{167.94 + 162.91}{2} = 165.43^\circ\text{C}$$

##### The actual volumetric flow rate of the flue gas:

$$\dot{M}_{VFlue}^{ECO2} = \frac{273.15 + \bar{t}_{(GH)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 165.43}{273.15} \cdot 23.49 = 37.71 \text{ m}^3/\text{s}$$

##### The cross sectional area that the flue gas flows through:

$$A_{duct}^{ECO2} = H \cdot L - H \cdot D_{tube}^{ECO2} \cdot n_{tube/r}^{ECO2} - H \cdot 2 \cdot h_{fin}^{ECO2} \cdot t h_{fin}^{ECO2} \cdot n_{fin}^{ECO2} \cdot n_{tube/r}^{ECO2}$$



$$A_{duct}^{ECO2} = 6.78 \cdot 1.76 - 6.78 \cdot 0.0337 \cdot 32 - 6.78 \cdot 2 \cdot 0.008 \cdot 0.001 \cdot 200 \cdot 32 = 3.93 \text{ m}^2$$

**Speed of the flue gas passing through ECO<sub>2HP</sub>:**

$$W_{Flue}^{ECO2} = \frac{\dot{M}_{Flue}^{ECO2}}{A_{duct}^{ECO2}} = \frac{37.71}{3.93} = 9.6 \text{ m/s}$$

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

#### 4.7.4 Reduced heat transfer coefficient outside the tubes of ECO<sub>2HP</sub>, through the flue gas

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

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{ECO2} = \frac{S_1^{ECO2}}{D_{tube}^{ECO2}} = \frac{0.05415}{0.0337} = 1.6$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{ECO2} = \frac{S'^{ECO2}}{D_{tube}^{ECO2}} = \frac{\sqrt{\left(\frac{S_1^{ECO2}}{2}\right)^2 + (S_2^{ECO2})^2}}{D_{tube}^{ECO2}} = \frac{\sqrt{\left(\frac{0.05415}{2}\right)^2 + 0.09^2}}{0.0337} = 2.79$$

Coefficient of the relative tube spacing:

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

$$\varphi_{\sigma}^{ECO2} = \frac{\sigma_1^{ECO2} - 1}{\sigma_2'^{ECO2} - 1} = \frac{1.6 - 1}{2.79 - 1} = 0.34$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through ECO<sub>2HP</sub> has been determined in previous calculations regarding the flue gas speed verification.

$$\text{Temperature} \quad \bar{t}_{(GH)} = 165.43 \text{ }^{\circ}\text{C}$$

$$\lambda_{Flue}^{ECO2} = \left[ \frac{165.43 - 100}{100} \cdot (39.228 - 30.89) + 30.89 \right] \cdot 10^{-3} = 3.635 \cdot 10^{-2} \text{ W/(m} \cdot \text{K)}$$

$$\nu_{Flue}^{ECO2} = \left[ \frac{165.43 - 100}{100} \cdot (32.36 - 21.41) + 21.41 \right] \cdot 10^{-6} = 2.857 \cdot 10^{-5} \text{ m}^2/\text{s}$$

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

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 2 lateral rows of tubes in economizer ECO<sub>2HP</sub>.

$$c_z^{ECO2} = 0.87$$

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

$$\alpha_c^{ECO2} = 0.23 \cdot c_z^{ECO2} \cdot (\varphi_\sigma^{ECO2})^{0.2} \cdot \frac{\lambda_{Flue}^{ECO2}}{s_{fin}^{ECO2}} \cdot \left(\frac{D_{tube}^{ECO2}}{s_{fin}^{ECO2}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{ECO2}}{s_{fin}^{ECO2}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{ECO2} \cdot s_{fin}^{ECO2}}{v_{Flue}^{ECO2}}\right)^{0.65}$$

$$\alpha_c^{ECO2} = 0.23 \cdot 0.87 \cdot 0.34^{0.2} \cdot \frac{0.03635}{0.005} \cdot \left(\frac{0.0337}{0.005}\right)^{-0.54} \cdot \left(\frac{0.008}{0.005}\right)^{-0.14} \cdot \left(\frac{9.6 \cdot 0.005}{2.857 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{ECO2} = 48.81 \text{ W}/(m^2K)$$

### Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of  $\beta^{ECO2} \cdot h_{fin}^{ECO2}$  and the quotient of  $D_{fin}^{ECO2} / D_{tube}^{ECO2}$ , where  $\beta^{ECO2}$  is a coefficient.

Coefficient  $\beta^{ECO2}$ :

$$\beta^{ECO2} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{ECO2}}{th_{fin}^{ECO2} \cdot \lambda_{fin} \cdot (1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO2})}} = \sqrt{\frac{2 \cdot 0.85 \cdot 48.81}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 48.81)}}$$

$$\beta^{ECO2} = 43.05 \text{ m}^{-1}$$

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

$$\beta^{ECO2} \cdot h_{fin}^{ECO2} = 43.05 \cdot 0.008 = 0.34$$

$$\frac{D_{fin}^{ECO2}}{D_{tube}^{ECO2}} = \frac{0.0497}{0.0337} = 1.47$$

Coefficient determined using the graph:

$$E^{ECO2} = 0.95$$

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

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

$$\frac{S_{fin}^{ECO2}}{S_{out}^{ECO2}} = \frac{\left(\frac{D_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)^2 - 1}{\left(\frac{D_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)^2 - 1 + 2 \cdot \left(\frac{S_{fin}^{ECO2}}{D_{tube}^{ECO2}} - \frac{th_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)} = \frac{\left(\frac{0.0497}{0.0337}\right)^2 - 1}{\left(\frac{0.0497}{0.0337}\right)^2 - 1 + 2 \cdot \left(\frac{0.005}{0.0337} - \frac{0.001}{0.0337}\right)}$$

$$\frac{S_{fin}^{ECO2}}{S_{out}^{ECO2}} = 0.83$$

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

$$\frac{S_{out-fin}^{ECO2}}{S_{out}^{ECO2}} = 1 - \frac{S_{fin}^{ECO2}}{S_{out}^{ECO2}} = 1 - 0.83 = 0.17$$

**The reduced heat transfer coefficient outside the tubes:**

$$\alpha_{r:out}^{ECO2} = \left(\frac{S_{fin}^{ECO2}}{S_{out}^{ECO2}} \cdot E^{ECO2} \cdot \mu + \frac{S_{out-fin}^{ECO2}}{S_{out}^{ECO2}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{ECO2}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO2}}$$

$$\alpha_{r:out}^{ECO2} = (0.83 \cdot 0.95 \cdot 1 + 0.17) \cdot \frac{0.85 \cdot 48.81}{1 + 0.0043 \cdot 0.85 \cdot 48.81} = 33.8 \text{ W}/(\text{m}^2 \text{ K})$$

#### 4.7.5 Overall heat transfer coefficient for ECO<sub>2HP</sub>

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

$$k^{ECO2} = \alpha_{r:out}^{ECO2} = 33.8 \text{ W}/(\text{m}^2 \text{ K})$$

#### 4.7.6 Logarithmic mean temperature difference across ECO<sub>2HP</sub>

$$\Delta t_1^{ECO2} = t_G^{real} - t_7^{HP} = 167.94 - 151.4 = 16.53 \text{ K}$$

$$\Delta t_2^{ECO2} = t_H' - t_8^{HP} = 162.91 - 139.4 = 23.51 \text{ K}$$

$$\Delta t_{ln}^{ECO2} = \frac{\Delta t_2^{ECO2} - \Delta t_1^{ECO2}}{\ln\left(\frac{\Delta t_2^{ECO2}}{\Delta t_1^{ECO2}}\right)} = \frac{23.51 - 16.53}{\ln\left(\frac{23.51}{16.53}\right)} = 19.82 \text{ K}$$

#### 4.7.7 The number of lateral rows of tubes required for ECO<sub>2</sub>HP

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

$$S_{out}^{ECO_2} = \frac{Q_{ECO_2}^{HP}}{k^{ECO_2} \cdot \Delta t_{ln}^{ECO_2}} = \frac{158.21}{33.8 \cdot 19.82} = 236.18 \text{ m}^2$$

The outer surface area of one fin:

$$S_{1fin}^{ECO_2} = 2 \cdot \pi \cdot \frac{(D_{fin}^{ECO_2})^2 - (D_{tube}^{ECO_2})^2}{4} + \pi \cdot D_{fin}^{ECO_2} \cdot th_{fin}^{ECO_2}$$

$$S_{1fin}^{ECO_2} = 2 \cdot \pi \cdot \frac{0.0497^2 - 0.0337^2}{4} + \pi \cdot 0.0497 \cdot 0.001 = 0.0023 \text{ m}^2$$

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

$$S_{out/1m}^{ECO_2} = \pi \cdot D_{tube}^{ECO_2} \cdot (1 - n_{fin}^{ECO_2} \cdot th_{fin}^{ECO_2}) + n_{fin}^{ECO_2} \cdot S_{1fin}^{ECO_2}$$

$$S_{out/1m}^{ECO_2} = \pi \cdot 0.0337 \cdot (1 - 200 \cdot 0.001) + 200 \cdot 0.0023 = 0.5351 \text{ m}$$

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

$$S_{out/r}^{ECO_2} = S_{out/1m}^{ECO_2} \cdot H \cdot n_{tube/r}^{ECO_2} = 0.5351 \cdot 6.78 \cdot 32 = 116.1 \text{ m}^2$$

The number of lateral rows of tubers in ECO<sub>2</sub>HP:

$$n_{row}^{ECO_2} = \frac{S_{out}^{ECO_2}}{S_{out/r}^{ECO_2}} = \frac{236.18}{116.1} = 2.03$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{ECO_2} = 2$$

#### 4.7.8 Calculating the actual heat transfer rate of ECO<sub>2</sub>HP

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

$$S_{out}^{ECO_2:real} = S_{out/r}^{ECO_2} \cdot n_{row}^{ECO_2} = 116.1 \cdot 2 = 232.21 \text{ m}^2$$

The actual heat transfer rate of ECO<sub>2</sub>HP:

$$Q_{ECO_2}^{HP:real} = k^{ECO_2} \cdot S_{out}^{ECO_2:real} \cdot \Delta t_{ln}^{ECO_2} = 33.8 \cdot 232.21 \cdot 19.82 = 155.55 \text{ kW}$$

Error verification:

$$\%Error^{ECO_2} = \left| \left( \frac{155.55}{158.21} - 1 \right) \cdot 100 \right| = 1.68\%$$

The discrepancy between the theoretical and the actual heat transfer rate of economizer ECO<sub>2HP</sub> is less than 2%, thus the current configuration of ECO<sub>2HP</sub> is acceptable.

#### 4.7.9 Determining the actual parameters of the flue gas exiting ECO<sub>2HP</sub>

Due to the discrepancy between the theoretical and the actual heat transfer rate of economizer ECO<sub>2HP</sub> the parameters of the flue gas exiting ECO<sub>2HP</sub> at point H must be recalculated.

##### Actual enthalpy of the flue gas at point H:

$$I_H^{real} = I_G^{real} - \frac{Q_{ECO2}^{HP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 224.81 - \frac{155.55}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 218.15 \text{ kJ/Nm}^3$$

##### Actual temperature of the flue gas at point H:

$$t_H^{real} = \frac{I_H^{real} - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{218.15 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 163 \text{ }^\circ\text{C}$$

### 4.8 Design of the Low Pressure Economizer ECO<sub>LP</sub>

Several parameters are required in the calculations associated with the design of the low pressure economizer ECO<sub>LP</sub>. Parameters, other than those describing the geometry of the economizer, that are necessary for the calculations regarding the design and sizing of economizer ECO<sub>LP</sub>, some of which were determined in previous calculations, are organized in Table 4.20 below.

|       | Parameters   | Symbol                 | Value  | Units                 |
|-------|--|------------------------|--------|-----------------------|
|       | Heat transfer rate required of ECO <sub>LP</sub> (theoretical value) | $Q_{ECO}^{LP}$         | 365.82 | [kW]                  |
| Flue  | Actual temperature of flue gas entering into ECO <sub>LP</sub>       | $t_H^{real}$           | 163    | [°C]                  |
|       | Actual enthalpy of flue gas entering into ECO <sub>LP</sub>          | $I_H^{real}$           | 218.15 | [kJ/Nm <sup>3</sup> ] |
| water | Mass flow rate of water in the high pressure circuit                 | $\dot{M}_{Steam}^{LP}$ | 0.88   | [kg/s]                |
|       | Temperature of water entering ECO <sub>LP</sub>                      | $t_5^{LP}$             | 53     | [°C]                  |
|       | Pressure of water entering ECO <sub>LP</sub>                         | $p_5^{LP}$             | 0.86   | [MPa]                 |
|       | Temperature of water exiting ECO <sub>LP</sub>                       | $t_4^{LP}$             | 151.15 | [°C]                  |
|       | Pressure of water exiting ECO <sub>LP</sub>                          | $p_4^{LP}$             | 0.56   | [MPa]                 |

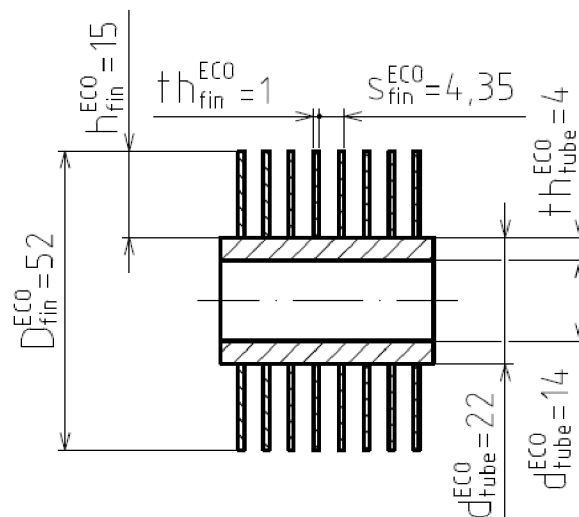
Table 4.20 Parameters necessary for economizer ECO<sub>LP</sub> design, not including parameters describing the geometry of ECO<sub>LP</sub>

### 4.8.1 Geometry of the low pressure economizer $ECO_{LP}$

The chosen dimensions of the finned tubes used in the low pressure economizer ( $ECO_{LP}$ ) are shown in *Table 4.21* in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in *Table 4.21* are selected and later on adjusted in order to supply the economizer with an acceptable water flow speed through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

|      | Parameter                | Symbol            | Value | Units |
|------|--------------------------|-------------------|-------|-------|
| tube | Outer tube diameter      | $D_{tube}^{ECO}$  | 22    | [mm]  |
|      | Tube wall thickness      | $th_{tube}^{ECO}$ | 4     | [mm]  |
|      | Inner tube diameter      | $d_{tube}^{ECO}$  | 14    | [mm]  |
| fins | Fin thickness            | $th_{fin}^{ECO}$  | 1     | [mm]  |
|      | Number of fins per meter | $n_{fin}^{ECO}$   | 230   | [mm]  |
|      | Fin spacing              | $s_{fin}^{ECO}$   | 4.35  | [mm]  |
|      | Fin height               | $h_{fin}^{ECO}$   | 15    | [mm]  |
|      | Outer fin diameter       | $D_{fin}^{ECO}$   | 52    | [mm]  |

*Table 4.21* Parameters selected for the finned tubes used in the low pressure economizer  $ECO_{LP}$



*Figure 4-17* Tube geometry  $ECO_{LP}$

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

**Chosen layout dimensions:**

Lateral gap between tubes  $a_{ECO} = 9.75 \text{ mm}$

Longitudinal tube spacing  $s_{2(ECO)} = 90 \text{ mm}$

### Lateral tube spacing in economizer ECO<sub>LP</sub>:

$$s_{1(ECO)} = D_{tube}^{ECO} + 2 \cdot h_{fin}^{ECO} + a_{ECO} = 0.022 + 2 \cdot 0.015 + 0.00975 = 0.06175 \text{ m}$$

### Number of tubes in each lateral row of tubes in ECO<sub>LP</sub>:

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{ECO} = \frac{L}{s_{1(ECO)}} - 0.5 = \frac{1.76}{0.06175} - 0.5 = 28.002$$

$$n_{tube/r}^{ECO} = 28$$

### Gap verification:

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

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

$$gap = 1.76 - 28 \cdot 0.06175 - \frac{0.022}{2} - 0.015 = 0.005 \text{ m}$$

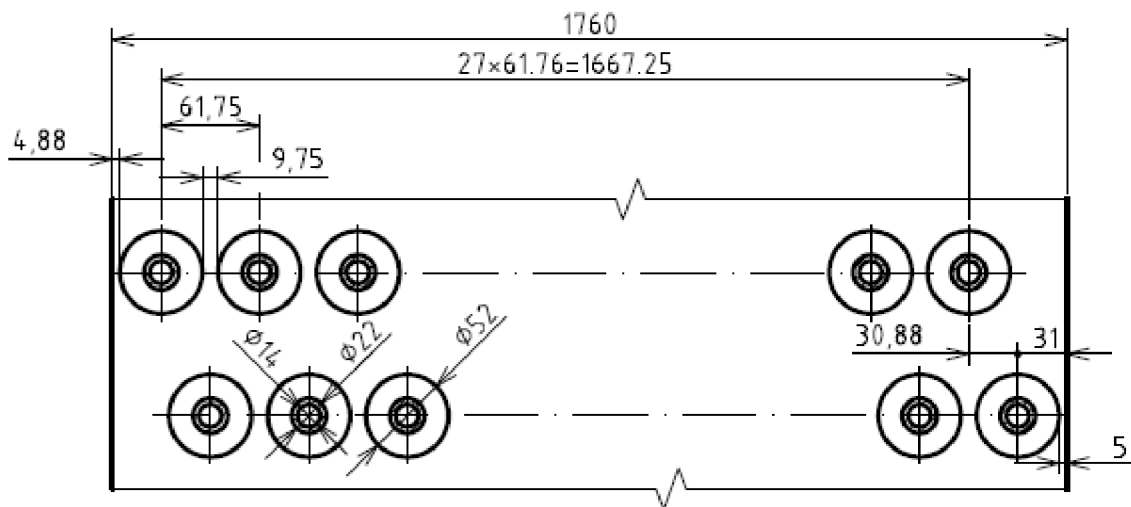


Figure 4-18 Tube layout ECO<sub>LP</sub>

### 4.8.2 Verification of the speed of water passing through economizer ECO<sub>LP</sub>

The speed of the water passing through economizer ECO<sub>LP</sub> is calculated according to average specific volume of the water. The specific volume of the water is determined through X-Steam according to the average pressure and temperature of the water passing through ECO<sub>LP</sub>.

$$\bar{t}_{ECO} = \frac{t_4^{LP} + t_5^{LP}}{2} = \frac{151.15 + 53}{2} = 102.08 \text{ } ^\circ\text{C}$$

$$\bar{p}_{ECO} = \frac{p_4^{LP} + p_5^{LP}}{2} = \frac{0.56 + 0.86}{2} = 0.71 \text{ MPa}$$

$$\bar{v}_{ECO} = 0.001 \text{ m}^3/\text{kg} \quad (\text{determined using X-Steam, } f(p,t))$$

**Speed of water, before splitting economizer into sections:**

$$W_{Steam}^{ECO'} = \frac{\dot{M}_{Steam}^{LP} \cdot \bar{v}_{ECO}}{\frac{\pi \cdot (d_{tube}^{ECO})^2}{4} \cdot n_{tube/r}} = \frac{0.88 \cdot 0.001}{\frac{\pi \cdot 0.014^2}{4} \cdot 28} = 0.203 \text{ m/s}$$

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

$$\text{Number of sections} \quad n_{section}^{ECO} = 4$$

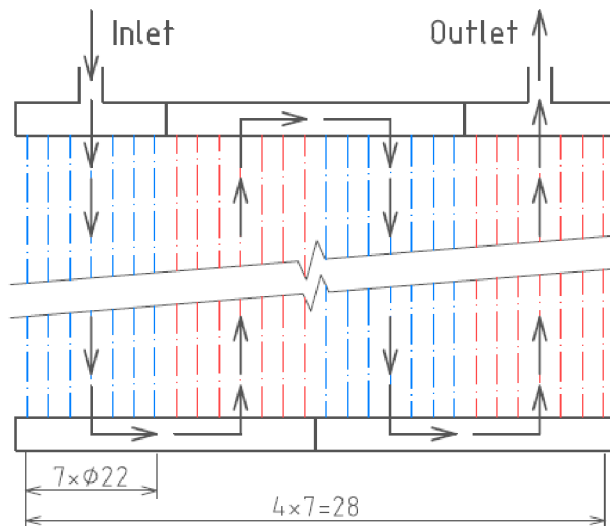


Figure 4-19 Splitting schematic  $ECO_{LP}$

**Actual speed of water flowing through  $ECO_{LP}$ :**

$$W_{Steam}^{ECO} = W_{Steam}^{ECO'} \cdot n_{section}^{ECO} = 0.203 \cdot 4 = 0.81 \text{ m/s}$$

#### 4.8.3 Verification of the speed of flue gas passing through economizer $ECO_{LP}$

**Recalculated flue gas enthalpy and temperature at point I:**

The parameters of the flue gas exiting the economizer at point I are recalculated according to the actual flue gas parameter entering  $ECO_{LP}$ .



$$I'_I = I_H^{real} - \frac{Q_{ECO}^{LP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 218.15 - \frac{365.82}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 202.5 \text{ kJ/Nm}^3$$

$$t'_I = \frac{I'_I - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{202.5 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 151.38^\circ\text{C}$$

**Average flue gas temperature:**

$$\bar{t}_{(HI)} = \frac{t_H^{real} + t'_I}{2} = \frac{163 + 151.38}{2} = 157.19^\circ\text{C}$$

**The actual volumetric flow rate of the flue gas:**

$$\dot{M}_{VFlue}^{ECO} = \frac{273.15 + \bar{t}_{(HI)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 157.19}{273.15} \cdot 23.49 = 37.01 \text{ m}^3/\text{s}$$

**The cross sectional area that the flue gas flows through:**

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

$$A_{duct}^{ECO} = 6.78 \cdot 1.76 - 6.78 \cdot 0.022 \cdot 28 - 6.78 \cdot 2 \cdot 0.015 \cdot 0.001 \cdot 230 \cdot 28 = 6.45 \text{ m}^2$$

**Speed of the flue gas passing through ECO<sub>LP</sub>:**

$$W_{Flue}^{ECO} = \frac{\dot{M}_{VFlue}^{ECO}}{A_{duct}^{ECO}} = \frac{37.01}{6.45} = 5.74 \text{ m/s}$$

#### 4.8.4 Reduced heat transfer coefficient outside the tubes of ECO<sub>LP</sub>, through the flue gas

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

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{ECO} = \frac{s_1(ECO)}{D_{tube}^{ECO}} = \frac{0.06175}{0.022} = 2.81$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{ECO} = \frac{s_1'^{ECO}}{D_{tube}^{ECO}} = \frac{\sqrt{\left(\frac{s_1(ECO)}{2}\right)^2 + (s_2(ECO))^2}}{D_{tube}^{ECO}} = \frac{\sqrt{\left(\frac{0.06175}{2}\right)^2 + 0.09^2}}{0.022} = 4.32$$

Coefficient of the relative tube spacing:

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

$$\varphi_\sigma^{ECO} = \frac{\sigma_1^{ECO} - 1}{\sigma_2'^{ECO} - 1} = \frac{2.81 - 1}{4.32 - 1} = 0.54$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through ECO<sub>LP</sub> has been determined in previous calculations regarding the flue gas speed verification.

$$\text{Temperature} \quad \bar{t}_{(HI)} = 157.19 \text{ }^{\circ}\text{C}$$

$$\lambda_{Flue}^{ECO} = \left[ \frac{157.19 - 100}{100} \cdot (39.228 - 30.89) + 30.89 \right] \cdot 10^{-3} = 3.566 \cdot 10^{-2} \text{ W}/(\text{m} \cdot \text{K})$$

$$\nu_{Flue}^{ECO} = \left[ \frac{157.19 - 100}{100} \cdot (32.36 - 21.41) + 21.41 \right] \cdot 10^{-6} = 2.767 \cdot 10^{-5} \text{ m}^2/\text{s}$$

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

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 2 lateral rows of tubes in economizer ECO<sub>LP</sub>.

$$c_z^{ECO} = 0.87$$

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

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

$$\alpha_c^{ECO} = 0.23 \cdot 0.87 \cdot 0.54^{0.2} \cdot \frac{0.03566}{0.00435} \cdot \left( \frac{0.022}{0.00435} \right)^{-0.54} \cdot \left( \frac{0.0015}{0.00435} \right)^{-0.14} \cdot \left( \frac{5.74 \cdot 0.00435}{2.767 \cdot 10^{-5}} \right)^{0.65}$$

$$\alpha_c^{ECO} = 42.41 \text{ W}/(\text{m}^2\text{K})$$

### Coefficient characterizing the effectiveness of the fins:

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

Coefficient  $\beta^{ECO}$ :

$$\beta^{ECO} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{ECO}}{th_{fin}^{ECO} \cdot \lambda_{fin} \cdot (1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO})}} = \sqrt{\frac{2 \cdot 0.85 \cdot 42.41}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 42.41)}}$$

$$\beta^{ECO} = 40.53 \text{ m}^{-1}$$

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

$$\beta^{ECO} \cdot h_{fin}^{ECO} = 40.53 \cdot 0.015 = 0.61$$

$$\frac{D_{fin}^{ECO}}{D_{tube}^{ECO}} = \frac{0.052}{0.022} = 2.36$$

Coefficient determined using the graph:

$$E^{ECO} = 0.85$$

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

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

$$\frac{S_{fin}^{ECO}}{S_{out}^{ECO}} = \frac{\left(\frac{D_{fin}^{ECO}}{D_{tube}^{ECO}}\right)^2 - 1}{\left(\frac{D_{fin}^{ECO}}{D_{tube}^{ECO}}\right)^2 - 1 + 2 \cdot \left(\frac{S_{fin}^{ECO}}{D_{tube}^{ECO}} - \frac{th_{fin}^{ECO}}{D_{tube}^{ECO}}\right)} = \frac{\left(\frac{0.052}{0.022}\right)^2 - 1}{\left(\frac{0.052}{0.022}\right)^2 - 1 + 2 \cdot \left(\frac{0.00435}{0.022} - \frac{0.001}{0.022}\right)}$$

$$\frac{S_{fin}^{ECO}}{S_{out}^{ECO}} = 0.94$$

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

$$\frac{S_{out-fin}^{ECO}}{S_{out}^{ECO}} = 1 - \frac{S_{fin}^{ECO}}{S_{out}^{ECO}} = 1 - 0.94 = 0.06$$

### The reduced heat transfer coefficient outside the tubes:

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

$$\alpha_{r,out}^{ECO} = (0.94 \cdot 0.85 \cdot 1 + 0.06) \cdot \frac{0.85 \cdot 42.41}{1 + 0.0043 \cdot 0.85 \cdot 42.41} = 26.68 \text{ W}/(\text{m}^2 \text{ K})$$

### 4.8.5 Overall heat transfer coefficient for ECO<sub>LP</sub>

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

$$k^{ECO} = \alpha_{r,out}^{ECO} = 26.68 \text{ W}/(\text{m}^2 \text{ K})$$

#### 4.8.6 Logarithmic mean temperature difference across ECO<sub>LP</sub>

$$\Delta t_1^{ECO} = t_H^{real} - t_4^{LP} = 163 - 151.15 = 11.83 \text{ K}$$

$$\Delta t_2^{ECO} = t_1' - t_5^{LP} = 151.38 - 53 = 98.38 \text{ K}$$

$$\Delta t_{ln}^{ECO} = \frac{\Delta t_2^{ECO} - \Delta t_1^{ECO}}{\ln\left(\frac{\Delta t_2^{ECO}}{\Delta t_1^{ECO}}\right)} = \frac{98.38 - 11.83}{\ln\left(\frac{98.38}{11.83}\right)} = 40.88 \text{ K}$$

#### 4.8.7 The number of lateral rows of tubes required for ECO<sub>LP</sub>

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

$$S_{out}^{ECO} = \frac{Q_{ECO}^{HP}}{k^{ECO} \cdot \Delta t_{ln}^{ECO}} = \frac{365.82}{26.68 \cdot 40.88} = 335.48 \text{ m}^2$$

The outer surface area of one fin:

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

$$S_{1fin}^{ECO} = 2 \cdot \pi \cdot \frac{0.052^2 - 0.022^2}{4} + \pi \cdot 0.052 \cdot 0.001 = 0.0037 \text{ m}^2$$

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

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

$$S_{out/1m}^{ECO} = \pi \cdot 0.022 \cdot (1 - 230 \cdot 0.001) + 230 \cdot 0.0037 = 0.8928 \text{ m}$$

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

$$S_{out/r}^{ECO} = S_{out/1m}^{ECO} \cdot H \cdot n_{tube/r}^{ECO} = 0.8928 \cdot 6.78 \cdot 28 = 169.5 \text{ m}^2$$

The number of lateral rows of tubers in ECO<sub>3HP</sub>:

$$n_{row}^{ECO} = \frac{S_{out}^{ECO2}}{S_{out/r}^{ECO}} = \frac{335.48}{169.5} = 1.98$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{ECO} = 2$$

#### 4.8.8 Calculating the actual heat transfer rate of ECO<sub>LP</sub>

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

$$S_{out}^{ECO:real} = S_{out/r}^{ECO} \cdot n_{row}^{ECO} = 169.5 \cdot 2 = 338.99 \text{ m}^2$$

The actual heat transfer rate of ECO<sub>2HP</sub>:

$$Q_{ECO}^{LP:real} = k^{ECO} \cdot S_{out}^{ECO:real} \cdot \Delta t_{ln}^{ECO} = 26.68 \cdot 338.99 \cdot 40.88 = 369.65 \text{ kW}$$

**Error verification:**

$$\%Error^{ECO} = \left| \left( \frac{369.65}{365.82} - 1 \right) \cdot 100 \right| = 1.05\%$$

The discrepancy between the theoretical and the actual heat transfer rate of economizer ECO<sub>LP</sub> is less than 2%, thus the current configuration of ECO<sub>LP</sub> is acceptable.

#### 4.8.9 Determining the actual parameters of the flue gas exiting ECO<sub>LP</sub>

Due to the discrepancy between the theoretical and the actual heat transfer rate of economizer ECO<sub>LP</sub> the parameters of the flue gas exiting ECO<sub>LP</sub> at point I must be recalculated.

**Actual enthalpy of the flue gas at point I:**

$$I_I^{real} = I_H^{real} - \frac{Q_{ECO}^{LP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 218.15 - \frac{369.65}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 202.34 \text{ kJ/Nm}^3$$

**Actual temperature of the flue gas at point I:**

$$t_I^{real} = \frac{I_I^{real} - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{202.34 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 151.26^\circ\text{C}$$

### 4.9 Design of the High Pressure Economizer ECO<sub>IHP</sub>

Several parameters are required in the calculations associated with the design of the high pressure economizer ECO<sub>IHP</sub>. Parameters, other than those describing the geometry of the economizer, that are necessary for the calculations regarding the design and sizing of economizer ECO<sub>IHP</sub>, some of which were determined in previous calculations, are organized in *Table 4.22* below.

|       | Parameters  | Symbol                 | Value   | Units                 |
|-------|---|------------------------|---------|-----------------------|
|       | Heat transfer rate required of ECO <sub>1HP</sub><br>(theoretical value)                              | $Q_{ECO1}^{HP}$        | 1119.61 | [kW]                  |
| Flue  | Actual temperature of flue gas entering into ECO <sub>1HP</sub>                                       | $t_l^{real}$           | 151.26  | [°C]                  |
|       | Actual enthalpy of flue gas entering into ECO <sub>1HP</sub>  | $I_l^{real}$           | 202.34  | [kJ/Nm <sup>3</sup> ] |
| water | Mass flow rate of water in the high pressure circuit<br>(only 95% passes through ECO <sub>1HP</sub> ) | $\dot{M}_{Steam}^{HP}$ | 3.24    | [kg/s]                |
|       | Temperature of water entering ECO <sub>1HP</sub>  | $t_9^{HP}$             | 53      | [°C]                  |
|       | Pressure of water entering ECO <sub>1HP</sub>   | $p_9^{HP}$             | 5.1     | [MPa]                 |
|       | Temperature of water exiting ECO <sub>1HP</sub>   | $t_8^{HP}$             | 139.4   | [°C]                  |
|       | Pressure of water exiting ECO <sub>1HP</sub>  | $p_8^{HP}$             | 5       | [MPa]                 |

Table 4.22 Parameters necessary for economizer ECO<sub>1HP</sub> design, not including parameters describing the geometry of ECO<sub>1HP</sub>

#### 4.9.1 Geometry of the high pressure economizer ECO<sub>1HP</sub>

The chosen dimensions of the finned tubes used in the high pressure economizer (ECO<sub>1HP</sub>) are shown in Table 4.23 in units of millimeters, however in any calculations these dimensions must be substituted into the equations in units of meters (SI units). The parameters in Table 4.23 are selected and later on adjusted in order to supply the economizer with an acceptable water flow speed through its tubes. These dimensions are also adjusted so that the required heat transfer rate of this heat exchanger is approximately met with a whole number of lateral rows of tubes making up the heat exchanger.

|      | Parameter                | Symbol             | Value | Units |
|------|--------------------------|--------------------|-------|-------|
| tube | Outer tube diameter      | $D_{tube}^{ECO1}$  | 33.7  | [mm]  |
|      | Tube wall thickness      | $th_{tube}^{ECO1}$ | 4     | [mm]  |
|      | Inner tube diameter      | $d_{tube}^{ECO1}$  | 25.7  | [mm]  |
| fins | Fin thickness            | $th_{fin}^{ECO1}$  | 1     | [mm]  |
|      | Number of fins per meter | $n_{fin}^{ECO1}$   | 220   | [mm]  |
|      | Fin spacing              | $s_{fin}^{ECO1}$   | 4.55  | [mm]  |
|      | Fin height               | $h_{fin}^{ECO1}$   | 12    | [mm]  |
|      | Outer fin diameter       | $D_{fin}^{ECO1}$   | 57.7  | [mm]  |

Table 4.23 Parameters selected for the finned tubes used in the high pressure economizer ECO<sub>1HP</sub>

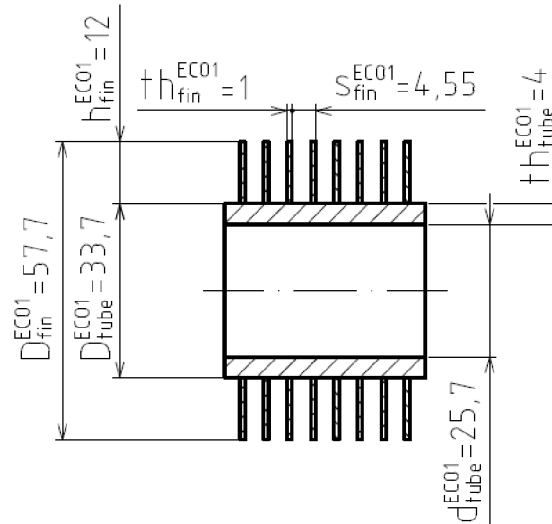


Figure 4-20 Tube geometry  $ECO_{1HP}$

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

#### Chosen layout dimensions:

Lateral gap between tubes  $a_{ECO1} = 4 \text{ mm}$

Longitudinal tube spacing  $s_{2(ECO1)} = 90 \text{ mm}$

#### Lateral tube spacing in economizer $ECO_{2HP}$ :

$$s_{1(ECO1)} = D_{tube}^{ECO1} + 2 \cdot h_{fin}^{ECO1} + a_{ECO1} = 0.0337 + 2 \cdot 0.012 + 0.004 = 0.0617 \text{ m}$$

#### Number of tubes in each lateral row of tubes in $ECO_{1HP}$ :

Naturally this value must be rounded to the nearest whole number

$$n_{tube/r}^{ECO1} = \frac{L}{s_{1(ECO1)}} - 0.5 = \frac{1.76}{0.0617} - 0.5 = 28.03$$

$$n_{tube/r}^{ECO1} = 28$$

#### Gap verification:

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

$$gap = L - n_{tube/r}^{ECO1} \cdot s_{1(ECO1)} - \frac{D_{tube}^{ECO1}}{2} - h_{fin}^{ECO1}$$

$$gap = 1.76 - 28 \cdot 0.0617 - \frac{0.0337}{2} - 0.012 = 0.00355 \text{ m}$$

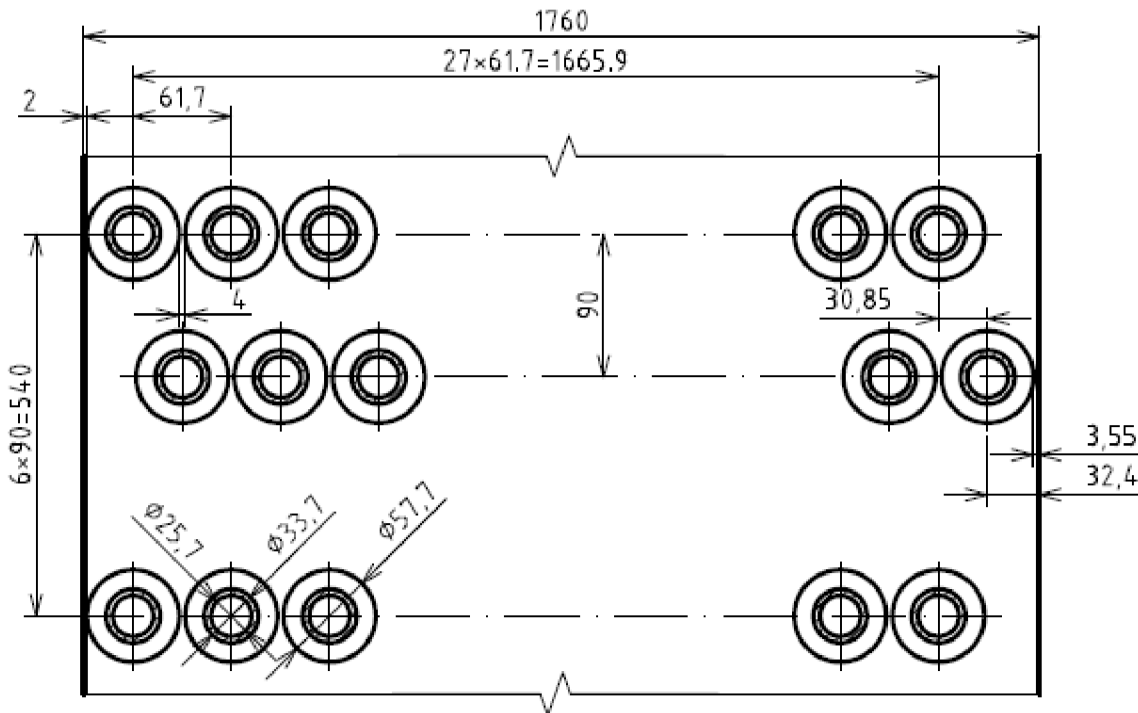


Figure 4-21 Tube layout  $ECO_{1HP}$

#### 4.9.2 Verification of the speed of water passing through economizer $ECO_{1HP}$

The speed of the water passing through economizer  $ECO_{1HP}$  is calculated according to average specific volume of the water. The specific volume of the water is determined through X-Steam according to the average pressure and temperature of the water passing through  $ECO_{1HP}$ .

$$\bar{t}_{ECO1} = \frac{t_8^{HP} + t_9^{HP}}{2} = \frac{139.4 + 53}{2} = 96.2 \text{ } ^\circ\text{C}$$

$$\bar{p}_{ECO1} = \frac{p_8^{HP} + p_9^{HP}}{2} = \frac{5 + 5.1}{2} = 5.05 \text{ MPa}$$

$$\bar{v}_{ECO1} = 0.001 \text{ m}^3/\text{kg} \quad (\text{determined using X-Steam, } f(p,t))$$

**Speed of water, before splitting economizer into sections:**

$$W_{Steam}^{ECO1'} = \frac{0.95 \cdot \dot{M}_{Steam}^{HP} \cdot \bar{v}_{ECO1}}{\frac{\pi \cdot (d_{tube}^{ECO1})^2}{4} \cdot n_{tube/r}} = \frac{0.95 \cdot 3.24 \cdot 0.001}{\frac{\pi \cdot 0.0257^2}{4} \cdot 28} = 0.22 \text{ m/s}$$

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

Number of sections  $n_{section}^{ECO1} = 4$



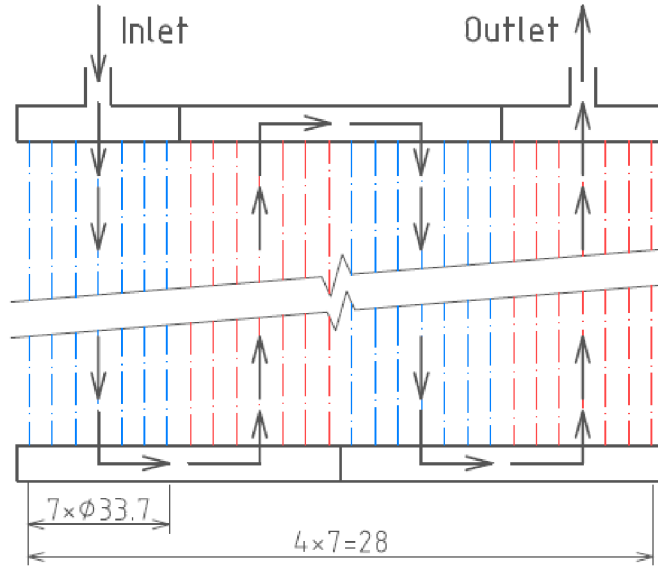


Figure 4-22 Splitting schematic  $ECO_{1HP}$

**Actual speed of water flowing through  $ECO_{1HP}$ :**

$$W_{Steam}^{ECO1} = W_{Steam}^{ECO1'} \cdot n_{section}^{ECO1} = 0.22 \cdot 4 = 0.88 \text{ m/s}$$

#### 4.9.3 Verification of the speed of the gas passing through economizer $ECO_{1HP}$

**Recalculated flue gas enthalpy and temperature at point H:**

The parameters of the flue gas exiting the economizer at point H are recalculated according to the actual flue gas parameter entering  $ECO_{1HP}$ .

$$I_j' = I_j^{real} - \frac{Q_{ECO1}^{HP}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 202.34 - \frac{1119.61}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 154.43 \text{ kJ/Nm}^3$$

$$t_j' = \frac{I_j' - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{154.43 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 115.71^\circ\text{C}$$

**Average flue gas temperature:**

$$\bar{t}_{(IJ)} = \frac{t_j^{real} + t_j'}{2} = \frac{151.26 + 115.71}{2} = 133.49^\circ\text{C}$$

**The actual volumetric flow rate of the flue gas:**

$$\dot{M}_{VFlue}^{ECO1} = \frac{273.15 + \bar{t}_{(IJ)}}{273.15} \cdot \dot{M}_{VFlue} = \frac{273.15 + 133.49}{273.15} \cdot 23.49 = 34.97 \text{ m}^3/\text{s}$$

**The cross sectional area that the flue gas flows through:**

$$A_{duct}^{ECO1} = H \cdot L - H \cdot D_{tube}^{ECO1} \cdot n_{tube/r}^{ECO1} - H \cdot 2 \cdot h_{fin}^{ECO1} \cdot t h_{fin}^{ECO1} \cdot n_{fin}^{ECO1} \cdot n_{tube/r}^{ECO1}$$

$$A_{duct}^{ECO1} = 6.78 \cdot 1.76 - 6.78 \cdot 0.0337 \cdot 28 - 6.78 \cdot 2 \cdot 0.012 \cdot 0.001 \cdot 220 \cdot 28 = 4.53 \text{ m}^2$$

**Speed of the flue gas passing through ECO<sub>IHP</sub>:**

$$W_{Flue}^{ECO1} = \frac{\dot{M}_{VFlue}^{ECO1}}{A_{duct}^{ECO1}} = \frac{34.97}{4.53} = 7.71 \text{ m/s}$$

#### 4.9.4 Reduced heat transfer coefficient outside the tubes of ECO<sub>IHP</sub>, through the flue gas

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

Lateral tube spacing relative to the outer tube diameter:

$$\sigma_1^{ECO1} = \frac{S_1^{(ECO1)}}{D_{tube}^{ECO1}} = \frac{0.0617}{0.0337} = 1.83$$

Diagonal tube spacing relative to the outer tube diameter:

$$\sigma_2'^{ECO1} = \frac{S'^{ECO1}}{D_{tube}^{ECO1}} = \frac{\sqrt{\left(\frac{S_1^{(ECO1)}}{2}\right)^2 + (S_2^{(ECO1)})^2}}{D_{tube}^{ECO1}} = \frac{\sqrt{\left(\frac{0.0617}{2}\right)^2 + 0.09^2}}{0.0337} = 2.82$$

Coefficient of the relative tube spacing:

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

$$\varphi_{\sigma}^{ECO1} = \frac{\sigma_1^{ECO1} - 1}{\sigma_2'^{ECO1} - 1} = \frac{1.83 - 1}{2.82 - 1} = 0.46$$

Flue gas Parameters:

The thermal conductivity, together with the kinematic viscosity of the flue gas are demined using *Table 4.2* and *Table 4.3* respectively, according to the average flue gas temperature. The average temperature of the flue gas passing through ECO<sub>IHP</sub> has been determined in previous calculations regarding the flue gas speed verification.

$$\text{Temperature} \quad \bar{t}_{(f)} = 133.49 \text{ }^{\circ}\text{C}$$

$$\lambda_{Flue}^{ECO1} = \left[ \frac{133.49 - 100}{100} \cdot (39.228 - 30.89) + 30.89 \right] \cdot 10^{-3} = 3.368 \cdot 10^{-2} \text{ W/(m} \cdot \text{K)}$$

$$\nu_{Flue}^{ECO1} = \left[ \frac{133.49 - 100}{100} \cdot (32.36 - 21.41) + 21.41 \right] \cdot 10^{-6} = 2.508 \cdot 10^{-5} \text{ m}^2/\text{s}$$

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

This correction coefficient is determined according to a graph from source [2] on page 116. The correction coefficient is determined assuming that there are 7 lateral rows of tubes in economizer ECO<sub>1HP</sub>.

$$c_z^{ECO1} = 0.98$$

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

$$\alpha_c^{ECO1} = 0.23 \cdot c_z^{ECO1} \cdot (\varphi_\sigma^{ECO1})^{0.2} \cdot \frac{\lambda_{Flue}^{ECO1}}{s_{fin}^{ECO1}} \cdot \left(\frac{D_{tube}^{ECO1}}{s_{fin}^{ECO1}}\right)^{-0.54} \cdot \left(\frac{h_{fin}^{ECO1}}{s_{fin}^{ECO1}}\right)^{-0.14} \cdot \left(\frac{W_{Flue}^{ECO1} \cdot s_{fin}^{ECO1}}{v_{Flue}^{ECO1}}\right)^{0.65}$$

$$\alpha_c^{ECO1} = 0.23 \cdot 0.98 \cdot 0.46^{0.2} \cdot \frac{0.03368}{0.00455} \cdot \left(\frac{0.0337}{0.00455}\right)^{-0.54} \cdot \left(\frac{0.0012}{0.00455}\right)^{-0.14} \cdot \left(\frac{7.71 \cdot 0.00455}{2.508 \cdot 10^{-5}}\right)^{0.65}$$

$$\alpha_c^{ECO1} = 46.81 \text{ W}/(m^2K)$$

### Coefficient characterizing the effectiveness of the fins:

This coefficient is determined through a graph from page 114 of source [2] according to the product of  $\beta^{ECO1} \cdot h_{fin}^{ECO1}$  and the quotient of  $D_{fin}^{ECO1}/D_{tube}^{ECO1}$ , where  $\beta^{ECO1}$  is a coefficient.

Coefficient  $\beta^{ECO1}$ :

$$\beta^{ECO1} = \sqrt{\frac{2 \cdot \Psi_{fin} \cdot \alpha_c^{ECO1}}{t h_{fin}^{ECO1} \cdot \lambda_{fin} \cdot (1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO1})}} = \sqrt{\frac{2 \cdot 0.85 \cdot 46.81}{0.001 \cdot 38 \cdot (1 + 0.0043 \cdot 0.85 \cdot 46.81)}}$$

$$\beta^{ECO1} = 42.29 \text{ m}^{-1}$$

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

$$\beta^{ECO1} \cdot h_{fin}^{ECO1} = 42.29 \cdot 0.012 = 0.51$$

$$\frac{D_{fin}^{ECO1}}{D_{tube}^{ECO1}} = \frac{0.0577}{0.0337} = 1.71$$

Coefficient determined using the graph:

$$E^{ECO1} = 0.9$$

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

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

$$\frac{S_{fin}^{ECO1}}{S_{out}^{ECO1}} = \frac{\left(\frac{D_{fin}^{ECO1}}{D_{tube}^{ECO1}}\right)^2 - 1}{\left(\frac{D_{fin}^{ECO1}}{D_{tube}^{ECO1}}\right)^2 - 1 + 2 \cdot \left(\frac{S_{fin}^{ECO1}}{D_{tube}^{ECO1}} - \frac{th_{fin}^{ECO1}}{D_{tube}^{ECO1}}\right)} = \frac{\left(\frac{0.0577}{0.0337}\right)^2 - 1}{\left(\frac{0.0577}{0.0337}\right)^2 - 1 + 2 \cdot \left(\frac{0.00455}{0.0337} - \frac{0.001}{0.0337}\right)}$$

$$\frac{S_{fin}^{ECO1}}{S_{out}^{ECO1}} = 0.9$$

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

$$\frac{S_{out-fin}^{ECO1}}{S_{out}^{ECO1}} = 1 - \frac{S_{fin}^{ECO1}}{S_{out}^{ECO1}} = 1 - 0.9 = 0.1$$

**The reduced heat transfer coefficient outside the tubes:**

$$\alpha_{r:out}^{ECO1} = \left(\frac{S_{fin}^{ECO1}}{S_{out}^{ECO1}} \cdot E^{ECO1} \cdot \mu + \frac{S_{out-fin}^{ECO1}}{S_{out}^{ECO1}}\right) \cdot \frac{\Psi_{fin} \cdot \alpha_c^{ECO1}}{1 + \varepsilon \cdot \Psi_{fin} \cdot \alpha_c^{ECO1}}$$

$$\alpha_{r:out}^{ECO1} = (0.9 \cdot 0.9 \cdot 1 + 0.1) \cdot \frac{0.85 \cdot 46.81}{1 + 0.0043 \cdot 0.85 \cdot 46.81} = 30.94 \text{ W}/(\text{m}^2\text{K})$$

#### 4.9.5 Overall heat transfer coefficient for ECO<sub>1HP</sub>

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

$$k^{ECO1} = \alpha_{r:out}^{ECO1} = 30.94 \text{ W}/(\text{m}^2\text{K})$$

#### 4.9.6 Logarithmic mean temperature difference across ECO<sub>1HP</sub>

$$\Delta t_1^{ECO1} = t_j^{real} - t_8^{HP} = 151.26 - 139.4 = 11.86 \text{ K}$$

$$\Delta t_2^{ECO1} = t'_j - t_9^{HP} = 115.71 - 53 = 62.71 \text{ K}$$

$$\Delta t_{ln}^{ECO1} = \frac{\Delta t_2^{ECO1} - \Delta t_1^{ECO1}}{\ln\left(\frac{\Delta t_2^{ECO1}}{\Delta t_1^{ECO1}}\right)} = \frac{62.71 - 11.86}{\ln\left(\frac{62.71}{11.86}\right)} = 30.53 \text{ K}$$

#### 4.9.7 The number of lateral rows of tubes required for ECO<sub>1HP</sub>

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

$$S_{out}^{ECO1} = \frac{Q_{ECO1}^{HP}}{k^{ECO1} \cdot \Delta t_{ln}^{ECO1}} = \frac{1119.61}{30.94 \cdot 30.53} = 1185.18 \text{ m}^2$$

The outer surface area of one fin:

$$S_{1fin}^{ECO1} = 2 \cdot \pi \cdot \frac{(D_{fin}^{ECO1})^2 - (D_{tube}^{ECO1})^2}{4} + \pi \cdot D_{fin}^{ECO1} \cdot th_{fin}^{ECO1}$$

$$S_{1fin}^{ECO1} = 2 \cdot \pi \cdot \frac{0.0577^2 - 0.0337^2}{4} + \pi \cdot 0.0577 \cdot 0.001 = 0.0036 \text{ m}^2$$

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

$$S_{out/1m}^{ECO1} = \pi \cdot D_{tube}^{ECO1} \cdot (1 - n_{fin}^{ECO1} \cdot th_{fin}^{ECO1}) + n_{fin}^{ECO1} \cdot S_{1fin}^{ECO1}$$

$$S_{out/1m}^{ECO1} = \pi \cdot 0.0337 \cdot (1 - 220 \cdot 0.001) + 220 \cdot 0.0036 = 0.8805 \text{ m}$$

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

$$S_{out/r}^{ECO1} = S_{out/1m}^{ECO1} \cdot H \cdot n_{tube/r}^{ECO1} = 0.8805 \cdot 6.78 \cdot 28 = 167.16 \text{ m}^2$$

The number of lateral rows of tubers in ECO<sub>1HP</sub>:

$$n_{row}^{ECO1} = \frac{S_{out}^{ECO1}}{S_{out/r}^{ECO1}} = \frac{1185.18}{167.16} = 7.09$$

Naturally tis value must be rounded to the nearest whole number

$$n_{row}^{ECO1} = 7$$

#### 4.9.8 Calculating the actual heat transfer rate of ECO<sub>1HP</sub>

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

$$S_{out}^{ECO1:real} = S_{out/r}^{ECO1} \cdot n_{row}^{ECO1} = 167.16 \cdot 7 = 1170.1 \text{ m}^2$$

The actual heat transfer rate of ECO<sub>1HP</sub>:

$$Q_{ECO1}^{HP:real} = k^{ECO1} \cdot S_{out}^{ECO1:real} \cdot \Delta t_{ln}^{ECO1} = 30.94 \cdot 1170.1 \cdot 30.53 = 1105.36 \text{ kW}$$

Error verification:

$$\%Error^{ECO1} = \left| \left( \frac{1105.36}{1119.61} - 1 \right) \cdot 100 \right| = 1.27\%$$

The discrepancy between the theoretical and the actual heat transfer rate of economizer ECO<sub>1HP</sub> is less than 2%, thus the current configuration of ECO<sub>1HP</sub> is acceptable.

#### 4.9.9 Determining the actual parameters of the flue gas exiting ECO<sub>1HP</sub>

Due to the discrepancy between the theoretical and the actual heat transfer rate of economizer ECO<sub>1HP</sub> the parameters of the flue gas exiting ECO<sub>1HP</sub> at point J must be recalculated.

##### Actual enthalpy of the flue gas at point J:

$$I_J^{real} = I_I^{real} - \frac{Q_{ECO1}^{HP:real}}{\left(1 - \frac{Q_{RC\%}}{100}\right) \cdot \dot{M}_{VFlue}} = 202.34 - \frac{1105.36}{\left(1 - \frac{0.49}{100}\right) \cdot 23.49} = 155.04 \text{ kJ/Nm}^3$$

##### Actual temperature of the flue gas at point J:

$$t_J^{real} = \frac{I_J^{real} - I_{Flue:100}}{I_{Flue:200} - I_{Flue:100}} \cdot (200 - 100) + 100 = \frac{155.04 - 133.27}{268.01 - 133.27} \cdot 100 + 100 = 116.16^\circ\text{C}$$

#### 4.10 Overview of the Calculated Values

An overview of the main values calculated in this section are presented in *Table 4.24* and *Table 4.25*. They include values regarding individual heat exchanger as well as actual flue gas parameters.

| parameters    |                    | Flue gas speed | Water/steam speed | Actual heat transfer rate | Longitudinal tube spacing | Lateral tube spacing | Number of tubes per lateral row | Number of lateral rows |
|---------------|--------------------|----------------|-------------------|---------------------------|---------------------------|----------------------|---------------------------------|------------------------|
| Symbol        |                    | $W_{Flue}$     | $W_{Steam}$       | $Q^{real}$                | $s_2$                     | $s_1$                | $n_{tube/r}$                    | $n_{row}$              |
| Units         |                    | [m/s]          | [m/s]             | [kW]                      | [mm]                      | [mm]                 | [-]                             | [-]                    |
| High pressure | SH <sub>2HP</sub>  | 10             | 19.79             | 807.23                    | 117                       | 74.8                 | 23                              | 4                      |
|               | SH <sub>1HP</sub>  | 9.73           | 15.98             | 1297.87                   | 117                       | 74.8                 | 23                              | 4                      |
|               | EV <sub>HP</sub>   | 10.45          | -                 | 5165.08                   | 117                       | 106.5                | 16                              | 17                     |
|               | ECO <sub>3HP</sub> | 10.27          | 1                 | 1468.96                   | 90                        | 54.05                | 32                              | 9                      |
|               | ECO <sub>2HP</sub> | 9.6            | 0.8               | 155.55                    | 90                        | 54.15                | 32                              | 2                      |
|               | ECO <sub>1HP</sub> | 7.71           | 0.88              | 1105.36                   | 90                        | 61.7                 | 28                              | 7                      |
| Low pressure  | SH <sub>LP</sub>   | 7.15           | 19.54             | 35.35                     | 117                       | 76                   | 23                              | 1                      |
|               | EV <sub>LP</sub>   | 9.27           | -                 | 1833.98                   | 117                       | 100.5                | 17                              | 8                      |
|               | ECO <sub>LP</sub>  | 5.74           | 0.81              | 369.65                    | 90                        | 61.75                | 28                              | 2                      |

Table 4.24 Parameters regarding individual heat exchangers

| Point | Temperature [°C] |        | Enthalpy [kJ/Nm <sup>3</sup> ] |        |
|-------|------------------|--------|--------------------------------|--------|
|       | symbol           | value  | symbol                         | value  |
| A     | $t_A$            | 490    | $I_A$                          | 678.69 |
| B     | $t_B$            | 466.32 | $I_B$                          | 644.15 |
| C     | $t_C$            | 428.24 | $I_C$                          | 588.62 |
| D     | $t_D$            | 272.23 | $I_D$                          | 367.64 |
| E     | $t_E$            | 226.66 | $I_E$                          | 304.79 |
| F     | $t_F$            | 225.57 | $I_F$                          | 303.27 |
| G     | $t_G$            | 167.94 | $I_G$                          | 224.81 |
| H     | $t_H$            | 163    | $I_H$                          | 218.15 |
| I     | $t_I$            | 151.26 | $I_I$                          | 202.34 |
| J     | $t_J$            | 116.16 | $I_J$                          | 155.04 |

Table 4.25 Actual flue gas parameters at various points throughout the boiler





## 5 MATERIAL SELECTION

The material that the heat exchanger tubes are made of is selected according standards (ČSN EN 12952-3) which outline the maximum temperature that each material can be used for. For finned tubes the reference temperature, used to determine which materials can be used, is the temperature of the water/steam exiting the heat exchanger with an additional safety margin. For smooth tube the reference temperature is simply the temperature of the water/steam exiting the heat exchanger. The Material chosen for the fins on all the heat exchanger tubes is CS, as the flue gas temperature does not exceed 500 °C [7]. The materials chosen for the tubes of each heat exchanger are shown in *Table 5.1*.

### Calculations used to determine the reference temperature (finned tubes):

#### Reference temperature:

$$t_{ref} = t_{steam:out} + \Delta t_{safe} \quad (5.1)$$

Where:

$t_{steam:out}$  is temperature water/steam exiting the heat exchanger [°C]

$\Delta t_{safe}$  is additional safety margin [°C]

#### Safety margin (for superheaters):

$$\Delta t_{safe} = 35 \text{ °C} \quad (5.2)$$

#### Safety margin (for all other heat exchangers):

$$\Delta t_{safe} = 15 + 2 \cdot th_{tube(mm)} \quad (5.3)$$

Where:

$th_{tube(mm)}$  is the thickness of the tube wall [mm]

|               | Parameter          | Exiting steam temperature | Safety margin     | Reference temperature | Selected tube material (ČSN EN 12952-3) |
|---------------|--------------------|---------------------------|-------------------|-----------------------|---|
|               | Symbol             | $t_{steam:in}$            | $\Delta t_{safe}$ | $t_{ref}$             | -                                       |
|               | Units              | [°C]                      | [°C]              | [°C]                  | -                                       |
| High pressure | SH <sub>2HP</sub>  | 450                       | 35                | 485                   | 16Mo3                                   |
|               | SH <sub>1HP</sub>  | 408.46                    | 35                | 443.46                | 16Mo3                                   |
|               | EV <sub>HP</sub>   | 261.4                     | 23                | 284.4                 | P235GH                                  |
|               | ECO <sub>3HP</sub> | 256.4                     | 23                | 279.4                 | P235GH                                  |
|               | ECO <sub>2HP</sub> | 151.4                     | 23                | 174.4                 | P235GH                                  |
|               | ECO <sub>1HP</sub> | 139.4                     | 23                | 162.4                 | P235GH                                  |
| Low           | SH <sub>LP</sub>   | 170                       | 0                 | 170                   | P235GH                                  |
|               | EV <sub>LP</sub>   | 156.15                    | 23                | 179.15                | P235GH                                  |
|               | ECO <sub>LP</sub>  | 151.15                    | 23                | 174.15                | P235GH                                  |

*Table 5.1 Materials selected for heat exchanger tubes*



## 6 STEAM DRUM DESIGN

Both the high and low pressure steam drums serve the same purpose. They physically separate the saturated steam from the water, with the help of various separators housed inside. This phase separation insured that only steam enters into the superheaters. The design of a steam drum entails defining the geometry, and then verifying that the drum load is acceptable. The outer diameter of the drum  $D_{drum}$  is chosen according to the steam production rate  $\dot{M}_{Steam}$  of the respective circuit, for steam production rates of less than 15 t/h the outer drum diameter is suggested to be 1.2 m. The length of the drum  $L_{drum}$  is chosen to be the same as the width of the flue gas duct  $L$ . The wall thickness of the steam drum  $th_{drum}$  is selected from a suggested range of between 40 and 120 mm [7].

### Equations used in the design of steam drums

#### Load verification:

$$z_{drum} < z_{max} \quad (6.1)$$

Where:

$z_{drum}$  is the steam drum load [kg/(s·m<sup>3</sup>)]

$z_{max}$  is the acceptable steam drum load [kg/(s·m<sup>3</sup>)]

#### Steam drum load:

$$z_{drum} = \frac{\dot{M}_{Steam}}{V_{drum}} \quad (6.2)$$

Where:

$\dot{M}_{Steam}$  is the steam production rate of the respective circuit [kg/s]

$V_{drum}$  is half of the steam drum volume [m<sup>3</sup>]

#### Half of the steam drum volume:

$$V_{drum} = \frac{\pi \cdot (d_{drum})^2 \cdot L_{drum}}{8} \quad (6.3)$$

Where:

$d_{drum}$  is the inner steam drum diameter [m]

$L_{drum}$  is steam drum length [m]

### 6.1 High Pressure Drum Design

The steam drum geometry is chosen according to the steam production rate of the high pressure circuit, which is 3.24 kg/s or approximately 11.67 t/h:

$$D_{drum}^{HP} = 1.2 \text{ m}$$

$$th_{drum}^{HP} = 0.04 \text{ m}$$

$$d_{drum}^{HP} = D_{drum}^{HP} - 2 \cdot th_{drum}^{HP} = 1.2 - 2 \cdot 0.04 = 1.12 \text{ m}$$

$$L_{drum}^{HP} = L = 1.76 \text{ m}$$

**Half of the steam drum volume:**

$$V_{drum}^{HP} = \frac{\pi \cdot (d_{drum}^{HP})^2 \cdot L_{drum}^{HP}}{8} = \frac{\pi \cdot 1.12^2 \cdot 1.76}{8} = 0.87 \text{ m}^3$$

**Steam drum load:**

$$z_{drum}^{HP} = \frac{\dot{M}_{Steam}^{HP}}{V_{drum}^{HP}} = \frac{3.24}{0.87} = 3.74 \text{ kg}/(\text{s} \cdot \text{m}^3)$$

**Load verification:**

The acceptable steam drum load is determined from *Table 6.1*, through linear interpolation, according to the pressure in the steam drum. This pressure has been determined in previous calculations.

$$\text{Pressure} \quad p_{drum}^{HP} = p_4^{HP} = 4.8 \text{ MPa}$$

| Parameter           | Symbol     | Units                      |       |      |      |      |     |
|---------------------|------------|----------------------------|-------|------|------|------|-----|
| Steam drum pressure | $p_{drum}$ | [MPa]                      | 0.932 | 1.47 | 2.65 | 4.22 | 6.8 |
| Acceptable load     | $z_{max}$  | [kg/(s · m <sup>3</sup> )] | 2.25  | 2.8  | 4.25 | 5.4  | 6.9 |

*Table 6.1 Acceptable steam drum load according to pressure*

$$z_{max}^{HP} = \frac{4.8 - 4.22}{6.8 - 4.22} \cdot (6.9 - 5.4) + 5.4 = 5.74 \text{ kg}/(\text{s} \cdot \text{m}^3)$$

The value of the steam drum load is lower than the acceptable load, thus the current geometry of the high pressure steam drum is acceptable.

## 6.2 Low Pressure Drum Design

The steam drum geometry is chosen according to the steam production rate of the high pressure circuit, which is 0.88 kg/s or approximately 3.18 t/h:

$$D_{drum}^{LP} = 1.2 \text{ m}$$

$$th_{drum}^{LP} = 0.04 \text{ m}$$

$$d_{drum}^{LP} = D_{drum}^{LP} - 2 \cdot th_{drum}^{LP} = 1.2 - 2 \cdot 0.04 = 1.12 \text{ m}$$

$$L_{drum}^{LP} = L = 1.76 \text{ m}$$

**Half of the steam drum volume:**

$$V_{drum}^{LP} = \frac{\pi \cdot (d_{drum}^{LP})^2 \cdot L_{drum}^{LP}}{8} = \frac{\pi \cdot 1.12^2 \cdot 1.76}{8} = 0.87 \text{ m}^3$$

**Steam drum load:**

$$z_{drum}^{LP} = \frac{\dot{M}_{Steam}^{LP}}{V_{drum}^{LP}} = \frac{0.88}{0.87} = 1.02 \text{ kg}/(s \cdot m^3)$$

**Load verification:**

The acceptable steam drum load is determined from *Table 6.1*, through linear interpolation, according to the pressure in the steam drum. This pressure has been determined in previous calculations.

$$\text{Pressure} \quad p_{drum}^{LP} = p_4^{LP} = 0.56 \text{ MPa}$$

$$z_{max}^{LP} = \frac{0.56 - 0.932}{1.47 - 0.932} \cdot (2.8 - 2.25) + 2.25 = 1.87 \text{ kg}/(s \cdot m^3)$$

The value of the steam drum load is lower than the acceptable load, thus the current geometry of the high pressure steam drum is acceptable.

**6.3 Overview of the Calculated Values**

An overview of the main values calculated in this section are presented in *Table 4.24*. They include values regarding individual heat exchanger as well as actual flue gas parameters.

| Parameter                  | Symbol      | Units                      | High pressure steam drum | Low pressure steam drum |
|----------------------------|-------------|----------------------------|--------------------------|-------------------------|
| Steam drum pressure        | $p_{drum}$  | [MPa]                      | 4.8                      | 0.56                    |
| Acceptable steam drum load | $z_{drum}$  | [kg/(s · m <sup>2</sup> )] | 3.74                     | 1.02                    |
| Steam drum load            | $z_{max}$   | [kg/(s · m <sup>2</sup> )] | 5.74                     | 1.87                    |
| Outer diameter             | $D_{drum}$  | [m]                        | 1.2                      | 1.2                     |
| Wall thickness             | $th_{drum}$ | [m]                        | 0.04                     | 0.04                    |
| Length                     | $L_{drum}$  | [m]                        | 1.76                     | 1.76                    |

*Table 6.2 Overview of calculated values*



## 7 DESIGN AND SIZING OF THE DOWNCOMER AND RISER TUBES

Both the high and low pressure circuits in this boiler depend on natural circulation for the production of saturated steam. The driving force of this circulation is the difference in density between the water in the downcomer tubes and the water/steam mixture rising through the evaporator tubes. Downcomer tubes supply water to the evaporator tubes, and riser tubes carry the water/steam mixture from the evaporator tubes back into the steam drum. The downcomer and riser tubes are sized according to their cross sectional area in relation to the cross sectional area of the respective evaporator tubes. These sizes are also dependent on the approximate height of the steam drum, or the height of the evaporator circulation loop. The calculations regarding sizing of the downcomer and riser tubes are carried out according to source [5].

### Equations used in the sizing of downcomer and riser tubes:

#### Cross sectional area of the downcomer tubes [m<sup>2</sup>]:

$$A_{Down} = A_{EV} \cdot (0.06 + 0.016 \cdot p_{drum} + 0.005 \cdot h_{drum}) \quad (7.1)$$

Where:

- $A_{EV}$  is the cross sectional area of the evaporator tubes [m<sup>2</sup>]  
 $h_{drum}$  is the approximate steam drum height [kg/(s·m<sup>3</sup>)]

#### Steam drum height:

The approximate height steam drum, or the height of the evaporator circulation loop is calculated according to equation (7.2), approximately 3 meters higher than the flue gas duct height [7].

$$h_{drum} = H + 3 \quad (7.2)$$

#### Cross sectional area of the evaporator tubes:

$$A_{EV} = \frac{\pi \cdot (d_{tube})^2}{4} \cdot n_{tube/r} \cdot n_{row} \quad (7.3)$$

#### Cross sectional area of the riser tubes [m<sup>2</sup>]:

$$A_{Riser} = A_{EV} \cdot (0.1 + 0.01 \cdot p_{drum} + 0.01 \cdot h_{drum}) \quad (7.4)$$

### 7.1 Downcomer Tube Sizing

#### Approximate high and low pressure Steam drum height:

$$h_{drum}^{HP} = h_{drum}^{LP} = H + 3 = 6.78 + 3 = 9.78 \text{ m}$$

### 7.1.1 High pressure downcomer tube sizing

#### Cross sectional area of the evaporator tubes:

$$A_{EV}^{HP} = \frac{\pi \cdot (d_{tube}^{EV1})^2}{4} \cdot n_{tube/r}^{EV1} \cdot n_{row}^{EV1} = \frac{\pi \cdot 0.049^2}{4} \cdot 16 \cdot 17 = 0.513 \text{ m}^2$$

#### Cross sectional area of the downcomer tubes:

$$A_{Down}^{HP} = A_{EV}^{HP} \cdot (0.06 + 0.016 \cdot p_{drum}^{HP} + 0.005 \cdot h_{drum}^{HP})$$

$$A_{Down}^{HP} = 0.51 \cdot (0.06 + 0.016 \cdot 4.8 + 0.005 \cdot 9.78) = 0.095 \text{ m}^2$$

The geometry of the downcomer tubes is chosen and then used to determine the number of downcomer tubes required.

| Parameter           | Symbol           | Value | Units |
|---------------------|------------------|-------|-------|
| Outer tube diameter | $D_{Down}^{HP}$  | 244.5 | [mm]  |
| Wall thickness      | $th_{Down}^{HP}$ | 10    | [mm]  |
| Inner tube diameter | $d_{Down}^{HP}$  | 224.5 | [mm]  |

Table 7.1 High pressure downcomer tube dimensions

#### Number of downcomer tubes required:

The number of downcomer tubes is rounded up to a whole number

$$n_{Down}^{HP} = \frac{A_{Down}^{HP}}{\frac{\pi \cdot (d_{Down}^{HP})^2}{4}} = \frac{0.513}{\frac{\pi \cdot 0.2245^2}{4}} = 2.4$$

$$n_{Down}^{HP} = 3$$

### 7.1.2 Low pressure downcomer tube sizing

#### Cross sectional area of the evaporator tubes:

$$A_{EV}^{LP} = \frac{\pi \cdot (d_{tube}^{EV})^2}{4} \cdot n_{tube/r}^{EV} \cdot n_{row}^{EV} = \frac{\pi \cdot 0.049^2}{4} \cdot 17 \cdot 8 = 0.256 \text{ m}^2$$

#### Cross sectional area of the downcomer tubes:

$$A_{Down}^{LP} = A_{EV}^{LP} \cdot (0.06 + 0.016 \cdot p_{drum}^{LP} + 0.005 \cdot h_{drum}^{LP})$$

$$A_{Down}^{LP} = 0.51 \cdot (0.06 + 0.016 \cdot 0.56 + 0.005 \cdot 9.78) = 0.03 \text{ m}^2$$

The geometry of the downcomer tubes is chosen and then used to determine the number of downcomer tubes required.



| Parameter           | Symbol           | Value | Units |
|---------------------|------------------|-------|-------|
| Outer tube diameter | $D_{Down}^{LP}$  | 244.5 | [mm]  |
| Wall thickness      | $th_{Down}^{LP}$ | 10    | [mm]  |
| Inner tube diameter | $d_{Down}^{LP}$  | 224.5 | [mm]  |

Table 7.2 Low pressure downcomer tube dimensions

**Number of downcomer tubes required:**

The number of downcomer tubes is rounded up to a whole number

$$n_{Down}^{LP} = \frac{A_{Down}^{LP}}{\frac{\pi \cdot (d_{Down}^{LP})^2}{4}} = \frac{0.256}{\frac{\pi \cdot 0.2245^2}{4}} = 0.76$$

$$n_{Down}^{LP} = 1$$

**7.2 Riser Tube Sizing****7.2.1 Height pressure riser tube sizing****Cross sectional area of the riser tubes:**

$$A_{Riser}^{HP} = A_{EV}^{HP} \cdot (0.1 + 0.01 \cdot p_{drum}^{HP} + 0.01 \cdot h_{drum}^{HP})$$

$$A_{Riser}^{HP} = 0.513 \cdot (0.1 + 0.01 \cdot 4.8 + 0.01 \cdot 9.78) = 0.126 \text{ m}^2$$

The geometry of the riser tubes is chosen and then used to determine the number of riser tubes required.

| Parameter           | Symbol            | Value | Units |
|---------------------|-------------------|-------|-------|
| Outer tube diameter | $D_{Riser}^{HP}$  | 133   | [mm]  |
| Wall thickness      | $th_{Riser}^{HP}$ | 8     | [mm]  |
| Inner tube diameter | $d_{Riser}^{HP}$  | 117   | [mm]  |

Table 7.3 High pressure riser tube dimensions

**Number of riser tubes required:**

The number of riser tubes is rounded up to a whole number

$$n_{Riser}^{HP} = \frac{A_{Riser}^{HP}}{\frac{\pi \cdot (d_{Riser}^{HP})^2}{4}} = \frac{0.126}{\frac{\pi \cdot 0.117^2}{4}} = 11.73$$

$$n_{Riser}^{HP} = 12$$

## 7.2.2 Low pressure riser tube sizing

### Cross sectional area of the riser tubes:

$$A_{Riser}^{LP} = A_{EV}^{LP} \cdot (0.1 + 0.01 \cdot p_{drum}^{LP} + 0.01 \cdot h_{drum}^{LP})$$

$$A_{Riser}^{LP} = 0.513 \cdot (0.1 + 0.01 \cdot 0.56 + 0.01 \cdot 9.78) = 0.052 \text{ m}^2$$

The geometry of the riser tubes is chosen and then used to determine the number of riser tubes required.

| Parameter           | Symbol            | Value | Units |
|---------------------|-------------------|-------|-------|
| Outer tube diameter | $D_{Riser}^{HP}$  | 168.3 | [mm]  |
| Wall thickness      | $th_{Riser}^{HP}$ | 8     | [mm]  |
| Inner tube diameter | $d_{Riser}^{HP}$  | 152.3 | [mm]  |

Table 7.4 Low pressure riser tube dimensions

### Number of riser tubes required:

The number of riser tubes is rounded up to a whole number

$$n_{Riser}^{HP} = \frac{A_{Riser}^{HP}}{\frac{\pi \cdot (d_{Riser}^{HP})^2}{4}} = \frac{0.052}{\frac{\pi \cdot 0.1523^2}{4}} = 2.86$$

$$n_{Riser}^{HP} = 3$$

## 8 FLUE GAS PRESSURE LOSSES THROUGH THE BOILER

The total flue gas pressure loss through the boiler  $\Delta p_{L:total}$  must be lower than the maximum acceptable pressure loss  $\Delta p_{Lmax} = 1500 Pa$ . The total flue gas pressure loss is calculated as the sum of the pressure losses across each heat exchanger and the pressure loss across the stack.

### 8.1 Flue Gas Pressure Losses across Individual Heat Exchangers

Calculation regarding pressure losses across the individual heat exchangers are carried out according to source [3], calculation regarding heat exchangers with smooth walls are calculated according to source [5].

#### Equations used to determine flue gas pressure losses:

**Pressure loss across a heat exchanger (with finned or smooth tubes):**

$$\Delta p_L = \zeta \cdot \frac{(W_{Flue})^2}{2} \cdot \rho_{Flue} \quad (8.1)$$

Where:

- $\Delta p_L$  is flue gas pressure loss across individual heat exchangers [Pa]
- $\zeta$  is the pressure loss coefficient, determined from the geometry of the heat exchanger [-]

**The pressure loss coefficient (for heat exchangers with finned tubes):**

$$\zeta = k_4 \cdot n_{row} \cdot \left(\frac{h_{fin}}{D_{tube}}\right)^{k_1} \cdot \left(\frac{s_{fin}}{D_{tube}}\right)^{-k_2} \cdot (Re)^{-k_3} \quad (8.2)$$

Where:

- $Re$  is the Reynolds number of the flue gas [-]
- $k_{1,2,3,4}$  are constants, which depend on the configuration of the tubes in the heat exchangers [-]

For heat exchangers with a staggered tube configuration the values of the constants used are presented in *Table 8.1*.

| Constant  | $k_1$ | $k_2$ | $k_3$ | $k_4$ |
|-----------|-------|-------|-------|-------|
| Value [-] | 0     | 0.72  | 0.24  | 2     |

*Table 8.1 Pressure loss coefficient constants*

**The pressure loss coefficient (for heat exchangers with smooth tubes):**

This equation is applicable for cases where  $s_1/D_{tube} < s_2/D_{tube}$

$$\zeta = (4 + 6.6 \cdot n_{row}) \cdot (Re)^{-0.28} \quad (8.3)$$

**Reynolds number:**

$$Re = \frac{(W_{Flue} \cdot d_e)^2}{\nu_{Flue}} \quad (8.4)$$

Where:

$d_e$  is equivalent diameter [m]

$\nu_{Flue}$  is the kinematic viscosity of the flue gas, according to the average temperature of the tarnished outer tube walls [m<sup>2</sup>/s]

**Equivalent diameter of the flue gas duct:**

$$d_{e:duct} = \frac{4 \cdot A_{E:duct}}{PER_{duct}} = \frac{4 \cdot L \cdot H}{2 \cdot (L + H)} = \frac{4 \cdot 1.76 \cdot 6.78}{2 \cdot (1.76 + 6.78)} = 2.79 \text{ m} \quad (8.5)$$

Where:

$A_{E:duct}$  is the cross sectional area of the flue gas duct [m<sup>2</sup>]

$PER_{duct}$  Is the perimeter of the flue gas duct form the view of the cross section [m]

**Average temperature of the tarnished outer tube walls:**

$$t_{wall} = \bar{t}_{Steam} + \Delta t \quad (8.6)$$

Where:

$\bar{t}_{Steam}$  is the average temperature of the water/steam inside the tubes [Pa]

$\Delta t$  is a given temperature difference, for gashouse fuels ( $\Delta t = 25^\circ\text{C}$ ) [°C]

Values that will be used to calculate the flue gas pressure drop across the individual heat exchangers are organized in *Table 8.2* below. This table also includes the Reynolds number and outer tube wall surface temperature, which have been calculated according to the equations above.

|               | Parameters         | Outer tube diameter | Fin height | Fin spacing | Flue gas speed | Number of lateral rows | Average wall temperature | kinematic viscosity  | Reynolds number   |
|---------------|--------------------|---------------------|------------|-------------|----------------|------------------------|--------------------------|----------------------|-------------------|
|               | Symbol             | $D_{tube}$          | $h_{fin}$  | $s_{fin}$   | $W_{Flue}$     | $n_{row}$              | $t_{wall}$               | $\nu_{Flue}$         | $Re$              |
|               | Units              | [mm]                | [mm]       | [mm]        | [m/s]          | [-]                    | [°C]                     | [m <sup>2</sup> /s]  | [-]               |
| High pressure | SH <sub>1HP</sub>  | 31.8                | 15         | 9.09        | 10             | 4                      | 424.23                   | $6.36 \cdot 10^{-5}$ | $1.23 \cdot 10^7$ |
|               | SH <sub>2HP</sub>  | 31.8                | 15         | 7.69        | 9.73           | 4                      | 359.93                   | $5.41 \cdot 10^{-5}$ | $1.37 \cdot 10^7$ |
|               | EV <sub>HP</sub>   | 57                  | 19         | 6.67        | 10.45          | 17                     | 238.9                    | $4.33 \cdot 10^{-5}$ | $1.97 \cdot 10^7$ |
|               | ECO <sub>3HP</sub> | 31.8                | 8          | 5.26        | 10.27          | 9                      | 228.9                    | $3.61 \cdot 10^{-5}$ | $2.28 \cdot 10^7$ |
|               | ECO <sub>2HP</sub> | 33.7                | 8          | 5           | 9.6            | 2                      | 170.4                    | $2.91 \cdot 10^{-5}$ | $2.47 \cdot 10^7$ |
|               | ECO <sub>1HP</sub> | 33.7                | 12         | 4.55        | 7.71           | 7                      | 121.2                    | $2.37 \cdot 10^{-5}$ | $1.96 \cdot 10^7$ |
| Low           | SH <sub>LP</sub>   | 38                  | -          | -           | 7.15           | 1                      | 188.08                   | $3.11 \cdot 10^{-5}$ | $1.28 \cdot 10^7$ |
|               | EV <sub>LP</sub>   | 57                  | 19         | 4.35        | 9.27           | 8                      | 178.65                   | $3 \cdot 10^{-5}$    | $2.23 \cdot 10^7$ |
|               | ECO <sub>LP</sub>  | 22                  | 15         | 4.35        | 5.74           | 2                      | 127.08                   | $2.44 \cdot 10^{-5}$ | $1.06 \cdot 10^7$ |

Table 8.2 Values for calculations regarding flue gas pressure losses

### 8.1.1 Pressure losses across superheater SH<sub>2HP</sub>

The pressure loss coefficient for SH<sub>2HP</sub>:

$$\zeta_{SH2} = k_4 \cdot n_{row}^{SH2} \cdot \left( \frac{h_{fin}^{SH2}}{D_{tube}^{SH2}} \right)^{k_1} \cdot \left( \frac{s_{fin}^{SH2}}{D_{tube}^{SH2}} \right)^{-k_2} \cdot (Re_{(SH2)})^{-k_3}$$

$$\zeta_{SH2} = 2 \cdot 4 \cdot \left( \frac{15}{31.8} \right)^0 \cdot \left( \frac{9.09}{31.8} \right)^{-0.72} \cdot (1.23 \cdot 10^7)^{-0.24} = 0.39$$

Pressure loss across SH<sub>2HP</sub>:

$$\Delta p_L^{SH2} = \zeta_{SH2} \cdot \frac{(W_{Flue}^{SH2})^2}{2} \cdot \rho_{Flue} = 0.39 \cdot \frac{10^2}{2} \cdot 1.2772 = 25.04 \text{ Pa}$$

### 8.1.2 Pressure losses across superheater SH<sub>1HP</sub>

The pressure loss coefficient for SH<sub>1HP</sub>:

$$\zeta_{SH1} = k_4 \cdot n_{row}^{SH1} \cdot \left( \frac{h_{fin}^{SH1}}{D_{tube}^{SH1}} \right)^{k_1} \cdot \left( \frac{s_{fin}^{SH1}}{D_{tube}^{SH1}} \right)^{-k_2} \cdot (Re_{(SH1)})^{-k_3}$$

$$\zeta_{SH1} = 2 \cdot 4 \cdot \left( \frac{15}{31.8} \right)^0 \cdot \left( \frac{7.69}{31.8} \right)^{-0.72} \cdot (1.37 \cdot 10^7)^{-0.24} = 0.43$$

**Pressure loss across SH<sub>1HP</sub>:**

$$\Delta p_L^{SH1} = \zeta_{SH1} \cdot \frac{(W_{Flue}^{SH1})^2}{2} \cdot \rho_{Flue} = 0.43 \cdot \frac{9.73^2}{2} \cdot 1.2772 = 26.04 \text{ Pa}$$

**8.1.3 Pressure losses across evaporator EV<sub>HP</sub>****The pressure loss coefficient for EV<sub>HP</sub>:**

$$\zeta_{EV1} = k_4 \cdot n_{row}^{EV1} \cdot \left(\frac{h_{fin}^{EV1}}{D_{tube}^{EV1}}\right)^{k_1} \cdot \left(\frac{S_{fin}^{EV1}}{D_{tube}^{EV1}}\right)^{-k_2} \cdot (Re_{(EV1)})^{-k_3}$$

$$\zeta_{EV1} = 2 \cdot 17 \cdot \left(\frac{19}{57}\right)^0 \cdot \left(\frac{6.67}{57}\right)^{-0.72} \cdot (1.97 \cdot 10^7)^{-0.24} = 2.83$$

**Pressure loss across EV<sub>HP</sub>:**

$$\Delta p_L^{EV1} = \zeta_{EV1} \cdot \frac{(W_{Flue}^{EV1})^2}{2} \cdot \rho_{Flue} = 2.83 \cdot \frac{10.45^2}{2} \cdot 1.2772 = 197.3 \text{ Pa}$$

**8.1.4 Pressure losses across economizer ECO<sub>3HP</sub>****The pressure loss coefficient for ECO<sub>3HP</sub>:**

$$\zeta_{ECO3} = k_4 \cdot n_{row}^{ECO3} \cdot \left(\frac{h_{fin}^{ECO3}}{D_{tube}^{ECO3}}\right)^{k_1} \cdot \left(\frac{S_{fin}^{ECO3}}{D_{tube}^{ECO3}}\right)^{-k_2} \cdot (Re_{(ECO3)})^{-k_3}$$

$$\zeta_{ECO3} = 2 \cdot 9 \cdot \left(\frac{8}{31.8}\right)^0 \cdot \left(\frac{5.26}{31.8}\right)^{-0.72} \cdot (2.28 \cdot 10^7)^{-0.24} = 1.13$$

**Pressure loss across ECO<sub>3HP</sub>:**

$$\Delta p_L^{ECO3} = \zeta_{ECO3} \cdot \frac{(W_{Flue}^{ECO3})^2}{2} \cdot \rho_{Flue} = 1.13 \cdot \frac{10.27^2}{2} \cdot 1.2772 = 75.96 \text{ Pa}$$

**8.1.5 Pressure losses across economizer ECO<sub>2HP</sub>****The pressure loss coefficient for ECO<sub>2HP</sub>:**

$$\zeta_{ECO2} = k_4 \cdot n_{row}^{ECO2} \cdot \left(\frac{h_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)^{k_1} \cdot \left(\frac{S_{fin}^{ECO2}}{D_{tube}^{ECO2}}\right)^{-k_2} \cdot (Re_{(ECO2)})^{-k_3}$$

$$\zeta_{ECO2} = 2 \cdot 2 \cdot \left(\frac{8}{33.7}\right)^0 \cdot \left(\frac{5}{33.7}\right)^{-0.72} \cdot (2.47 \cdot 10^7)^{-0.24} = 0.27$$

**Pressure loss across ECO<sub>2HP</sub>:**

$$\Delta p_L^{ECO2} = \zeta_{ECO2} \cdot \frac{(W_{Flue}^{ECO2})^2}{2} \cdot \rho_{Flue} = 0.27 \cdot \frac{9.6^2}{2} \cdot 1.2772 = 15.65 \text{ Pa}$$

### 8.1.6 Pressure losses across economizer ECO<sub>1HP</sub>

The pressure loss coefficient for ECO<sub>1HP</sub>:

$$\zeta_{ECO1} = k_4 \cdot n_{row}^{ECO1} \cdot \left( \frac{h_{fin}^{ECO1}}{D_{tube}^{ECO1}} \right)^{k_1} \cdot \left( \frac{S_{fin}^{ECO1}}{D_{tube}^{ECO1}} \right)^{-k_2} \cdot (Re_{(ECO1)})^{-k_3}$$

$$\zeta_{ECO1} = 2 \cdot 7 \cdot \left( \frac{12}{33.7} \right)^0 \cdot \left( \frac{4.55}{33.7} \right)^{-0.72} \cdot (1.96 \cdot 10^7)^{-0.24} = 1.05$$

Pressure loss across ECO<sub>1HP</sub>:

$$\Delta p_L^{ECO1} = \zeta_{ECO1} \cdot \frac{(W_{Flue}^{ECO1})^2}{2} \cdot \rho_{Flue} = 1.05 \cdot \frac{7.71^2}{2} \cdot 1.2772 = 40.03 \text{ Pa}$$

### 8.1.7 Pressure losses across superheater SH<sub>LP</sub>

The pressure loss coefficient for SH<sub>LP</sub>:

$$\zeta_{SH} = (4 + 6.6 \cdot n_{row}^{SH}) \cdot (Re_{(SH)})^{-0.28} = (4 + 6.6 \cdot 1) \cdot (1.28 \cdot 10^7)^{-0.28} = 0.11$$

Pressure loss across SH<sub>LP</sub>:

$$\Delta p_L^{SH} = \zeta_{SH} \cdot \frac{(W_{Flue}^{SH})^2}{2} \cdot \rho_{Flue} = 0.11 \cdot \frac{10^2}{2} \cdot 1.2772 = 3.53 \text{ Pa}$$

### 8.1.8 Pressure losses across evaporator EV<sub>LP</sub>

The pressure loss coefficient for EV<sub>LP</sub>:

$$\zeta_{EV} = k_4 \cdot n_{row}^{EV} \cdot \left( \frac{h_{fin}^{EV}}{D_{tube}^{EV}} \right)^{k_1} \cdot \left( \frac{S_{fin}^{EV}}{D_{tube}^{EV}} \right)^{-k_2} \cdot (Re_{(EV)})^{-k_3}$$

$$\zeta_{EV} = 2 \cdot 8 \cdot \left( \frac{19}{57} \right)^0 \cdot \left( \frac{4.35}{57} \right)^{-0.72} \cdot (2.23 \cdot 10^7)^{-0.24} = 1.76$$

Pressure loss across EV<sub>LP</sub>:

$$\Delta p_L^{EV} = \zeta_{EVH} \cdot \frac{(W_{Flue}^{EV})^2}{2} \cdot \rho_{Flue} = 1.76 \cdot \frac{9.27^2}{2} \cdot 1.2772 = 96.41 \text{ Pa}$$

### 8.1.9 Pressure losses across economizer ECO<sub>LP</sub>

The pressure loss coefficient for ECO<sub>LP</sub>:

$$\zeta_{ECO} = k_4 \cdot n_{row}^{ECO} \cdot \left( \frac{h_{fin}^{ECO}}{D_{tube}^{ECO}} \right)^{k_1} \cdot \left( \frac{S_{fin}^{ECO}}{D_{tube}^{ECO}} \right)^{-k_2} \cdot (Re_{(ECO)})^{-k_3}$$

$$\zeta_{ECO} = 2 \cdot 2 \cdot \left( \frac{15}{22} \right)^0 \cdot \left( \frac{4.35}{22} \right)^{-0.72} \cdot (1.06 \cdot 10^7)^{-0.24} = 0.27$$

**Pressure loss across ECO<sub>LP</sub>:**

$$\Delta p_L^{ECO} = \zeta_{ECO} \cdot \frac{(W_{Flue}^{ECO})^2}{2} \cdot \rho_{Flue} = 0.27 \cdot \frac{5.74^2}{2} \cdot 1.2772 = 5.58 \text{ Pa}$$

**8.1.10 Flue gas pressure losses across all the heat exchangers in the boiler**

The pressure losses across all the heat exchangers in the boiler are calculated intuitively as the sum of the pressure losses across the individual heat exchangers, as seen in equation (8.7)

$$\Delta p_{L1} = \Delta p_L^{SH2} + \Delta p_L^{SH1} + \Delta p_L^{EV1} + \Delta p_L^{ECO3} + \Delta p_L^{ECO2} + \Delta p_L^{ECO1} + \Delta p_L^{SH} + \Delta p_L^{EV} + \Delta p_L^{ECO} \quad (8.7)$$

$$\Delta p_{L1} = 25.04 + 26.04 + 197.3 + 75.96 + 15.65 + 40.03 + 3.53 + 96.41 + 5.58 = 485.53 \text{ Pa}$$

**8.2 Flue Gas Pressure Losses across the Stack**

The total pressure loss across the stack is the sum of the local pressure losses at the stack inlet, outlet, across the stack silencer, and the pressure loss due to the friction of the inner surface of the stack. It is also important to take into account the stack effect (draft effect) which is counteracting the pressure losses. Chosen values that are used in the calculations regarding pressure losses are organized in *Table 8.3*.

| Parameter  | Symbol                | Value  | Units                 |
|--|-----------------------|--------|-----------------------|
| Functional stack height                                | $H_{stack}$           | 16     | [m]                   |
| Inner stack diameter                                   | $d_{stack}$           | 1.05   | [m]                   |
| Coefficient of friction inside the stack (sheet metal) | $\lambda_{stack}$     | 0.03   | [-]                   |
| Inlet pressure loss coefficient                        | $\zeta_{stack}^{in}$  | 1      | [-]                   |
| Outlet pressure loss coefficient                       | $\zeta_{stack}^{out}$ | 1      | [-]                   |
| Pressure loss across the stack silencer                | $\Delta p_L^{sil}$    | 200    | [Pa]                  |
| Stack inlet temperature                                | $t_j$                 | 116.16 | [°C]                  |
| Density of air   | $\rho_{air}$          | 1.275  | [kg/Nm <sup>3</sup> ] |
| Acceleration due to gravity                            | $g$                   | 9.81   | [m/s <sup>2</sup> ]   |

*Table 8.3 Values for stack pressure loss calculations*

**8.2.1 Pressure loss across the stack due to friction**

The pressure loss across the stack due to friction is calculated according to equation (8.8).

$$\Delta p_L^{fric} = \lambda_{stack} \cdot \frac{H_{stack}}{d_{stack}} \cdot \frac{(W_{Flue})^2}{2} \cdot \rho_{Flue} \quad (8.8)$$



Where:

- $\lambda_{stack}$  is coefficient of friction inside the stack [-]  
 $H_{stack}$  is functional stack height [m]  
 $d_{stack}$  is the inner stack diameter [m]

The speed of the flue gas in the stack is calculated intuitively according to the cross sectional area of the stack and the actual volumetric flow rate of the flue gas.

Cross sectional area of the stack:

$$A_{stack} = \frac{\pi \cdot (d_{stack})^2}{4} = \frac{\pi \cdot 1.05^2}{4} = 0.86 \text{ m}^2$$

Actual volumetric flow rate of the flue gas in the stack:

$$\dot{M}_{VFlue}^{stack} = \dot{M}_{VFlue} \cdot \frac{t_j + 273.15}{273.15} = 23.49 \cdot \frac{116.16 + 273.15}{273.15} = 33.48 \text{ m}^3/\text{s}$$

Speed of the flue gas in the stack:

$$W_{Flue}^{stack} = \frac{\dot{M}_{VFlue}^{stack}}{A_{stack}} = \frac{33.48}{0.86} = 38.93 \text{ m/s}$$

Pressure loss due to friction:

$$\Delta p_L^{fric} = \lambda_{stack} \cdot \frac{H_{stack}}{d_{stack}} \cdot \frac{(W_{Flue}^{stack})^2}{2} \cdot \rho_{Flue} = 0.03 \cdot \frac{16}{1.05} \cdot \frac{38.93^2}{2} \cdot 1.2772 = 109.09 \text{ Pa}$$

## 8.2.2 Local pressure losses at the inlet and outlet of the stake

The pressure losses at the inlet and outlet of the stack are calculated in the same manner as the calculation of pressure losses across the individual heat exchangers, where equation (8.1) was used.

$$\Delta p_L^{in:out} = (\zeta_{stack}^{in} + \zeta_{stack}^{out}) \cdot \frac{(W_{Flue}^{stack})^2}{2} \cdot \rho_{Flue} = (1 + 1) \cdot \frac{38.93^2}{2} \cdot 1.2772 = 477.28 \text{ Pa}$$

## 8.2.3 Stack effect

The stack effect acts against the pressure losses and aids in the expulsion of cooled flue gas from the boiler. The stack effect is calculated according to equation (8.9).

$$\Delta p_{eff} = H_{stack} \cdot \left( \rho_{air} - \rho_{Flue} \cdot \frac{273.15}{273.15 + t_j} \right) \cdot g \quad (8.9)$$

$$\Delta p_{eff} = 16 \cdot \left( 1.275 - 1.2772 \cdot \frac{273.15}{273.15 + 116.16} \right) \cdot 9.81 = 59.53 \text{ Pa}$$

### 8.2.4 Overall pressure loss across the stack

The overall stack pressure loss is calculated, as seen in equation (8.10), as the sum of the losses at the inlet and outlet, the loss across the stack silencer, the losses due to friction, and the stack effect opposing these losses.

$$\begin{aligned}\Delta p_L^{stack} &= \Delta p_L^{fric} + \Delta p_L^{in:out} + \Delta p_L^{sil} - \Delta p_{eff} \\ \Delta p_L^{stack} &= 109.09 + 477.28 + 200 - 59.53 = 726.84 \text{ Pa}\end{aligned}\tag{8.10}$$

### 8.3 Overall Flue Gas Pressure Losses through the HRSG

The overall flue gas pressure drop is calculated according to equation (8.11), as the sum of the overall pressure loss across the stack and the pressure loss across all the heat exchangers in the boiler.

$$\Delta p_{L:total} = \Delta p_L^{stack} + \Delta p_{L1} = 726.84 + 485.53 = 1212.38 \text{ Pa}\tag{8.11}$$

The total pressure loss is lower than the maximum acceptable flue gas pressure loss of 1500Pa. Thus the stack dimensions and the configuration of the heat exchangers in the boiler are acceptable.

## CONCLUSION

The aim of this thesis was to design a dual pressure horizontal heat recovery steam generator (HRSG), utilizing flue gas from a natural gas burning turbine, including the sizing of the heat exchangers. The initial step in the design of this boiler was choosing the general configuration of the heat exchangers in the boiler, the number of heat exchangers and the order in which they are arranged. The steam production rates of both the high and low pressure circuits, along with the theoretical heat transfer rates required of each heat exchanger were calculated according to the given flue gas parameters and the required steam outlet parameters.

The geometry of the first heat exchanger to come in contact with the flue gas (superheater SH<sub>2HP</sub>) was selected according to the required heat transfer rate and the suggested steam and flue gas flow speeds. The inner dimensions of the HRSG casing (flue gas duct dimensions) were then determined according to the geometry of superheater SH<sub>2HP</sub>. The geometry of the remaining heat exchangers, along with the specific geometry of the tubes and tube layout, were determined according to the dimensions of the flue gas duct and the heat transfer requirements of each individual heat exchanger. The discrepancy between the required heat transfer rates and the actual heat transfer rates, resulting from the geometry of each heat exchanger, was kept below 2%. The parameters of the flue gas exiting each heat exchanger were accurately recalculated according to the actual heat transfer rate of the heat exchanger. Materials for the heat exchanger tubes were then selected according to the temperature of the water/steam exiting each heat exchanger.

Both the high and low pressure steam drums were sized according to the respective high and low pressure steam production rates. The steam drum loads were then calculated and verified to be lower than the limiting load value for the given steam drum pressure. The downcomer and riser tubes were also sized for both circuits to insure natural circulation of the water from the steam drum into the evaporator, through the evaporator tubes, where it becomes a water/steam mixture and back into the steam drum. The final step in the design of this dual pressure HRSG was to calculate the flue gas pressure losses across the heat exchangers and the stack, and to verify that the total pressure loss does not exceed the maximum allowable pressure loss of 1500 Pa.

The assembly drawing of the dual pressure HRSG (seen in appendix O-DP-HRSG/01) shows the main dimensions of the HRSG and its components, including the general layout of the individual heat exchangers inside the HRSG casing, which has been split into six modules for transportation purposes.



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## LIST OF SYMBOLS

### Abbreviations:    Meaning:

|                    |  |
|--------------------|--|
| HRSG               | heat recovery steam generator                          |
| HP                 | high pressure circuit                                  |
| LP                 | low pressure circuit                                   |
| <br>               |  |
| ECO <sub>LP</sub>  | low pressure economizer                                |
| ECO <sub>1HP</sub> | first heat exchanger of the high pressure economizer   |
| ECO <sub>2HP</sub> | second heat exchanger of the high pressure economizer  |
| ECO <sub>3HP</sub> | third heat exchanger of the high pressure economizer   |
| EV <sub>LP</sub>   | low pressure evaporator                                |
| EV <sub>HP</sub>   | high pressure evaporator                               |
| SH <sub>LP</sub>   | low pressure superheater                               |
| SH <sub>1HP</sub>  | first heat exchanger of the high pressure superheater  |
| SH <sub>2HP</sub>  | second heat exchanger of the high pressure superheater |

### Index:            Meaning:

|              |  |
|--------------|--|
| 1,2, ...,9   | points on the temperature profile graph (water/steam)  |
| A, B, ..., J | points on the temperature profile graph (flue gas)     |
| <br>         |  |
| <i>HP</i>    | high pressure circuit                                  |
| <i>LP</i>    | low pressure circuit                                   |
| <i>ECO</i>   | low pressure economizer                                |
| <i>ECO1</i>  | first heat exchanger of the high pressure economizer   |
| <i>ECO2</i>  | second heat exchanger of the high pressure economizer  |
| <i>ECO3</i>  | third heat exchanger of the high pressure economizer   |
| <i>EV</i>    | low pressure evaporator                                |
| <i>EV1</i>   | high pressure evaporator                               |
| <i>SH</i>    | low pressure superheater                               |
| <i>SH1</i>   | first heat exchanger of the high pressure superheater  |
| <i>SH2</i>   | second heat exchanger of the high pressure superheater |
| <br>         |  |
| <i>Ar</i>    | argon  |
| <i>CO2</i>   | carbon dioxide   |
| <i>H2O</i>   | water  |
| <i>N2</i>    | nitrogen   |
| <i>O2</i>    | oxygen   |

|              |                      |
|--------------|----------------------|
| <i>drum</i>  | steam drum           |
| <i>duct</i>  | flue gas duct        |
| <i>Down</i>  | downcomer tubes      |
| <i>fin</i>   | tube fins            |
| <i>Flue</i>  | flue gas             |
| <i>real</i>  | actual value         |
| <i>Riser</i> | riser tubes          |
| <i>stack</i> | HRSG stack           |
| <i>Steam</i> | water/steam          |
| <i>tube</i>  | heat exchanger tubes |

| <b>Symbol:</b>      | <b>Meaning:</b>   | <b>Units:</b>     |
|---------------------|---|-------------------|
| $\alpha_c$          | convective heat transfer coefficient outside the heat exchanger tubes (find tubes)                    | $[W/(m^2K)]$      |
| $\alpha_{c1}$       | convective heat transfer coefficient outside the heat exchanger tubes (smooth tubes)                  | $[W/(m^2K)]$      |
| $\alpha_{in}$       | heat transfer coefficient inside the heat exchanger tubes (smooth tubes)                              | $[W/(m^2K)]$      |
| $\alpha_{out}$      | heat transfer coefficient outside the heat exchanger tubes (smooth tubes)                             | $[W/(m^2K)]$      |
| $\alpha_r$          | coefficient of heat transfer through radiation outside the tubes of the heat exchanger (smooth tubes) | $[W/(m^2K)]$      |
| $\alpha_{r:in}$     | reduced heat transfer coefficient inside heat exchanger tubes (find tubes)                            | $[W/(m^2K)]$      |
| $\alpha_{r:out}$    | reduced heat transfer coefficient outside heat exchanger tubes (find tubes)                           | $[W/(m^2K)]$      |
| $\beta$             | coefficient   | $[1/m]$           |
| $\varepsilon$       | correction coefficient for fin fouling  | $[m^2K/W]$        |
| $\lambda_{fin}$     | thermal conductivity of the fins  | $[W/(m \cdot K)]$ |
| $\lambda_{Flue}$    | thermal conductivity of the flue gas  | $[W/(m \cdot K)]$ |
| $\lambda_{stack}$   | coefficient of friction inside the stack  | $[-]$             |
| $\lambda_{Steam}$   | thermal conductivity of the water/steam   | $[W/(m \cdot K)]$ |
| $\mu$               | coefficient characterizing fin taper  | $[-]$             |
| $\mu_{Steam}^{SH}$  | dynamic viscosity of the water/steam  | $[Pa \cdot s]$    |
| $\nu_{Flue}$        | kinematic viscosity of the flue gas   | $[m^2/s]$         |
| $\nu_{Steam}^{SH2}$ | kinematic viscosity of the water/steam  | $[m^2/s]$         |
| $\xi$               | coefficient of the degree to which the flue gas is utilized   | $[-]$             |
| $\rho_{air}$        | density of air  | $[kg/Nm^3]$       |
| $\rho_{Flue}$       | densities of individual flue gas  | $[kg/Nm^3]$       |
| $\rho_{O2,...H2O}$  | densities of individual flue gas components   | $[kg/Nm^3]$       |



|                       |   |                     |
|-----------------------|---|---------------------|
| $\rho_{Steam}$        | density of the water/steam  | $[kg/m^3]$          |
| $\zeta$               | pressure loss coefficient   | $[-]$               |
| $\zeta_{stack}^{in}$  | stack inlet pressure loss coefficient   | $[-]$               |
| $\zeta_{stack}^{out}$ | stack outlet pressure loss coefficient  | $[-]$               |
| $\sigma_1$            | lateral tube spacing relative to the outer tube diameter  | $[-]$               |
| $\sigma_2'$           | diagonal tube spacing relative to the outer tube diameter   | $[-]$               |
| $\varphi_\sigma$      | coefficient characterizing the relative tube spacing  | $[-]$               |
| $\Psi$                | coefficient of thermal effectiveness (smooth tubes)   | $[-]$               |
| $\Psi_{fin}$          | coefficient characterizing the uneven distribution of the convective heat transfer coefficient ( $\alpha_c$ ) along the surface of the fins | $[-]$               |
| $a$                   | lateral gap between tubes, or finned tubes where applicable   | $[m]$               |
| $a_{wall}$            | coefficient of emissivity of the tube walls   | $[-]$               |
| $a_{Flue}$            | coefficient of emissivity of the flue gas   | $[-]$               |
| $A$                   | cross sectional area  | $[m^2]$             |
| $A_{duct}$            | cross sectional area between the heat exchanger tubes and the walls of the flue gas duct  | $[m^2]$             |
| $A_{E:duct}$          | cross sectional area of the empty flue gas duct   | $[m^2]$             |
| $A_{EV}$              | cross sectional area of the evaporator tubes  | $[m^2]$             |
| $b$                   | coefficient of radiation decline  | $[1/(m \cdot MPa)]$ |
| $b_{tri}$             | coefficient of radiation decline caused by triatomic gases  | $[1/(m \cdot MPa)]$ |
| $c_l$                 | correction coefficient dependent on the tube length relative to the tube diameter   | $[-]$               |
| $c_m$                 | correction coefficient for fluid flow between concentric tubes  | $[-]$               |
| $c_s$                 | correction coefficient characterizing the layout of the heat exchanger tubes (smooth tubes)   | $[-]$               |
| $c_t$                 | correction coefficient dependent on the temperature of the water/steam and tube wall temperature  | $[-]$               |
| $c_z$                 | correction coefficient for depending on the number of lateral rows  | $[-]$               |
| $c_{z1}$              | correction coefficient depending on the number of lateral rows (smooth tubes)   | $[-]$               |
| $C$                   | constant for calculating the rate of heat loss through radiation and convection   | $[-]$               |
| $d$                   | inner diameter  | $[m]$               |
| $d_e$                 | equivalent diameter   | $[m]$               |
| $D$                   | outer diameter  | $[m]$               |
| $E$                   | coefficient characterizing the effectiveness of the fins  | $[-]$               |
| $\%Error$             | discrepancy between the theoretical and actual heat transfer rates  | $[\%]$              |
| $g$                   | acceleration due to gravity   | $[m/s^2]$           |

|                       |  |                        |
|-----------------------|--|------------------------|
| $gap$                 | gap between the duct wall and the outer tubes  | [m]                    |
| $h_{drum}$            | is the approximate steam drum height   | [m]                    |
| $h_{fin}$             | fin height   | [m]                    |
| $H$                   | height of the flue gas duct  | [m]                    |
| $H_{stack}$           | functional stack height  | [m]                    |
| $i$                   | enthalpy of the steam/water  | [kJ/kg]                |
| $\Delta i$            | enthalpy difference  | [kJ/kg]                |
| $I$                   | enthalpy of the flue gas   | [kJ/Nm <sup>3</sup> ]  |
| $I'$                  | recalculated enthalpy of the flue gas  | [kJ/Nm <sup>3</sup> ]  |
| $k$                   | is the overall heat transfer coefficient   | [W/(m <sup>2</sup> K)] |
| $k_{1,2,3,4}$         | constants depending on tube configuration  | [–]                    |
| $L$                   | width of the flue gas duct   | [m]                    |
| $L_{drum}$            | is steam drum length   | [m]                    |
| $\dot{M}_{Flue}$      | mass flow rate of flue gas   | [kg/s]                 |
| $\dot{M}_{Steam}$     | mass flow rate of steam (steam production rate)  | [kg/s]                 |
| $\dot{M}_{VFlue}$     | volumetric flow rate of flue gas   | [Nm <sup>3</sup> /s]   |
| $\dot{M}_{W\%}$       | percentage of high pressure feedwater used to regulate outlet steam parameters (water injection) | [%]                    |
| $n$                   | number tubes   | [–]                    |
| $n_{fin}$             | number of fins per meter tube length   | [–]                    |
| $n_{row}$             | number of lateral row of tubes   | [–]                    |
| $n_{tube/r}$          | number of lateral tubes per lateral row  | [–]                    |
| $p$                   | pressure   | [MPa]                  |
| $p_1$                 | flue gas pressure (assuming atmospheric)   | [MPa]                  |
| $p_{p:tri}$           | partial pressure of triatomic gases in the flue gas  | [MPa]                  |
| $\bar{p}$             | average pressure   | [MPa]                  |
| $\Delta p$            | water/steam pressure losses through the hrsg   | [MPa]                  |
| $\Delta p_{eff}$      | flue gas pressure difference due to the stack effect   | [Pa]                   |
| $\Delta p_L$          | flue gas pressure losses across individual heat exchangers                                       | [Pa]                   |
| $\Delta p_{L1}$       | flue gas pressure loss across all of the heat exchangers   | [Pa]                   |
| $\Delta p_L^{fric}$   | flue gas pressure loss across the stack due to friction  | [Pa]                   |
| $\Delta p_{Lmax}$     | maximum acceptable flue gas pressure loss  | [Pa]                   |
| $\Delta p_L^{in:out}$ | local flue gas pressure losses at the stack inlet and outlet                                     | [Pa]                   |
| $\Delta p_L^{sil}$    | flue gas pressure loss across the stack silencer   | [Pa]                   |
| $\Delta p_L^{stack}$  | the overall flue gas pressure loss across the stack  | [Pa]                   |
| $\Delta p_{L:total}$  | overall flue gas pressure losses through the hrsg  | [Pa]                   |

|                               |  |                   |
|-------------------------------|--|-------------------|
| $PER_{duct}$                  | perimeter of the flue gas duct cross section   | [m]               |
| $Pr$                          | prandtl number   | [-]               |
| $Q$                           | rate of heat transfer  | [W]               |
| $Q_N$                         | maximum usable thermal power   | [W]               |
| $Q_{RC}$                      | rate of heat loss through radiation and convection   | [W]               |
| $Q_{RC\%}$                    | percentage of usable thermal power that is lost through radiation and convection                                 | [W]               |
| $Re$                          | reynolds number  | [-]               |
| $s$                           | effective radiation layer thickness  | [m]               |
| $s_{fin}$                     | fin spacing  | [m]               |
| $s_1$                         | lateral tube spacing   | [m]               |
| $s_2$                         | longitudinal tube spacing  | [m]               |
| $s'$                          | diagonal tube spacing  | [m]               |
| $S_{1fin}$                    | outer surface area of one fin  | [m <sup>2</sup> ] |
| $S_{out}$                     | theoretical outer surface area of all the heat exchanger tubes needed to meet the heat transfer requirements     | [m <sup>2</sup> ] |
| $S_{out}^{real}$              | actual outer surface area of all the heat exchanger tubes  | [m <sup>2</sup> ] |
| $S_{out/1m}$                  | the outer surface area of one tube per meter length  | [m <sup>2</sup> ] |
| $S_{out/r}$                   | the outer surface area of all the tubes in one lateral row   | [m <sup>2</sup> ] |
| $S_{in/1m}$                   | the inner surface area of one tube per meter length  | [m <sup>2</sup> ] |
| $\frac{S_{fin}}{S_{out}}$     | the fraction of the total outer surface area of the finned tubes that is made up of the fins themselves          | [-]               |
| $\frac{S_{out-fin}}{S_{out}}$ | the fraction of the total outer surface area of the finned tubes that is made up of the exposed outer tube walls | [-]               |
| $t$                           | temperature  | [°C]              |
| $t_{Flue}$                    | inlet flue gas temperature   | [°C]              |
| $t_{FW}$                      | feedwater temperature  | [°C]              |
| $t_{ref}$                     | reference temperature for material selection   | [°C]              |
| $t_{Steam:out}$               | temperature of water/steam exiting the heat exchanger  | [°C]              |
| $t_{wall}$                    | average temperature of the tarnished outer tube walls  | [°C]              |
| $th$                          | thickness or wall thickness  | [m]               |
| $th_{tube(mm)}$               | tube wall thickness  | [mm]              |
| $t'$                          | recalculated temperature   | [°C]              |
| $\bar{t}$                     | average temperature  | [°C]              |
| $\Delta t$                    | temperature difference   | [°C]              |
| $\Delta t_{Ap}$               | approach point   | [°C]              |
| $\Delta t_{ln}$               | logarithmic mean temperature difference  | [K]               |

|                   |   |                            |
|-------------------|---|----------------------------|
| $\Delta t_{pi}$   | pinch point   | [°C]                       |
| $\Delta t_{safe}$ | safety margin   | [°C]                       |
| $\Delta t_1$      | temperature difference between the flue gas entering and the water/steam exiting the heat exchanger | [K]                        |
| $\Delta t_2$      | temperature difference between the flue gas exiting and the water/steam entering the heat exchanger | [K]                        |
| $T$               | the absolute temperature  | [K]                        |
| $\bar{v}$         | average specific volume   |                            |
| $V_{duct}$        | half of the steam drum volume   | [m <sup>3</sup> ]          |
| $W$               | speed of flow   | [m/s]                      |
| $W_{steam}'$      | water/steam speed before splitting lateral rows into sections                                       | [m/s]                      |
| $x$               | volumetric concentration  | [-]                        |
| $x_{tri}$         | volumetric concentration of triatomic gases in the flue gas   | [-]                        |
| $z_{drum}$        | steam drum load   | [kg/(s · m <sup>3</sup> )] |
| $z_{max}$         | acceptable steam drum load  | [kg/(s · m <sup>3</sup> )] |

## LIST OF FIGURES

- Figure 0-1 Combined cycle power plants [10]
- Figure 1-1 HRSG schematic
- Figure 2-1 HRSG preliminary temperature profile graph
- Figure 2-2 Feedwater injection
- Figure 2-3 HRSG Temperature profile
- Figure 3-1 Tube geometry SH<sub>2HP</sub>
- Figure 4-1 Tube spacing
- Figure 4-2 Tube layout SH<sub>2HP</sub>
- Figure 4-3 Tube geometry SH<sub>1HP</sub>
- Figure 4-4 Tube layout SH<sub>1HP</sub>
- Figure 4-5 Tube geometry EV<sub>HP</sub>
- Figure 4-6 Tube layout EV<sub>HP</sub>
- Figure 4-7 Tube geometry ECO<sub>3HP</sub>
- Figure 4-8 Tube layout ECO<sub>3HP</sub>
- Figure 4-9 Splitting schematic ECO<sub>3HP</sub>
- Figure 4-10 Tube geometry SH<sub>LP</sub>
- Figure 4-11 Tube layout SH<sub>LP</sub>
- Figure 4-12 Tube geometry EV<sub>LP</sub>
- Figure 4-13 Tube layout EV<sub>LP</sub>
- Figure 4-14 Tube geometry ECO<sub>2HP</sub>
- Figure 4-15 Tube layout ECO<sub>2HP</sub>
- Figure 4-16 Splitting schematic ECO<sub>2HP</sub>
- Figure 4-17 Tube geometry ECO<sub>LP</sub>
- Figure 4-18 Tube layout ECO<sub>LP</sub>
- Figure 4-19 Splitting schematic ECO<sub>LP</sub>
- Figure 4-20 Tube geometry ECO<sub>1HP</sub>
- Figure 4-21 Tube layout ECO<sub>1HP</sub>
- Figure 4-22 Splitting schematic ECO<sub>1HP</sub>

## LIST OF TABLES

|            |   |
|------------|---|
| Table 1.1  | Volumetric composition, concentrations  |
| Table 2.2  | Selected parameters values  |
| Table 0.3  | Densities of individual flue gas components   |
| Table 2.4  | Enthalpy of the flue gas (calculated) and flue gas components at normal conditions, a temperature of 0°C and pressure of 0.101 MPa.   |
| Table 2.5  | Water/steam parameters at various points throughout each circuit  |
| Table 2.6  | Flue gas parameters at various points throughout the boiler   |
| Table 2.7  | Heat transfer rates of individual heat exchangers in the high and low pressure circuits   |
| Table 2.8  | Other important values  |
| Table 3.1  | Parameters selected for the finned tubes used in the high pressure superheater (SH <sub>2HP</sub> )                                   |
| Table 3.9  | Calculated values defining duct and superheater (SH <sub>2HP</sub> ) geometry, as well as actual flow speeds.                         |
| Table 4.10 | Collective values used in the heat transfer calculations  |
| Table 4.11 | Flue gas coefficient of thermal conductivity, $\lambda_{\text{Flue}} \cdot 10^3$ [W/(m·K)]  |
| Table 4.12 | Flue gas coefficient of kinematic viscosity, $\nu_{\text{Flue}} \cdot 10^6$ [m <sup>2</sup> /s]                                       |
| Table 4.13 | Parameters necessary for superheater SH <sub>2HP</sub> design, not including parameters describing the geometry of SH <sub>2HP</sub>  |
| Table 4.14 | Values describing the geometry of superheater SH <sub>2HP</sub>   |
| Table 4.15 | Parameters necessary for superheater SH <sub>1HP</sub> design, not including parameters describing the geometry of SH <sub>1HP</sub>  |
| Table 4.16 | Parameters selected for the finned tubes used in the high pressure superheater SH <sub>1HP</sub>                                      |
| Table 4.17 | Parameters necessary for evaporator EV <sub>HP</sub> design, not including parameters describing the geometry of EV <sub>HP</sub>     |
| Table 4.18 | Parameters selected for the finned tubes used in the high pressure evaporator EV <sub>HP</sub>  |
| Table 4.19 | Parameters necessary for economizer ECO <sub>3HP</sub> design, not including parameters describing the geometry of ECO <sub>3HP</sub> |
| Table 4.20 | Parameters selected for the finned tubes used in the high pressure economizer ECO <sub>3HP</sub>                                      |
| Table 4.21 | Values used in the calculations for heat exchangers with smooth tubes   |
| Table 4.22 | Prandtl number of the flue gas, $Pr_{\text{Flue}}$ [-]  |
| Table 4.23 | Parameters necessary for superheater SH <sub>LP</sub> design, not including parameters describing the geometry of SH <sub>LP</sub>    |

|            |   |
|------------|---|
| Table 4.24 | Parameters selected for the smooth tubes used in the high pressure superheater $SH_{LP}$                                |
| Table 4.25 | Parameters necessary for evaporator $EV_{LP}$ design, not including parameters describing the geometry of $EV_{LP}$     |
| Table 4.26 | Parameters selected for the finned tubes used in the high pressure evaporator $EV_{LP}$                                 |
| Table 4.27 | Parameters necessary for economizer $ECO_{2HP}$ design, not including parameters describing the geometry of $ECO_{2HP}$ |
| Table 4.28 | Parameters selected for the finned tubes used in the high pressure economizer $ECO_{2HP}$                               |
| Table 4.29 | Parameters necessary for economizer $ECO_{LP}$ design, not including parameters describing the geometry of $ECO_{LP}$   |
| Table 4.30 | Parameters selected for the finned tubes used in the low pressure economizer $ECO_{LP}$                                 |
| Table 4.31 | Parameters necessary for economizer $ECO_{1HP}$ design, not including parameters describing the geometry of $ECO_{1HP}$ |
| Table 4.32 | Parameters selected for the finned tubes used in the high pressure economizer $ECO_{1HP}$                               |
| Table 4.33 | Parameters regarding individual heat exchangers   |
| Table 4.34 | Actual flue gas parameters at various points throughout the boiler  |
| Table 5.35 | Materials selected for heat exchanger tubes   |
| Table 6.36 | Acceptable steam drum load according to pressure  |
| Table 6.2  | Overview of calculated values   |
| Table 7.37 | High pressure downcomer tube dimensions   |
| Table 7.38 | Low pressure downcomer tube dimensions  |
| Table 7.39 | High pressure riser tube dimensions   |
| Table 7.40 | Low pressure riser tube dimensions  |
| Table 8.41 | Pressure loss coefficient constants   |
| Table 8.42 | Values for calculations regarding flue gas pressure losses  |
| Table 8.43 | Values for stack pressure loss calculations   |





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**APPENDIX**

0-DP-HRGS/01      STEAM BOILER (HRSG)