Experimental Research of Flue Gas Condensing Unit

EDGARS VĪGANTS*, DAGNIJA BLUMBERGA, IVARS VEIDENBERGS, ĢIRTS VĪGANTS, CLAUDIO ROCHAS
Institute of Energy Systems and Environment
Riga Technical University
Kronvalda blvd. 1, LV-1010, Riga
LATVIA
edgars.vigants@rtu.lv www.videszinatne.lv

* - corresponding author. Riga Technical University, Institute of Energy Systems and Environment, Kronvalda blvd. 1, LV-1010, Riga, Latvia. Mob. +371 29212789, Fax +371 67089908

Abstract: - A flue gas condenser is a system that helps to increase the effectiveness of boiler plant operations by condensing vapour from flue gases. Flue gas condensers can be used in boiler plants that burn gaseous and liquid fuels as well as biomass. This article analyses the operations of a empty flue gas condenser without package elements in a wood chip fired boiler plant. The study begins with a model of the processes in a flue gas condenser and the selection of constructive parameters. The system was installed in 2010. Data from the industrial experiment were obtained, mathematically processed, and evaluated for uncertainty.

Keywords: - Gas condenser, Chip fuelled boiler plant, Energy efficiency improvement, Industrial experiment, Renewable energy, Condenser capacity

1. Introduction

Member states of the European Union have resolved to reduce Europe's dependence on imported fuels and to increase the proportion of renewable energy resources in the energy balances of the individual states. This can be achieved by acquiring new technologies for the use of renewable energy resources, but also by effectively exploiting those energy resources that are already currently in use.

One of these resources is wood fuel, the use of which is often quite inefficient. This can especially be observed in furnaces that burn damp wood. Wood chips can have widely differing levels of moisture content, from the 25% standard envisioned by the European Union to snow-covered wood with a moisture content of over 55% and sometimes even 60%.

There are two ways of increasing the effectiveness of boiler rooms relying on wood chips as fuel. The first option is to dry the wood chips before feeding them into the furnace. The organisation of the drying process depends on the specific needs of the boiler room. A dryer must be used to obtain wood chips with low moisture content, but dryers consume relatively large amounts of energy that cannot be used for other purposes. A second option involves the use of green or damp wood chips in the furnace and the installation of a flue gas condenser behind the boiler system.

The combustion process in the furnace changes the water content in the fuel into water vapour. The vaporisation process of this water consumes heat, and each kg of vapour carries ~2500 kJ of heat away with it. A flue gas condenser recovers this heat from the vapour by condensing it. The higher the moisture content of the fuel, the more heat can be recovered in a flue gas condenser.

Several complex heat and mass transfer processes take place in the flue gas condenser simultaneously: heat conduction, convection, vaporisation, and condensation [1]. Based on process modelling, which is gone into more detail in the authors' publication [2], this article analyses the operations of an already established flue gas condenser.

2. Description of the condenser and the industrial experiment

The flue gas condenser is installed in a boiler room behind the hot water boiler, which has a capacity of 8 MW. The boiler furnace burns wet wood chips. The furnace has slanted coils upon which the fuel goes through all the stages of combustion: the heating of the wood chips, vaporisation of moisture, combustion of volatile substances, and combustion of coke. After exiting the furnace, the flue gases are cooled by water flowing through the heating surfaces of the boiler. After the boiler, the flue gases reach the condenser, which consists of two connected parts: the horizontal vaporiser, in which the sprayed water vaporises, and the vertical condenser, in which the droplets of sprayed water act as condensing surfaces. Some of the measuring gauges are built into the condenser; others are distributed among the systems in the boiler room. The gauge indicators are placed on a control board, where the levels can be read. The placement of gauges in the system diagram is shown in Figure 1.

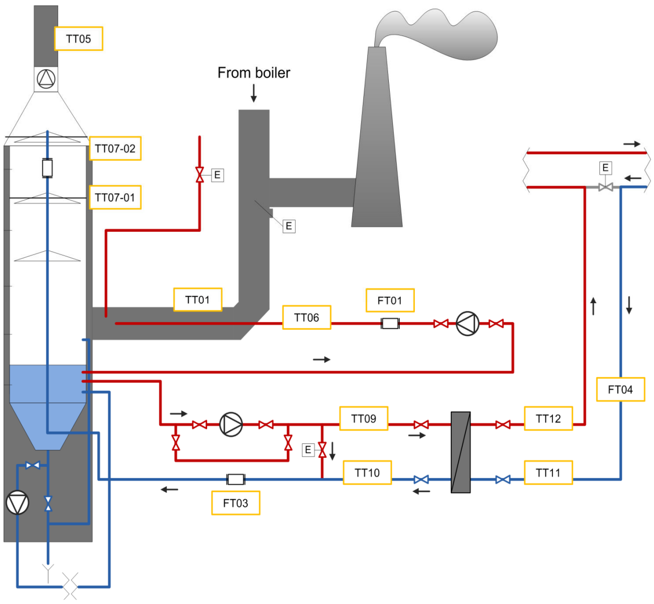

Fig.1 Diagram of the placement of measuring gauges in the system

System descriptions and measurement parameters used for measurements in the diagram:

- ***temperature sensors:***
1) TT01 – flue gas temperature by boiler t_{1g};
2) TT05 – flue gas temperature by condenser t_{2g};
3) TT06 – temperature of water sprayed into horizontal scrubber t_{in};
4) TT07-01, TT07-02 – temperature of sprayed water t_{2c};
5) TT09 – water temperature by condenser t_{1c};
6) TT10 – water temperature by net heat exchanger;
7) TT11 – return water temperature of the heating nets t_{1n};
8) TT12 – outgoing water temperature of the heating nets t_{2n}.

- ***flow measurements:***
1) FT01 – amount of water sprayed into horizontal scrubber G_{in};
2) FT03 – amount of water sprayed into vertical scrubber G_c;
3) FT04 – amount of water in heating net G_n.

Readings of measurements are taken once every three hours and recorded in a log. The set of data processed for this study is from 201 operating regimes.

Combined list of quantities established from measurements and their units of measurement:

- boiler capacity N_b, MW;
- condenser capacity N_c, MW;
- amount of sprayed water G_c, m^3/h;
- flue gas temperature before and after condenser t_{1g}; t_{2g}; ^0C;
- temperature of sprayed water t_{2c}, ^0C;
- water temperature after condenser t_{1c}, ^0C;
- return water temperature of heating net t_{1n}, ^0C;
- oxygen content in flue gases O_2, %.

A calculation programme of the combustion process establishes the following for each experimental regime:

- flow of dry flue gases L_{dg}, kg/s;
- flue gas moisture content before condenser x, kg/kg dry gas.

To calculate the density of condenser spraying, the following equation is used:

$$H_s = G_c/(3.6 \cdot F_c),\ kg/(m^2 \cdot s), \quad (1)$$

where

F_c – area of condenser cross-section, m^2.

Data regarding the boiler's efficiency and fuel combustion heat are obtained from boiler check logs.

3. Method for processing empirical data

The empirical data were processed using the correlation analysis and regression analysis methods of statistical processing for data. Correlation analysis determined the link between two quantities in the form of an equation and the closeness of this link as the correlation coefficient square R^2. The results are shown in the form of graphs. Regression analysis was used to determine the statistical significance of the multiple regression model and its coefficients [3]. The statistical significance of the model's coefficients is evaluated with t–statistics, if the confidence level is 95%.

The statistical processing of the data and the creation of a multiple empirical model were done with the STATGRAPHICS Plus computer programme.

3.1. Regression analysis of the data from the industrial condenser experiment

ISBN: 978-1-61804-122-7

The goal of a regression analysis is to obtain a multiple empirical equation that gives a quantitative description of the condenser capacity depending on typical and statistically significant indices of the boiler system as well as to serve as a basis for the prognosis and evaluation of results of the system's operating regimes.

The results of a regression analysis are correct if the necessary conditions for its use have been met. There are several main conditions for the use of regression analysis, some of which can be tested before the regression analysis, while others can be tested only after obtaining the regression equation. Whether the dependent variable quantity – the capacity of the flue gas heat utilisation system condenser N_c – conforms to a normal distribution rule is tested before the data processing.

Several essential issues must always be addressed when creating empirical models in the form of regression equations. Does the model include all the independent variables that characterise the phenomenon under consideration, and are there no superfluous, unimportant variable quantities that might make the model unnecessarily complicated? The answer to these questions can be found by evaluating the statistical significance of the quantities included in the model and by performing a dispersion analysis [4]. The study results in a regression equation that determines the condenser capacity depending on the moisture content of the flue gas x, the spray density H_s, the boiler capacity N_b, the flue gas temperature before the condenser t_{1g}, the water temperature after the condenser, t_{1c}, the return water temperature of the heating nets t_{1n}, and the temperature of the water sprayed into the condenser t_{2c}.

$$N_c = -0.763+1.126\cdot x+0.0423\cdot H_s+0.02\cdot N_b+ 0.00051\cdot t_{1g}+0.0183\cdot t_{1c}-0.0213\cdot t_{1n}+0.0092\cdot t_{2c} \qquad (2)$$

The statistical processing of the data from the created empirical model results in determining the value of R^2 as 0.96. This means that the created model (2) explains 96% of the changes in the analysed data. The remaining 4% can be attributed to the effect of the independent variables not included in or not defined by the equation or to the mutual influence between them.

Other tests of the correct use of a regression analysis, such as autocorrelation, multicollinearity, and heteroscedasticity tests, can be performed after the data has been processed. It has been determined that no autocorrelation is observed because the Durbin–Watson statistic is 1.79. The multicollinearity evaluation shows that no mutual correlation between the independent variable parameters in the multiple regression equation is observed. A coefficient correlation matrix with no correlation coefficients greater than 0.5 was used for the analysis. The heteroscedasticity test was performed by graphically analysing the distribution of the remainder depending on the independent variable parameters. The graphs do not show changes in the remainder variations depending on changes in the value parameters, and this proves that the standard data error was calculated correctly. Together, the tests indicate that the use of multiple regression analysis in the processing of experimental data is correct.

One of the essential issues in using empirical equations is whether and to what extent the results of a regression equation correlate with the empirical data. Only in the case of a satisfactory correlation is it possible to assert that the model adequately describes the actual situation as observed in practice and that its use in the modelling of the situation is correct. To test the adequacy of the empirical equation, actual condenser capacity measurements are compared to data about condenser capacity calculated by the equation (2). The data from this comparison are shown in the graph in Figure 2.

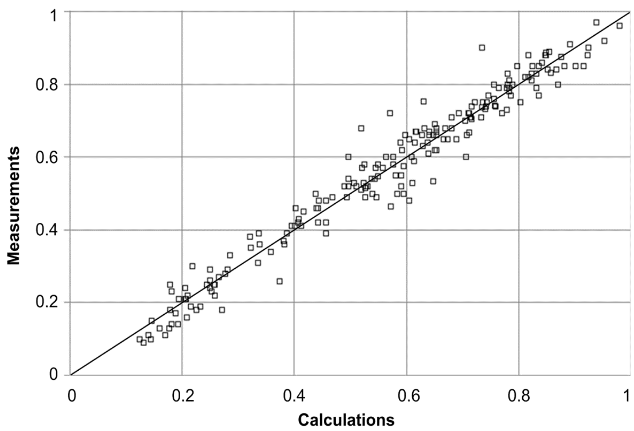

Fig.2 Comparison of data from condenser capacity measurements and calculations

A good correlation between both sets of data can be observed. There is a slightly greater dispersion around in the middle values of condenser capacity.

3.2. Uncertainty of measurements

Information regarding measurement results and calculated quantities based on those results is complete only if the range of the uncertainty of measurements is indicated when reporting about the obtained value. This means that the actual value of the amount is found within the indicated range.

Uncertainty is shown as an expanded uncertainty that corresponds to a 95% level of confidence [5].
The uncertainty of condenser capacity measurements obtained in the experiment is evaluated with a B type evaluation method [6]. This means that the measured quantity is subjected to an equal probability rectangular distribution rule and the coverage factor is 1.65 [7].
The measurement equation for determining condenser capacity is the expression:

$$N_c = 1.164 \cdot G_c \cdot (t_{1c} - t_{2c}),\ kW \qquad (3)$$

As seen in the expression, the main sources of uncertainty are:

- temperature measurements of water introduced into the heat exchanger t_{1c};
- temperature measurements of water cooled in the heat exchanger t_{2c};
- measurements of water flow through net heat exchanger G_c.

Calculations of uncertainty show that the expanded uncertainty with a coverage factor of 1.65 is 30 kW if the condenser capacity is 200 kW and $N_c = 200 \pm 30$ kW. If the condenser capacity is 800 kW, the expanded uncertainty is 72 kW and $N_c = 800 \pm 72$ kW.
The difference for a boiler capacity of 7 MW is 18%. A comparison of results shows that the congruence is satisfactory and the results of the modelling can be used to explain changes in the local parameters of the condenser.

4. Analysis of data from the experimental study of the condenser

The deep cooling of flue gases and recovery of heat has its own specific character. It is not enough to analyse only the operations of the boiler and the condenser itself, in which the deep cooling of the flue gases takes place. The recovered heat energy has a low potential (temperature) and a consumer must be found for it. Such a consumer could be a heat supply system that uses the recovered heat for primary heating of water within the net. The influence of a heat supply system on the use of the heat is connected with the temperature level in the heating nets [8]. The return temperature is significant. Changes in condenser capacity depending on the return temperature of net water can be seen in Figure 3.
As the return temperature of net water rises, the heat capacity of the condenser decreases because the heat transfer between the heating net and the condenser's water circuits in the net water heat exchange weakens. The water heated in the condenser is cooled less effectively in the net heat exchange and is sprayed through nozzles into the condenser at a higher temperature. When the temperature of the sprayed water is higher, the dry heat transfer between the droplets and the flue gases as well as the mass transfer both weaken. In the case of condensation, the partial pressure of the water vapour on the surface of the water droplet is lower than the partial pressure in the flue gases. As the temperature of the water droplets rises, so does the partial pressure of the vapour on the surface of the droplet, and the difference between the partial pressures decreases. As a result, the condensation process weakens and the flue gases exit the condenser with higher moisture content. A weakened dry heat transfer correlates to a higher temperature of the exiting flue gases. On the whole, a less intense heat and mass transfer results in a poorer use of flue gas heat and decreased efficiency of the boiler room [9].

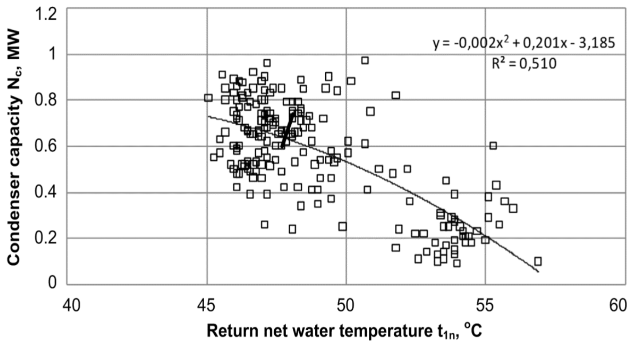

Fig.3 Changes in condenser capacity depending on the return temperature of net water

Water heated in the direct contact heat exchange is sprayed through nozzles into the upper section of the system. The flue gases and sprayed water move in counterflow and a heat and mass transfer between the two environments takes place throughout the entire condenser. The amount of sprayed water is characterised by the density of spray.

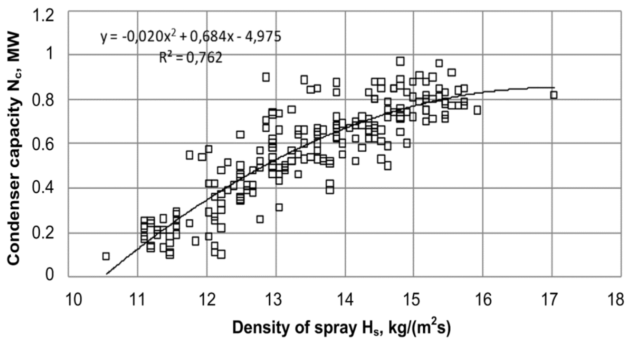

Fig.4 Changes in condenser capacity depending on density of spray

Changes calculated in the experiment of condenser capacity depending on the density of spray can be seen in Figure 4. As the density of the spray increases, condenser capacity increases non-linearly. In the case of a greater amount of sprayed water, its final heating temperature t_{1c} and spray temperature t_{2c} is decreased. When the droplet surface temperature is lower, the partial pressure of the vapour on the surface is lower and the difference in partial pressure on the droplet surface and in the flue gases increases and the mass transfer is intensified [10]. In the case of a lower sprayed water temperature t_{2c}, the difference in temperature between the flue gases and the water increases and the dry heat transfer between the gases and the water increases. The processes of dry heat transfer as well as mass transfer are such that the flue gases are more deeply cooled and heat recovery in the condenser increases.

The capacity of the wood chip fired boiler changed from 1 MW to 6.5 MW during the industrial experiment. Changes in the condenser capacity depending on the capacity of the boiler are shown in Figure 5.

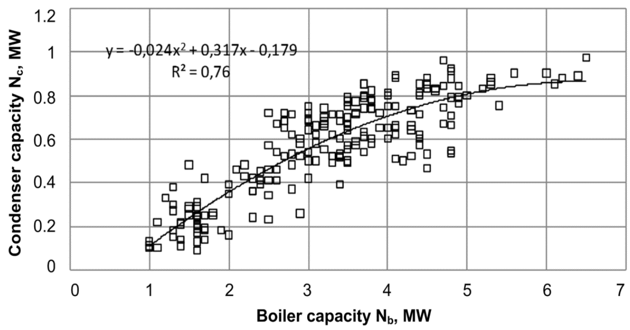

Fig.5 Changes in condenser capacity depending on boiler capacity

Changes in condenser capacity depending on boiler capacity are mainly influenced by two factors: changes in the amount of exiting flue gases and changes in flue gas temperature. As boiler capacity increases, fuel consumption also increases. If the amount of oxygen in the flue gases and the moisture content of the fuel are kept constant, the amount of flue gases is directly proportional to fuel consumption, which is in turn inversely proportional to the efficiency of the boiler.

$$V_{fg} = \frac{N_b}{\eta_b \cdot LHV} \cdot V_g = B \cdot V_g \;\; m^3/s, \qquad (4)$$

where

N_b – boiler capacity, kW;

η_b – boiler efficiency;

LHV – lowest temperature for combustion of fuel;

V_g – amount of flue gas per kg of fuel burned, m^3/kg;

B – fuel consumption, kg/s.

Efficiency values in modern boilers stay approximately the same for a nominal to a 25% load. Efficiency values at lower loads decrease non-linearly.

Condenser capacity in the case of deep cooling of flue gases is determined as

$$N_c = L_{dg} \cdot (h_1 - h_2), \; kW, \qquad (5)$$

where

h_1 – flue gas enthalpy before condenser, kJ/kg dry gas;

h_2 – flue gas enthalpy after condenser, kJ/kg dry gas.

The flow of dry flue gases L_{dg} depends on the amount of flue gases V_g and their moisture content. The enthalpy of wet gases is determined by their temperature and moisture content. The parameters are linked by the equation:

$$h_g = c_p \cdot t_g + x \cdot (2500 + 1.93 \cdot t_g), \; kJ/kg \; dry \; gas, \qquad (6)$$

where

c_p – specific thermal capacity of dry flue gases, kJ/(kg dry gas·K);

x – moisture content of flue gases, kg/kg dry gas.

As the amount of flue gases increases, so does the speed of gases in the condenser. Because there is a counterflow between the sprayed water and flue gases, intense turbulence in the gas and water mixture can be noticed as the speed of the gases increases. As a result of this turbulence, the heat and mass transfer between both environments intensifies and the recovered heat in the condenser increases.

The figure shows that the changes in condenser capacity are non-linear. This is explained by the character of the changes in boiler efficiency and flue gas temperature depending on boiler capacity.

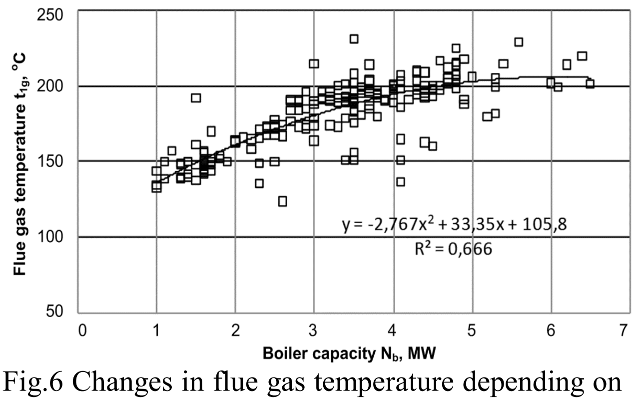

Fig.6 Changes in flue gas temperature depending on boiler capacity

Changes calculated by the experiment in flue gas temperature after leaving the boiler can be seen in Figure 6. At greater boiler capacities the furnace simultaneously burns more fuel, more heat is discharged, and combustion temperature increases. As a result the temperature of exiting flue gases also increases.

On the whole, as boiler capacity increases so does the potential for usable heat – the amount of heat in the flue gases increases as does the temperature, resulting in a non-linear increase in condenser capacity. The condenser capacity at small boiler capacities is 17% of the boiler capacity; condenser capacity is 14% of boiler capacity for boilers with a capacity of 6 MW.

Condenser operations are associated with flue gas cooling below the dew point temperature. Such deep cooling of flue gases results in moisture condensation and phase transition (vaporisation) heat is recovered. The amount of heat that can be recovered as a result of condensation depends on the moisture content of the flue gases, the temperature of the sprayed water, the spray density, the time flue gases have spent in the condenser, and other conditions [11]. Data from the industrial experiment regarding changes in condenser capacity depending on the moisture content of flue gases in the post-condenser entry are shown in Figure 7.

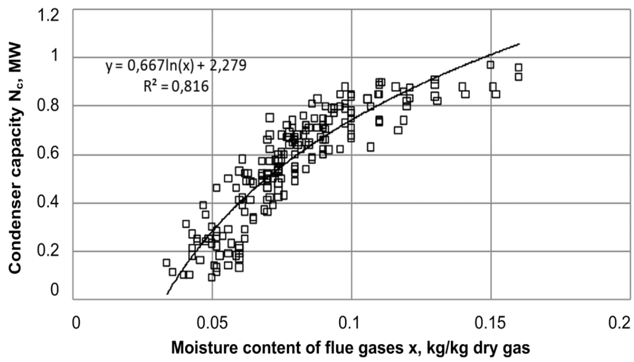

Fig.7 Changes in condenser capacity depending on moisture content of flue gases

An increasing tendency in capacity correlating to an increase in the moisture content of the flue gases can be observed. A dispersion of data can also be seen, which can be explained by the simultaneous influence of other above-mentioned parameters. The regression equation obtained as the result of processing the experimental data is used to isolate the influence of other parameters. Other parameters in the calculations are considered constant.

The effects of the flue gas moisture content on the heat capacity of the condenser as evaluated by the regression equation can be seen in Figure 8.

The effect of moisture content is evaluated at boiler capacities of 2 MW and 6 MW. The temperature of the sprayed water t_{2c} in both cases is 55 ^{0}C. The temperature of flue gases in the condenser entry is assumed to correspond with the boiler capacities from Figure 6. The density of the sprayed water is 12 kg/(m^2·s) for a boiler capacity of 2 MW and 16 kg/(m^2·s) for a capacity of 6 MW.

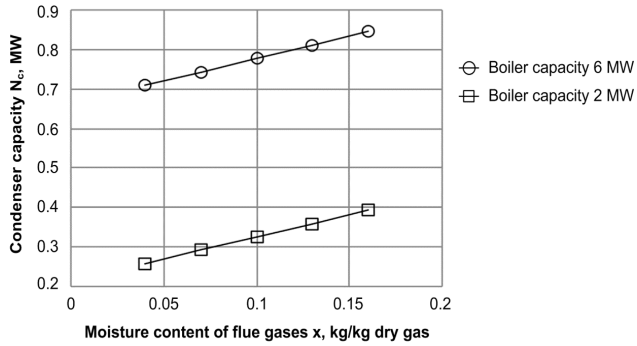

Fig.8 Changes in condenser capacity depending on the moisture content of flue gases after leaving the boiler

The figure shows that as the moisture content of the flue gases increases, so does the condenser capacity, because the flue gases contain a larger amount of vapour, the condensation of which results in the recovery of a greater amount of phase transition heat. But, in order for vapour condensation to take place, appropriate conditions must be created. The temperature of the sprayed water must be such that the partial pressure of vapour on the surface of the sprayed water droplets is lower than the partial pressure of saturated vapour in the gases. The temperature of the sprayed water t_{2c} is in turn affected by the heat transfer in the net water heater. The lower the return temperature of the net water, the better the sprayed water is cooled and the lower the t_{2c} temperature value. In order for condensation to take place in the entire vertical space of the scrubber, the temperature of the sprayed water must be lower than the dew point temperature of the flue gases t_d in the entire scrubber. The dew point temperature of the flue gases is determined by the temperature of the flue gases and their moisture content. In the case of flue gas and water counterflow in direct contact condensers with no filler elements, the flue gas temperature is 7 – 10 ^{0}C higher than the water temperature. Lower temperature differences can be observed in condensers with filler elements.

5. Analysis of condenser modelling data

An industrial experiment is able to determine the parameters in the condenser entry and exit and thereby evaluate its operation on the whole. A more detailed analysis of parameter changes is done using a two-part model calculating the process of heat and mass transfer [2]. The first part describes the processes in the horizontal scrubber of the condenser, in which flue gases are introduced after the boiler. A concurrent flow of the sprayed water and flue gases is achieved in the first section. After the gases have been processed in the horizontal scrubber, they move to the vertical scrubber, where heat recovery from the flue gases takes place.

Modelling assumes the boiler capacity is 6 MW, the flue gas temperature after leaving the boiler is 200 ^{0}C, and the moisture content of the flue gases after leaving the boiler is 0.04 kg/kg dry gas. This can be observed in situations where extra air is fed to the burning fuel. From the standpoint of heat and mass transfer, the task of the horizontal scrubber is primarily to cool the flue gases to 90 ^{0}C and raise the moisture content of the gases to the maximum, which is achieved by spraying water with a temperature higher than the dew point temperature of the flue gases. Modelling assumes the temperature of the sprayed water is 62 ^{0}C. The parameter changes in the horizontal scrubber can be seen in Figure 9.

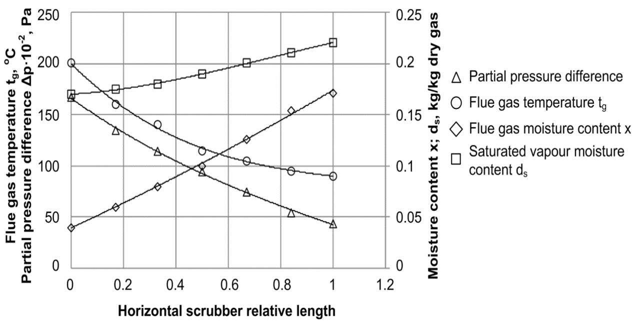

Fig.9 Parameter changes in the horizontal scrubber obtained through modelling

Sprayed water droplets vaporise and flue gases become saturated with water vapour in the scrubber [12]. It is essential that the condition of flue gases in the horizontal scrubber exit be such that, no matter the moisture content of the flue gases after leaving the boiler, moisture condenses in the vertical part of the condenser [13]. The driving force of the vaporisation process is the difference between the partial pressure of saturated vapour from the sprayed water and the partial pressure of vapour in the gases. The figure shows that the difference in partial pressures in the scrubber decreases and the vaporisation process is more intense in the entry to the scrubber. The difference between partial pressures is proportional to the difference between the moisture content in the saturated condition d_s and the flue gas moisture content x. In the case of vaporisation, $d_s > x$ and mass transfer takes place from the water droplets to flue gases. As a result, the gases introduced into the vertical scrubber have higher moisture content than the gases leaving the boiler.

In order to evaluate the parameter changes in the vertical scrubber of the condenser, modelling of the processes was performed. The results of this modelling are shown in Figures 10 and 11.

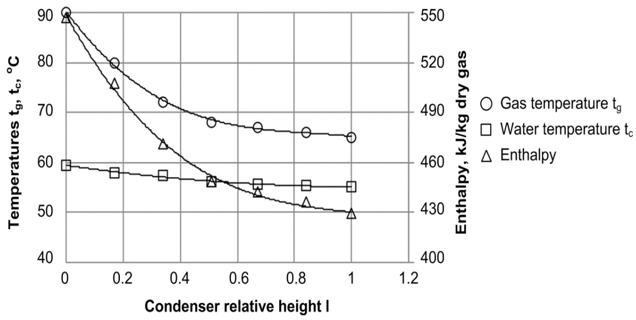

Fig.10 Modelled changes in temperature and enthalpy in the vertical scrubber

Condensation of moisture in the flue gases takes place in the vertical scrubber. In order for this condensation to take place, the temperature of the sprayed water (55 ^{0}C) must be lower than the dew point (58.5 ^{0}C) in the area of spraying. Water in the scrubber flows from top to bottom against the flue gases and is heated to 60 ^{0}C. The dew point of flue gases in the entry of the vertical scrubber is 61.8 ^{0}C. This means that moisture condensation takes place throughout the entire scrubber and phase transition heat is recovered [14, 15]. As a result of the condensation, the quantity of sprayed water increases by the quantity of condensed vapour. This means that the amount of water in the condenser circuit increases and, if water droplets are not carried away with the exiting flue gases, then a part of the water must be taken out of the condenser. As the temperature of the flue gases decreases, physical heat from the gases is also recovered; the heat is received by the sprayed water, whose temperature thereby increases. The sprayed water can theoretically be heated to the temperature of the wet thermometer in the entry to the vertical scrubber. The temperature of the wet thermometer corresponding to the flue gas temperature and the moisture content is 63.3 ^{0}C. The actual limit temperature of water heating is lower, because heat is lost in the system. But from the standpoint of ensuring condensation, the water temperature must not be equal to or higher than the dew point (61.8 ^{0}C). Changes in the heat content of the flue gases are characterised by the enthalpy curve in the figure. The difference in enthalpy at the beginning and end of the scrubber characterises the recovered heat and is directly proportional to the capacity of the condenser.

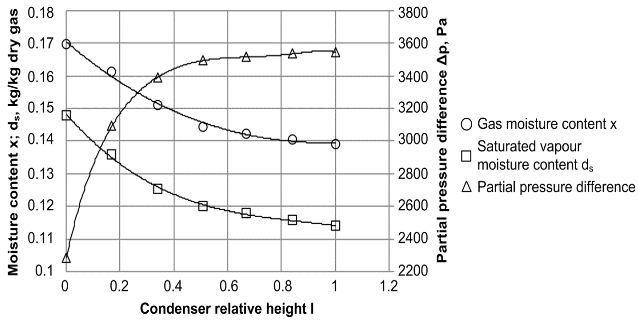

Fig.11 Modelled changes in differences between moisture content and partial pressures in the vertical scrubber

Changes in the intensity of the condensation process are characterised by the changes in the difference between moisture content and partial pressures. In the condensation process, the difference in partial pressures is defined as the difference between the partial pressure of vapour in the flue gases and the partial pressure of water saturated vapour. It is evident that the difference in partial pressures in the lower section of the scrubber (1=0) is smaller and condensation is weaker compared to the upper sections (1=1). As the moisture content of the flue gases decreases, it can happen that a vaporisation zone – instead of condensation – forms in the lower sections of the vertical scrubber. This means that heat will not be recovered in a part of the scrubber and the capacity of the condenser decreases. This can happen if the flue gases are dry after leaving the boiler and are not adequately moistened in the horizontal scrubber.

In order to define the moisture content limit values at which condensation begins in the vertical scrubber, process modelling for a 6 MW boiler with various spray densities was performed. The results can be seen in Figure 12.

It can be seen that as the moisture content of the flue gases decreases, the difference between the partial pressures of saturated vapour in the flue gases and the partial pressure of vapour on the surface of sprayed water droplets decreases. Since the difference in partial pressures is the driving force of the mass transfer process in condensation, its value determines the intensity of the process. When the difference between the partial pressures is zero, condensation does not take place. The results of the modelling indicate that the zero values for the difference in partial pressures can be observed at higher moisture content if the density of spray is lower. When the density of spray is lower, less water is introduced into the condenser and it heats to a higher temperature t_{1c}.

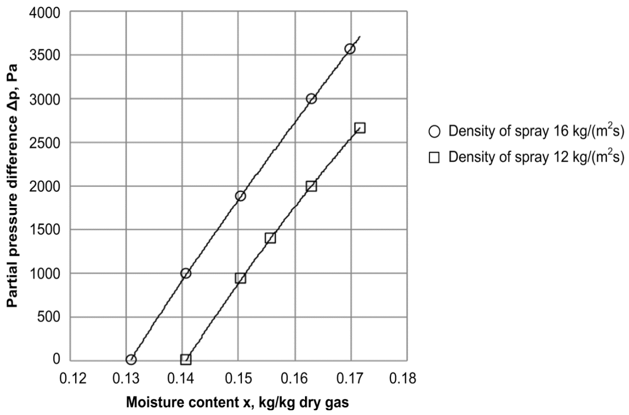

Fig.12 Modelled changes in partial pressures of vapour in the entry to the scrubber (1=0) depending on flue gas moisture content

The modelled values for the final temperature of the sprayed water t_{1c} depending on the moisture content of the flue gases can be seen in Figure 13. The figure also shows the changes in dew point temperature for the flue gases.

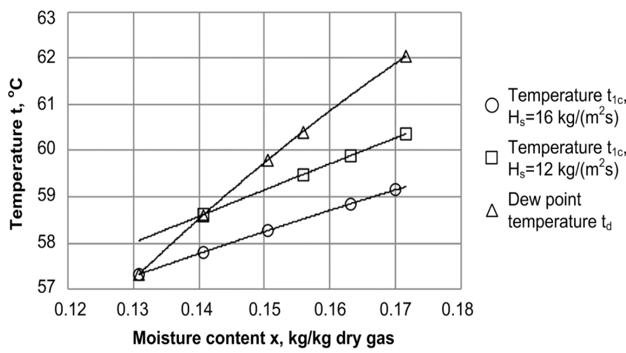

Fig.13 Changes in final temperature of water t_{1c} depending on moisture content of flue gases (1=0).

Modelling was performed for a condenser installed next to a 6 MW wood chip fired boiler, if the temperature of the flue gases in the entry to the condenser's vertical scrubber is 90 ^{0}C, the temperature of the sprayed water t_{2c} is 55 ^{0}C, and the spray densities are 12 kg/(m^2·s) and 16 kg/(m^2·s).

As mentioned earlier, condensation will take place in the vertical scrubber of the condenser if the dew point temperature $t_d > t_{1c}$. This means that the partial pressure of saturated vapour in the flue gases is also higher than the partial pressure of sprayed water vapour on the surface of the droplets $p_s > p_w$ and, as a result of the difference in partial pressures, mass is transferred from the flue gases to the water droplets. If the temperatures are equal $t_d = t_{1c}$, then the observed partial pressures are equal, their difference is zero, and condensation does not take place. If $t_d < t_{1c}$, the partial pressures' difference symbol changes and vaporisation of the sprayed water droplets takes place in the lower section of the vertical scrubber. Such a process is undesirable because it decreases the recovered heat by one part of heat of the mass transfer.

The results of the modelling show that by a spray density of 12 kg/(m^2·s) condensation begins when the flue gases have reached a moisture content of 0.14 kg/kg dry gas. As the spray density increases to 16 kg/(m^2·s), condensation begins when the flue gases have reached a moisture content of 0.13 kg/kg dry gas.

An analysis of the results of the modelling indicate that the moisture content of the flue gases and the spray density are interconnected quantities and a certain flue gas moisture content corresponds to the minimal value for spray density at which moisture condensation takes place in the entire vertical scrubber.

6. Comparison of modelled and experimental data

A study of the parameter changes characterising flue gas condenser processes can be done by experiment or by modelling the heat and mass transfer process in the condenser. The drawback to the experimental method is that it is not able to determine the local parameters of the condenser along the length of the horizontal scrubber or throughout the height of the vertical scrubber. It is linked to the measuring of the parameters of the water and flue gas mixture. The local parameters are determined by a model created with Excel. The two methods supplement each another and allow the processes in the condenser to be better understood [16]. An essential issue when analysing the results of the modelling is how adequately the results of the modelling describe the actual processes. To answer this question, a comparison of the parameters determined by modelling and the regression equation (2) was performed. The changes in condenser capacity depending on boiler capacity determined by both methods were compared. The compared changes can be seen in Figure 14.

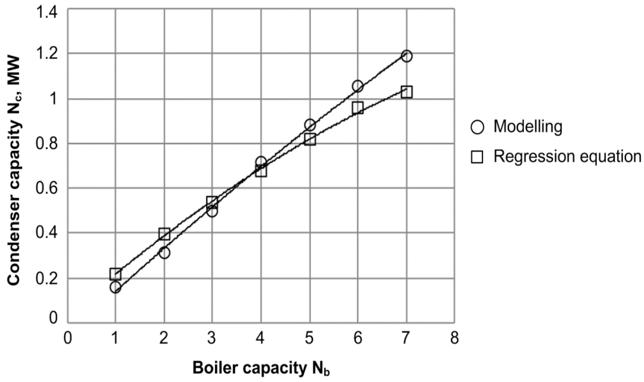

Fig.14 Comparison of condenser capacity values determined by modelling and regression equation

Parameter values characteristic of boiler capacity observed in the experiment were used in the regression equation and modelling calculations. The figure shows a good congruence of results in the capacity range from 3 MW to 5 MW. The values for condenser capacity determined by the regression equation are higher in the case of small boiler capacity. The difference in results between a boiler capacity of 1 MW as applied to the value determined by the regression equation is 15%. The difference for a boiler capacity of 7 MW is 18%. The comparison of results indicates that the congruence is satisfactory and that results of modelling can be used to explain changes in local parameters of a condenser.

7. Conclusions

1. The regression equation has resulted in a multiple empirical equation that quantitatively determines the capacity of a condenser depending on characteristic and statistically significant indices of the operations of a boiler system and serves as a basis for the prognosis and evaluation of the results of a system's operation regimes. The regression equation obtained as the result of the study determines the condenser capacity depending on the flue gas moisture content x, the spray density H_s, the boiler capacity N_b, the flue gas temperature before entering the condenser t_{1g}, the water temperature after leaving the condenser t_{1c}, the return temperature of net water t_{1n}, and the temperature of water sprayed into the condenser t_{2c}.
2. The results of the analysis show that as the return temperature of the net water increases, the heat capacity of the condenser decreases because the heat transfer between the heating net and the condenser's water circuits in the net water heat

exchange weakens. As a result, the temperature of the sprayed water increases and both the dry heat transfer between the droplets and the flue gases and the mass transfer weaken.

3. As the spray density increases, the condenser's capacity also increases. Capacity increases more quickly when spray density is low, whereas capacity decreases and tends to saturation level when values are higher. As the amount of sprayed water increases, its final heating temperature t_{1c} and the temperature of the sprayed water t_{2c} decreases. The flue gases are cooled more deeply when water at a lower temperature is sprayed. More complete cooling of the flue gases ensures a more intense dry heat transfer. When the temperature of the surface of the droplets is lower, the partial pressure of the vapour on the surface is lower, the difference in partial pressures on the droplet surface and in the flue gases increases, and the mass transfer intensifies. Both the dry heat transfer and mass transfer processes are such that the flue gases are cooled more deeply and heat recovery in the condenser increases.
4. As boiler capacity increases, so does the condenser capacity. These changes are mainly influenced by two factors: an increase in the amount of exiting flue gases and an increase in the flue gas temperature. As the amount of flue gases increases, so do the speeds of flue gases in the condenser and the turbulence of the gas-water mix intensifies. As a result, the heat and mass transfer between both environments intensifies and the amount of heat recovered in the condenser increases. Studies show that for small capacity boilers the condenser capacity is 17% of the boiler capacity, and condenser capacity is 14% if the boiler capacity is 6 MW.
5. The results of the study show that as the moisture content of flue gases increases, so does the condenser capacity, because the flue gases contain a greater amount of vapour, which recovers a larger amount of phase transition heat as it condenses. In order to ensure condensation, the water temperature must be such that the partial pressure of vapour on the sprayed water droplet surfaces is lower than the partial pressure of saturated vapour in the gases. This means that the temperature of sprayed water in the entire vertical scrubber must be lower than the dew point temperature.
6. The changes in condensation intensity are characterised by the changes in the difference between moisture content and partial pressures. The difference in partial pressures during condensation is defined as the difference between the partial pressure of vapour in the flue gases and the partial pressure of water saturated vapour. The difference in partial pressures is smaller and the condensation is weaker in the lower section of the scrubber than it is in the upper section. As the flue gas moisture content decreases, it is possible that a vaporisation zone is created in the lower section of the vertical scrubber instead of condensation. This means that heat recovery does not take place in a part of the scrubber and condenser capacity is decreased. This can happen in cases where the flue gases after leaving the boiler are dry and are not adequately moistened in the horizontal scrubber.
7. The task of the horizontal scrubber is primarily to cool the flue gases and to maximally increase the moisture content of the gases, which is achieved by spraying water that has a higher temperature than the dew point temperature of the flue gases. The driving force of the vaporisation process is the difference between the partial pressure of vapour saturated with sprayed water and the partial pressure of vapour in the gases. The higher the temperature of the sprayed water, the more intense the vaporisation process. Water is sprayed from the condenser's volume of water and its maximum temperature cannot exceed the temperature of the wet thermometer in the entry to the vertical scrubber. The study shows that the difference in partial pressures in the horizontal scrubber decreases and the vaporisation process is more intense in the entry to the scrubber. It is important to create a flue gas situation at the exit of the horizontal scrubber that promotes condensation of moisture in the vertical scrubber regardless of the moisture content of the flue gases after leaving the boiler.
8. An analysis of the modelling results indicates that the moisture content of the flue gases and the density of spray are interconnected quantities and that specific flue gas moisture content corresponds to a minimum value of spray density at which condensation of moisture takes place in the entire vertical scrubber. If the value of spray density is lower than the minimum, then the temperature of the flue gas dew point is lower than the temperature of the sprayed water. As a result, the partial pressure of saturated vapour in the flue gases is lower than the partial pressure of sprayed water vapour and vaporisation of the sprayed water droplets takes place in the lower section of the vertical scrubber of the condenser. Such a process is undesirable because it

decreases the recovered heat by one part of heat of the mass transfer.

References

[1] Y. Li, J. F. Klausner, R. Mei, J. Knight, Direct condensation in packed beds, *International Journal of Heat and Mass Transfer*, Vol. 49, 2006, pp. 4751–4761.

[2] I. Veidenbergs, D. Blumberga, E. Vigants, G. Kozuhars, Heat and mass transfer processes in scrubber of heat recovery device, *Scientific Proceedings of Riga Technical University*, Vol. 4, 2010, pp. 109–115.

[3] M. Blasnik, The need for statistical analysis and reporting requirements: some suggestions for regression models, Energy Program Evaluation Conference, Chicago, 1995.

[4] I. Revina, *Ekonometrija* [*Econometrics*], EuroFaculty, 2002.

[5] International Organization of Standardization (ISO), Guide to the Expression of Uncertainty in Measurement, Geneva, 1995.

[6] L. Kirkup, *Calculating and expressing uncertainty in measurement*, University of Technology, Sydney, 2003.

[7] H. Castrup, *Distributions for uncertainty analysis*, Integrated Sciences Group, California, 2004.

[8] D. Che, Y. Liu, C. Gao, Evaluation of retrofitting a conventional natural gas fired boiler into a condensing boiler, *Energy Conversion and Management,* Vol. 45, 2004, pp. 3251–3266.

[9] P. Neuenschwander, J. Good, T. Nussbaumer, Combustion efficiency in biomass furnaces with flue gas condensation, Biomass for Energy and Industry, 10th European Conference and Technology Exhibition, Würzburg, 1998.

[10] K. Jeong, M. J. Kessen, H. Bilirgen, E. K. Levy, Analytical modeling of water condensation in condensing heat exchanger, *International Journal of Heat and Mass Transfer*, Vol. 53, 2010, pp. 2361–2368.

[11] F. Pena, J. M. Blanco, Evaluation of the physical dew point in the economizer of a combined cycle burning natural gas, *Applied Thermal Engineering*, Vol. 27, 2007, pp. 2153–2158.

[12] G. Miliauskas, S. Sinkunas, G. Miliauskas, Evaporation and condensing augmentation of water droplets in flue gas, *International Journal of Heat and Mass Transfer*, Vol. 53, 2010, pp. 1220–1230.

[13] A. Yu Val'dberg, O. V. Zhigun, Designing hollow spray scrubbers for evaporative gas cooling. Part 1. Determining the minimum possible gas temperature, *Chemical and Petroleum Engineering*, Vol. 1, 2008, pp. 11–17.

[14] K. Jeong, M. J. Kessen, H. Bilirgen, E. K. Levy, Analytical modeling of water condensation in condensing heat exchanger, *International Journal of Heat and Mass Transfer*, Vol. 53, 2010, pp. 2361–2368.

[15] K. Boulama, N. Galanis, J. Orfi, Heat and mass transfer between gas and liquid streams in direct contact, *International Journal of Heat and Mass Transfer*, Vol. 47, 2004, pp. 3669–3681.

[16] M. Lemouari, M. Boumaza, A. Kaabi, Experimental analysis of heat and mass transfer phenomena in a direct contact evaporative cooling tower, *Energy Conversion and Management*, Vol. 50, 2009, pp. 1610–1617.