DE GRUYTER OPEN

Chemical and Process Engineering 2014, 35 (4), 447-461
DOI: 10.2478/cpe-2014-0034

PAN POLSKA AKADEMIA NAUK

METHODS OF THERMAL CALCULATIONS FOR A CONDENSING WASTE-HEAT EXCHANGER

Paweł Rączka*, Kazimierz Wójs

Wrocław University of Technology, Faculty of Mechanical and Power Engineering,
Wyb. Wyspiańskiego 27, 50-370 Wrocław, Poland

The paper presents the algorithms for a flue gas/water waste-heat exchanger with and without condensation of water vapour contained in flue gas with experimental validation of theoretical results. The algorithms were used for calculations of the area of a heat exchanger using waste heat from a pulverised brown coal fired steam boiler operating in a power unit with a capacity of 900 MW_e. In calculation of the condensing part, the calculation results obtained with two algorithms were compared (Colburn-Hobler and VDI algorithms). The VDI algorithm allowed to take into account the condensation of water vapour for flue gas temperatures above the temperature of the water dew point. Thanks to this, it was possible to calculate more accurately the required heat transfer area, which resulted in its reduction by 19 %. In addition, the influence of the mass transfer on the heat transfer area was taken into account, which contributed to a further reduction in the calculated size of the heat exchanger - in total by 28% as compared with the Colburn-Hobler algorithm. The presented VDI algorithm was used to design a 312 kW pilot-scale condensing heat exchanger installed in PGE Belchatow power plant. Obtained experimental results are in a good agreement with calculated values.

Keywords: condensation heat recovery, calculation of the heat and mass transfer, heat exchanger, waste heat

1. INTRODUCTION

The solutions used so far for heat recovery from flue gas generated by power boilers come down to installing in the flue gas duct of an exchanger that collects sensible heat from the flue gas. Such a solution was used for example in the *Lippendorf* Power Plant near Leipzig. The heat recovered in the exchanger is transferred to the condensate flowing from the turbine condenser to the boiler, which contributes to an increase in the electric power and efficiency of the unit. The cooled flue gas is then directed to a cooling tower. A similar solution was used in a block with a capacity of 460 MW_e in the Łagisza Power Plant. The heat from the flue gas leaving the electrostatic precipitator is used for pre-heating the combustion air (downstream the rotary air heater in the boiler).

However, condensing waste-heat exchangers have not been used in power units so far. The first such a heat exchanger in Poland is just being built in the Białystok S.A. Combined Heat and Power Plant. This heat exchanger is to recover heat from the flue gas generated by a biomass boiler with a fluidised bed furnace. The heat will be transferred to utility water and directed to the district heating system of Białystok. This solution will allow to increase thermal power in the combined heat and power plant, improving its efficiency and competitiveness, as well as increasing the production of "green" heat.

*Corresponding author, e-mail: pawel.raczka@pwr.edu.pl

For the above reasons, calculation algorithms for condensing heat exchangers, especially in relation to flue gas from coal combustion, have not been developed so far (Szulc et al., 2013). Only two examples of calculations concerning a condensing heat exchanger used for flue gas from coal combustion were found in the literature (Jeong et al., 2010; Levy et al., 2008). Both models referred to the well-known Colburn-Hougen model (Colburn and Hougen, 1934) and introduced various modifications to it. In other works, mathematical models of condensing heat exchangers used at combustion of fuel oil (Ball et al., 1984; Osakabe, 2000) or gas (Jia et al., 2001; Liang et al., 2007; Shi et al., 2011) were presented.

2. MATHEMATICAL MODELS

Heat transfer in the condensing heat exchanger, which occurs between the flue gas stream and the pipe wall covered with a film of water vapour condensate, takes place through the interface layer near the wall. In this layer, further away from the inlet towards the exchanger and closer from the gas core towards the wall in each cross section, the fraction of inert gas increases, while the fraction of water vapour decreases. So the mechanism responsible for heat transmission is simultaneous and mutually dependent on heat and mass transfer. The share of each mechanism is variable on the path of the flue gas flow - heat transfer may predominate at the inlet of the heat exchanger, while mass transfer may be predominant at the outlet (due to the condensation of water vapour and a low fraction of water vapour in the mixture with inert gas at the outlet).

Calculations of heat transfer in the condensing part are based on the model composed of: α_G - the coefficient of convective heat transfer from gas to the condensate film/pipe surface, α_F - the coefficient of convective heat transfer from the condensate layer to the pipe wall, $\frac{s}{\lambda}$ - the thermal resistance of the wall, and α_K - the coefficient of convective heat transfer from the wall to the cooling water. The thermal resistance of the condensate is not taken into account.

On this basis, the total heat transfer coefficient can be expressed as

$$k = \left(\frac{1}{\alpha_G} + \frac{1}{\alpha_F} + \frac{s}{\lambda} + \frac{1}{\alpha_K} \right)^{-1} \tag{1}$$

Among the above listed coefficients, the biggest problem in calculations is the thermal resistance of the boundary layer of the inert gas containing water vapour $\frac{1}{\alpha_G}$. This is caused by the necessity to calculate the partial pressure of water vapour in this layer, near the pipe wall or the condensate, where inert gas with a volume fraction different than that in the centre of the stream occurs. If calculations concern flue gas, the problem is complicated further by the presence of superheated water vapour in flue gas.

In the Colburn's method (Colburn and Hougen, 1934), the following values must be known for the dry part of the exchanger:

- inlet temperature of flue gas T_{G1}, mass flow rate and inlet temperature of coolant T_{K1}; hence the coolant and flue gas properties are known,
- tube wall thickness s and material properties.

The flue gas T_{G2} and coolant T_{K2} temperatures can be calculated from the energy balance equations

$$\dot{Q} = \frac{\overline{T}_G - \overline{T}_K}{\dfrac{1}{\alpha_G A_o} + \dfrac{\ln\left(\dfrac{r_o}{r_i}\right)}{2\pi K_G L} + \dfrac{1}{\alpha_K A_i}} \tag{2}$$

$$\dot{Q} = \dot{m}_K C_{p,K} \left(T_{K2} - T_{K1} \right) = \dot{m}_G C_{p,G} \left(T_{G1} - T_{G2} \right) \tag{3}$$

The heat transfer coefficients on the flue gas side α_G and cooling water side α_K are calculated using correlations of forced convective heat transfer for tube banks and inside a tube, respectively (VDI-GVC Editor, 2010) as for the economiser in a pulverised fuel steam boiler.

When a part of a heat exchanger is in an unsaturated or saturated region the following quantities are known:

- inlet parameters of flue gas - the temperature T_{G1} and partial pressure of water vapour P_{V1}, and hence the thermodynamic properties of the flue gas,
- mass flow rate and inlet temperature T_{K1} of cooling water as well as its other thermodynamic parameters.

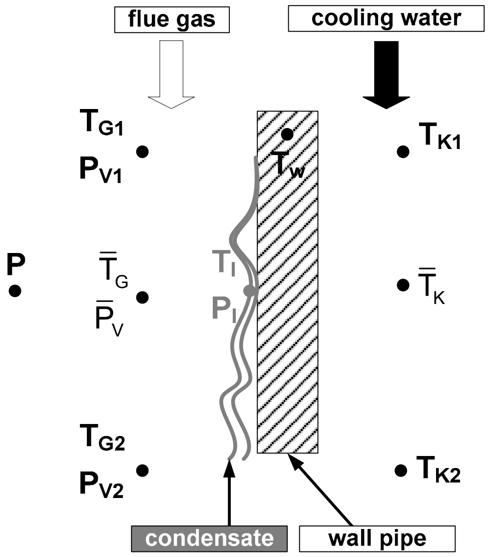

Fig. 1. Section used for calculations in the Colburn's method

To satisfy an energy balance in the unsaturated region, the flue gas outlet temperature and vapour pressure, interface temperature, coolant outlet temperature, and heat exchanger surface area must be calculated (in relation to the section used for calculations - Fig. 1)

Therefore, there are five unknowns and appropriate equations. The first one is the energy balance equation in saturated region (Colburn and Hougen, 1934)

$$\alpha_G\left(\bar{T}_G - T_I\right) + K_G M_V L\left(\bar{P}_V - P_I\right) = k\left(T_I - \bar{T}_K\right) \tag{4}$$

where the first and second terms on the left side are the sensible and latent heat transferred from flue gas respectively. The term on the right side is the heat transferred through the condensate film and heat exchanger wall. There is only one unknown in that equation - P_I because of saturated conditions (P_I is a function of T_I). The value of T_I is iteratively adjusted until convergence is achieved.

The second equation with one unknown T_{K2} is the balance of energy on the side of coolant and flue gas

$$\dot{m}_G\left(H_1 - H_2\right) - \dot{m}_C C_{p,C} T_I = \dot{m}_K C_{p,K}\left(T_{K2} - T_{K1}\right) = \dot{Q} \tag{5}$$

where the first term on the left side is the flux of heat transferred on the side of flue gas. The second term is the heat lost with the condensate being removed, while the third term is the flux of heat taken by the cooling water.

Since the flue gas outlet temperature and vapour pressure for unsaturated condensation are independent variables, an equation is needed to relate the heat and mass transfer of the cooling flue gas. The following equation provides a relationship between the outlet vapour pressure P_{V2}, the outlet flue gas T_{G2} and interface temperatures T_I (Colburn and Hougen, 1934)

$$\frac{P_{V1}-P_{V2}}{T_{G1}-T_{G2}}=\frac{\left(P-\bar{P}_V\right)\left(\bar{P}_V-P_I\right)\left(e^a-1\right)}{P_{m1}\left(\bar{T}_G-T_I\right)a}\left[\frac{\Pr}{Sc}\right]^{\frac{2}{3}} \tag{6}$$

where $a=\dfrac{M_V C_{p,V}}{M_m C_{p,m}}\left[\dfrac{\Pr}{Sc}\right]^{\frac{2}{3}}\ln\left(\dfrac{P-P_I}{P-\bar{P}_V}\right)$.

The fourth equation is the relationship between the difference between the bulk cooling water and condensate interface temperature on the pipe surface and the total heat flux

$$A=\frac{\dot{Q}\left(\dfrac{1}{\alpha_K}+\dfrac{\ln\left(\dfrac{r_o}{r_i}\right)}{2\pi K_G L}+R_C\right)}{T_I-\bar{T}_K} \tag{7}$$

The above Equations (4-7) are solved iteratively using the assumed value of the flue gas outlet temperature T_{G2}. In the heat exchanger region, where the condensation of saturated water vapour occurs, Equation (6) is not needed since the outlet vapour pressure is fixed by T_{G2}.

The results obtained at the end of the section provide input data for the next section. The outlet parameters for the last section will also constitute the outlet parameters for the heat exchanger, while the summarised heat transfer areas in individual sections will give the total transfer area.

Hobler (1986) proposed a method for calculating the partial pressure in the interface layer, which is more precise than that in the Colburn's model. He compared the formulas for α_G obtained in the same Colburn’s experiment, but relating in the first case to the core of the inert gas and water vapour mixture, and in the second case - to the interface layer. He obtained the dependence for the ratio of partial pressures of water vapour and inert gas (air):

$$\frac{P_{V,F}}{P_{G,F}}=\frac{48}{82.5}\frac{P_{V,B}}{P_{G,B}}=0.58\frac{P_{V,B}}{P_{G,B}} \tag{8}$$

Thus, the average mole fraction of water vapour in the interface layer is

$$X_{V,F}=0.58X_{V,B} \tag{9}$$

In the case of turbulent flow of the gas mixture, he determined from it an equivalent convective heat transfer coefficient for the condensation of water vapour from the saturated gas mixture

$$\alpha^*=\alpha_G\left(1+C\zeta X_{V,B}\right) \tag{10}$$

where C= 0.684.

The above equation allowed to take into account the simultaneous heat and mass transfer occurring at the condensation of water vapour from the air. Although the equation was formulated for saturated mixtures, it may also be used for unsaturated mixtures - without a large error (Hobler 1986).

In the VDI model (VDI-GVC Editor, 2010) the specific flux of heat transferred from flue gas to the condensate film forming on the wall is (Fig. 2)

$$q=\alpha_G E_T\left(T_G-T_F\right) \tag{11}$$

where the Ackermann correction factor is

$$E_T = \frac{\phi_T}{1 - e^{-\phi_T}} \tag{12}$$

with $\phi_T = \dfrac{\tilde{C}_{p,V}}{\tilde{C}_{p,G} Le^{0.6}} \ln\left(\dfrac{\tilde{y}_{G,F}}{\tilde{y}_{G,B}}\right)$.

The correction factor allows to include in the calculations the mass transfer that occurs simultaneously with the heat transfer.

The heat flux transferred to the coolant is expressed by the equation

$$q = k'\left(T_F - T_K\right) \tag{13}$$

where the coefficient of heat transfer from the condensate film T_F to the coolant with the temperature T_K reads

$$\frac{1}{k'} = \frac{1}{\alpha_F} + \frac{s}{\lambda} + \frac{1}{\alpha_K} \tag{14}$$

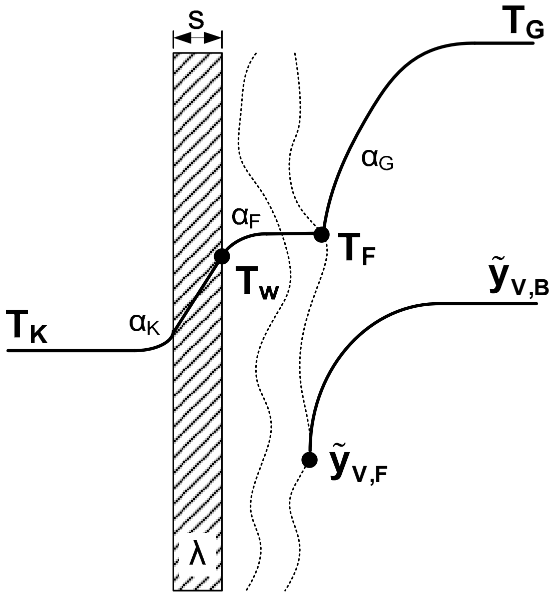

Fig. 2. The temperature and concentration profiles in the condensate film and near it (VDI model)

In order to determine the amount of condensing water vapour and thermal power of the heat exchanger, it is necessary to determine the temperature of the condensate film on the pipe surface. The following balance equation is used for this purpose (VDI-GVC Editor, 2010):

$$k'\left(T_F - T_K\right) = \alpha_G \phi_T \left(\frac{\Delta\tilde{h}_V}{\tilde{C}_{p,V}} + \frac{T_G - T_F}{1 - e^{-\phi_T}}\right) \tag{15}$$

In order to determine the heat transfer area, it is necessary to know the predominant mechanism in the water vapour condensation process. For this purpose, the temperature of the condensate film at the inlet and outlet of the heat exchanger condensation regions needs to be determined (Eq. 15). Then, using the heat balance equation

$$q = k\left(T_{dew} - T_K\right) = k'\left(T_F - T_K\right) = \alpha_G\left(T_{dew(\tilde{y}_{V,B})} - T_F\right) \tag{16}$$

where $T_K < T_w < T_F < T_{dew}$, the values of $\left(\frac{k'}{\alpha_G}\right)_1$ and $\left(\frac{k'}{\alpha_G}\right)_2$ are calculated. If the calculated value of $\left(\frac{k'}{\alpha_G}\right)_2$ is lower than 0.5, the whole condensation process may be regarded as controlled by the heat transfer mechanism (VDI-GVC Editor, 2010). The heat transfer area can be then calculated from the equation

$$A = \frac{\dot{Q}}{k' \Delta T_{log}} \tag{17}$$

where ΔT_{log} is the logarithmic difference of the inlet and outlet temperature of the condensate film and cooling water.

In turn, for $\left(\frac{k'}{\alpha_G}\right)_1 > 2$, the entire condensation process is controlled by the mass transfer mechanism, while the heat transfer area can be calculated from the equation (VDI-GVC Editor, 2010)

$$A = \frac{\dot{N}_G}{n_G \beta_G \bar{\bar{y}}_{G,F}} \left(\frac{\tilde{y}_{G,F,2}}{\tilde{y}_{G,B,1}} - \frac{\tilde{y}_{G,F,2}}{\tilde{y}_{G,B,2}} + \ln \frac{\dfrac{\tilde{y}_{G,F,2}}{\tilde{y}_{G,B,1}} - 1}{\dfrac{\tilde{y}_{G,F,2}}{\tilde{y}_{G,B,2}} - 1} \right) \tag{18}$$

where $\bar{\bar{y}}_{G,F} = 0.5\left(\tilde{y}_{G,F,1} + \tilde{y}_{G,F,2}\right)$.

If the predominant mechanism at the inlet to the condensing region is heat transfer, while the predominant mechanism at the outlet is mass transfer, the area of the heat exchanger should be calculated from Equations (17) and (18) and averaged.

3. CALCULATION DATA AND ASSUMPTIONS

Calculations for the waste-heat exchanger were carried out for flue gas from combustion of pulverised brown coal. The adopted parameters of coal are given in Table 1, while Table 2 shows the remaining parameters adopted for the process.

The value of the cooling water temperature at the inlet to the heat exchanger was adopted as equal to the temperature of the condensate leaving the turbine condenser (25 °C), assuming that it will be cooled in an open system. The calculations of the water temperature at the outlet from the heat exchanger were performed with the view of utilising the heat contained in it (attempts were made to maximise both its temperature and the amount of condensing water vapour - both processes are opposing).

Table 1. Technical and elemental analysis of as-received brown coal

Quantity	Unit	Value
Calorific value	MJ/kg	7.75
Moisture content	mass fraction	0.5140
Ash content		0.1140
C content		0.2320
H content		0.0192
O content		0.1050
N content		0.0032
S content		0.0126

Table 2. Input data for calculations

Quantity	Unit	Value
Mass flow of the fuel burned	kg/s	248.35
Temperature of flue gas at the inlet of the heat exchanger	°C	170
Coefficient of excess air in flue gas	-	1.2
Flue gas volumetric flow rate	m^3_{STD}/s	889
Cooling water mass flow rate	kg/s	850
Cooling water temperature at the inlet of the heat exchanger	°C	25
Temperature of flue gas at the outlet of the heat exchanger	°C	60
Initial volume fraction of water vapour in flue gas	-	0.250
Calculated water dew point	°C	65.0

The thermal calculations of the condensing heat exchanger (Fig. 3) use two models:

- the Colburn-Hagen model modified by Hobler (1986),
- the VDI model (VDI-GVC Editor, 2010).

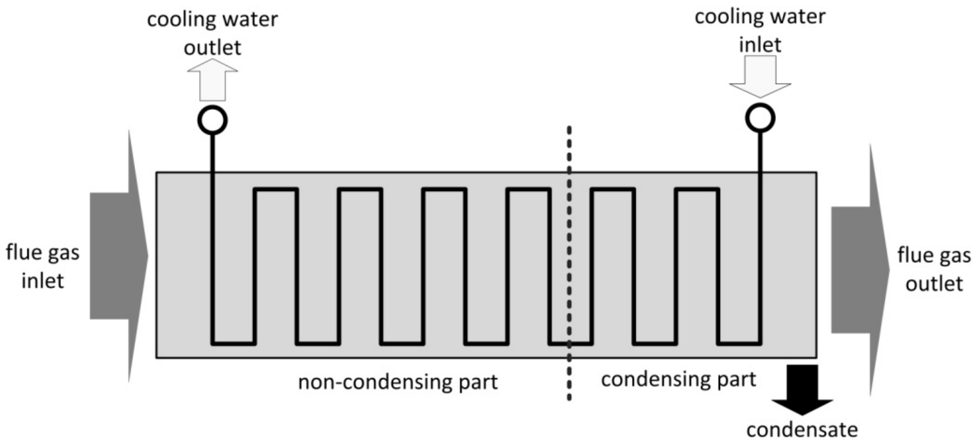

Fig. 3. Diagram of a cross-flow heat exchanger

In order to perform calculations, approximating functions were created for dynamic viscosity, Prandtl number, as well as for thermal conductivity for water and water vapour. Special functions were used for calculating e.g. dynamic viscosity and thermal conductivity for mixtures of inert gases and water vapour (VDI-GVC Editor, 2010).

The following geometry of the flue gas duct was adopted: height – 8 m, width – 16 m, while the length of the heat exchanger is the resulting value. The cross-flow heat exchanger was made of fluoroplastic coating steel coil pipes with the outer diameter of 13.5 mm and the wall thickness of 1.8 mm. The transverse and longitudinal pitch of the pipes was 40.5 mm. The flue gas velocity was approx. 15 m/s and the average flue gas pressure: 0.1 MPa. It was assumed that the system would include a single heat exchanger, through which the entire flue gas stream would flow. The velocity of the cooling water flowing in the pipes was 1 m/s, resulting from the mass flow rate of cooling water (850 kg/s) and the adopted geometry of the heat exchanger. The flue gas was cooled from 170 to 60 °C. All the calculations were carried out for the values of arithmetic means of thermodynamic parameters of flue gas and cooling water between the inlet and the outlet, separately for the non-condensing and condensing parts of the heat exchanger.

In the calculations, the heat exchanger was hypothetically divided into two parts: the first part without condensation and the second one with the condensing water vapour from flue gas (Fig. 3).

In the Colburn-Hobler model, the flue gas is cooled in the first part to the water dew point (65 °C) and transfers only sensible heat to cooling water. Then the flue gas flows to the second part of the heat exchanger, where it is cooled to a temperature below the dew point (60 °C) and transfers the sensible heat and the heat of water vapour condensation. The generated condensate is carried away from the heat exchanger.

Although a similar approach was used in the calculations based on the VDI model, the distribution of the temperature between flue gas and cooling water at the inlet and outlet of the condensing part was additionally calculated. The flue gas temperature, at which condensation begins due to the temperatures of the coolant, the pipe wall and condensate, was determined iteratively. From Eq. 15 T_F for inlet and outlet conditions was calculated. If calculated T_F (and T_w) was lower than the water dew point then condensation occurred (Tab. 4). For the calculated flue gas temperature of 78 °C (i.e. higher than the water dew point 65°C), condensate temperature T_F was 64.9°C (and T_w is 62.7 °C). Therefore 78 °C was the temperature of division of the heat exchanger between the non-condensing part and the condensing part in the VDI model.

In the non-condensing part of the heat exchanger, the calculation formulas for economiser in pulverised coal-fired steam boilers were used. In order to calculate the heat and mass transfer in the condensing part, the influence of the thermal and diffusion resistance of the inert gas layer (dry flue gas) between the core of the wet flue gas stream (far from the wall) and the condensate on the pipe wall was taken into account. For both models, the influence of thermal resistance of the condensate film on the wall was treated as negligibly small (Hobler, 1986), while the resistance of heat conduction through the pipe wall (which reaches a significant value if the coil pipe of the heat exchanger is made of plastic) was taken into account. It was assumed that the value of the thermal conductivity coefficient of the pipe material was 14 W/(m·K) (Lee et al., 2005).

In order to carry out calculations for the entire heat exchanger a system of equations describing heat and mass transfer was derived. It was possible to determine the most important design parameters (heat transfer area, number of pipes, pipe rows, etc.). It was assumed in both models that $25.3\%_{vol.}$ of the water vapour contained in flue gas would undergo condensation, which results from the calculations performed in the range of the condensation of water vapour from flue gas cooled down to 60 °C.

4. RESULTS AND DISCUSSION

4.1. Colburn-Hobler model

By dividing the heat exchanger into a part, in which flue gas is cooled without condensation, and the condensing part (depending on the water dew point), the maximum dimensions of the heat exchanger can be determined. In the case of brown coal, the total heat transfer area of 35,665 m^2 was obtained (Tab. 3).

A considerable difference in the calculated heat transfer area of both parts of the heat exchanger is visible (82% - non-condensing part, 18% - condensing part), while the share in the total thermal power transferred to the coolant is 53.5% and 46.5 %, respectively (Fig. 4).

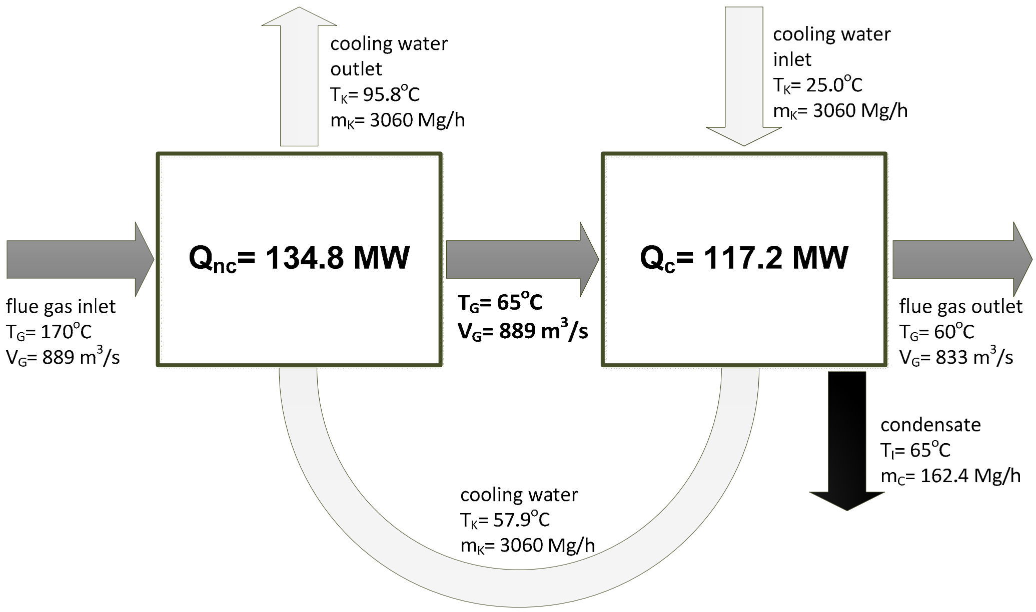

Fig. 4. Results of heat transfer calculations for the cross-flow heat exchanger for recovery of waste heat from brown coal flue gas; plastic coil pipes – Colburn-Hobler model

Table 3. The results of the calculations performed with use of the Colburn- Hobler model

Quantity	Unit	Value
Total calculated heat transfer area	m^2	35,665
non-condensing part	m^2	29,247
condensing part	m^2	6,419
Heat transfer coefficient in the non-condensing part	$W/(m^2·K)$	165
Convective heat transfer coefficient in the condensing part		
from flue gas to the wall (without condensation)	$W/(m^2·K)$	157
from flue gas to the condensate on the wall (with condensation), α*	$W/(m^2·K)$	1,549
from the wall to cooling water	$W/(m^2·K)$	5,765
Heat transfer coefficient in the condensing part	$W/(m^2·K)$	1,123

4.2. VDI model

Since the distribution of temperatures between the gas and the coolant was calculated in the VDI model, the temperature at the outlet from the non-condensing part of the heat exchanger was selected so that condensation occurs on the coil wall surface, despite the fact that the gas temperature in the bulk of the stream is above the water dew point. This should allow to calculate the heat and mass transfer in a more precise and realistic way, as well as to reduce the calculated heat transfer area.

The obtained results are presented in Fig. 5 and Table 4. Nearly at the same total thermal power as in the Colburn-Hobler model, a smaller total area of the heat exchanger (25,655 m^2) was obtained.

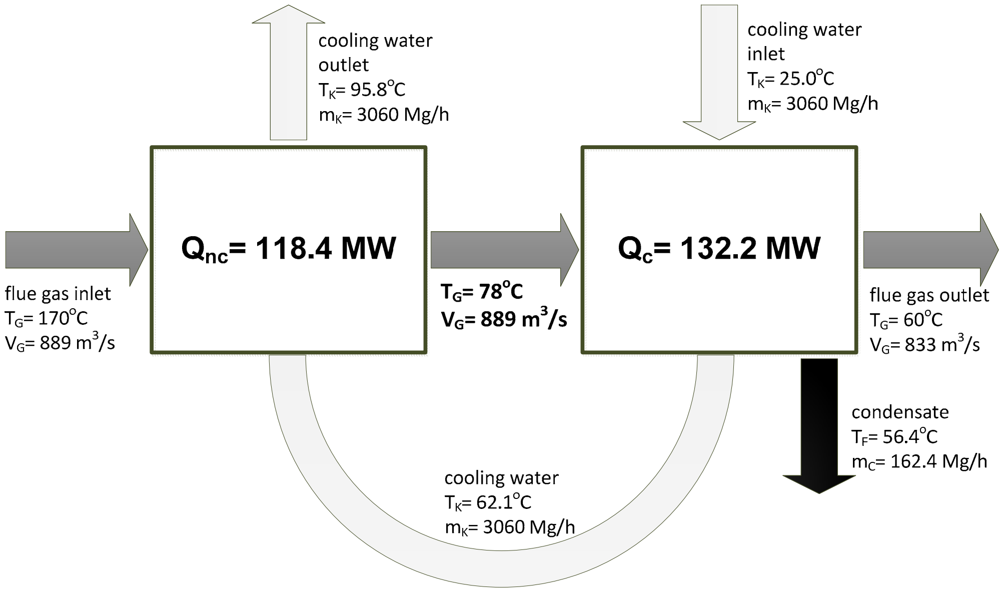

Fig. 5. Results of heat transfer calculations for the cross-flow heat exchanger for recovery of waste heat from brown coal flue gas; plastic coil pipes – VDI model

Table 4. Results of calculations performed with the VDI model

Quantity	Unit	Value
Total calculated heat transfer area without / with mass transfer	m^2	28,787 / **25,655**
non-condensing part	m^2	19,406
condensing part	m^2	9,381
condensing part, including the mass transfer	m^2	6,249
Temperature distribution in the condensing part (inlet to the condensing part/outlet from the condensing part)		
flue gas, T_G	°C	78.0 / 60.0
condensate on the pipe wall, T_F	°C	64.9 / 47.8
pipe wall, T_w	°C	62.7 / 26.3
cooling water, T_K	°C	62.1 / 25.0
Heat transfer coefficient in the non-condensing part	$W/(m^2 \cdot K)$	167
Convective heat transfer coefficient in the condensing part		
from flue gas to the condensate film on the wall, α_G	$W/(m^2 \cdot K)$	152
from the condensate film to the pipe wall, α_F	$W/(m^2 \cdot K)$	2,295
from the pipe wall to cooling water, α_K	$W/(m^2 \cdot K)$	5,679
Heat transfer coefficient, k'	$W/(m^2 \cdot K)$	1,470
Mass transfer coefficient, β_G	-	0.182
$\left(\frac{k'}{\alpha_G}\right)_1$	-	0.006
$\left(\frac{k'}{\alpha_G}\right)_2$	-	0.537

As in the Colburn-Hobler model, a considerable difference in the calculated heat transfer area of both parts of the heat exchanger is visible (75.6% - non-condensing part, 24.4% - condensing part), while the shares in the total thermal power transferred to the coolant is 47.2% and 52.8%, respectively.

Therefore, this proves again that the efficiency of the heat exchange in the condensing part is much higher, despite the fact that the temperature difference is much lower.

In addition, the values of the last two rows of the table above indicate that for the VDI model (with a division of the flue gas temperatures in both parts of the heat exchanger different than in the Colburn-Hobler model), both heat and mass transfer constitute a mechanism responsible for heat exchange. Hence, as described by Equations (17) and (18), a smaller heat transfer area in the condensing part can be adopted (6,249 m^2, i.e. smaller by 33%), and thereby the total heat transfer area of the heat exchanger will decrease by 11%.

The presented calculations based on VDI model was used to design a 46 kW laboratory scale condensing heat exchanger test bench - Fig. 6 (Rączka and Wójs, 2014). The obtained results were then used to design a 312 kW pilot-scale condensing heat exchanger installed in PGE Bełchatów power plant equipped with brown coal PF boilers (Fig. 7). Because of a high corrosive gases content in flue gas from brown coal the pilot-scale exchanger was made of fluoroplastic (PFA) coil pipes. The next section presents the obtained experimental results.

Fig. 6. Laboratory scale heat condensing exchanger with equipment (from right – flue gas generator, heat exchanger, cooling water tank)

Fig. 7. Pilot scale 312 kW condensing heat exchanger in PGE Bełchatów power plant

5. EXPERIMENTAL RESULTS

The obtained experimental results for the condensing heat exchanger made of PFA coil pipes utilising waste heat from brown coal PF boiler flue gas in PGE Bełchatów power plant are presented in Fig. 8. The t.G2 to t.G7 are the values of flue gas temperature along the 312 kW heat exchanger. The results (especially flue gas and cooling water temperature on the outlet of the heat exchanger and thermal power transferred to the cooling water) are in good agreement with calculated values – Tab. 5.

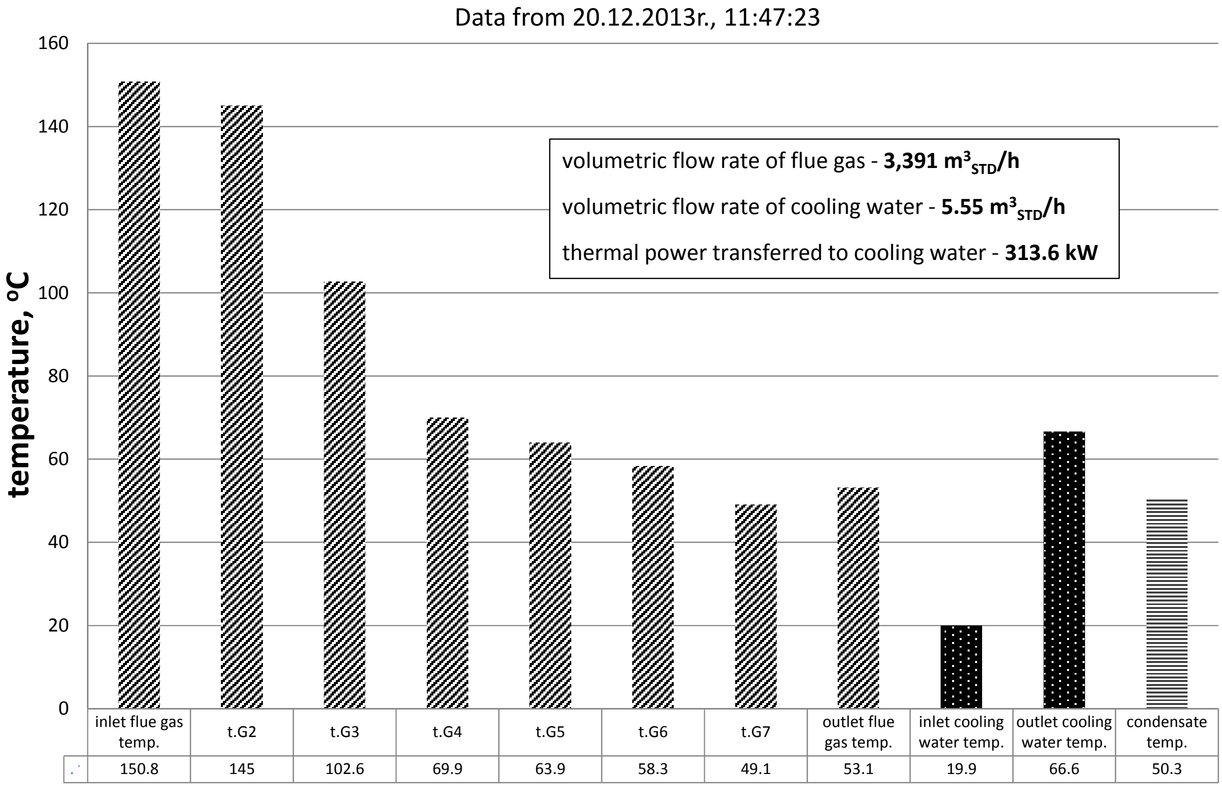

Fig. 8. Experimental results for the condensing heat exchanger utilising waste heat from brown coal PF boiler flue gas (design point of a heat exchanger)

Table 5. Calculated and experimental values comparison for the 312 kW pilot-scale condensing heat exchanger

Quantity	Unit	Value	
		design	experimental
Volumetric flow rate of flue gas	m_{STD}^3/h	3,367	3,391
Temperature of flue gas at the inlet of the heat exchanger	°C	**160.0**	**150.8**
Temperature of flue gas at the outlet of the heat exchanger	°C	55.0	53.1
Cooling water temperature at the inlet of the heat exchanger	°C	20.0	19.9
Cooling water temperature at the outlet of the heat exchanger	°C	73.3	66.6
Cooling water mass flow rate	m_{STD}^3/h	**5.04**	**5.55**
Temperature of condensate at the outlet of the heat exchanger	ºC	58.4	50.3
Total heat transferred to cooling water	kW	**312.1**	**313.6**

A small difference in the obtained results (mainly the outlet flue gas and the cooling water temperature) comes from differences in design values and experimental data of cooling water flow, lower inlet flue gas temperature and flue gas composition (mainly excess air). This is a confirmation that the VDI algorithm is applicable for designing flue gas condensing waste-heat exchangers.

5. CONCLUSIONS

The paper presents the construction and applications of the calculation algorithms for a flue gas/water waste-heat exchanger with and without condensation of water vapour contained in flue gas. The calculations were carried out for flue gas from a brown coal-fired boiler of a power unit with a capacity of 900 MW_e. The heat exchanger was divided into the non-condensing and the condensing parts. The calculations for the non-condensing part were carried out in the same way as for the economiser in a pulverised brown coal fired steam boiler. The calculations for the condensing part were carried out using two models: Colburn-Hobler and VDI. In the Colburn-Hobler model, the value of the water dew point for flue gas (65 °C) was adopted as the point of division into two parts. In turn, by using additional equations in the VDI model, the distribution of temperatures between flue gas and cooling water was determined, and on this basis also the temperature of flue gas and water, at which condensation starts (78 °C). This temperature was adopted as the point of division into the non-condensing and the condensing parts. Thanks to that, the total area of the heat exchanger was reduced - from 35,665 m^2 (Colburn-Hobler model) to 28,787 m^2 (VDI model).

In addition, the area of the condensing part of the heat exchanger, which results only from the mass transfer, was calculated in the VDI model. This allowed to increase the accuracy of the calculations, and thus to reduce the area of this part by 33% and the area of the entire heat exchanger by almost 11% (to 25,655 m^2).

The presented VDI algorithm was used to design a 312 kW pilot-scale condensing heat exchanger installed in PGE Bełchatów power plant. The obtained experimental results are in good agreement with the calculated values.

The results presented in this paper were obtained in research co-financed by the National Centre for Research and Development under the contract SP/E/1/67484/10 - Strategic Research Programme - Advanced technologies for energy generation: Development of a technology for highly efficient zero-emission coal-fired power units integrated with CO_2 capture from flue gas.

SYMBOLS

A	area of the heat exchange, m^2
C_p	specific heat at constant pressure, kJ/(kg K)
$\tilde{C}_p$	molar specific heat at constant pressure, kJ/(kmol K)
E_T	Ackermann correction factor
H	specific enthalpy of gas, kJ/kg
$\Delta\tilde{h}$	molar heat of condensation, kJ/kmol
K_g	molar mass transfer coefficient of water vapour, kmol/(m^2s)(atm)
k, k'	overall heat transfer coefficients, W/(m^2 K)
L	heat of condensation, kJ/kg
M	molar mass, kg/kmol
$\dot{m}$	mass flow rate, kg/s, Mg/h
$\dot{N}$	molar flow, mol/s
n	molar density, mol/m^3
P	total pressure, the partial pressure, atm, kPa
Pr	Prandtl number of inert gas
$\dot{Q}$	thermal power, kW
q	heat flux, kW/m^2

R	thermal resistance, $(m^2 K)/W$
r	radius of the pipe, m
s	thickness of the pipe, m
Sc	Schmidt number of inert gas
T	temperature, K
ΔT_{log}	logarithmic difference of the inlet and outlet temperature of the condensate film and cooling water, K
$\dot{V}$	volume flow, m^3/s
X	mole fraction of the component in flue gas
$\tilde{y}$	molar fraction of the component in flue gas
$\overline{x}$	average

Greek symbols

α	convective heat transfer coefficient, $W/(m^2 K)$
β	coefficient of mass transfer
ϕ_T	dimensionless mass flow rate
λ	thermal conductivity coefficient for the pipe material, W/(m K)
ζ	dimensionless value

Subscripts

B	core of the stream
c	condensing part
C	condensate
dew	water dew point
F	condensate film
G	inert gas, flue gas
I	interface
K	cooling water
m	flue gas mixture
nc	non-condensing part
STD	standard
V	water vapour
w	wall
1	inlet
2	outlet

REFERENCES

Ball D.A., White E.L., Lux J.J., Razgaitis R., Markle R.A., 1984. Condensing heat exchanger systems for residential/commercial furnaces and boilers. Phase III, *DOE Contract Number AC02-76CH00016*.

Colburn A.P., Hougen O.A., 1934. Design of cooler condensers for mixtures of vapors with noncondensing gases. *Ind. Eng. Chem.*, 11, 1178–1182. DOI: 10.1021/ie50299a011.

Hobler T., 1986. *Ruch ciepła i wymienniki*. 6th edition, WNT, Warszawa.

Jeong K., Kessen M.J., Bilirgen H., Levy E.K., 2010. Analytical modeling of water condensation in condensing heat exchanger. *Int. J. Heat Mass Transfer*, 53, 2361–2368. DOI: 10.1016/j.ijheatmasstransfer.2010.02.004.

Jia L., Peng X.F., Yan Y., Sun J.D., Li X.P., 2001. Effects of water vapor condensation on the convection heat transfer of wet flue gas in vertical tube. *Int. J. Heat Mass Transfer*, 44, 4257–4265. DOI: 10.1007/s00231-006-0148-0.

Lee J., Kim T-J., Kim M.H., 2005. Experimental study on the heat and mass transfer of teflon-coated tubes for the latent heat recovery. *Heat Transfer Eng.*, 26:2, 28-37, DOI: 10.1080/01457630590897079.

Levy E., Bilirgen H., Jeong K., Kessen M., Samuelson Ch., Whitcombe Ch., 2008. Recovery of water from boiler flue gas. *DOE Award Number DE-FC26-06NT42727*.

Liang Y., Che D., Kang Y., 2007. Effect of vapor condensation on forced convection heat transfer of moistened gas. *Heat Mass Transfer*, 43, 677–686. DOI: 10.1007/s00231-006-0148-0.

Osakabe M., 2000. Latent heat recovery from oxygen-combustion flue gas. *Energy Conversion Engineering Conference and Exhibit*. Las Vegas, USA, 24-28 July 2000, 804–812. DOI: 10.1109/IECEC.2000.870877.

Rączka P. Wójs K., 2014. Projektowanie kondensacyjnego wymiennika ciepła odpadowego. *Rynek Energii*, 111, 87-92.

Shi X., Che D., Agnew B., Gao J., 2011. An investigation of the performance of compact heat exchanger for latent heat recovery from exhaust flue gases. *Int. J. Heat Mass Transfer*, 54, 606-615. DOI: 10.1016/j.ijheatmasstransfer.2010.09.009.

Szulc P., Tietze T., Rączka P., Wójs K., 2013. Porównanie wybranych konstrukcji wymienników ciepła pracujących w układzie odzysku ciepła odpadowego ze spalin wylotowych (in Polish). *Archiwum Energetyki*, 1-4, 11-31.

VDI-GVC Editor, 2010. *VDI Heat Atlas*. 2nd edition, Springer-Verlag, Berlin Heidelberg, 919-932.

Received 30 October 2013
Received in revised form 11 March 2014
Accepted 24 July 2014