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A pilot-scale condensing waste heat exchanger

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Abstract

This paper presents a calculation algorithm, design assumptions and results of studies concerning a flue gas/water heat exchanger with the condensation of water vapor contained in flue gas from the combustion of brown coal. The algorithm was used for design calculations of a pilot-scale heat exchanger with capacity of 380/312 kW. A cross-counter flow heat exchanger with capacity of 312 kW and coils made of PFA (perfluoroalkoxy polymer) was designed and installed. Waste heat is recovered from flue gas produced by a pulverized brown coal fired subcritical steam boiler operated in a power unit with capacity of 370 MW_e. The heat exchanger was theoretically divided into a non-condensing part (sensible heat recovery) and a part with the condensation of water vapor contained in flue gas (recovery of sensible and latent heat). The point of the division is the temperature of flue gas in the stream core (higher than near the pipe wall) where the condensation of water vapor occurs on the pipe surface. The heat transfer in the non-condensing part was calculated using the same formulas as for the economizer in a pulverized-fuel boiler, while the calculations of the heat and mass transfer in the condensing part were performed using the VDI algorithm. The results of the thermal calculations and the geometry of the heat exchanger together with the place of installation of the entire test rig are presented. The results of the calculation are then compared with the test results. Good correlation was achieved between the test results and the assumptions and results of the design calculations. Calculations for full scale exchanger for 370 MW_e brown coal fired power unit showed a 1.18% net efficiency increase with improving wet flue gas desulphurization process (EUR 3.7 million annual savings of fuel consumption and CO_2 emission).

Keywords: condensing heat exchanger, waste heat, power boiler, chimney loss reduction, latent heat

1 Introduction

In light of the Energy Efficiency Act of 2011 and Directive 2012/27/EU, which introduce, inter alia, an obligation to achieve a 20% reduction in the primary energy consumption by 2020 (27% by 2030) and revised EU ETS Directive with 40% reduction of greenhouse gases emission to 2030 (new proposition of 55% to 2030 is now introduced), solutions that improve efficiency, such as waste heat recovery from flue gas from a power boiler, in addition to offering obvious benefits for the boiler operator, will be able to receive support in the form of so-called "white certificates". Additional benefits for the power plant's owner are the lower cost of fuel used and lower CO_2 emission due to higher efficiency of the power plant. The latter is particularly attractive due to the growing price of carbon emission allowances (EUA).

Boiler outlet flue gas temperature has the greatest impact on thermal efficiency. The latent heat from water vapor content in flue gas is recovered in small, residential gas and oil fired water boilers mainly. Only a few examples of calculations concerning a condensing heat exchanger used for flue gas from hard coal combustion were found in the literature [1]; [2]; [3]. In all cases pilot-scale heat exchangers were used to validate the analytical models. Models, with various modifications, referred to the well-known Colburn-Hougen model [4] used for calculations of heat and mass transfer at low water vapor concentration in the main stream with a high portion of an inert gas. In other works, mathematical models and calculations of condensing heat exchangers used at combustion of fuel oil [5]; [6], natural gas [7]; [8]; [9]; [10] and biomass [11] were presented. No design and validation of the model for brown coal flue gas was presented in any publication.

There are only a few examples of commercial scale waste heat exchangers and possible utilization of recovered low temperature heat (sensible and latent) in a power unit or CHP plant. One commercial scale condensing heat exchanger system was started in February 2015 in Białystok CHP, Poland. The heat (sensible and latent) is recovered from the flue gas water vapor leaving one of the two 75 MW CFB biomass boilers working in a 55 MW_e duo-block. The heat in the amount of 20 MW maximum is transferred to utility water and directed to the district heating system of Białystok city. The recovered heat increases the thermal power and efficiency of the CHP plant. Additionally, all the recovered heat is "green" heat. Other examples are flue gas condensers following quenching at the wood-fired power plant at Örtofta and design of a heat exchanger for Lomma power plant in Sweden. Additionally, condensate from water vapor (after treatment) can be used as make-up water for a boiler and a district heating network [11]. The model of waste heat recovery from flue gas presented in [12] referred to a 900 MW_e hard coal fired power unit, but without condensation of water vapor in flue gas.

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Construction of new highly efficient power units combined with the desire to improve the efficiency of existing ones mean that waste heat sources not used to date will become attractive. These sources include flue gas from brown coal or biomass fired steam boilers, which discharge huge amounts of heat, including water vapor condensation heat, into the atmosphere. An additional benefit is the positive impact of the reduced temperature of the flue gas entering the wet flue gas desulfurization system (make-up water stream reduction). In the near future this may attract increased interest from the power industry.

2 Mathematical model

Calculations of heat transfer in the condensing heat exchanger are based on a model composed of [13] - Fig. 1:

- coefficient of heat transfer from the flue gas to the condensate film/pipe surface,
- coefficient of heat transfer from the condensate layer to the pipe wall,
- thermal resistance of the pipe wall,
- coefficient of heat transfer from the pipe wall to cooling water.

The thermal resistance of the condensate film on the outer surface of the pipe is omitted.

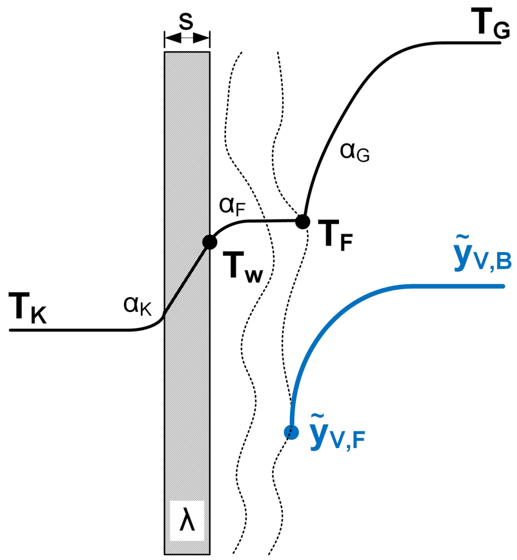

Figure 1: Temperature and concentration profiles in the condensate layer and immediate vicinity

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

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

From among the above coefficients, the largest problem in calculations is the resistance of the boundary layer of flue gas containing water vapor . This results from the necessity to calculate the partial pressure of water vapor in this layer near the pipe wall (condensate film on the pipe), where the composition of the flue gas varies both in the cross-section of the stream and along the length of the exchanger (Fig. 1). The mass transfer in the heat transfer path should also be taken into account. If calculations concern flue gas, the presence of superheated water vapor in the flue gas make the problem even more complicated.

In the VDI model, the specific flux of heat transferred from flue gas to the condensate film layer forming on the wall is

$$\dot{q} = \alpha_G E_T \left(T_G - T_F\right) \tag{2}$$

here the Ackermann correction factor E_T is

$$E_T = \frac{\phi_T}{1 - e^{-\phi_T}} \tag{3}$$

in which the dimensionless mass flow rate is

$$\phi_T = \frac{\widetilde{C}_{p,V}}{\widetilde{C}_{p,G} Le^{0.6}} \ln\left(\frac{\widetilde{y}_{G,F}}{\widetilde{y}_{G,B}}\right) \tag{4}$$

Using the correction factor it is possible to include in the calculations the mass transfer that occurs simultaneously with the heat transfer.

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

$$\dot{q} = k' \left(T_F - T_K\right) \tag{5}$$

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

$$k = \left(\frac{1}{\alpha_F} + \frac{s}{\lambda} + \frac{1}{\alpha_K}\right) \tag{6}$$

In order to determine the amount of condensing water vapor and the thermal power of the heat exchanger, it is necessary to determine the temperature of the condensate film layer on the pipe surface. The following balance equation is used for this purpose [13]

$$k'\left(T_F - T_K\right) = \alpha_G \phi_T \left(\frac{\Delta \widetilde{h}_v}{\widetilde{C}_{p,V}} + \frac{T_G - T_F}{1 - e^{-\phi_T}} \right) \quad (7)$$

In order to determine the heat transfer area, it is necessary to know the predominant mechanism in the process of condensation of water vapor from flue gas. For this purpose, it is necessary to calculate the temperature of the condensate film layer on the pipe at the inlet and outlet of the condensing part of the heat exchanger (using equation 7). Then, using the heat balance equation

$$\begin{aligned} \dot{q} &= k\left(T_{dew} - T_K\right) \\ &= k'\left(T_F - T_K\right) \\ &= \alpha_G \left(T_{dew(\widetilde{y}_{V,B})} - T_F\right) \end{aligned} \quad (8)$$

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 recent one is lower than 0.5, the condensation process may be regarded as controlled only by the heat transfer mechanism (without taking mass transfer into account). The heat transfer area can be then calculated from the well-known equation

$$A = \frac{\dot{Q}}{k' \Delta T_{log}} \quad (9)$$

where ΔT_{log} is the logarithmic difference of the inlet and outlet temperatures of the condensate film layer and cooling water in the counterflow heat exchanger.

In turn, when $\left(\frac{k'}{\alpha_G}\right)_2 > 2$, the condensation process is controlled by the mass transport mechanism, and the necessary heat transfer area can be calculated from the equation [13]

$$\begin{aligned} A = &\frac{\dot{N}_G}{n_G \beta_G \overline{\overline{y}}_{G,F}} \\ &\cdot \left(\frac{\widetilde{y}_{G,F,2}}{\widetilde{y}_{G,B,1}} - \frac{\widetilde{y}_{G,F,2}}{\widetilde{y}_{G,B,2}} + ln \frac{\frac{\widetilde{y}_{G,F,2}}{\widetilde{y}_{G,B,1}} - 1}{\frac{\widetilde{y}_{G,F,2}}{\widetilde{y}_{G,B,2}} - 1} \right) \end{aligned} \quad (10)$$

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

If heat transfer is the predominant mechanism at the inlet to the condensing part, while mass transfer is the predominant mechanism at the outlet, the heat transfer area should be calculated using equations 9 and 10, and then averaged.

3 Assumptions and input data

The test rig will be used for studies on the condensation of water vapor contained in flue gas from the combustion of brown coal. The aim is to determine inter alia the following: the impact of the temperature and mass flow rate of cooling water and the volumetric flow rate of flue gas on the thermal power and temperature of the cooling water leaving the heat exchanger. It will also enable verification of the correctness of the algorithm used for calculating the heat transfer area.

An assumption was made that the pilot-scale heat exchanger to be built will be a cross-counter flow membrane heat exchanger with an in-line arrangement of coils. For the needs of the calculation, reference was made to a numerical library of thermodynamic parameters of water and water vapor, physico-chemical properties of air and flue gas components, as well as functions for heat transfer coefficients. The authors also created their own functions to calculate the following values: Prandtl number, specific heat at constant pressure, thermal conductivity coefficient for water and air, diffusion coefficients, and properties of wet and dry gas mixtures [14]; [13].

The heat exchanger will be installed in the by-pass flue gas duct from the 370 MW_e brown coal fired subcritical steam boiler BB-1150, parallel to the main duct. Flue gas will be drawn by an additional fan from the main duct located downstream of the ID flue gas fan and will be returned to this duct before the ID fan.

The heat exchanger was divided theoretically into two parts: the non-condensing part and the part with the condensation of water vapor from flue gas (Fig. 2). The parameter used for division of the heat exchanger is the temperature of the flue gas stream core, where the condensation of water vapor starts.

The heat exchanger will be built from PFA fluoroplastic polymer pipe coils with an outer diameter of 10 mm and wall thickness of 1 mm. The velocity of flue gas was assumed to be approx. 10 m/s. The velocity of cooling water flowing in the pipes will be less than 1 m/s. The flue gas will be cooled from 160°C to 50°C. The condensate will be carried away from the heat exchanger. All thermal calculations will performed for the values of arithmetic means of the flue gas and cooling water thermodynamic parameters between the inlet and the outlet, separately for

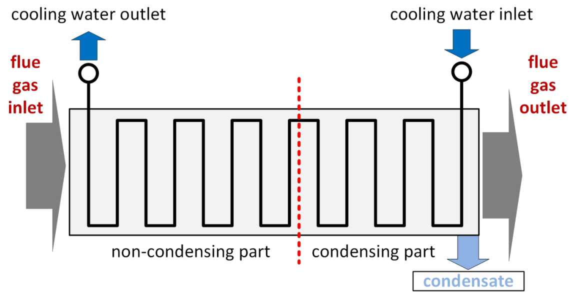

Figure 2: Schematic diagram of the cross-counter flow condensing heat exchanger

the non-condensing and condensing parts of the heat exchanger.

Design assumptions for the model are presented in Table 1.

The calculations of the heat transfer in the non-condensing part of the heat exchanger were carried out in the same way as for the economizer in a pulverized fuel-fired steam boiler [15]. The main assumption is to adopt a value of inlet flue gas temperature for the condensing part of the heat exchanger at which the condensation of water vapor on pipes begins. Based on the calculations of the temperature distributions for flue gas, the condensate, pipe wall area, and coolant at a specified mass flow rate, it was determined iteratively that this temperature is 66°C, which is slightly higher than the water dew point temperature (64.8°C) for the inlet parameters of flue gas (Fig. 3).

In the non-condensing part of the heat exchanger, flue gas is cooled from 160°C to 66°C, transferring only sensible heat (no condensation occurs) to the cooling water. The flue gas then flows to the second part of the heat exchanger, where it is cooled to 50°C, below the water dew point. Then the flue gas transfers the sensible heat and the condensation heat from water vapor contained in the flue gas. At a temperature close to the water dew point, the condensate is carried away from the heat exchanger. In the first part, the driving force of the heat transfer is the temperature difference, while in the second part - the difference between water vapor temperatures and concentrations caused by the gradient of the water vapor concentration in flue gas and near the condensate film layer on the pipe wall. An additional thermal resistance appears in the form of a layer of inert gas (flue gas) in which the content of water vapor is decreasing. The impact of the thermal resistance of the condensate layer on the pipe wall was assumed as negligibly low [13], while the resistance of the heat conduction through the pipe wall was taken into account (it reaches a significant value, if the coil of the exchanger is made of fluoroplastic polymer – the thermal conductivity coefficient for PFA is 0.25 W/mK only).

Table 1: The input data and the assumptions for calculations of the pilot-scale heat exchanger

Quantity	**Unit**	**Value**
Inlet flue gas temperature	°C	160
Inlet flue gas pressure	kPa	104.8
Partial pressure of water vapor in inlet flue gas	kPa	24.8
Coefficient of excess air in inlet flue gas	-	1.3
Outlet flue gas temperature	°C	50
Outlet flue gas pressure	kPa	102.1
Partial pressure of water vapor in outlet flue gas	kPa	12.3
Inlet temperature of cooling water	°C	20
Thermal efficiency of the heat exchanger	%	98
Technical and elemental analysis of as-received brown coal		
Low heating value	MJ/kg	7.75
Moisture	%	51.50
Ash	%	11.40
C content	%	23.10
H content	%	1.90
O content	%	10.52
N content	%	0.32
S content	%	1.26

The calculations of the water temperature at the outlet of the heat exchanger were performed with a view to utilizing the heat contained in it (attempts were made to maximize the value of this temperature and also to maximize the amount of condensing water vapor, but these two goals were in opposition to each other). In order to perform calculations for both parts of the heat exchanger, a system of equations was built describing the heat and mass transfer and computing the most important design parameters (heat transfer area, number of pipes and pipe rows, number of media intersections, etc.). According to the results of the calculations, approx. 50% of the water vapor contained in the flue gas will be condensed when the flue gas is cooled to the temperature of 50°C.

4 Results of the design calculations

The results of the calculations presented in Table 2 indicate that a heat exchanger with capacity of 380 kW designed in accordance with the aforementioned assumptions would require a total length of approx. 28 m. This exceeded the technical capabilities in terms of installing such a large device in the power plant. A decision was made to limit the length of the heat exchanger to approx. 10 m by reducing the thermal power of the heat exchanger – the outlet temperature of flue gas leaving the heat exchanger was raised to 55°C. The results of the repeated calculations for the heat exchanger with capacity of 312 kW are shown in Table 2 (values in brackets and bold).

Raising the temperature of the flue gas leaving the heat exchanger caused a reduction in the heat transfer area and the length of the exchanger, a small change in the ratio of thermal powers of the two parts, as well as:

- a change in the temperature of the division into non-condensing and condensing parts – currently it is 80°C (Fig. 3); the condensation of water vapor on the pipe wall starts at this temperature,
- an increase in the partial pressure of water vapor at the outlet of the heat exchanger to 15.7 kPa; due to this, approx. 36% of water vapor contained in flue gas will be condensed,
- a decrease in the temperature of cooling water at the outlet of the heat exchanger to 73.3°C.

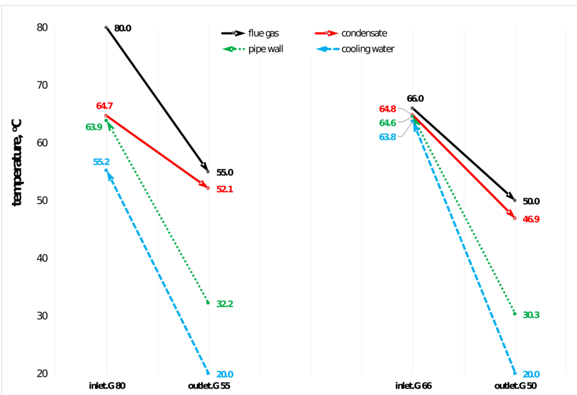

Figure 3: The temperature distribution at the inlet and at the outlet of the condensing part of the heat exchanger for the following parameters of flue gas (inlet/outlet): 80°C/55°C and 66°C/50°C

Further calculations showed that the values of $\left(\frac{k'}{\alpha_G}\right)_1$ and $\left(\frac{k'}{\alpha_G}\right)_2$ are respectively 0.0009 and 0.0418, and therefore it can be assumed that the heat transfer is the predominant mechanism in the water vapor condensation process.

Table 2: Results of the calculations of the condensing heat exchanger with capacity of 380 kW (values for the capacity of 312 kW are in brackets and bold)

Quantity	**Unit**	**Non-condensing part**	**Condensing part**
Thermal power of the heat exchanger	kW	124.4 **(106.0)**	257.0 **(206.1)**
Heat exchanger division temperature	°C	66 **(80)**	66 **(80)**
Temperature of cooling water at the outlet	°C	85.0 **(73.3)**	63.8 **(55.2)**
Cooling water flow rate	kg/s	1.4	1.4
Volumetric flow rate of flue gas (standard temperature and pressure)	m^3_{STP}/s	0.935 **(0.935)**	0.812 **(0.844)**
Condensate flow rate	kg/h	-	355 **(263)**
Temperature of the condensate	°C	-	55.8 **(58.4)**
Design dimensions of the heat exchanger (WxHxL)	m	0.8x0.4x7.9 **(0.8x0.4x2.8)**	0.8x0.4x20.1 **(0.8x0.4x6.7)**
Average velocity of flue gas in the heat exchanger	m/s	7.9 **(8.0)**	7.9 **(8.0)**
Design heat transfer area	m^2	79.3 **(26.9)**	151.7 **(50.6)**
Number of pipes per row	pcs	38	38
Number of media intersections	pcs	167 **(59)**	424 **(141)**
Cooling water flow rate	m/s	0.85	0.85

5 Test rig

The diagram below (Fig. 4) shows the concept of the pilot-scale test rig. The test rig includes:

- a heat exchanger with a system for cleaning the condensing part,
- a flue gas fan with an inverter, drawing flue gas from the flue gas duct located downstream of the ID fan for boiler flue gas,
- a cooling water pump with an inverter and pipelines,
- control and instrumentation (thermocouples, water, flue gas and condensate flow meters, hygrometers, water control and shut-off valves, a controller),
- a liquid trap at the outlet of flue gas from the heat exchanger,
- flue gas ducts with cut-offs.

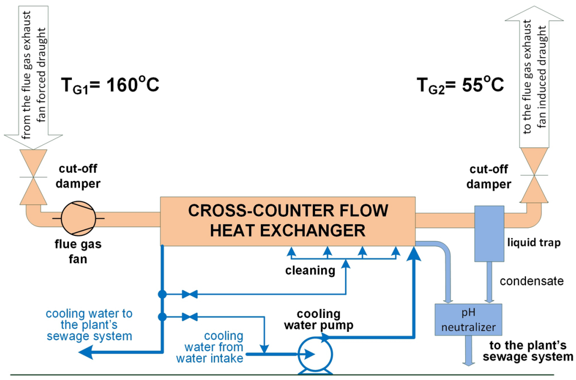

Figure 4: Schematic diagram of the pilot-scale test rig

The photos below show a condensing waste heat exchanger with capacity of 312 kW installed at the branch of the flue gas duct downstream of an electrostatic precipitator of the BB-1150 steam boiler. A pipe cleaning system was designed in the condensing part in order to remove fly ash deposits formed during the condensation. To enable observation of the processes occurring inside the heat exchanger, glazed inspection ports were designed in one of its side walls (Fig. 5 and 6). To collect the condensate and to measure the condensate flow rate (and also the flow rate of water discharged when cleaning the pipes), an outlet was designed in the form of a gutter running through the entire length of the heat exchanger bottom, together with a flow meter and thermocouple installed at its end. Ports were made in the side walls and in two vertical cross sections in order to mount the thermocouples. These ports are also used when measuring the flue gas temperature distribution along the path of the flue gas flow and in selected vertical cross sections of the heat exchanger. Hygrometers were installed at the inlet and at outlet of the heat exchanger to measure the moisture content in flue gas.

The operation of the test rig is automatically controlled with a *SIEMENS SIEMATIC* controller.

Figure 5: The 312 kW heat exchanger with the following elements visible (from left to right): cooling water manifold, pipe cleaning water manifold, inspection windows, a liquid trap, and the outlet flue gas duct

Figure 6: The interior of the heat exchanger with PFA coils and the cooling water outlet manifold

6 Results of the research

The diagram below (Fig. 7) shows sample measurement values obtained from the start of the operation to nominal rated steady state of the test rig.

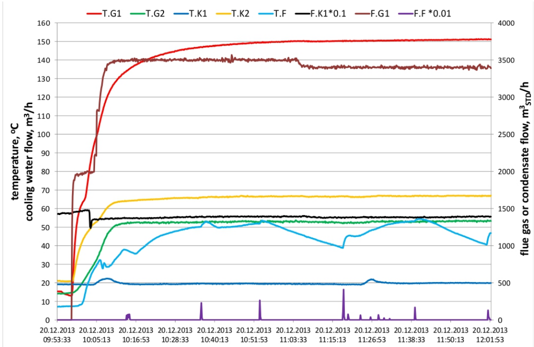

Figure 7: Sample temperature and flow rate profiles obtained near the nominal rated operating point of the pilot-scale test rig

The values of the outlet flue gas and cooling water temperatures as well as the values of thermal power transferred to cooling water, which were measured near the nominal rated operating point of the 312 kW heat exchanger, correlate well with the values obtained from the calculations. The shape of the condensate temperature plot (light blue line) correlated well with the time of water cleaning injection, but the measured rate of outgoing flow from installation (condensate and cleaning water together) is not adequate for the amount of injected cleaning water. The problem was to continuously and accurately measure such a small flow. It was made by a periodically opened valve with a flow meter installed downstream.

A small discrepancy between the calculated and measured values of the temperatures of the outlet flue gas and cooling water leaving the heat exchanger may result from: a higher flow rate of cooling water, a lower temperature of the inlet flue gas (by approx. 9°C), an excess air factor value (1.347) higher than the design value, and small differences in the composition of the coal combusted during the measurements (moisture= 50.9%, ash= 13.5%, sulfur= 0.54%, LHV= 7.77 MJ/kg). The last two differences change the calculated water dew point from 64.8 °C to approx. 64.3°C and lowered the absorbed latent heat flux. Some of above specified parameters (inlet flue gas temperature, excess air in flue gas and the composition of coal) are independent of the authors.

Calculations made for a full scale condensing heat exchanger for a subcritical 370 MW_e brown coal fired power unit showed a 1.18% net efficiency increase. The fuel consumption and CO_2 emission dropped about 123 000 and 109 000 Mg per year respectively [16]. If we assume the fuel cost EUR 1.5 per 1 GJ (LHV= 8 MJ/kg) and the CO_2 emission cost EUR 20 per 1 EUA we have EUR 1.5 million and EUR 2.2 million of annual savings respectively. There are some additional effects of water vapor condensation in the heat exchanger e.g. lower water demand for the flue gas desulfurization absorber, because of lower flue gas temperature and lower fly ash and SO_x content in flue gas in the absorber inlet.

7 Summary

This paper presents a calculation algorithm, design assumptions and results of studies concerning a flue gas/water heat exchanger with the condensation of water vapor contained in flue gas from the combustion of brown coal. The algorithm was used for design calculations of a pilot-scale heat exchanger with capacity of 380/312 kW. A cross-counter flow heat exchanger with capacity of 312 kW and coils made of PFA polymer was designed and installed. Waste heat is recovered from flue gas produced by a pulverized brown coal fired subcritical steam boiler operated in a power unit with capacity of 370 MW_e. The heat exchanger was divided theoretically into a non-condensing part (sensible heat recovery) and a part with the condensation of water vapor contained in flue gas (recovery of sensible and latent heat). The point of the division is the temperature of flue gas in the stream core (higher than near the pipe wall), where the condensation of water vapor on the pipe surface starts. The heat transfer in the non-condensing part was calculated using the same formulas as for the economizer in a pulverized-fuel steam boiler, while the calculations of the heat and mass transfer in the condensing part were performed using the VDI algorithm. Results of the thermal calculations and the geometry of the heat exchanger together with the place of installation of the entire test rig were presented. The results of the design calculation were then compared with the test results. There was good correlation between the test results and the assumptions and results of the design calculations. Calculations made for a full scale condensing heat exchanger for a subcritical 370 MW_e brown coal fired power unit showed a 1.18% net efficiency increase (EUR 3.7 million annual savings of fuel consumption and CO_2 emission). Additional effects of water vapor condensation in heat exchanger are limited water demand for the flue gas desulfurization absorber and lower fly ash and SO_x content in flue gas in the absorber inlet (wet scrubber effect). It reduces the operating costs of flue gas desulfurization and increases its efficiency.

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8 Symbols

A - heat transfer area, m^2

$\widetilde{C}_p$ - molar specific heat at a constant pressure, kJ/(kmol K)

F - volumetric flow, $m^3{}_{STP}/h$

$\Delta\widetilde{h}$ - molar heat of condensation, kJ/kmol

k, k' - heat transfer coefficient, W/(m^2 K)

Le - Lewis number

M - molar mass, kg/kmol

$\dot{m}$ - mass flow rate, kg/s, Mg/h

$\dot{N}$ - molar flow rate, mol/s

n - molar density, mol/m^3

$\dot{Q}$ - thermal power, kW

$\dot{q}$- specific heat flux, kW/m^2

s - pipe wall thickness, m

T - temperature, °C

$\overline{x}$ - mean value

$\widetilde{y}$ - molar fraction, kmol/m^3

Greek symbols

α - convective heat transfer coefficient, W/(m^2 K)

β - mass transfer coefficient

λ - pipe material thermal conductivity, W/(m K)

ϕ_T - dimensionless mass flow rate

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B - flue gas stream core

dew - water dew point temperature

F - condensate film layer

G - inert component, flue gas

K - cooling water

V - water vapor

w - wall

1 - inlet

2 - outlet

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