EXPERIMENTAL INVESTIGATION OF WATER VAPOR CONDENSATION FROM FLUE GAS IN DIFFERENT ROWS OF A HEAT EXCHANGER MODEL

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ABSTRACT

Condensing heat exchangers (HE) are used in many applications because of their usability with different fluids and a wide operating range in terms of pressure, temperature and power. Despite that, the thermal design of condensing heat exchangers is still not optimized, due to the complexity of the condensation process and lack of related research.

This paper presents results of experimental investigations of biofuel flue gas water vapor condensation on vertical tubes in different rows of a tube bundle in a crossflow. The effects of water vapor mass fraction, inlet flue gas temperature and the Reynolds number on heat transfer when the inlet cooling water temperature and flow rate are constant were analyzed. The results obtained showed that the main parameters which had the most influence on the condensation process were the water vapor mass fraction in the flue gas and its temperature at the inlet to the test section. In the range of inlet flue gas Reynolds numbers investigated, the Re effect on heat transfer was not as significant as the effect of the parameters indicated above. However, the Re number had some influence on the heat transfer variation along the inline tube bundle. A comparison of the average Nu number in the case of dry air with the experimentally determined average Nu number, even with low condensable gas mass fraction (6 %), showed that it increased considerably. A correlation was proposed, which helps to determine the average Nu number for the heat exchanger in the range of experiments performed.

Keywords: Biofuel flue gas, water vapor condensation, vertical tubes bundle, crossflow, different rows heat transfer, condensate mass along HE.

1. INTRODUCTION

Energy demand will be increasing considerably in the coming years as a result of population growth and economic development [1]. Having in mind that the increase concerns related to climate change and rising energy prices [2], it is necessary to find ways to increase energy efficiency by using renewable and alternative fuel sources or by improving the efficiency of old energy systems by optimizing them.

Power and boiler plants are primary sources of energy. The flue gas from these plants, if directly released into the atmosphere, contains a considerable amount of moisture in vapor form. If the flue gas temperature could be reduced below the dew point, the water vapor would begin to condense and sensible heat (convection) as well as latent heat (condensation) of the flue gas could be recovered.

Also, water in the form of condensate could be used repeatedly in the plant after its acidity had been neutralized [3, 4]. The latent heat losses from power plants that incinerate fossil fuels could be reduced to ~15 percent, and in the case of biomass incineration the reduction could be up to 10 percent by recovering the waste heat from the flue gas [5, 6]. Thus, the release of flue gas directly into the atmosphere is neither profitable nor ecological, whereas the recovery of latent heat greatly improves the overall efficiency of the energy system. The way to recover this energy is to use specially designed recovery systems. The condensing heat exchanger (condensing economizer) is the key device for this type of a system [7].

Water vapor condensation is an important phenomenon in various industrial applications. Therefore, many condensation investigations have been done with pure water vapor or water vapor containing small amounts of non-condensable gases (NCG) inside tubes and different channels [8-10, etc.]. These investigations mainly conclude that high heat and mass transfer rates can be obtained in the presence of pure water vapor condensation and that even small amount of non-condensable gases in water vapor complicate the condensation process and thus deteriorate heat and mass transfer.

Pure vapor condensation in horizontal and vertical tube banks in crossflow is also widely investigated, but mostly concerning the average heat transfer of heat exchangers. In [11, 12], condensation heat transfer is investigated in different rows (10 tube rows) along the vapor flow. It was obtained that the mean heat transfer coefficient was decreasing along the rows: in the last row it was almost twice smaller than in the first row.

In the case of flue gas, the amount of non-condensable gases is high and the amount of water vapor is small. Such a combination also influences heat transfer, and in the case of a horizontal tube and horizontal tube heat exchangers it was rather widely investigated [13-15, etc.]. The numerical investigation of heat transfer characteristics from a horizontal tube in a crossflow when the water vapor mass fraction was 0.02–0.12, the flue gas inlet temperature 145°C and the Reynolds number was 3400 is described in [13]. The analysis of the Nusselt number distribution around the perimeter of the tube showed that the Nu number was different. The lowest Nusselt number was at the separation point (which was at ~100 degrees around the tube) where both latent and sensible heats reached minimum values because of an increased condensate film and a thicker boundary layer. Beyond the separation point, the Nusselt number increased because of the condensate recirculation zone. The results showed that the water vapor mass fraction is an important parameter in terms of condensation heat transfer. As the water vapor fraction increased, the condensation process intensified. The investigations also showed that the latent heat transfer coefficient increased rapidly with the increase in the water vapor fraction, while the sensible heat transfer coefficient remained the same.

Condensation in a tubes bundle in the case of flue gas from a natural gas boiler was studied in [14]. The tubes bundle was composed of six horizontal tubes of ~22 mm outer diameter and three tubes in one row with a horizontal pitch of 33.7 mm and vertical pitch of 43.4 mm. The flue gas entering the test section was of 120-140 °C, the dew point temperature 55 °C and the flue gas Reynolds number was varied in the range of 2100-13500. This study revealed that the condensation-convection heat flux in the second stage of tubes located downward the flue gas flow direction was bigger than in the first stage. The enhancement was due to the wake turbulence of the first row tubes.

Heat transfer analysis in a horizontal tubes bundle in a crossflow when the flue gas Reynolds number is between 1000 and 3600 and the water vapor mass fraction is about 0.12 was presented in [15]. In this study the outside tube diameter was 5.5 mm, transversal and longitudinal pitches were 13.6 mm and 8.8 mm, respectively and the inlet flue gas temperature was within the range of 60-88 °C. The test section was composed of 78 tubes arranged in 12 rows. The results showed that heat transfer could be complicated by the effects of condensate inundation. Also, with an increase in the flue gas inlet temperature, it was found that the condensation rate decreased linearly and the convective heat flux increased linearly. The Nusselt number variation in different bundle rows revealed that the parameter increased with increasing flue gas velocity and in the first rows it was higher than in the last rows. For the analyzed heat exchanger, the authors proposed only correlation for calculation of convective heat transfer of tubes bundles, which was based on Zukauskas correlation [16].

In the case of vertical tubes and vertical tubes heat exchangers, condensation heat transfer investigations are rather limited [17-19]. Analysis and comparison of water vapor condensation heat transfer in a vertical tube bundles and in a single tube in crossflow is described in [17]. The tubes used in the test section were 34 mm outer diameter. The vertical tubes bundles were composed of three tubes with a spacing of 46 mm. The length of the tubes was 2.23 m. The tubes' surface temperature was measured using thermocouples welded at tube surfaces, the gas temperature was measured by thermocouples installed between the tubes. During the experiments, the air mass fraction was changed from 5 % to 65 %. The results showed that when the air mass fraction was low (5 %), the average heat transfer rate of the tubes bundles was about two times higher in comparison with that of the single tube. With an increase in the air mass fraction, the heat transfer rate of the bundles rapidly decreased, while for the single tube, the decrease was much less pronounced. The experimental results also showed that when the air mass fraction reached about 40 %, the heat transfer rates of the single tube and the tubes bundle were almost the same. With a further increase in the air mass fraction, the difference between the single tube and the tubes bundle was negligible. The authors also concluded that the main factors which affected the condensation heat transfer were the water vapor flow rate and the thickness of the non-condensable gas layer. It was also determined that for the same air mass fraction the condensation heat transfer increased with pressure increase. The authors proposed a formula for the average heat transfer coefficient calculation, which was valid for the pressures in the system between 0.3 MPa and 0.6 MPa, and for the gas and the tube outer surface temperature difference from 5 °C to 22°C.

Experimental results of flue gas condensation in three groups of connected multi-row three vertical staggered tube bundle heat exchangers were presented in [18]. One heat exchanger was composed of 72 tubes (6 transverse and 12 longitudinal rows) with the outside diameter of 38 mm. In the study, natural gas was incinerated and routed to the test section. As this gas contained low amount of water vapor, the water was injected into the gas flow before entering the test section in order to increase the water vapor mass fraction (4-16 %) in the flue gas. The effects of various parameters (flue gas velocity, water vapor fraction, cooling water flow rate and temperature) on the condensate collection rate and heat transfer were analyzed. The results showed that the convective heat transfer was at least two times lower and its characteristics were different when compared to the average heat transfer in each of the condenser groups with the change in the flue gas velocity. With the increase in the flue gas velocity from 2.5 m/s to 7 m/s, the convective heat transfer increased linearly in all three groups, meanwhile the average increased only up to a certain flue gas velocity and then started to decrease. Also, the differences between the average heat transfer cases in all three groups were rather significant, but with convective heat transfer they were negligible. The authors also emphasized that the water vapor fraction was an important factor influencing the heat transfer. The change of the water vapor fraction from 4 % to 16 % resulted in an increase in the average heat transfer coefficient from about 125 W/(m^2·°C) to about 300 W/(m^2·°C). The increase in the cooling water flow rate from 3 l/min to 18 l/min also had a similar effect on the total heat transfer. The analysis of the cooling water temperature variation at the inlet revealed that a temperature increase from 10 °C to 16 °C induced lower condensate mass.

Experimental results of the condensation of water vapor from flue gas on a vertical tube bundles heat exchanger in a crossflow were presented in [19]. In total, six stages of this type of heat exchangers were connected in series. Flue gas was generated by the boiler incinerating oil, gas and coal and flue gas inlet temperature at the test section was about 150 °C. The tubes used in heat exchangers had an outside diameter or 12.7 mm. Transverse and longitudinal pitches of the tubes were 0.7 and 2.0, respectively. During the experiments, the flue gas containing CO_2, O_2, H_2O, and N_2 gases was supplied with a flow rate from 150 to 192 kg/h. It was determined that the condensation efficiency decreased almost linearly with the increase in the cooling water inlet temperature. The results showed that a cooling water temperature increase from about 24 °C to almost 38 °C gave a decrease in the condensation efficiency of about 30 %, i.e., from 75 % to 46 %. It was observed that in the first two stages of heat exchangers, almost no condensate was collected because they mostly operated for the reduction of flue gas temperature, which, at the inlet to the test section (i.e., before reduction), was about 150 °C. Most of the condensate was formed in the fourth stage, and in stages five and six the condensate collection was decreasing. During this study, an analytical model was also developed for predicting average heat transfer and condensation efficiency. The accuracy of the results predicted by the model was verified experimentally. The results predicted by the model were compared with the experiments and the error was within the 10 percent margin.

The literature analysis showed that there are wide investigations of condensation heat transfer performed in tubes and different channels in the case of pure water vapor or water vapor containing small amounts of non-condensable gases but large amounts of water vapor. In the case of flue gas, which contains rather small amounts of water vapor, investigations with horizontal tubes and horizontal tube heat exchangers are also rather wide. There are some investigations of heat transfer in different rows of horizontal tube bundles or even of local heat transfer around the perimeter of the tube. For vertical tubes bundles, some condensation heat transfer investigations are also conducted; however, we have not succeeded in finding how condensation heat transfer depends on the position of the tubes in the bundle. Therefore, this paper, for the first time, presents and discusses experimental condensation-convection results in different rows of a vertical tubes heat exchanger in a crossflow of gas containing rather small amounts of water vapor, using flue gas generated from incineration of wood pellets. The obtained results provide the quantification of the effects and the mapping of the distribution of heat transfer behavior in the bundle. Therefore, this study also contributes to the knowledge on the local processes occurring in different rows in heat exchangers.

Nomenclature

G volumetric flow rate (m^3/s)	*Greek letters*	*Subscripts*
P pitch of the tubes	α heat transfer coefficient (W/m^2·°C)	cd condensate
r latent heat of condensation (kJ/kg)	η condensation efficiency (%)	cw cooling water
c_p specific heat (kJ/kg·°C)	λ thermal conductivity (W/m·°C)	f flue gas
d outer diameter (m)		i row
l length (m)		in inlet
m mass flow rate (kg/s)		lo longitudinal
Nu Nusselt number		ln logarithmic
Re Reynolds number		out outlet
S area (m^2)		t condensation-convection
t temperature (°C)		tr transverse
Q heat quantity (W)		v water vapor
q heat flux (W/m^2)		w wall

2. EXPERIMENTAL SETUP

The experimental setup with the test section used to investigate condensation heat transfer on vertical tubes in a crossflow is shown in Fig. 1. The main components of the experimental setup are the following: a model of serpentine tube heat exchanger (the test section), dampers, a boiler, a flue gas exhauster, cooling water lines and flue gas lines.

Figure 1. *Experimental setup (a): (1) boiler, (2) flue gas exhauster, (3) damper, (4) inlet flue gas temperature and humidity measurement point, (5) test section, (6) serpentine tube bundle, (7) condensate collection bottle, (8) bypass, (9) chimney; test section (b); view of tubes bundle (c)*

The flue gas was generated by incinerating wood pellets in an automatic boiler. The maximum power of this boiler is 50 kW.

The gas composition of non-condensable fractions, when incinerating wood pellets, was determined using a flue gas analyser IMR2000. The measured typical composition is presented in Table 1.

Table 1. Flue gas composition during incineration of wood pellets at the inlet to the test section

N_2, vol. %	CO_2, vol. %	O_2, vol. %	CH_4, vol. %	CO, ppm	SO_2, ppm	NO_2, ppm	NO, ppm
79.1	8.6	11.9	0.051	184	1	0.2	63.4

During the experiments, the flue gas from the boiler was supplied to the inlet of the test section (heat exchanger). The stability of the flue gas entering the test section was ensured by using the flue gas exhauster as shown in Figure 1. Besides, the adjustment of dampers (see Figure 1, position 3) also allowed to increase the stability of the flue gas flow as a certain part of the flue gas was entering the test section and another part flew directly into the bypass pipe. So, the combination of use of the flue gas exhauster together with the bypass pipe and the dampers ensured the stability of the flue gas entering the test section.

When incinerating almost dry wood pellets, the water vapor mass fraction in the flue gas was about 6 % at the inlet to the test section. However, in reality biofuel may be rather wet. In order to simulate wet incinerated biofuel the water was sprayed into the furnace of the boiler using fog nozzles, and this increased water vapor mass fraction to about 14 % and decreased the inlet flue gas temperature. At the outflow of the test section, the flue gas was directed to the chimney and exhausted to the atmosphere.

The test section used in the experiments in cooperation with Brunel University London was designed at the Lithuanian Energy Institute (LEI) and the experiments were also performed at LEI. The heat exchanger of the test section was composed of three longitudinal stainless steel tubes of serpentine shape (outer diameter 18 mm, wall thickness 2 mm) forming tube bundles, which were placed inside a rectangular stainless steel frame. The internal dimensions of the frame are 1.230×0.108×0.302 m (L×W×H). Thus, an inline tube bundle arrangement (transverse pitch $P_{tr} = s_1/d = 1.5$ and longitudinal pitch $P_{lo} = s_2/d = 7.2$; here s_1 is the transverse distance between centers of the pipes (27 mm) and s_2 is the longitudinal distance between centers of the pipes (130 mm)) was formed with eight vertical tube rows in the longitudinal direction and three in the transverse direction. The total outer surface area of the serpentine tube bundles (i.e., total heat transfer area) was 0.49 m^2.

The surface temperature of the eight tube rows was measured in the central (middle) tube. Three shielded 0.2 mm diameter thermocouples were embedded and welded in the tube's wall at it's front, side and back as shown in Fig. 1. Flue gas temperatures were measured at the inlet and outlet of the heat exchanger (i.e., before the first and after the last row of the tubes) and also in the cross sections between the tube rows (three thermocouples in each cross-section at upper, middle and lower parts of the cross-section) as presented in Fig. 1. Cooling water inside the tubes was circulating in the counter-current direction to the flue gas flow. Cooling water temperatures were also measured at the inlet (in the water mixer) and the outlet (in the water mixer) and along the central tube in seven elbows of the tubes (see Fig. 1). All the temperatures were measured using calibrated chromel-copel type thermocouples (wire diameter 0.2 mm, accuracy ±0.3 %).

At the inlet of the test section, the flue gas relative humidity (RH) and temperature were measured using an electronic temperature-humidity probe KIMO C310 (RH accuracy measurement ±0.88 %, temperature ±0.3 %). From the temperature and relative humidity the absolute humidity (g/m^3) was calculated. Since the flue gas flow rate (m^3/h) was determined, the water vapor flow rate was also obtained.

During the experiments, all the thermocouple readings were collected using a Keithley automatic data acquisition system (accuracy ±0.25 %). Information about the stability of the boiler operating regime, and of the boiler parameters, such as flue gas temperature at the exit from the boiler, boiler supply and water return temperature, and suction, was recorded with an automatic system KD7 every 5 minutes.

The inlet flow rate of the flue gas was measured using a nozzle with installed Pitot-Prandtl tubes connected to a manual micromanometer (accuracy ±1.5 %). The cooling water flow rate supplied to the test section was measured with an electromagnetic flowmeter (Isoil, accuracy ±0.4 %), which has been installed in the cooling water supply line at the inlet to the test section. Calibration by the weighing method using balancing valves had been performed to have the same cooling water flow rate in all three serpentine tubes in the test section.

The condensate along the tube bundles was collected at four positions from every two rows (from rows no. 1+2, 3+4, 5+6 and 7+8) in collection bottles (see Fig. 1). The condensate flow rate was determined based on the condensate mass collected in the bottle and duration (0.5-1.0 hour depending on the condensate flow rate) during which the condensate was collected. Every empty bottle before the experiment and the bottles with condensate after the experiment were weighted on the scales (Beurer, accuracy ±1 g).

The entire test section as well as the chimney was thermally insulated using Rockwool insulation. The thickness of the insulation used for the test section was 8 cm, and for the chimney 5 cm.

The experimental section was insulated with two layers of insulation: 5 cm and 3 cm (total 8 cm). The thermocouples were installed between the 3 cm thickness insulation layer. Calculations showed that thermal losses relative to the heat loads during the experiments were in the range between 2-3%.Therefore, it was assumed that the test section had been insulated sufficiently.

3. METHODOLOGY

Experiments were carried out at different inlet flue gas velocities (i.e., Reynolds number Re_{in}), temperature, and humidity. During these experiments, the cooling water flow rate and its inlet temperature were not changed. All the experiments were performed at the flue gas atmospheric pressure.

The parameters were calculated at 8 positions along the heat exchanger (represented by index *i* of the row in following formulas). All the properties (c_p, λ, etc.) of the flue gas, i.e. the mixture of flue gas and water vapor, were calculated using formulas for mixture property calculation [20], taking into account the decrease in the mass fraction of water vapor and flue gas temperature along the heat exchanger.

Specific heat flux obtained by the cooling water was determined as:

$$q_{t_i} = Q_{cw}/S_i = \frac{m_{cw_i}\cdot\left(c_{p_i+1,}\cdot t_{cw,out,_i} - c_{p_i}\cdot t_{cw,in,i}\right)}{\pi\cdot d\cdot l_i} \tag{1}$$

where Q_{cw_i} is heat quantity obtained by the cooling water, W; S_i is the outer surface area of the serpentine tubes, m^2; m_{cwi} is cooling water mass flow rate, kg/s; c_p is the specific heat of the water at outlet and inlet temperatures, kJ/kg·°C; $t_{cw,in,\ ,-out}$ are the temperatures of the cooling water at the inlet and outlet of each elbow of the tube, °C; *d* is the outer diameter of the tube, m; *l* is the length of the tube measured along the tube axis between the elbows of the tube.

The condensation-convection heat transfer coefficient (W/m^{2}·°C) for every row was calculated as follows:

$$\alpha_{t_i} = q_{t_i}/\left(t_f - t_w\right)_i \tag{2}$$

where t_f the average of the flue gas temperature measured at the center of the test section before and after respective row, °C; t_w is the measured outer surface tube temperature at a single longitudinal location, °C.

The condensation-convection Nusselt number for each row:

$$Nu_{t_i} = \propto_{t_i} \cdot d/\lambda_{fi} \tag{3}$$

where λ is the thermal conductivity of the flue gas based on t_f, W/m·°C.

To evaluate the performance of the heat exchanger, the condensation efficiency parameter was used [19]. Condensate capture rate or condensation efficiency was calculated according to the formula:

$$\eta = (\Sigma m_{cd_i}/m_{v,in}) \cdot 100 \tag{4}$$

where Σm_{cd} is the total condensate flow rate, kg/s, $m_{v,in}$ is the water vapor flow rate at the inlet into the test section, kg/s.

Heat quantity released due to flue gas cooling and water vapor condensation was obtained:

$$Q_f = m_f \cdot \left(t_{f,in} \cdot c_{pf,in} - t_{f,out} \cdot c_{pf,out}\right) + m_{cd} \cdot r \tag{5}$$

where m_f is the flue gas flow rate, kg/s; c_{pf} is the specific heat of the flue gas, kJ/kg·°C; m_{cd} is condensate flow rate, kg/s; r is latent heat of condensation, kJ/kg.

Average heat transfer coefficient:

$$\bar{\alpha} = \Sigma q_{t_i}/\Delta t_{ln} \tag{6}$$

where Δt_{ln} is the logarithmic mean temperature, calculated as:

$$\Delta t_{ln} = \frac{(t_{f,in} - t_{cw,out}) - (t_{f,out} - t_{cw,in})}{ln\left[(t_{f,in} - t_{cw,out})/(t_{f,out} - t_{cw,in})\right]} \tag{7}$$

Average Nusselt number:

$$\overline{Nu} = \bar{\alpha} \cdot d/\lambda \tag{8}$$

In this formula, thermal conductivity λ was evaluated based on average flue gas flow temperature $\bar{t} = (t_{f,in} + t_{f,out})/2$.

Calculation of the flue gas inlet Reynolds number is based on the formula:

$$Re_{in} = \frac{G_{f,in} \cdot d \cdot \rho_{f,in}}{S_{min} \cdot \mu_{f,in}}, \tag{9}$$

where $G_{f,in}$ is the volumetric flue gas flow rate, m^3/s; S_{min} is the minimal cross section of the bundle, m^2; $\rho_{f,in}$ is flue gas density at the inlet to the test section, kg/m^3 and $\mu_{f,in}$ is flue gas dynamic viscosity at the inlet to the test section, kg/(m·s).

The average $\overline{Re}$ number was calculated as:

$$\overline{Re} = \frac{G_f \cdot d \cdot \rho_f}{S_{min} \cdot \mu_f}, \tag{10}$$

Where $G_f = (G_{f,in} + G_{f,out})/2$ is the average volumetric flue gas flow rate, m^3/s; ρ_f is flue gas density and μ_f is flue gas dynamic viscosity based on $\bar{t}$.

4. RESULTS AND ANALYSIS

4.1. Water vapor mass fraction ~6 %

Figure 2a presents a typical distribution of temperatures and the convection-condensation Nusselt number (Nu_t) along the test section rows when the inlet Reynolds number (Re_{in}) was 320, the inlet flue gas temperature (i.e., temperature before the first row of tubes, see Fig. 1) 95 ºC and the water vapor mass fraction (i.e. condensable gas mass fraction m_{v-in}) was 6 %. The dew point temperature at the inlet to the test section for this case was about 50 ºC. The inlet cooling water temperature ($t_{cw,in}$) during all the experiments was about 35 ºC and flow rate was 2 kg/min.

Figure 2. *Temperature, Nusselt number, dew point temperature (a), condensate mass (b) variation along the test section when $Re_{in}=320$, $t_{in}\approx95^oC$, $m_{cw}=2kg/min$, $t_{cw,in}\approx35^oC$. (1) average flue gas temperature, (2) average outside tube surface temperature, (3) cooling water temperature, (4) dew point temperature, (5) Nusselt number*

The results in Figure 2a indicate that the flue gas temperature up to row no. 4 decreased from 95 ºC to 65 ºC (difference 30 ºC) and then from row no. 4 to row no. 8 it decreases from 65 ºC to 44 ºC (difference 21 ºC), the average outside tube surface temperature decreased from 42 ºC to 36 ºC and average cooling water temperature increased only by a few degrees: from 36 ºC (from the inlet at row no. 8) to about 40 ºC (at the outflow of row no. 1). The distribution of the temperatures indicates that the condensation on the tubes surfaces was present in all rows because the temperature of the tube's outside surface was always below the dew point temperature (dew point temperature was calculated using formulas presented in [21] and taking into account the collected condensate mass in respective row). This was also confirmed by the data on condensate collection (Fig. 2b). Moreover, the linear flue gas temperature decrease is visible in the temperature distribution (Fig. 2a, curve 1) up to rows no. 4–5. However, from row no. 6, the rate of the temperature decrease slowed down because of a more intensive condensation of water vapor (Fig. 2b). It should be kept in mind that along the tube bundle, the water vapor mass fraction in the flue gas was also decreasing. Yet, because of the low cooling water temperature at the inlet (i.e., at row no. 8), the condensation in the last rows of the tube bundle was rather intensive and the condensate mass collected here was the largest (Fig. 2b).

The results of the Nusselt number distribution (Fig. 2a, curve 5) show that at the beginning of the test section, the Nusselt number was the smallest. The same was true for the condensate, where its mass (Fig. 2b) was about 0.45 kilograms per hour from a square meter of the heat transfer surface ($kg/(h\cdot m^2)$). Further, as the collected condensate mass along the test section was noticeably increasing (rows no. 5÷6), the Nusselt number was also continuously increasing and reached the value of about 55 at row no. 7. Then, at rows no. 7÷8, as some amount of the water vapor had already condensed in rows no. 1÷6, the temperature difference between the flue gas and the tube wall temperature became small and the condensate mass decreased. The same was obtained for the Nusselt number. In general, the Nusselt number correlated well with the mass of the collected condensate (Fig. 2b). The condensation efficiency in this case was determined to be about 14 %.

The results at a higher inlet flue gas Re_{in} number are presented in Figure 3. As the figure shows, the flue gas temperature decreased from 115 ºC to 61 ºC, the average outside tube surface temperature decreased from 45 ºC to 36 ºC and average cooling water temperatures increased from 32 ºC (from inlet at row no. 8) to about 40 ºC at the outflow (at row no. 1). The dew point temperature at the inlet to the test section for the case presented was almost 52 ºC.

Figure 3. *Temperature, Nusselt number, dew point temperature (a), condensate mass (b) variation along the test section in the heat exchanger at $Re_{in}\approx930$, $t_{in}\approx115^oC$, $t_{cw,in}\approx35^oC$. (1) average flue gas temperature, (2) average outside tube surface temperature, (3) cooling water temperature, (4) dew point temperature, (5) Nusselt number*

In this case, the inlet flue gas temperature was higher than the dew point temperature and remained such along the test section. As for other aspects, the variation of the flue gas, the cooling water and surface temperatures remained similar to the previously analyzed case at smaller Re number.

In the case of a higher inlet flue gas Re number, the condensation-convection Nusselt number (Fig. 3a, curve 5) increased up to the value of about 65 when it reached row no. 7. After that, it decreased slightly. In general, the manner of the Nu number variation is similar in both Re_{in} cases (cf. Fig. 2a and 3a).

The collected condensate mass (Fig. 3b) from rows no. 1 to no. 6 was rather small and varied from 0.2 to 0.4 kg/(h·m^2). Hence, it looks like the influence of the condensation on heat transfer in these rows was less significant. However, the largest mass of the condensate was collected from rows no. 7+8, i.e., about 2.2 kg/(h·m^2), and the Nu_t number is also the highest in these rows (Fig. 3a, curve 5). The biggest condensate mass in rows no. 7+8 could be associated with the fact that the flue gas temperature decreases relatively fast (compared to vapor removal) along the heat exchanger and such a mixture becomes increasingly humid. Here, the cooling water temperature is of the lowest temperature and also enhances the condensation.

The condensation efficiency was determined to be about 10 %, and it was about 1.4 times smaller than in case of smaller inlet flue gas Re_{in} number.

4.2. Water vapor mass fraction ~14 %

As it has been mentioned above, in order to investigate the condensation dependency on water vapor content in the flue gas, experiments were performed with a water vapor mass fraction of about 14 % in the flue gas. The investigation was conducted under similar conditions as presented in the previous section (i.e., inlet flue gas Reynolds numbers, temperatures and cooling water flow rate were kept almost the same).

The dew point temperature for the flue gas at the inlet to the test section for the further case presented was in the range of 55–61 ºC. The inlet cooling water temperature was about 34 ºC.

Temperature variations along the test section are presented in Fig. 4a for the case of the smaller Re_{in}. It can be seen that variations of the flue gas, cooling water and surface temperatures are very similar to the case with smaller water vapor mass fraction (Fig. 2a). The tube surface temperature is below the flue gas dew point temperature, and therefore the condensation of water vapor occurs in all rows of the heat exchanger.

Figure 4. *Temperature, Nusselt number, dew point temperature variation along the test section when (a) Re_{in}≈320, t_{in}≈85°C, $t_{cw,in}$≈34°C (b) Re_{in}≈930, t_{in}≈100 °C, $t_{cw,in}$≈35 °C. (1) average flue gas temperature, (2) average outside tubes surface temperature, (3) cooling water temperature, (4) dew point temperature, (5) Nusselt number*

The cooling water temperature and tube surface temperature changed almost linearly (Fig. 4a, curves 3 and 2, respectively). The cooling water temperature increased by 10 ºC, and the outside tube surface temperature decreased by 6 ºC through the entire test section.

Condensation-convection Nusselt number variation (Fig. 4a, curve 5) is also similar to the case with the smaller water vapor mass fraction (Fig. 2a), and the maximum value of the Nusselt number is not very high, being about 120.

The condensation efficiency in this case was almost 49 %, meaning it was almost 4 times higher compared to the case with condensable gas mass fraction of ~6 %.

Figure 4b shows the temperature and Nu_t variation along the rows when Re_{in} is higher than that presented in Fig. 4a. Flue gas, cooling water and tube surface temperatures are very similar to the case of the smaller water vapor mass fraction (Fig. 3a). The tube surface temperature is always below the flue gas dew point temperature, and therefore the condensation of the water vapor occurs in all rows.

The profiles of the tube wall and cooling water temperatures (Fig. 4a and 4b) are rather similar. However, the cooling water temperature increases intensively (Fig. 4b) because of the heat from row no. 8 (i.e., from inlet) until row No. 4. As for the remaining rows up to the outlet, a much slower temperature increase is observed. A comparison of the cases when $Re_{in}\approx320$ and when $Re_{in}\approx930$ shows that for the latter case the decrease in the temperature of the outer tube surface from the beginning up to row no. 4 is steeper than that from row no. 4 up to row no. 8.

When the water vapor mass fraction was increased from 6 % to 14 %, the condensation efficiency increased from 10 % to 31 %, i.e., by about three times for the same Reynolds number ($Re_{in} \approx 930$). However, in the case of a higher Re_{in}, the condensation efficiency was smaller, almost half in comparison with a smaller Re_{in} ($Re_{in} \approx 320$) for the same water vapor mass fraction.

4.3. Comparison of the results for different inlet water vapor mass fractions and temperatures

Fig. 5a shows the comparison of the condensation-convection Nusselt number at different water vapor mass fractions (m_{v-in}) and flue gas inlet Re numbers. The results show that water vapor mass fraction has a significant influence on heat transfer.

Figure 5. *Comparison of (a) the Nusselt number along the test section of the heat exchanger at different water vapor mass fractions and inlet Reynolds numbers and (b) comparison of the Nusselt number at the same $Re_{in}\approx930$ but different flue gas inlet temperatures and the same (6 %) water vapor mass fraction and cooling water flow rate*

At a smaller Re_{in} (Figure 5a), with the increase in water vapor mass fraction from 6 % to 14 % the peak value of the Nu_t number went from 55 to 120. Hence, the higher water vapor mass fraction resulted in about two times a higher Nu_t.

For a higher Re_{in} number (Figure 5a), the increase in water vapor mass fraction resulted in an increased Nu_t peak value from 60 to 160. Hence, it can be concluded that the condensation highly depends on the water vapor mass fraction in the flue gas flow.

Comparison of the variation in the Nu_t obtained for different Re_{in} and the same water vapor mass fraction (Fig. 5a) in general shows that Nu_t also depends on Re_{in} number although the dependence is much less obvious than in the case of vapor mass fraction.

Additionally, the influence of the inlet flue gas temperature on the Nu_t number was analyzed. The variation of the Nusselt number at the same flue gas Re_{in} numbers but different flue gas inlet temperatures is presented in Fig. 5b. At a higher flue gas inlet temperature, the Nu_t number stays much lower almost throughout the entire test section compared to the case of a lower flue gas inlet temperature. This happens because a higher flue gas inlet temperature means worse conditions for condensation. Although the Nu_t number is also affected by the flue gas to tube wall temperature difference, the condensation efficiency at a higher inlet flue gas temperature was only about 4 %. Hence, the flue gas inlet temperature is also an important factor influencing condensation efficiency and Nusselt number.

4.4. Correlation of data and comparison with other investigation results

Experimental results on average $\overline{Nu}$ number based on logarithmic temperature difference for different WVMF is presented in Figure 6a. With the increase of water vapor mass fraction from 6 % to 14 % the increase of heat transfer is about 2.5 times, for both Re_{in} numbers (Fig. 6a). These results correlate rather well with the data presented in [18] for vertical staggered tubes bundle where with the increase of water vapor mass fraction from 4 % to 16 % the increase of heat transfer is about 2.4 times.

A correlation for calculation of the average $\overline{Nu}$ number due to condensation was proposed based on the analysis of the results:

$$\overline{Nu} = 104 \cdot m_{v-in} \cdot \overline{Re^{0.22}} \tag{11}$$

The formula is valid when the flue gas Re_{in} number is in the range between 320–930, inlet water vapor mass fraction (m_{v-in}) varies in the range of 0.06–0.14 at a cooling water inlet temperature of 35 °C with its flow rate of 2 kg/min and the flue gas inlet temperature is in the range between 85 °C and 115 °C. Such limitations indicate that it yet cannot be widely used for engineering calculations and of course require additional investigations.

The comparison of the experimentally determined and calculated average $\overline{Nu}$ is presented in Fig. 6b. The results in the figure show good agreement as the maximum relative error between them does not exceed 7 %. The changes in the cooling water inlet temperature and flow rate will require some corrections in the proposed formula; however, this is a task for further investigations.

Fig. 6a presents also a comparison between the results of this study and the average $\overline{Nu}$ number for dry air in the case of an inline tube bundle with maximum available transverse and longitudinal pitches 2.6×2.6 [22]. This average $\overline{Nu}$ number for dry air at Re numbers between 100 and 1000 can be calculated as:

$$\overline{Nu}_{dry-air} = 0.52 \cdot Re^{0.5} \cdot Pr^{0.36} \cdot (Pr/Pr_w)^{0.25} \tag{12}$$

Pr/Pr_w can be ignored in the above formula because the Pr number for the temperature interval typical for this study changes only negligibly. According to the above formula, the average Nu numbers obtained in the case of dry air for $Re_{in} \approx 320$ and $Re_{in} \approx 930$ are 8.4 and 14, respectively.

Thus, the enhancement of the average $\overline{Nu}$ number for $Re_{in} \approx 320$ in comparison with the dry air Nu number in the case of a water vapor mass fraction of 6 % was about a factor of three and about a factor of six when mass fraction was 14 %. When the Re_{in} was ≈ 930 and the water mass fraction was 6 %, the average Nu number increased by about a factor of two; when the water mass fraction was 14 %, it increased by a factor of four. The results presented in Fig. 6a also indicate that the average Nu number for dry air is affected more by the Re_{in}, while for the air containing a certain amount of water vapor it is affected less.

Figure 6. *Variation of the (a) average $\overline{Nu}$ number with $\overline{Re}$ number and (b) comparison of the experimentally determined and calculated average $\overline{Nu}$ number*

The obtained heat transfer results for our vertical tubes bundle were compared with the results for horizontal staggered tubes bundle [23] (Figure 6a). In [23] the correlation for calculation of the average $\overline{Nu}$ number for the horizontal staggered tubes bundle was proposed:

$$\overline{Nu} = 4.2682 \cdot Re^{0.7586} \cdot Pr^{\frac{1}{3}} \cdot \left(\frac{X}{1+\tilde{C}}\right)^{0.8876}, \tag{13}$$

where X is the condensation factor (X= x/(1-x)) and $\tilde{C}$ *is* the dimensionless concentration of ash particles ($\tilde{C} = c/\rho_f$). The above indicated formula is valid for $576<\overline{Re}<1160$, water vapor mass fraction $0<x\leq 25$ % and concentration of solid particles $1\leq c\leq 6.5$ g/m^3.

In our case the concentration of solid particles was very small, therefore using formula for X it was assumed to be 1 g/m^3. For the case of the horizontal tubes bundle, the average $\overline{Nu}$ number is more dependent on the $\overline{Re}$ number, exponent of $\overline{Re}$ is close to 0.8, which is characteristic for turbulent flows. In our case exponent of $\overline{Re}$ is 0.22. In general, the results of the average $\overline{Nu}$ number in the case of the horizontal tubes bundle show that it is smaller at smaller $\overline{Re}$ numbers, and larger at larger $\overline{Re}$ numbers in comparison with the results obtained for the vertical tubes bundle. The maximal discrepancy in the analysed $\overline{Re}$ numbers range for both water vapor mass fractions is up to 20 %.

Experiments performed by other authors do not exactly correspond to the inlet parameters of the flue gas, cooling water, etc. of our investigation, and therefore direct comparison with other authors' data is very complicated. However, a comparison with condensation efficiency in staggered tube bundles for a similar condensable gas mass fraction presented in [18] was also done (Fig. 7a). The main difference is that in [18], the cooling water temperature was 10–16 ºC and in our work it was higher. There was also a slight difference in water vapor mass fractions, and in [18] the flue gas temperature at the inlet to the test section was adjusted to be about 60 ºC. Results in [18] show that with the increase in the flue gas velocity (Re number), the condensation efficiency is decreasing. The results of our study indicate the same tendency.

In our case and in other studies [17, 19], the condensation efficiency usually increases with increasing water vapor mass fraction; however, in [18] the situation is the opposite. In [18] it is explained that this difference can be due to the fact that the heat exchanger used was not able to condense a higher amount of water vapor entering the test section. Besides, if in our experiments the flue gas temperature before the test section could be reduced to 60 ºC (close to the dew point temperature), this would result in some additional increase in condensation efficiency.

Figure 7. *Comparison of: (a) condensation efficiency at different water vapor mass fractions presented in [18] for staggered tube bundles with inline tube bundles obtained during current investigations, (b) heat balance.*

A comparison of heat balances, i.e., the condensation-convection ($Q_t = Q_{cw}$, obtained by cooling water) and that released by flue gas and water vapor condensation (Q_f+Q_{cd}), is presented in Fig. 7b. For $Re_{in} \approx 320$, the results indicate that the increase in water vapor mass fraction from 6 % to 14 % results in a doubling of the total heat quantity (from 400 to 800 W) and for $Re_{in} \approx 930$ it is almost twice as high (from 1000 W to 1800 W). The highest relative error between the heat quantities ($\Delta=[(Q_t+(Q_f+Q_{cd}))/Q_t]\times100$) was obtained for $Re_{in} \approx 320$ and a low water vapor mass fraction. With the increase in the water vapor mass fraction and the inlet flue gas Re number, the error between the heat quantities decreased (see Fig. 7b).

The uncertainties were evaluated using the methodology presented in [24]. For the Nusselt number, the highest uncertainties are at the end of the test section, where the temperature difference between the flue gas temperature and the tube wall temperature is the smallest. These uncertainties are between 6 - 12 %.

5. CONCLUSIONS

The results of the investigations of water vapor condensation heat transfer from biofuel flue gas in different rows of the model of the condensing heat exchanger at different water vapor mass fractions, Re_{in} numbers, and inlet flue gas temperatures, lead to the following conclusions:

1. In most tests performed, at the inlet of the tube the wall temperature was below the flue gas dew point temperature, and therefore the condensation of water vapor from flue gas was observed in all the rows along the heat exchanger. This was also confirmed by the condensate collected.
2. A significant variation in the Nu_t number occurred along the tube bundle, with its maximum recorded at rows no. 6–7.
3. The larger the water vapor mass fraction in the flue gas, the higher the condensation-convection heat transfer Nu number. The gas inlet temperature also had a noticeable effect on the condensation-convection Nu number in different rows of HE: the lower the flue gas inlet temperature, the better conditions for condensation resulting in an increase in the Nu_t number. With an increase in the Re number, the Nu_t number was also increasing.
4. Even a small water vapor mass fraction (about 6 %) increased the average Nu number by about at least two times compared to the dry air Nusselt number.
5. The analysis showed that the increase in water-vapor mass fraction from 6 % to 14 % resulted in the increase in the condensation efficiency from 14 % to 49 % for $Re_{in} \approx 320$. Meanwhile, at

a higher Re_{in} ($Re_{in} \approx 930$), the condensation efficiency for the same water-vapor mass fraction ranges increased much less: from10 % to 31 %.

6. A correlation for calculating the average $\overline{Nu}$ number for the heat exchanger was proposed for the range of the experimental results.

ACKNOWLEDGMENT

The research presented in this paper was partly funded from the European Union's H2020 programme iWAYS under grant agreement numbers 958274.

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