archives
of thermodynamics
Vol. **44**(2023), No. 4, 463–482
DOI: 10.24425/ather.2023.149729

Experimental investigation of limit parameters of direct contact condensation in the heat exchanger for waste heat recovery

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Abstract In the present study, a suitable composition of parameters has been obtained to provide an efficient process of cooling flue gas with complete condensation of water vapour from air-water vapour mixture on a water film in co-current upward flow in the tube of the direct contact heat and mass exchanger. The results showed that the value of the irrigation density depends on the velocity of the air-water vapour mixture and the initial vapour content and should be calculated from an empirical equation. The active pipe height depends on the velocity of the air-water vapour mixture and the initial vapour content and should also be calculated from an empirical equation. For example, if the initial vapour content of the air-water vapour mixture is 11%, the velocity of the mixture is 20.8 m/s the height of the channel should not exceed 0.460 m. The value of the water heating limit temperature increases from 46°C to 62°C with a change in the initial vapour content from 11% to 30%. The present experimental results could be helpful in the design of direct contact heat and mass exchangers for waste heat recovery.

Keywords: Direct contact heat mass exchanger; Direct contact condensation; Process of cooling flue gas; Air-water vapour mixture

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Nomenclature

c_p	–	specific heat, kJ/kg K
d	–	diameter, m
D_i	–	irrigation density, m^2/s
$\dot{m}$	–	mass flow rate, kg/s
h	–	enthalpy, kJ/kg
H	–	height of the test section, m
p_{par}	–	partial pressure of water vapour, Pa
Q	–	heat rate, W
Re	–	Reynolds number
T	–	temperature, °C
V	–	volume flow rate, m^3/s
W	–	average velocity, m/s

Greek symbols

φ	–	initial volume vapour content, %
ω	–	humidity ratio, kg water vapour/kg dry air

Subscripts and superscripts

cond	–	condensation
dr	–	droplet removal
in	–	tube inlet
lim	–	limiting value
m	–	air-water vapour mixture
min	–	minimum value
out	–	tube outlet
v	–	water vapour
w	–	water liquid

Acronyms

DCHME	–	direct contact heat mass exchanger
AWVM	–	air-water vapour mixture

1 Introduction

The recovery of a portion of the energy in hot stack gases has been practiced for many years. Recently, because of environmental reasons, the use of natural gas in boilers has become more common. As this clean fuel, compared to other fuels, containes more hydrogen than carbon, there is more water vapour in the exhaust flue gas accompanied by latent heat [1].

Natural gas-fired boilers have gradually dominated the process heating sector due to their high thermal efficiency, relatively low emissions, and convenient fuel delivery system [2]. Nowadays, we are facing a deep natural gas crisis. This year's winter gas season begins with extreme natural gas price levels, with the market tightness expected to continue well into 2026 [3]. When the cost of gas, which is now the most common fuel for industry, reducing heat losses with flue gases can be the most significant energy saving way and the main task. So, the recovery of the latent heat from flue gas to increase the thermal efficiency is very important for natural gas-fired boilers [1, 4]. The products of natural gas combustion contain a sufficiently large amount of water vapour, on the formation of which consumes part of the heat from combustion of fuel. The amount of latent heat loss of the exhaust flue gas from gas-fired boilers is very high. If we recover sensible and latent heats at the same time, the overall energy efficiency can be increased about 10–20% [4, 5].

The thermal efficiency of a gas boiler is mainly influenced by the flue gas temperature. The higher the flue gas temperature, the lower the thermal efficiency. When the flue gas temperature drops from 200°C to 60°C, only sensible heat is recovered and of the thermal efficiency increases slowly. However, if the flue gas is cooled to the dew point, which is around 55°C, the water vapour will start to condense, causing a sharp increase in thermal efficiency. Therefore, cooling the flue gas below the dew point is important for heat recovery [6].

Effective waste heat recovery of flue gases of boilers that uses the heat of condensation of water vapour, can be carried out by direct contact heat exchangers. Boilers can generally lose 20% of the combustion energy in the flue gas, and thanks to flue gas condensers, they can recover 50% of this lost energy depending to the operating conditions.

In direct contact heat exchangers it is possible to cool gases to a temperature below the dew point temperature because cooling the gases by direct contact coolant allows the use of the heat condensation component of the gases.

A direct contact heat exchanger has several advantages over surface heat exchangers such as the elimination of a metallic heat transfer surface between the fluids, which is prone to corrosion and fouling, and the reduction of heat transfer resistance. It can be operated at very low temperature differences or heat transfer driving forces and allows a high heat transfer coefficient (about 20–100 times higher than single-phase or surface type heat exchangers [7]).

In such apparatuses, the heat transfer from the exhausted combustion gases to the heated liquid takes place in direct contact with the coolants, which significantly improves the conditions for heat and mass transfer and allows nitrogen oxide (NO_x) emissions to be reduced. A significant reduction of harmful emissions occurs due to the scrubbing action of the heat exchanger as they are absorbed by the water [8].

These heat exchangers are also called a direct contact heat mass exchangers (DCHME) and it typically have gas and water as the two fluid streams. The transfer processes in such apparatuses occur under the contact between the liquid phase and the gas phase at the interphase surface. Depending on the method of interaction of the contacting phases, the most commonly used designs of heat exchangers are the following types: film, drop, bubble type DCHME [7]. In these types of heat exchangers, the principle of countercurrent movement of heat coolants is applied. The water flows from the top and the gas flows from the bottom. The advantage of the countercurrent flow is that the minimum temperature of the gases at the apparatus outlet is achieved, as the outgoing gases meet the coldest water. The water can be heated to a higher temperature due to a higher average temperature difference. However, the flow rate in DCHME of gas and liquid that moving in countercurrentl is limited by the flooding phenomenon. This phenomenon is characterized by the loss of flow stability and the entrainment of water from the DCHME under the action of a gas flow.

It is necessary to know the parameters that provide an efficient process for cooling gases with complete condensation of water vapour on a liquid film. The process of heat and mass transfer during the contact of gases and water is complex and depends on many factors. In some cases, at certain ratios of the parameters of a two-phase system of wet flue gases and water, the mode of condensation of water vapour can be reversed to the evaporation of water from the surface of the film, leading to a sharp decrease in the efficiency of the DCHME. Unfortunately, it was not possible to find any papers or studies on this issue.

However, the research has been divided into three lines. The first group of papers is devoted to new technologies for waste heat recovery and the efficiency of heat recovery. The technologies involving the use of direct contact heat exchangers for flue gas heat recovery have been extensively studied in the references [4, 9–14]. For example, papers [9–11, 14] present a boiler waste heat recovery technology based on heat pumps and the direct contact total heat exchangers. These technologies can improve the heat capacity of gas boilers by nearly 8–14%. These studies have focused on system performance and not on the heat and mass transfer characteristics during the direct condensation of water vapour from flue gases.

Analytical and experimental solutions of the heat and mass transfer process in different types of direct contact heat exchangers for flue gas heat recovery have been described in the second group [15–22]. In all these studies, however, the principles of countercurrent movement of heat coolants are applied. Men, Liu and Zhang [15] note that when a gas-water spray tower in a waste heat recovery system is optimized, the maximum heat recovery efficiency and the maximum heat transfer rate are achieved. Optimising a heat exchanger means determining the parameters at which the cooling of gases will be the most effective. The optimum spray water flow rate corresponding to the minimum thermal resistance [15], spray nozzle height and type (hollow and full cone), droplet diameter distribution, inlet air velocity and humidity, and water spray velocity are key parameters required to establish a clear understanding of the direct heat exchange performance in spray type heat exchangers [16].

Third group of papers are about heat and mass transfer between gas and liquid two-phase flows in pipes [20, 23–27]. Heat transfer and phase change processes of water droplets in humid airflow were investigated by performing experiments and numerical simulations of heat recovery from biofuel exhaust gas at 40–250°C, which is characteristic for condensing heat exchangers [20]. Dong, Qi and Zhang [24] developed a mechanistic model of flow and heat transfer for upward two-phase slug flow in vertical pipes. The authors found an improvement in heat transfer in two-phase slug flow compared to single-phase flow. This can be attributed to an increase in the turbulence of the liquid due to the injection of air and a reduction in the thickness of the thermal boundary layer due to the frequent alternation between liquid slug and the Taylor bubble. Jia and Dong [25] formulated a mathematical model for flow and heat transfer in two-phase annular flows, which based on the two-fluid concept. The effect of the orientations and flow parameters on the heat transfer was comprehensively investigated. The dependence of the two-phase heat transfer coefficients on the void fractions and pressure multipliers was quantitatively identified and a simple heat transfer correlation was developed. Bhagwat and Ghajar [26] analysed experimental data for void fraction, pressure drop and heat transfer coefficient for their dependence on phase flow rates and pipe inclination. The experiments were carried out in 0.0127 m internal diameter polycarbonate and 0.0125 m stainless steel pipes using air-water as the fluid combination. The results show that increasing the pipe inclination from the horizontal significantly affects the void fraction, total pressure drop and heat transfer coefficient at low values of gas and liquid flow rates. As the gas and liquid flow rates increase, the effect of pipe inclination on the two phase flow variables is observed to diminish. Dong and Hibiki [27] developed a robust correlation of the heat transfer coefficient for two-component two-phase slug flows in vertical pipes. This correlation can predict the heat transfer coefficient under various conditions such as vertical upward and downward flows, developing and fully developed flows, laminar and turbulent flows, and all two-phase flow regimes. In summary, the interaction processes of different flows under different conditions have been studied. However, the question of studying the process of direct contact condensation in co-current upward flow flue gases and water in the DCHME tube has not been investigated and remains open.

It is necessary to know the parameters that provide an efficient process of cooling gases with complete condensation of water vapour on a liquid film, because the process of heat and mass transfer during the contact of gases and water is complex and depends on many factors.

In this paper, measurements of the outlet temperature of the air-vapour mixture and the outlet temperature of the water along the single vertical tube of DCHME were caried out. The experiments were conducted under the conditions of co-current upward flow of air-vapour mixture and water. The effect of flue gas velocity, irrigation density, humidity ratio of the flue gas on limiting temperature of water heating and the active height of the tube were examined. It was also necessary to experimentally determine the value of limit temperature for water heating in direct contact with the upward co-current flow of the air-water vapour mixture (AWVM).

The main aim of the present study was to find an adequate composition of parameters assuring maximum efficiency of cooling flue gas with complete condensation of water vapour on a water film in DCHME with co-current upward flow. The basic design of the DCHME is shown in [28].

Based on an extensive literature review in the field of the already existing studies on direct contact condensation in the heat exchangers for waste heat recovery it was found that there are currently no scientific studies that specifically investigating the process of direct condensation specifically in the ascending co-movement of gases and water. Furthermore, there are no recommendations for the practical application of such a flow configuration in heat exchangers. We assume that the processes of heat and mass transfer in a co-current upward flow of gas and liquid are effective. It will also be easy to regulate the flow rate of gas and heated liquid in a wide range with a constant metal consumption of the DCHME. We expect that this will make it possible to design more efficient DCHME compared to the countercurrent heat exchangers considered above. The equations proposed in this paper can be used for engineering calculations of such heat exchangers.

2 Method

To solve the problems formulated above, experimental methods were used to study the heat and mass exchange processes during the condensation of water vapour from a flue gas with an accompanying upward movement with a water film. During the experiments, the wet flue gas was modelled as an air-water vapour mixture. Under laboratory conditions at the university, it is not always possible to use the combustion products of natural gas directly due to several reasons related to safety. However, in conducting such research, moist air can be used with a sufficient degree of adequacy [28]. Natural gas contains mainly methane. The combustion process is described by the formula: $CH_4 + 2O_2 = CO_2 + 2H_2O$. Oxygen is taken from the blast air, while the remaining nitrogen passes through the combustion chamber without undergoing any reaction. The amount of water vapour is determined by the composition of the fuel burnt, its humidity, and the air excess coefficient. The moisture content of the flue gases is 0.15–0.108 kg/kg$_{d.g}$, depending on the air excess coefficient ($\alpha = 1{-}1.4$) [29]. Thus, the composition of the combustion products of natural gas differs from that of moistened air by the replacement of oxygen by carbon dioxide, which does not affect the study of the condensation process of water vapour. A multitude of research and experiments in the field of flue gas analysis are conducted using a mixture of air and water vapour, making it a standard and reliable choice for modelling. When switching to flue gases, it is only necessary to recalculate their moisture content, which can be computed according to the ratio derived in [30].

The study was carried out in a specially constructed test facility. Figure 1 shows the diagramt of the installation, which consists of the test section, the systems for supplying liquid water, air and water vapour, and the measuring systems. The test facility was thermally insulated to prevent heat loss to the environment.

Test section (2) is a vertical cylindrical DCHE pipe of made of stainless steel with an internal diameter of 0.017 m and different heights (from 0.2 m to 1.404 m). The object of the research was an upward two-phase flow, consisting of the air-water vapour mixture flow and the water flow.

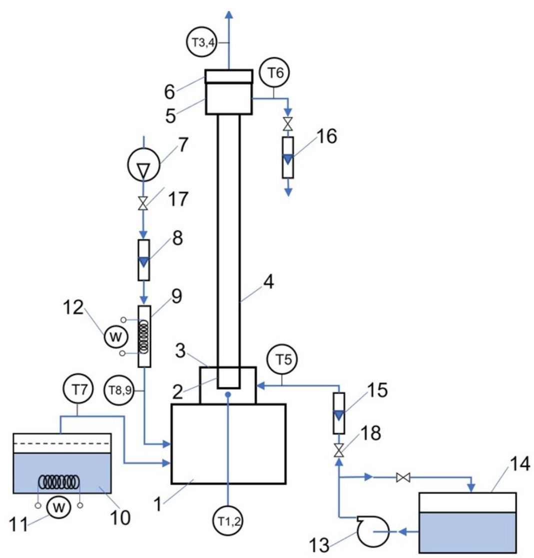

Figure 1: Schematic diagram of experimental installation: 1 – chamber for mixing air and water vapour; 2 – porous insert; 3 – water inlet chamber; 4 – test pipe; 5 – hot water collection chamber; 6 – separator; 7 – compressor; 8, 15, 16 – rotameters; 9 – electric heater; 10 – steam generator; 11, 12 – watt-meters; 13 – pump; 17, 18 – valves; T1–T9 – thermocouples.

The AWVM was produced by mixing air from a compressor (3) with a flow of water vapour produced by a special vapour generator (10) and supplied to the test section through the mixing chamber (1). The required amount of water vapour in the AWVM was provided by regulating the flow of the vapour and air. The upward two-phase flow was created by the flow of an AWVM and water, which was introduced into the working channel through the porous insert (2) and drawn into the upward flow in the form of a film due to the action of the AWVM.

The air was supplied to the unit from the compressor (3) through the rotameter (4). The air flow was regulated by a valve (17). An electric heater (5) was used to heat the air. The power supplied to the electric heater was regulated and controlled by voltage regulators and a wattmeter (12). The heated air was supplied into the mixing chamber (1), the walls of which were thermally insulated. The preheated mixing chamber (1) also received water vapour from the vapour generator (10). In order to prevent carry-over of droplet moisture from the steam generator a separator was installed in its vapour volume. The separator was made in the form of a perforated sheet with a baffle plate. The vapour flow was regulated by changing the voltage applied to the electric heaters of the steam generator. The heating power was controlled by a wattmeter (11). The air and water vapour were mixed in chamber 1. The formed AWVM entered the test pipe (2).

Cold water was supplied from the water tank by pump (13) through valve (18) into the water inlet camber (3). The water flow was monitored by the rotameter (15). The liquid was injected into the test pipe through the porous insert (2). After passing through the working channel, the water was separated from the air by the separator (6) and fell by gravity into the hot water collection chamber (5), from where it was drained. The water flow rate at the outlet of the unit was measured using the rotameter (16).

The air temperature upstream of the compressor was measured with dry (T8) and wet (T9) thermocouples. The temperature of the AWVM at the inlet to the working channel was measured with dry and wet chromel-copel thermocouples (T1) and (T2), and at the outlet – with dry and wet thermocouples (T3) and (T4). The water temperatures at the inlet and outlet of the working channel were measured with thermocouples (T5) and (T6). The thermocouple readings were recorded using a digital device.

The intensity of the heat and mass transfer processes was calculated from the measured values of the initial and final temperatures and the flow rates of the refrigerants using known methods. The thermodynamic properties of water and humid air were determined using the CoolProp library [31]. In this case $\omega_{m,\mathrm{in}}$, $\omega_{m,\mathrm{out}}$, $p_{\mathrm{par,in}}$, $p_{\mathrm{par,out}}$, $h_{\mathrm{par},w}$ were determined from the values of atmospheric pressure, dry and wet bulb temperatures.

The flow organization and the range of flow parameters of the phases mainly provided film flow regimes with the movement of the water film on the inner walls of the channel.

Before the experiments a series of set-up experiments were carried out. The design of the experimental set-up allowed some parameters to be varied independently while keeping others constant, which is very important when studying heat and mass transfer processes during condensation of water vapour from an AWVM on a liquid film. Table 1 lists the equipment used in the experimental rig along with their respective measurement errors.

The experimental data were processed by drawing up an energy balance shown in Table 2 and in Fig. 2(a).

The condensate mass flow rate was calculated as

$$\dot{m}_{\mathrm{cond}} = (\omega_{m,\mathrm{in}} - \omega_{m,\mathrm{out}})\dot{m}_{m,\mathrm{dry}}$$

Table 1: List of the elements used in the test rig with measurement accuracy.

Element	Type	Operating rate	Measurement uncertainty
Thermocouple	TCC-1172	0–500°C	±0.2°C
Barometer	–	97.8–100.8 kPa	±0.05 kPa
Water flow rotameter	RS-3	0.063 m^3h^{-1}	±2.5%
Air flow rotameter	RS-5	16 m^3h^{-1}	±2.5%
Air flow rotameter	RS-7	25 m^3h^{-1}	±2.5%
Wattmeter	D50162	1.5 kW	±2%
Digital device	A565	0–5 mA	±0.15%
Voltage regulator	RNO-250-10	0–250 V	–

and a water mass balance (Fig. 2(b)) is

$$\dot{m}_{w,\mathrm{in}} = \dot{m}_{w,\mathrm{out}} + \dot{m}_{w,\mathrm{dr}} - \dot{m}_{\mathrm{cond}}\,.$$

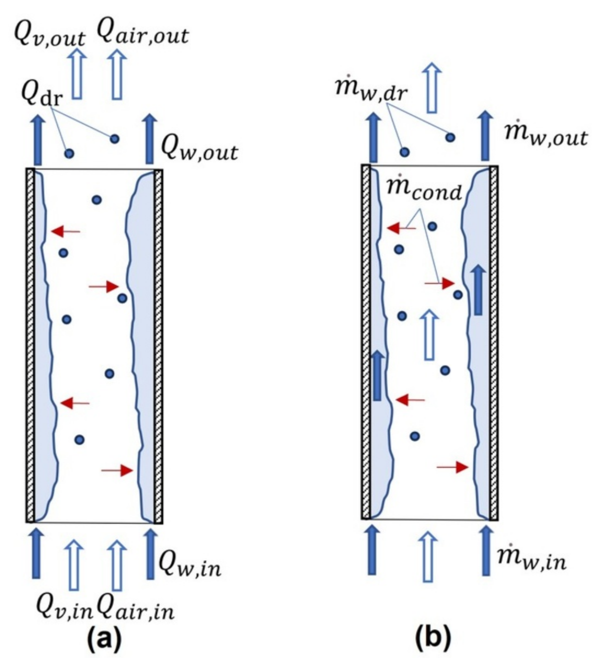

Figure 2: (a) Energy balance. (b) Water mass balance.

Table 2: Energy balance.

$Q_{\mathrm{air,in}} + Q_{v,\mathrm{in}} + Q_{w,\mathrm{in}} = Q_{\mathrm{air,out}} + Q_{v,\mathrm{out}} + Q_{w,\mathrm{out}} + Q_{\mathrm{dr}}$		
$Q_{\mathrm{air,in}}$	$\dot{m}_{\mathrm{air,in}}\, h_{\mathrm{air,in}}$	Inlet air heat rate
$Q_{v,\mathrm{in}}$	$\dot{m}_{w,\mathrm{in}}\, h_{w,\mathrm{in}}$	Inlet water vapour heat rate
$Q_{w,\mathrm{in}}$	$Q_{w,\mathrm{in}} = \dot{m}_{w,\mathrm{in}}\, c_{pw,\mathrm{in}} T_{w,\mathrm{in}}$	Inlet water heat rate
$Q_{\mathrm{air,out}}$	$\dot{m}_{\mathrm{air,out}}\, h_{\mathrm{air,out}}$	Outlet air heat rate
$Q_{v,\mathrm{out}}$	$(\dot{m}_{v,\mathrm{in}} - \dot{m}_{\mathrm{cond}}) h_{v,\mathrm{out}}$	Outlet water vapour heat rate
$Q_{w,\mathrm{out}}$	$Q_{w,\mathrm{out}} = \dot{m}_{w,\mathrm{out}} c_{pw,\mathrm{out}} T_{w,\mathrm{out}}$	Outlet water heat rate
Q_{dr}	$[\dot{m}_{w,\mathrm{in}} - (\dot{m}_{w,\mathrm{out}} - \dot{m}_{\mathrm{cond}})]\, c_{pw,T_{\mathrm{air,out}}} T_{\mathrm{air,out}}$	Entrained water droplets

3 Results and discussion

Firstly, the dependence of the temperature and humidity ratio of the AWVM at the outlet of the experimental section on the irrigation density was studied by varying the regime parameters (Fig.3), since one of the main characteristics of contact condensers is the outlet parameters of the air-water vapour flow.

The irrigation density was calculated using the formula

$$D_i = \frac{V_{w,\mathrm{in}}}{\pi d_{\mathrm{in}}} \,. \tag{1}$$

Figures 3(a) and (b) show the dependencie of the mixture outlet temperature $T_{m,\mathrm{out}}$ and humidity ratio $\omega_{m,\mathrm{out}}$ on the irrigation density D_i and the AWVM velocity W_m at the mixture inlet temperature of $T_{m,\mathrm{in}} = 105°\mathrm{C}$ and initial vapour content $\varphi = 11\%$ in the channel height $H = 1.404$ m. It can be seen from the figures that at low irrigation densities the AWVM velocity has almost no effect on both the outlet temperature of the mixture and the outlet humidity ratio of the AWVM flow. At high irrigation densities the effect of air-vapour mixture velocity on the mixture output parameters is more significant.

Secondly, the dependencie of the outlet temperature $T_{m,\mathrm{out}}$ and the humidity ratio $\omega_{m,\mathrm{out}}$ on the irrigation density D_i and the vapour content φ at the inlet temperature $T_{m,\mathrm{out}} = 105°\mathrm{C}$ and the velocity $W_m = 20.8$ m/s in the channel height $H = 1.404$ m (Figs. 3(c) and (d)). It can be seen that the outlet temperature $T_{m,out}$ increases as the initial vapour content increases. Also, as the initial water vapour content increases, the output moisture ratio increases, but this effect is more pronounced at low irrigation densities than at high ones. As can be seen in Figs. 3(e) and (f), the outlet temperature $T_{m,out}$ and the final humidity ratio $\omega_{m,\text{out}}$ are insignificantly dependent on the initial temperature $T_{m,\text{in}}$.

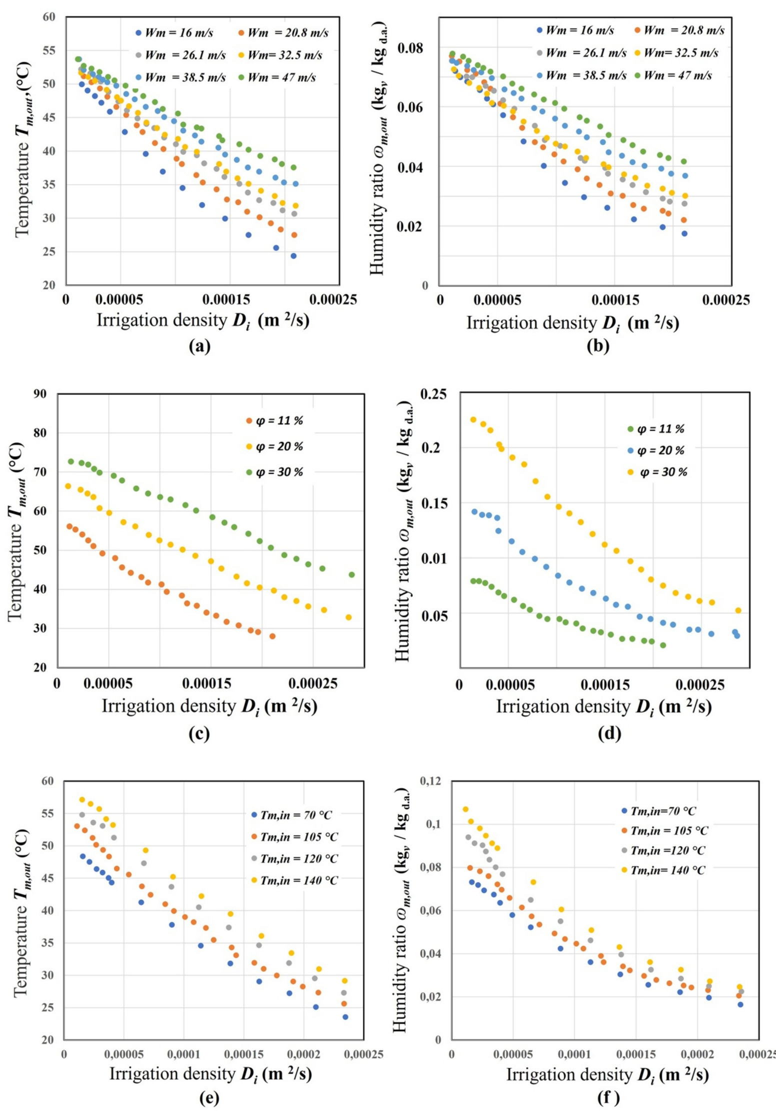

Figure 3: Outlet temperature and humidity ratio of the air-water vapour mixture versus irrigation density for different values: (a), (b) velocity of the AWVM; (c), (d) vapour content of the AWVM; (e), (f) initial temperature of the AWVM.

Figures 3(b), (d) and (f) show the higher humidity ratio at the outlet when the irrigation density is reduced. This can be explained by the fact that when a small amount of cooling water was supplied, it heated up almost immediately to the limit temperature at the entrance of the pipe. With further interaction, the moisture flow moved in the opposite direction, moistening the mixture. When more water was supplied, a greater amount of moisture condensed, and a smaller amount of moisture evaporated.

The experimental data in Fig. 4 are explained by the fact that at irrigation densities below the limit value $D_i < D_i^{\lim 1}$ the water heats up to the limit temperature $T_w^{\lim}$. The water heating limit refers to the reaching of the temperature of a wet bulb thermometer of the AWVM, which is approximately equal to the boiling temperature of water at a partial pressure of water vapour in the AWVM [32]. When a certain limit irrigation density $D_i > D_i^{\lim}$ is reached, the water is not heated to the limit temperature. In this case the water is heated to a certain equilibrium temperature from the cooling conditions of the mixture.

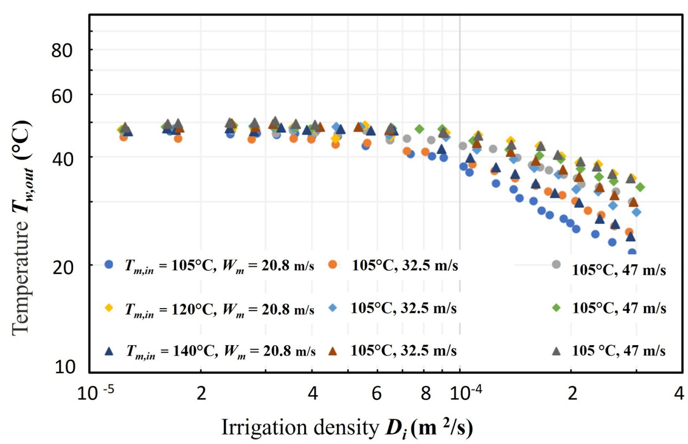

Figure 4: Effect of inlet temperature and velocity of air-vapour mixture on outlet water temperature.

The experiments showed that the value of the limit temperature of water heating increases with increase of the inlet vapour content φ (Fig. 5). This can be explained by the fact that, the value of the wet bulb temperature increases as the value of the vapour content increases. When the inlet vapour content is increased from $\varphi = 11\%$ to $\varphi = 30\%$, the limit temperature of the heated water increases from $T_w^{\lim} = 46°C$ to $T_w^{\lim} = 62°C$.

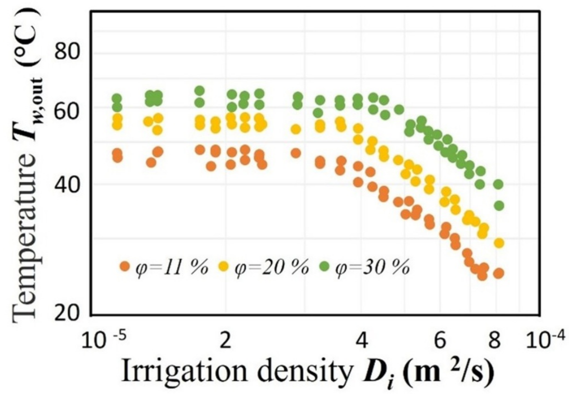

Figure 5: Outlet water temperature versus irrigation density for three different inlet vapour content.

Finally, the water temperature at the outlet of the test section depends on the following parameters:

$$T_{w,\mathrm{out}} = T_w^{\mathrm{lim}}(\varphi) \qquad \text{when} \quad D_i < D_i^{\mathrm{lim}}, \tag{2}$$

$$T_{w,\mathrm{out}}\,(D_i, W_m, \varphi) < T_w^{\mathrm{lim}} \qquad \text{when} \quad D_i > D_i^{\mathrm{lim}}. \tag{3}$$

The following formula is derived to determine the limit temperature of the water to be heated:

$$T_{w,\mathrm{out}} = T_w^{\mathrm{lim}} = 88.94\varphi^{0.3}. \tag{4}$$

Also, it is necessary to note, that the range of variation of the irrigation densities, at which water heats up to the limiting temperature, extends both with increasing of the initial vapour content φ (Fig. 5), and with increasing of the velocity W_m (Fig. 4), as one would intuitively expect. That is, the value of the limiting irrigation density depends not only on the velocity of the AWVM, but also on the initial vapour content: $D_i^{\mathrm{lim}} = f(W_m, \varphi)$.

Based on the above, the presence of the evaporation section is associated with the heating of the water to the limit temperature at irrigation densities below a certain limit value $D_i < D_i^{\mathrm{lim}}$, depending on the velocity and initial vapour content of the vapour mixture.

The temperature of the heated water T_w varies along the height of the tube for different values of irrigation density (Fig. 6). The Fig. 5 shows that there is a limit temperature of water heating, which for the given conditions ($T_{m,\mathrm{in}} = 105$°C, $\varphi = 11\%$, $W_m = 20.8$ m/s) is approximately $T_w^{\mathrm{lim}} = 46$°C. The water is not heated above this temperature. For each irrigation density there is a limit value to the working height of the test section, above which the temperature of the heated water does not increase further. For irrigation densities $D_i < D_i^{\lim}$ this is due to reaching the heating water temperature limit, and for $D_i < D_i^{\lim}$ it is due to the maximum cooling of the air-vapour mixture under energy balance conditions.

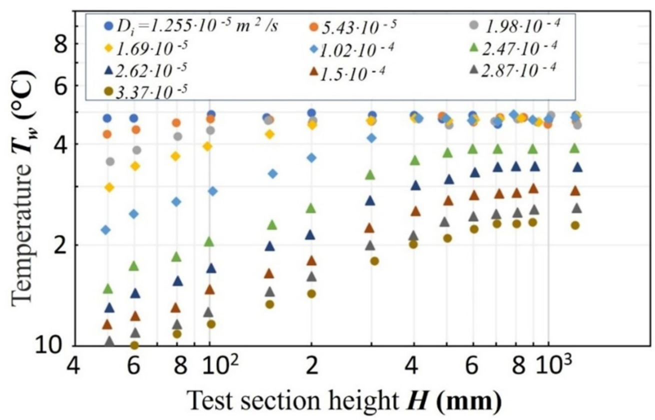

Figure 6: Heating water temperature distribution over the pipe height at initial mixture temperature $T_{m,\text{in}} = 105°\text{C}$, initial vapour content $\varphi = 11\%$ and mixture velocity $W_m = 20.8$ m/s.

Thus, at irrigation densities below a certain limit value $D_i < D_i^{\lim}$ the water is heated to the limit temperature $T_{w,\text{out}} = T_w^{\lim}$ at a certain working height of the test section. For the rest of the pipe section the opposite process takes place – the evaporation.

The experimental data allowed to obtain the dependence of the minimum operating height of the test section of the pipe on the irrigation density (Fig. 7). When $D_i < D_i^{\lim}$ at the working height of the experimental section the water was heated to the limit temperature. When $D_i > D_i^{\lim}$ the water was heated to a certain equilibrium temperature from conditions of energy balance. During the experiments, it was found that with increasing the velocity W_m and the initial vapour content φ, the minimum operating height of the test section decreases. The equation for determining the minimum operating height of the test section of the pipe is as follows:

a) the water has been heated to $T_w^{\lim}$

$$H_{\min} = 9.2 \cdot 10^8 D_i^{1.38} W_m^{-0.66} \varphi^{-0.4} \qquad \text{for} \quad D_i < D_i^{\lim}, \tag{5}$$

$$H_{\min} = 4.1 \cdot 10^3 D_i^{0.23} W_m^{-0.14} \varphi^{-0.2} \qquad \text{for} \quad D_i > D_i^{\lim}. \tag{6}$$

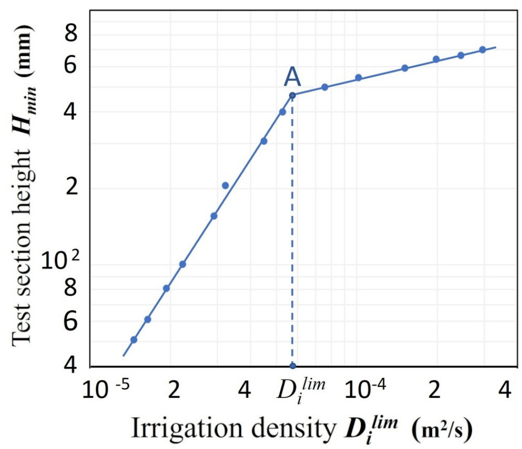

Figure 7: Limiting operating active height of the test section of the pipe versus irrigation density.

The aim of the present work is to find an adequate composition of parameters assuring maximum efficiency, that provide an efficient process of cooling gases with complete condensation of water vapour on a water film. For this purpose, we have obtained the dependency of the limiting operating active height $H_{\min}$ of the test section of the pipe on the irrigation density D_i (Fig. 7). The practical significance in Fig. 6 is point A. At the point A, the height of the test section ensures two conditions that are important in practice: heating of water to the limit temperature $T_w^{\lim}$ and, at the same time, the maximum irrigation density under this condition, which corresponds to the maximum heating rate of the tube.

The empirical relationships for the irrigation density and the height of the pipe at point A result from the joint solution of the following equations:

$$D_i^{\lim} = 2.22 \cdot 10^{-5} W_m^{0.45} \varphi^{0.17}, \tag{7}$$

$$H_A = 3.48 \cdot 10^{-1} W_m^{-0.04} \varphi^{-0.17} \tag{8}$$

or

$$(H/d)_A = 26.77 \cdot \mathrm{Re}_m^{-0.04} \varphi^{-0.17}. \tag{9}$$

Our study addressed the question of finding an adequate composition of parameters that provide an efficient process of cooling flue gas with complete condensation of water vapour in DCHME with co-current upward flows. Namely, we have obtained an empirical equation for determining the limiting heating temperature of water heating (4), the limiting irrigation density (7) and the active height of the tube of the DCHME (9).

On the basis of our experimental data, we realized that the processes taking place in the experimental section. At low irrigation densities and a sufficiently long working section, the cooling of the AWVM is first accompanied by condensation processes with a decrease in moisture content, and then by evaporation processes with some increase in humidity ratio of the mixture. As the irrigation density increases, the evaporation section decreases.

At irrigation densities above a certain some limit value, the cooling of the mixture is accompanied only by condensation processes. Since the presence of the evaporation process worsens the performance of the condenser surface due to its irrational use, we have determined the irrigation density, at which the mixture cooling is accompanied only by condensation processes.

According to the experimental results at irrigation densities below the limit value $D_i < D_i^{\lim}$, the water heats up to the limit temperature $T_w^{\lim}$. The limit heating of the water refers to reaching the temperature of a wet bulb thermometer of the mixture, which is approximately equal to the boiling temperature of the water at a partial pressure of water vapour in the AWVM [31]. When a certain limiting irrigation density $D_i > D_i^{\lim}$ is reached, the water has not been heated to the limit temperature. In this case the water is heated to some equilibrium temperature from the energy balance conditions. We have obtained the empirical Eq. (7) to determine the value of the limiting irrigation density that depends on the initial vapour content and the AWVM velocity.

In addition, it should be noted that for the first time the value of the water heating limit temperature has been experimentally determined for the ascending downward movement of the AWVM and the water film. These results converge with previous findings [33] that when comparing the two flow patterns of hot gas and cold water in the heat exchanger, the counterflow allows to get more heated water at the outlet. The results show that the value of the limit temperature of water heating is only 5–7°C lower than the corresponding value of the limit temperature for the countercurrent heating scheme according to Aronov [32].

The results obtained are consistent with the findings presented in [34] that the active height should lead to a complete condensation of the water vapour from the flue gas. Therefore, it directly affects the cost of the heat exchanger by its volume. The active height of the DCHME pipe can be determined by using empirical Eq. (9).

4 Conclusions

In the present study, the limit temperature of water heating, the irrigation density limit and the active height of a pipe direct contact heat exchanger were determined experimentally. At the same time, the velocity of the AWVM, the initial vapour content and the initial temperature of the AWVM were varied. From these data, the appropriate composition of parameters to provide an efficient flue gas cooling process with complete condensation of water vapour in DCHME with co-current upward flows was obtained. The results show that, for the first time, the value of the limit temperature of water heating at ascending upward movement of AWVM and water film was experimentally determined The value of the limit temperature of water heating increases from 46°C to 62°C when the initial vapour content changes from 11% to 30%. The dependence (7) for calculating the irrigation density at which there is no evaporation zone in the working channel is obtained. It is found that the effective height of the working channel should not exceed the value determined by Eq. (9), since further increase of the height of the working channel does not change the heat capacity of the working channel. For example, if the initial vapour content of the mixture is 11%, and the velocity of the air-vapour mixture is 20.8 m/s the height of the channel should not exceed 460 mm.

Finally, the experimental data obtained suggest that it is advisable to work with specific values of irrigation density and pipe height. The value of the irrigation density depends on the velocity of the AWVM and the initial vapour content and it should be calculated from empirical Eq. (7). The active pipe height depends on velocity of the AWVM and the initial vapour content and it should be calculated from (9). Under these conditions, the efficiency of the condenser will be high; its capital cost will be minimal. The present experimental results could be helpful in the design of direct contact heat exchangers for waste heat recovery.

Received 25 July 2023

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