

# Modelling of Technological Solutions to 4th Generation DH Systems

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**Abstract** – Flue gas evaporation and condensing processes are investigated in a direct contact heat exchanger – condensing unit, which is installed after a furnace. By using equations describing processes of heat and mass transfer, as well as correlation coherences for determining wet gas parameters, a model is formed to create a no-filling, direct contact heat exchanger. Results of heating equipment modelling and experimental research on the gas condensing unit show, that the capacity of the heat exchanger increases, when return temperature of the district heating network decreases. In order to explain these alterations in capacity, the character of the changes in water vapour partial pressure, in the propelling force of mass transfer, in gas and water temperatures and in the determining parameters of heat transfer are used in this article. The positive impact on the direct contact heat exchanger by the decreased district heating (DH) network return temperature shows that introduction of the 4th generation DH system increases the energy efficiency of the heat exchanger. In order to make an assessment, the methodology suggested in the paper can be used in each particular situation.

**Keywords** – Wood chip boiler house; condensing unit; direct contact heat exchanger; 4th generation DH systems

## Nomenclature

$c_p$	specific heat	kJ/kg K
$d_d$	droplet diameter	mm
$d_{noz}$	nozzle diameter	mm
$E_{cd}$	$(E_{cd} = N_{cd} \cdot 100 \% / N_b)$ efficiency indicator of the flue gas condenser	%
$f$	specific droplet surface in kg water	m <sup>2</sup> /kg
$F_d$	droplet surface in column m <sup>3</sup>	m <sup>2</sup> /m <sup>3</sup>
$g$	gravity acceleration	m/s <sup>2</sup>
$G$	mass flow rate	kg/s
$Gu$	Guhman criteria	
$h$	gas enthalpy	kJ/kg dry gas
$H_w$	water spray intensity	kg/m <sup>2</sup> s
$L_{dg}$	dry gas flow rate	kg dry gas/s
$N_b$	boiler capacity	kW

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$N_{cd}$	capacity of direct contact device	kW
$Nu$	Nusselt criteria	
$p$	wet gas pressure	Pa
$p_g$	gas partial pressure	Pa
$p_v$	vapour partial pressure	Pa
$p_{sat}$	water vapour saturation pressure	Pa
$P_{noz}$	water pressure before nozzle	MPa
$Pr$	Prandtl criteria	
$r$	water latent heat	kJ/kg
$Re$	Reynolds criteria	
$T$	temperature	°C
$\Delta T_{lm}$	logarithmic mean temperature difference	
$u_d$	droplet velocity	m/s
$u_r$	relative droplet velocity	m/s
$w$	gas absolute humidity	kg/kg dry gas
$V$	volume	
$V_a$	active volume	
$n$	number of sections	
$l_k$	height of the column	
<i>Greek symbols</i>		
$\alpha$	heat transfer coefficient	W/m <sup>2</sup> K
$\lambda$	thermal conductivity	W/m K
$\nu$	cinematic viscosity	m <sup>2</sup> /s
$\rho$	density	kg/m <sup>3</sup>
$\tau$	time	
<i>Subscripts</i>		
w	water	
v	vapour	
dg	dry gas	
wg	wet gas	
sat	saturation	
wet	wet bulb	
dew	dew point	
i-1	element input	
i	element output	
r	return	
s	supply	
g	gas	

## 1. INTRODUCTION

Creation of a 4th generation district heating system is simpler and easier to justify for those systems, which already possess reduced temperatures, or in which the operational efficiency of the source depends on the water temperature level at the heat network [1]. This pertains to systems with flue gas condensers installed after the boilers in particular. Energy is the main factor affecting national economies of countries, therefore it is in everyone's best interests to improve energy efficiency. One way to do that is by developing and implementing heat recovery from industrial processes, rather than creating waste heat in industry. There are different technologies available for heat recovery, for example, heat pumps, heat exchangers, boilers, etc. [2]. The choice of a certain technology is dependent on the heat source. In this paper, focus is placed on heat recovery from the power industry, more precisely – heating system, with a condenser.

Condensing unit is a heat exchanger used for energy recovery from boiler outlet flue gas. It reduces the temperature of water vapour below the dew point, recovering sensible and latent heat from the flue gas in the process. The use of condensing economizers also reduces the amount of particulate matter in the flue gas. These actions improve the overall efficiency of the boiler system [3]. The heat in the flue gas is in a range of 30 °C to 250 °C and it makes up about 15 %–40 % of the total heat content in the fuel used [4].

Flue gas contains acids and particulates which are mostly removed by the use of scrubbers and filters before reaching the condenser. Heat recovery that can be reached by a condenser is usually between 10 % and 15 %, in the case when natural gas is used as a fuel. In the case of biomass, these levels of heat recovery are similar. However in this case it is possible to recover energy from two sources – moisture evaporation and water from fuel components, therefore decreasing the necessity of fuel drying before combustion [5].

There are two types of condensers in use – direct and indirect contact economizers. The cooling matter used in condensers is either air or water. Indirect contact condensers are more common in real life applications, because they recover latent heat from flue gas by letting gas through condensing heat exchangers. These condensers can be divided in groups such as pipe, lamella and combi condensers [3], [6]. Direct contact condensers consist of two chambers, the first is meant for vapour conditioning, but the second is meant for flue gas cooling with counter – current sprays. They can be with or without fillings and are mostly applied to gas-fired energy sources, with new and developing applications to wood-fired and oil-fired boilers as well [5]. The main advantage of a condenser is the recovery of latent heat, which leads to thermal efficiency improvement and can exceed 100 %, when referencing it against the fuels' lowest heating value. Another advantage is an increase in environmental benefits, created by a decreased amount of flue gas emissions. This is due to condensate that creates a film, removing particle matter and dissolving acids [7].

One disadvantage to a condenser is that a larger amount of sensible heat is transferred to non-condensable gas, leading to smaller vapour amounts and a higher thermal conductivity coefficient ( $\lambda$ ), which affects the efficiency of the boiler. Another disadvantage of the technology is a high corrosion risk caused by combustion products. It can lead to the creation of cracks in exchangers, if the material is not strong enough or the point of damage has high mechanical stress [8]. This disadvantage can be reduced by detailed and precise installation and the selection of corrosion-resistant materials; however, this may lead to higher costs [9]. When using indirect condensers, there is a possibility of fouling in the tubes, therefore extra attention should be brought to the feed water characteristics [10].

One of the most important factors in the operation of heating systems is the return water temperature, which should be about 30 °C. It serves as a heat sink in the system. If the water temperature is below that which is desired, a heat pump has to be added to the system, increasing energy consumption and costs [11].

Condensers, their possible applications and types have been studied in various cases. A lot of research devoted to condensing heat exchangers, their application and possible implementation has been conducted at the Lehigh University in Bethlehem, USA. Some of the studied topics are described further. Jeong and Levy [12] have worked on the determination of acid condensation, to make it possible to create a suitable heat exchanger for use in predicted conditions. They used mass and energy balances of the system to study characteristics of flue gas and their condensation, leading to the conclusion, that modelling methodology can be applied to real life heat exchangers, because theoretically obtained data of acids and water condensation are in agreement with experimental data [12]. Samuelson [13] has extensively studied recovery of water from boiler flue gas, focusing on the condensing heat exchanger for use in the oil-fired boiler. The designed technology leads to 80 % water recovery and the same efficiency for acid condensation, which was conducted separately [13]. Hazell has modified an existing computer simulation code for heat exchanger to predict condensation rates, heat transfer and pressure changes. The costs of the heat exchanger were studied, which showed that the most significant costs for implementing a condensing heat exchanger is capital costs, while operating costs make up a small part of total costs. The heat exchanger was optimized by using corrosive resistant materials and calculations of tube spacing, water temperature effects on heat and mass transfer and implementation. It was concluded that the most cost effective solution for condensing heat exchanger placement is downstream of the flue gas desulfurization unit [14].

Hill [15] has studied heat recovery and its possible applications in real life systems in his master thesis, creating a heat recovery system design for power plants. The issues noted previously, were brought out in the paper, because they slow down wide implementation of heat recovery systems in power plants [15].

Li et al. [16] have studied vapour condensation theoretically in environments containing non-condensable gas. Mass and heat balance equations have been used to develop a mathematical model to predict results and make comparisons with existing experimental results. The developed model can be applied to co-current and counter flow condensers. Predicted results agree with the experimental data available. Osakabe has developed a heat exchanger for latent heat recovery from flue gas gained from oxygen – combustion boiler and added a design of result prediction method. The experimental results focused on water and gas temperatures, mass and heat transfer changes and agreed with prediction, making the method applicable to existing heat exchangers in oxygen-combustion systems [17], [18]. Sun et al. [19] studied condensation of flue gas on tubes in experimental conditions. Dew point defined separation in dry and condensation sections. It was concluded that the condensation pattern was close to dropwise condensation. A decrease in wall temperature lead to larger sections of film covered wall. Predictions agreed with experimental results.

To begin the progress towards the 4th generation district heating system, it is necessary not only to analyse the existing system, but also to understand what the main leading parameter is. The analysis of the operational data for the Ludza district heating system shows that energy efficiency of the system can be increased if the return water temperature is decreased. This can be explained by the fact that wet wood chips (moisture content  $W_d = 45\% - 55\%$ ) are burned in the boiler house [20].

Another option is to form a heat exchanger, into which flue gases transfer their heat to the spray water through direct contact, and later use this heat in the heating system. Both types of technologies have the same principle of operation – direct contact exchange of heat and mass between the heat carriers. In both cases, the processes are substantially affected by the spray water temperature, which, in turn, is determined by the network return temperature. Only the flue gas temperature differs, which in one case is fixed after the heater, but in the other case – at the furnace exit. Usually there are two types of direct contact heat exchanger – with or without filling. The condensing unit with filling is researched and inquired more widely [21]. It possesses higher energy efficiency and is smaller in size, however it has higher aerodynamic resistance and therefore also a higher level of energy consumption. A drawback of this heat exchanger is that, in case purified gas flows are used, the filling gets clogged up.

No-filling spray in technologies is widely used in the heat exchanger to form direct contact of liquid and gas. Water is sprayed into the columns, and it forms drops with a high interphase surface for heat and mass transfer to take place between the flows. The characteristic values of columns are hard to determine, as a complex hydrodynamic system is formed and thus the values are affected by several factors: size and division of drops, speed of drops, crashes and mergers of drops, circulation of drops within the condensing unit, deformation of drops, and flows of gases and liquids. In the research [22] influence on the drop size by the speed of gas and liquid, interphase surface size and aggregate ratio of gas and liquid mass transfer are analysed. Results can be used in case the heat carrier speed is small.

Although spray technologies are widely known, only a few methodologies are known for their calculations, which are related to one-dimensional modelling and empirical research [23]. The drawback of empirical research is the limited use of its results, as these are correct only for research of similar situations.

Most of the existing studies focus on improving efficiency of latent heat and water recovery from flue gas, without drastically changing existing technologies. Here, attention is brought to the use condenser without filling for latent heat recovery from flue gas. It is important to develop and publish information on heat recovery technologies to provide solutions for improvement of energy efficiency in different industries. As sustainable development is becoming more and more important in industrial zones, energy efficiency and waste energy recovery will receive more attention in the future.

In the paper, the methodology for the calculation of parameters of the heat and mass exchanger – condensing unit is created, equipment calculations are performed, and results are compared with the experimental research data of industrial flue gas-condensing units installed after boiler house technologies.

## **2. RESEARCH METHODOLOGY**

### **2.1. Characterization of Condensing Unit**

The research analysed direct contact heating equipment, which can be used for 4th generation DH systems with a decreased network water temperature graphic. The condensing unit is located directly after the furnace, at which stage the wood chips are burnt, and this heat exchanger substitutes the boiler. The scheme of the direct contact water heating equipment can be seen in Fig. 1.

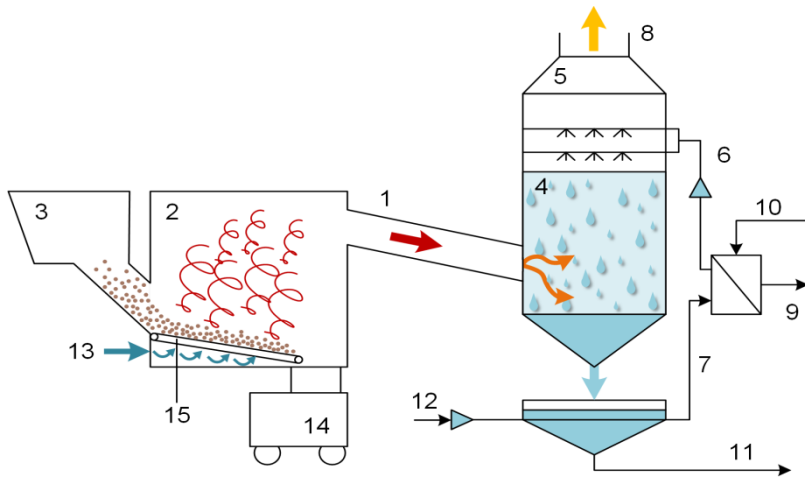


Fig. 1. Principal scheme of direct contact heating equipment.

1 – introduction of hot gases; 2 – furnace; 3 – wood chip storage; 4 – evaporation and condensing part; 5 – water drop separator; 6 – spray nozzles; 7 – DH heat exchanger; 8 – exit of flue gases; 9 – DH supply water; 10 – DH return water; 11 – pulp outlet; 12 – feeding water inlet; 13 – air supply; 14 – ash container; 15 – furnace grates.

Hot wet flue gases flow into the heat exchanger (1) directly from the furnace. The flue gas temperature can vary from 900 °C to 1000 °C, depending on the fuel quality. The absolute humidity of flue gases reaches  $w = 0.192$  kg/kg dry gas, and depends on the wood chip moisture content  $W_d$  (often 55 %), and oxygen content in the flue gases  $O_2 = 7$  %. From the wood chip storage (3) fuel is supplied to the furnace (2). Combustion process is implemented on furnace grates (15). The air required for combustion (13) is supplied under grates. Initial parameters of flue gases are determined with the help of calculations for wood fuel combustion processes. Flue gases are tangentially supplied to the condensing unit (4), thus furthering better mixing with the water drops, and resulting in more comprehensive gas treatment from solid particles. It is known that spinning of the flow intensifies the transfer of heat and mass [24]. The water is sprayed into flue gases through the nozzles located at the upper part of the heat exchanger (6). The mixture of flue gases, vapour and water drops form flows, after which condensing of vapour from flue gases and deep cooling of exhaust gases takes place. The water comes from the bottom part of the condensing unit, and is supplied to the spray nozzles through the heat exchanger, at which point water transfers its heat to the DH network water. In order to condense as much vapour as possible from flue gases' vapour, the water temperature should be as low as possible. The spray water temperature is determined by the heat transfer into the heat exchanger, which, in turn, depends on the DH return water temperature (10). The effectiveness of the condensing unit depends on the heat transferred to the DH network (9). The lower the return water temperature of the network, the higher the useful heat of the condensing unit. Condensate is formed and stored at the bottom part of the direct contact heat exchanger envelope, and exits from the condensing unit together with the collected solid particles (11) (with pulp). Condensate contains dissolved oxides from the flue gases, which form acid environment liquid phase, to neutralize which NaOH is introduced as well (12). Cooled and dried flue gases are connected to the stack with the help of a tube (8). At the upper part of condensing unit, the filling or drop separator (5) is located in order to catch water drops.

## 2.2. Process Calculations of Direct Contact Heat Exchanger

At direct contact condensing unit, heat carriers – wet flue gases and spray water – come into direct contact, and there is no fixed heat exchange surface between them. The heat and mass exchange surface is formed by the phase transfer surface of water drops and gases. The phase transfer surface varies, and, to a large extent, it is determined by the hydrodynamic regimes of both flows. The cooling process of wet flue gases is connected with the heat and mass transfer. Hot flue gases and water drops sprayed into the condensing unit form the counter flow.

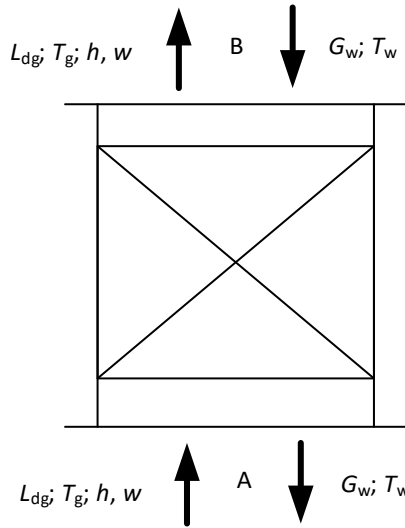


Fig. 2. Flow scheme and parameters of heat carriers in direct contact heat exchanger.

After the furnace, flue gases are supplied to the bottom part A of the direct contact heat exchanger, and water is sprayed into the upper part B. Counter flow of the heat carrier provides for a more comprehensive cooling of flue gases and increases the water temperature. The surface of heat exchange is formed either by the pellicle flowing downwards or by the spray water drops. In order to ensure the condensation of vapour, the spray water temperature should be lower than the temperature of the gas dew point  $T_w < T_{dew}$ . Moreover, the lower the water temperature, the more comprehensive the condensation of the water vapour. Owing to the low return water temperature (30 °C), at the 4th generation DH systems, it is possible to reach effective vapour condensing at the direct contact heat exchanger.

Heat and mass exchange processes do not take place directly between gases and water drops, but they take place at the bordering layers. When water vaporizes from the surface of a water drop or pellicle, a bordering layer of saturated vapour is formed [25]. In this layer, partial pressure of vapour corresponds to the saturation  $p_{sat}$  pressure, and the value of it is determined by the water temperature, which changes from  $T_{wB}$  at the spraying place to  $T_{wA}$  at the heat exchanger inlet. After the condensing unit, water temperature cannot be higher than the temperature of the wet thermometer  $T_{wet}$ . In case vapour condensing takes place at the whole heat exchanger, the water temperature is equal to the dew point temperature  $T_{wA} = T_{dew}$ .

The heat and mass exchange constitutes a complex process, at which exchange of sensible heat and mass takes place simultaneously. The propelling force of sensible heat exchange is the

difference in temperature of heat carriers – flue gases and spray water. As the flue gas temperature  $T_g$  is higher than the water temperature  $T_w$ , the energy is transferred from the flue gases to the water, as a result of which water heats up, but flue gases cool down. Simultaneously with the heat transfer, mass transfer also takes place, which is determined by two propelling forces. The difference between the partial pressure of water vapour at the saturated gas bordering layer  $p_{\text{sat}}$  and partial pressure of vapour at gases  $p_v$ , or the difference between humidity content at the saturation condition  $w_{\text{sat}}$  and humidity content of gases  $w$  can be used as propelling forces to characterize the mass transfer.

Alterations in parameters of heat and mass transfer processes can be shown graphically, while gases flow through the contact heat exchanger. In order to determine all the necessary parameters presented, the conditions at the heat exchanger exit and calculations of the processes taking place therein should be made. Knowing the gas temperature at the inlet  $T_{gA}$  and assuming the gas temperature at the exit  $T_{gB}$ , the scrubber is divided into elementary  $n$  sections  $(T_{gA}-T_{gB})/n$  with a determined gas temperature at the end of sections  $T_{g1}, T_{g2} \dots T_{gi}, T_{gi-1}$  and so on. The calculations scheme  $T$ - $w$  of elements  $i-1$  and  $i$  are reflected as a coordinate grid in Fig. 3.

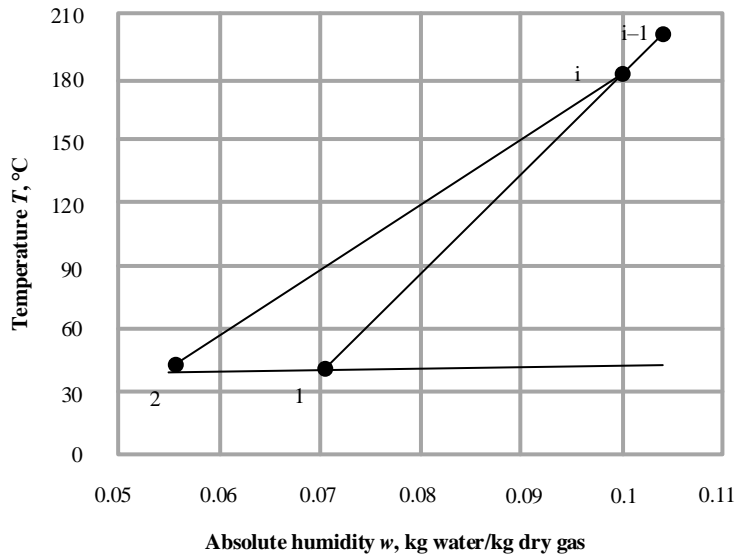


Fig. 3. Element of calculations scheme of direct contact heat exchanger.

At the horizontal axis, the values of the gas absolute humidity are indicated. At the vertical axis, values of the gas dry thermometer are shown. Points 1–2 are located in close proximity on the vapour saturation curve, at which the relative humidity is 100 %. Condition  $i-1$  characterizes gas parameters selected at the inlet of the direct contact heat exchanger. After inlet into the heat exchanger, wet gases come into contact not with the water mass, but with the saturated vapour layer, which is located above the water surface (conditions 1 and 2 in Fig. 3). This means that the contact takes place between two gases possessing different parameters. The results of the mixing process of two gases possessing different parameters  $T$ - $w$  at the diagram are reflected on the line, which connects points characterizing gas conditions.

At the beginning of tests, the following is measured: pressure at the direct contact heat exchanger  $p$ ; gas temperature at the inlet  $T_{gA}$  and exit  $T_B$ ; absolute humidity of gases  $w_A$ ; mass



flow of dry gases  $L_{dg}$ ; specific heat of dry gases  $c_{pdg}$ ; specific heat of water vapour  $c_{pv}$ ; vaporization heat  $r$ ; flow of spray water  $G_w$ ; temperature  $T_{wB}$  and  $T_{wA}$  of spray water.

Partial pressure of spray water vapour saturation is calculated by using the correlation obtained from the approximation of table data of the saturated vapour:

$$p_{sat} = 612,51748842 + 43,675694339 \cdot T_W + 1,484260898 \cdot T_W^2 + 0,0252485257 \cdot T_W^3 + 0,000292033 \cdot T_W^4 + 0,0000027053 \cdot T_W^5. \quad (1)$$

By using this correlation (1), absolute humidity of the saturated vapour is calculated as follows:

$$w_{sat} = \frac{0,6221 \cdot p_{sat}}{p - p_{sat}}. \quad (2)$$

Partial pressure of water vapour into gases is calculated by using the following formula:

$$p_v = \frac{p \cdot w}{0,6221 - w}. \quad (3)$$

The humidity content of gas flows at the exit of heat exchanger element  $i$  and inlet of element  $i-1$  are:

$$w_i = w_{i-1} - \Delta w_i. \quad (4)$$

Humidity content changes at element  $i$ :

$$\Delta w_i = \frac{w_{i-1} - w_{sat,i-1}}{T_{gi-1} - T_{g,sat,i-1}} \cdot \Delta T_g. \quad (5)$$

Flue gas enthalpy of process at element  $i$  is calculated by using the following correlation:

$$h_i = c_{pgi} \cdot T_{gi} + w_i \cdot (r + c_{pv} \cdot T_{gi}). \quad (6)$$

As vapour condensing takes place, the amount of spray water increases. Water amount at the end of section  $i$  can be determined as:

$$G_{wi} = G_{wi-1} - L_{dg} \cdot (w_{i-1} - w_i). \quad (7)$$

Changes of water temperature at the elementary sections of scrubber can be determined through the heat and mass balance equation. The heat balance equation of an elementary section connects the heat conducted and exited at this particular section. For this calculation, parameters of heat carriers at the section inlet and exit should be determined, as well as the direction of heat carriers should be observed. The flows and parameters of heat carriers of direct contact heat exchanger are shown in Fig. 2.

The heat balance equation for the elementary section  $i$  connects the heat conducted in and out during this section:

$$Q_{gi-1} + Q_{wi} = Q_{gi} + Q_{wi-1}. \quad (8)$$

After determining values of the Eq. (8), the heat balance of the elementary section can be expressed as the following equation:

$$L_{dg} \cdot (c_{pdg} \cdot T_{gi-1} + w_{i-1} \cdot (r + c_{pv} \cdot T_{gi-1})) + (G_{wi-1} - L_{dg} \cdot (w_{i-1} - w_i)) \cdot c_{pw} \cdot T_{wi} = L_{dg} \cdot (c_{pdg} \cdot T_{gi} + w_i \cdot (r + c_{pv} \cdot T_{gi})) + G_{wi-1} \cdot c_{pw} \cdot T_{wi-1}. \quad (9)$$

By using gas enthalpy Eq. (6), the balance can be expressed as follows:

$$L_{dg} \cdot h_{i-1} + (G_{wi-1} - L_{dg} \cdot (w_{i-1} - w_i)) \cdot c_{pw} \cdot T_{wi} = L_{dg} \cdot h_i + G_{wi-1} \cdot c_{pw} \cdot T_{wi-1}. \quad (10)$$

The calculation expression for determination of water temperature at the exit of elementary section is obtained from the heat balance Eq. (10):

$$T_{wi} = \frac{G_{wi-1} \cdot c_{pw} \cdot T_{wi-1} - L_{dg} \cdot (h_{i-1} - h_i)}{(G_{wi-1} - L_{dg} \cdot (w_{i-1} - w_i)) \cdot c_{pw}}. \quad (11)$$

Software is being developed in order to prepare a detailed process analysis based on the descriptive equations of the heat and mass process (1...7), (11), as well as the correlation for wet gas parameter calculations. At the end of the calculations, the spray water temperature is determined  $T_{wB}$  and compared with the value assumed at the beginning. In case the difference is 2 °C or more, the calculation is repeated by changing the flow of the spray water  $G_{wA}$ . The iteration method, justified assumptions and simplifications are used for calculations [26]. The processes at the heat exchanger can be reflected graphically by using the calculated inter-condition data. The process can be seen in Fig. 4.

The heat capacity of the condensing unit can be determined from the heat balance Eq. (10) by applying it to the whole direct contact heat exchanger (Fig. 2) and changing it. This can be determined by examining the heat released by wet gases or heat received by water. The two following correlations can be used accordingly:

$$N_{cd} = L_{dg} \cdot (h_A - h_B), \quad (12)$$

$$N_{cd} = G_{wA} \cdot c_{pw} \cdot (T_{wA} - T_{wB}) + L_{dg} \cdot (w_A - w_B) \cdot c_{pw} \cdot T_{wB}. \quad (13)$$

The peculiarity of a direct contact heat exchanger is that in case the spray water temperature is higher than the dew point temperature  $T_{wB} > T_{dew}$  or end temperature of the spray water heating is close to the temperature of the wet thermometer  $T_{dew} < T_{wB} \leq T_{wet}$ , the water vaporization will take place at the inlet part of the heat exchanger. Vaporization heat forms the heat of the process, which is not desired. The heat of the process is later recovered during the process of condensing, and this does not change heat capacity of the condensing unit; however, a larger volume and size of heat exchanger are required for this. It is important, whether the processes at the direct contact condensing unit take place with the humidity condensing, starting with the introduction of wet gases, or with the vaporization and condensing following it.

### 2.3. Exchange of Heat and Mass at Direct Contact Heat Exchanger

After the heat exchanger process calculations are completed, one can move forward to determining the geometrical dimensions of the heat exchanger. Several methodologies for calculations are known. One of them relates to the use of a complex heat and mass exchange ratio per column volume unit. Then, knowing the heat capacity of the heat exchanger, the necessary volume and height of the heat exchanger can be determined. The height should be such that the necessary temperature is reached between the spray water and introduced gases. In the direct contact heat exchanger, both spraying columns with empty co-flows and counter-flows, and columns with partitions for gas co-flows and counter-flows, and columns with a freely poured filling or orderly placed filling are used. Differently from columns with filling, the empty columns do not possess defined phase transfer surfaces. This encumbers calculations, and forces one to look for other solutions. One such solution is the calculation of energy transfer units and determination of unit height [27]. The basis of another method is formed by an empirically determined aggregate ratio of the heat and mass exchange per column volume unit [28]. As the number of affecting factors is large, the ratios are not always determined in the circumstances, which correspond to the analysed situation, and therefore their calculated values are approximate. This method can be used in situations, which are similar to that which was researched.

In the article, the approximate calculation method is used for the no-filling direct contact heat exchanger. The aggregate heat transfer ratio between wet gases and water drops for the heat carriers can be determined by using the following empiric formula [29]:

$$Nu = \frac{\alpha \cdot d_d}{\lambda_{wg}} = 2 + 1,107 \cdot (Re)^{0,48} \cdot (Pr)^{0,23} \cdot (Gu)^{0,175}. \quad (14)$$

In the formula, the Guhman criteria  $Gu$  considers impact of the mass exchange on the heat exchange.

The diameter of the spray water drop is affected by the type, and diameter of a nozzle, and spray pressure prior to the nozzle. The following empiric correlation can be used to calculate the drop diameter for the mechanic centrifugal nozzles [30]:

$$d_d = 0,154 \cdot P_{noz}^{-0,44} d_{noz}^{0,23}. \quad (15)$$

In case it is assumed that drops are spherical, their area within 1 kg of water is

$$f' = n \cdot \pi \cdot d_d^2, \quad (16)$$

and the volume used by them is:

$$V = \frac{n \cdot \pi \cdot d_d^3}{6}. \quad (17)$$

The specific area of drops within 1 kg of water is obtained by dividing Eq. (17) with Eq. (18):

$$f = \frac{f'}{V} = \frac{6}{d_d}. \quad (18)$$

The Reynolds criteria is determined by the relation of water drop speed and gas speed [31]:

$$u_r = u_d + u_{wg}. \quad (19)$$

In order to determine the water drop speed at the column, forces should be analysed which affect the drop: gravity force, Archimedes force, which is contrary to the first one, and frontal resistance force of the drop, which is also contrary to the first one [32]. In the case of a laminar drop movement, the Stocks formula can be used to determine its frontal resistance force, and the speed  $u_d$  is calculated as:

$$u_d = \frac{d_d^2(\rho_w - \rho_{wg}) \cdot g}{18 \cdot \nu_{wg} \cdot \rho_{wg}}. \quad (20)$$

The time needed for a drop to move for 1 meter at the column is:

$$\tau = \frac{1}{u_d}. \quad (21)$$

Drop surface in  $1 \text{ m}^3$  of a column is:

$$F_d = f \cdot \tau \cdot H_w. \quad (22)$$

By determining the heat exchange ratio, the active volume of the column can be calculated from the Eq. (14):

$$V_a = \frac{N_{cd}}{\alpha \cdot \Delta T_{lm} \cdot F_d}. \quad (23)$$

In order to determine the active height of column, the following correlation can be used:

$$l_k = \frac{V_a \cdot H_w}{G_w}. \quad (24)$$

### 3. RESEARCH RESULTS AND THEIR ANALYSIS

This section shows the modelling results by using the previously described model. Modelling is performed for the direct contact heat exchanger, into which flue gases from wood chip burning are introduced with the humidity content  $w = 0,192 \text{ kg/kg}$  dry gas and temperature  $T_{gA} = 1000 \text{ }^\circ\text{C}$ . The following parameters of heat carriers were used:  $G_{wA} = 10 \text{ kg/s}$ ,  $L_{dg} = 0,675 \text{ kg/s}$ .

Alterations into the heat exchanger caused by the process parameters obtained during the modelling are indicated in Fig. 4.

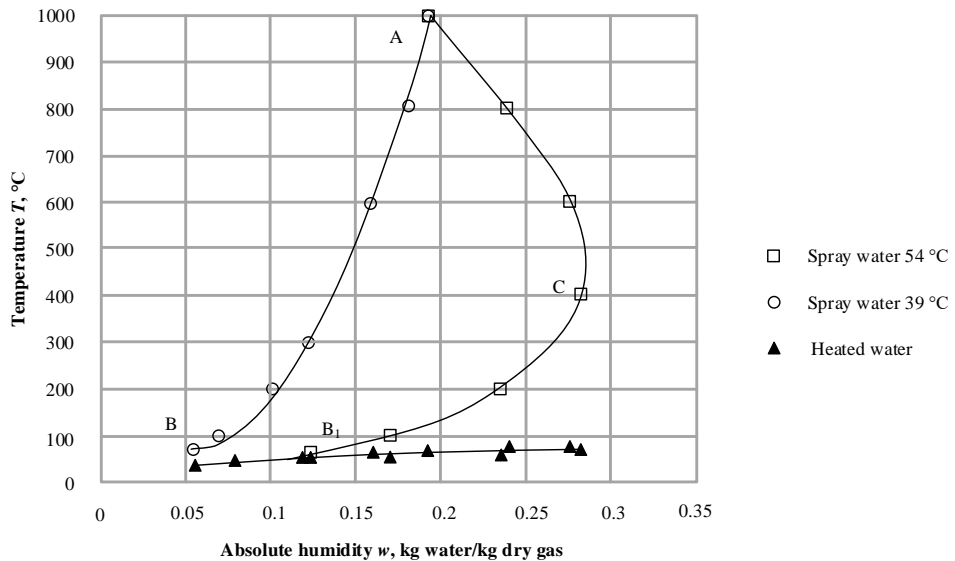


Fig. 4. Alterations of gases, water temperature and humidity content in direct contact heat exchanger.

In Fig. 4, two processes are compared, which differ in terms of spray water temperature. In both cases, parameters of the introduced flue gases are equal and correspond to the condition (A). End-conditions of processes are (B) and (B1). At the process described by the curve (A-B) the temperature of spray water is 39 °C, and it is observed that the humidity content of gas end-condition (B) is lower than at the condition (B1). This indicates that by decreasing the spray water temperature, a more comprehensive humidity condensing of flue gases takes place. In process A-B, condensing takes place at the whole heat exchanger, because the spray water temperature is lower than the dew point temperature, and spray water is heated up to the dew point temperature  $T_{\text{dew}} = 64$  °C. By increasing the spray water temperature to 54 °C, which corresponds to the network return temperature 45 °C, the process in the heat exchanger is described by the curve A-C-B1. It can be seen that at the starting section of the heat exchanger, the gas humidity content rises, which proves vaporization process at this section. In case of vaporization, the gas humidity content grows, and water can be heated up to a temperature, which is higher than the dew point temperature, but lower than the temperature of wet thermometer  $T_{\text{dew}} < T_w < T_{\text{wet}}$ . At condition A, the temperature of wet thermometer is 83 °C. The water temperature determined by the modelling is  $T_w = 73$  °C. Following vaporization, humidity condensing commences in condition C, into which the humidity content is  $w = 0.28$  kg water/kg dry gas, gas temperature  $T_g \approx 400$  °C and dew point temperature  $T_{\text{dew}} = 70$  °C. In condition C, the heat exchanger reaches the secondary dew point, at which vapour vaporization commences. The vaporization process at the heat exchanger inlet A-C and the following vapour condensing C-D decreases the capacity of heat exchanger, but increases the temperature of the to-be-heated water.

At the direct contact condensing unit, exchange of sensible and latent heat takes place, and each of these processes has its own propelling forces. Sensible heat exchange takes place due to the difference between the gas and water temperatures, and this value affects the intensity of the process. It can be seen that the largest temperature difference is at the flue gas entry of heat exchanger, and at this section the sensible heat transfer is more intense. These observations coincide with the research results [33]. In case the difference in heat carrier temperatures

decreases, the sensible heat transfer also decreases, and the lowest value of it is at the place for water spraying.

The propelling force of the latent heat transfer is the difference of partial force of water vapour, which is formed by the vapour partial pressure at the saturated gas bordering-layer surrounding the drop, and the vapour partial pressure at gases. Difference alterations into the partial pressure obtained at the heat exchanger as a result of modelling in the case of different spray temperatures are shown in Fig. 5.

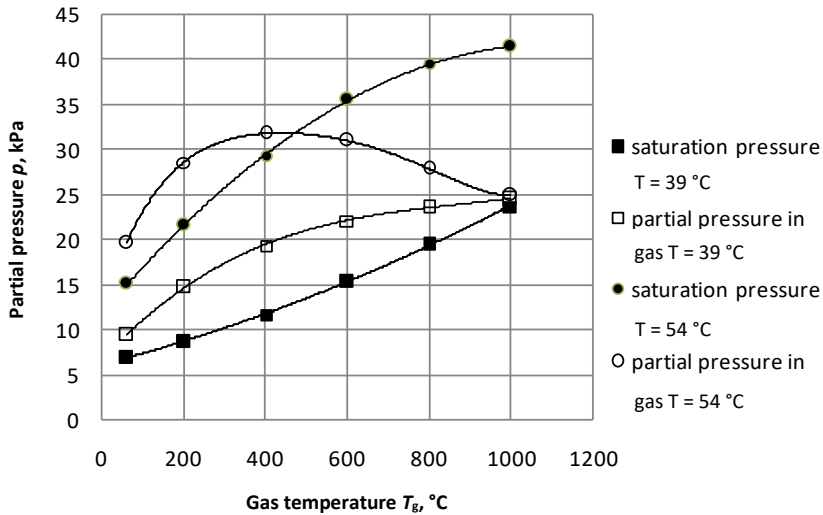


Fig. 5. Alterations of partial pressure in the heat exchanger.

As reflected in the results, when the spray water temperature is 39 °C, the condensing of water vapour from wet flue gases takes place at all heat exchangers; however, as shown by alterations in the difference between partial pressures, the intensity of the process is different. At the gas introduction place, the condensing is weak; afterwards its intensity increases up to the particular flue gas temperature, after which it decreases again. Flue gas temperatures, at which the most intensive water vapour condensing takes place, at a range within the limits of 400 °C–600 °C. In the case when the spray water temperature is 54 °C, the partial pressure of saturated water vapour at the inlet of heating equipment is higher than the vapour partial pressure in gases, and mass transfer takes place from the spray water to gas. When reaching gas temperature  $T_g \approx 450$  °C, the relation of partial pressures changes and mass transfer takes place from gas to spray water.

Relative capacity changes of water heat exchanger determined with the help of modelling can be seen in Fig. 6.

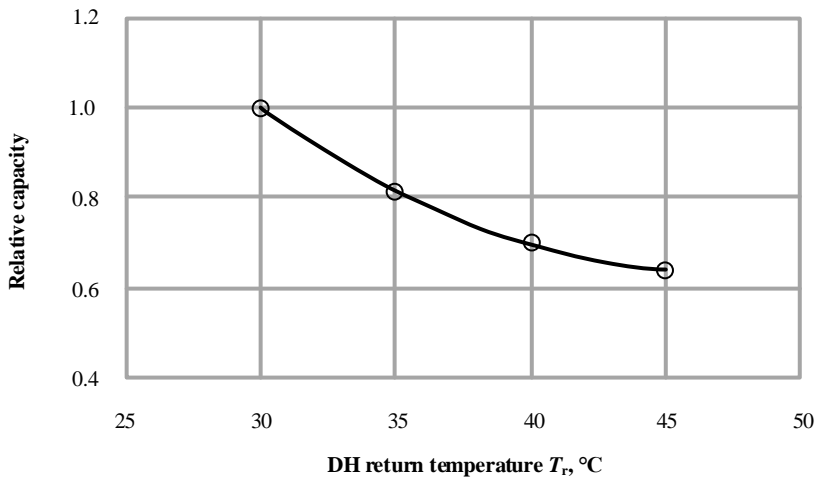


Fig. 6. Capacity decrease of direct contact heat exchanger depending on return temperature increase.

Analysis shows, that absolute values of partial pressure differences of mass transfer propelling forces vary depending on different spray water temperatures. By decreasing the spray water temperature, a more intensive vapour condensing at the heat exchanger is provided for, and as can be seen in Fig. 4, a lower humidity end-content of flue gases can also be achieved. This means that larger phase transfer heat is conducted to the water at the heat exchanger, and the heat capacity of the heating equipment increases. Analysis of gas and water temperature alterations shows that the average logarithmic temperature is higher in the case of the lowest spray water temperature, which are 240 °C and 185 °C, respectively. This means that the sensible heat exchange is also more intense in the case of a lower spray water temperature.

Experimental research on the flue gas condenser-unit of the direct contact heat exchanger is performed at the boiler house of the Ludza city heating system [34]. The modelled alterations in the capacity of water heat exchanger and gas condenser unit, depending on the network return temperature, cannot be numerically compared, as gas introduction temperatures, gas flow, spray water amount and other parameters are different because of the fact, that the condenser is installed after the 8 MW wood chip boiler. Nonetheless, the tendency of alterations can be compared, and this coincides for the heat exchanger with a similar operation principle – the condensing unit capacity decreases, if return temperature increases.

Changes in the condenser efficiency indicator depending on the network return temperature are shown in Fig. 7.

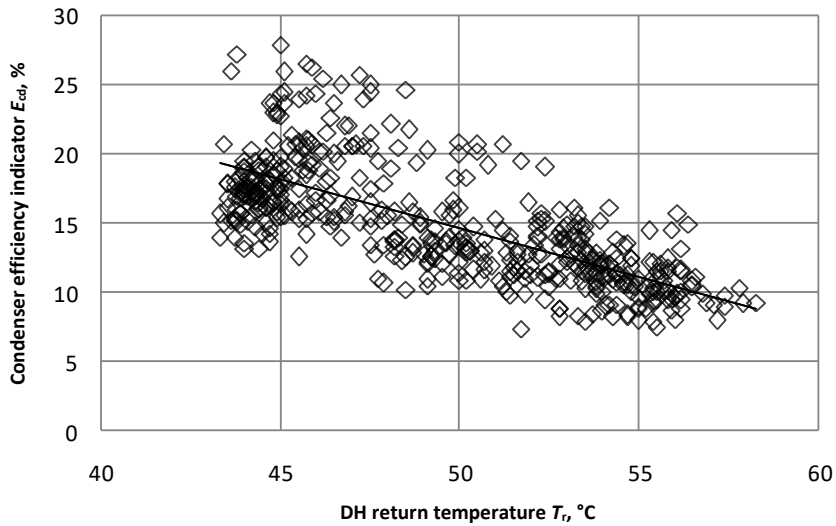


Fig. 7. Changes in condenser efficiency indicator depending on DH network return temperature.

A correlation has been found between the effectiveness of the condenser and the return water temperature:

$$E_{cd} = 49,698 - 0,7022 \cdot T_r. \quad (25)$$

It can be observed that by lowering the return water temperature by 1 °C, the value of the condenser efficiency indicator increases by 0.7 %. The DH network return temperature is a variable and it is influenced by the outdoor temperature, system heat load, mode of its adjusting, as well as the operation of consumer heating units and technical condition of the internal systems. The effect of various indicators explains the data dispersion seen in Fig. 7.

Cooling of the spray water is done in the network heat exchanger with the help of return water. In this way, network water performs the function of a cooler, and the efficiency of the cooler is determined by the return water temperature. This means that the first condition is fulfilled on the way to creating a 4th generation DH system, and the added value is obtained for an engineering and technical solution for transition to a low temperature DH system.

## 4. CONCLUSION

In order to carry out a detailed process analysis based on the descriptive equations of the heat and mass process, as well as the correlation for wet gas parameters calculations, software is being developed. Modelling results show that the capacity of a direct contact heat exchanger decreases, when the return temperature of the DH system increases. This trend is confirmed by experimental research performed on the gas condenser unit installed after the boiler at the boiler house of Ludza city.

Analysis of the results shows that absolute values of differences in partial pressures of mass transfer propelling force differ at different spray water temperatures. By decreasing spray water temperature, a more intense vapour condensing and lower humidity end-content in flue gases are



achieved in the heat exchanger. This means that larger phase transfer heat takes place at the heat exchanger. Analysis of gas and water temperature changes shows that the average logarithmic temperature is higher in case of the lowest spray water temperature, and exchange of sensible heat is more intense in case of a lower spray water temperature. Both factors facilitate an increase in the heat exchanger's capacity.

Positive impact of decreased DH network return temperature on the direct contact heat exchanger shows that introduction of the 4th generation of DH systems will increase the energy efficiency of the heat exchanger. The methodology suggested in the article can be used to assess it in each particular situation.

## REFERENCES

- [1] Gadd H., Werner S. Achieving low return temperatures from district heating substations. *Applied Energy* 2014;136:59–67. [doi:10.1016/j.apenergy.2014.09.022](https://doi.org/10.1016/j.apenergy.2014.09.022)
- [2] Huang F., Zheng J., Baleynaud J. M., Lu J. Heat recovery potentials and technologies in industrial zones. *J. Energy Inst.* 2016;In Press. [doi:10.1016/j.joei.2016.07.012](https://doi.org/10.1016/j.joei.2016.07.012)
- [3] Butcher T., Litzke W. Condensing economizers for small coal-fired boilers and furnaces. Project Report, BNL Informal, 1994.
- [4] Comakli K. Economic and environmental comparison of natural gas fired conventional and condensing combi boilers. *J. Energy Inst.* 2008;81(4):242–246. [doi: 10.1179/014426008x371031](https://doi.org/10.1179/014426008x371031)
- [5] Chen Q., Finney K., Li H., Zhang X., Zhou J., Sharifi V., Swithenbank J. Condensing boiler applications in the process industry. *Appl. Energy* 2012;89(1):30–36. [doi:10.1016/j.apenergy.2010.11.020](https://doi.org/10.1016/j.apenergy.2010.11.020)
- [6] Terhan M., Comakli K. Design and economic analysis of a flue gas condenser to recover latent heat from exhaust flue gas. *Appl. Therm. Eng.* 2016;100:1007–1015. [doi:10.1016/j.applthermaleng.2015.12.122](https://doi.org/10.1016/j.applthermaleng.2015.12.122)
- [7] Grohn A., Suonmaa V., Auvinen A., Lehtinen K. E. J., Jokiniemi J. Reduction of fine particle emissions from wood combustion with optimized condensing heat exchangers. *Environ. Sci. Technol.* 2009;43(16):6269–6274. [doi:10.1021/es8035225](https://doi.org/10.1021/es8035225)
- [8] Jeong K., Kessen M. J., Bilirgen H., Levy E. K. Analytical modeling of water condensation in condensing heat exchanger. *Int. J. Heat Mass Transf.* 2010;53(11–12):2361–2368. [doi:10.1016/j.ijheatmasstransfer.2010.02.004](https://doi.org/10.1016/j.ijheatmasstransfer.2010.02.004)
- [9] Street O. S. A Market Assessment for Condensing Boilers in Commercial Heating Applications. Boston: 2001.
- [10] TEMA. Standards of the Tubular Exchanger, 8th Edition, 1999.
- [11] Energy S. T. The 1st PRO-TEM Network Conference Sustainable Thermal Energy Management in Process Industry.
- [12] Jeong K., Levy E. K. Theoretical prediction of sulfuric acid condensation rates in boiler flue gas. *Int. J. Heat Mass Transf.* 2012;55(25–26):8010–8019. [doi:10.1016/j.ijheatmasstransfer.2012.08.037](https://doi.org/10.1016/j.ijheatmasstransfer.2012.08.037)
- [13] Levy E. Recovery of water from boiler flue gas, 2008.
- [14] Hazell D. Modeling and Optimization of Condensing Heat Exchangers for Cooling Boiler Flue Gas. ProQuest Dissertations Publishing, 2011.
- [15] Hill J. M. Study of low-grade waste heat recovery and energy transportations systems in industrial applications. ProQuest Dissertations Publishing, 2011.
- [16] Li J.-D., Sarairoh M., Thorpe G. Condensation of vapor in the presence of non-condensable gas in condensers. *Int. J. Heat Mass Transf.* 2011;54(17–18):4078–4089. [doi:10.1016/j.ijheatmasstransfer.2011.04.003](https://doi.org/10.1016/j.ijheatmasstransfer.2011.04.003)
- [17] Osakabe M. Latent heat recovery from actual flue gas, 2009.
- [18] Osakabe M. Latent heat recovery from Oxygen-combustion flue gas, 2000. [doi:10.1109/IECEC.2000.870877](https://doi.org/10.1109/IECEC.2000.870877)
- [19] Sun X., Kotake S., Suzuki Y., Senoo M. Condensation heat transfer on tubes in actual flue gas. *Heat Transfer – Asian Research* 2001;30(2):139–151. [doi:10.1002/1523-1496\(200103\)30:2<139::AID-HTJ5>3.0.CO;2-0](https://doi.org/10.1002/1523-1496(200103)30:2<139::AID-HTJ5>3.0.CO;2-0)
- [20] Vigants E., Blumberga D., Veidenbergs I., Vigants G., Rochas C. Experimental research of flue gas condensing unit. *Advances in Environment, Biotechnology and Biomedicine* 2012:179–189.
- [21] Li Y., Klausner J. F., Mei R., Knight J. Direct contact condensation in packed beds. *International Journal of Heat and Mass Transfer* 2006;49:4751–4761. [doi:10.1016/j.ijheatmasstransfer.2006.06.013](https://doi.org/10.1016/j.ijheatmasstransfer.2006.06.013)
- [22] Turpin A., Couvert A., Laplanche A., Paillier A. Mass transfer and deodorization efficiency in a countercurrent spray tower for low superficial gas velocities. *The Canadian Journal of Chemical Engineering* 2009;87(1):53–59. [doi:10.1002/cjce.20084](https://doi.org/10.1002/cjce.20084)
- [23] Codolo M. C., Bizzo W. A. Experimental study of the SO<sub>2</sub> removal efficiency and volumetric mass transfer coefficients in a pilot-scale multi-nozzle spray tower. *International Journal of Heat and Mass Transfer* 2013;66:80–89. [doi:10.1016/j.ijheatmasstransfer.2013.07.011](https://doi.org/10.1016/j.ijheatmasstransfer.2013.07.011)
- [24] Javed K. H., Mahmud T., Purba E. Enhancement of mass transfer in a spray tower using swirling gas flow. *Chemical Engineering Research and Design* 2006;84(A6):465–477. [doi:10.1205/cherd.05119](https://doi.org/10.1205/cherd.05119)

- [25] Boulama K., Galanis N., Orfi J. Heat and mass transfer between gas and liquid streams in direct contact. *International Journal of Heat and Mass Transfer* 2004;47:3669–3681. [doi:10.1016/j.ijheatmasstransfer.2004.04.004](https://doi.org/10.1016/j.ijheatmasstransfer.2004.04.004)
- [26] Jeong K., Kessen M. J., Bilirgen H., Levy E. K. Analytical modeling of water condensation in condensing heat exchanger. *International Journal of Heat and Mass Transfer* 2010;53:2361–2368. [doi:10.1016/j.ijheatmasstransfer.2010.02.004](https://doi.org/10.1016/j.ijheatmasstransfer.2010.02.004)
- [27] Fair J. R., Bravo J. L. Energy recovery by direct contact gas-liquid heat exchange. Proceedings from the Tenth Annual Industrial Energy Technology Conference, Houston, TX, September 13–15, 1988.
- [28] Rusanov A. A. Hand-Book on Dust and Ash Capture. Moscow: Energia, 1975.
- [29] Nesterenko A. V. Bases of thermodynamic calculations for ventilating and air conditioning (in Russian). Moscow: Vysshaja Shkola, 1971.
- [30] Galustov V. S. The direct – flow spray devices in heat – power engineering (in Russian). Moscow: Energoatomizdat, 1989.
- [31] Shilyaev M. I., Khromova E. M. Modelling of heat and mass transfer and absorption-condensation dust and gas cleaning in jet scrubbers. *Mass Transfer - Advances in Sustainable Energy and Environment Oriented Numerical Modelling* 2013:163–194. [doi:10.5772/53094](https://doi.org/10.5772/53094)
- [32] Rahimi A., Bakhshi A. A simple one-dimensional model for investigation of heat and mass transfer effects on removal efficiency of particulate matters in a venturi scrubber. *Iranian Journal of Chemical Engineering* 2009;6(4):1–14.
- [33] Mizonov V., Yelin N., Yakimych P. A cell model to describe and optimize heat and mass transfer in contact heat exchangers. *Energy and Power Engineering* 2011;3:144–149. [doi:10.4236/epe.2011.32018](https://doi.org/10.4236/epe.2011.32018)
- [34] Vigants G., Galindoms G., Veidenbergs I., Vigants E., Blumberga D. Efficiency diagram for district heating system with gas condensing unit. *Energy Procedia* 2015;72:119–126. [doi:10.1016/j.egypro.2015.06.017](https://doi.org/10.1016/j.egypro.2015.06.017)



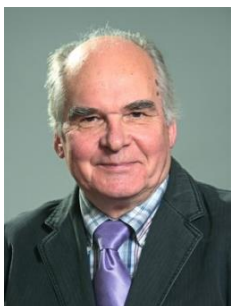
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