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THE CONDENSATION OF WATER VAPOUR IN A MIXTURE CONTAINING A HIGH CONCENTRATION OF NON-CONDENSABLE GAS IN A VERTICAL TUBE

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Abstract. This paper deals with the condensation of water vapour possessing a content of non-condensable gas in vertical tubes. The condensation of pure steam on a vertical surface is introduced by the Nusselt condensation model. However, the condensation of water vapour in a mixture with non-condensable gas differs from pure vapour condensation and is a much more complex process. The differences for the condensation of water vapour in a mixture containing a high concentration were theoretically analysed and evaluated. In order to investigate these effects, an experimental stand was built. Experiments were carried out in regards to the case of pure steam condensation and the condensation of water vapour with a non-condensable gas mixture to evaluate the influence of the variable non-condensable gas content during the process. A non-condensable gas in a mixture with steam decreases the intensity of the condensation and the condensation heat transfer coefficient. A gradual reduction of the volume and partial pressure of steam in the mixture causes a decrease in the condensation temperature of steam, and the temperature difference between steam and cooling water. The increasing non-condensable gas concentration restrains the transportation of steam towards the tube wall and this has a significant effect on the decrease in the condensation rate.

Keywords: Condensation, non-condensable gas, vertical tube condenser.

1. Introduction

In many cases in industry, water vapour is not present as a pure separate substance, but it may mix or become polluted by other gases due to infiltration, chemical reaction or the presence of other impurities. These mixed gases influence the condensation process of water vapour and have to be taken into account in the design of condensing heat exchangers. Depending on the concentration of non-condensable gases, the total heat transferred in the exchanger is reduced [1]. The content of non-condensable gases in a mixture with water vapor may be high in some industrial applications (flue gas in energy applications, waste vapor from the process industry, etc.) thus it significantly affects the intensity of heat transfer.

For applications where water vapour condenses in a mixture with a non-condensable gas (NGC), there are two commonly used forms of condensation: direct contact condensation [2] and surface condensation [3]. In the case of the condensation of waste vapour with a presence of mechanical impurities, a surface condenser operates with the lower amount of the outgoing condensate which may be contaminated at the outlet of the condenser [4].

The basic configurations of surface condensers are horizontal tube condensers and vertical tube condensers [4]. Condensation on horizontal tube bundles of different configurations has a wide application in industry [5]. The heat transfer rate in the condensation processes is mainly affected by the external flow velocity and the presence of non-condensable impurities [6]. For the condensation of steam with non-condensable gases outside a horizontal tube, a decrease in the condensation heat transfer coefficient (HTC) begins even in low concentrations of NCG and the decrease significantly rises with concentrations of NCG [7]. For air-steam condensation on a vertical tube, a decrease in the HTC below the value of $1000\,\mathrm{W/m^2K}$ was observed in a concentration of more than 10 % of a NCG [8, 9]. For the condensation of water vapour in a mixture with a NCG, which may also contain small mechanical impurities, it is suitable to use condensers in a vertical tube-side configuration [4]. These solid particles, which can stick to the tube wall, are spontaneously carried away from the tubes by the condensate flowing out. This design of the condenser is very flexible and is suitable where a particularly low pressure drop is specified for the condensing fluid [4]. Condensation in vertical tubes in the presence of NCG flowing downward in such tubes was experimentally investigated in [10]. The condensate flows down the tubes in the form of an annular film of liquid, thereby maintaining a good contact with both the cooling surface and the remaining vapour [11]. The disadvantages are that the coolant, which is often more prone to fouling, is on the shell side, and the use of finned tubes is precluded.

A determination of the overall HTC, which is necessary for the design of the condenser's heat transfer area, is well described in the literature for the case of pure steam condensation on a vertical surface by the Nusselt condensation model [1, 3, 4]. However, the condensation of water vapour in a mixture with NCG differs from pure vapour condensation and is a much more complex process [4, 12]. The aim of this article is to provide a theoretical analysis of the modifications of condensation in the presence of NGC with a high concentration in a vertical tube and to carry out an experimental investigation into the effect of NCG on the condensation process.

2. Condensation in vertical tubes

The basic heat-transfer model for the surface condensation introduced by Nusselt describes how a pure saturated vapour condenses on a vertical wall, forming a thin film of condensate that flows downward due to gravity [4]. The operating conditions of real condensers may differ from the assumptions adopted in the basic Nusselt theory [13]. The following differences may occur in the condensation of the steam mixture with inert gas in a vertical tube. During a condensation inside vertical tubes, flowing steam works on the surface of the condensate film through shear force, the film flow accelerates and the condensation HTC increases slightly [1, 4]. On the bottom of the high vertical walls or long tubes, the thickness of the film grows and the laminar flow can change to turbulent flow increasing the HTC [1, 4]. Furthermore, a subcooling of the condensing mixture may occur due to a decrease in the partial pressure of steam and a decrease in the condensation temperature [14].

NCG restrains the diffusion of the steam molecules through the gas towards the vapour-liquid interface, and it results in a decrease in the partial pressure of the vapour and reduces the HTC. Since the total pressure of the mixture remains constant, the partial pressure of the inert gas increases with the decreasing partial water vapour pressure. Steam concentrations decrease along the length of the tube equally with corresponding steam partial pressures [4, 12].

The modifications of the Nusselt condensation model were theoretically analysed for the condensation of an air-steam mixture in a vertical tube (see Tab. 1). The flow of a low-pressure steam mixture with a low velocity is assumed. When steam condenses its partial pressure decreases and the NGC concentration increases. The velocity decreases as well as the volume flow of the mixture.

The presence of non-condensable gases profoundly affects the condensation process, providing a great reduction in the condensation HTC. Other effects have a lesser influence on the value of the condensation HTC and, conversely, they increase the HTC.

3. Experimental setup

A schematic diagram of the experimental apparatus is shown in Fig. 1. The condenser was designed as a vertical double pipe heat exchanger consisting of two concentric stainless tubes. The inner tube of the heat exchanger is 2 000 mm long with an inner diameter of 23.7 mm (d_i) and a wall thickness of 1.6 mm. The outer tube is 1 500 mm long with an inner diameter of 29.7 mm (D_i) and a wall thickness of 2 mm. The material of the tubes is stainless steel 1.4301 (AISI 304). The annulus made from concentric tubes is 1.6 mm wide. Stainless pins are used as spacers at three circumferential positions to keep the annulus concentric.

Steam from a steam generator with a regulated pressure close to 1 bar and a temperature from 100 to 130 °C is mixed with pressurised air. The steam temperature is controlled so that the mixture is in a saturated or slightly superheated state after mixing superheated steam with cold air before it enters the condenser. A mixture of water vapour and air enters the condenser at the top and is directed vertically downward through a calming section before flowing over the inner vertical tube. The cooling water flows upwards in the annulus. The heat exchanger is in a counter-current configuration. The condensate flowing out of the pipe is collected in a tank and its production is determined by weighing. The excessive steam-air mixture in the heat exchanger outlet is released to the ambient. The position of the temperature, pressure, weight and flow measurements are shown in Fig. 1.

The experimental loop may operate in two modes - condensation of pure steam or condensation of steam in a mixture with air. In the case of a pure steam condensation mode, the air compressor is disconnected.

4. Evaluation procedure

The calculation of the HTC is based on the heat balance of the condenser [2]. The total heat performance Q is given by the equation

$$Q = M_w \cdot c_w \cdot (T_{w,out} - T_{w,in})\,, \qquad (1)$$

where M_w is the cooling water flow, $c_(w)$ is the specific heat capacity of water, $T_{w,out}$ is the outlet cooling water temperature, and $T_{w,in}$ is the inlet cooling water temperature. The overall HTC U is given by the equation

$$U = \frac{Q}{A \cdot \Delta T_{log}}, \qquad (2)$$

where A is the heat transfer surface of a condenser tube and ΔT_{log} is the logarithmic mean temperature difference. The overall HTC for condensation in a vertical tube is determined by the heat transfer balance (see Eq. 3). The right-hand side consists of the term corresponds to the tube side condensation heat transfer coefficient h_{cNG} (condensation in the presence of NCG), the term corresponds to the tube wall conduction heat transfer coefficient, and the term corresponds to the outside tube heat transfer coefficient h_w (cooling water side). The value of U is related to

Effect	Occurrence condition	Operating condition	Influence	Influence on HTC	Reference
Non-condensable gas	concentration more than approx. 0.1 %	high concentration	**decrease**	up to several times	[15, 16]
Flow mode of the film	thick condensate film film Re > 30: waviness flow film Re > (400 - 1600): pure turbulent flow	small film thickness	increase	up to 25 % for waviness interval; more for turbulent	[12, 16]
Sub-cooling of condensing mixture	Well temperature is below saturation (change in released enthalpy at the wall)	temperature change due to concentration profile change	increase	few %	[12]
Shear stress of flow	vapour is flowing at a high velocity	low velocities	increase	negligible under 5 m/s	[13]

Table 1. The influence of operating conditions on the condensation process.

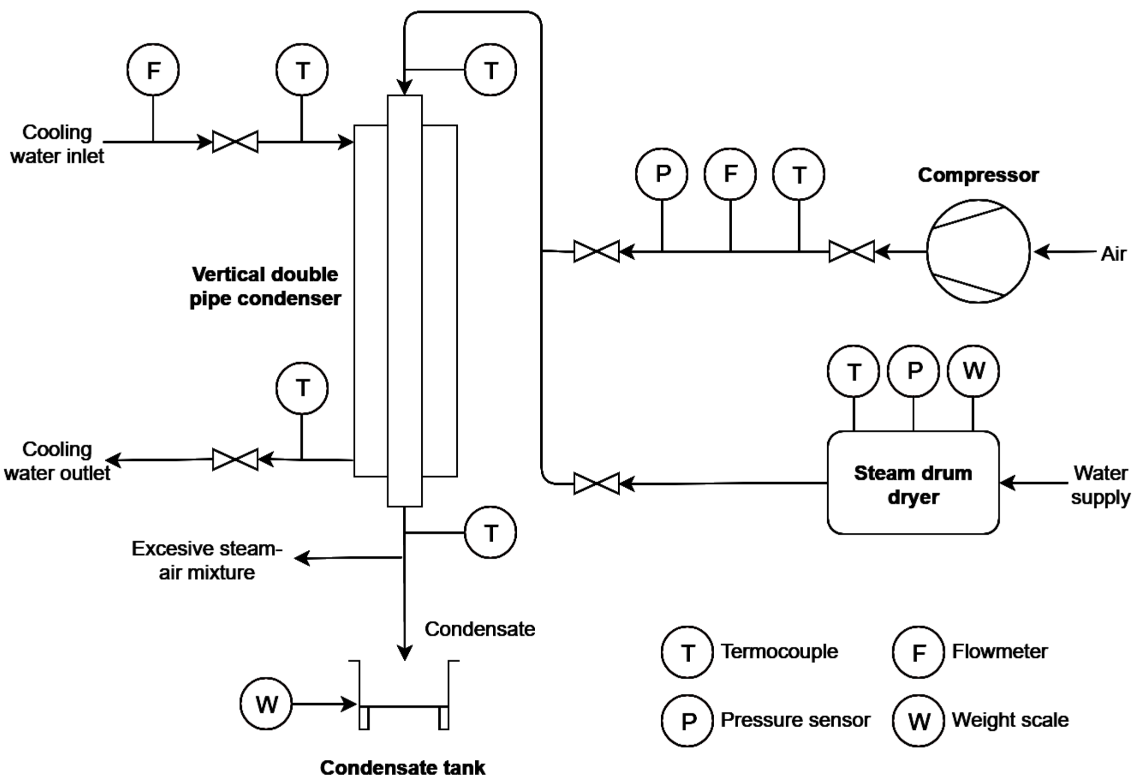

Figure 1. Experimental loop with a vertical tube condenser.

outer surface of the tube.

$$U = \frac{\frac{1}{d_o}}{\frac{1}{d_i \cdot h_{cNG}} + \frac{1}{2k} \ln\left(\frac{d_o}{d_i}\right) + \frac{1}{d_o \cdot h_w}}, \quad (3)$$

where d_i is the inside diameter of the tube, k is the thermal conductivity of the tube and d_o is the outside diameter of the tubes.

The evaluation procedure consists of two steps: Experimental determination of the HTC on the cooling water side and subsequently experimental determination of the HTC of the condensation of water vapour in a mixture with NCG.

4.1. Determining the cooling water HTC

From the point of view of reducing the inaccuracy of the determination of the HTC from the experiment, it is advisable to achieve low thermal resistance on the cooling water side, so that its effect on the overall HTC is as small as possible. Therefore, there is an effort to achieve high HTC on the cooling water in the annulus, which has been achieved by reducing the annulus spacing, and should increase the flow velocity as well as the convective HTC [3, 4]. Moreover, the heat transfer can be influenced by the small width of the annulus. The proposed condenser was designed with an annulus width of 1.4 mm. An annulus with such a restricted width approaches the scale of micro-channels where the heat transfer rapidly increases [17, 18]. It is generally considered that a micro-channel is any channel with a hydraulic diameter in the range of a micrometer, i.e. less than 1 mm. Thus, it can be assumed that the flow of cooling water in the annulus of the proposed condenser approaches this phenomenon and it can have an influence on the HTC of the cooling side of the condenser. Therefore, it is suitable to determine the HTC experimentally for the tested heat exchanger [19].

Experiments with pure steam condensation were performed to determine the cooling water HTC. In the case of pure steam condensation, it is possible to calculate the condensation HTC h_c according to the Nusselt model of pure steam condensation on the vertical wall [4, 12, 20].

$$h_c = 0.943 \left[\frac{g\rho_L(\rho_L - \rho_v)k_L^3 r'_{FG}}{\mu_L(T_{sat} - T_s)L}\right]^{1/4}, \quad (4)$$

where ρ_L is the density of the condensate, ρ_p is the density of water vapour, h'_{fg} is the latent heat of condensation, k_L is the thermal conductivity of the condensate, μ_L is the dynamic viscosity of the condensate, ΔT_{sat} is the difference between the saturation temperature and the wall temperature, and L is the wall length. According to [20], the inaccuracy given by using Eq. 4 is up to 3 % for $c_L \cdot (T_{sat} - T_s)/r'_{FG} \leq 0.1$ and $1 \leq \mathrm{Pr} \leq 100$.

Knowing the value of h_c, the cooling water HTC h_w in the annulus side for the operating range of the device (characterised by Reynolds number) was evaluated by the equation for determining the overall HTC as in the case of pure steam condensation

$$U = \frac{\frac{1}{d_o}}{\frac{1}{d_i \cdot h_c} + \frac{1}{2k} \ln\left(\frac{d_o}{d_i}\right) + \frac{1}{d_o \cdot h_w}}. \quad (5)$$

The HTC on the cooling water side is dependent on medium properties (resp. fluid velocity and temperature) [4, 20, 21]. Therefore, it is suitable to use the Nusselt number for evaluating h_w in various operating conditions. The Nusselt number is equal to the dimensionless temperature gradient at the surface and it essentially provides a measure of the convective heat transfer. The Nusselt number Nu may be viewed as the ratio of the conduction resistance of a material to the convection resistance of the same medium [21]

$$Nu = \frac{h_w \cdot D_e}{k}, \quad (6)$$

where k is the thermal conductivity, D_e is the characteristic diameter which is defined as

$$D_e = \frac{4 \cdot flow\ area}{wetted\ perimeter} = D_i - d_o, \text{ for annulus.} \quad (7)$$

In single phase fluid flow heat transfer, the Nusselt number for forced convection is generally in the form of

$$Nu = f(C, Re, \mathrm{Pr}), \quad (8)$$

where C is the term given by the geometry characteristic, the Reynolds number Re is defined as

$$Re = \frac{w_w \cdot D_e}{\mu_w}, \quad (9)$$

the Prandtl number Pr is defined as

$$\mathrm{Pr} = \frac{c_w \cdot \mu_w}{k_w}, \quad (10)$$

where, μ_w, k_w are the velocity, the dynamic viscosity resp. the thermal conductivity of cooling water.

The dependence of Nu on Re is influenced by the geometry and regime of flow, therefore it is generally difficult to describe. The term C can be considered as a constant for a specific type of tested heat exchanger. To take into account changes in the cooling water temperature, the correction for the Prandtl number has to be introduced assuming a raise in the value to the power of 0.33 in accordance with the standard practice for the forced convection [3, 4]. The heat transfer on the cooling water side is described by the dependence (see Fig. 2)

$$Y = Nu/Pr_{exp}^{0.33} = f(Re_{exp}) \quad (11)$$

For various conditions with Re in the analysed range and corresponding Pr, it is possible to determine h_w for the experimental condenser as

$$h_w = Y \cdot Pr^{0.33} \cdot \frac{k_w}{D_e} \quad (12)$$

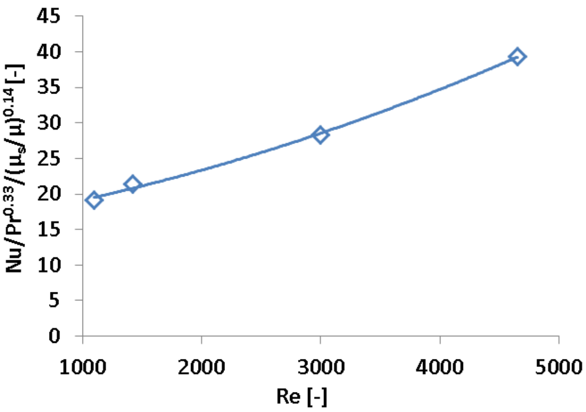

Figure 2. Heat transfer for the cooling water side.

4.2. Determining the condensation HTC

After the experimental determination of the cooling water HTC h_w, the value of condensation HTC h_{cNG} for various concentrations of air in a mixture with steam is derived from Eq. 3.

$$h_{cNG} = \frac{\frac{1}{d_i}}{\frac{1}{d_o \cdot U} - \frac{1}{2k}\ln\left(\frac{d_o}{d_i}\right) - \frac{1}{d_o \cdot h_w}}. \tag{13}$$

5. Results

5.1. The cooling water HTC

Experiments with pure steam condensation were carried out to evaluate the cooling water HTC. The experiments were made in the design operating range of cooling water flow in the condenser described by the Reynolds number from 1 000 to 5 000 and a cooling water temperature of around 20 °C within the atmospheric parameters of condensing steam. Based on the dependence in Fig. 2, it is possible to determine the value of the HTC when changing the conditions on the cooling water side.

5.2. The condensation HTC

After the evaluation of the cooling water HTC, experiments with various concentrations of air in a mixture with steam were carried out for the concentration range of 23 % to 69 % (see Tab. 2). Finally, the experimentally determined values of the condensation HTC for water vapour in a mixture with air were compared with the value for pure steam condensation calculated according to the Nusselt condensation model for corresponding operating parameters. Inlet mixture velocity is calculated based on the molar weight of air M_a, resp. vapour M_v as

$$u_g = \frac{M_a + M_v}{\rho_g \cdot \pi \frac{d_i^2}{4}} \tag{14}$$

with mixture density ρ_g defined as

$$\rho_g = \frac{M_a + M_v}{\frac{M_a}{\rho_a} + \frac{M_v}{\rho_v}} \tag{15}$$

where ρ_a is the air density and ρ_v is the vapour density.

The inlet and outlet temperatures of the mixture correspond to the partial pressure of steam in the mixture. In the measurements 1 and 2, the steam condensation rate at the inlet of the condenser was sufficiently high. This resulted in a higher temperate drop in the second part of the condenser. Due to low steam mass fraction at the end of the condenser, condensed steam caused a higher temperature drop in the mixture as compared to measurements 3 to 6. Therefore the outlet temperatures in the measurements 1 and 2 are lower.

When compared, the thermal resistances in Eq. 13, the value of thermal resistance of the condensation term $1/h_{cNG}$ is significantly higher than the value of thermal resistance of the cooling water term $1/h_w$. Thus the sensitivity of the condensation HTC value on the overall HTC is more significant in comparison with the cooling HTC. A deviation of 20 % in the determination of the h_w value (part 4.1) corresponds with a change up to 1 % in the resulting h_{cNG} value, resp. 2.5 % for a deviation of 50 %. Therefore, it can be said that a deviation in the determination of the value of h_w has a minimal effect on the result of the h_{cNG} values.

5.3. The effect of the operation conditions on the condensation process

The shear stress of flow

The shear stress caused by the flowing gas-vapour mixture depends on kinetic energy and the flow direction of the mixture. The mixture flow accelerates the condensate film since it flows in the same direction as the condensate flow. This causes a reduction in the thickness of the film which improves the heat transfer rate.

In all cases, the inlet velocity of the mixture during the experiments was less than 5.2 m/s. According to the calculation based on [21], the calculated improvement of the HTC was lower than 2 % for the all measurements. This is in a good agreement with the equation introduced in [22], where the theoretical and experimental analysis of the local HTC during the condensation of water vapour in the presence of a NCG in a vertical tube condenser was conducted. This study focused on the effect of the shear stress and created a new empirical factor which incorporates the influence of the shear stress on the heat transfer. The analysis shows that the effect of the shear stress is higher with a decreasing tube diameter for the same Reynolds number of the mixture. On the contrary, the effect of the shear stress decreases with an increasing concentration of the NCG in the mixture.

Measurement	1	2	3	4	5	6
Air mass concentration [%]	23	26	41	52	60	69
Inlet mixture velocity [m/s]	2.8	2.5	3.7	5.1	5.2	4.3
Inlet mixture temperature [°C]	94.9	93.9	91.1	85.3	82.1	76.4
Outlet mixture velocity [°C]	0.66	0.7	1.7	1.2	3.7	2.9
Outlet mixture temperature [°C]	43.6	40.6	74.6	71.1	64.6	51.6
Cooling water temperature [°C]	23.4	22.0	22.8	22.7	20.2	19.1
Cooling water HTC h_w [W/m^{2}K]	5916	5987	5946	5703	6087	6525
Air - water vapour mixture condensation HTC h_{cNG} [W/m^{2}K]	314	271	214	175	168	125
Corresponding calculated Nusselt HTC pure steam h_c [W/m^{2}K]	5887	5881	5884	5996	5873	5746
Reducing to [%]	5.3	4.6	3.6	3.1	2.9	2.1

Table 2. Experimental results.

In conclusion, an increase of the mixture velocity or a decrease of the tube diameter increases the effect of the shear stress, which has a positive effect on heat transfer. However, a decrease in the heat transfer due to the presence of air has a far higher effect.

Waviness on the film surface

Waviness formations on the film surface can be theoretically characterised by the critical *Re*. At certain values of *Re*, waves form on the film surface and improve the HTC of the film due to an increase in the heat transfer area. However, the exact value of Re_{crit} is often very difficult to determine and this effect is not taken into account until the empirically evaluated value $Re = 30$ [16].

The maximal calculated Re of the film during experiments was in all cases lower than 30. In study [23] an empirical number for the Nusselt formula was introduced which enables the waviness formation effect on the HTC. For the range of $5 < Re < 100$ and for the presented experimental setup, this gives an enhancement of the HTC in the range of 1.05˘1.2. These values are in accordance with the empirical factor of 1.15 as presented in [12].

As the air concentration in the mixture for the presented measurements increases, the condensation rate decreases for the same experimental parameters. This means that the amount of condensate on the tube surface is lower which corresponds to the lower *Re* of the film. Hence the effect of waviness is quite low and it is not necessary to take it into account.

Superheating and subcooling

It is well proven that subcooling of the condensate occurs very often during a film condensation [12]. In the case when steam condenses in the presence of air, superheating of the mixture also occurs since the temperate of the mixture changes according to the local saturation temperature of steam.

The temperature difference of the film at the wall and the saturation temperature is calculated and shown in the Tab. 3. As the concentration of air increases, the difference between the wall temperature of the film and the saturation temperature decreases because the steam has smaller partial pressure in the mixture and the saturation temperature is less. During experiments, the difference in additional heat transferred due to the superheating and subcooling was at maximum 4 % and 10 %, respectively. Such values have a negligible effect on the overall HTC.

Temperature dependent variables

The film flowing on the cold surface does not have a constant thickness along the vertical tube. Thermodynamic properties along the width of the film differ due to a temperature difference in the film. The most important parameters, which are influenced and are often important to include are the density, the dynamic viscosity and the heat conductivity.

As discussed in previous studies [24], the temperature dependent variables of the film have a very small effect on the HTC during the steam condensation. Especially when the saturation temperature and the wall temperature are similar. Generally, the film temperature is between the saturation temperature and temperature of the wall so the Drew reference temperature or the mean temperature is often used.

In [12, 25], the influence of temperature dependent material properties on heat transfer in the film condensation were analysed. It has been concluded that in the case when a difference between the wall temperature and the saturation temperature is less than 50 °C, a deviation from the Nusselt´s theory is less than 3 %. In [26], it was shown that a temperature difference lower than 100 °C results in a change of the HTC to just a few percent with a maximal deviation for 100 °C - 5.1 %. During experiments the mean tem-

Effect	Values during experiment	Influence on HTC during experiments	Influence with increased air concentration (constant Re)
NCG	23 – 62 mass. %	Decrease	
Flow mode of the film	Less than 30	5 – 20 %	Decrease
Superheating	15.1 – 53.3 °C	Few %	
Subcooling	42 – 57 °C (for average temperatures)	Few %	
Shear stress of flow	2.5 – 5.2 m/s	Below 2 %	Decrease
Temperature dependent variables	$T_{sat} - T_w < 60\,\mathrm{K}$ (for average temperatures)	Few %	

TABLE 3. Calculated influence of considered effects.

perature difference had no higher value than 60 °C so a deviation in the Nusselt number in the range of few percent is assumed.

5.4. An evaluation of the effects

An evaluation of the effects mentioned above and analysed for the measured conditions is given in Tab 3. Although these effects have a generally positive effect on heat transfer, the negative influence of NCG on the HTC is so high that it overcomes all other considered effects. Thus, it can be stated that a significant decrease in the condensation HTC is caused by the presence of an NCG.

5.5. Effect of non-condensable gas

For the condensation of a water vapour mixture with a mass air concentration from 23 % to 69 %, a drop in the condensation HTC to the level of 5.3 % to 2.1 % compared to the value for pure steam condensation was evaluated for the described condenser (see Tab. 2).

Moreover, the heat transfer rate is reduced by a decrease in the mixture temperature as it passes the tube (see Tab. 2), more precisely, by the temperature difference between the condensing mixture and the cooling water.

These results confirm the trend of decreasing the HTC with increasing the concentration of NGC [9, 10, 27] and expand the range of the investigation by higher concentrations of NCG. For air-steam condensation in a vertical tube, a decrease in the HTC below the value of 200 W/m^{2}K was observed in a concentration of more than 50 % of NCG.

6. Conclusion

The condensation of water vapour with a high content of a non-condensable gas in vertical tubes was experimentally investigated. The influence of the operating conditions on the condensation process was theoretically analysed. The presence of a non-condensable gas reduces the steam condensation temperature and reduces the HTC. In order to evaluate the influence of the non-condensable gas, experiments on a condensation of water vapour with air content were carried out in a vertical tube condenser. A significant decrease in the HTC value was experimentally verified. For the air concentration in a steam mixture of 23 % to 69 %, the condensation HTC decreases by a level of several percentages compared to the value of the condensation of a pure steam. A non-condensable gas in a mixture with steam decreases the intensity of the condensation. A gradual reduction of the volume and partial pressure of steam in the mixture causes a decrease in the condensation temperature of steam respective to the temperature difference between the mixture and the wall film of the condensate. A growing non-condensable gas concentration restrains the transportation of steam to the wall and this has a significant effect on the decrease of the condensation rate.

Other effects on the condensation process (the flow mode of the film, subcooling of the condensing mixture, the shear stress of flow, temperature dependent variables) have a lesser influence on the value of the condensation HTC and, conversely, they increase the HTC value. The content of non-condensable gases in the steam restrains the condensation process, which results in a reduction in the HTC.

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