A novel concept of enhanced direct-contact condensation of vapour- inert gas mixture in a spray ejector condenser

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Abstract

A novel analytical model of direct steam condensation in a spray ejector condenser (SEC) in the presence of inert gas, namely carbon dioxide has been developed. The model incorporates the continuity, momentum and energy equations to describe the phenomenon. Crucial in the process of direct condensation is atomisation of the motive fluid. Such breakup of a jet into droplets has been considered, so that the droplet diameter can subsequently be determined, on which, in turn steam is condensed. Although the temperature of droplet along the length of throat is exhibiting a rising trend, the temperature of mixture of steam is confronted a decreasing trend. Rising the value of initial velocity of droplet causes decline of the temperature of droplet and mixture at the end of the length of throat. Increasing the droplet diameter leads to decrease of the temperature of droplet and rise of the temperature of mixture of steam.

Keywords: spray ejector condenser, direct-contact condensation, jet breakup
1. Introduction

Direct-contact condensation (DCC) plays a pivotal role for both engineering and natural sciences that has been exploited in many fields [1-5]. DCC is when a gas/vapour stream comes into direct contact with a subcooled liquid. Such process is associated with much higher heat transfer coefficients than in conventional heat exchange processes [6]. DCC of vapour with inert gas on a spray of subcooled liquid exists in a number of technical applications such as for example in the nuclear industry (e.g. depressurization under normal operating conditions, safety analyses) and in the chemical industry (e.g. mixing-type heat exchanger, degasser, sea water desalting). Another application is when the supersonic steam jet flows combines with cold water in the mixing nozzle in the emergency cooling system of nuclear reactor, causing DCC. The condensing-injector is envisaged as a potential heat exchanger or energy-efficient pump due to its higher heat exchange coefficient, low-grade thermal energy utilization and ability to pressurize without rotating components [7]. Applications of SEC are considered in gas power cycles for condensing steam in exhaust gases [4, 5, 8]. The problem is also essential to the modelling and analysis of some fundamental phenomena of two-phase flow, such as a condensation in power engineering condenser of steam power plant. The phenomenon of direct-condensation heat transfer is primarily characterized by the transport of heat and mass through a moving vapour-liquid interface. Accurate predictions of the properties of DCC, particularly the heat and mass transfer phenomena near the vapour-liquid interface, are crucial for upgrades and designs. Literature survey shows that this problem is not sufficiently understood and described [9-14].

Condensation of vapour on a spray of drops is an even more complicated process. Drop size distribution, drop velocity, and condensation on droplets should be investigated in order to characterize it. A jet coming out of the nozzle dissipates and breaks up into various-sized drops. Condensation of vapour on the drops occurs when saturated vapour comes into contact with subcooled drops (their temperature is lower than the saturation temperature of the vapour at the given pressure). Instead of the drop size distribution that occurs in the spray, Brown [15] applied mean diameters (arithmetic mean diameter for the trajectory of drops and Sauter mean diameter for the heat
transfer rate) in his analysis of the process. Heat transfer coefficients up to 27200 W m\(^{-2}\) K\(^{-1}\) have been achieved in his experimental research of steam condensation on a spray of subcooled water drops (dia. 125-520 \(\mu m\)), suggesting the high efficiency of this type of condensation. A one parameter drop size distribution function and the volume mean diameter were utilized in the analysis of Isachenko et al. [16]. Comparing the experimental and theoretical results, they have found that the obtained rate of the process is larger than that theoretically expected [17]. Madejski et al. [18] concluded that their review of the numerical investigation shows that various types of direct contact condensation modelling approaches are still developing because of the immense diversity and complexity of the phenomenon.

In this paper, an analytical approach of direct contact condensation in a spray ejector condenser (SEC) in the presence of inert gas (CO\(_2\)) has been postulated to study the phenomenon. The model is based on consideration of the twofold influence on the process, namely due to the convective heat transfer of the mixture of steam and CO\(_2\) and the mass flow due to concentration of vapour on interface of droplets. Such analytical model allows to analyse the influence of various parameters on the DCC process, which subsequently develops knowledge on better design procedures of SEC. Moreover, effect of breakup of jets on droplets is investigated and best estimate for droplet diameter has been obtained. Also, impact of initial velocity of droplet and mixture of steam and inert gas on temperature of droplet and mixture of gases as well as the effect of diameter of droplet and geometry of throat has been scrutinised.

2. Physical ejector model and its mathematical description

The theoretical analysis in the paper concerns the issue of DCC of vapour with inert gas within ejector creating a subcooled water spray. The physical situation considered is shown in Fig. 1.
In order to maximize the device efficiency a proper ejector design and analysis is required. The ejector, being the crucial component of a considered thermodynamic cycle, which in our case is a negative CO$_2$ emission gas power plant [4, 5, 8], determines the overall performance and efficiency of the condensing steam from mixture of vapour-inert gas system. For the ejector design, the study focuses on the ejector nozzles, pre-mixing chamber, mixing section and the diffuser part.

### 2.1 Pre-mixing chamber

The considered spray ejector condenser is basically a converging-single nozzle with the condensation in mixing section inside combined with cyclone separator of droplets instead of diffuser. Fig. 2 demonstrates the division of the ejector into elementary control volumes from the inlet up to the exit.
It is assumed that the inlet properties of the fluids, which are supplied at a state denoted as “inlet” in Fig. 2, are all known. At that location the liquid jets have a subscript 1 whereas the mixture of vapour and non-condensable is denoted as g. In order to determine the properties at state 0 the procedure of reduction of inlet parameters due to change of flow geometry and friction is required. Before premixing section (state 0), the liquid jet inside the nozzle and a mixture of gases (around the nozzle) are flowing as single phase fluids. It ought to be borne in mind that the mixture of gases flowing with the velocity higher than 0.3 Ma number should be treated as a flow of compressible fluid. In the paper non compressible fluids are considered. Of our subsequent interest is determination of the pressure and mass flow rate of the steam-inert gas mixture at state 1. That location is the starting point of steam condensation process. For this purpose, the conservation equations and thermodynamic relations are applied.

In order to calculate, for a particular ejector design, the suction mass, \( m_g \) and the pressure at the beginning of the mixing zone, it is necessary to know the equations resulting from the flow in the inlet zone resulting from mass, energy and momentum conservation equations for gas mixture:

\[
m_g = u_{gin} \rho_g A_{gin} = u_{g0} \rho_g (A_0 - A_1) \tag{1}
\]

Form Bernoulli equation:

\[
p_{g, in} + \frac{1}{2} \rho_{g,0} u_{g, in}^2 = p_{g,0} + \frac{1}{2} \rho_{g,0} u_{g,0}^2 + \Delta p_{loss,1} \tag{2}
\]

The pressure drop due to losses can be evaluated from the :

\[
\Delta p_{loss,1} = K_m \frac{\rho_{g,0} u_{g,0}^2}{2} \tag{3}
\]

\( K_m \) is the coefficient of local losses which in the present calculations has been modelled as \( K_m=0.0015 \). Suction pressure \( p_{g,1} \) of mixed gases caused by liquid jet can be determined from the balance of momentum between pre mixing section and entrance to the mixing section, that means between states 0 and 1. Precise parameters are required to determine the pressure at the beginning of condensation process, namely location 1. Transition from location 0 to location 1 can be determined from the momentum balance equation of the mixture:
\[ p_{g,0}A_{g,0} + m_g u_{g,0} + p_l A_l + m_l u_l = p_{g,1}A_1 + (m_g + m_l)u_l \]  \hspace{1cm} (4)

Determination of pressure in locations 0 and 1 can be accomplished when we assume values of inlet velocity of motive liquid and steam-gas mixture, i.e. \( u_l \) and \( u_g \). Then the respective mass flow rates can be determined. Due to mixing after location 1 the total mass flow rate has velocity \( u_l \) stemming from (4).

\[ m_g = u_g \rho_g (A_0 - A_l) \]  \hspace{1cm} (5)

In the subsequent analysis it is assumed that \( m_{g,0} = m_{g,1} = m_g \). We assume also that the liquid velocity will not change over that distance, \( u_l \) value is assumed for the entire flow. Following re-arrangements, the pressure after pre-mixing is obtained:

\[ p_{g,1} = p_{g,0} a + p_l b - \frac{m_g (u_l - u_g)}{A_1} \]  \hspace{1cm} (6)

In (6) introduced are two terms related to surface area ratios, namely \( a = \frac{A_{g,0}}{A_1} \) and \( b = \frac{A_l}{A_1} \).

From the equations (1-3) we can determine:

\[ p_{gin} - p_{g0} = \frac{1}{2} \rho_g u_{g0}^2 (1 + K_m) - \frac{1}{2} \rho_g u_{gin}^2 \]  \hspace{1cm} (7)

Substituting the relation for mass flow of gas mixture into (7) we obtain the expression:

\[ p_{gin} - p_{g0} = \frac{1}{2} \rho_g A_0 u_{gin}^2 (1 + K_m) \left( 1 + \frac{A_0}{A_{in}} \right) \]  \hspace{1cm} (8)

Hence the mass flow rate of steam-inert gas mixture reads:

\[ m_g = A_0 \sqrt{\frac{2 (p_{gin} - p_{g0}) \rho_g A_0}{1 + K_m - \frac{A_0}{A_{in}}}} \]  \hspace{1cm} (9)

The pressure at location 1 can be determined from the expression (6).

At the location 0, we determined the mass flow rate of the mixture which subsequently will take place in the condensation process and the pressure at location 1, described by equation (6).

Conditions leading to state 2 can be found by a marching procedure outlined in the next section. In order to determine the properties at state 2 after an increment distance dL, the conservation equations and thermodynamic relations are applied. The aforementioned equations are applied successively for every given distance until the exit of the nozzle is reached.
2.2 Mixing section

2.2.1- Determination of state 2

The cylindrically-shaped region of the ejector with a constant-area is the mixing section. It is where the primary and secondary fluids start to interact. Then a fully mixed flow travels towards the inlet of the diffuser. Having a trial value of mass stream of vapour gas, we can determine molar concentration of vapour in the mixing chamber. This allows finding a partial pressure of vapour and gas. Partial pressure of vapour permits to evaluate temperature of saturation from which varies from location 1 to location 2. The above initial value allows to proceed calculations forward to subsequent values of coordinate x.

In order to get the values in state 2 conservation equations for mass, energy and momentum were applied together for the adiabatic process.

A cold liquid stream creating the jet has the inlet mass flowrate $m_l$, temperature, $T_{l,0}$, velocity, $u_{l,0}$, and flows along the ejector driving a vapour-gas stream having the mass flow rate $m_{m,0}$, velocity $v_{g,0}$ and temperature $T_g > T_{\text{sat}}$ – where the saturation temperature on interface depends on gas concentration in the gas-vapour mixture. Coordinates are chosen in such a way that the y axis is directed from the interface outward the liquid spray, and its origin is always located on the inlet cross section of the entrance of sub cooled liquid, whereas the x axis origin is located at the inlet cross section. The following assumptions and simplifications are made in the analysis:

1. Inlet flow rate of sub cooled liquid is $m_{l,0}$, temperature $T_{l,0}$, velocity $u = u_{l,0}$. Initial mass flow rate of steam and inert gas mixture $m_{g,0}$ is known. Mixture temperature at inlet is $T_{m,0}$ and pressure $p_g$.
2. Breakdown of liquid jets into droplets occurs after some distance from the injection of liquid to mixing chamber. At the beginning of calculations this distance is assumed zero.
3. Diameter of droplets after breakdown is $d_d$
4. Both phases are flowing under steady-state conditions. The amount of droplets in the mixing throat is constant for a given flow rate of liquid

5. Physical properties of the liquid droplets and vapour-gas mixture are constant

6. Thermal diffusivity of liquid layer is constant. The distribution of temperature in liquid spray and gas mixture due to vapour condensation on interface is described by mass and energy conservation equations

2.2.2 Effect of breakup of droplets

Firstly, it is assumed that there will be an extra length of jet prior to the liquid breakup into droplets, followed by the breakup. The liquid jet breaks into droplets under the influence of waves formed on the surface of the jet, due to instability. These waves cause a loss of stability and the formation of primary droplets. The size of these droplets results from the volume of the wave on jet and depends on jet diameter and length of waves. Developed droplets feature a minimum surface energy. For certain outflow conditions of a liquid jet, there is an optimum wavelength whose length is greater than the jet circumference, at which the wave amplitude increases and the liquid jet bursts. The droplets so formed during primary atomization further undergo secondary breakup resulting in much smaller droplets. Weber developed a relation for the size of secondary droplets:

\[
\frac{d}{D_j} = 1.436 \left( 1 + 3 \frac{We^{0.5} Re}{\sigma l} \right)^{1/6} \tag{10}
\]

where: \( We = \frac{\rho_l \bar{u} D_j}{\sigma_l} \) - Weber number, \( Re = \frac{\rho_l u \bar{D}_j}{\mu_l} \) - Reynolds number

The criterion for the beginning of primary droplets breakup is \( We > We_{cr} \). Following Jain et al [19] the value \( We_{cr} = 12 \) has been assumed.

Length of the zone from which starts the formation of primary droplets for turbulent liquid jets reads [20]:

\[
\frac{L}{D_j} = 5.0 We^{0.5} \tag{11}
\]
Secondary droplet breakup occurs under the influence of aerodynamic forces with account of viscous forces. Primary droplets are deformed and broken up by aerodynamic drag forces and surface tension forces. The maximum secondary droplets size can be determined from the definition of Weber number relationship with \( We_{cr}=12 \).

\[
D_{\text{max}} = \frac{\sigma We_{cr}}{\rho_d (u_l - u_g)^2} \quad (12)
\]

The average droplet size can be assumed as [21]:

\[
D_m \approx \frac{D_{\text{max}}}{2} \quad (13)
\]

A more accurate value could be calculated from the droplet size distribution spectrum.

### 2.2.3 Liquid droplet flow

Let us now consider the averaged mass and force balance equations for the liquid drops. Stream of drops leaving the supplying water nozzles has the mass flow rate the same as the stream of liquid on inlet channel, so the mass of droplets is the same as the mass of liquid within the length of mixing chamber:

\[
n_j m_d = \frac{\pi d_j^2 \rho_j}{6} n_j = m_{j0} = \frac{\pi D_d}{4} l \rho_l \quad (14)
\]

Where: \( n \) - total number of droplet in one jet exchanging mass and heat in mixing chamber

\[
n = \frac{3}{2} \left( \frac{D_d}{d_d} \right)^2 \frac{l}{d_d} \text{denotes total number of droplets in mixing zone}
\]

\[
n_x, \text{ number of droplets in one jet being within distance } x.
\]

\[
n_x = \frac{x}{l} \times n
\]

The motion of the liquid due to condensation of droplets in the x-direction is described by the force balance of inertia and drag forces in line with the equation:

\[
\frac{d(m_d u_d)}{dt} = -\frac{1}{2} \rho_m C_D (u_d - \bar{u}_m)^2 \frac{\pi d_d^2}{4} \quad (15)
\]

Or following the expansion of the left hand side

\[
m_d \frac{d(u_d(x))}{dx} u_d(x) + \frac{d(m_d)}{dt} - u_d = -\frac{1}{2} \rho_m C_D (u_d - \bar{u}_m)^2 \frac{\pi d_d^2}{4} \quad (16)
\]
Where: \( \frac{dm_d}{dt} = m_v \) denotes the mass flow rate of droplets with condensation of vapor on the droplet surface, which will be determined in the later.

Rearranging (16) we get:

\[
\frac{du_d}{dx} + \frac{1}{\frac{8}{\rho_m C_D (u_d - u_m) \frac{d}{n} \frac{d}{d^2}} + \frac{m_v}{m_d}} = 0
\]  

(17)

Following separation of variables in (17) and integrating the resulting relation we obtain the expression for the function linking the ratio of velocities, \( z = \frac{u_d}{u_m} \), in function of the distance \( x \):

\[
\int \frac{z \, dz}{z^2 - 2 \left( 1 - \frac{4 m_v}{\rho_m C_D u_m d^2} \right) z + 1} = -\frac{1}{8} \frac{\rho_m C_D d^2}{m_d} x + C
\]  

(18)

In general, the integral on the left hand side of (18) can be presented in a closed form:

for \( b^2 - 4 > 0 \)

\[
\int \frac{z \, dz}{z^2 + bz + 1} = \frac{1}{2} \ln(z^2 + bz + 1) - \frac{1}{2} \frac{b}{\sqrt{b^2 - 4}} \ln \left( \frac{2z + b - \sqrt{b^2 - 4}}{2z + b + \sqrt{b^2 - 4}} \right)
\]  

(19a)

for \( 4 - b^2 > 0 \)

\[
\int \frac{z \, dz}{z^2 - bz + 1} = -\frac{b}{\sqrt{4 - b^2}} \frac{1}{\arctan \left( \frac{2z + b}{\sqrt{4 - b^2}} \right)}
\]  

(19b)

where: \( b = -2 \left( 1 - \frac{4 m_v}{\rho_m C_D d^2} \right) \).

Exact solution of equations (19) can be sought with the following boundary conditions:

for \( x = 0 \) \( u_d = u_m \) and \( z = 1 \)

(20)

The solution can be obtained in the form:

\[
x(z) = \frac{\left( A(z) - A(1) \right)}{e}
\]  

(21)

In equation (21) the term \( e \) is defined as:

\[
e = 3 \frac{\rho_m}{4 \rho_l} d_d
\]  

(22)
Where $\rho_m = \frac{1}{\rho_v} + \frac{1-x}{\rho_{m_0}}$ (23)

And $\rho_v = \frac{\rho_{v_0}}{\rho_{v_m}}$ (24)

We can formulate a simple approximate relation, corresponding motion of droplet in stagnant environment, which can be serving as a first iteration of the problem in numerical procedure allowing to solve (19) as:

$$u_d = u_d0 \exp\left(-\frac{3\rho_mC_D}{4d_d\rho_l}x\right)$$

(25)

### 2.2.4 Condensation of vapour on subcooled droplet stream

Mass balance on the droplet’s interface gives the relation:

$$\frac{dm_d}{dt} = \rho_v \omega \pi d_d^2$$

(26)

Where: $\dot{m}_v = \rho_v \omega_v$ - mass flux of condensing vapour, which will be analysed later.

The mass flux of condensing vapour can be determined taking advantage of the Fick’s Law:

$$\rho_v \omega_v = -D_v \frac{d\rho_v}{dn}$$

(27)

Introducing equation (27) to (26) returns the expression for the rate of change of droplets in function of mass fractions in the vicinity of the droplet and freestream concentration:

$$\frac{dm_d}{dt} = -\rho_v D_v \frac{d\omega}{dn} \pi d_d^2 = -\rho_v D_v \frac{\omega_d - \omega_\infty}{d_d} \pi d_d^2$$

(28)

Where: $\omega = \rho_v/\rho_m$ yields the mass fraction, whereas $\omega_d - \omega_\infty$ is a difference in concentrations between the vicinity of the droplet and the freestream concentration, which lead to the differences in partial pressures between these locations, and subsequently promoting the condensation process.

After rearranging (28) and introducing the definition of Sherwood number $Sh$ the following form of the rate of mass droplets is obtained:

$$\frac{dm_d}{dt} = \pi \rho_v l_m^2 D_v (\omega_\infty - \omega_d) Sh = S$$

(29)

Where $l_m$ represents the distance between the centres of neighbouring droplets, $l_m = \frac{d_d (D_v)}{2}$.
Following the introduction of the chain rule of differentiation, i.e. \( \frac{d}{dt} = \left( \frac{d}{dx} \right) \left( \frac{dx}{dt} \right) = \left( \frac{d}{dc} \right) u_c(x) \), the change of the mass of droplets along the distance of the mixing chamber is obtained:

\[
\frac{dm_d}{dx} = \frac{\pi \rho_d D_v (\omega_c - \omega_d)}{u_d(x)} Sh = \frac{S}{u_d(x)}
\]

(30)

Where \( S \) is defined by equation (29).

In our subsequent calculations we will also utilise the following relations for the longitudinal rate of mass of droplets:

\[
\frac{dm_d}{dx} = \frac{d(m_d)}{dt} \frac{dt}{dx} = \frac{dS}{dx}
\]

(31)

Where the definition of the Sherwood number \( Sh \) denotes the dimensionless concentration gradient at the droplet surface:

\[
Sh = h_{m,g} \frac{d_d}{D_v}
\]

(32)

The Sherwood number can be expressed from established empirical correlations. In the present study the Ranz-Marshall correlation has been used, which takes into account the difference in velocity between the droplet and the surrounding gas, when information about the Reynolds number and the Schmidt number are known:

\[
Sh = 2 + 0.6 Re^{0.5} Sc^{0.33}
\]

(33)

Where the droplet Reynolds number yields:

\[
Re = \frac{d_d(u_d - u_m)}{v_t}
\]

(34)

And the Schmidt number reads:

\[
Sc = \frac{v}{D_v}
\]

(35)

In (32) \( h_m \) denotes the mass transfer coefficient, which can be approximately evaluated through the Lewis number definition, \( Le = \frac{a}{D_v} \), in relation to the convection heat transfer coefficient \( h_{m,g} \):

\[
h_m = \frac{h_{m,g}}{\rho_e g c_{p,fg}}
\]

(36)
Bearing in mind that the mass flow rate of vapour and gas is known we can determine the molar concentration of vapour and inert gas at the arbitrary throat location in function of the flow rate of mixture of gases. Concentration of inert gas yields:

\[ n(x) = \frac{\frac{m_v(x)}{M_v}}{\frac{m_v(x)}{M_v} + \frac{m_g}{M_g}} \]  

(37)

Where the amount of droplets condensing from the flow can be evaluated from the mass continuity:

\[ m_v(x) = m_{v0} - n_x \int_0^x \frac{dm_d}{dx} dx = m_{v0} - n_x S x = m_{v0} - n S \frac{x}{l} \]  

(38)

This allows to find the partial pressure of water vapour and inert gas:

\[ p_v(x) = \frac{\frac{m_v(x)}{M_v}}{\frac{m_v(x)}{M_v} + \frac{m_g}{M_g}} p = n(x) p \]  

(39)

Where, p denotes the total pressure. Partial pressure of gas fraction results from the Dalton’s Law:

\[ p_g = p - p_v \]  

(40)

The mass fraction difference in concentrations between the vicinity of the droplet and the freestream concentration \( \omega_d - \omega_{ao} \), can be calculated using the equation of state for ideal gas:

\[ p_v = \rho_v R_v T_m \]  

(41)

When a non-condensable gas is present in the condensation space, both heat and mass transfer must be analysed. There exists a boundary layer in which the partial pressure of the condensable vapour \( p_v \) decreases from the constant concentration in the free stream to the value \( p_{ad} \), where the vapour is condensing to a liquid. The condensing liquid must diffuse through this boundary layer to the liquid-vapour interface. The partial pressure of the non-condensable gas \( p_g \) on the other hand increases from the pressure in the core \( p_{an} \) to the value \( p_{ad} \) at the liquid-vapour interface. At any point in space and time the summation of the partial pressures of this mixture must equal the total pressure, \( p = p_v + p_g \) (Dalton’s Law).

Decreasing partial pressure of vapour in the end approaches the pressure at interface and its corresponding saturation temperature \( T_{sat}(p_v) \).
The pressure of condensing vapour above the liquid droplet can be determined from Clausius - Clapeyron equation:

\[
\frac{dp_v}{dT_d} = \frac{h_{lv}}{(v_v - v_l)T_d}
\]  \hspace{1cm} (42)

Assuming that specific volume of vapour is significantly greater than that of liquid, \(v_v \gg v_l\) and taking advantage of the equation of state we get:

\[
\frac{dp_v}{dT_d} = \frac{h_{lv}p_v}{R_vT_d^2}
\]  \hspace{1cm} (43)

Integrating (43) we obtain the distribution of vapour pressure:

\[
p_v = C \exp\left(-\frac{h_{lv}}{R_vT_d}\right)
\]  \hspace{1cm} (44)

where C is a constant dependent on the fluid, \(h_{lv}\) – latent heat of condensation dependent on the fluid.

In effect the total mass flow rate of droplets is:

\[
\dot{m}_l(x) = \dot{m}_{l0} + n_x \int_0^x d \frac{m_d(x)}{dx} dx
\]  \hspace{1cm} (45)

Taking advantage of (28) and (31) we can find approximate values of liquid mass flow rate:

\[
\dot{m}_l(x) = \dot{m}_{l0} + n_x S = \dot{m}_{l0} + \frac{x}{l} S
\]  \hspace{1cm} (46)

The length required for a particular stream of droplets to completely condensate from the mixture of gases can be determined approximately from the mass balance equation, assuming that steam occupies the total volume of the mixing chamber:

\[
l_{end} = \frac{4nS}{\pi D^2 \rho_v}
\]  \hspace{1cm} (47)

**2.2.5 Heat balance of droplet stream**

The energy balance can be used to determine temperature of a stream of droplets. That means that on one side the change of enthalpy of a stream of droplets and heat obtained due to condensation are forming the rate of heat transferred to the mixture of vapour and non-condensable gas:

\[
\frac{d}{dx}(c_p \dot{m}_d T_d) + \frac{h_{iv}}{d} \dot{m} = \frac{dQ}{dx}
\]  \hspace{1cm} (48)

That rate of heat is subsequently transferred by convection to the interface of the droplet surface:
\[ \frac{dQ}{dx} = \pi d_t^2 n \eta h_m g (T_m - T_d) \]  

(49)

where \( h_m g \) is a convective heat transfer coefficient, whereas \( \frac{dm_d}{dx} \) is described by (28).

Introducing (29) and (49) to (48) and rearranging the longitudinal droplet temperature gradient is obtained:

\[ \frac{dT_d}{dx} = - \left( \frac{T_d}{m_d} + \frac{h_{lu}}{c_{pu} m_d} \right) \frac{dm}{dx} + \frac{\pi d_t^2 n h_m g (T_m - T_d)}{c_{pu} m_d} \]  

(50)

Equation (50) is a first order ordinary linear differential equation with variable coefficients.

To find the solution for temperature \( T_d \) the knowledge of temperature of flowing vapour-gas mixture \( T_m \) is required. For this purpose, balance of energy for the mixture of vapour will be developed.

2.2.6 Balance of mass and heat for vapour-gas mixture

Balance of mass for the mixture of vapour and gas rate yields:

\[ m_m(x) = m_{m0} - \int_0^1 \frac{dm_d(x)}{dx} dx \]  

(51)

Incorporating into (51) equation (31) the following expression is obtained:

\[ m_m(x) = m_{m0} - n \int_0^x \frac{dm_d(x)}{dx} dx = m_{m0} - \frac{x}{l} S \]  

(52)

Mass flow rate of mixture vapour-gas allows calculating velocity distribution of mixture, assuming the area of the flow is not changing:

\[ u_m(x) = u_{m0} \frac{m_m(x)}{m_{m0}} \]  

(53)

Temperature of flowing vapour-gas mixture can be determined from the balance of heat for the mixture of vapour-gas flow as:

\[ -c_{pm} d m_m(T_m) = dQ = \frac{h_m d_t^2 n}{l}(T_m - T_d) dx \]  

(54)

or in the form when the left hand side of eq. (54) is expanded:

\[ -c_{pm}(m_m d T_m + d m_m T_m) = \frac{h_m d_t^2 n}{l}(T_m - T_d) dx \]  

(55)

Eq. (55) can be rearranged to provide the equation for temperature gradient for the mixture of gases:
\[
\frac{dT_m}{dx} = -\frac{T_m dm}{m_m dx} - \frac{h_{m \pi d^2 n}}{l m_m c_{pm}} (T_m - T_d) \tag{56}
\]

As can be seen we obtained a first order ordinary linear differential equation with variable coefficients. Equations (50) and (56) can be solved numerically. To find approximate solution we subtract eq. (50) from (56) obtaining:

\[
\frac{dT_m - T_d}{dx} + \frac{h_{m \pi d^2 n}}{l (m_m c_{pm} + m_d c_{pl})} \left( \frac{T_d}{m_m} - \frac{T_m}{m_d} + \frac{h_{m \nu}}{m_m c_{pm}} \right) \frac{dm}{dx} \tag{57}
\]

In line with equation (30) \( \frac{dm}{dx} = \frac{nS}{T} \). Assuming that coefficients in eq. (57) are not changing too much and are approximately constant we can find the solution for the difference of temperatures \((T_m - T_d)\) assuming the following initial conditions:

For \( x=0 \rightarrow T_m = T_{m0} \) and \( T_d = T_{d0} \) \( \tag{58} \)

The solution to eq. (57) yields:

\[
(T_m - T_d) = \frac{B n S}{l f} (1 - \exp(fx)) + (T_{m0} - T_{d0}) \exp(-f) \tag{59}
\]

where:

\[
B = \left( \frac{T_d}{m_m} - \frac{T_m}{m_d} + \frac{h_{m \nu}}{m_m c_{pm}} \right), \quad f = \frac{h_{m \pi d^2 n}}{l (m_m c_{pm} + m_d c_{pl})} \left( \frac{T_d}{m_m} - \frac{T_m}{m_d} + \frac{h_{m \nu}}{m_m c_{pm}} \right)
\]

Having the above difference of streams temperatures, we can determine temperature of the vapour-inert gas mixture as well as that of the cold water on which vapour is condensing.

Introducing (56) to (59) we obtain the temperature gradient of vapour-inert gas mixture:

\[
\frac{dT_m}{dx} = -\frac{h_{m \pi d^2 n}}{l m_m c_{pm}} (T_m - T_d) - \frac{T_m dm}{m_m dx} = \frac{K(T_m - T_d)}{m_m} - \frac{T_m n S}{m_m l} \tag{60}
\]

Where \( K = -\frac{h_{m \pi d^2 n}}{l m_m c_{pm}} \)

Integrating eq. (60) with initial conditions, for \( x=0 \) \( T_m = T_{m0} \) the temperature of the mixture is obtained:

\[
T_m = \frac{T_{m0} n S}{m_m l} \frac{h_{m \pi d^2 n}}{l m_m c_{pm}} \left( \frac{T_m - T_d}{f} \exp(fx) + \frac{B n S}{l f} \left( x + \frac{\exp(-fx)}{f} \right) \right) + C_1 \tag{61}
\]

Where: \( C_1 = T_{m0} + \frac{h_{m \pi d^2 n}}{l m_m c_{pm}} \left( \frac{T_m - T_d}{f} + \frac{B n S}{l f} \right) \) \( \tag{62} \)
Having determined temperature $T_m$ and temperature difference between streams we can find temperature of droplets surface $T_d$ as:

$$T_d = - (T_m - T_d) + T_m$$  \hspace{1cm} (63)

Equations (59) and (61) have terms containing expression $S$ responsible for process of condensation. To see the effect of condensation, for comparison $S=0$ can be assumed in eq. (59) and (61), which means the lack of condensation on droplets. In such case we can also get approximate equations describing the problem and by applying a similar procedure we determine the difference of streams of temperatures and then temperatures of the streams themselves.

Heat obtained from the mixture of vapour and inert gas yields:

$$dQ = -c_{pm} m_m dT_m = \frac{h_m \pi d_{pm}^2}{l} (T_m - T_d) dx$$  \hspace{1cm} (64)

Heat transferred to the droplet surface:

$$dQ = -c_{pd} m_d dT_d = \frac{h_m \pi d_{pd}^2}{l} (T_m - T_d) dx$$  \hspace{1cm} (65)

Determining the difference between the two of them we obtain:

$$d(T_m - T_d) = -\left(\frac{1}{m_{cm_{pm}}} + \frac{1}{m_{cm_{pd}}}ight) \frac{h_m \pi d_{pm}^2}{l} (T_m - T_d) dx$$  \hspace{1cm} (66)

Solution of (66) yields:

$$T_m - T_d = (T_{m0} - T_{d0}) \exp(-fx)$$  \hspace{1cm} (67)

Where: $f = -\left(\frac{1}{m_{cm_{pm}}} + \frac{1}{m_{cm_{pd}}}ight) \frac{h_m \pi d_{pm}^2}{l}$.

From (67) we can determine temperature of the mixture of gases and water flow:

$$dT_m = - \frac{h_m \pi d_{pm}^2}{l m_{cm_{pm}}} (T_{m0} - T_{d0}) \exp(-fx) dx$$  \hspace{1cm} (68)

Integration of eq. (68) leads to determination of the mixture of gases and droplet temperatures, respectively:

$$T_m = T_{m0} - \frac{h_m \pi d_{pm}^2}{l m_{cm_{pm}}} (1 - \exp(-fx))$$  \hspace{1cm} (69)

$$T_d = T_m - (T_{m0} - T_{d0}) \exp(-fx)$$  \hspace{1cm} (70)

The process of condensation stops when temperature of the drops will reach temperature of saturation.
2.2.7 Pressure drop in mixing chamber

The pressure drop of the Venturi throat is the sum of friction and momentum terms [22]:

\[
\Delta p = \frac{2}{\rho_v \cdot d} \cdot G^2 \cdot I_z + G^2 \cdot \left( \frac{x_1^2}{\varphi_1} - \frac{(1-x_1)^2}{\rho_1} \right)_{z=z_1} - G^2 \cdot \left( \frac{x_0^2}{\varphi_0} - \frac{(1-x_0)^2}{\rho_0} \right)_{z=z_0}
\]

(71)

Where:

\[ I_z = \int_{z=z_0}^{z=z_1} f_v \cdot x^2 \cdot \varphi_v^2 \, dz \]  

(72)

\[ \varphi_v^2 = (1 + C \cdot X + X^2) \]  

(73)

\[ X = \frac{-2 \cdot f_l \cdot G^2 \cdot (1-x)^2}{\rho_l \cdot d} \]  

(74)

\[ f_v = B \cdot Re_v^{-n} \]  

(75)

\[ f_l = B \cdot Re_l^{-n} \]  

(76)

Where \( B = 0.079 \) and \( n = 0.25 \) or \( B = 16 \) and \( n = 1 \) for the turbulent and laminar flow, respectively.

The value of the constant \( C \) differs depending on the flow regime associated with the vapour flow alone through the Venturi throat. In the research studies presented in the current paper only two flow regimes have been considered, namely turbulent (vapour)-turbulent (liquid) and turbulent (vapour)-viscous (liquid) two-phase flow. Accordingly, the value of the parameter \( C \) used in the calculus was 20 or 12, respectively. Following Butterworth, the general form of the void fraction correlations is introduced through the following equation [22]:

\[ \varphi = \left[ 1 + B_B \cdot \left( \frac{1-x}{x} \right)^n \cdot \left( \frac{\rho_v}{\rho_l} \right)^{n_2} \cdot \left( \frac{\mu_v}{\mu_l} \right)^{n_3} \right] \]

(77)

and \( \varphi_1 \) and \( \varphi_0 \) were computed using formula. In the current research studies there were set the following values of the parameters \( B_B = 1, \, n = 1, \, n_2 = 1, \, n_3 = 0 \), corresponding to the homogenous two-phase flow regime. The analytical two-phase fluid flow pressure drop was calculated using formula which is commonly known and available in the literature. In the pressure drop calculus all needed parameters such as the friction factor for vapour coming alone in the tube, the quality, the
two-phase multiplier and the void fraction come from the analytical model results. The pressure drop was calculated in two ways. In the first one there were used the two-phase fluid flow parameters for the total flow in the Venturi. In the second one there were used parameters of the core of the flow. The total pressure drop and the pressure drop in the flow core should be equal. Calculating the pressure drop based on the core flow parameters requires the knowledge about the flow morphology in the core which results directly from the analytical model results presented in the paper. The core pressure drop was calculated with use of the following equation [22]:

\[
\Delta p = 2 \cdot \frac{G^2_{\text{core}}}{\rho_v} \cdot I_x \cdot \frac{d}{l} + \frac{G^2_{\text{core}}}{\varphi_{c,1}} \left( \frac{x^2_1}{\rho_v} - \frac{(1-x_1)^2}{\rho_l} \right) - \frac{G^2_{\text{core}}}{\varphi_{c,0}} \left( \frac{x^2_0}{\rho_v} - \frac{(1-x_0)^2}{\rho_l} \right) \bigg|_{z=z_1} - \frac{G^2_{\text{core}}}{\varphi_{c,0}} \left( \frac{x^2_0}{\rho_v} - \frac{(1-x_0)^2}{\rho_l} \right) \bigg|_{z=z_0} \tag{78}
\]

\[
\varphi_c = -\frac{\rho_l \cdot u_l - \rho_v \cdot u_v}{1 - \frac{\rho_l}{\rho_v} \cdot \frac{u_l}{u_v}} \tag{79}
\]

Above equation was derived for the purpose of the core void fraction estimation. After substituting the needed parameters mentioned just above, the core pressure drop can be estimated. If the two-phase flow is fully mixed presser drop in core flow is the same as total pressure flow in throat.

### 2.3 Diffuser

In diffuser mainly noncondensing gas and liquid water droplets are flowing. Pressure drop the diffuser can be analysed as for a two-phase flow case. For simplicity pressure drop between entry and exit and space-averaged quantities are used for velocities. From Bernoulli equation is:

\[
p_2 + \frac{1}{2} \rho_2 V_{\frac{2}{2}}^2 - f \frac{1}{2} \rho_2 V_{\frac{2}{2}}^2 = p_3 + \frac{1}{2} \rho_3 V_{\frac{3}{2}}^2 \tag{81}
\]

Where f-friction coefficient is from the experiments. The diffuser of the ejector is responsible for compressing the fluid to the back pressure. The available kinetic energy at the diffuser inlet is used to elevate the pressure. Thus, the flow velocity decreases as the fluid passes through the diffuser. The analysis of the controlled element in the diffuser is similar to the analysis used in the diverging nozzle.
3. Results and discussion

In this section, the effects of initial velocity of mixture of steam and droplets, diameter and breakup of droplets and throat diameter have been investigated with respect to the length of the process. For calculation of $u_d$, $S$, $\bar{m}_m(x)$ and $u_m(x)$, equations (26), (31), (52) and (54) have been used, respectively. Calculations have been using the MATLAB code. The MATLAB code applies an iterative solution approach in which the value of $x$ is assumed and the $T_d$ and $T_m$ values are to be determined. Since the solution method exploits an iterative process all other parameters will be given as inputs (Table.1) and the dependence of parameters on $T_d$ and $T_m$ is evaluated. The parametric study has been accomplished by varying the dependent parameters ($u_d$, $u_m$ and $n$), whilst keeping other as constants. The convergence of the solution has been acquired by keeping the error tolerance as $1 \times 10^{-6}$. As mentioned earlier, the length required for a particular stream of droplets to completely evaporate can be determined as $z_{end} = \frac{m_v(0) l_{throat}}{n S}$, where $z_{end}$ is the length of condensation zone and if $z_{end} > l_{throat}$, it means that $l_{throat} = 0.2$ (m) is appropriate assumption. So,

$$m_{v0} = r_{steam} \bar{m}_{m0}$$  (88)
Taking the data for the case study from Table 1, the value of $z_{end}$ is greater than $l_{throat}$, so in this study a value of $l_{throat} = 0.2$ has been considered.

Table 1. Calculated and input parameters

<table>
<thead>
<tr>
<th>Calculated Parameters</th>
<th>Input data</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$u_d$</td>
<td>$u_{d0}$</td>
<td>20-100 (m/s)</td>
</tr>
<tr>
<td></td>
<td>$\rho_m$</td>
<td>$\frac{1}{\rho_{steam}^{x} + \rho_{inert gas}^{1-x}}$ (kg/m$^3$)</td>
</tr>
<tr>
<td></td>
<td>$d_d$</td>
<td>1-20 (mm)</td>
</tr>
<tr>
<td>$u_m$</td>
<td>$u_{m0}$</td>
<td>0-45 (m/s)</td>
</tr>
<tr>
<td></td>
<td>$m_{m0}$</td>
<td>$\frac{x n}{l} \cdot S$ (kg/s)</td>
</tr>
<tr>
<td></td>
<td>$m_{steam}$</td>
<td>$m_{steam} + m_{inert gas}$ (kg/s)</td>
</tr>
<tr>
<td></td>
<td>$m_{inert gas}$</td>
<td>$r_{steam} \times m_g \times \frac{M_{steam}}{M_{mix}}$</td>
</tr>
<tr>
<td></td>
<td>$m_g$</td>
<td>$r_{inert gas} \times m_g \times \frac{M_{inert gas}}{M_{mix}}$</td>
</tr>
<tr>
<td></td>
<td>$M_{steam}$</td>
<td>$(A_0 - A_j) \times \rho_g \times u_{m0}$ (kg/s)</td>
</tr>
<tr>
<td></td>
<td>$M_{inert gas}$</td>
<td>18 (kg/kmol)</td>
</tr>
<tr>
<td></td>
<td>$r_{steam}$</td>
<td>44 (kg/kmol)</td>
</tr>
<tr>
<td></td>
<td>$r_{inert gas}$</td>
<td>0.9</td>
</tr>
<tr>
<td></td>
<td>$\rho_v$</td>
<td>0.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\frac{p_s M_s}{R_u T_{m0}}$ (kg/m$^3$)</td>
</tr>
<tr>
<td>S</td>
<td>( l_m )</td>
<td>( \frac{d_d\left(D_d\right)^2}{2\left(D_v\right)^2} )</td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>( D_v )</td>
<td>( 1.5 \times 10^{-3} \frac{(m^2)}{s} )</td>
<td></td>
</tr>
<tr>
<td>( \omega_{\infty} )</td>
<td>1.573</td>
<td></td>
</tr>
<tr>
<td>( \omega_d )</td>
<td>0.062</td>
<td></td>
</tr>
<tr>
<td>( Sh )</td>
<td>( v_l )</td>
<td>( 31.6 \times 10^{-6} \frac{(m^2)}{s} )</td>
</tr>
<tr>
<td>( m_d )</td>
<td>( \dot{m}_d + \frac{x n}{l} S )</td>
<td></td>
</tr>
<tr>
<td>( h_{lv} )</td>
<td>( 2460.6 \frac{(kJ)}{kg} )</td>
<td></td>
</tr>
<tr>
<td>( C_{pl} )</td>
<td>( 4.190 \frac{(kJ)}{kg.K} )</td>
<td></td>
</tr>
<tr>
<td>( n )</td>
<td>Eq.13</td>
<td></td>
</tr>
<tr>
<td>( l )</td>
<td>0.2 (m)</td>
<td></td>
</tr>
<tr>
<td>( h_m )</td>
<td>( 580.5 \times 10^{-6} \frac{(m)}{s} ) (Eq.33)</td>
<td></td>
</tr>
<tr>
<td>( T_m )</td>
<td>( r_{steam} C_{pl} + r_{inert gas} C_{inert gas} \left( \frac{kJ}{kg.K} \right) )</td>
<td></td>
</tr>
<tr>
<td>( C_{pm} )</td>
<td>0.971 ( \frac{(kJ)}{kg.K} )</td>
<td></td>
</tr>
</tbody>
</table>

### 3.1 Effect of initial velocity of mixture of steam

Temperature of droplet and mixture of steam are depicted in Figs. 3 and 4, respectively. It can be seen vividly that temperature of droplet along the length of throat is faced with a rising trend, whereas temperature of mixture of steam is experiencing a decreasing trend. In addition, having studied the data from Figs 3 and 4 it can be considered that temperature of droplet at the length of throat \( (x=0.2 \text{ m}) \) for \( u_{m0} = 0, 5, 15, 25, 35 \) and \( 45 \text{ (m/s)} \) is 331.55, 332.08, 333.25, 334.66, 336.50 and 339.54 K, respectively, whilst temperature of mixture of steam for mentioned \( u_{m0} \) is 339.31, 340.24, 342.40, 345.15, 348.99 and 356.21 K, respectively. So, increasing the value of \( u_{m0} \) not only results in increasing the droplets temperature, but also leads to rise in temperature of mixture of steam at the
length of throat. Moreover, the lowest value for temperature of droplet and mixture of steam is when the drop is moving in stagnant environment ($u_m(x) = 0$).

Fig. 3. Distribution of temperature of droplet for different initial velocity of mixture at $U_d_0=50$ m/s and diameter of droplet 1 mm

Fig. 4. Distribution of temperature of mixture of steam for different initial velocity of mixture at $U_d_0=50$ m/s and diameter of droplet 1 mm

3.2 Effect of initial velocity of droplet

Figures 5 and 6 indicate the temperature of droplet and mixture of steam along the length of throat for wide range of initial velocity of droplet (20-150 (m/s)), respectively. Results show that although temperature of droplet faces a rising trend for $U_d_0 = 20$, 50 and 100, it is reaching a peak and then falls after $z=0.15m$ for $U_d_0 = 150$ (m/s). In addition, increasing the value of initial velocity of droplet from 20 to 150 (m/s) causes decline the temperature of droplet at the length of throat $(x=0.2 \ m)$, so that its value for 20, 50, 100 and 150 (m/s) is 336.5596, 331.9848, 328.1053 and 325.4665 (K), respectively. On the other hand, the temperature of mixture of steam has been faced with a falling trend for mentioned initial velocity of droplet. Moreover, augmenting the initial velocity of droplet from 20 to 150 (m/s) result in decline the temperature of mixture of steam, so that it decreases by 5.2% between 20 and 150 (m/s) at the full length of throat $(x=0.2 \ m)$. 

This preprint research paper has not been peer reviewed. Electronic copy available at: https://ssrn.com/abstract=4426192
6.3 Effect of diameter of droplet

According to Eq. 17, there is a direct relation between the number of droplet and its diameter under the constant geometry of the throat. Distribution of temperature of droplet and mixture of steam at different diameter of droplets have been represented in Figs. 7 and 8, respectively. Initial velocity of droplet and mixture is 50 m/s and 20 m/s, respectively, and diameter of throat \(D_0\) is equal to 9 mm. Results demonstrate that although temperature of droplet except for 1, 1.5 and 2 mm of diameter has a surging trend along length, temperature of mixture of steam is faced with a decreasing trend. For 1 and 1.5 mm of diameter of droplet, maximum value of temperature of droplet takes place at 8 and 12 cm of length, respectively, and then exhibits a slight reduction. For 2 mm, it boosts up to 16 cm of length and then remains approximately constant. In addition, increasing the diameter of droplet leads to decrease the temperature of droplet and rise the temperature of mixture of steam. As it can be seen, distribution of temperature of mixture of steam for 1mm, apart from a brief rise again from 8 to 12 cm, gradually declined from 12 to 20 cm of length of throat.
4. Experimental facility

The process of direct contact condensation is taking place in the main part of SEC that is in the throat. In the developed facility the physical processes in the throat only are investigated experimentally by authors of the present paper, as the processes in converging and diverging parts of Venturi nozzle are pretty well known in literature for single or two-phase flow. The test section of the spray ejector throat is made in the form of a transparent tube of 80 mm internal diameter. A schematic diagram of the facility together with a photograph is presented in Fig. 9. A motive fluid is supplied through the 1 mm nozzle. The nozzle is fed with motive water at average inlet temperature of 8°C and pressure ranging from 5.68 to 10.17 bar at the nozzle inlet what corresponds to mass flow rates of water ranging from 17.56 to 22.90 g/s. The value of inlet pressure equal to 5.68 bar has been noticed as a minimum value at which the spray of drops is formed and the jet breakup takes place. The length of the throat is 600 mm.

A mixture of water vapour and carbon dioxide is fed into the mixing length through the mixing chamber and an inlet manifold. The wet vapour is received from the Battistella Saturno MAX/S generator at the pressure close to atmospheric and temperature of about 100 degrees Celsius. The maximum vapour feed used for experiment is 6 g/s and its regulation takes place with the use of precision valve and a by-pass. Maximum carbon dioxide feed is 6.7 m³/h. The mixing chamber with a volume of 2.4 dm³ ensures mixing, temperature equalization and removal of the excess water condensate.
The steam-gas mixture is fed to the throat where it meets spray of drops and the process of direct contact condensation occurs. The process of jet breakup is not taking place right away, but after some distance (depending on the inlet pressure of motive water). Then the products of the process go to the separation chamber, 6, with a volume of 1.9 dm$^3$ through the outlet manifold. The vapour – all or partially, depends on experimental conditions – goes out from the siphoned outlet on the bottom of mixing chamber and is collected for titration or weighing. The subtraction of the known amount of
water from the nozzle from the total volume gives information about the amount of condensed vapour during the process. Carbon dioxide with possible excess, non-condensed vapour is released into the atmosphere through the outlet located on the upper part of the separation chamber.

The water feeds the nozzle under municipal water supply pressure (3 bar), raised by the Lead Fluid model CT3001S gear pump. This device ensures a pulse-free flow and, under measurement conditions, stabilization of the flow with an accuracy of ±5 ml/min. A flow rate measurement is made using Atrato AT740 mass flowmeter and is independently verified by a titration at the beginning and the end of each measurements series.

The measurement series consists in going through the full vapour input range for a fixed spray of drops feed and, optionally, a fixed carbon dioxide flow rate or in going through the full carbon dioxide input range at a fixed rate of vapour and spray of drops. It gives the 5-10 measuring points each time.

The vapour and carbon dioxide flow rates are measured using appropriately calibrated Krohne glass rotameters. Water pressure measurement at the nozzle inlet is performed with the Trafag NAH type 8253.80.2317 transducer. The measurements are carried out using T-type thermocouples with an accuracy of ±0.3 K.

The analytical distributions of temperature of mixture of steam were compared with predictions resulting from the model described earlier. Figures 10 and 11 show the distribution of temperature at the inlet and outlet of mixing length, respectively, for different mass flow rates of steam (1.2, 2.4 and 3.6 g/s) for a constant mass flow rate of liquid jet set to 29 (g/s). It is apparent that increasing mass flow rate of steam leads to increased levels of temperature at the inlet and outlet of the mixing length, whilst at the constant mass flow rate of steam, enhancement of flow rate of CO₂ causes both inlet and outlet temperature of mixing length to decrease. The distributions of temperature along mixing length for different mass flow rate of steam (1.2, 2.4 and 3.6 g/s) when flow rate of CO₂ is 0, 2.6, 4.6 and 6.7 (m³/h) at liquid jet flow rate = 29(g/s) have been illustrated in Fig. 12.
Fig. 10. Distribution of inlet temperature at liquid jet mass flow rate = 29 (g/s)

Fig. 11. Distribution of outlet temperature at liquid jet mass flow rate = 29 (g/s)

Fig. 12. Distribution of temperature along mixing length for flow rate of CO$_2$ (m$^3$/h): a) 0, b) 2.6, c) 4.6 and d) 6.7 (m$^3$/h) at liquid jet flow rate = 29 (g/s)
5. Model validation

5.1. Throat model validation

For validation purposes of the present model, different mass flow rate of steam was set as: 2.4, 3.6, 4.8 and 6 (g/s), respectively. The results of comparisons of distributions of temperature of mixture of steam and condensation mass flow rate have been presented in Fig. 13. As it can be seen in Fig. 13 a good agreement was achieved between the experimental results data. In Fig. 14 presented is the rate of acquired condensate from the flow in function of mass flow rate of supplied vapour. As can be noticed, at the 6 g/s of mass flow rate of vapour, condensation mass flow is 3.9 g/s, which means that it is possible that some of the vapor has condensed, but there is still some uncondensed vapor remaining in the experimental facility. A longer mixing section is required to accomplish full condensation.

Fig. 13. Distribution of temperature of mixture at liquid jet mass flow rate = 29 (g/s)

Fig. 14. Condensation mass flow rate at CO\textsubscript{2} flow rate = 2.6 (m\textsuperscript{3}/h) and liquid jet flow rate = 29 (g/s)

5.2. Pressure distribution validation in the throat
The analytical pressure data calculations were compared with predictions resulting from the experimental models described by Silva [23]. In the experimental Venturi scrubber facility the throat was of a diameter of 125 mm and a total length of 1.27 m. The throat gas velocity was within the range between 34 and 70 m/s and the liquid flow rate was set between 0.013 and 0.075 (kg/s). For validation of the present model, the liquid flow rate of 0.038 kg/s was considered and gas flow rate was set as: 0.483, 0.736, 0.861 and 0.987(kg/s), respectively. The results of comparisons of pressure at inlet from to the test section as well as at the outlet has been presented in Fig. 15. As it can be seen a good agreement was achieved between the experimental results data taken from the studies by Silva. That consistency is showing up for all values of velocities considered. Unfortunately, there were no temperature values found for the relevant comparisons.

![Fig. 15 Validation of pressure distribution along the Venturi scrubber](image)

6. Conclusions

In the paper a comprehensive simple analytical model validated with experimental data of direct contact condensation on a jet of fine droplets in the throat of the Venturi throat is presented. The direct condensation process is complex especially in the light of the fact of the presence of inert gas in vapour. In such case the mass transfer due to the presence of inert gases makes the process of direct
contact condensation less effective. The model is based on the solution of mass, momentum and energy conservation equations for the droplets flow as well as the mixture of steam and inert gas. The model of direct condensation considers the convective heat transfer mechanism along with a mass transfer due to different concentrations of vapour in the core of the flow and on the droplet. The developed model exhibits satisfactory agreement of presented novel modelling with experimental results. In effect temperature distributions of droplets and the steam-gas mixture are obtained together with the rates of condensate.

Through the thermal analysis of studied case, the following results were found:

- Although the temperature of droplet along the length of throat is accosted a rising trend, the temperature of mixture of steam is confronted a decreasing trend.

- Growth the diameter of droplet leads to decrease of the temperature of droplet and rise of the temperature of mixture of steam.

- By adjusting appropriately the length of the throat the complete condensation of steam can occur in the throat, which can be regarded as potential application in power plant condensers.

ACKNOWLEDGMENTS

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NOMENCLATURE

- thermal diffusivity, m²/s,
- specific heat, J/(kg K)
C - constant

g - gravitational acceleration, m/s²

hₘₐₙ - mass transfer coefficient

hₘ,g - convection heat transfer coefficient

K₆ - loss coefficient

Le - Lewis number

lₘ - distance between the centres of neighbouring droplets

m - mass flow rate, kg/s

mᵥ - mass flow rate of condensation of vapour on the droplet surface

N - total number of droplet

nₓ - number of droplets being within distance x

p - pressure, N/m²

q - heat flux, W/m²

R - individual gas constant

Re - Reynolds number

Sh - Sherwood number

t - time, s

T - temperature, °C

Tₜ - temperature of droplet

Tₘ - temperature of mixture of steam

We - Weber number

u - velocity in the x-direction, m/s

x - coordinate along the flow measured from throat outlet

x₀ - thermal entry (development length, m)

z - ratio of droplet velocity to mixture velocity
Greek Symbols

\( \alpha \) - local heat transfer coefficient, W/(m\(^2\) K)

\( \delta \) - liquid film thickness, m

\( \mu \) - dynamic viscosity, kg/(m s)

\( \nu \) - kinematic viscosity, m\(^2\)/s

\( \rho \) - density, kg/m\(^3\)

\( \tau \) - shear stress, N/m\(^2\)

\( \omega \) - mass fraction

\( \kappa \) - the ratio of specific heat at constant pressure to specific heat at constant volume

Subscripts

\( c \) - condensation

\( l \) - liquid,

\( 0 \) - initial parameter (at the slit outlet)

\( v \) - vapour

\( t \) - turbulent

\( x \) - axial component

\( w \) - wall

Superscript

\( + \) - dimensionless value

REFERENCES:


15. Brown, G., Heat transmission by condensation of steam on a spray of water drops. 1948, University of London (Imperial College of Science and Technology).


