Calculating investigation of low temperature desalination processes characteristics

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Abstract. The technology of low-temperature process of desalting of water is offered. Realization of this technology assumes partial evaporation of water in adiabatic steam generator, formation of two-phase stream, separation and condensation steam-phase. Results of numerical modeling of the processes in steam generator are presented, also productivity steam generator is determined.

1. Introduction. Water is fundamental and extremely important resource. Fresh water reserves are estimated to be 35.8 million km$^3$, being only 2.5 % of all water on Earth. Despite big fresh water reserves, some regions are facing severe need for fresh water. One of the reasons for such need is the fact that fresh water reserves are disproportionally distributed through regions – such, Baykal lake has 23 615.39 km$^3$ of high quality fresh water in it, which is 19 % of all fresh water reserves. Another factor is caused by increase of industrial demand for fresh water. As following, development of new water desalination methods is in high demand.

Known methods can be separated in two groups: without water phase state change (electrodialysis, ion exchange, reversed osmosis, extraction) and with it (distillation, freezing). Most widely used method is distillation. Though, this method has some disadvantages, such as high energy consumption, salts drop-out, water contamination [1, 2]. These disadvantages may be partially overcome by keeping process pressure below atmospheric and temperatures within range 40 – 70 C.

2. Desalinization. Technological process of low-temperature desalination is described on fig. 1. Saline water from reservoir 1 is pumped by pump 3 into heat exchanger 4, in which water gets heated to temperature not more than 70 C. Such temperature can be reached by low-potential heat sources or solar energy collectors. Heated water gets into two phase fluid generator 5 having temperature $T_o$ and pressure $P_o$. Within generator 5, water is exposed to adiabatic expansion in the throttle channel and to pressure drop from $P_o$ to $p_1$ after the generator.
Fig. 1. Desalination apparatus arrangements. 1 – salt water tank, 2 -filter, 3 - pump, 4 – heat exchanger, 5 – throttle, 6 - separator, 7 - condenser, 8- cooler, 9 -pump, 10 – fresh water tank. During adiabatic expansion in the throttle channel, following conditions must be met: \( p_0 < p_f(T_0) \), where \( p_f(T_0) \) is bubble point pressure for temperature \( T_0 \), and pressure ratio must be \( (p_0/p_f) < 0.5 \). Ensuring first condition is met provides vapor phase formation, and meeting second conditions provides critical flow, when flow rate does not depend on counter pressure \( p_f \) variation [3]. 

Lavale nozzles (channels with variable cross section). Generated two phase flow gets into separator 6, and separated vapor phase gets into condenser 7, which is being a mixing type heat exchanger with water of temperature 20-35 C as a working fluid, which ensures pressure range of \( p_0 = 2...5 \) kPa. From condenser, water gets pumped by pump 10 into fresh water tank 10, and part of it gets returned through three-way valve into condenser 7 through cooler 8. Apparatus efficiency is defined by water flow rate, vapor phase fraction formed during adiabatic expansion and separation quality.

3. Two-phase flow simulation. Two phase flow in generator is formed during adiabatic expansion within throttle. Axially-symmetric Lavale nozzle is used as a throttle. Two-phase flow parameters have been simulated for the non-equilibrium case of boiling flow through throttle. Both one- and two-dimensional case have been modeled. Numerical simulations provided critical flow rates through Lavale nozzle, phase concentration distributions and integral vapor content in the throttle outlet. One-dimensional model is based on the following assumptions: flow is one-dimensional, vapor phase gets formed in a metastable region at certain liquid constant overheating, liquid phase is overheated, vapor phase is saturated, phases have same velocities, density and enthalpy of system are additive to phase properties, and only liquid phase-pipe surface tension is taken into account. 

Mentioned assumptions provide following form of mass and impulse conservation equations:

\[
\frac{1}{\varphi \rho_n + (1-\varphi) \rho_x} \left[ \varphi \frac{d \rho_n}{dz} + (\rho_n - \rho_x) \frac{d \varphi}{dz} + (1-\varphi) \frac{d \rho_x}{dz} \right] + \frac{1}{w} \frac{dw}{dz} + \frac{1}{f} \frac{df}{dz} = 0
\]

\[
\varphi \rho_n w \frac{dw}{dz} + (1-\varphi) \rho_x w \frac{dw}{dz} + \frac{dp}{dz} + \Phi_\text{tr} = 0
\]

\[
x \frac{dh_n}{dz} + (h_n - h_x) \frac{dx}{dz} + (1-x) \frac{dh_x}{dz} + w \frac{dw}{dz} = 0
\]

Where \( \varphi \) is vapor volume fraction, \( \rho \) - density, \( w \) –velocity, \( h \) –enthalpy, \( \Phi_\text{tr} \) – friction force per volume unit, \( z \) – coordinate, \( f \) – field. System of equations (1) may be solved if pipe geometry \( f=f(z) \) is known, as well as phase thermodynamic properties, including metastable region: \( \rho_n=p^\varphi(T), \rho_x=p(T_x), h_n=h^\varphi(p), h_x=h(T_x) \). Liquid overheating value is defined as \( \Delta T = T_x - T_s(p) \), a difference between liquid temperature and liquid saturation temperature for pressure \( p \). Overheat providing phase transfer start in the metastable region have been set according to [4].

Simulation method included critical flow condition, at which flow rate through nozzle is aspiring to critical nozzle \( q=q_{cr} \) and pressure gradient along all the pipe is negative \( \frac{dp}{dz} < 0 \). Results of calculation of the given critical expense of \( j \) \([\text{kg} / (\text{m}^3 \cdot \text{s})]\) depending on pressure on an inlet to channel \( p_0 \) for a case of the outflow of water through Lavale nozzle from a saturation state are presented on fig. 2. In fig. 3. results of calculations of mass dryness fraction \( x \) in the output section of a nozzle depending on drop of pressure \( \Delta p = p_0 - p_1 \) are shown. During calculations of the dryness fraction pressure \( p_0 \) varied and counter-pressure remained constant equal to \( p_1 = 4000 \) kPa. Results of calculation of characteristics of a stream on equilibrium model are given in the same drawings. As seen on figs 2 and 3, equilibrium model provides overestimated values of vapor fraction and underestimated values of flow rate, compared to more sophisticated non-equilibrium model.

Noteworthy, non-equilibrium adiabatic expansion simulation for water were performed for “limiting” case, where flow is not affected by anything that could accelerate vapor fraction formation. In real conditions, due to different thermal, acoustic, mechanical and other effects, real phase transfer might deviate less from equilibrium case.
Fig. 2. Critical flow simulations results and experimental data. ● – experimental data from [5]; ▲ - experimental data from [6]

Fig. 3. Vapor mass fraction calculation results for different pressure drops $\Delta p = p_0 - p_1$; 1- equilibrium model; 2- non-equilibrium model

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Two-dimensional model is based on following assumptions: phase have different velocities and cross-interact, heat exchange between flow and pipe walls is absent. Flow continuity equation will be as following:

$$\frac{\partial \rho_m}{\partial \tau} + \nabla (\rho_m \bar{w}_m) = \dot{m}$$

(2)

where $\bar{w}_m$ - average velocity $\rho_m$ – mixture density, $\phi_i$ – vapor volume fraction of $i$ –th phase, $\dot{m}$ - mass increment per time per volume unit, $\tau$ – time.
Impulse conservation equation is obtained by summation of n impulse equations for each component of system:

\[
\frac{\partial}{\partial t}\left( \rho_m \tilde{w}_m \right) + \nabla \left( \rho_m \tilde{w}_m \tilde{w}_m \right) = -\nabla \rho + \nabla \left[ \mu_m \left( \Delta \tilde{w}_m \right) \right] + \nabla \left( \sum_{i=1}^{n} \varphi_i \rho_i \tilde{w}_{dr,i} \tilde{w}_{dr,i} \right),
\]

(3)

where \( \mu_m = \sum_{i=1}^{n} \varphi_i \mu_i \) mixture dynamic viscosity, \( \tilde{w}_{dr,i} = \tilde{w}_i - \tilde{w}_m \) – velocity slip factor of phase \( i \). Energy conservation equation is:

\[
\frac{\partial}{\partial t} \sum_{i=1}^{n} \left( \varphi_i \rho_i E_i \right) + \nabla \sum_{i=1}^{n} \left( \varphi_i w_i (\rho_i E_i + \rho_i) \right) = -\nabla \left[ \lambda_{ef} \nabla T \right]
\]

(4)

where \( \lambda_{ef} = \lambda + \lambda_{turb} \) is effective heat conductivity, \( \lambda_{turb} \) is turbulent heat conductivity (as per selected \( k - \varepsilon \) model), \( E_i \) – partial energy of phase \( i \). Details regarding equation system (2)-(4) and vapor phase formation specifics are given in [7]. Vapor phase formation mechanics have been realized using Dyne hypothesis, saying vapor phase is being formed within micro-whirls which are generated in turbulent flow. Number of vapor fraction nuclei have been provided by realization of [8] method. Vapor phase formation conditions have been modeled for two-dimensional case for the channel of variable cross-section. Reverse case for set flow rate have also been simulated. Flow have been determined for one-dimensional case as per equation system (1). Flow simulations have been performed in the “FLUENT” simulator package. Vapor fraction distribution through Lavale nozzle during adiabatic expansion is presented on fig. 4.

![Vapor volume fraction at po=1,752 MPa, To=203,8 C.](image)

Two phase flow in the vapor generator channel is characterized by structural anisotropy, which fits experimental data [5]. Periferal region of the nozzle has vapor fraction close to unity, while flow core is dominantly liquid with vapor fraction maximum value being 0.5. Such phenomenon must be taken into account for industrial applications.

4. Separation quality calculation. Case of wet gas (also contaminated by micro particles) in a centrifugal separator is simulated. It is assumed, that particles do not coagulate, phase transfer is not happening and particles and carrier gas have same temperature. Wet gas injection is tangential to the separator cylindrical vessel. Case setup and solution details are given in [8]. Results of calculations of influence of the drops’ sizes on quality of separation \( k \) are given in fig. 5. As appears from fig. 5, drops with sizes less than 10 microns unsatisfactorily separate in a vortex separator that can make negative impacts on characteristics of desalinated water. For increase of quality of separation it is offered to carry out coagulation of drops as mechanically [10], and carrying out wave impact on a vapor-liquid stream, according to the method stated in [11].

As a result, efficiency of a desalinization apparatus have been determined during research. According to the scheme at fig. flowrate of fresh water is \( j_{amp} = j \cdot x \cdot k \), where \( j \) is water flowrate through throttle per throttle cross-section field unit, \( x \) - vapor phase mass fraction, \( k \) - separation efficiency.
Fig. 5. Separation quality for different size particles at separator cross section \( z = 0.1 \) m for three different values of simulation layer thickness \( h = 2.5 \) mm (1), 5 mm (2) and 7.5 mm (3).

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