Обговорено проблеми усунення дефіциту води в посушливих регіонах планети і виконаний аналіз сучасних систем отримання води. Показані перспективи отримання води безпосередньо з атмосферного повітря при охолодженні його нижче точки роси за допомогою холодильних агрегатів.

Як холодильні агрегати в районах з надлишком сонячної енергії запропоновано застосовувати системи охолодження абсорбційного типу з водоаміачних розчином як робоче тіло (АВХА). Відзначено, що широке застосування АВХА в системах отримання води з атмосферного повітря ускладнено через невисокі енергетичні характеристики тепловикористовуючого холодильного циклу, причому основні проблеми пов'язані з нерозрахованими втратами холодильного агенту (аміаку) на етапі транспортування по дефлегматора АВХА. Особливо помітний цей внесок при експлуатації АВХА в широкому діапазоні температур зовнішнього повітря.

Для пошуку методів усунення транспортних втрат аміаку було виконано моделювання процесів тепломасообміну підйомної ділянки дефлегматора АВХА. В основі модельних уявлень лежали рівняння балансів тепла і маси, а при моделюванні враховувалося опір процесу дифузії при радіальному русі парового потоку до стінки дефлегматора. Попередній аналіз термічного опору плівки флегми показав її малий внесок у сумарне опір і в подальшому не враховувався.

В результаті моделювання виявлено значну (до 36 °C) різницю температур між потоком всередині дефлегматора і його стінкою. Результати моделювання підтверджені в процесі експериментальних досліджень серійного АВХА українського виробництва.

Отримані результати дозволили запропонувати оригінальну конструкцію теплоізоляційного кожуха дефлегматора АВХА із змінним термічним опором при відповідній зміні температури зовнішнього повітря. Це дозволило підвищити енергетичну ефективність від 18 до 36 % і продуктивність систем отримання води з атмосферного повітря

Ключові слова: системи отримання води, атмосферне повітря, точка роси, сонячні абсорбція холодильні агрегати

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IMPROVING THE ENERGY EFFICIENCY OF SOLAR SYSTEMS FOR OBTAINING WATER FROM ATMOSPHERIC AIR

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1. Introduction

Water will become the most valuable resource on the planet very soon: this trend will only grow in the near future [1]. The United Nations General Assembly declared 2005 2015 the International Decade for "Water for Life" Action to help to resolve this problem [2].

Hydrogen is one of the most promising fuels as an efficient and environmentally friendly energy carrier. However, we can also obtain water from it since the exothermic reaction of hydrogen in the oxygen medium releases water. Such method does not relate to practically effective methods of water obtaining, and it is not rational in terms of resource use [3].

Electrodialysis is a process of changing of the electrolyte concentration in a solution under an influence of electric current. People use electrodialysis for desalination of water, separation of salts from solutions. Electrodialysis refers to electro-membrane processes and combines features of both electrochemical and membrane processes. The use of the method assumes presence of sources of salt water, from which a subsequent removal of salts under an influence of an electric field and with a help of partially permeable membranes is possible [4].

The phenomenon of osmosis is a transition through a semipermeable membrane of a solvent from a solution with a low concentration of impurities to a solution with a higher concentration. Reverse osmosis proceeds in the direction opposite to a direct osmosis. Solvent passes through a semipermeable membrane from a solution with salts to an area with a pure solvent under an influence of increased pressure. The driving force of reverse osmosis is a pressure difference. The method also assumes a kind of desalination, so we need a source of seawater for this method [5].

Ion exchange is a reversible chemical reaction. An exchange of ions between an electrolyte solution and a solid ion exchanger takes place during it. Ion-exchange resins are synthetic organic ion exchangers – high-molecular synthetic compounds with a three-dimensional helium and macroporous structure. Ion exchange resins are solid polymers, insoluble, boundedly swelling in solutions of electrolytes and organic solvents, capable of ion exchange in aqueous and aqueous-organic solutions [6].

Distillation is rundown, evaporation of liquid followed by cooling and condensation of vapors. This is the most famous method of obtaining of fresh water, which originates in the I century, we can see it mentioned in works of the Greek alchemists in Alexandria (Egypt). Distillation refers to methods where the main source of fresh water is the sea [7].

Rectification is a process of separation of double or multicomponent mixtures due to countercurrent mass exchange and heat exchange between vapor and liquid. Rectification is separation of liquid mixtures into almost pure components, which differ in boiling points, by repeated evaporation of liquid and condensation of vapors. We can obtain fresh water from the air using this physical process if we reach the required conditions for condensation of liquid [8].

Each of methods [4–8] is effective and fulfills the task of obtaining of fresh water. Similarity of each method is in the use of seawater as an initial source. As water covers about 70 percent of the surface of the globe, and it is 97.5 percent of salt water [9].

Meanwhile, the bulk of fresh water lies in a one-kilometer layer of the planet's atmosphere. According to researchers [10], the average absolute humidity near the earth's surface is 11 g/m^3, it reaches 25 g/m^3 and higher in the tropical zones. Moreover, a large number of countries suffer from the lack of fresh water in the tropical zone, although its content in the atmosphere is very high.

Production of water of the atmospheric air is also possible with a supersaturation of air mass (a condensation effect when temperature reaches a temperature below the "dew point") [11].

Such method is unique, because it uses atmospheric air as a source of water. Of course, it is necessary to carry out a number of thermodynamic processes with air and this requires electric energy.

Paper [12] shows that the greatest prospects show methods associated with operation of artificial cooling generators – refrigeration units, which provide air temperature below the dew-point temperature.

The main volume of the market of equipment for separation of water from air comes from systems that have an electrically driven compression refrigeration system that guarantees a temperature below the dew point in their composition [13].

However, there are many places in the world with electricity problems in addition to water problems. These are not only the countries of the Mediterranean, Africa, South-East Asia, South America, but also the southern region of Ukraine. [14].

We should note that humanity sought for ways to minimize energy costs for obtaining of water even under the most unfavorable climatic conditions since ancient times. People obtained fresh water by collection of condensed droplets from atmospheric air due to natural daily radiative cooling of the earth's surface [11]. It is possible to isolate 10–14 g of water from each cubic meter when temperature lowers by 10–15 °C.

People use various intensifying technologies such as cold accumulators (crushed stone), evaporative-condensing heat transfer devices and moisture sorbents (12) to increase efficiency of the water production process in modern conditions.

At the same time, it is possible to carry out air cooling processes with a use of alternative energy sources, for example, solar thermal energy – solar installations based on solar collectors and solar batteries [15–18]. Solar energy is redundant in the arid regions of our planet.

In this connection, developers of modern systems for obtaining of water from atmospheric air pay attention to heat-using refrigerator units (HRU). They can use solar energy as a source of heat [19].

There are two types of HRU represented in the current market of refrigerating equipment: steam-jet refrigerating unit (SJRU) and absorption refrigerating units (ARU) [20].

SJRU have limited application due to a low coefficient of conversion of heat energy into the cold and instability of characteristics when temperature conditions of operation change [21].

There are two types of ARU: bromine-lithium units and water ammonia units. In the first type, the refrigerant is water, and the absorber is a concentrated aqueous solution of lithium bromide. In the second type, the refrigerant is ammonia, and the absorber is a concentrated water ammonia solution (WAS).

Despite rather high-energy characteristics of bromine-lithium schemes, we cannot use them in systems for obtaining of water from atmospheric air. This is due to a weak absorption capacity of lithium bromide water vapor and, consequently, a need for intensive heat removal in the absorber. We can organize intensive heat removal by liquid cooling with the subsequent use of a cooling tower. Cooling of circulating water occurs due to partial evaporation of water into the air in cooling towers. This is unacceptable for places where water resources are scarce.

Large-scale water-ammonia refrigeration units also use liquid cooling, but medium and small units can efficiently work with air-cooling, including operation in natural convection mode, without external circulation drivers (fans) [22].

It is also important to note that a working substance of water-ammonia refrigerating units is a natural substance – WAR, which does not affect the ecosystem of the planet adversely [23]. In this case, ammonia, as a refrigerant, gives possibility to expand a scope of application in terms of lowering of temperature below 0 °C, for example, for tasks of air conditioning or storage of food and raw materials.

Thus, absorption water-ammonia refrigeration units (AWRU) can solve cooling problems in systems for obtaining of water from atmospheric air in absence of sources of electrical energy effectively. The source of heat energy for AWRU can be solar collectors and solar batteries, at the same time, AWRU gives possibility to organize operation of water production systems at places of its consumption directly.

At the same time, insufficiently high-energy efficiency of a refrigeration cycle limits a wide use of solar water production systems based on AWRU.

As the analysis of studies [23–25] shows, this is largely due to low efficiency of transportation of a refrigerating agent – ammonia – through an AWRU refluxer at operation in a wide range of outside air temperatures. There are unclear losses of ammonia at low outside temperatures, and a significant amount of water enters an AWRU evaporator at elevated temperatures, which leads to a decrease in efficiency of the cooling process.

Solar systems work during daylight hours at various temperatures of atmospheric air. Temperatures are minimum in mornings and evenings and maximum in the middle of the day. The design of AWRU at a certain fixed level (minimum or maximum) of outside air temperatures will lead to a decrease in energy efficiency of both AWRU and water production systems for at least half of daylight hours.

In this regard, it seems expedient to search for energy-saving modes of transportation of a refrigerant agent through an AWRU refluxer in a wide range of outside air temperatures. It is necessary to have an appropriate theoretical substantiation based on model representations of heat and mass transfer processes in an AWRU refluxer to solve the problem.

This determines the relevance of studies on thermal modes of AWRU refluxes aimed at increasing of energy efficiency of AWRU cycles and, accordingly, increasing of productivity of systems for obtaining of water from atmospheric air.

2. Literature review and problem statement

Fig. 1 shows a typical circuit diagram of AWRU, which operates in solar systems for obtaining of water from atmospheric air.

Solar generator 1 and strong WAR receiver 4 are communicating vessels where the liquid WAR is below ∇1 level.

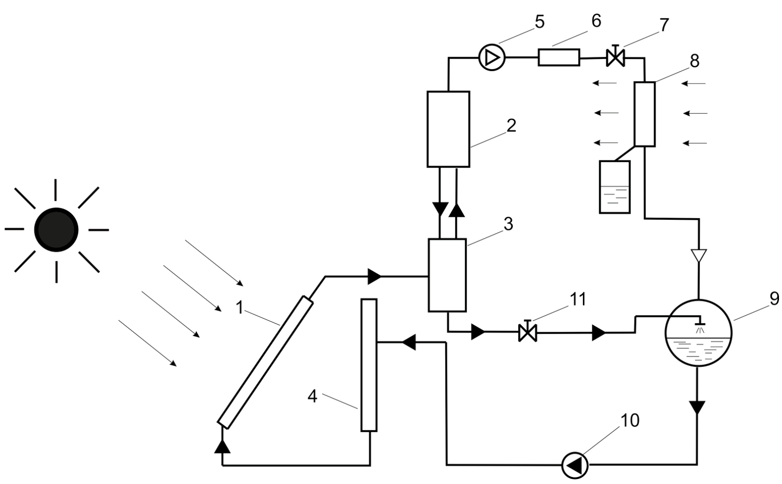

Fig. 1. Schematic diagram of a solar system for obtaining of water from atmospheric air based on AWRU: 1 – solar generator; 2 – refluxer; 3 – weak WAR receiver; 4 – strong WAR receiver; 5 – booster compressor; 6 – condenser of ammonia vapor; 7 – throttle of liquid ammonia; 8 – evaporator; 9 – absorber; 10 – low WAR circulating pump; 11 – throttle of weak WAR

The heat Q_s of solar radiation goes to solar generator 1 and a predominantly low-boiling component, ammonia, is evaporated from the strong WAR. Solar generator 1 is at an angle to the horizon for maximum absorption of solar radiation. The angle of inclination depends on the latitude of the terrain.

Vapor bubbles push depleted in ammonia (weak) WAR into the upper part of the solar collector. The solution flows into the receiver of weak WAR 3, and vapor water-ammonia mixture (WAM) enters vertical refluxer 2. Water vapor condenses predominantly to form reflux in refluxer 2.

Reflux flows into receiver 3. From refluxer 2, booster-compressor 5 compresses purified ammonia vapor to P_k pressure and sends it to condenser 6. P_k pressure creates a necessary temperature drop for the condensation process with the removal of Q_k heat to the environment.

Liquid ammonia arrives to throttle 7 from condenser 6, where its pressure decreases to a pressure level of P_0 in evaporator 8 and absorber 9. In evaporator 8, ammonia boils at a pressure of P_0 and temperature of T_0 with the removal of Q_o heat from a cooling object – a flow of outside air.

The temperature T_0 is maintained below the temperature of the "dew point" $T_{d\text{-}p}$ in the solar system for obtaining water from atmospheric air. Condensate falls out of the air, and the air itself cools. A constant process of absorption of ammonia vapor in absorber 9 by weak WAR that enters absorber 9 from the receiver of weak WAR 3 through throttle valve 11 maintains P_0 pressure in evaporator 8. The strong WAR formed after absorption of ammonia by circulation pump 10 is fed back to strong solution receiver 4 and the cycle goes again.

At present, there is a paradoxical situation – cooling capacity of AWRU decreases at low outside temperatures. This is due to the modes of purification and transportation of ammonia in refluxer 2. A refluxer performs the function of final purification of ammonia vapor from water vapor in the known designs of AWRU [23, 26]. A heat-insulating casing covers the lower part of refluxer 2, and the upper part remains free (not covered with heat-insulation). Thus, purification of ammonia vapor occurs both in the area of installation of a heat-insulation (partially) and in the open sections of the refluxer. In the ideal mode, the ammonia purification process ends at the end of the refluxer lift, reflux flows into receiver 3, booster compressor 5 pumps pure ammonia vapor into condenser 6.

Under real conditions, either non-purified ammonia vapor enters condenser 6, or condensation of ammonia begins at the top of reflux condenser 2 already. Both factors affect cooling capacity of the AWRU evaporator adversely.

Developers [24, 27, 28] propose and perform a thermodynamic analysis of various AWRU schemes based on solar collectors to improve energy performance.

Work [24] pays attention to efficiency of an AWRU generator set in minimization of heat losses from a working surface. The mentioned review presents data on various working bodies of AWRU of a non-pump type.

Paper [27] gives an analysis of AWRU with a solar collector based on two-phase thermal syphons of gravitational type (a heat source is in the lower part and a heat consumer at the top part). Authors consider a solution of lithium bromide and methanol, which can provide the necessary level of cooling temperatures for obtaining of water from atmospheric air, as a working substance of ARU and the temperature of a heating source is 65–75 °C.

Work [28] considers both absorption and adsorption schemes for air conditioning systems based on solar collectors. It proposes the methodology for their thermodynamic calculation and analysis.

At the same time, initial data for calculation and analysis are constant outside air temperatures in all cases [24, 27, 28]. Developers [24, 27, 28] do not give an answer to the question: how does the energy efficiency of this or another AWRU scheme change when the outside temperature changes?

Papers [29–31] present methodologies for thermodynamic calculation and analysis of various cycles of AWRU which makes it possible to define a complex of thermodynamic and energy characteristics of refrigeration cycles in a wide range of operation parameters, including modes of obtaining of water from atmospheric air. At the same time, there is no information on internal processes of heat and mass transfer, which makes impossible to use them for the design of real structures of AWRU and systems for obtaining of water from atmospheric air, in papers [29–31].

Authors of paper [32] consider AWRU operation with the addition of salts in the operation body to reduce a partial pressure of an absorbent (water), which reduces energy consumption by reducing of a proportion of absorbent in vapor. A paper considers lithium nitrate ($LiNO_3$) and sodium thiocyanate (NaSCN) as salts. At the same time, paper [32] does not consider corrosion processes, which can take place in a system with a salt filler.

Authors of work [33] find a similar solution to the problem of increasing of energy characteristics of absorption system cycles, but they use TFE-TEGDME additive, which makes possible to eliminate problems with corrosion of internal working surfaces of heat exchange AWRU.

Works with additions to a working body of AWRU [32, 33] does not consider modes of ammonia transportation through a refluxer; however, they can differ from traditional ones significantly.

Authors of paper [34] presented theoretical grounds for existence of an optimum (the maximum energy efficiency) of a thermodynamic cycle of AWRU for some values of outside air temperatures and temperatures of a heating source. They showed that it is possible to choose only two characteristic temperatures for cooling arbitrarily among three characteristic temperatures of a cooling object, the environment and a heating source, and the third is always a derivative of the choice. At the same time, authors of [34] did not propose ways for choosing energy effective modes at a change in temperature of the outside air, for example, during a day light.

Authors of work [35] propose to intensify processes of natural circulation of a vapor-gas mixture in AWRU using a jet ejector installed at the output of a generator. They assume that forced convection will arise in the internal evaporator-absorber circuit because of ejection and, accordingly, intensity of heat and mass exchange processes during absorption and evaporation will increase substantially, and in general, cooling capacity of a cooling unit will increase also. At the same time, authors of paper [35] do not consider changes in temperature of the outside air, and this change is sufficiently critical for efficient operation of an ejector.

3. The aim and objectives of the study

The objective of this study is to determine energy-efficient modes of refluxers and to develop methods for their realization during operation of AWRU in a wide range of temperatures of the outside air.

It is necessary to solve the following tasks to achieve the objective:

– development of a methodology for modeling heat and mass transfer modes of AWRU refluxers and conduction of analytical studies in a range of operation parameters;

– development of promising ways to increase energy efficiency of AWRU in a wide range of operation temperatures at operation in solar systems for producing of water from atmospheric air.

4. Modeling the heat and mass transfer modes in AWRU refluxer and analysis of the results obtained

The basis of model approaches and representations is a “verification” calculation of some design of a partially insulated refluxer of AWRU with known overall and thermal characteristics:

a) dimensions of a refluxer (thermal insulation): length $L_D(L_{IS})$, internal (d_{in} (d_{in}^{is})) and outer (d_{out} (d_{out}^{is})) diameters;

b) coefficients of thermal conductivity of wall material of a pipe of a refluxer (λ_D) and thermal insulation material (λ_{IS}).

We assume conditions for heat removal to ambient air as known: air temperature – t_{ENV}, a convective heat transfer coefficient – α_K, a degree of blackness of a wall surface of a refluxer – ε. For most models, ε=0.876 (black shiny enamel) [36].

Saturated WAM with a mass consumption – G''_{ent}, temperature – ϑ_{ent}, and a mass concentration – ξ''_{ent}. enters the inlet (to the bottom part) of a partially insulated refluxer [25].

Outer surfaces of a refluxer are in thermal interaction with the ambient air and have a temperature lower than WAM flow temperature. Partial condensation of a predominantly high-boiling component of a mixture, water vapor, occurs in the presence of a difference in temperatures of the wall of a refluxer and a saturated WAM flow.

A difference in water vapor concentrations emerges in WAM between a wall layer and a main flow at water vapor condensation. The difference in concentrations leads to appearance of a diffusion flow of water vapor mass from the main core of a WAM flow to the wall. When a condensate, which consists of water mainly, falls out, the integral equilibrium temperature of a flow ϑ decreases on the inner wall of a refluxer.

A decrease in flow temperature will take place until complete purification of ammonia vapor from water vapor. Subsequently, condensation of pure ammonia vapor at a constant temperature and flow, and walls of the refluxer ($\vartheta_{NH_3}^{S}$) will proceed.

A WAM counterflow along the inner wall of the refluxer is a reflux, which contains water mainly.

At the first stage of modeling, we consider the problem of laminar film condensation of WAM vapor on a vertical wall of a refluxer.

We accept the following generally accepted assumptions in the study [37]:

a) inertia forces, which arise in a condensate film, are negligible compared to forces of viscosity and gravity forces;

b) convective heat transfer in a film, as well as thermal conductivity along it, are insignificant in comparison with thermal conductivity across a film;

c) friction at the boundary between vapor and liquid phases is taken into account by means of tangential stress from the gas side;

d) temperature of an external surface of a condensate film is constant;

e) physical parameters of the reflux do not depend on temperature;

f) surface tension forces on a free surface of a film do not affect a nature of its flow;

g) vapor density of WAM is small comparing with density of reflux;

h) we assume wall temperature constant due to intensive forced cooling.

Taking into account the above model representations, we calculate parameters of a reflux flow and a vapor mixture in a vertical refluxer.

Input data for calculation:

a) cooling capacity is 1,000 W;

b) temperature (pressure) of boiling in an evaporator is 0°C (0.4 MPa);

c) pressure in the generator is 1 MPa;

d) ambient temperature is 32 °C;

e) temperature of a vapor steam mixture at the inlet (outlet) into a refluxer is 100(40) °C;

e) height of refluxer is 1.0 m.

Evaluation of thermal and hydrodynamic parameters of a draining reflux flow based on model concepts [37], taking into account the above assumptions and initial data, showed their insignificant effect on integral modes of refluxer operation and therefore we do not take into account the presence of reflux in the subsequent model representation.

Evaluation of thermal and hydrodynamic parameters of a draining reflux flow [37] showed little effect on operating modes of a refluxer and we do not take into account the presence of reflux in a further model representation.

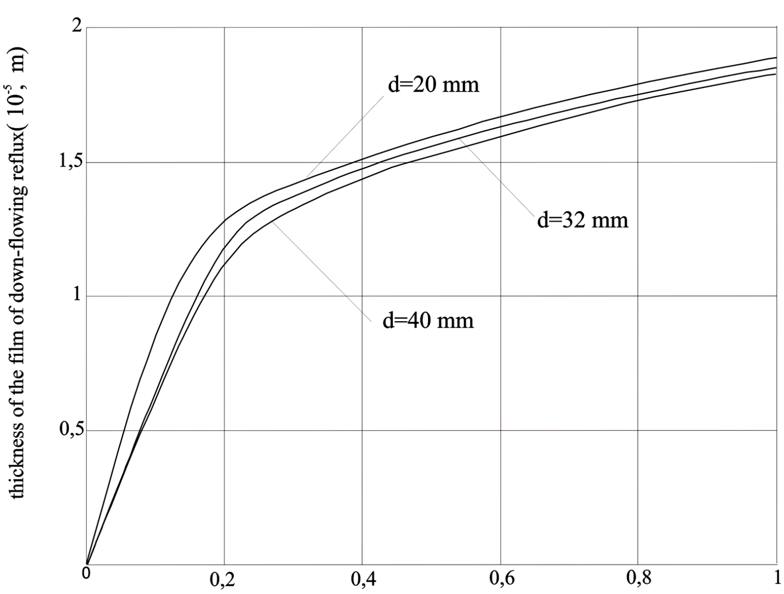

Fig. 2. Change of thickness in the film of down-flowing reflux along the height of a vertical refluxer for various internal diameters (0 — is an upper part)

At the thermal interaction of a vapor flow of WAM and a wall of a refluxer, a partial cooling of a flow (Q_V) occurs and heat (Q_{Ph}) of phase transition (reflux heat) is released (reflux heat Q_D):

$$Q_D = Q_{Ph} - Q_V. \quad (1)$$

Some heat is released into environment (Q_{ENV}), and some – in the form of axial leakages is redistributed along a wall of a refluxer (Q_k), and the total heat flow is always directed to the upper part of a refluxer.

The bases of the mathematical model of AWRU refluxer are heat and mass conservation equations, which for an elementary i-th section of dx refluxer:

$$\Delta Q_{Di}(\tau) = Q_{Di} + Q_{ENVi} + Q_{Ki}, \quad (2)$$

$$G''_{ent} = G''_{\Delta x} + G'_{\Delta x}, \quad (3)$$

where $G''_{\Delta x}$ and $G'_{\Delta x}$ are mass consumptions of a vapor WAM at the exit of Δx section and reflux, respectively, kg/s.

We can present the mathematical model of AWRU refluxer as an object with distributed parameters. We can represent it in general form by the equation of the thermal balance of unit cells of a body in partial derivatives [23]:

$$\frac{\partial}{\partial \tau}(\rho_D c_D \Theta_D) = \frac{\partial}{\partial x}\left(\lambda_D \frac{\partial \Theta_D}{\partial x}\right) + Q(x), \quad (4)$$

where Θ_D is the temperature of a wall of AWRU refluxer, °C; ρ_D, c_D, λ_D is density, heat capacity, coefficient of thermal conductivity of a refluxer wall, respectively, kg/m^3; J/(kg·K); W/(m·K); $Q(x)$ is the integral heat flow from internal heat sources along a height of a refluxer, W; τ is time, s.

The left part of equation (4) is a change in internal energy of a unit cell of a refluxer wall in time. The right part is the sum of:

a) convective heat flows at cooling and reflux of vapor WAM Q_D;

b) heat dissipation into the environment Q_{ENV};

c) conductive heat transfer from neighboring cells (in general, both sides $Q_{K(i+1)}$ and $Q_{K(i-1)}$).

As a result, in general, initial values of temperature in this region $\Theta_{Di}(\tau=0)$ and a total heat flow to it determine temperature $\Theta_{Di}(\tau)$ of each i-th unit cell of a refluxer

$$\Theta_{Di}(\tau) = \Theta_{Di}(\tau=0) + \frac{1}{m_{Di} c_{Di}} \int_0^r \Delta Q_i(\tau) d\tau, \quad (5)$$

where

$$\Delta Q_i(\tau) = \Delta Q_D + \Delta Q_{ENV} + \Delta Q_{K(i+1)} + \Delta Q_{K(i-1)}$$

is the algebraic sum of heat flows into unit cells of a refluxer, W.

At the first stages of modeling of AWRU refluxes [23], we assumed the process of internal mass transfer during reflux ideal. We believed that there is an ideal mixing of a flow of WAM vapor in the radial direction.

Let us consider validity of such a situation using the example of the Ukrainian AWRU model of "Kiev-410" ASh-160 type with a pipe refluxer with an external diameter of 0.016 m and an inner diameter of 0.0146 m [38].

In the operating mode of such AWRU [25], linear velocity of a WAM flow lies in the range of 0.0075–0.0130 m/s. This speed range corresponds to a laminar flow mode with Reynolds numbers of 350–600.

The presence of a laminar stream of a WAM flow does not imply perfect mixing of a flow in the radial direction [25, 39, 40]. In this case, there is a kind of resistance to a transverse flow of matter, and it seems expedient to take into account such a factor to create the most adequate model of a refluxer.

Thus, for cylindrical coordinates (a height of a unit cell dx), the following dependence determines a mass flow (kg/s) [41]:

$$dm_W = \frac{\mu_W}{8314 \cdot T} \cdot \frac{P_{W(r=r_o)} - P_{W(r=r_D)}}{\ln \dfrac{d_{in}}{d_D}} \cdot 2 \cdot \pi \cdot D_{AB} \cdot dx, \qquad (6)$$

where $\mu_W = 18$ kg/kmol is the molar mass of diffusing component-water vapor; $P_{W(r=r_o)}$, $P_{W(r=r_D)}$ are the partial pressures of water vapor in a vapor WAM flow, respectively, in the core of a flow and in a wall zone of a refluxer, Pa; D_{AB} is a coefficient of diffusion of water vapor in a vapor WAM, m^2/s; T is absolute temperature of a diffusing component, K; d_D is the current numerical value of the diameter of a refluxer, m.

We can calculate a diffusion coefficient of binary systems by the classical Fuller, Schlatter and Giddings relationship [42]

$$D_{AB} = \frac{10^{-7} \cdot T_s^{1,75} \cdot \left(\dfrac{M_x + M_{H_2}}{M_x \cdot M_{H_2}} \right)^{\frac{1}{2}}}{P_0 \cdot \left(\left(\sum \nu\right)_X^{\frac{1}{3}} + \left(\sum \nu\right)_{H_2}^{\frac{1}{3}} \right)^2}, \qquad (7)$$

where

$$\left(\sum \nu\right)_X^{\frac{1}{3}} = 14{,}9; \quad \left(\sum \nu\right)_{H_2}^{\frac{1}{3}} = 7{,}07$$

are atomic diffusion volumes, respectively, of ammonia and hydrogen, $[P_0$=bar], $[T_S=K]$.

We made the following assumptions for modeling of thermal modes of AWRU refluxer condenser:

a) temperatures of a wall and a vapor WAM flow are constant in the axial direction along a length of the elementary section of a reflux condenser dx, respectively, $\Theta = \text{const}$ and $\vartheta = \text{const}$;

b) mass concentration of a WAM vapor flow along a length of the elementary section of a refluxer dx varies logarithmically according to a relation (6).

Taking these assumptions into account, the algebraic sum of heat flows into the unit cells of a refluxer takes the following form:

a) the inlet (initial) section:

$$\tilde{\alpha}_D \left(\vartheta_{ent} - \Theta_1\right) = \alpha_{D(1)} \left(\vartheta_{ent} - \Theta_1\right) \cdot \Delta F_{in} +$$
$$+ K_l \left(\Theta_1 - t_{env}\right) \cdot \Delta x + \frac{\lambda}{\Delta x} \left(\Theta_1 - \Theta_2\right) \cdot F_{\sec}; \qquad (8)$$

b) the output (final) section (K):

$$\tilde{\alpha}_D \left(\vartheta_{ent} - \Theta_K\right) = \alpha_{D(K)} \left(\vartheta_K - \Theta_K\right) \cdot \Delta F_{in} +$$
$$+ K_l \left(\Theta_K - t_{env}\right) \cdot \Delta x + \frac{\lambda}{\Delta x} \left(\Theta_{K-1} - \Theta_K\right) \cdot F_{\sec}; \qquad (9)$$

c) the intermediate section i:

$$\tilde{\alpha}_D \left(\vartheta_{ent} - \Theta_i\right) = \alpha_{D(i)} \left(\vartheta_i - \Theta_i\right) \cdot \Delta F_{in} +$$
$$+ K_l \left(\Theta_i - t_{env}\right) \cdot \Delta x + \frac{\lambda}{\Delta x} \left(\Theta_{i-1} - 2\Theta_i + \Theta_{i+1}\right) \cdot F_{\sec}, \qquad (10)$$

where $\tilde{\alpha}_D$ is the conditional heat transfer coefficient at reflux of a vapor WAM flow, which takes into account a phase transition process, W/(m^2×K); ϑ, Θ are temperatures in Δx section of a WAM flow and a wall of a refluxer, respectively, °C; $\alpha_{D(i)}$ is a coefficient of convective heat exchange between a wall of a refluxer and a WAM flow, W/(m^2×K); K_l is a linear coefficient of heat transfer between condensable reflux and ambient air in Δx section, W(m×K); ΔF_{in} and $F_{\sec}$ is the area of an inner wall of Δx section and an axial section of a refluxer pipe, respectively, m^2.

We use the empirical formula [43] to calculate external convective heat transfer of a vertical pipe in the natural convection mode (α_{conv}), and the Stefan-Boltzmann formula to calculate radiative heat transfer (α_{rad}).

We calculate a total coefficient of complex radiant-convective heat transfer as an additive sum of components [43]:

$$\alpha_\Sigma = \alpha_{conv} + \alpha_{rad} =$$
$$= \left[1{,}67 + 3{,}6 \cdot 10^{-3} \cdot \left(\frac{\Theta_D + \Theta_{ENV}}{2} \right) \cdot \left(\Theta_D - \Theta_{ENV}\right)^{\frac{1}{3}} \right] +$$
$$+ 4{,}96692 \cdot 10^{-8} \cdot \left(\frac{\left(273 + \Theta_D\right)^4 - \left(273 + \Theta_{ENV}\right)^4}{\Theta_D - \Theta_{ENV}} \right). \qquad (11)$$

Conditional coefficient of heat transfer at reflux of a vapor WAM flow:

$$\tilde{\alpha}_D = 28{,}5 \cdot \frac{\lambda_{mix}}{d_{in}^{0,2}} \left(\frac{Q_{Ph}}{\Delta F_{in} \cdot r \cdot \eta''_{mix}} \right)^{0,8} \cdot \Pr_{mix}^{0,33}, \qquad (12)$$

where a coefficient of thermal conductivity of a vapor WAM [39]:

$$\lambda_{mix} = \lambda_A \cdot \left(1 - \xi''\right) + \lambda_X \cdot \xi'', \qquad (13)$$

where λ_A, λ_X are coefficients of thermal conductivity of water vapor and ammonia vapor at WAM parameters, respectively, W/(m·K); ξ'' is the mass concentration of ammonia vapor in a saturated vapor WAM; Q_{Ph} is the heat flow at phase transition during reflux process, W:

$$Q_{Ph} = dm_W \cdot r, \qquad (14)$$

where r is the specific heat of phase transition at WAM reflux at P_0 pressure in a system and ξ''; mass concentration of ammonia in a vapor mixture; $\Pr_{mix}$ is the Prandtl number for a vapor WAM at P_0 pressure in a system and ξ'' mass concentration of ammonia in a vapor mixture; η''_{mix} is a coefficient of dynamic viscosity of a vapor WAM [44]:

$$\eta''_{mix} = (0{,}866 + 0{,}0000753 \cdot T_S + 0{,}00001 \cdot T_s^2 +$$
$$+ 0{,}09163 \cdot \xi'' + 0{,}00952 \cdot T_s \cdot \xi'' -$$
$$- 0{,}000106 \cdot T_s^2 \cdot \xi'' + 0{,}172 \cdot \left(\xi''\right)^2 -$$
$$- 0{,}0079 \cdot T_S \cdot \left(\xi''\right)^2 + 0{,}000104 \cdot T_S^2 \cdot \left(\xi''\right)^2) \cdot 10^{-5}, \text{ Pa·s}; \qquad (15)$$

where T_S is absolute temperature of saturation of a vapor WAM, K.

We considered a case of convective heat transfer in a confined space to study the hypothesis presented above on a prospect of regulation of heat dissipation conditions from an outer surface of a refluxer when temperature of the air in a room changed.

Features of heat transfer in a confined space are well studied for air cylindrical layers with a thickness δ≥10 mm in ref. [43], authors recommend the following relation for a coefficient of convective heat transfer

$$\alpha = 0{,}91 \cdot \frac{\delta}{d_1 \ln\left({d_s}/{d_1} \right)} \cdot \sqrt[4]{\frac{\Delta t}{\delta}}, \tag{16}$$

where d_s, d_l are diameters of the cylindrical air interlayer, the smaller and larger, respectively, m; Δt is the temperature difference on walls of the air interlayer, °C.

According to the same recommendations [43], we used air interlayers of less than 5 mm to create almost adiabatic conditions, in which natural convection is absent and the process of heat transfer is conducted by conductor means.

A coefficient of convective heat transfer between a wall of a refluxer and a laminar WAM flow, which satisfies the relation

$$\mathrm{Re} \cdot \mathrm{Pr} \cdot \frac{d}{l} \succ 10, \tag{17}$$

we calculated it based on dependence [41]:

$$Nu = 1{,}86 \cdot (\mathrm{Re} \cdot \mathrm{Pr})^{0{,}33} \cdot \left(\frac{d}{l} \right)^{0{,}33} \cdot \left(\frac{\mu_b}{\mu_s} \right)^{0{,}14}, \tag{18}$$

where $\left(\frac{\mu_b}{\mu_s} \right)^{0{,}14}$ is an empirical correction factor, taking into account an effect of temperature on viscosity of liquids. For gases and vapors, it is almost equal to one.

The preliminary numerical analysis showed possibility of using of a relation (18) in a range of flow parameters, composition and temperatures of WAM in typical AWRU of household execution.

When WAM vapor flow moves along a length of a refluxer, not only loss of a high boiling component occurs, but also a decrease in flow temperature happens due to convective heat exchange with a wall of a pipe.

It is possible to obtain a dependence for temperature of a vapor flow at its outlet ϑ'', from heat balance and mass balance equations in some Δx section of a refluxer, if we know temperature at the input of a section ϑ', heat exchange conditions and flow parameters:

$$\vartheta'' = \left[\frac{G''_{\Delta x} \cdot c_p \cdot \vartheta' - \alpha \cdot \Delta F_{in} \cdot (\vartheta/2 - \Theta)}{\alpha \cdot \Delta F_{in}/2 + c_p \cdot (G''_{\Delta x} - dm_W)} \right], \tag{19}$$

where c_p is the mass isobaric heat capacity of a vapor WAM in Δx section, J/(kg·K); α is a convective heat transfer coefficient between a WAM vapor flow and a refluxer wall calculated by formula (16), W/(m^2·K).

We realized model concepts of thermal and hydraulic modes of an AWRU refluxer in the Simulink simulation environment of the MATLAB program.

According to the above algorithm, we performed calculation of temperature fields for a refluxer with a diameter of 16×1.4 mm. Pipe material is structural steel (d_{in}= =45 W/(m×K)). Thermal insulation material of a casing is a fiberglass cloth (λ_{ti}=0.056 W/(m×K)).

A WAM flow with a temperature varying in dependence on the ambient air temperature goes to the inlet of AWRU refluxer.

Taking into account the same results, we set mass flow rates at the inlet of a refluxer at thermal loads: 70; 100 and 150 W. Further, we modelled thermal modes of a refluxer of a size of 0.20 m with 20 elementary sections the size of 0.01 m.

We considered two options of refluxer operation: without heat insulation of a lifting section and operation under ideal adiabatic conditions.

In the first case, the modeling went at temperatures: 10; 17; 25 and 32 °C, and in the second case, at 10 and 25 °C. The lower limit of the modeling range (10°C) corresponds to the international class of performance of a household appliance SN^*, and the upper limit – to the moderate climate [45].

5. Discussion of results of modeling the processes of heat and mass exchange of AWRU refluxers

Analysis of modeling results made it possible to reveal a significant calculation difference of temperatures between a WAM flow and a wall of a AWRU refluxer.

Table 1 shows the data obtained.

Table 1

Calculation difference of temperatures difference between a WAM flow and a wall of an AWRU refluxer

Lifting area refluxer without thermal insulation				
Thermal loading of AWRU solar generator, W	Ambient air temperature, °C			
	10	17	25	32
70	19	18	15	16
100	29	28	28	24
150	36	34	32	29
Lifting area refluxer with thermal insulation coating				
Thermal loading of AWRU solar generator, W	Ambient air temperature, °C			
	10	–	25	–
70	13	–	16	-
100	23	–	23	–
150	28	–	28	–

An analysis of the results obtained showed that the minimum temperature difference between a WAM flow and a wall of an AWRU refluxer takes place under adiabatic operation conditions of a refluxer when there is no environmental effect.

The obtained results confirmed the well-known fact [46, 47] about inoperability of AWRU of a standard structure under conditions of low ambient temperature (10 °C) and minimum thermal loads on a thermosyphon generator (70 W).

When a thermal load increases, a launch of AWRU starts (at 100 and 150 W). At 100 W of thermal load, the temperature at the outlet of a refluxer is 64 °C. At a pressure in the system of 2.0 MPa, mass concentration of a vapor WAM is 0.997, i. e., almost pure ammonia is moving. At a thermal load of 150 W, the temperature at the outlet of a refluxer is 73 °C, and mass concentration is 0.994.

Starting with the ambient air temperature, which corresponds to 17 °C, even at a minimum load, almost pure ammonia vapor flows into a condenser. At the same time, there is a WAM flow with the mass concentration of 0.990 at half a length of a refluxer (in the upper part).

At 25 °C and a thermal load of 150 W, a vapor front advances to the end of a refluxer and a regular WAM cleaning mode is realized.

The same effect takes place with a thermal load of 70 W, but at an ambient air temperature of 32 °C already.

Under adiabatic conditions, passing of a WAM BAC flow at outdoor air temperature of 10 °C takes place even with a minimum thermal load of 70 W. We can determine this mode by the wall temperature of a refluxer, which in some tests [38, 46, 47] is about 65 °C. The temperature corresponds to the minimum temperature in a refrigerating chamber and temperature at the end of boiling of WAM not exceeding 170 °C. The limitation on boiling temperature relates to an exponential increase in intensity of corrosion processes in a heat input zone of a solar generator [30].

With a thermal load of 100 W in the final section of a refluxer, temperature reaches 88 °C, which corresponds to a mass concentration of ammonia in WAM flow – 0.985, and at a thermal load of 150 W, flow temperature is 93 °C and a mass concentration is 0.970.

At an air temperature of 25 °C and a thermal load of 70 W, temperature of a WAM flow at the outlet of a refluxer is 81 °C, and the mass concentration of ammonia in WAM flow is 0.996, at a thermal load of 100 W and 150 W – temperatures are, respectively, 88 °C and 93 °C, and mass concentrations are 0.985 and 0.977.

We verified the above modeling results in the framework of experimental studies. We got convergence sufficient for engineering calculations, which does not exceed 5 °C.

We proposed an original refluxer casing with variable thermal resistance for energy-saving operation of AWRU in a wide range of outside air temperatures (10–32 °C) (Fig. 3, 4).

The casing is installed in the upper part of an AWRU refluxer and is capable of changing of heat exchange conditions on the outer surface of a refluxer.

A base of the principle of operation of the new heat-insulating casing is an effect of changing the volume of bellows 24 at internal pressure changes.

The internal pressure in bellows 24 depends on saturation temperature of liquid partially filling bellows 24, which in turn is determined by the temperature of the outside air.

The additional casing on refluxer 3 has the form of a coaxial cylinder with an air cavity in the intertubular space.

There are two modes of operation of refluxer casing 3:

a) "adiabatic" mode with a fully enclosed intertubular space;

b) "open" mode with the access of external air to the internal cavity.

Here the term "adiabatic" implies not a complete absence of heat exchange of a wall of the refluxer 3 with the surrounding medium, but only the possible minimum.

The intertubular gap should be made taking into account recommendations [43] – if an air gap is equal to or less than 5 mm, convection is completely absent and the process of heat transfer goes in the thermal conductivity mode.

The thermal resistance of the air gap of 5 mm thick at a temperature of 70 °C is 0.17 (K·m^2)/W. The thermal resistance of the open section of the refluxer 3, taking into account convective and radiant components of the total heat transfer coefficient, will lie in the range of 0.067–0.110 (K·m^2)/W. The total thermal resistance to the process of heat transfer through the additional casing of the refluxer 3 will be not less than 0.40 (K·m^2)/W, that is, almost 4 times more than in the case of an open section.

Estimating calculations showed that a heat flow in the section of the refluxer of 0.20 m length with a temperature difference between a WAM BAC flow and the outside air of 60 °C will be 0.15 W, that is, one can speak about almost adiabatic mode in the area of installation of the additional casing at a typical thermal load of an AWRU refluxer of 12–15 W.

The presence of common axis 17 ensures movement of casing flaps 15 and 16. Fig. 3, *a* shows the casing in the "adiabatic" mode of operation, and in Fig. 3, *b* – in the "open" mode.

The "adiabatic" mode of operation of the bellows is realized with the minimum temperature of the outside air.

When the temperature of the outside air increases, so does saturation temperature of low-boiling liquid partially filling bellows 24 and, correspondingly, saturation pressure. The growth of the internal pressure causes movement of the end walls of bellows 24, which lead to movement of flaps 15 and 16 of the additional casing through axes 22 and 23.

When designing bellows 24, with respect to rigidity, dimensions and type of low-boiling liquid, it is necessary to provide for both the "adiabatic" mode of a fully enclosed casing and the maximum opening mode for operation under conditions of high ambient air temperatures.

At intermediate values of outside air, bellows 24 must provide a partial air passage to the inner volume of a casing.

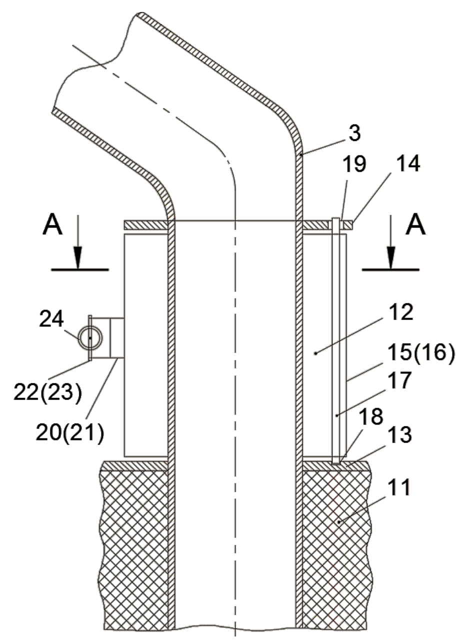

Fig. 3. Scheme of the lifting section of the refluxer with an additional casing: 3 – lifting part of the refluxer; 11 – stationary heat-insulating casing of a generator unit; 13 and 14 – washers for fastening the additional casing; 15 and 16 – flaps of the additional casing; 17 – axis of fastening of an additional casing; 18 – slot for the axis; 19 – hole for the axis; 20 and 21 – semi-oval plates; 22 and 23 – fastening axes; 24 – bellows

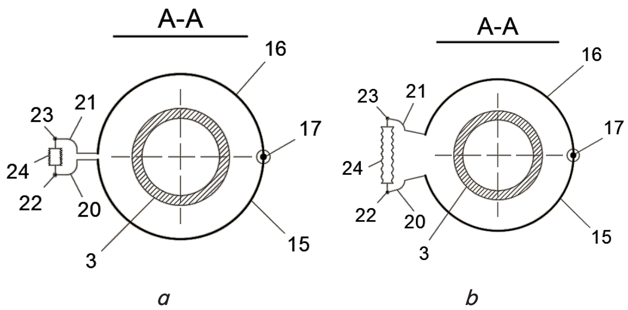

Fig. 4. Cross-sections of the lift section of the refluxer with an additional casing: *a* — closed position of the additional casing with minimum outside air temperatures (adiabatic approximation); *b* — open position of the additional casing at elevated outdoor temperatures; 3 — lifting part of the refluxer; 15 and 16 — flaps of additional casing; 17 — axis of fastening of an additional casing; 20 and 21 — semi-oval plates; 22 and 23 — fastening axes; 24 — bellows

Thus, the proposed design of the AVHA reflux condenser with a heat-insulated casing of a variable design will allow, in an automatic mode (without user participation), to ensure the most efficient modes of ammonia transportation to an evaporator and the corresponding growth of both the cooling capacity and systems for obtaining of water from the ambient air throughout the daylight hours.

6. Discussion of results of modeling the heat and mass exchange processes in AWRU refluxer

The presented model representations of processes in an AWRU refluxer (equations (1) to (6) and (8) to (10)) take into account actual physical features of heat exchange of a refluxer with external air in the temperature range from 10 °C to 32 °C and presence of the diffusion process at condensation of water vapor from a vapor WAM flow. We performed modeling for both stationary and transient modes of operation.

We found a significant calculation difference of temperatures (from 10 to 36 °C) between a WAM flow and a reflux wall in all operation modes. We should consider this result when adjusting a thermal load in an AWRU solar generator by a temperature sensor installed on a wall of a refluxer.

Studies showed that in order to reduce losses during transportation of ammonia vapor to an AWRU evaporator, thermal insulation in a lifting part of a refluxer is suitable at low ambient air temperatures (10–17 °C) only.

The analysis showed also that installation of thermal insulation throughout the whole lifting section of an AWRU refluxer contributes not only to reduction in losses during transportation of ammonia, but also to a decrease in time of the output of a refrigeration unit to the operation mode.

The conducted experimental researches showed convergence of modeling results and real characteristics acceptable for engineering design, for example, temperature fields of a wall of a refluxer – no more than 5 °C.

Taking into account the results of the modeling, we proposed the original design of an AWRU refluxer, which is able to change heat exchange conditions when the ambient air temperature changes automatically without user intervention.

The obtained results make possible to increase energy efficiency of AWRU significantly (up to 36 %) and, accordingly, proportionally to increase productivity of a solar system for obtaining of water from atmospheric air under temperature conditions that are changing during daylight hours.

The found regularities can find application in the design of domestic and commercial absorption refrigerators, which operate according to *SN** class in the range of air temperatures in the room of 10–32 °C [26]. A preliminary estimation of the method of changing of heat exchange conditions in the zone of a refluxer of household Ukrainian device Ash-160, type “Kiev”, showed an increase in energy efficiency from 18 to 36 %.

We should note that the conducted studies do not take into account the orientation of elements of AWRU in space, and the key point is maximum sun orientation at creation of high-performance solar water producing systems [48]. With a daily movement of the sun over the sky, one should also carry out movement of an AWRU refluxer in space. In this case, appropriate modeling is necessary for not only heat and mass exchange, but also for modes of reflux flow along inclined surfaces.

7. Conclusions

1. We developed a technique for modeling of heat and mass exchange modes of AWRU refluxers in the composition of solar water producing systems from atmospheric air. A distinctive feature of the presented technique is accounting for resistance to mass transfer in the process of diffusion of water vapor from the center of a flow to a wall of a refluxer. We confirmed the reliability of the modeling methodology in the course of experimental studies of a typical household absorption refrigeration unit. The method made possible analytical determination of numerical values of a temperature difference between a vapor WAM flow and a refluxer wall. We can use modeling results to design systems for automatic control of energy-saving modes of AWRU at operation in a wide range of outdoor temperatures.

2. We developed an approach to creation of systems of energy-saving management of AWRU in a wide range of outdoor air temperatures (10–32 °C) at operation in solar systems for obtaining of water from atmospheric air. The base of the approach is changing of heat exchange conditions at the outlet of a refluxer in the automatic control mode using a bellows filled with a low-boiling liquid.

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