MODELING OF THERMAL MODES OF THE REFLUX CONDENSER OF THE ABSORPTION REFRIGERATION UNIT

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Abstract

Currently, developers of modern refrigeration equipment, in accordance with the plans of the UN, are moving to natural refrigerants (hydrocarbons, carbon dioxide and ammonia) that do not have an adverse technological impact on the ecosystem of the planet. In domestic refrigeration technology, one of the options is absorption refrigeration units, the working body of which is an aqueous ammonia mixture with the hydrogen addition. Having a number of unique advantages over compression analogs, absorption systems are characterized by lower energy characteristics.

As the analysis shows, the maximum thermodynamic losses in the absorption aggregates are concentrated in the generating unit when the ammonia is evaporated, it is purified from water vapor and transported to the evaporator. In this connection, the mathematical modeling of the thermal regimes of the reflux condenser is performed, which is responsible for purification and transportation of ammonia vapor.

Modeling is carried out on standard designs of absorption refrigeration units taking into account reasonable assumptions and results of own experimental researches. A cellular model is used. Stationary operating modes are modeled due to the high thermal inertia of the processes in the reflux condenser.

As a result, the perspective of the thermal insulation installation throughout the reflux section is shown, which makes it possible to increase the energy efficiency by 17...22 %.

Keywords: thermal modes; energy efficiency; absorption refrigerator; reflux condenser; thermal insulation.

1. Introduction

Household absorption refrigerators (AR) based on absorption refrigeration units (ARU) are popular with consumers due to a wide range of operating temperatures – from –24...–18 °C to 12 °C, which allows long-term storage of various food products [1].

The ARU working fluid is a water-ammonia solution with the addition of an inert gas (hydrogen, helium or their mixture) that is environmentally safe, i. e. has zero values of the ozone-depleting potential and potential of the "greenhouse" effect [2].

ARUs have a number of unique qualities [2–4]:

a) Noiselessness, high reliability and long service life, absence of vibration, magnetic and electric fields during operation;

b) the possibility of using several different energy sources in one device, both electric and non-electric;

c) the ability to work with low-quality sources of electrical energy with a voltage in the network of up to 160 V.

Their advantages are also the minimal cost in comparison with existing types of household refrigeration equipment [5].

At the same time, ARUs have increased power consumption in comparison with similar compression models [6–10]. In our opinion, this situation is due not only to the imperfection of their refrigerating cycle, but also to the lack of appropriate scientific and engineering developments.

The relatively low ARUs energy efficiency causes a narrow area of their use, mainly as mini-refrigerators, and a small share in the domestic refrigeration market.

When searching for energy-saving ARU regimes, special attention must be paid to the efficiency of ammonia transport to the evaporator, especially under operating conditions at low outside air temperatures. At present, there is a paradoxical situation – at low temperatures in the room, ARU energy consumption increases. This position is determined by the modes of cleaning and transportation of ammonia in the elevating part of the reflux condenser. In the well-known ARUs designs [11–13], which are designed for operation in the positional control mode, the lifting section of the reflux condenser performs the function of final purification of the ammonia vapor from the water vapor. The geometric dimensions of the reflux section are determined not by calculation, but from the experience of practical development and the layout of the working elements in the concrete ARU. For example, in all modern ARU designs, the inner diameter of the lifting section does not exceed 18 mm. This limitation is associated with the problems of completely removing of the vapor-gas mixture from the reflux condenser and condenser into the absorber and evaporator during the starting period. With a larger pipe diameter, the front of the gas-vapor mixture is degraded, some of the inert gas remains in the reflux and condensation zones and significantly reduces the intensity of these processes [5, 11].

The length of the reflux section is determined by the location of the condenser and the ARU rectifier. In single-chamber absorption refrigerators this is approximately 0.8 m, in double chamber – about 1.25 m.

The lower part of the lift section of the reflux condenser in the generator zone is closed by a common heat-insulating jacket. The thickness of the heat-insulating jacket on the lift section of the reflux condenser is not calculated, but is actually determined by the arrangement of the elements of the generator assembly having a complex spatial configuration. The upper part of the lift section of the reflux condenser remains free (not closed with thermal insulation).

Thus, ammonia vapor purification after the rectifier occurs both in the zone of installation of the thermal insulation (partially) and in the open sections of the lifting reflux condenser. In the ideal mode, at the end of the lifting section of reflux condenser, the ammonia purification process ends, the reflux flows into the rectifier, and the pure ammonia vapor enters the condenser.

At the same time, with the existing approach to the design of the reflux section, ideal modes of its operation are practically unrealizable. This is due to a non-optimal choice of the dimensions of the open and heat-insulated zone and thickness of the heat-insulating jacket in the lower part of the reflux condenser.

As a result of this approach, either the crude ammonia vapor enters the condenser, or the condensation of ammonia begins already at the top of the lift section of the reflux condenser. Both these factors adversely affect the refrigerating capacity of the evaporator and ARU energy efficiency.

The first factor is easily eliminated by increasing the surface of the heat exchanger, for example, by installing ribs [13] or by increasing the length of the tube due to the zigzag bending of the pipe in the vertical plane [13].

The second factor can be eliminated by increasing the thickness of the thermal insulation coating of the lower part of the reflux condenser, but it will have an adverse effect on ARU operation in a wide range of ambient temperatures – to cause condensation of ammonia vapors.

In such situation, development of a mathematical model for the operating conditions of the lifting section of the ARU reflux condenser is becoming topical.

2. Mathematical model of thermal regimes of the ARU reflux condenser

The functional diagram of the lift section operation of the ARU reflux condenser is shown in **Fig. 1.** In general, based on general physical concepts, two modes of the reflux condenser operation can be distinguished:

a) the mode of cleaning the vapor mixture, when the wall temperature (t_S) exceeds the condensation temperature of ammonia ($\vartheta_{NH_3}^S$);

b) partial condensation of pure ammonia vapors $\left(t_S \leq \vartheta_{NH_3}^S \right)$.

With the ideal operating mode of the reflux condenser, the "a" mode is realized.

The initial data for modeling are:

a) the dimensions of the lifting section of the reflux condenser (length L_D, inner d_{in} and outer d_{out} diameters of the pipe);

b) the coefficients of thermal conductivity of the wall material of the reflux condenser pipe (λ_S) and the warm insulation material (λ_{ti});

c) the mass flow rate of ammonia vapor at the outlet of the lifting section of the reflux condenser G''_{ex};

d) the parameters of the vapor mixture flow at the inlet of the lifting section of the reflux condenser (temperature, mass concentration).
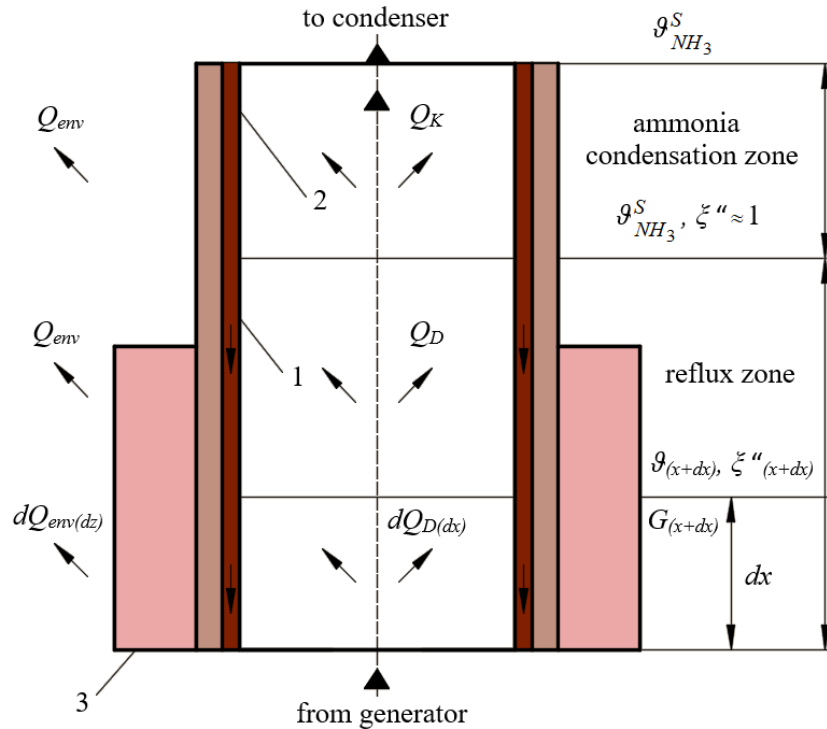

Fig. 1. Diagram of heat and mass flows in the reflux condenser

At the inlet of the lifting section of the reflux condenser comes a vapor water-ammonia mixture with the parameters ϑ_{ent}, ξ''_{ent}, G''_{ent}.

At a partial reflux at the initial (lower) section due to the difference in wall and flow temperatures, ammonia concentration in it increases. The equilibrium temperature of the vapor mixture flow (ϑ) decreases, and at the next higher section of the reflux condenser, the wall temperature will be lower than at the initial stage.

By changing the wall temperature along the height of the reflux condenser, there will be axial heat transfers along the pipe section (Q_{ax}).

Chilled reflux countercurrent flows of the vapor mixture along the inner wall of the reflux condenser. The heat of reflux (Q_D) after reflux heating (Q_F) is discharged to the environment (Q_{env}) both in the installation area of the thermal insulation jacket and from the free surface of the pipe.

The mathematical model is based on the equations of heat and mass conservation, which for the part of the reflux condenser of height Δx have the form

$$\Delta Q_{D(dx)} = \Delta Q_{env(dx)} + \Delta Q_F + \Delta Q_{ax}, \tag{1}$$

$$G''_{ent} = G''_{dx} + G'_{dx}, \tag{2}$$

where G''_{dx} and G'_{d} are the mass flow rate of the vapor mixture at the outlet of the Δx section and the reflux produced in this section, respectively, kg/s.

For the initial section, the axial heat transfers will enter the left side of equation (1) with a plus sign, and for the final section with a minus sign. In the intermediate sections, the contribution of the axial flows represents the difference between the inflow of heat from the lower section and the outflow of heat to the upper section.

When writing the components of equation (1), the following assumptions are applied:

a) thermal resistance of the reflux film is insignificant and can be neglected when modeling the thermal regimes of a reflux condenser [14];

b) temperatures of the reflux and the walls of the reflux condenser are equal and are constant at the portion Δx;

c) the wall temperature of the reflux condenser at the section Δx is constant in the axial and radial directions.

Taking these assumptions into account, equation (1) takes the form:

a) the inlet (initial) section 1:

$$\alpha_{D(1)}\left(\vartheta_{ent} - t_1\right)\cdot \Delta F_{in} = K_1\left(t_1 - t_{env}\right)\cdot \Delta x +$$
$$+G_{F(1)}\cdot C_{F(1)}\left(t_1 - t_2\right)+\frac{\lambda}{\Delta x}\left(t_1 - t_2\right)\cdot F_{sec}; \tag{3}$$

b) outlet (final) section K:

$$\alpha_{D(K)}\left(\vartheta_K - t_K\right)\cdot \Delta F_{in} = K_1\left(t_K - t_{env}\right)\cdot \Delta x +\frac{\lambda}{\Delta x}\left(t_{K-1} - t_K\right)\cdot F_{sec}; \tag{4}$$

c) intermediate section i (i=2...K−1):

$$\alpha_{D(i)}\left(\vartheta_i - t_i\right)\cdot \Delta F_{in} = K_1\left(t_i - t_{env}\right)\cdot \Delta x +$$
$$+G_{F(i)}\cdot C_{F(i)}\left(t_i - t_{i+1}\right)+\frac{\lambda}{\Delta x}\left(t_{i-1} - 2t_i + t_{i+1}\right)\cdot F_{sec}; \tag{5}$$

where $\alpha_{D(i)}$ – coefficient of heat exchange during condensation of water-ammonia vapor flow, W/(m²×K);

ϑ, t – temperatures in the section Δx of the vapor flow and the wall (reflux), respectively, °C;

K_1 – linear coefficient of heat transfer between the condensing vapor flow and the environment in the area, W/(m×K);

ΔF_{in} and F_{sec} – the area of the inner wall of the section Δx and the axial section of the reflux tube, respectively, m²;

G_F, C_F – mass flow rate and mass heat capacity of incoming reflux, respectively, kg/s and J (kg×K).

To find the unknown parameters of the vapor flow at the exit of the section Δx ($\vartheta_{(x+dx)}$, $G_{(x+dx)}$ and $\xi''_{(x+dx)}$) to equations (3)–(5), the equations of material balance along the lifting section of the reflux condenser are added.

In view of the fact that the pure ammonia stream G'' must leave the reflux condenser and reflux containing both water G^D_w and ammonia G^D_a must return to a rectifier, the expressions for the equilibrium mass concentration can be represented as:

a) for a vapor mixture

$$\xi'' = \frac{G'' + G^D_a}{G'' + G^D_a + G^D_w}; \tag{6}$$

b) for a liquid mixture (reflux)

$$\xi' = \frac{G^D_a}{G^D_a + G^D_w}. \tag{7}$$

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The equilibrium concentrations are determined by the temperature ϑ_{in} and the total pressure in the system.

Solving the system (6), (7) with respect to G_a^D and G_w^D, let's find

$$G_a^D = G'' \cdot \frac{1 - \xi''}{\xi' - 1 + \xi'' \left(\dfrac{1 - \xi'}{\xi'} \right)}, \qquad (8)$$

$$G_w^D = G_a^D \cdot \frac{1 - \xi'}{\xi'}. \qquad (9)$$

From equations (8) and (9), one can determine the total flow of reflux flowing into the rectifier

$$G_F = G_a^D + G_w^D. \qquad (10)$$

Thus, at a given input flow concentration of the vapor mixture and the flow rate of purified ammonia at the outlet, it is possible to unambiguously determine the flow rate of the vapor mixture at the inlet to the lifting section of the reflux condenser

$$G_{in}'' = G'' + G_F. \qquad (11)$$

At Δx, reflux heat can also be expressed in terms of the heat of the phase transition, r:

$$\Delta Q_{D(\Delta x)} = G_{F(\Delta x)} \cdot r. \qquad (12)$$

The mass concentration of reflux is determined from the wall temperature of the reflux condenser, which can be taken as equal to the flow temperature in the first approximation.

Let's find the fraction of ammonia and water in the reflux stream on the section Δx

$$G_{a(\Delta x)}^D = G_{F(\Delta x)} \cdot \xi'_{(\Delta x)}, \qquad (13)$$

$$G_{w(\Delta x)}^D = G_{F(\Delta x)} \cdot \left(1 - \xi'_{(\Delta x)} \right). \qquad (14)$$

Taking into account the liquefied reflux, let's write the equation for the mass concentration of the flow at the outlet of the section Δx

$$\xi''_{ex(\Delta x)} = \frac{G'' + G_a^D - G_{a(\Delta x)}^D}{G'' + G_w^D - G_{w(\Delta x)}^D + G_w^D - G_{w(\Delta x)}^D}. \qquad (15)$$

The equilibrium temperature of the flow of the vapor mixture at the outlet of the section Δx is determined from the concentration value $\xi''_{ex(\Delta x)}$.

The presented model allows to carry out both design and verification calculation of the lifting section of the ARU reflux condenser.

Minimization of the inevitable losses of ammonia when it is transported to the condenser when operating in conditions of low and comfortable air temperatures in the room is proposed to be carried out on the basis of the following considerations.

In temperate latitudes ARUs in "hard" mode are operated 2−3 months in the year. The rest of its time passes at room air temperatures from 18 to 25 °C [15].

It is assumed that the optimal approach will be to calculate the heat-insulating jacket of the lifting section of the reflux condenser from the condition of complete purification of ammonia vapor at room temperature of 32 °C. Such approach to design will allow ARUs operation under severe conditions and minimize inevitable losses during transportation in conditions of moderate and low air temperatures in the room.

As a result of the design calculation of the lifting section of the reflux condenser, taking into account the initial data, it is necessary to determine the diameter (d_{ti}) and height (h_{ti}) of the heat-insulating jacket, at which the complete cleaning of ammonia vapor ($\zeta''=1$) is ensured at room temperature of 32 °C.

At the first stage of the design calculation, let's determine the diameter of the heat-insulating jacket on the basis of the approximate relationship

$$G''_{ent} \cdot i''_{ent} - G'' \cdot i'' = \frac{\overline{\vartheta} - t_{env}}{R_{ti}}, \tag{16}$$

where i''_{ent} and i'' – the specific enthalpy of the vapor water-ammonia mixture at the inlet of the lift section of the reflux condenser and the ammonia vapor at the outlet, respectively, J/kg;

$\overline{\vartheta}$ – average temperature of the vapor mixture in the lifting part of the reflux condenser, °C;

$$\overline{\vartheta} = \frac{\vartheta_{ent} + \vartheta^S_{NH_3}}{2};$$

t_{env} – air temperature in the room (assume 25 °C);

R_{ti} – total thermal resistance of thermal insulation, K/W.

As a result of the calculation, let's obtain the distribution of temperatures and concentrations along the height of the lift section of the reflux condenser and perform an evaluation of the parameters of the vapor flow at the outlet. In the case that a complete purification of ammonia is not achieved, let's reduce the thickness of the thermal insulation, and in the case that the cleaning occurs at some distance to the outlet – let's increase. Calculation is finished when within the specified accuracy, a complete purification of the ammonia vapor is achieved at the outlet of the lifting section of the reflux condenser. Let's fix the thickness of the heat-insulating jacket.

At the second stage, at a comfortable room temperature of 22 °C, let's perform a verification calculation of the heat-insulating jacket. Let's determine the inevitable losses of ammonia during transportation in two cases − with the traditional design with a partial installation of thermal insulation in the lifting section of the reflux condenser and with its complete thermal insulation.

Thus, the developed mathematical model allows not only to determine the optimal design dimensions of the heat-insulating jacket in the lifting of the reflux condenser, but also to determine the inevitable losses of ammonia during transportation at moderate and low temperatures, including under different heat loads on the boiler-generator.

The system of equations (3)–(5)–(12)–(15) is solved using an iterative method based on the direct determination of the temperature at each node from the difference equation of energy balances [13].

For a section of a cylindrical pipe with a length Δx, taking into account the assumptions, three types of equations can be written – for the inlet and outlet sections and for the intermediate sections (**Fig. 2**).

For the case "a", the energy balance equation

$$\alpha_D \pi d_{in} \cdot \left(\vartheta_{ent} - T_0\right)\Delta x + G_{F(1)} \cdot C_{F(1)} \cdot \left(T_1 - T_0\right) +$$
$$+ \frac{\lambda}{\Delta x} F_{sec}\left(T_1 - T_0\right) + \alpha_k \pi d_{out}\left(T_{env} - T_0\right)\Delta x = 0. \tag{17}$$

After the designation:

$$A = \alpha_D \pi d_{in} \cdot \Delta x; \ \ B = G_{F(1)} \cdot C_{F(1)}, \ \ C = \frac{\lambda}{\Delta x} F_{sec}; \ \ D = \alpha_k \pi d_{out} \cdot \Delta x,$$

equation (17) takes the form

$$\frac{A\,\vartheta_{ent}}{\left(A + B + C + D\right)} + \frac{B + C}{\left(A + B + C + D\right)} \cdot T_1 + \frac{D}{\left(A + B + C + D\right)} \cdot T_{env} = T_0. \tag{18}$$
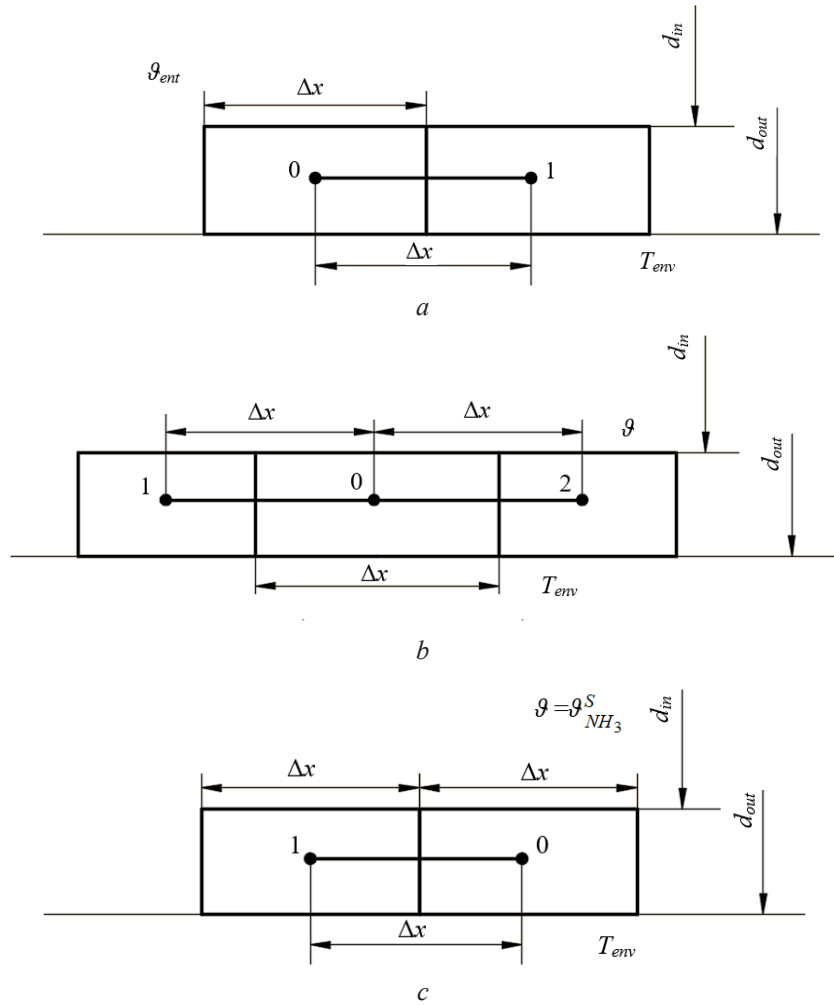
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Fig. 2. Thermal diagrams of typical units of the lifting section of the ARU reflux condenser without heat-insulating jacket: *a* – inlet section; b – intermediate section; *c* – outlet section

Denoting the constants in Eq. (18) in an appropriate manner, let's obtain the ratio for the temperature of the node "0"

$$T_0 = a_0 + a_1 T_1 + a_2 T_{env}. \tag{19}$$

For the case «b», the energy balance equation

$$\alpha_D \pi d_{in} \cdot (\vartheta - T_0) \Delta x + G_{F(2)} \cdot C_{F(2)} \cdot (T_2 - T_0) +$$

$$+ \frac{\lambda}{\Delta x} F_{sec} (T_1 - T_0) + \frac{\lambda}{\Delta x} F_{sec} (T_2 - T_0) + \alpha_k \pi d_{in} (T_{env} - T_0) \Delta x = 0. \tag{20}$$

After the designation:

$$A = \alpha_D \pi d_{in} \cdot \Delta x; \ B = G_{F(2)} \cdot C_{F(2)}; \ C = \frac{\lambda}{\Delta x} F_{sec}; \ D = \alpha_k \pi d_{in} \cdot \Delta x,$$

equation (20) takes the form

$$\frac{A}{(A+B+2C+D)} \cdot \vartheta + \frac{C}{(A+B+2C+D)} \cdot T_1 + \frac{B+C}{(A+B+2C+D)} \cdot T_2 +$$

$$+ \frac{D}{(A+B+2C+D)} \cdot T_{env} = T_0. \tag{21}$$

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Applying similar designation, let's obtain the relation for the temperature of the node "0"

$$T_0 = b_0 + b_1 \vartheta + b_2 T_1 + b_3 T_2. \qquad (22)$$

For the case "c", the energy balance equation

$$\alpha_D \pi d_{in} \cdot (\vartheta - T_0) \Delta x + \frac{\lambda}{\Delta x} F_{sec} (T_1 - T_0) + \alpha_k \pi d_{in} (T_{env} - T_0) \Delta x = 0. \qquad (23)$$

After the designation:

$$A = \alpha_D \pi d_{in} \cdot \Delta x; \quad B = \frac{\lambda}{dx} F_{sec}; \quad C = \alpha_k \pi d_{in} \cdot \Delta x,$$

equation (20) takes the form

$$\frac{A}{(A+B+C)} \cdot \vartheta + \frac{B}{(A+B+C)} \cdot T_{env} + \frac{C}{(A+B+C)} \cdot T_1 = T_0. \qquad (24)$$

Simplifying (24), let's obtain the ratio for the temperature of the node "0"

$$T_0 = c_0 + c_1 \vartheta + c_2 T_1. \qquad (25)$$

The calculation by equations of the type (19), (22) and (25) is carried out by the Gauss-Seidel method [14], which allows to significantly reduce the time of convergence due to the constant updating of the current design temperatures.

Calculation of thermal conditions in the installation area of the thermal insulation jacket is carried out in a similar way, taking into account the thermal resistance of the cylindrical layer of thermal insulation.

3. Research results

The results of calculations of lifting reflux condensers of typical household ARU are given in **Table 1**.

Table 1

Results of calculation of the lifting reflux condensers of typical household ARU

Designation of calculation parameters	Basic Objects		
	Cooling capacity of the ARU evaporator, W		
	40	50	60
	Height of the lifting section of the reflux condenser, m		
	0,80	1,00	1,25
Mass flow rate $G \cdot 10^5$, kg/s			
– ammonia in reflux	0,198	0,248	0,297
– water in reflux	0,388	0,486	0,583
– reflux	0,586	0,733	0,878
– ammonia vapor at the outlet of the reflux condenser	3,330	4,171	5,002
– the vapor mixture at the inlet of the reflux condenser	3,916	4,905	5,882
Thermal load of the lifting section of the reflux condenser, W	20,9	26,2	31,5
Thermal resistance of thermal insulation at air temperature, K/W:			
– 32 °C	2,54	2,02	1,68
– 22 °C	3,01	2,40	2,00
Thermal losses at a comfortable room temperatur (22 °C), W:			
– when installing thermal insulation on the entire reflux condenser	1,8	2,2	3,3
– in basic construction	7,4	10,9	13,7
The increase in the cooling capacity of the ARU evaporator in comparison with the basic design, %	**17**	**22**	**22**

The calculations are carried out for a reflux tube with a diameter of 16´1.5 mm. The pipe material is steel (λ_S=45 W/(m×K)). The thermal insulation material of the jacket is a fiberglass cloth (λ_{ti}=0.056 W/(m×K)). Water-ammonia vapor mixture with a temperature of 120 °C is fed at the inlet of the lifting section of the reflux condenser.

4. Discussion of research results

Calculation shows that in order to ensure a complete purification of the ammonia vapor stream under the severe conditions of ARU operation, the thickness of the thermal insulation of the refluxing section in the form of a fiberglass cloth should be 3...4 mm thick.

Analysis of the calculation results in Table 1 shows that the installation of a heat-insulating jacket along the entire height of the reflux section of the main basic ARU designs makes it possible to increase the refrigerating capacity of the evaporator in comparison with the traditional partial thermal insulation of the lifting section of the reflux condenser by 17...22 %.

The developed model is of particular interest when optimizing the ARU operating conditions with variable thermal loads in the generator, including with afterburner. Having the dependence of the flow rate of the vapor mixture at the inlet to the reflux condenser on the applied heat load, it is possible to control the location of the ammonia vapor purification zone at any air temperature in the room and to realize the energy saving modes of operation of household ARUs using electronic control systems.

5. Conclusions

ARs possess a number of undoubted operational advantages (reliability, long life, noiselessness in operation, minimum cost) in comparison with compression analogs, as well as versatility in the use of energy sources, in the presence of energy-saving technologies, can expand their presence in the domestic refrigeration market.

One of the effective and low-budget methods for increasing the ARU energy efficiency is the technology to reduce losses when transporting ammonia to the artificial cold zone (evaporator). A key role in this process is performed by the ARU reflux condenser, which purifies ammonia vapor by removing the heat of a phase transition into the environment in the temperature range from 10 to 32 °C.

To reduce the losses during the transportation of ammonia through the ARU reflux condenser, it is necessary to install a thermal insulation that would efficiently purify ammonia vapor (at high ambient temperatures) within the working temperature range and promote its minimum condensation of ammonia (at low ambient temperatures).

Modeling of the thermal modes of the reflux condenser that are performed within the framework of this work allows to obtain such optimal parameters of heat insulation, and it is expedient to install thermal insulation along the entire length of the reflux condenser, which is not practiced in the latest developments of leading manufacturers.

As a result, it is shown the prospect of installing thermal insulation throughout the reflux section, which makes it possible to increase the energy efficiency by 17...22 %.

Particular importance of this study is for ARU energy-saving control systems, which use temperature indicators at the characteristic points of the reflux condenser to produce a control action [16].

References

[1]   Tassou, S. A., De-Lille, G., Ge, Y. T. (2009). Food transport refrigeration – Approaches to reduce energy consumption and environmental impacts of road transport. Applied Thermal Engineering, 29 (8-9), 1467–1477. doi: 10.1016/j.applthermaleng.2008.06.027

[2]   Rodríguez-Muñoz, J. L., Belman-Flores, J. M. (2014). Review of diffusion–absorption refrigeration technologies. Renewable and Sustainable Energy Reviews, 30, 145–153. doi: 10.1016/j.rser.2013.09.019

[3] Ngouateu Wouagfack, P. A., Tchinda, R. (2014). Optimal performance of an absorption refrigerator based on maximum ECOP. International Journal of Refrigeration, 40, 404–415. doi: 10.1016/j.ijrefrig.2013.11.025

[4] Yildiz, A. (2016). Thermoeconomic analysis of diffusion absorption refrigeration systems. Applied Thermal Engineering, 99, 23–31. doi: 10.1016/j.applthermaleng.2016.01.041

[5] Ersöz, M. A. (2015). Investigation the effects of different heat inputs supplied to the generator on the energy performance in diffusion absorption refrigeration systems. International Journal of Refrigeration, 54, 10–21. doi: 10.1016/j.ijrefrig.2015.02.013

[6] Dincer, I., Ratlamwala, T. A. H. (2016). Developments in Absorption Refrigeration Systems. Green Energy and Technology, 241–257. doi: 10.1007/978-3-319-33658-9_8

[7] Acuña, A., Velázquez, N., Cerezo, J. (2013). Energy analysis of a diffusion absorption cooling system using lithium nitrate, sodium thiocyanate and water as absorbent substances and ammonia as the refrigerant. Applied Thermal Engineering, 51 (1-2), 1273–1281. doi: 10.1016/j.applthermaleng.2012.10.046

[8] Yildiz, A., Ersöz, M. A., Gözmen, B. (2014). Effect of insulation on the energy and exergy performances in Diffusion Absorption Refrigeration (DAR) systems. International Journal of Refrigeration, 44, 161–167. doi: 10.1016/j.ijrefrig.2014.04.021

[9] Zhang, N., Lior, N. (2007). Development of a Novel Combined Absorption Cycle for Power Generation and Refrigeration. Journal of Energy Resources Technology, 129 (3), 254. doi: 10.1115/1.2751506

[10] Zohar, A., Jelinek, M., Levy, A., Borde, I. (2007). The influence of diffusion absorption refrigeration cycle configuration on the performance. Applied Thermal Engineering, 27 (13), 2213–2219. doi: 10.1016/j.applthermaleng.2005.07.025

[11] Mazouz, S., Mansouri, R., Bellagi, A. (2014). Experimental and thermodynamic investigation of an ammonia/water diffusion absorption machine. International Journal of Refrigeration, 45, 83–91. doi: 10.1016/j.ijrefrig.2014.06.002

[12] Dincer, I., Ratlamwala, T. A. H. (2016). Developments in Absorption Refrigeration Systems. Green Energy and Technology, 241–257. doi: 10.1007/978-3-319-33658-9_8

[13] Titlov, A. S. (2007). Sovremennyiy uroven razrabotok i proizvodstva byitovyih absorbtsionnyih holodilnyih priborov. Holodilnyiy Biznes, 8, 12–17.

[14] Kirillov, V. Kh., Zub, V. V., Titlov, A. S., Shirshkov, A. K. (2016). Komp'yuternoye modelirovaniye fizicheskikh i tekhnologicheskikh protsessov. Teoriya, algoritmy, programmy. Odessa: BMB, 565.

[15] Selivanov, A. P. (2013). Absorbtsionnyie holodilnyie apparatyi sezonnogo tipa. Sovremennoe sostoyanie i tendentsii razvitiya. Zbirnik Naukovih Prats NUK, (5-6), 82–88.

[16] Titlova, O. A. (2011). Avtomatizirovannoe rabochee mesto issledovatelya teplovyih protsessov v absorbtsionnyih holodilnyih priborah. Avtomatizatsiya Tehnologicheskih I Biznes-protsessov, 5-6, 60–64.