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## IMPROVEMENT OF OPERATION MODES OF THE EVAPORATOR OF THE ABSORPTION REFRIGERATING UNIT

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### Abstract

Absorption refrigeration units (ARU), which are part of absorption refrigeration devices (ARD) with a natural working fluid (water, ammonia and hydrogen) have a number of unique qualities. These qualities include: noiselessness, high reliability and long life; the possibility of using several energy sources in one device. At the same time, ARDs have increased energy consumption compared to similar compression models, and this does not allow them to expand their presence in the market of household refrigeration equipment.

The ARU evaporator provides a predetermined temperature level in the chambers of the refrigeration appliance and the required cooling capacity. In this regard, it is relevant to search for the operating modes of the evaporator that provide the ARU maximum energy efficiency, which is the aim of this work.

The thermal conditions of the direct-flow three-pipe design of the evaporator are simulated. The calculated ratio for a once-through evaporator is obtained taking into account the assumption of the adiabaticity of the evaporation process, when all the heat of the phase transition is used to cool the incoming flows of the purified vapor-gas mixture (VGM) and liquid ammonia to a minimum temperature.

The analysis of the results of calculating the operating modes of the evaporator made it possible to determine the directions of ways to increase the energy efficiency of both the evaporator itself and the ARU in general:

- preliminary cooling of the purified VGM flow at the inlet of the adiabatic section of the evaporator with an under-recovery of up to 5 °C and up to 10 °C;
- preliminary cooling of the liquid ammonia flow at the inlet of the adiabatic section of the evaporator with an under-recovery of up to 5 °C for all ARU types;
- increasing the purification degree of the VGM flow in the absorber allows increasing the temperature of the purified VGM flow at the inlet of the adiabatic section of the evaporator by 4...6 °C, i. e. to reduce the costs of useful cooling capacity for pre-cooling by 10...15 %.

**Keywords:** absorption refrigerating devices and units, evaporator, heat and mass transfer, energy saving.

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## 1. Introduction

The modern transition of refrigeration systems to natural refrigerants [1] makes developers of household systems pay close attention to absorption refrigerators. According to existing regulatory requirements and definitions [2], absorption refrigerators are recommended to be called “absorption refrigeration devices” (ARD). ARDs are equipped with absorption refrigeration units (ARU) with a natural working fluid (ammonia-water solution with the addition of inert gas-hydrogen) and can be considered as one of the alternative options for switching to environmentally friendly refrigerants [3].

ARD have a number of such unique qualities as [4]:

- a) noiselessness, long life, lack of vibration, magnetic and electric fields during operation;
- b) possibility of using several sources of energy in one device – both electric and alternative (heat of combustion of organic fuel, solar radiation, exhaust gases of internal combustion engines);
- c) ability to work with low-quality energy sources, including electric, in the voltage range of the network 160...240 V.

The ARD advantages include the lower cost compared to compression counterparts, which often determines their popularity in the market of household refrigeration equipment.

The main segment of ARD production falls on mini-refrigerators and minibars with a useful volume of up to 50 dm<sup>3</sup>, when it is not practical to use compression refrigeration units. ARD with burner devices is widely used by tourists, as they have no alternative in areas with a lack of electricity.

The possibility of using ARD with solar thermal energy sources [5], as well as compact cold generators [6], is also considered.

At the same time, ARDs have higher energy consumption compared to similar compression models, and this does not allow them to expand their presence in the market of household refrigeration equipment.

This situation is associated with the features of the ARU.

Firstly, with the presence of an additional “thermal” circuit in the ARU thermodynamic cycle, which performs the discharge function of the compressor [7].

The problem here is the thermodynamic properties of the ammonia-water solution, when, when the refrigerant (ammonia) is evaporated, a considerable amount of water (absorbent) also passes into the vapor state.

The presence of a rectifier, a reflux condenser, and regenerative heat exchangers do not fundamentally solve the problem of increased energy consumption [8].

Secondly, with the presence of a low-intensity diffusion component in the processes of ammonia evaporation into a vapor-gas hydrogen ammonia medium (VGM) and the absorption of ammonia vapor from VGM [9], including in the presence of helium, as an inert gas [10], which has higher compared with hydrogen, diffusion characteristics [11].

The evaporator and absorber in the ARU make up the so-called “natural circulation loop” [12], in which VGM moves at a speed of 0.10...0.50 m/s as a natural convection [13], which also reduces the intensity of internal heat and mass transfer processes.

The ARU evaporator provides a predetermined temperature level in the chambers of an absorption refrigeration device and the required cooling capacity.

In this regard, many developers and researchers are actively searching for the operating modes of the evaporator, providing ARD maximum energy efficiency as a whole.

## 2. Materials and methods

Despite the compactness and complexity of the design of modern designs of ARU evaporators, made, as a rule, in the form of a three-flow heat exchanger [14], the main studies are concentrated in the theoretical field [15–17].

For ARD, due to the complexity and interdependence of heat and mass transfer and gas dynamics, this is a completely justified method that allows, if not quantitatively, to qualitatively determine the trends in the influence of regime parameters on the geometric characteristics of elements.

Known studies [15–17] of the ARU operating characteristics are thermodynamic modeling of cycles with the determination of the specific consumption of components, enthalpy, and conversion coefficient from given temperatures, pressures, and composition at characteristic points (input-output of elements).

However, they do not take into account the features of thermophysical evaporation processes and the relationship of thermal and diffusion processes. In particular, no analysis is made of the amount of ballast inert gas in various operating modes and possible methods for minimizing it.

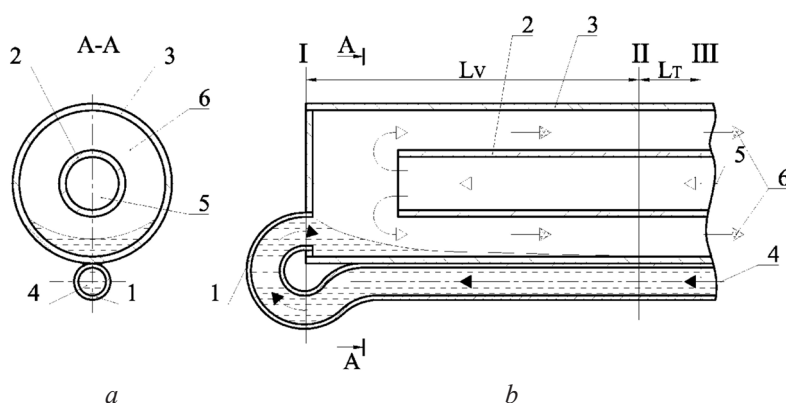
In this regard, it seems relevant to carry out mathematical modeling of heat and mass transfer processes in an evaporator of modern ARU, which should take into account the features of the physical processes of diffusion mass transfer.

As noted above, the long-term experience of the ARU developers has allowed to dwell on a straight-through three-pipe evaporator design, in which the process of producing artificial cold is carried out by evaporating the flowing flow of liquid ammonia into the VGM flow [14].

This design provides for pre-cooling and flow of liquid ammonia and VGM (**Fig. 1**).

Liquid ammonia 4 enters the initial section of the evaporator (section I) through channel 1, where it contacts a flow of purified VGM 5 coming in channel 2. At the initial stage, at a low partial pressure of ammonia in the flow of purified VGM (about 1.0...1.5 bar [18]) evaporation of liquid ammonia occurs at a temperature of  $-33...-28$  °C. This is the lowest evaporator temperature.

Further, in the process of evaporation, VGM is saturated with ammonia vapor. The ammonia concentration in VGM and the VGM density in this case increase.



**Fig. 1.** Flow pattern in the initial section of a straight-through evaporator:  
*a* – cross section; *b* – initial section of the evaporator; 1 – channel of liquid ammonia;  
 2, 3 – channel flow of purified (saturated) VGM; 4 – flow of liquid ammonia;  
 5, 6 – flow of purified (saturated) VGM

Firstly, this leads to an increase in the partial pressure of ammonia and, accordingly, to an increase in evaporation temperatures. With complete evaporation of liquid ammonia in the VGM, they will have a maximum evaporation temperature (section II).

Secondly, a density difference arises between saturated ammonia and unsaturated ammonia of VGM, which sets in motion the natural circulation circuit between the evaporator and the ARU absorber.

When flows flow through the evaporator in section I–II, heat exchange processes take place:

- between the flow of evaporating ammonia in the inter-pipe section and the flow of liquid ammonia passing to the initial portion of the evaporator;
- between a flow of cold saturated VGM and a flow of purified VGM.

Both processes are in counterflow mode and contribute to lowering the temperature in the initial section of the evaporator.

In section II–III, evaporation ends and the cooling flows of liquid ammonia and purified VGM are cooled by a flow of cold saturated VGM moving in the inter-pipe section.

The thermal conditions of the straight -flow ARU evaporator are determined by the temperature and concentration parameters of the VGM and liquid ammonia flows entering the initial section.

The mass composition of the purified VGM flow depends on the purification degree efficiency in the absorber, and the temperature of the VGM and ammonia flows depends on the fraction of useful cooling capacity used to supercool these flows.

In the initial section of the direct-flow evaporator, the minimum temperature of ammonia evaporation is reached upon contact with purified VGM. Given the small size of the contact zone (an area of the order of 1 cm<sup>2</sup> [18]), it can be assumed that the interaction process occurs under adiabatic conditions.

Based on this assumption, the relations are obtained for the minimum evaporation temperature, respectively, for the counterflow mode [19] and forward flow mode [20] of the flow of the evaporating liquid and the VGM flow. The relations for the forward flow [20] is obtained with the assumption that the Lewis number is equal to  $Le=1$  during the evaporation of liquid ammonia in the ASG.

The analysis shows that the use of this assumption in the calculation of various chemical technological processes gives satisfactory results only for the case of evaporation of water into air and leads to significant discrepancies in a number of other cases.

Experimental studies of the evaporation of ammonia in VGM [21] give Lewis numbers in the range  $Le=0.745...0.758$ .

In further modeling and analysis, let's select the results of experimental studies as the most suitable for the working conditions of ARD evaporators, and  $Le=0.750$  is adopted.

The calculated ratio for a straight-through evaporator is obtained taking into account the assumption of the adiabaticity of the evaporation process, when all the heat of the phase transition is used to cool the incoming VGM flows and liquid ammonia to a minimum temperature ( $v_{min}$ )

$$\alpha \cdot (\theta_v^s - v_{min}) = \beta_p \cdot r_{v(min)} \cdot (P_{x(min)}^s - P_{x(min)}), \quad (1)$$

where  $\alpha$  – the convective heat transfer coefficient during the evaporation of ammonia in the VGM, W/(m<sup>2</sup>·K);  $\theta_v^s$  – temperature of the purified VGM flow at the beginning of the evaporation process, °C;  $v_{min}$  – minimum evaporation temperature of liquid ammonia, °C;  $\beta_p$  – mass transfer coefficient related to the difference in pressure of ammonia at the liquid surface and in the VGM flow, kg/(Pa·s);  $r_{v(min)}$  – specific heat of ammonia vaporization at a minimum evaporation temperature  $v_{min}$ , J/kg;  $P_{x(min)}^s$ ,  $P_{x(min)}$  – vapor pressure of ammonia, respectively, of saturation at an evaporation temperature  $v_{min}$  and partial in the VGM flow at the inlet of the adiabatic section, Pa.

Given the known relations [22]

$$Le = \frac{\alpha}{\beta_p \rho C_p} \quad \text{and} \quad \beta = \beta_p \frac{R}{\mu_A} \theta_v^s$$

equation (1) takes the form

$$Le \frac{R}{\mu_G} \cdot \theta_v^s \cdot \rho C_p (\theta_v^s - v_{min}) = r_{v(min)} (P_{x(min)}^s - P_{x(min)}), \quad (2)$$

where  $R=8314$  – universal gas constant, J/(kmol·K);  $\mu_G$  – relative molecular weight of purified ASG;  $\rho$ ,  $C_p$  – respectively, density and isobaric mass heat capacity of purified VGM at temperature  $\theta_v^s$  and pressure  $P_{x(min)}$ , kg/m<sup>3</sup> and J/(kg·K).

The numerical values of the minimum evaporation temperature  $v_{min}$  of modern ARU is in the range  $-30...-20$  °C [23, 24] and the dependence of ammonia pressure on the temperature on the saturation line, obtained by approximating tabular data [25], has the form

$$P_{x(\min)}^S = 0.07 v_{\min} - 15.98, \text{ bar}, \quad (3)$$

where  $v_{\min}$  – minimum temperature of ammonia evaporation, K.

Equation (2), taking into account (3), can be solved with respect to temperatures  $\theta_v^S$  or  $v_{\min}$ .

In the first case, equation (2) is converted to a quadratic equation

$$\theta_v^S - v_{\min} \cdot \theta_v^S - n = 0, \text{ K}, \quad (4)$$

where  $\theta_v^S$  – temperature of the purified VGM flow at the beginning of the evaporation process, K;

$$n = \frac{0.07 v_{\min} - 15.98 - P_{x(\min)}^S}{C};$$

$$C = \frac{\text{Le} \cdot R \cdot \rho \cdot C_p}{\mu_G \cdot r_{v(\min)} \cdot 10^5}.$$

The thermophysical meaning of the solution of equation (4) is that the temperature of the purified VGM flow is at the inlet of the adiabatic section of the evaporator, which provides a given minimum evaporation temperature for a given composition of the purified VGM.

In the second case, the solution of equation (2) gives the relation

$$v_{\min} = \frac{m \theta_v^S + r_{v(\min)} \cdot 10^5 \cdot (P_{x(\min)}^S + 15.98)}{0.07 \cdot 10^5 \cdot r_{v(\min)} + m}, \quad (5)$$

where

$$m = \text{Le} \frac{R}{\mu_G} \theta_v^S \rho C_p.$$

The thermophysical meaning of the second solution is that the minimum temperature of ammonia evaporation at the inlet of the adiabatic section of the evaporator is determined, depending on the specified composition and flow temperature of the purified VGM.

Equations (4) and (5) are the basis for the analysis of the influence of the parameters of the incoming flows of VGM and liquid ammonia on the operating modes of the ARU countercurrent evaporator (**Fig. 1**).

### 3. The results of modeling the ARU evaporator thermal operating conditions

The calculations were carried out in the range of operating parameters characteristic of modern ARD using the above mathematical model of heat and mass transfer processes in the ARU evaporator.

The minimum evaporation temperature varied from minus 35 °C (freezers) to minus 20 °C (refrigerators with a low-temperature compartment – LTC). These temperatures correspond to a saturation pressure of ammonia of 0.93 bar and 1.9 bar [26].

The driving pressure drop in the once-through evaporator is determined by the level of minimum evaporation temperatures and the purification degree of the VGM flow in the absorber. At the outlet of the absorber, the VGM flow interacts with the weak WAM flow, whose minimum concentration in ammonia is limited by the corrosion processes in the ARU generator and in almost all modern schemes is  $\xi_w \approx 15\%$  [27].

With this composition and WAM temperature at the outlet of the absorber  $T'_A = 44$  °C, the partial pressure of ammonia in the VGM flow can't be less than 0.7 bar. At temperatures  $T'_A = 40$  °C and  $T'_A = 20$  °C, the partial pressure is 0.5 bar and 0.2 bar, respectively. With these considerations

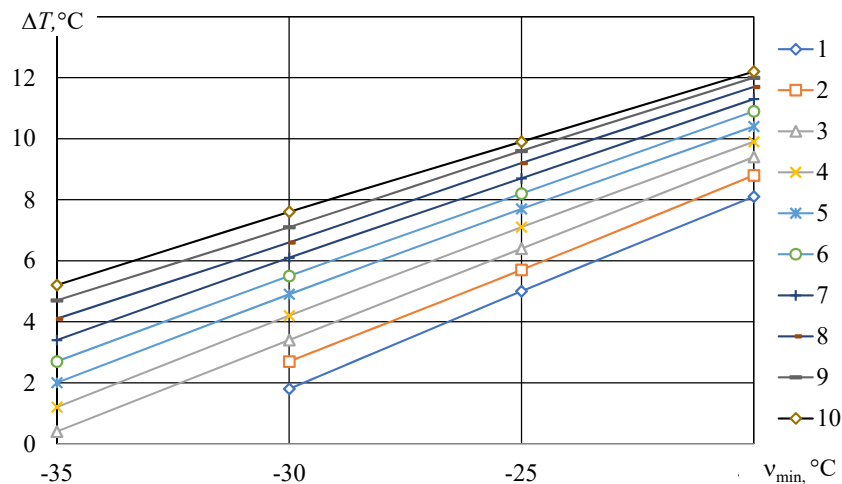
in mind, the numerical values of the difference in the partial pressure of ammonia during the evaporation  $\Delta P$  were taken in the range from 0.1 to 1.0 bar.

At the first stage, the influence of the Lewis number on the numerical values of the calculated temperatures was studied. The appropriateness of the assumptions made was assessed, taking into account the differences in regime and design parameters.

Lewis numbers ranged from 0.75 to 1.00.

The calculations showed that they slightly affect the temperature values in the range  $v_{\min} = -35 \dots -20$  °C (maximum up to 1 °C). Nevertheless, in further calculations, the value of  $Le=0.75$  was used, as the closest to the real processes of ammonia evaporation in VGM.

The results of calculating the temperature difference in the adiabatic part of the evaporator are shown in **Fig. 2**.



**Fig. 2.** The effect of the minimum evaporation temperature and the difference in the partial pressure of ammonia during the evaporation process on the temperature difference in the initial (adiabatic) section of the evaporator: the difference in the partial pressure of ammonia during the evaporation process, bar: 1 – 1; 2 – 0.9; 3 – 0.8; 4 – 0.7; 5 – 0.6; 6 – 0.5; 7 – 0.4; 8 – 0.3; 9 – 0.2; 10 – 0.1

**Fig. 3** shows the calculated dependence of the temperature of the purified VGM flow on the minimum evaporation temperature in the adiabatic section of the evaporator.

Presented in **Fig. 2, 3**, the calculation results allow to conclude that it is necessary to significantly subcool the flow of purified VGM, especially in the case of low-temperature performance of ARD (freezers).

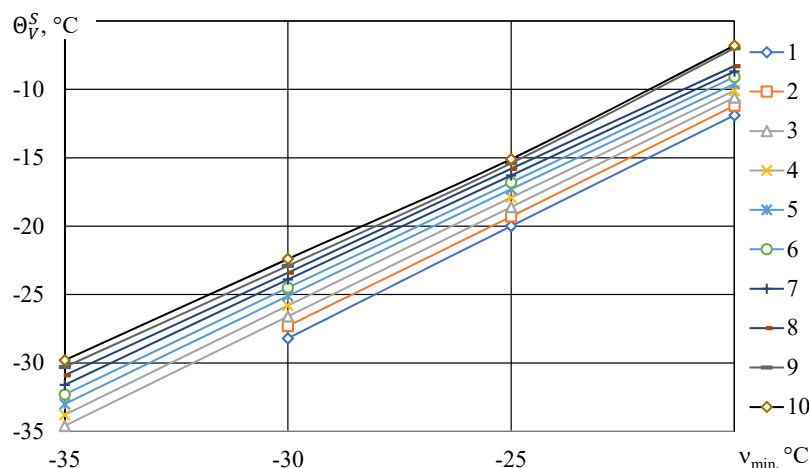
So, in the temperature range  $v_{\min} = -35 \dots -30$  °C, the average value of the temperature of the VGM flow at the inlet of the adiabatic section lies in the range from minus 32 to minus 26 °C. For refrigerators with LTC, the situation is more favorable – VFM flow must be cooled, on average, to minus 10 °C.

As for the influence of the purification degree of the inert gas in the absorber on the operating modes of the evaporator. The best VGM purification from ammonia in the absorber allows to maintain a higher temperature level of the VGM flow at the inlet of the adiabatic section and thus reduce the consumption of cold for pre-cooling the AVGM flow.

In the range of partial pressures under study, due to a higher degree of VGM purification in the absorber, it is possible to increase the VGM temperature at the inlet of the adiabatic section of the evaporator by 4...6 °C for all ARDs.

In the presence of under-recovery processes during absorption in real ARU, it is not possible to completely purify VGM from ammonia vapor. VGM will contain some “ballast” amount of ammonia vapor, which will reduce the driving pressure drop during evaporation and, ultimately, reduce the energy efficiency of the ARU thermodynamic cycle.

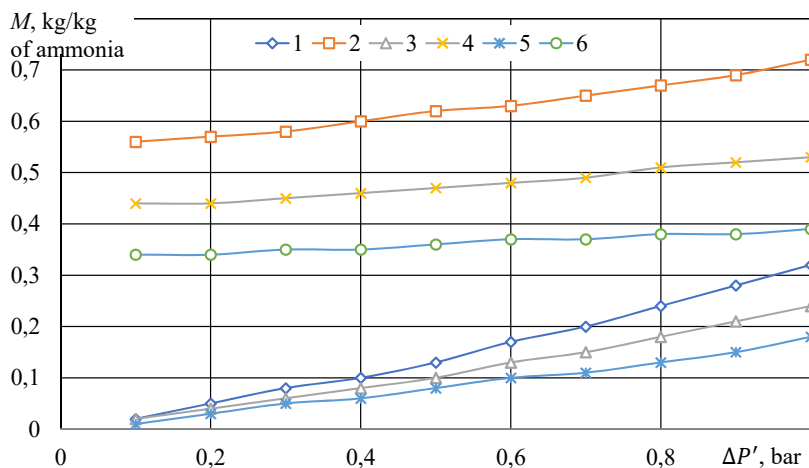




**Fig. 3.** The dependence of the temperature of purified VGM on the minimum evaporation temperature and the difference in the partial pressure of ammonia during evaporation on the adiabatic part of the evaporator: the difference in the partial pressure of ammonia during the process of evaporation, bar: 1 – 1; 2 – 0.9; 3 – 0.8; 4 – 0.7; 5 – 0.6; 6 – 0.5; 7 – 0.4; 8 – 0.3; 9 – 0.2; 10 – 0.1

Let's evaluate such an unfavorable component of the evaporation process. The calculation of the specific consumption of inert gas-hydrogen and the "ballast" mass of ammonia was carried out for three operating modes at temperatures at the end of the evaporation process (at the outlet of the evaporator):  $-5^{\circ}\text{C}$ ;  $0^{\circ}\text{C}$ ;  $5^{\circ}\text{C}$  (**Fig. 4**). The mass concentration of ammonia in VGM for these temperature conditions is 0.648, respectively; 0.700; 0.747.

Such modes are implemented, respectively, in freezing low-temperature ARU, refrigerators with LTC and mini-refrigerators.



**Fig. 4.** The results of calculations of the specific flows of inert gas and the "ballast" mass of ammonia under different operating conditions of the ARU evaporator: mass fraction of ammonia vapor in the saturated VGM flow at the evaporator ( $\xi_{\max}$ ): 1, 2 – 0.648; 3, 4 – 0.700; 5, 6 – 0.747; temperature of the end of evaporation: 1, 2 – minus  $5^{\circ}\text{C}$ ; 3, 4 –  $0^{\circ}\text{C}$ ; 5, 6 –  $5^{\circ}\text{C}$ ; 1, 3, 5 – specific flow of the "ballast" mass of ammonia; 2, 4, 6 – specific flow of inert gas-hydrogen

#### 4. Discussion and analysis of modeling results

An analysis of the calculated results shows that with an increase in the pressure drop during the evaporation process from 0.1 to 1.0 bar:

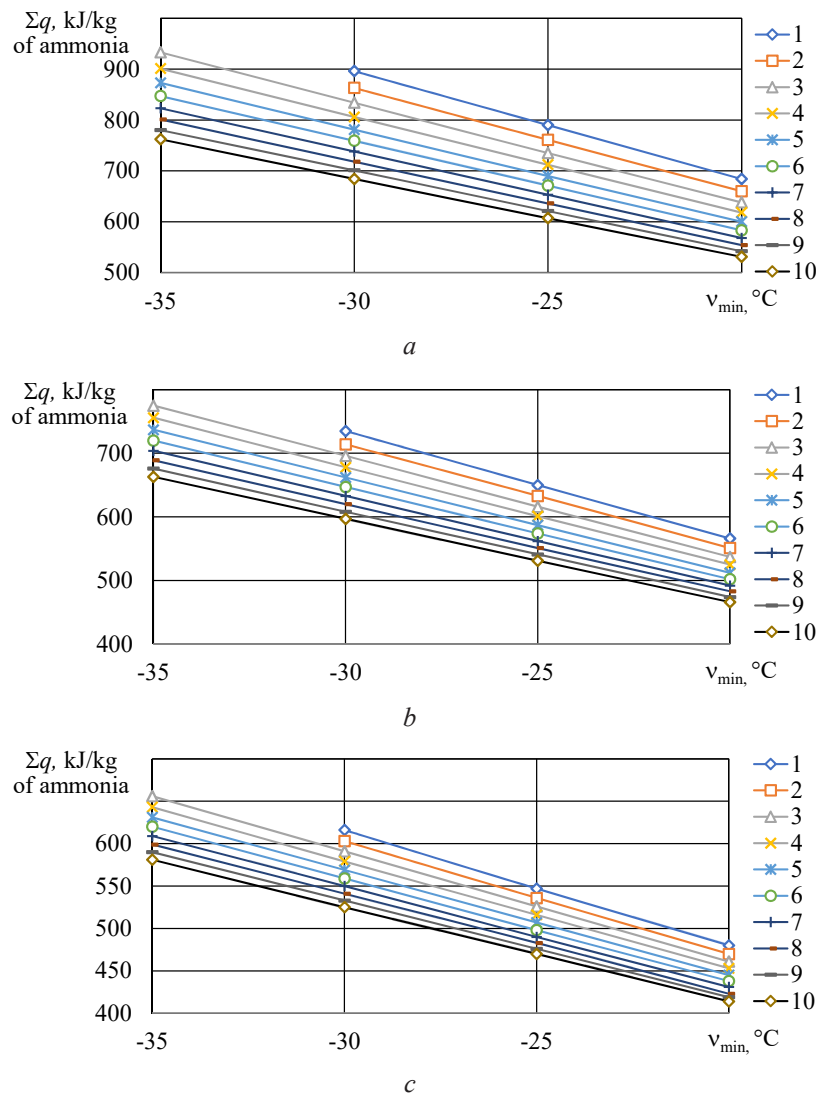
a) the specific amount of circulating inert gas-hydrogen increases insignificantly (1.1...1.3 times);

- b) the specific amount of “ballast” ammonia increases 9...16 times;  
c) the largest specific amount of “ballast” ammonia circulates in low-temperature ARU (freezers) with the highest possible difference in partial pressure during evaporation.

An assessment is made of the useful cooling capacity used to cool the flows of purified VGM and liquid ammonia. The calculation results are shown in **Fig. 5**.

It is shown that the maximum cooling capacity spent on cooling the VFM and liquid ammonia flows occurs in low-temperature ARDs (freezers). If to take such a “ballast” cooling capacity for 100 %, then for medium-temperature models of ARD it will be 80 %, and for mini-refrigerators –70 %.

An increase in the purification degree of the VGM flow in the absorber allows one to reduce the “ballast” cooling capacity by 20 %.



**Fig. 5.** The results of calculating the total specific cooling capacity spent on cooling the flows of purified VGM and liquid ammonia: the difference in the partial pressure of ammonia in the flow of purified VGM at the inlet to the evaporator, bar: 1 – 1; 2 – 0.9; 3 – 0.8; 4 – 0.7; 5 – 0.6; 6 – 0.5; 7 – 0.4; 8 – 0.3; 9 – 0.2; 10 – 0.1; mass fraction of ammonia vapor in the purified VGM flow at the outlet of the evaporator ( $\xi_{\max}$ ): a – 0.648; b – 0.700; c – 0.747

Analysis of graphical dependencies in **Fig. 5** shows that a decrease in the minimum evaporation temperature in all cases leads to an increase in the cooling capacity spent on



cooling the flows of liquid ammonia and VGM. Presented on **Fig. 5** graphical dependencies allow developers to accurately assess the possible costs of “ballast” cooling capacity in ARD for various purposes.

So, for example, in universal ARDs operating in a wide range of refrigeration storage temperatures from minus 18 °C to plus 12 °C, about 65 % of useful cooling capacity is spent on cooling VGM and liquid ammonia flows. 60 % are spent in VGM with LTC (temperatures in two chambers from minus 12 °C to plus 5 °C), in mini-refrigerators (temperature plus 5 °C) – 55 %.

Calculated results presented in **Fig. 5** were obtained under the assumption of complete hypothermia of liquid ammonia at the inlet to the adiabatic section of the evaporator, i. e. bringing its temperature to the temperature of the beginning of evaporation.

It is known [19] that in order to increase the cooling capacity in evaporative systems, the temperature of the liquid at the entrance to the contact zone should be kept as close as possible to the temperature of the beginning of evaporation ( $v_{\min}$ ).

Otherwise, throttling of liquid ammonia, which is not cooled to saturation temperature, will be throttled to a two-phase region, where saturated vapor and liquid will be in thermodynamic equilibrium.

The presence of additional ammonia vapor in the adiabatic section of the evaporator will lead to a decrease in the driving drop during the evaporation and to an increase in both the minimum temperature ( $v_{\min}$ ) and all temperatures in the evaporator. The temperature in the ARD refrigerator compartment will increase accordingly.

Let's evaluate the effect of the temperature of the incoming liquid ammonia flow on the thermal conditions of the adiabatic section of the evaporator.

In the calculation, let's use the well-known relation [4] for the specific enthalpy in the two-phase region

$$h_x = h''(1-x) + h'x, \quad (6)$$

where  $h'$ ,  $h''$  – the specific enthalpies of saturated liquid and saturated vapor, respectively;  
 $x = \frac{M_v}{M_v + M_L}$  – dryness degree of wet steam in the two-phase region.

From relation (6) let's find the dryness degree of the wet ammonia vapor

$$x = \frac{h_x - h'}{h'' - h'}. \quad (7)$$

Given the conditions  $M_v + M_L = 1$ , let's obtain the ratio for the amount of ammonia vapor in the adiabatic section of the evaporator

$$M_v = \frac{h_x - h'}{h'' - h'}. \quad (8)$$

The results of calculations of the specific amount of ammonia vapor in the adiabatic section of the evaporator due to incomplete cooling of liquid ammonia to the saturation temperature are presented in **Table 1**.

Analysis of the calculation results given in **Table 1** allows to draw the following two conclusions.

Firstly, with a fixed temperature difference between the flow of liquid ammonia and a minimum evaporation temperature in the initial adiabatic section of the evaporator, the amount of ammonia vapor after throttling is practically unchanged.

Secondly, with an increase in the temperature difference of the liquid ammonia flow, the amount of ammonia vapor after throttling increases linearly.

These calculation results confirm the earlier conclusions about the need for maximum cooling of the liquid ammonia flow at the inlet to the ARU evaporator.

**Table 1**

The results of the calculation of the specific amount of ammonia vapor on the adiabatic section of the evaporator due to incomplete cooling of liquid ammonia to a saturation temperature,  $M_{\nu}$ , kg/kg of ammonia

The temperature difference between the flow of liquid ammonia and the minimum evaporation temperature in the initial adiabatic section of the evaporator, °C	Minimum evaporation temperature, $t_{\min}$ , °C			
	–35	–30	–25	–20
5	0.016	0.016	0.017	0.017
10	0.032	0.033	0.033	0.034
15	0.049	0.049	0.050	0.051
20	0.065	0.066	0.068	0.068

From the analysis of the Table 1 it can also be concluded that for all types of ARU, a decrease in the temperature of liquid ammonia at the inlet of the evaporator by 5 °C, compared with the temperature of evaporation, will lead to an increase in temperature in the evaporator by an average of 1.5 °C. Such an increase practically does not affect the temperature field of the cold stone, but expands the range of possible structural solutions of the evaporator during design.

#### 4. Conclusions

1. A mathematical model of the thermal conditions of the ARU evaporator is developed, taking into account the features of the physical processes of diffusion evaporation. In contrast to the existing approaches [15–17], the model contains the relationships for Lewis numbers obtained experimentally.

2. Analysis of the results of calculating the thermal operation of the ARU evaporator makes it possible to determine the directions of ways to increase the energy efficiency of both the evaporator and ARU in general:

a) pre-cooling the flow of purified VGM at the inlet of the adiabatic section of the evaporator with under-recovery of up to 5 °C (for freezers) and up to 10 °C (for models with LTC), for example, using intensive external cooling [28];

b) preliminary cooling of the liquid ammonia flow at the inlet of the adiabatic section of the evaporator with an under-recovery of up to 5 °C for all types of ARU;

c) an increase in the purification degree of the VGM flow in the absorber allows the flow temperature of the purified VGM at the inlet of the adiabatic section of the evaporator to increase by 4...6 °C, for example, due to the intensification of internal mass transfer [13] and external heat transfer [28]. This reduces the consumption of useful cooling capacity for pre-cooling by 10 (in the case of freezers)...15 % (in the case of refrigerators with LTC).

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