ХОЛОДИЛЬНА ТЕХНІКА ТА ЕНЕРГОТЕХНОЛОГІЇ

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Analysis of temperature modes and temperature differences in heat exchangers of solar-powered absorption refrigeration systems

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Solar cooling technologies include a solar power plant as a heat source and heat-using refrigeration machine as a cooling production. The article presents the analysis of temperature regimes and temperature differences in heat exchange devices of absorption refrigeration systems from the standpoint of ensuring the workability and high efficiency of processes. The analysis was performed on the example of a single-effect water-ammonia absorption machine. The analysis is based on the following conditions: the presence of the system limit cycle. That is, the interdependence of the temperature potentials of three heat sources, two of which are chosen arbitrarily, the third is a function of the first two; temperature regimes and temperature differences in the heat exchangers are given in interaction with the temperatures of external energy sources .The presence of a system total temperature differences, consistent with the limit cycle at known temperature levels of the heat sources, was established. The total head takes into account: the generator temperature range, the temperature at the generator hot edge, the temperature at the solution heat exchanger cold edge, the temperature differences at the absorber cold edge, at the condenser smallest temperature differences, water heating in the condenser. The issue of optimizing temperature differences in system elements is proposed to be solved by considering the variable component of capital costs associated with the cost of each heat exchanger depending on the temperature differences. For the practical implementation of cycles with the considered temperature regimes and solar powered generator on the basis of energy saving, cycles of absorption systems "with an extended degassing zone" are proposed. The cycles are implemented under conditions of different nature of temperature changes of energy sources and working fluid in the heat exchange processes between them. Solving problems is recommended by creating complex schemes and cycles of hybrid water-ammonia refrigeration systems.

Keywords: *Solar cooling technology; Water-ammonia absorption system; Heat exchanger; Temperature difference; Degassing zone.*

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1. Introduction

Clean and renewable, with a low price and natural availability, solar energy is one of the sources for solving the social problems of the modern world with huge potential for applications in air conditioning and refrigeration. In particular, the need for commercial and residential air conditioning during the hot season is constantly increasing. In recent years, many countries have faced difficulties in generating and storing electricity for the operation of systems with high energy consumption, such as refrigeration and heat pump technology [1]. The large capacities absorption refrigeration systems operating with water/ ammonia (for cooling purposes) or water-lithium bromide (for air conditioning) are actually used in many countries. Most of them operate by means of heat recovery from industrial processes and, just a few of them are actually operating with solar energy. Solar refrigeration systems may be very important in the development of small communities. Those are away far from the electrical net. Peoples living in these areas can produce the ice needed to conserve their food, and also to store and transport f fish, fruits or vegetables directly to the customers. Thus, solar cooling technologies have become the focus of attention in the world [2,3].

Solar cooling technologies include a solar power plant as a heat source and heat-using refrigeration machine as a cooling production. Heat exchangers that interact with external heat sources are essential elements of absorption refrigeration systems, the need for their use is due to the very principle of the system. The inclusion of auxiliary devices in the technological scheme of the machine is not fundamentally mandatory, but their use improves the performance of the machines. It increases the reliability and efficiency of their work. In energy and refrigeration systems, especially in areas with a lack of fresh water, air-cooled heat exchangers (condensers, dry coolers, absorbers) are widely used. In such conditions, when determining the technical and economic indicators of machines, it is important to take into account the time of using the installed capacity of the equipment.

Refrigeration systems, as you know, even when maintaining a constant temperature in the cooled object, operate in a variable mode. The main influence on the regime is exerted by the ambient temperature.

The increased ambient temperature in countries with a temperate climate, which the system usually counts on, is observed for a short time – 1000-2000 hours a year, moderate temperatures last 3000-4000 hours, the rest of the time is dominated by low. In the calculations of energy systems, the average ambient temperature is taken, and the time of using the installed capacity of the equipment is reasonably obtained to be 4000-5000 hours per year. Refrigeration systems are calculated in the mode of the highest ambient temperature, therefore, the determination of the operating time of the equipment requires special study. Failure to take this factor into account leads to inefficient use of heat exchange equipment.

Numerous calculations performed have shown that in heat exchangers cooled by the environment, an additional heat transfer surface affects the energy efficiency of the system only during a period of elevated ambient temperature. At a moderate temperature, the energy efficiency is almost equal to the calculated one, at a low temperature, the additional surface only leads to additional operating costs, an increase in heat losses and complicates the automatic control of the system. Thus, when determining the optimal dimensions of the surfaces of heat exchangers of refrigeration systems, the time of using the maximum installed power should be taken as no more than 1000-2000 hours per year. It is necessary to summarize the costs in three modes when determining the annual energy consumption and divide the system elements into base and peak. The base elements, designed for long-term operation, should be characterrized by high energy efficiency, the peak elements – with low initial investment. Differentiation of regimes can be achieved by changing the seasonal parameters of heat sources. Based on these provisions, generalized characteristics of the operation of absorption machines are outlined.

However, to determine the cost of the equipment, it is necessary to clarify the optimal temperature differences in the heat exchangers. The issues of choosing temperature differences in the heat exchangers of compressor machines are considered in many works and have become classic in refrigeration engineering [4]. To date, little attention has been paid to the determination of temperature differences in heat exchangers of absorption systems.

A review of studies of absorption systems driven by solar energy, given in [5], showed that modern studies are represented by scheme-cycle design, there is no analysis of processes and designs of heat exchangers. Meanwhile, the specificity of the thermodynamic cycles of absorption systems does not allow solving optimization issues by analogy with compressor systems and requires independent analysis.

2. Temperature modes of absorption refrigeration systems

Let us turn to the technological scheme of a single-effect absorption refrigeration system, which is shown in Fig.1

Absorption refrigeration systems must have three different temperature potentials energy sources to implement the thermodynamic cycle (Fig. 1):

• high-potential (T_h), used as a source that heats the generator;

•intermediate potential (T_w), which is the external environment (water or air) for cooling the condenser and absorber;

• low-potential (T_{cold}), cooling effect is produced at a low temperature

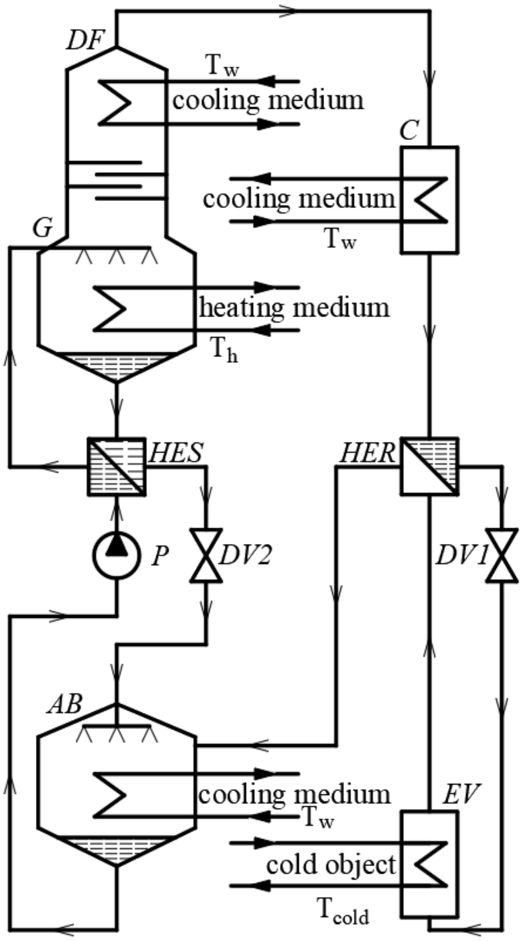

Figure 1 *– Technological scheme of a single-effect absorption refrigeration system: A – absorber; C – condenser; D – dephlegmator; DV – throttle device; EV – evaporator; G – generator; HER – refrigerant heat exchanger; HES – solution heat exchanger; P – pump*

The three temperature levels are interconnected so that only two can be chosen arbitrarily, the third is always a function of any two. The dependence between temperatures is presented in Fig. 2 in graphic form $T_h = f\ (T_w,\ T_{cold})$ according to the data of works [6,7] The dependence is valid in the temperature range T_{cold} = 223...293 K, T_w = 283...313.

Solar, geothermal and other types of renewable and non-traditional energy sources are characterized by a low temperature potential, which for climat of Eastern Europe does not exceed 100 °C.

From the point of view of the great energy industry, it has been proven that heat with a temperature potential of T_h = 70°C is no longer operational, and its emissions into the atmosphere are ecologically safe. Small energy industry does not confirm such a situation. Theoretically, it is possible to create a single-effect cycle at T_h = 80 + 100 °C due to the presence of a similar spin T_{cold} and T_w (Fig. 2). Real temperature relation is much more complicated. The real values of T_h are higher than those shown in Fig. 2.

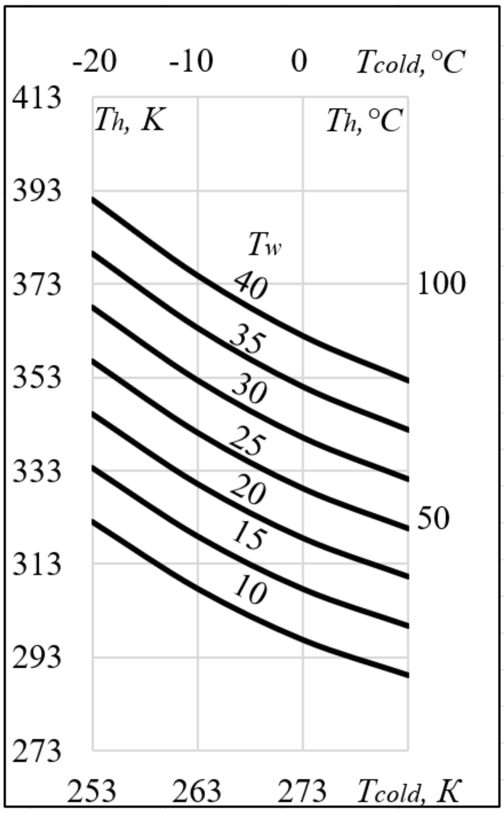

Figure 2 *– Theoretical $T_h = f(T_w, T_{cold})$ of a single-effect absorption refrigeration system*

It is known from the general experience of the design absorption systems, that it is impossible even theoretically to implement a single-effect cycle at T_h =+ 100 °C, T_w = 40 °C and T_{cold} = 0 °C (air conditioning)

The economic efficiency of any and, in particular, absorption refrigeration systems, significantly depends on the temperature differences in the heat exchangers. The heat exchange surface decreases with an increase of the temperature differences and, as a consequence, capital investments are reduced. At the same time, the external irreversibility of heat transfer processes increases, which in most cases leads to an increase in energy and cooling medium consumption. Obviously, there are temperature differences, at which the optimal efficiency of the absorption system is achieved.

3. Temperature differences in the heat exchangers of the absorption refrigeration system

Evaporator, condenser, solution heat exchanger and refrigerant heat exchanger in absorption system – primary dual-flow heat exchangers (Fig. 1).

The processes that are involved in them are connected with the heat exchange of single-phase flows of the working fluid or phase transformations of the pure components of the mixture. The absorber and the generator are heat exchangers with the processes of mixing and condensation, or boiling with separation of the mixture.

Consider the cycle of a single-effect water-ammonia refrigeration system in interaction with the external energy sources and temperature differences of the heat exchangers (Fig. 3)

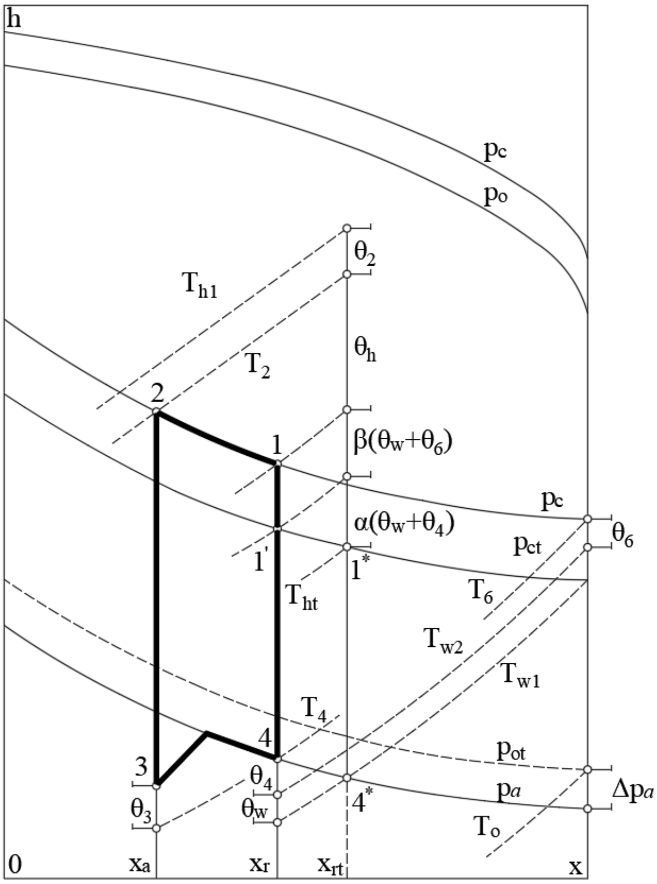

Figure 3 *– Processes in the absorption machine for h-x diagrams: 1-2 at the generator; 2-3 and 4-1 at the solution heat exchanger, 3-4 in the absorber*

Fig.3 shows:

T_{h1} – heating medium temperature at the generator inlet;

T_{w1} and T_{w2} – cooling medium temperatures at the condenser inlet and outlet;

T_0 – refrigerant evaporating temperature at the evaporator,

θ_2 – temperature difference at the generator hot edge;

θ_3 – temperature difference at the solution heat exchanger cold edge;

θ_4 – temperature difference at the absorber cold edge;

θ_6 – smallest temperature difference at the condenser;

θ_w – $(T_{w2} - T_{w1})$ – cooling medium heating in the condenser.

Sequential cooling of the condenser and absorber is proposed. If we assume a constant pressure difference $(p_0 - p_a)$ at all the evaporator and absorber modes, then the segment 1^*-4^* determines the limit cycle of the solution. Such a cycle has a system with infinitely large surfaces of heat exchangers and infinitely large flows of solution, heating and cooling mediums.

In the limit cycle, the temperature of the solution in the generator coincides with the theoretically lowest temperature T_{ht}. In a real cycle, the temperature T_h is always higher than T_{ht}. The temperature difference $\theta tot = (T_h - T_{ht})$ makes it possible to obtain the final temperature differences in the heat exchangers and the temperature difference at the generator $\theta_h = (T_2 - T_1)$.

The total temperature difference in the cycle has the form

$$\theta_{tot} = \theta_h + \theta_2 + \alpha(\theta_w + \theta_4) + \beta(\theta_w + \theta_6). \qquad (1)$$

Here α – the coefficient that takes into account the effect of the pressure change between p_a and p_{ct} on the saturation temperature according to the known concentration difference $(x_r - x_{rt})$; β – the coefficient that takes into account the effect of x_r to the saturation temperatures, corresponding to the pressure difference $(p_c - p_{ct})$; p_{ct} and x_{rt} – condensation pressure and solution concentration in the limit cycle. For a water-ammonia solution, usually α = 1,15...3; β = 1,2...1,35.

It can be seen from the (1) *G* that the heat exchangers temperature differences are mutually related at a given temperature difference $(T_h - T_{ht})$, with a change in one, the others also change or the temperature interval in the generator changes = $(T_2 - T_1)$. Obviously, as the value of θ decreases, the temperature differences in individual heat exchangers also decrease. Since the value of these θ_{im} should not be taken less than 2 ... 3 deg. Then, for small values of the argument θ_{sum}, one has to deviate from the recommended values. The issues of optimizing the temperature differences in the elements of the system are solved by considering the variable component of capital costs K associated with the cost of each heat exchanger depending on the temperature difference. This value can be represented as a sum

$$K = s_g F_g + s_a F_a + s_c F_c + s_{hes} F_{hes} + ..., \qquad (2)$$

where s_g, s_a, s_c, s_{hes} – variable components of the cost of the generator, absorber, condenser,solution heat exchanger, etc., related to the surface unit, cu/m^2; F_g, F_a, F_c, F_{hes} – surfaces of heat exchangers.

Heat exchange surface F_i of the skin heat exchanger, depends on the middle temperature difference θ_{im} and through it, the heat load of the heat exchanger Q_i is the temperature difference in the others, to which the value of K can be given by a function of many variable arguments

$$K\left(\theta_{gm}, \theta_{am}, ..., \theta_{im}...\right) = \sum s_i \frac{Q_i}{k_i \theta_{im}}, \qquad (3)$$

where Q_i – heat load of the heat exchanger, kW; k_i – heat transfer coefficient, W/m^{2}K.

Calculations of the absorption refrigeration system are carried out by the method of iterations, setting the values of the temperature differences θ_2, θ_3,…, θ_i…, θ_w. Then, for an arbitrary increment $\Delta\theta_i$ of one of the heat exchangers, the system is calculated again. If the obtained values differ significantly from the initial ones, the calculation is repeated. The optimal values of the average temperature differences in heat exchangers correspond to the optimal values θ_2, θ_3, θ_4, θ_6, θ_w. It is obvious that the choice of the desired temperature differences is to be agreed by the parameters of the limit cycle. In order to avoid the fixed parameters of the limit cycle by the method of iterations, with a fixed value of T_{ht}, you can use the dependencies shown in Fig. 2.

Determining the technical and economic characteristics of heat-using refrigeration machines, as well as any systems that convert energy, is associated with significant difficulties. First of all, it is necessary to have the initial cost indicators, which are not independent values themselves and are closely related to the nature of the load. Finally, cost indicators are valid for a short time and change with the connection of new consumers, with the progress of technology and other market factors.

In single-effect absorption systems, the total temperature difference θ_{tot} decreases with a decrease the evaporation temperature T_0 and the heating medium temperature T_{h1}. This leads to a decrease the individual devices temperature differences to increase the metal mass of the system. At the same time, the degassing zone decreases, which will lead to an increase in the consumption of heat, electricity and cooling medium.

As an example of calculating the optimal temperature differences in absorption chillers, we present the results of scientific research, conducted in the 1960s by the Department of Refrigeration Machines OIFRT under the guidance of Prof. Minkus B.A. and published in the scientific reports of the department. The calculations were carried out for large water-ammonia absorption systems with film shell-and-tube heat exchangers, heating medium with water at a temperature T_h = 102 °C, water for cooling the dephlegmator and condenser T_w = 25 °C and T_0 = 0 °C. It has been established that θ_2 = 10,7 deg; θ_4 = 4,1 deg; θ_6 = 7,2 deg; θ_w = 5,1 deg; θ_3 = 3,2 deg can be considered optimal temperature differences. For comparison, a machine with upgraded light-weight film heat exchangers and dephlegmator with nozzle, cooled by the solution was used. High energy efficiency in machines is achieved at different values of θ_{tot}, and $\theta_{tot2} > \theta_{tot1}$. Modernization of equipment has shown the possibility of significantly reducing metal consumption and capital investment, and increasing the energy efficiency of the system. Characteristics calculated according to a single method, if they do not correspond to the accuracy of absolute values, still quite objectively reflect the relative relationship between indicators and can be used to compare machines.

The transition to modern plate heat exchangers ensures an increase in the total temperature difference θtot and, as a result, a reduction in the mass of equipment and capital investment1 per 1 kW of installled heat power. Modern designs of plate heat exchangers for water/ammonia absorption systems are manufactured by AlfaLaval [8,9,10]. Using the above information, it is possible to evaluate the change in the general temperature indicators of installations only due to the modernization of heat exchangers.

In single-effect absorption systems, the total temperature difference θ_{tot} decreases with decreasing T_0 and T_h. This leads to a decrease in the temperature difference in individual devices θ_{im} and an increase in the metal mass of the system. At the same time, the degassing zone decreases, which will lead to an increase in the consumption of heat, electricity and cooling medium.

4. Heat exchangers temperature differences and degassing zone at the cycle

Temperature modes and degassing zone

With a different combination of initial temperatures in the absorption cycle, the degassing zone $\Delta x = (x_r - x_a)$ can take the following values: $\Delta x > 0$, but not less than 0,06, which allows the implementation of a single-effect cycle; $\Delta x = 0$; $\Delta x < 0$. A single-effect cycle cannot be carried out even theoretically with zero and negative values Δx. However, in modern conditions of low energy use of solar energy or the use of a cooling medium with an elevated temperature (tropical climate), such cases occur most often. The dynamics of the reduction of the degassing zone Δx under such temperature modes is shown in Fig. 4. The decrease in the degassing zone is determined by: a decrease the heating medium temperature

(Fig.4, cycle 1), an increase in the temperature and pressure of condensation (Fig. 4, cycle 2) and a decrease in the temperature and pressure of evaporation (Fig. 4, cycle3).

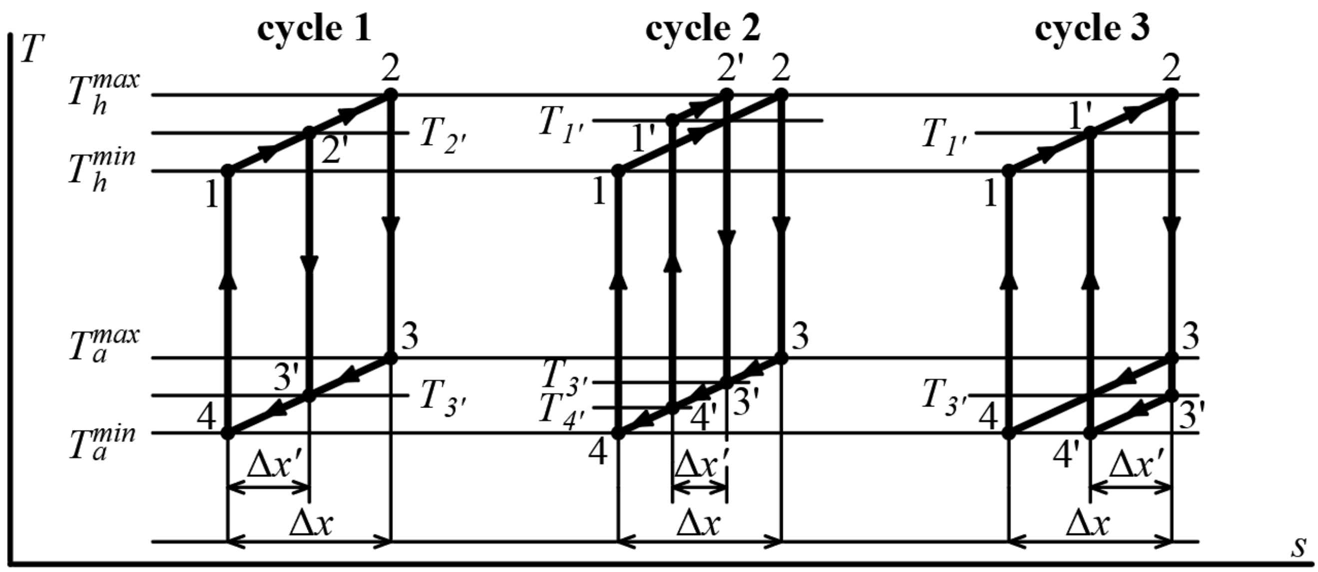

Figure 4 *– Dynamics of the reduction of the degassing zone: Cycle 1 – decrease in the temperature of the heating medium; Cycle 2 – an increase in the temperature of the cooling medium; Cycle 3 – decrease in the temperature of the cooling object*

For the practical implementation of cycles with the considered modern temperature regimes and solar powered of the generator, it is necessary to design absorption systems schemes "with an extended degassing zone". An increase in the degassing zone leads to a decrease in the solution circulation rate and an increase in the average temperature difference in the heat exchangers. Thanks to this, the heat transfer surface of devices, especially the solution heat exchanger, is reduced. As a result, the metal capacity of the installation decreases.

In the theoretical cycles of absorption systems, the refrigerant evaporation and condensation occur at constant temperatures, the processes with solutions in the absorber and generator – at variable temperatures. heat transfer processes between sources and working fluid are not considered (Fig. 5a).

In real conditions, the temperature of the cooled object may be variable, and the heating medium temperature may be constant. The discrepancy of temperature changes in the heat transfer process leads to an increase in the external irreversibility of the cycle.

Cycles with variable heating medium temperature

When the generator is heated and the absorber is cooled by sources with variable temperatures that are equidistant in relation to the temperature of the solution, minimal irreversible losses are observed, which are determined by the constant temperature difference θ_2 and θ_3 (Fig. 5 Cycle 2).

The efficiency of machines with generator heating with hot water, hot gas or other heat carriers, the temperature of which changes during heat transfer, depends on the degree of use of the available temperature difference θ_2. In single-effect absorption machines, the temperature difference between the hot and cold edges of the generator (Fig. 5, Cycle 2,) often turns out to be insignificant, and therefore it is rarely possible to perfectly use the available temperature range.

The use of combined machines will expand the degassing zone and thereby increase the temperature range in the generator. Especially importance is the cycles that reduce the initial boiling point T_{ht} of the solution. Such combined cycles are implemented at high absorption pressure, by pre-compressing the vapor in a booster compressor [5].

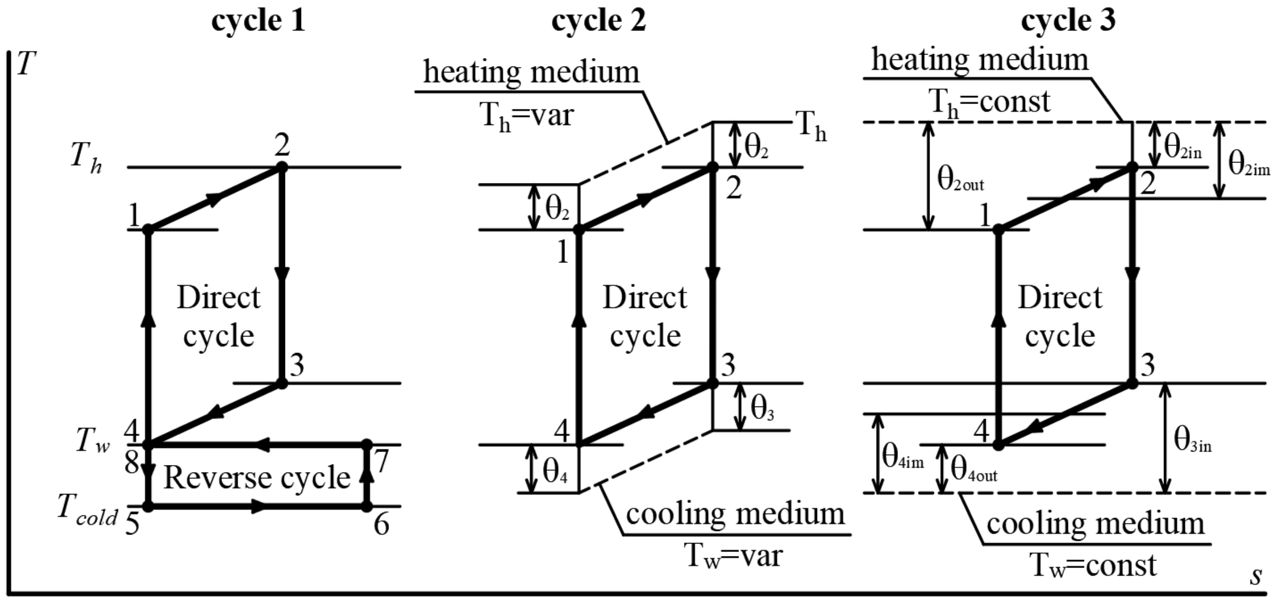

Figure 5 *– Heat transfer processes between sources and working fluid: Cycle 1 – theoretical cycle; Cycle 2 – cycles with variable heating and cooling mediums temperatures; Cycle 3 – cycle with constant heating and cooling mediums temperatures*

<u>Cycles with constant heating medium temperature</u>

In real conditions, the heating medium temperature can be constant (Fig. 5, Cycle 3). The effect is observed when heat carriers use substances in a two-phase state (condensed steam). Differences between θ_{2in} and θ_{2out} of the generator leads to an increase in external irreversibility. By creating a multi-effect generator, it is possible to give the generation process an isothermal character, and to reduce the average temperature difference θ_{2m}.

Similarly, when cooling the environment with a constant temperature, the external irreversibility of processes decreases with the introduction of a multi-effect absorber. Fig.6 shows the cycle of an absorption system with triple-effect generator and triple-effect absorber.

The multi-effect in the generator is created by the turbine-compressor unit. High-pressure vapors p_{g1}, received in the generator G_1, expand in the turbine to pressure p_{g2}, the turbine power is used to compress the vapors received in the G_3 from pressure p_{g3} to p_{g2}. As can be seen from Fig. 6, the introduction of a turbine unit allows, without increasing the consumption of heating source, to expand the degassing zone towards low concentrations.

Turning the turbine-compressor unit allows you to get cold at three temperature levels EV1, EV2, EV3 which is widely used in commercial systems such as supermarkets, etc. Problems in the systems are the ratio of the thermal power of the cooled objects and their temperature modes.

<u>Cycles with a constant temperature of the cooled object</u>

Heat removal at a constant temperature is the best operating conditions for any refrigeration systems based on the refrigerant phase transformation in the evaporator. However, at low evaporation temperatures, the degassing zone decreases, and the energy efficiency of the system decreases too.

In such cases, a cycle with several cooling objects, which can be implemented with a double-effect absorber, is highly effective. When the refrigerant vapor is absorbed in the high-pressure absorber, it is possible to expand the degassing zone in the direction of high concentrations and reduce the average temperature difference during the absorption process. The same effect can be obtained with one evaporator, booster compressor and absorber with double effect [5].

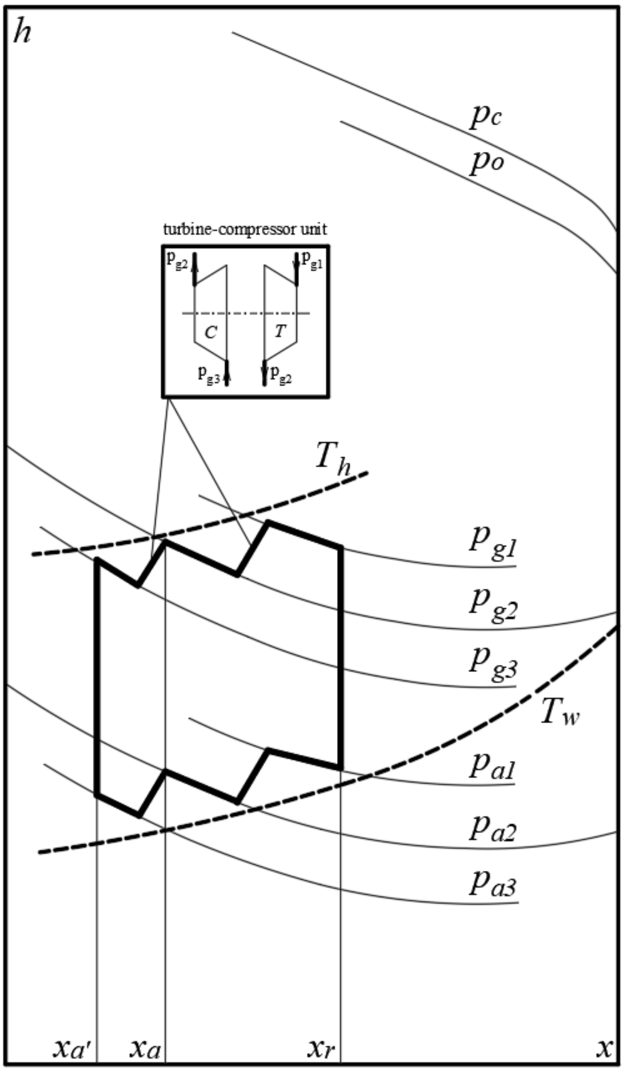

Figure 6 *– The cycle of an absorption system with the triple-effect generator .and triple-effect absorber*

Cycles with a variable temperature of the cooled object

When cooling liquid flows, the temperatures of which change during heat transfer, it is advisable to use absorption systems with absorbers double effect. However, direct cycles do not always agree well with the required temperature regime of reverse cycles in one system. In this case, the system works uneconomically. Coordination of the thermal power of the evaporator with the optimal distribution of the degassing zone and a decrease in the temperature difference in the absorption process is possible when using a booster compressor between the absorbers. When solving more complex problems, triple effect absorbers can be used (Fig. 6).

Cycles with low flow rates of cooling medium

When the flow rate of the cooling medium (water) is limited or an increased leaving water temperature is required, it is desirable to reduce the number of effects in the absorber and increase the number of effects in the condenser. The creation of such machines is not particularly difficult; they include combined machines with two cooling objects and one absorber cooled in series with the condenser. When designing a system with a minimum water flow, it should be borne in mind that the outlet water temperature Tw2 should not exceed the temperature Tht (Fig. 3)

5. Conclusions

Considering the temperature regime, the authors focus on the creation of cycles with evaporfting temperatures of -30...0 °C and several cooling objects, with coolant temperatures of 70...100 °C, which are directly related to solar installations of small energy. At the same time, the thermal power of absorption systems is consistent with the needs of commercial refrigeration.

The analysis demonstrating the possibility of implementing scheme-cycle design of absorption systems for completing a trigeneration system for small-energy. Practical implementation is connected with the choice of systems elements designs. The availability of modern designs of plate heat exchangers for water-ammonia absorption machines expands the nomenclature of heat-using refrigerating machines and the temperature range of the received cold.

The above method for calculating the temperature characteristics makes it possible to approximately determine the technical and economic indicators of absorption machines, which is very important for their further comparison with the indicators of hybrid systems based on simple effect machines.

In conclusion, we note that the considered examples do not exhaust the variety of practical problems and the possibilities of their solution by means of combining machines.

Aknowlegement

The authors of the article with deep respect and gratitude honor the memory of Doctor of Technical Sciences, Professor Boris Adolfovich Minkus, who gave his descendants a rich creative heritage.

CRediT author statement

Larisa Morosuk: Conceptualization, Formal analysis, Writing – Original Draft, Project administration. **Boris Kosoy**: Methodology, Writing – Review & Editing, Supervision. **Artem Kykoliev**: Investigation, Validation. **Sergii Psarov**: Funding acquisition, Data Curation. **Anatolij Basov**: Visualization, Resources, Software.

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Аналіз температурних режимів і температурних напорів у теплообмінниках абсорбційних холодильних систем на сонячній енергії

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Технології сонячного охолодження включають сонячну установку як джерело тепла та тепловикористальну холодильну машину для охолодження. У статті представлено аналіз температурних режимів та температурних напорів у теплообмінних апаратах абсорбційних холодильних систем з позицій забезпечення працездатності та високої ефективності процесів. Аналіз виконано на прикладі одноступеневої водоаміачної абсорбційної машини. В основу аналізу покладено умови: наявність граничного циклу працездатності системи, тобто взаємозалежність температурних потенціалів трьох джерел тепла, два з яких обрані довільно, третє – функція перших двох; температурні режими і температурні напори в апаратах надано у взаємодії з температурами зовнішніх джерел енергії. Встановлено наявність загального теспературного напору у системі, узгодженого з граничним циклом за відомих температурних рівнів джерел тепла. Загальний напір враховує: інтервал температур у генераторі, температурний напір на гарячому кінці генератора, температурний напір на холодному кінці теплообмінника розчинів, температурний напір на холодному кінці абсорбера – найменший, температурний напір у конденсаторі, підігрів води у кон-

денсаторі. Питання оптимізації температурних напорів в елементах системи запропоновано вирішувати шляхом розгляду змінної складової капітальних витрат, пов'язаної з вартістю кожного теплообмінника в залежності від температурного напору. Для практичного впровадження циклів з розглянутими температурними режимами та сонячним живленням генератора на засадах енергозбереження запропоновано цикли абсорбційних систем «з розширеною зоною дегазації». Цикли реалізовано за умови різного характеру зміни температур джерел енергії та робочої речовини у процесі теплообміну між ними. Вирішення проблем рекомендовано шляхом створення складних схем та циклів гібридних водоаміачних холодильних машин.

Ключові слова: *Технологія сонячного охолодження; Водоаміачна абсорбційна машина; Теплообмінник; Температурний напір; Зона дегазації*

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