The search for new and improvement of existing technical design of energy converter systems for specific consumers requires a reasonable choice of the most rational design for these objects.

Thermotransformers that operate on the reverse and mixed thermodynamic cycles, in combination with power plants utilizing renewable and non-traditional primary energy (fuel), are considered to be of interest for small-scale power generation (trigeneration systems), which is consistent with the concept of distributed energy generation.

Cold in trigeneration systems is provided by heat-using thermotransformers.

This paper reports a method for synthesizing a scheme-cycle designs of absorption water-ammonia thermotransformers that utilize renewable heat sources with a low-temperature potential of 90–250 °C.

A "cycle method" was applied to perform the thermodynamic analysis of the cycle of simple absorption thermotransformers with the expansion of the degazation zone with an increase in the temperature of the heating source; the technological schemes for the corresponding cycles have been substantiated.

The influence of changing the degazation zone on the energy efficiency of the machine has been established. A scheme-cycle designs of the thermochemical compressor for a thermotransformer with a return supply of solutions to the generator and absorber at "excess temperatures" has been proposed, as a way to improve the cycle energy efficiency.

A comparative analysis of the degree of thermodynamic perfection of the considered cycles has been performed using a specific example.

The thermodynamic analysis demonstrated that the practical implementation of the scheme-cycle designs "with excess temperatures" could provide energy-saving conditions in small-scale trigeneration systems

Keywords: thermodynamic analysis, "cycle method", absorption water-ammonia thermotransformer, degazation zone, excess temperature

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SYNTHESIS OF SCHEME-CYCLE DESIGNS OF ABSORPTION WATER-AMMONIA THERMOTRANSFORMERS WITH EXTENDED DEGAZATION ZONE

Boris Kosoy
Doctor of Technical Sciences, Professor,
Director of Institute
V. S. Martynovsky Institute of Refrigeration,
Cryotechnologies and Ecoenergetics**
Larisa Morozyuk
Corresponding author
Doctor of Technical Sciences, Professor*
E-mail: lara.morozyuk@mail.ru
Sergii Psarov
Postgraduate Student*
Artem Kukoliev*
*Department of Cryogenic Engineering**
**Odessa National Academy of Food Technologies
Kanatna str., 112, Odessa, Ukraine, 65039

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1. Introduction

The estimate of the thermal energy lost in various sectors of energy and industry is high enough. It is possible to save most of the thermal energy by improving energy-converting systems, increasing the reversibility of production processes and heat transformation.

The search for new and improvement of existing technical designs of energy converter systems for specific consumers requires a reasonable choice of the most appropriate ones for these objects. As regards the relevant technological schemes, it is necessary to take into consideration that such a choice should provide for the possibility to take into account a series of factors such as the scale of production, its energy, economic, and social potentials. Distributed electricity generation implies arranging electricity sources in close proximity to consumers. It becomes possible for the consumer to use not only the electricity generated, but thermal energy as well, for the needs of heating, hot water supply, and refrigeration supply to the owner of the power plant and third-party consumers.

That is confirmed by the scientific and technical information on the development and analysis of small-scale energy trigeneration systems [1, 2]. The rise of distributed generation, smart grids, and storage technologies are rapidly changing the way energy is used [3].

Modern requirements related to the environmental purity of working substances in *thermotransformer* thermotransformers operating on the reverse and mixed cycles indicate that absorption *thermotransformer*s are in demand on a par with compressor t *thermotransformers* .

Absorption thermotransformers (ATTs) have many positive qualities, which can be implemented in actual installations by involving new methods of thermodynamic analysis of cycles. Absorption thermotransformers have again become the object of thermodynamic study; moreover, experts have recently paid great attention to special types of AHTs. New methods of thermodynamic analysis make it possible to design AHTs taking into consideration energy saving while satisfying the social conditions of the consumer at the conversion coefficient COP=0.3...0.8. That indicates that the research topics related to scheme-cycle designs for absorption thermotransformers are relevant.

2. Literature review and problem statement

Absorption thermotransformers have a century-old history, their development paths were quite complex and contradictory at different times of the formation of refrigeration technology.

Working substances in absorption machines that are known and promising are still an aqueous solution of lithium bromide and a water-ammonia solution. The theory and development of lithium-bromide devices have reached their limit while practical use is limited to air conditioning and the production of process water; this is described in work [4]. The prospects for the development of absorption water-ammonia refrigeration machines in modern conditions are outlined in paper [5].

Everything new and promising in absorption refrigeration machines and heat pumps was earlier reported by scientists from different countries and scientific schools. The study of two low-temperature heat-using absorption-compressor refrigeration cycles based on the analysis of energy consumption, economy and ecology is given in work [6]. In [7], the authors investigate the influence of the temperature of the heating source on the energy efficiency of small absorption water-ammonia refrigeration systems. Work [8] estimated the energy efficiency of the absorption-resorption refrigeration machine in the trigeneration system of small-scale energy generation. The development of a combined absorption cycle to produce electricity and for cooling is described in [9].

A new scientific approach to the exergy analysis of absorption refrigeration machines using a “cycle method” is set out in work [10]. Multifaceted research is the scientific basis for all further work in the field of absorption technology. In [11], the authors investigate absorption-diffusion refrigeration machines from the point of view of improving energy efficiency. Paper [12] considers scheme-cycle designs with an expansion of the degazation zone by techniques that correspond to the thermodynamic cycles of absorption-diffusion machines.

The wide range of the proposed schemes was made possible due to the deep thermodynamic analysis and synthesis of the processes occurring in them, set forth in fundamental works [13–15]. Paper [13] addresses the issues related to the design of absorption thermotransformers, in particular schemes and cycles in thermochemical compressors. Work [14] tackles the energy efficiency of cycles of thermotransformers. Paper [15] reports procedures for calculating the characteristics of absorption machines and individual elements in their technological schemes.

The studies in the above works were aimed at ensuring the design of large (up to 5,000 kW of unit capacity) absorption plants. They implied the use of waste heat of the basic technological processes of own generation, as well as water as an intermediate heat carrier, in heat removal systems. Another area of design was household refrigeration absorption equipment (up to 1,000 W of unit power) with power supply and air cooling of heat exchangers. The development of autonomous small-scale energy equipment involving renewable sources of primary energy required the construction of refrigeration absorption units of a new generation: small and medium productivity. It was envisaged to expand the temperature intervals of the cold production, the use of air heat exchangers, a wide variety of climatic conditions. Previous scientific approaches to design did not meet the needs of individual consumers. The emergence of new methods of thermodynamic analysis of actual cycles makes it possible to design absorption thermotransformers, taking into consideration the requirements of modern consumers.

3. The aim and objectives of the study

The aim of this study is to find a method to synthesize a scheme-cycle design of an absorption water-ammonia thermotransformer using heat sources with a low-temperature potential of 90–250 °C, based on a comparative assessment of thermodynamic cycles with an extended degazation zone. That could make it possible to establish the practical feasibility of introducing and implementing new technical designs.

To accomplish the aim, the following tasks have been set:

– to conduct thermodynamic analysis of cycles with the expansion of the degazation zone at an increase in the temperature of the heat source;

– to conduct thermodynamic analysis of the cycle with excess temperatures;

– to conduct a comparative energy analysis of cycles with an extended degazation zone using a specific example.

4. The study materials and methods

The object of this study is the absorption water-ammonia thermotransformers under the trigeneration modes with a low temperature of the heat source and the expansion of the degazation zone. The research method is the thermodynamic analysis of processes in a thermochemical compressor and their numerical modeling. The basis of mathematical modeling is the equations of classical thermodynamics.

5. The results of synthesizing the circuit-cycle solutions of absorption water-ammonia thermotransformers with an extended degazation zone

5. 1. Thermodynamic analysis of cycles with an expansion of the degazation zone when the temperature of the heat source rises

The complex cycle of any absorption thermotransformer consists of two simple cycles, direct (power) and reverse (refrigeration).

The direct cycle is associated with a scheme design known as a “thermochemical compressor”. The structural scheme includes a generator, an absorber, a throttle valve, a pump, and any heat and mass transfer devices on the lines of strong and weak solutions (Fig. 1).

The reverse cycle is called the “main process”; the structural elements of the scheme include a condenser, a throttle valve, an evaporator, as well as heat exchangers on the refrigerant line.

In this case, the main process of absorption and compressor thermotransformers coincides. Of considerable interest is the multitude of scheme-cycle designs of the thermochemical compressor.

Below, we considered in detail the simple absorption thermotransformer as it has been widely applied in various industries, including large-and small-scale energy generation.

For absorption thermotransformers (ATTs) to implement a thermodynamic cycle, it is necessary to have three energy sources of different temperature potentials:

– high, for heating the generator, T_{hot};

– medium; the medium for heat removal in the condenser and absorber, T_{amb};

– low, for cooling the object, T_{cold}.

The three temperature levels in single-stage ATTs are interconnected so that only two can be chosen arbitrarily, the third is always a function of any two [10].

The dependence of temperature T_{hot} can be represented as a function of temperatures T_{cold} and T_{amb} using the following equation [9].

$$T_{hot} = 264 + \left(38{,}8 - 0{,}13T_{cold}\right)^2 + \left(4{,}0 - 6{,}7\cdot 10^{-3}T_{cold}\right)\left(T_{amb} - 273\right), \text{K}. \quad (1)$$

The values of temperatures T_{hot}, shown in the chart in Fig. 2, are valid in the temperature range T_{cold}=223...293 K and T_{amb}=283...313 K.

Fig. 2 highlights a temperature field corresponding to the practical implementation of a single-stage absorption thermotransformer.

In single-stage ATTs, there are two pressure levels determined by the conditions of the main process: condensing pressure p_c and evaporating pressure p_e. Conditionally accepted is the equality of evaporator pressure and absorber pressure, p_e=p_a, and the equality of condenser and generator pressures, p_c=p_g. The temperature of the heating source determines the final boiling point of the solution in a generator, T_2. The ambient temperature T_{amb} determines the final absorption temperature T_4.

The final boiling point of the solution, T_2, and the generator pressure, p_g, determine the concentration of the weak solution, x_a. The final absorption temperature T_4 and pressure p_a determine the concentration of the strong solution x_r in the cycle of a thermochemical compressor. The concentration of refrigerant x_d in the main process can be correctly taken to be equal to unity.

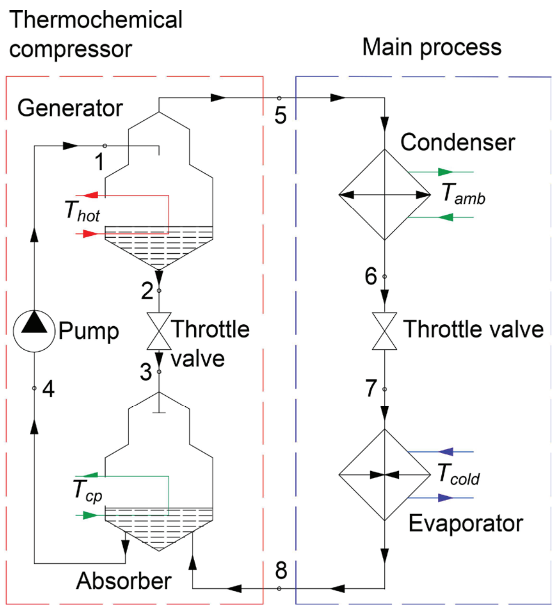


Fig. 1. Simple absorption thermotransformer

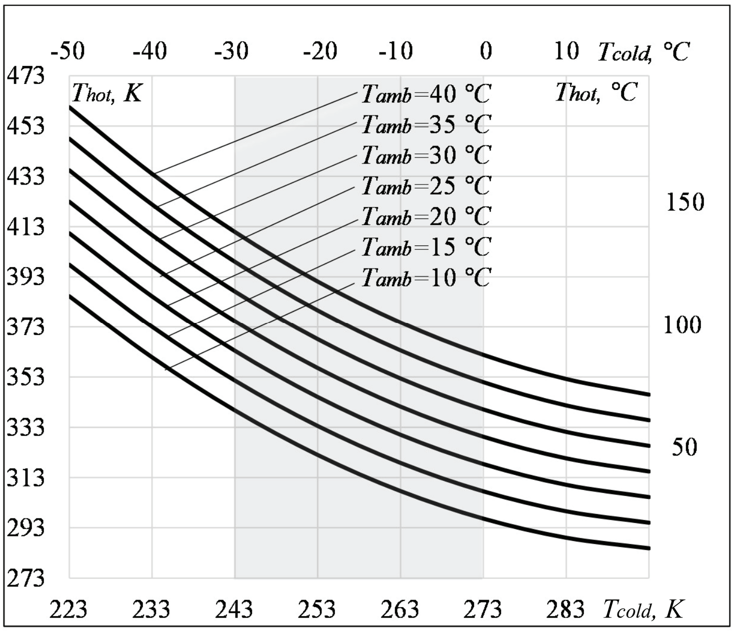


Fig. 2. Theoretical minimum temperature T_{hot} depending on temperatures T_{cold} and T_{amb}

The energy balance of an absorption thermotransformer

$$Q_G + Q_E = Q_A + Q_C, \quad (2)$$

where Q_G and Q_E is the heat supplied in the generator and evaporator, respectively; Q_A and Q_C is the heat removed in the absorber and condenser, respectively.

The energy balance attributed to 1 kg of the refrigerant circulating in the main process:

$$q_g + q_e = q_a + q_c. \quad (3)$$

We shall also attribute the mass flow rate of the solution through the thermochemical compressor to 1 kg of refrigerant in the main process.

$$f = F / D, \quad (4)$$

where F is the mass flow rate of a strong solution, kg/s; D is the mass flow rate of the refrigerant, kg/s.

The value of f affects the efficiency of any ATT operation; it is termed the multiplicity of circulation and can be calculated as

$$f = \frac{x_d - x_a}{x_r - x_a}. \quad (5)$$

The denominator of this fraction is the degazation zone (the difference in the concentrations of strong and weak solutions)

$$\Delta x = x_r - x_a. \quad (6)$$

With a different combination of initial temperatures in the cycle of a thermotransformer, the value of the degazation zone Δx can take the following values:

– Δx>0, but not less than 0.06, which is allowed for the practical implementation of the simplest cycle of single-stage ATT [13];

– $\Delta x=0$;

– $\Delta x<0$.

With zero and negative values of the magnitude of the degazation zone Δx, the cycle of a single-stage ATT is not feasible even theoretically. However, such cases are most common under modern conditions when using renewable energy sources as a heating source or using a cooling medium with high temperatures.

The solar, geothermal, and other types of renewable and non-traditional energy for the drive of a heat-using thermotransformer are characterized by a low-temperature level. For the conditions of Eastern Europe, the temperature does not exceed 100 °C. From the point of view of "large-scale" energy, it has been proven that heat with a temperature level of 70 °C is no longer operational, and its emissions into the atmosphere are environmentally safe. In the "small-scale" energy sector, the use of such energy resources is very important.

The narrowing of the degazation zone in the considered temperature modes of ATT operation is illustrated in Fig. 3. The cycles are shown in the coordinates "enthalpy-concentration".

The three heat sources in single-stage ATTs are interconnected so that only the temperatures of the two can be chosen arbitrarily. The third source is always a function of either two. An actual ATT differs from a theoretical one by the necessary set of auxiliary elements that make it possible:

– "to increase the reversibility of processes";

– and/or reduce the consumption of the heating source, cooling medium;

– and/or reduce the metal intensity, by combining several processes in one element, intensifying them.

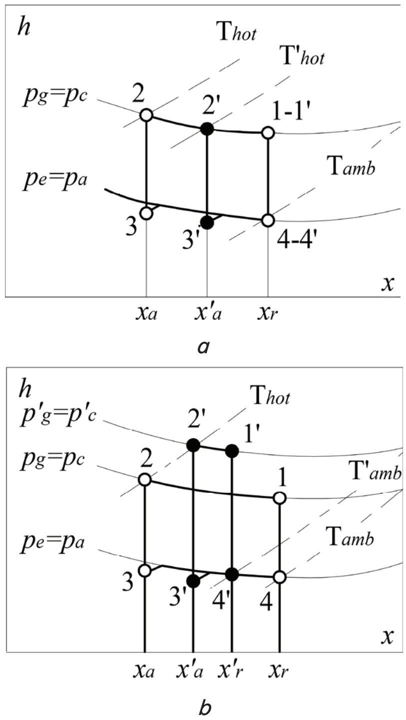


Fig. 3. Dynamics in the narrowing of the degazation zone in a thermochemical compressor: *a* – lowering the temperature of the heating source $\left(T'_{hot} < T_{hot}\right)$; *b* – increasing ambient temperature $\left(T'_{amb} > T_{amb}\right)$

A variety of circuit solutions for water-ammonia ATTs follows the path of improving the thermochemical compressor while the main process (reverse cycle) remains traditional.

A thermodynamic analysis is the first step in the design process. Based on its results, a would-be system turns out to be operational, energy-efficient, or its further advancement is impractical.

In international practice, the quantity that quantitatively describes the irreversibility in an energy-converting system, in general, is thermodynamic effectiveness of the cycle, η_{TEFF}.

The term implies the value of the coefficient of performance COP_{theor} of the theoretical cycle of the thermotransformer in question relative to the coefficient of performance of the corresponding Carnot cycle $\text{COP}_{\text{Carnot}}$ for this system

$$\eta_{TEFF} = \frac{COP_{\text{theor}}}{COP_{\text{Carnot}}}. \tag{7}$$

This quantity is also known as "cycle carnotization" since the Carnot cycle is recognized as a universal cycle model [14, 15]; both heat sources were conditionally accepted with constant temperatures (Fig. 4, *a*). The values of the energy coefficient of performance COP in the direct and reverse Carnot cycles are mathematically notated as

$$COP_{\text{Carnot}} = \frac{T_{\text{hot}} - T_{\text{amb}}}{T_{\text{hot}}}, \tag{8}$$

$$COP_{\text{Carnot}} = \frac{T_{\text{cold}}}{T_{\text{amb}} - {}_{\text{cold}}}. \tag{9}$$

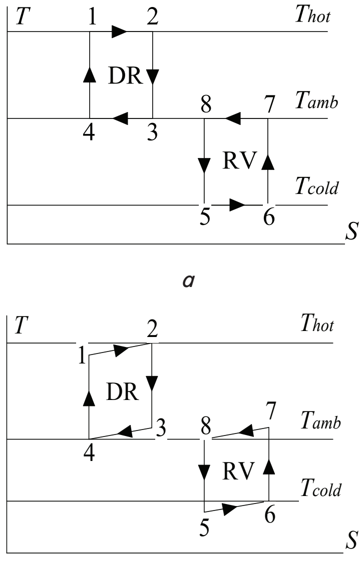


Fig. 4. Cycle models: a – Carnot; *b* – Lorenz

The choice of the Carnot cycle as a cycle model affects the conclusions from the thermodynamic analysis. In the technical literature, there are works that offer the choice of another cycle model, which is the closest to the working fluid and processes in thermotransformers [14].

In practice, there are heat sources that change their temperature during heat exchange. To meet the conditions of reversibility of the heat exchange process, the temperature of the working fluid must change in the same way as the temperature of the source. In this case, the Lorentz cycle can be used as a model: two iso-entropic processes and two polytropic processes with variable temperatures, whose nature exactly follows the change in the temperatures of the sources (Fig. 4, *b*).

The values of the energy coefficient of performance in the direct and reverse Lorentz cycles are mathematically notated as follows [14]

$$COP_{\text{Lorenz}} = 1 - \left(T_3 / T_2\right), \tag{10}$$

$$COP_{\text{Lorenz}} = \frac{1}{\left(T_7 / T_6\right) - 1}. \tag{11}$$

Mixtures consisting of polar components (for example, a water-ammonia solution) are characterized by high heat of mixing. The processes of heat supply and removal, accompanied by phase transformations, occur at a variable temperature. Thus, the Lorentz cycle can serve as a model for the cycles of water-ammonia absorption thermotransformers, while the processes of heat supply and removal are carried out at constant isobaric heat capacities.

Heat-using thermotransformers carry out a direct cycle "themselves" while the work of the direct cycle is directly used in the reverse one. Based on such a statement, the representation of two independent ideal cycle model corresponding to the direct and reverse cycles is widespread. For reversible cycle models, any combination of the thermodynamic cycles discussed above can be selected. A prerequisite is the equality of the work produced and utilized.

The combination of the two cycles is used for the absorption thermotransformer in Fig. 5. With an unlimited multiplicity of circulation in the generator and absorber, the processes can be considered isothermal, and, for a direct cycle, the Carnot cycle can be used (Fig. 5, cycle *A*). With a limited multiplicity of circulation, the processes in the generator and absorber are non-isothermal. Non-isothermality must necessarily be reflected in the Lorentz cycle, even with a simple representation of the cycle (Fig. 5, cycle *B*). In the reverse cycle, the non-isothermality in the processes of heat supply and removal is minimized or missing at all, so the reverse cycle can be described by the Carnot cycle. The effectiveness of a complex cycle is assessed as

$$COP_{\text{model}} = COP_{DR} \cdot COP_{RY}. \tag{12}$$

Using cycle models, we have synthesized a scheme-cycle design of the absorption machine with the expansion of the degazation zone by increasing the temperature of the heat source. The tool of thermodynamic analysis is the "cycle method" [15]. As a result, the ideal reversible cycle model and the corresponding ideal machine circuit are transformed into an actual scheme-cycle design; its energy efficiency is then evaluated.

Stage one. Considering the above, the direct Lorentz cycle 1234 was chosen as the model for the thermochemical compressor, and the reversed Carnot cycle 5678 was chosen for the main process as they are related through the equality of work (Fig. 5, cycle *B*).

Stage two. The existence of a single working substance in the direct and reverse cycles determines the complex cycle 12345678 (Fig. 5, cycle *C*), formed from the characteristic fragments of known independent cycle patterns. In this case, it is necessary to take into consideration the circulation multiplicity and the degazation zone of the solution in the direct cycle 1234 of the thermochemical compressor.

Stage three. The ambient temperature T_{amb} and the temperature of the cooled object T_{cold} for the reverse cycle 5678 remain constant (cold production conditions). Then the temperature of the heating source T_{hot} in the direct cycle 1234 (Fig. 5, cycle *D*1) would determine the multiplicity of circulations of the solution *f* and the degazation zone in the cycle. If the temperature of the heating source is lower than the corresponding value in equation (1), the cycle of a single-stage absorption machine cannot be carried out. Cycle *D*1 is operable with a minimum degazation zone according to equation (6). In this case, the condition is met: the initial boiling point T_1 is higher than the initial absorption temperature T_3. To improve the energy efficiency of such a direct cycle, heat recovery is used [13].

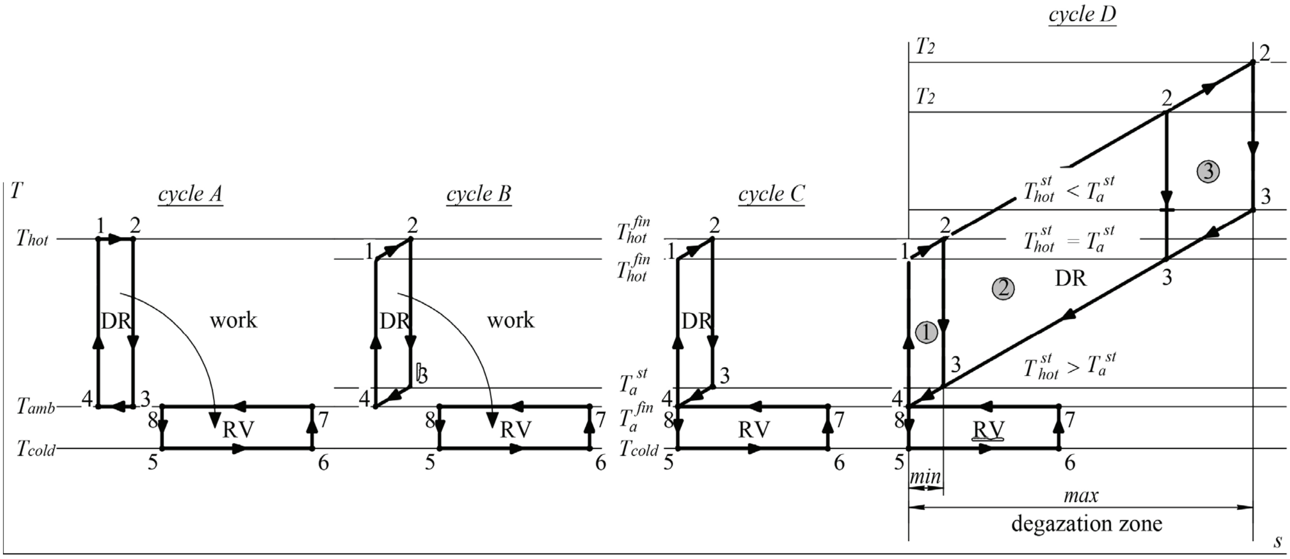


Fig. 5. "Cycle method" in the synthesis of an absorption thermotransformer

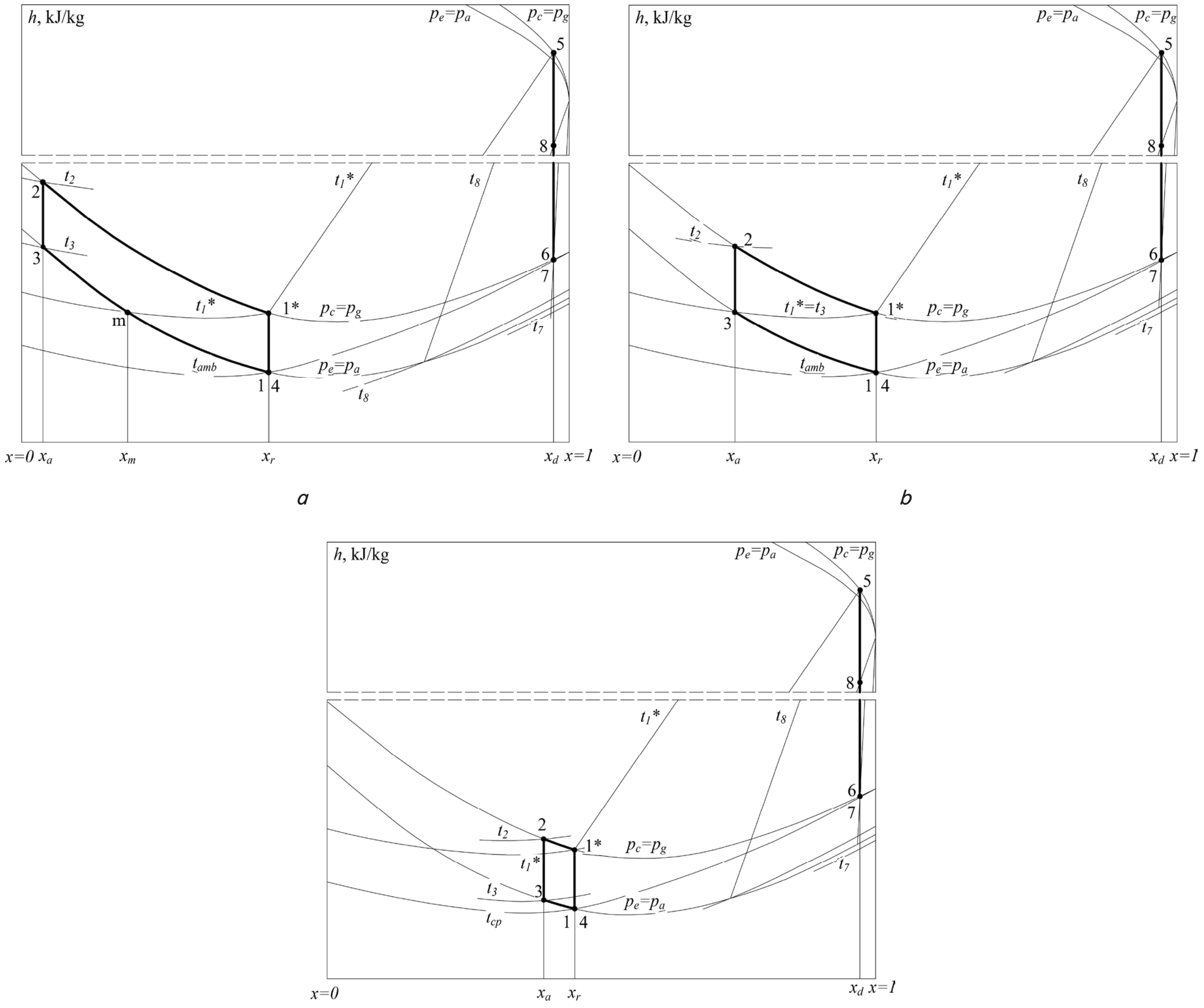
Stage four. An increase in the temperature of the heating source T_2 is accompanied by an expansion of the degazation zone of the solution with a corresponding decrease in the concentration of the weak solution (Fig. 5, cycle *D*2). There is a mode when the initial absorption temperature $T_{3''}$ and the initial boiling point in the generator T_1 become equal. Under such conditions, it is impossible to use heat recovery in a thermochemical compressor as a means of improving the efficiency of the cycle [13].

Stage five. With a further increase in the temperature of the heating source T_2, it is possible to reduce the concentration of the weak solution x_a that exits the generator so much that the initial absorption temperature T_3 would be higher than the initial boiling point T_1 in the generator (Fig. 5, cycle *D*3). In this case, it is possible to introduce into a direct cycle the process of "excess temperatures", that is, the "hot" end of the absorber should heat the "cold" end of the generator [13]. Such an additional process would lead to a decrease in the consumption of the heating source in the generator and the cooling medium in the absorber, which could improve the energy efficiency of AHT. For example, a direct cycle can be complicated by the process of "reverse" supply of the solution through the absorber and generator or by two cycles of the solution.

Stage six. Next, the analysis is supplemented with the actual working substance, the parameters of the working substance, and the state diagram "enthalpy–concentration".

The analysis is based on determining the characteristics for the theoretical cycle of a single-stage water-ammonia AHT with the expansion of the degazation zone. The boundary conditions are the minimum degazation zone, equal to $\Delta x=0.06$, and the highest temperature of the heating source $T_2=180$ °C. The boundary conditions were determined from the conditions of the practical operation of the machine and the thermophysical properties of ammonia [13, 15]. The cycles in the *hx* diagram are shown in Fig. 6.

The ambient temperature, $T_{\text{amb}}=30$ °C, and the temperature of the cooled object, $T_{\text{cold}}=-10$ °C, remain constant during the analysis and correspond to the actual operating modes of the machine.



a *b* *c*

Fig. 6. Cycles in the *hx* enthalpy–concentration diagram: *a* – cycle $\Delta x=0.06$; *b* – cycle $T_{1*}=T_3$; *c* – cycle $T_{1*}<T_3$

The estimation model for analysis:

– the multiplicity of circulation of a strong solution

$$f = \frac{x_d - x_a}{x_r - x_a}; \tag{13}$$

– the heat supplied in a generator,

$$q_g = (h_5 - h_2) + f(h_2 - h_1),\ \text{kJ/kg}; \tag{14}$$

– the heat removed in a condenser

$$q_c = h_5 - h_6,\ \text{kJ/kg}; \tag{15}$$

– the heat supplied in an evaporator,

$$q_e = h_8 - h_7,\ \text{kJ/kg}; \tag{16}$$

– the heat removed in an absorber,

$$q_a = (h_8 - h_3) + f(h_3 - h_4),\ \text{kJ/kg}, \tag{17}$$

where h is the enthalpy at the nodal points of the cycles according to Fig. 6;

– the energy balance of the machine

$$q_g + q_e = q_c + q_a,\ \text{kJ/kg}; \tag{18}$$

– the coefficient of performance of the theoretical cycle

$$COP_{\text{theor}} = q_e / q_g; \tag{19}$$

– the coefficient of performance of the Carnot-Carnot cycle model

$$COP_{CC\,\text{mod}el} = \frac{T_{\text{hot}} - T_{\text{amb}}}{T_{\text{hot}}} \cdot \frac{T_{\text{cold}}}{T_{\text{fmb}} - {}_{\text{cold}}}; \tag{20}$$

– the coefficient of performance of the Lorenz-Carnot cycle model

$$COP_{LC\,\text{mod}el} = \left(1 - \frac{T_3}{T_2}\right) \cdot \frac{T_{\text{cold}}}{T_{\text{amb}} - {}_{\text{cold}}}; \tag{21}$$

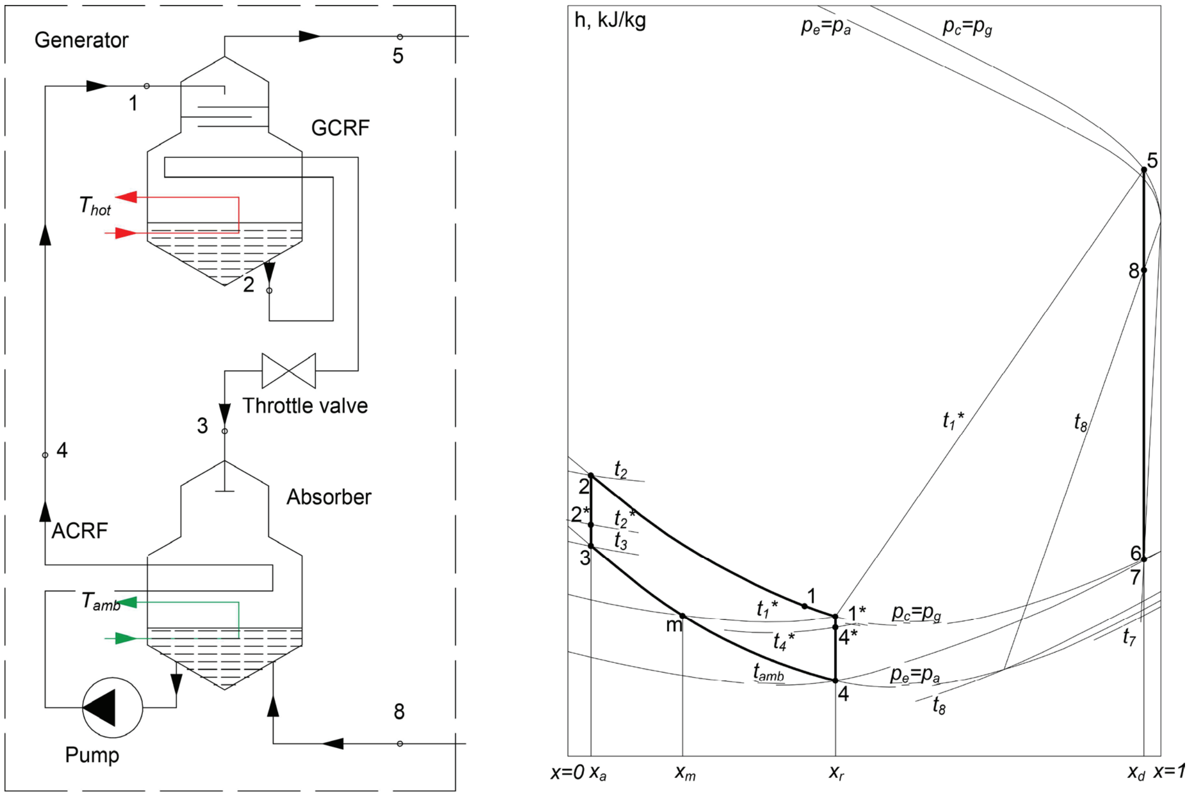
– the thermodynamic efftctiveness of the cycle

$$\eta_{TEFF} = \frac{COP_{\text{theor}}}{COP_{\text{mod}el}}. \tag{22}$$

The results obtained when solving (20) to (22) will make it possible to estimate the correctness of the choice of cycle model for the thermodynamic analysis of an absorption thermotransformer using a "cycle method".

5. 2. The thermodynamic analysis of a cycle with "excess temperatures"

Consider the scheme-cycle variant of a water-ammonia machine operating at excess temperatures, using the reverse supply of the solution through the generator and absorber [13]. The scheme and cycle are shown in Fig. 7.



a

b

Fig. 7. The scheme and cycle of a thermochemical compressor with a reverse supply of solution to the generator and absorber: *a* — technological scheme; *b* — cycle in the *h-x* diagram

The absorber pump delivers a strong solution to the coil of the reverse feed of the absorber ACRF. The heat of absorption is transferred to a strong solution, as a result of which the solution is heated to T_{1^*} and enters the generator. The end of the process is determined by point 1. A weak solution in state 2 through the coil of the reverse supply GCRF is returned to the generator, while its temperature is reduced to T_{2^*}. Next, through the throttle, the solution enters the absorber, where it absorbs vapor (point 8) from the evaporator. In the executed cycle, it is not necessary to use a traditional heat exchanger of solutions since heat exchange is carried out by the return supply of solutions.

In a cycle with excess temperatures (Fig. 7), the vapor from the generator (point 5) comes out in equilibrium to the solution in the state at point 1.

In a cycle with excess temperatures, a strong solution fully perceives the transferred heat.

A strong solution can be heated to a temperature of T_{4^*} (point 4*) using the absorption heat, which is released in the process 4–*m*. At the same time, the temperature should be practically 5–8 degrees below the temperature of point m. The change in the state of the solution in the process 4*–1 is the result of the supply of absorption heat, which is removed in the 3–*m* process.

The energy analysis of the cycle with excess temperatures is carried out in accordance with the scheme shown in Fig. 7.

The input parameters of the cycle remain the same:
- the ambient temperature T_{amb}, °C=30 °C;
- the cooled object temperature T_{cold}=10 °C;
- the heating source temperature T_{hot}=180 °C.

The derived parameters at the nodal points of the cycle

The enthalpy and concentration of the solution at point (*m*) are defined as the parameters of the point of intersection of the isotherm T_{1^*} and the isobar p_e. The amount of vapor absorbed in the process (3–*m*) is determined from the material balance of the absorber. The absorber receives a weak solution in the amount of $(f-1)x_a$ kg and 1 kg of vapor with a concentration of x_d. In the process, g_m kg of vapor with a concentration of x_d is absorbed to point (*m*), the amount of which with the parameters at point (*m*) is determined from the following equation:

$$g_m = (f-1)(x_m - x_a)/(x_d - x_m),\ \text{kg}. \tag{23}$$

The value of the thermal load of the absorber on section (3–*m*) is determined as

$$q_{a2} = g_m h_8 + (f-1)(h_3 - h_m) - g_m h_m,\ \text{kJ/kg}. \tag{24}$$

The heat perceived by a strong solution in the process 4–4*, is calculated from the following formula

$$q_{a1} = f\left(h_{4^*} - h_4\right),\ \text{kJ/kg}. \tag{25}$$

The solution is partially evaporated when reversed in the absorber and enters the generator in the form of wet vapor. The solution enthalpy h_1 is defined as

$$h_1 = h_{4^*} + q_{a2}/f,\ \text{kJ/kg}. \tag{26}$$

The estimation model for analysis:
- the multiplicity of solution circulation

$$f = \frac{x_d - x_a}{x_r - x_a}; \tag{27}$$

- the heat removed in a condenser,

$$q_g = h_5 - h_6,\ \text{kJ/kg}; \tag{28}$$

- the heat supplied in an evaporator,

$$q_e = h_8 - h_7,\ \text{kJ/kg}; \tag{29}$$

- the energy possessed by a weak solution,

$$q_{ws} = (f-1)(h_2 - h_3),\ \text{kJ/kg}; \tag{30}$$

- the heat removed with return supply in a generator,

$$q_{grem} = q_{ws} - f\left(h_{1^*} - h_4\right),\ \text{kJ/kg}; \tag{31}$$

- the state of a weak solution coming out of the generator

$$h_{2^*} = h_2 - q_{grem}/(f-1),\ \text{kJ/kg}; \tag{32}$$

- the heat supplied to the generator from an external source

$$q_g = h_5 - h_{2^*} + f\left(h_{2^*} - h_{1^*}\right),\ \text{kJ/kg}; \tag{33}$$

- the coefficient of performance of the theoretical cycle

$$COP_{\text{theor}} = q_e/q_g. \tag{34}$$

At a high temperature of the heating source in the absorption machine, a direct cycle can be carried out using complete heat recovery, which is the system with excess temperatures.

In the absence of complete heat recovery, the irreversible energy loss of the heating source is similar to a single-stage absorption machine without a solution heat exchanger.

5. 3. Comparative energy analysis of cycles with extended degazation zone using a particular example

As an example, the comparative analysis of the cycles depicted in Fig. 6, 7 is shown.

The estimated input data are given in Table 1.

Table 1

Input parameters

Input parameter name	Designation	Measurement unit	Cycle Δx=0.06	Cycle $T_{1^*}=T_3$	Cycle T_2=180 °C	Excess temperatures cycle
Ambient temperature	*T*	°C	30	30	30	30
Cooled object temperature	*T*	°C	–10	–19	–10	–10
Heating source temperature	*T*	°C	95	144	180	180
The temperature of boiling in a generator	*T*	°C	83	83	83	83

The temperature regime corresponds to the actual operating conditions of the refrigeration machine in a fruit storage facility. The maximum generation temperature is determined by the operating conditions of the waste heat of a small power plant. The minimum theoretical temperature of generation, in accordance with (1), is 83 °C. To simplify the calculations, the external irreversibility in the processes of heat supply and removal is taken to be zero.

The calculations of the parameters and characteristics of cycles (Fig. 6) were performed in accordance with (13) to (22); the characteristics of the cycle (Fig. 7) – in accordance with (23) to (34). The calculation results are given in Table 2.

Table 2

Calculation results

Designation	Measurement unit	Cycle Δx=0.06	Cycle $T_{1*}=T_3$	Cycle T_2=180 °C	Excess temperature cycle
f	kg/kg	10.1	3.18	2.30	2.30
q_g	kJ/kg	4,068	2,695	2,654	2,375
q_o	kJ/kg	1,025	1,025	1,025	1,025
q_k	kJ/kg	1,340	1,340	1,340	1,340
q_a	kJ/kg	4,398	2,843	2,760	2385
COP_{theor}	–	0.25	0.36	0.38	0.43
$COP_{CC\,\mathrm{mod}\,el}$	–	1.16	1.79	2.12	2.12
$COP_{LC\,\mathrm{mod}\,ek}$	–	0.95	0.96	0.97	0.97
η_{TEFF}^{KCC}	–	0.22	0.20	0.18	0.20
η_{TEFF}^{LLC}	–	0.263	0.375	0.391	0.443

In graphical form, the comparative analysis is shown in Fig. 8.

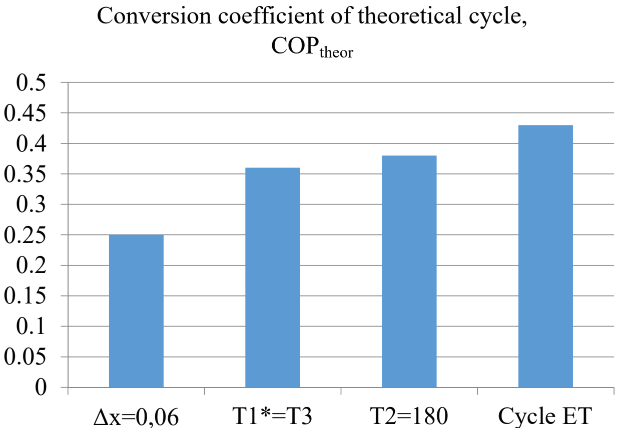


Fig. 8. The coefficient of performance of the theoretical cycle COP_{theor} with extended degazation zone

Fig. 9 shows the coefficient of performance for the cycle models.

The thermodynamic effectiveness of the studied cycles is shown in Fig. 10.

As our calculations have demonstrated, the coefficient of performance of the corresponding Carnot cycle doubles as the temperature of the heating source increases within a given temperature range. Under the same conditions, in the Lorenz cycle, energy efficiency remains constant. Such results were substantiated in detail in work [14]. Thus, the Lorenz cycle is correctly chosen as a sample for a thermochemical compressor.

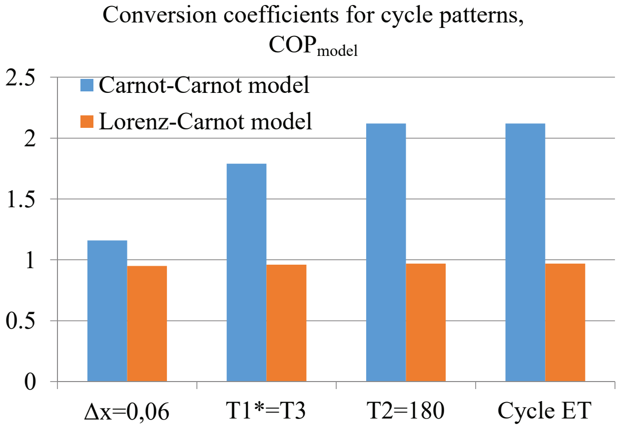


Fig. 9. The coefficient of performance for the cycle models with extended degazation zone

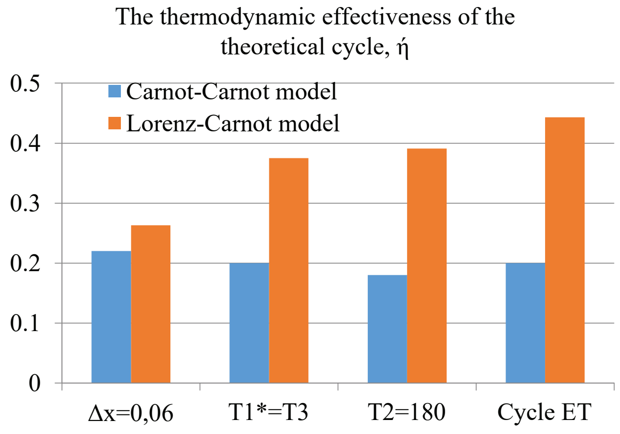


Fig. 10. The thermodynamic effectiveness of the theoretical cycle with extended degazation zone

6. Discussion of the thermodynamic analysis results

We have searched for scheme-cycle designs of ATTs operating with renewable sources of primary energy with a low-temperature potential of 90–250 °C using the thermodynamic analysis by a "cycle method".

In accordance with the conditions of analysis (1), the ambient temperature and the cooling object temperature were assumed to be constant. Then the degazation zone of the solution is determined by the temperature of the heating source and, accordingly, the energy efficiency of the cycle (Fig. 3, *a*). At a temperature of 90 °C, ATT remains operational with low energy efficiency (Fig. 5, cycle *D*1). When the temperature of the heating source rises to 250 °C (Fig. 5, cycle *D*3), the concentration of the weak solution decreases so much that the starting absorption temperature becomes higher than the starting boiling point in the generator, and the "excess temperatures" mode is observed. In this case, it is proposed to introduce internal heat recovery: the "hot" end of the absorber is to heat the "cold" end of the generator as a way to improve the energy efficiency of the ATT cycle.

The scheme-cycle variant of a water-ammonia machine has been considered, operating at "excess temperatures" using the reverse supply of the solution through the generator and absorber (Fig. 7). In this cycle, it is not necessary to use a traditional heat exchanger of solutions since heat exchange is carried out by the reverse supply of solutions with full heating of a strong solution (Fig. 7, *b*, process 4–1). We have built an estimation model of energy analysis of the cycle with "excess temperatures" in accordance with the studied scheme and cycle (23) to (34).

The comparative energy analysis was carried out for four cycles with the expansion of the degazation zone within the limits corresponding to the temperatures of the heating source of 90–180 °C (Table 1). The comparative energy analysis has shown that the COP of the corresponding Carnot cycle, built on the final temperatures of heat supply and removal (20), increases with the expansion of the degazation zone; at the same time, the COP of the Lorentz cycle remains constant (21) since the heat capacity in the processes of heat supply and removal are constant (p_g=const and p_a=const). Such results are reliable since they coincide with the data reported in work [14]. As a result, the degree of the thermodynamic perfection of the actual cycles relative to the cycle model is estimated ambiguously (Fig. 10). The processes in the Lorenz cycle, unlike the Carnot cycle, are similar to the processes in the actual cycles of a thermochemical compressor, so the Lorenz cycle reliably establishes the thermodynamic effectiveness of the cycles with the expansion of the degazation zone with an increase in the temperature of the heat source.

Thermodynamic analysis by a "cycle method", used in this work for the synthesis of scheme-cycle design of ATT, is the first and necessary stage of design. At the level of thermodynamic cycle model, the characteristics of actual cycles affecting energy efficiency have been established; the engineering parameters of installations have been estimated without resorting to complex calculations and design advancements. The scheme-cycle design of ATT with the reverse supply of solutions is the simplest from an engineering point of view and serves only as an example. Further studies of the possibilities of modes with excess temperatures will be performed when designing a new generation of heat-using thermotransformers with heat supply from renewable energy sources. This work is the first stage of ONAFT research on the topic No. MK 21/01 "Improving the reliability of data centers functioning by developing thermoregulation systems based on absorption refrigeration machines and heat pipes".

7. Conclusions

1. The thermodynamic analysis of ATT cycles using a "cycle method" has shown that at constant temperatures of the environment and the cooling object, the degazation zone of the solution and the energy efficiency of the cycle depend on the temperature of the heat source. The temperature of the heat source of 90 °C provides for a minimum degazation zone of 6 % and the performance of the ATT with low energy efficiency. The temperature of the heat source of 250 °C ensures the maximum degazation zone under an "excess temperatures" mode used as a way to improve the energy efficiency of the ATT cycle of internal heat recovery.

2. We have synthesized the scheme-cycle designs of a thermotransformer operating at "excess temperatures" with the implementation of the return supply of a hot weak solution through the generator and a cold strong solution through the absorber. The boundary temperatures of the return feeding processes set were the starting absorption temperature and the starting boiling point in the generator, which ensured the complete heating of the strong solution and abandoning the use of a traditional solution heat exchanger.

3. Comparative energy analysis was carried out for four cycles with the expansion of the degazation zone within the limits corresponding to the temperatures of the heat source of 90–180 °C. We have established that the COP of the Carnot-Carnot cycle model increases by almost 2 times, the COP of the Lorenz-Carnot cycle model remains constant, the COP of the theoretical cycle increases by 1.7 times, the thermodynamic effectiveness of the theoretical cycle relative to the Lorenz-Carnot cycle increased from 26 to 44 %.

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