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METHOD FOR THE OPTIMAL DESIGN OF VACUUM-EVAPORATIVE HEAT PUMPS

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A method is developed for optimally designing vacuum-evaporative heat pumps that use water (R718) as a refrigerant. This method is based on the autonomous method of the thermoeconomic optimization of thermodynamic systems, and makes it possible, when optimizing the design and choosing economical modes of system operation, to simultaneously take into account both thermodynamic and economic parameters. When solving the optimization problem, the resulting costs (RC) of creating and operating the system during the estimated service life are taken as the objective function. The minimum of RCs corresponds to the optimal characteristics of the system while maintaining its performance. The development of the thermo-economic model of the vacuum-evaporative heat pump made it possible to represent the objective function in the form of detailed analytical expressions that take into account the relationship between all the optimizing parameters of the system. The numerical solution to the problem of the thermoeconomic optimization of the operating and design parameters of the vacuum-evaporative heat pump embedded in the cooling system of the second circuit of thermal and nuclear power plants (TPP and NPP) allowed finding the optimal system parameters ensuring the conditions for achieving the minimum RCs. At the same time, for 25 years of operation, the estimated value of the RCs of this heat pump was reduced by 35% through a more rational distribution of energy flows therein. An analytical solution to the optimization problem in the form of a system of partial derivatives of the objective function of RCs for all optimizing variables is suitable for any heat pump operating according to the considered scheme and with a similar type of equipment. The influence of the electricity tariff variability and yearly active time of the vacuum-evaporative heat pump on the economic effect of its thermoeconomic optimization is investigated. The application of the developed methodology in practice should help reduce the financial costs for creating and operating vacuum-evaporative heat pumps that use water as a refrigerant, increase their competitiveness compared to traditional freon systems and create the conditions for their large-scale implementation.

Keywords: *thermoeconomic model, vacuum-evaporative heat pump, exergy losses, resulting costs.*

Introduction

In recent decades, refrigeration and heat pump equipment developers have been actively searching for refrigerants that would meet high environmental requirements (zero or close to zero GWP and ODP values) and at the same time have good thermodynamic parameters comparable to the refrigerants to be replaced. The impetus for these studies were the provisions of the Montreal and Kyoto Protocols concerning the prohibition and restriction of the use of chlorine and fluorine-containing compounds as the refrigerants of thermotransformers (refrigerating machines and heat pumps).

When choosing a refrigerant, the orientation on such indicators as ecological purity, low cost, availability, operational safety, and thermodynamic efficiency has revived interest in natural refrigerants. Of these, many criteria are met by water (R718). Its effect on the environment is well-known and quite predictable, and the thermodynamic characteristics even surpass the known synthetic hydrocarbon-based compounds in a number of indicators.

The use of water as a refrigerant allows for high energy cycle indicators compared with those of synthetic refrigerants, and creates prerequisites for a substantial simplification of the thermotransformer circuit, when water is both the low-potential heat carrier and refrigerant.

Today, the research work on the creation of thermotransformers with R718 refrigerant is at the stage of pilot drafts, which is why the information regarding their real operating characteristics is extremely limited and not generalized. In publications devoted to the development of plants of this type, main attention is paid to the creation of high-speed turbocompressors operating on water vapor and development of engineering methods for calculating them [1–3]. In [4], the authors presented a 3D model of the turbocompressor based on the finite element method for the study of its strength and aerodynamic characteristics. The authors of [5, 6] proposed a dynamic model of the characteristics of a vacuum-evaporative heat pump. This model is necessary to study the problem of starting the unit operating under vacuum, and allows determining ways to further improve the system by varying the geometric dimensions of its elements, as well as changing the configuration of the refrigerant circuit and operating parameters.

However, in modeling energy processes, these works use only the method of energy analysis, which does not allow taking into account the energy losses resulting from external and internal irreversibility in the individual units of a vacuum-evaporative system, which is insufficient for a reliable assessment of the system [7]. To more completely analyze the system, it is advisable to use exergy analysis based on the calculation of exergy losses in the individual processes and cycle of a thermodynamic system.

As is known, when designing thermotransformers, to increase the energy efficiency of their operation, it is necessary to strive to reduce the losses resulting from the irreversibility of thermodynamic processes. However, in practice, this often leads to higher installation costs [8]. At the same time, the high cost of a turbocompressor, due to the need to manufacture its blades from high-strength materials, such as titanium and fiber composites, is a major obstacle to the use of water vapor as a refrigerant. Based on this, when choosing the optimal operating conditions for vacuum-evaporating thermotransformers, it is especially necessary to take into account a number of economic factors.

During optimization calculations, exergy analysis-based thermoeconomic methods of the one-time consideration of thermodynamic and economic factors allow optimizing internal energy processes in the system as well as making an expedient choice of those compromise solutions that would ensure obtaining a minimum level of RCs for its creation and operation. Therefore, the creation of a method allowing to design or deeply modernize vacuum-evaporation thermotransformers to ensure their optimum design and operating characteristics while achieving a minimum level of RCs, is actual, and is the goal of this study.

Thermoeconomic Optimization of a Vacuum-evaporative Heat Pump with R718 Refrigerant

A method is developed for optimally designing vacuum-evaporation heat pumps that use water (R718) as a refrigerant. This method is based on the autonomous method of the thermoeconomic optimization of thermodynamic systems [9–12], and makes it possible, when optimizing the design and choosing the economical modes of system operation, to simultaneously take into account both thermodynamic and economic parameters.

A vacuum-evaporative heat pump operating according to a single-stage scheme [13, 14] is taken as the object to be optimized. It is equipped with a turbo-compressor (CMPR), a contact-type evaporator (EVAP), a horizontal shell-and-tube smooth-tube condenser with an intertubular boiling refrigerant (CO), a throttle valve (T) and piston pumps for pumping heat transfer fluids through the heat pump heat exchangers.

For the scheme under consideration, a thermoeconomic model is built [15–17], through whose conditional control boundary the exergy and heat flows enter and exit, with the flows being necessary for the functioning of the system (Fig. 1).

In Fig. 1, SPC refers to the pump for pumping water from the cooling system of the TPP or NPP second circuit; CP refers to the pump for the water heated in the condenser.

It is assumed that the heat pump uses the circulating water from the cooling system of the TPP or NPP second circuit as a source of low-potential heat, i.e. the water coming from the turbine condenser. This water has a constant design flow rate of $G_{\mathrm{wtr}}^{\mathrm{SCIR}}$ and fixed temperatures of $T_{\mathrm{wtr1}}^{\mathrm{SCIR}}$ and $T_{\mathrm{wtr2}}^{\mathrm{SCIR}}$ at the heat pump evaporator inlet and outlet, respectively. The values of these quantities are given on the basis of the conditions of technological processes at the TPP or NPP. Therefore, the cooling capacity of the evaporator Q_0 of this heat pump during its optimization is assumed to be unchanged. On the contrary, the flow rate $G_{\mathrm{wtr}}^{\mathrm{SCIR}}$ and temperature $T_{\mathrm{wtr}}^{\mathrm{CO}}$ of the water heated in the condenser can be varied by determining the value of the useful heat pump condenser heat productivity Q_{CO} on the basis of the required cooling capacity of the evaporator, and by compensating for a user the possible lack of the heat pump heat productivity during peak hours with an additional reheat unit.

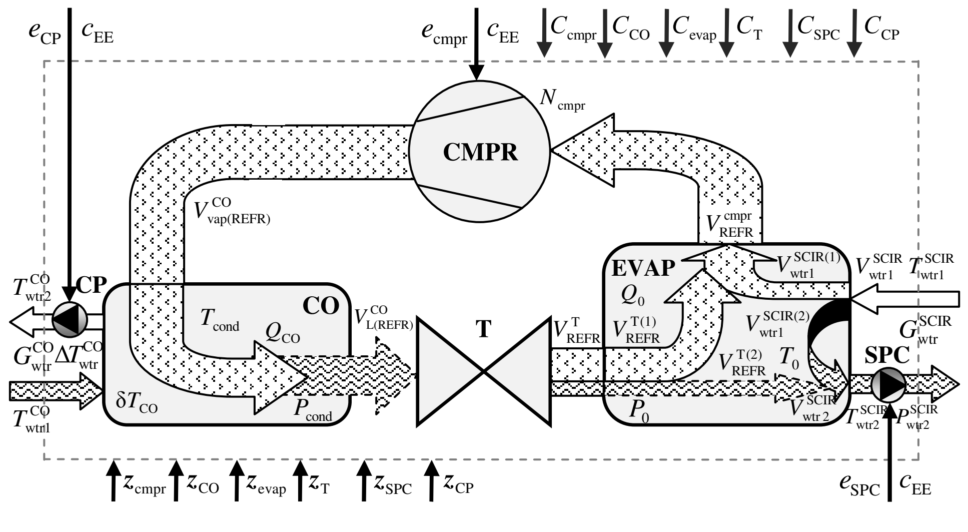

Fig. 1. Thermoeconomic model of a vacuum-evaporative heat pump

Therefore, when solving the optimization problem, the logarithmic mean temperature difference δT_{CO} and the heating of water (heat carrier) $\Delta T_{\mathrm{wtr}}^{\mathrm{CO}}$ in the heat pump condenser are selected as the variables optimizing the objective function. The objective function is the RCs determined from the expression

$$RC = \left[c_{\mathrm{EE}}\left(e_{\mathrm{cmpr}} + e_{\mathrm{SPC}} + e_{\mathrm{CP}}\right) + z_{\mathrm{cmpr}} + z_{\mathrm{evap}} + z_{\mathrm{SPC}} + z_{\mathrm{CO}} + z_{\mathrm{CP}} + z_{\mathrm{T}}\right]\tau_{\mathrm{at}}\, n_{\mathrm{s}} + \\ + C_{\mathrm{cmpr}} + C_{\mathrm{evap}} + C_{\mathrm{SPC}} + C_{\mathrm{CO}} + C_{\mathrm{CP}} + C_{\mathrm{T}}, \tag{1}$$

where c_{EE} is the electricity tariff; e_{cmpr}, e_{SPC} and e_{CP} are the exergy flows for the turbocompressor drive, SPC and CP, respectively; C_{cmpr}, C_{evap}, C_{SPC}, C_{CO}, C_{CP} and C_{T} are the costs of the turbocompressor, evaporator, SPC, condenser, CP and throttle valve, respectively, and z_{cmpr}, z_{evap}, z_{SPC}, z_{CO}, z_{CP} and z_{T} are the annual total deductions from these costs [11, 12, 17]; τ_{at} is the yearly active time of the system; n_{s} is the analyzed period of operation, in years.

The costs of equipment elements C_i, annual total deductions from these costs z_i, as well as the exergy delivered to each element e_i from an external source are included in the objective function (1), and can be expressed as functional dependencies on the given heat pump evaporator cooling capacity Q_0=const and selected optimizing variables

$$e_{\mathrm{cmpr}}, C_{\mathrm{cmpr}}, z_{\mathrm{cmpr}}, C_{\mathrm{evap}}, z_{\mathrm{evap}}, C_{\mathrm{CO}}, z_{\mathrm{CO}}, e_{\mathrm{CP}}, C_{\mathrm{CP}}, z_{\mathrm{CP}} = f\left(Q_0, \delta T_{\mathrm{CO}}, \Delta T_{\mathrm{wtr}}^{\mathrm{CO}}\right). \tag{2}$$

As a result of the above condition on the invariability of the evaporator cooling capacity (Q_0 =const), such parameters as e_{SPC}, C_{SPC}, z_{SPC}, C_{T} and z_{T}, which are functions of Q_0, remain constant.

To solve the optimization problem, the functional expressions (2) present in the objective function of RCs (1) were presented in the form of the detailed analytical dependencies describing the energy processes occurring in the individual elements of the heat pump under consideration. At that, state parameters (temperature, pressure), thermomechanical and thermophysical properties (specific volume, density, enthalpy, entropy, thermal conductivity, kinematic viscosity, dynamic viscosity, latent heat of vaporization, the Prandtl number) of water, which in the heat pump under consideration is also a refrigerant, were presented in the form of regression dependencies on the optimizing variables in the probable range of change of corresponding cycle parameters.

For example, the regression dependence of the entropy of vapor at the end of the actual compression process in the turbocompressor s_2=$f\,(i_2(T_{\mathrm{cond}}), P_{\mathrm{cond}}(T_{\mathrm{cond}}))$ in the range of enthalpy change $i_2(T_{\mathrm{cond}})$ from 2590 to 2750 kJ/kg and the condensation pressure $P_{\mathrm{cond}}(T_{\mathrm{cond}})$ from 0.04 to 0.08 bar for the vapor is the following:

$$s_2 = 7.4998612 - 16.9725647\, P_{\mathrm{cond}}\left(T_{\mathrm{cond}}\right) - 0.0013544\, i_2\left(T_{\mathrm{cond}}\right) + \\ + 68.9253235\left[P_{\mathrm{cond}}\left(T_{\mathrm{cond}}\right)\right]^2 + 7.6712786\cdot 10^{-7}\left[i_2\left(T_{\mathrm{cond}}\right)\right]^2 + 0.0002841\, P_{\mathrm{cond}}\left(T_{\mathrm{cond}}\right) i_2\left(T_{\mathrm{cond}}\right),$$

where $i_2(T_{\text{cond}})$ and $P_{\text{cond}}(T_{\text{cond}})$ are, respectively, regression dependencies of water vapor enthalpy at the end of the actual compression process in the turbocompressor i_2 and condensation pressures of vapor (refrigerant) P_{cond} from the condensation temperature T_{cond}, which in turn is a function of the optimizing variables

$$T_{\text{cond}} = T_{\text{wtr1}}^{\text{CO}} + \Delta T_{\text{wtr}}^{\text{CO}} \frac{e^{\frac{\Delta T_{\text{wtr}}^{\text{CO}}}{\delta T_{\text{CO}}}}}{e^{\frac{\Delta T_{\text{wtr}}^{\text{CO}}}{\delta T_{\text{CO}}}} - 1},$$

where $T_{\text{wtr1}}^{\text{CO}}$ is the water (heat carrier) temperature at the condenser inlet.

When calculating the volume of a contact-type vacuum evaporator, it is assumed that every second the evaporator receives water from the cooling system of the second circuit of the TPP or NPP with a volume of $V_{\text{wtr1}}^{\text{SCIR}}$. During this very second, some of the water, on getting into the vacuum, turns to vapor with a volume of $V_{\text{wtr1}}^{\text{SCIR(1)}}$ (Fig. 1), and the remaining part of the water, on getting cooled to the boiling point T_0, falls onto the evaporator bottom. Also, every second the evaporator receives (from the throttle valve) the saturated refrigerant vapor (a mixture of water and saturated vapor) with a temperature of T_0 and volume of $V_{\text{REFR}}^{\text{T}}$. Within this very second, one part of it, on expanding to the volume $V_{\text{REFR}}^{\text{T(1)}}$, fills up the volume of vapor $V_{\text{REFR}}^{\text{cmpr}} = V_{\text{wtr1}}^{\text{SCIR(1)}} + V_{\text{REFR}}^{\text{T(1)}}$, and the other part of $V_{\text{REFR}}^{\text{T(2)}}$ fills up the volume of liquid at the bottom of the evaporator $V_{\text{wtr2}}^{\text{SCIR}} = V_{\text{wtr1}}^{\text{SCIR(2)}} + V_{\text{REFR}}^{\text{T(2)}}$. The volume of vapor $V_{\text{REFR}}^{\text{cmpr}}$ is pumped out every second from the evaporator by the turbocompressor, and every second the volume of liquid is drawn by the SPC to the condenser of the TPP or NPP turbine.

Thus, assuming that every second the evaporator contains vapor with a volume of $V_{\text{REFR}}^{\text{cmpr}}$ and liquid with a volume of $V_{\text{wtr2}}^{\text{SCIR}}$ that are numerically equal, respectively, to the one-second volume flow rate of the refrigerant through the turbocompressor (turbocompressor volume capacity), m^3/s, and one-second volume water flow through the SPC, m^3/s, the evaporator volume, m^3, is calculated as $V_{\text{evap}} = \left(V_{\text{REFR}}^{\text{cmpr}} + V_{\text{wtr2}}^{\text{SCIR}}\right)\tau$, where the time τ=1 s.

Similarly, the internal volume of the condenser shell is calculated as $V_{\text{CO}} = \left((1-\overline{\varphi})V_{\text{vap(REFR)}}^{\text{CO}} + \overline{\varphi}\, V_{\text{L(REFR)}}^{\text{CO}}\right)\tau$, where $\overline{\varphi}$ is the fill factor, i.e. the ratio of the volume filled with liquid to the total volume (in particular, for a shell-and-tube condenser with the inter-tubular condensation $\overline{\varphi}$=0.3–0.4); $V_{\text{vap(REFR)}}^{\text{CO}}$ is the one- second volumetric flow rate of the compressed refrigerant vapor supplied to the condenser by the turbocompressor (Fig. 1); $V_{\text{L(REFR)}}^{\text{CO}}$ is the one-second volumetric flow rate of the liquid refrigerant at the condenser outlet; τ = 1 s is the time.

The costs of equipment elements C_i and the annual total deductions from these costs z_i were determined by using regression dependencies on the optimizing variables built on the basis of the cost functions presented in [14, 18, 19]. The exergy losses in the heat pump main elements were calculated using the entropy method [20].

Substituting the detailed expressions (2), obtained in expanded form, into the objective function (1) and solving the system of equations

$$\frac{\partial RC}{\partial \delta T_{\text{CO}}} = 0; \quad \frac{\partial RC}{\partial \Delta T_{\text{wtr}}^{\text{CO}}} = 0$$

allows determining the optimal values of the variables δT_{CO} and $\Delta T_{\text{wtr}}^{\text{CO}}$, whose substitution into the corresponding calculation formulas makes it possible to find the optimal (from the point of minimization of RCs) regime and geometry parameters of the vacuum-evaporation heat pump with R718 refrigerant.

Results of Calculating the Optimal Parameters of a Vacuum-evaporative Heat Pump

The following is taken as initial data for solving the optimization problem: the heat pump evaporator cooling capacity, Q_0=600 kW; the flow rate of the cooled water from the TPP or NPP second circuit cooling system through the evaporator, $G_{\text{wtr}}^{\text{SCIR}}$ =12.7632 kg/s; the water temperature at the evaporator inlet and outlet, $T_{\text{wtr1}}^{\text{SCIR}}$ =40 ºC and $T_{\text{wtr2}}^{\text{SCIR}}$ =T_0=25 ºC, respectively; the water pressure in the discharge pipe (after the SPC), $P_{\text{wtr2}}^{\text{SCIR}}$ =4 bar; the temperature of the heated water (heat carrier) at the condenser inlet, $T_{\text{wtr1}}^{\text{CO}}$ =25 ºC; the system yearly active time, τ_{at}=8000 h/year; the analyzed system operation time, n_{s}=25 years; the electricity tariff, c_{EE}=0.0727 USD/(kWh), the normal ratio of deductions from the equipment costs, $k_{\text{n}i}$=0.15; the ambient temperature, T_{env}=32 ºC.

The optimization results are shown in Table 1.

Table 1. Results of the thermoeconomic optimization of the vacuum-evaporative heat pump

Parameter	Initial version	Optimum version
condenser water heating $\Delta T_{\text{wtr}}^{\text{CO}}$, K	10	5.516
condenser logarithmic mean temperature difference δT_{CO}, K	10	2.968
turbocompressor outlet refrigerant temperature T_2, °C	120.95	63.98
condensation temperature T_{cond}=T_3, °C	40.82	31.53
condenser heat productivity Q_{CO}, kW	860.95	824.31
turbocompressor power consumption N_{cmpr}, kW	73.929	29.480
condenser water pump power consumption, kW	6.825	6.825
evaporator water pump power consumption, kW	2.175	10.409
total heat pump power consumption, kW	82.928	46.714
refrigerating factor ε	10.821	27.137
energy efficiency factor with the power consumption of auxiliary mechanisms (pumps) taken into account ε_{e}	9.647	17.125
evaporator exergy losses $E_{D\text{evap}}$, kW	20.089	20.089
condenser exergy losses $E_{D\text{CO}}$, kW	35.208	11.370
throttle valve exergy losses E_{DT}, kW	0.583	0.098
turbocompressor exergy losses $E_{D\text{cmpr}}$, kW	12.062	5.679
total heat pump exergy losses $\Sigma E_{D\text{HP}}$, kW	67.943	37.235
indicated turbocompressor output $N_{i\text{cmpr}}$, kW	60.954	24.306
turbocompressor exergy efficiency $\eta_{\text{exg cmpr}}$	80.211	76.638
evaporator water pump cost C_{SPC}, USD	907	907
condenser water pump cost in C_{CP}, USD	1 378	2 242
throttle valve cost C_{T}, USD	320	320
evaporator volume V_{evap}, m^3	14.607	14.373
evaporator cost C_{evap}, USD	12 859	12 798
condenser outer heat exchange surface area A_{CO}, m^2	24.5	59.7
condenser cost C_{CO}, USD	19 020	25 292
electric motor-driven turbocompressor cost C_{cmpr}, USD	22 292	13 558
total heat pump cost, USD	56 774	55 116
deductions from capital investments for 25 years, USD	269 678	261 803
capital costs for 25 years, USD	326 453	316 919
operating costs for 25 years, USD	1 205 398	679 015
resulting costs for 25 years, USD	1 531 851	995 934
economic effect resulted from thermoeconomic optimization, %	–	35.0

The data in Table 1 show that a reduction in the values of optimizing variables of water heating $\Delta T_{\text{wtr}}^{\text{CO}}$ and the logarithmic mean temperature difference δT_{CO} in the condenser leads to a decrease in the refrigerant condensation temperature T_{cond}, which at the fixed boiling point T_0=const contributes to an increase in the specific evaporator mass cooling capacity q_0, a decrease in the refrigerant mass flow G_{REFR}, and a reduction in the exergy losses during the drosselation E_{DT}. It should be noted that because of the steep rise of the left boundary curve for R718, the irreversibility losses during the throttling of this refrigerant are extremely small compared to those in the condenser and turbocompressor. A reduction in the refrigerant condensation pressure P_{cond} at P_0=const, which is concomitant with a decrease in T_{cond}, contributes to a reduction in the degree of the refrigerant compression in the turbocompressor and a decrease in the temperature of the compressed vapor on the pressure line T_2. This, in conjunction with a decrease in G_{REFR}, leads to a decrease in the condenser heat productivity Q_{CO} at the required Q_0=const, as well as a decrease in the exergy losses $E_{D\mathrm{CO}}$ from irreversibility therein.

A reduction in the temperature T_2 and consumption G_{REFR} also leads to a decrease in the turbocompressor exergy losses $E_{D\mathrm{cmpr}}$ as well as in its indicated output $N_{i\mathrm{cmpr}}$ and power consumption N_{cmpr}. The evaporator exergy losses $E_{D\mathrm{evap}}$ remain unchanged because at constant T_{env}, the water vapor (refrigerant) entropies at the turbocompressor inlet s_1, Q_0, $T_{\mathrm{wtr1}}^{\mathrm{SCIR}}$ and $T_{\mathrm{wtr2}}^{\mathrm{SCIR}}$, a reduction in the flow rate G_{REFR} is compensated by a decrease in the wet saturated water vapor (refrigerant) entropy at the end of the throttling process s_4, and thus the product $G_{\mathrm{REFR}}(s_1 - s_4)$ in the expression for calculating the evaporator exergy losses [20] remains constant. Despite this, the total exergy losses in the system, $\Sigma E_{D\mathrm{HP}}$, are significantly reduced, as is N_{cmpr}, which leads to a significant (2.5 times) increase in the refrigerating factor ε.

A significant decrease in the optimal logarithmic mean temperature difference δT_{CO} leads to a significant increase in the condenser heat exchange surface area A_{CO} and its cost C_{CO}, despite some compensation for this increase in the surface area A_{CO} by increasing the condenser heat transfer coefficient k_{CO} because of heat exchange intensification therein due to the increase in the heat carrier movement velocity $W_{\mathrm{wtr}}^{\mathrm{CO}}$. An increase in $W_{\mathrm{wtr}}^{\mathrm{CO}}$ is associated with a decrease in the condenser temperature difference $\Delta T_{\mathrm{wtr}}^{\mathrm{CO}}$, which leads to a significant increase in the heated water mass flow rate, CP power consumption and its cost C_{CP}. The evaporator cost C_{evap} decreases slightly, since at Q_0=const its volume V_{evap} decreases only as a result of a certain decrease in G_{REFR}. The increase in the costs of C_{CO} and C_{CP} is compensated by even a greater, in magnitude, reduction in the turbocompressor cost C_{cmpr} on account of a significant decrease in its N_{cmpr}. As a result, the total capital costs on the system are slightly reduced. At the same time, the operating costs of the system are significantly reduced due to a significant reduction in the turbocompressor power consumption, which more than compensates for the increase in electricity consumption by the CP. The total power consumption of the system is significantly reduced, and the energy efficiency factor ε_{e} increases significantly (1.77 times).

Some reduction in capital costs and a significant reduction in operating costs lead to a significant reduction in *RCs* for the estimated period of system operation. Given the initial data, the economic effect of thermoeconomic optimization is 35%.

Since the considered design of the vacuum-evaporative heat pump can be used to cool water at given parameters in a number of technological processes at enterprises of various industries and agriculture, the electricity tariff c_{EE}=0.0727 USD/(kWh) adopted in the calculation is the tariff for industrial consumers of electricity (Class II voltage up to 27.5 kV) for the currency rate in Ukraine as of mid-October 2017. In the case of using the heat pump in the technological cycle of the TPP or NPP, the cost of electricity for the turbocompressor and pump drives will be related to the cost of the plant's own needs, and will be assessed at the cost of production. Depending on the yearly duration of work on observing the graphs of the energy loads characterized by the number of hours of using the installed power of equipment, power stations are usually classified as basic (τ_{at}>6000 h/year, for all NPPs), semi-peak (τ_{at}=2000–5000 h/year) and peak (τ_{at}<2000 h/year) [21]. The cost of the electric energy, c_{EE}, consumed by the heat pump and its yearly active time τ_{at} can have a significant impact on the economic effect of thermoeconomic optimization.

Therefore, the optimal regime and geometrical parameters of the vacuum-evaporative heat pump were also calculated with the variation of the electricity tariff c_{EE} from 0.02 USD/(kWh) (approximate cost of electricity production at NPPs) to 0.12 USD/(kWh) (the most expensive tariff for Eastern European countries) [13, 14] and its active time τ_{at} from 2 000 to 8 000 h/year. Some of calculation results are shown in Figs. 2 and 3.

Fig. 2 shows that the optimal values of the refrigeration coefficient $\varepsilon_{\mathrm{opt}}$ and the energy efficiency factor $\varepsilon_{\mathrm{e\ opt}}$ increase with increasing c_{EE} and τ_{at}, and consequently, the heat pump profitability increases and the RC operational component decreases.

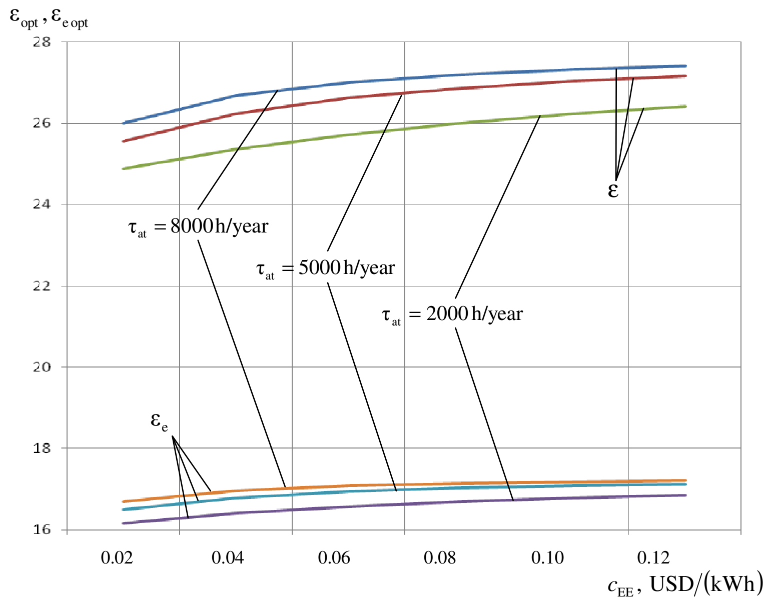

Fig. 2. Dependence of the optimal values of the cooling coefficient ε_{opt} and the heat pump energy efficiency factor $\varepsilon_{e\ opt}$ from c_{EE} and τ_{at}

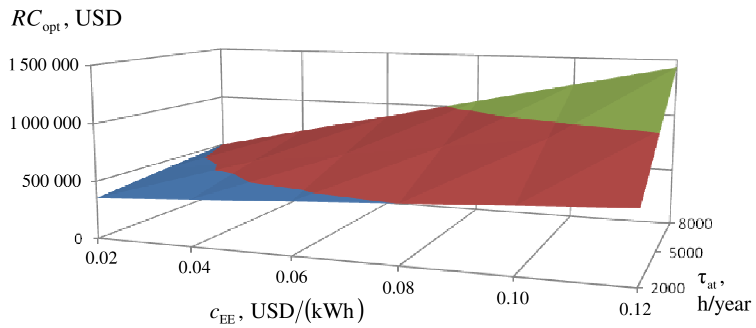

Fig. 3. Dependence of the RCs of the optimum vacuum-evaporative heat pump RC_{opt} from c_{EE} and τ_{at} for 25 years of service

The RCs of the optimal vacuum-evaporative heat pump for 25 years of service and the effect of its thermoeconomic optimization depending on changes in the values of c_{EE} and τ_{at} are shown in Fig. 3 and in Table 2.

Table 2. Effect of the thermoeconomic optimization of the vacuum-evaporative heat pump, %

Parameter		Electricity tariff c_{EE}, USD/(kWh)					
		0.02	0.04	0.06	0.08	0.10	0.12
heat pump yearly active time τ_{at}, h/year	8000	23.9	30.3	33.6	35.6	37.0	38.0
	5000	19.6	26.0	29.7	32.2	33.9	35.2
	2000	13.0	17.7	21.2	23.9	26.0	27.7

The data in Table 2 show that the cost of electricity c_{EE} and the heat pump yearly active time τ_{at} have a significant impact on the economic effect of its thermoeconomic optimization, which can vary from 13% (at c_{EE}=0.02 USD/(kWh) and τ_{at}=2000 h/year) up to 38% (at c_{EE}=0.12 USD/(kWh) and τ_{at}=8000 h/year). This is due to the fact that the greater the values of c_{EE} and τ_{at}, the greater economic effect can be obtained from increasing the thermodynamic perfection of the system, increasing its refrigeration coefficient ε and reducing the operating component of RCs, which has a much greater effect on the level of RCs than some increase in their capital component.

Discussion of the Results of Thermoeconomic Optimization of a Vacuum-evaporative Heat Pump with R718 Refrigerant

The developed methodology and software allow solving the problem of optimizing the operating and design parameters of vacuum-evaporative heat pumps using water (R718) as a refrigerant, taking into account the interrelations between the parameters of all major subsystems while ensuring the minimum level of RCs for their creation and operation.

The numerical solution to this problem allowed us to find the optimal parameters of the vacuum-evaporative heat pump built into the cooling system of the second circuit of a TPP or an NPP, ensuring the conditions for achieving the minimum level of RCs for different values of the electricity tariff and the yearly active time of the system.

One of the advantages of the proposed method is that the obtained unambiguous analytical solution in the form of a system of equations is suitable for the thermo-economic optimization of any heat pump operating according to the considered scheme and with a similar type of equipment.

Concurrently, this advantage imposes a restriction on the use of this method in optimizing the vacuum-evaporative heat pumps that operate according to the schemes different from the one considered here, or are equipped with a different type of equipment (for example, using a surface-type evaporator instead of a contact-type one). In this case, the computational algorithm must be adapted to the description of the energy processes in each particular scheme, taking into account the specificity of the design of all its basic elements.

The application of this method in practice should help reduce the financial costs of creating and operating vacuum-evaporative heat pumps that use water as a refrigerant, increase their competitiveness compared to traditional freon systems, and help create the conditions for their widespread implementation.

In the future, this method can be adapted to solving the problems of the thermoeconomic optimization of the operating and design parameters of thermotransformers working on water of another circuit-design solution or function.

Conclusions

The development of the thermoeconomic model of the vacuum-evaporative heat pump that uses water (R718) as a refrigerant allowed the objective function of RCs to be presented as detailed analytical expressions that take into account the relationship between all the optimizing parameters of the system. The analytical solution to the optimization problem in the form of a system of partial derivatives of the objective function of RCs for all optimizing variables is suitable for any heat pump operating according to the scheme considered and with a similar type of equipment. As a result of the thermoeconomic optimization of the operating and design parameters of the vacuum-evaporative heat pump built into the cooling system of the second circuit of a TPP or an NPP, it was possible to reduce the estimated value of RCs for 25 years of its service by 35% through a more rational distribution of energy flows therein. At the same time, the electricity tariff c_{EE} and the heat pump yearly active time τ_{at} have a significant influence on the economic effect of the thermo-economic optimization of the system. The greater the values of c_{EE} and τ_{at}, the greater economic effect can be obtained from increasing the thermodynamic perfection of the installation, increasing its refrigerating factor ε, and reducing its operating costs.

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Методика оптимального проектування вакуумно-випарних теплових насосів

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На базі автономного методу термоекономічної оптимізації термодинамічних систем розроблена методика оптимального проектування вакуумно-випарних теплових насосів, що використовують воду (R718) як холо-

доагент. Ця методика дозволяє під час оптимізації конструкції і вибору економічних режимів роботи системи одночасно враховувати як термодинамічні, так і економічні параметри. Розв'язуючи задачі оптимізації, як цільова функція прийняті зведені витрати на створення та експлуатацію системи протягом розрахункового терміну служби. Мінімум зведених витрат відповідає оптимальним характеристикам системи під час збереження її продуктивності. Розробка термоекономічної моделі вакуумно-випарного теплового насоса дозволила подати цільову функцію у вигляді розгорнутих аналітичних виразів, що враховують взаємозв'язок між усіма оптимізуючими параметрами системи. Числовий розв'язок задачі термоекономічної оптимізації режимно-конструктивних параметрів вакуумно-випарного теплового насоса, що вбудований в систему охолодження другого контуру теплових і атомних електростанцій (ТЕС) і (АЕС), дозволив знайти оптимальні параметри системи, що забезпечують умови досягнення мінімального рівня зведених витрат. За таких обставин розрахункове значення зведених витрат за 25 років експлуатації даного теплового насоса вдалося знизити на 35 % за рахунок більш раціонального розподілу енергетичних потоків в ньому. Аналітичний розв'язок задачі оптимізації у вигляді системи рівнянь частинних похідних від цільової функції зведених витрат за всіма оптимізуючими змінними є придатним для будь-якого теплового насоса, що працює за розглянутою схемою і з подібним типом обладнання. Досліджено вплив варіативності тарифу на електроенергію і тривалості роботи вакуумно-випарного теплового насоса протягом року на економічний ефект від його термоекономічної оптимізації. Застосування розробленої методики на практиці має сприяти зниженню фінансових витрат на створення і експлуатацію вакуумно-випарних теплових насосів, що використовують воду як холодоагент, підвищенню їхньої конкурентоспроможності порівняно з традиційними фреоновими системами і сприяти створенню умов для їх широкомасштабного впровадження.

Ключові слова: *термоекономічна модель, вакуумно-випарний тепловий насос, втрати ексергії, зведені витрати.*

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EXPERIMENTAL STRENGTH ANALYSIS OF VARIABLE STIFFNESS WAFFEL-GRID CYLINDRICAL COMPARTMENTS PART 2. Analysis Results

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This paper presents the results of the experimental analysis of the stress-strain state of the variable stiffness tail compartment (section) designed by the Yuzhnoye Design Bureau. Equivalent compressive forces in the cross-sections of the tail compartment without the transport-erector support are analyzed. It is established that the calculated and experimental compressive forces are extremely close. Deformations in the tail compartment were measured where resistance strain gages were installed. For the measurement of displacements, displacement gauges were installed. The displacements were measured at six points. They were studied at maximum loading values corresponding to the fifth and sixth stages of loading. Axial movements are always negative, which indicates that the shell is compressed in the axial direction. The stress-strain state of the launch vehicle tail compartment was experimentally investigated. The circumferential normal stresses are several orders of magnitude smaller than the longitudinal ones. Therefore, the circumferential stresses were not investigated. The results of the experimental studies were compared with the numerical simulation data in the NASTRAN software package. The purpose of the simulation was to confirm the workability of the tail compartment under the loads that occur during operation. In other words, the design must withstand the actual loads without destruction and the appearance of plastic deformations. Special attention was paid to the zones that were directly under the brackets. The experimental results and numerical simulation data are close.

***Keywords**: stress-strain state of tail compartment, equivalent compressive forces, displacement measurements.*

Introduction

In the first part of this article [1], a technique for the experimental study of the strength of the launch vehicle (LV) tail compartment was developed. The tail compartment is a waffle-grid thin-walled structure, which consists of two shells. This part of the article presents the results of experimental studies based on the methodology developed in the first part.