Роботу присвячено розробці методу оптимального проектування вакуумно-випарних теплонасосних установок (ТНУ) для системи охолодження технологічного обладнання другого контуру атомної електростанції (АЕС) з використанням сучасних методів термодинамічного аналізу та термоекономічної оптимізації.

Запропоновано дві схеми включення вакуумно-випарної ТНУ в систему охолодження другого контуру АЕС. Перша схема передбачає використання ТНУ в комплексі з існуючою градирнею і дозволяє доохолодити воду після градирні з 30 °С до 25 °С. За другою схемою тільки ТНУ здійснює охолодження води до необхідних параметрів.

Розроблено термодинамічну модель для прогнозування статичних характеристик ТНУ. Проаналізовано термодинамічні властивості води як холодоагенту, оцінено їх вплив на режимні параметри і енергетичну ефективність парокомпресійного циклу. Встановлено, що вода повністю відповідає всім вимогам екологічної безпеки, які пред'являються до робочих речовин теплових насосів. Її використання дозволяє забезпечити високі енергетичні показники циклу в порівнянні з синтетичними холодоагентами. До проблемних аспектів використання води можна віднести характерну для пароводяного циклу високу температуру в кінці процесу стиснення. Однак шляхом застосування двох секційного конденсатора з утилізацією теплоти зняття перегріву пари вплив цієї температури на енергетичні, а також експлуатаційні показники установки, можна нівелювати.

З використанням графоаналітичного апарату побудови С-кривих здійснено вибір раціональних схемно-циклових рішень вакуумно-випарної теплонасосної установки, які забезпечують ефективну роботу установки для охолодження технологічного обладнання другого контуру АЕС, оцінено капіталомісткість системи.

На основі моделювання теплогідравлічних процесів в контурі циркуляції холодоагенту виконано термоекономічну оптимізацію і визначено режимно-конструктивні характеристики установки, які відповідають мінімуму зведених витрат при її експлуатації

Ключові слова: вакуумно-випарна теплонасосна установка, атомна електростанція, термоекономічна модель, зведені витрати

UDC 621.577
DOI: 10.15587/1729-4061.2019.175679

DEVELOPMENT OF A VACUUM-EVAPORATIVE THERMOTRANSFORMER FOR THE COOLING SYSTEM AT A NUCLEAR POWER PLANT

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Received date 22.05.2019
Accepted date 06.08.2019
Published date 19.08.2019

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1. Introduction

Reliable water supply is a prerequisite for efficient and safe operation of the second circuit of a cooling system for nuclear power plants (NPPs).

An evaporative cooling tower typically provides heat dissipation in a cooling system of the second circuit at NPP. It uses energy resources of the environment. Ambient temperatures often reach critical levels in summer in subtropical, tropical and even in temperate climates in recent years. That is why an evaporative cooling tower is not able to provide the required level of water cooling (25 °C) during summer. This circumstance can significantly affect efficiency and reliability of NPP core equipment, as it leads to a decrease in its capacity. Construction of another cooling tower, in addition to the existing one, may be advisable only if we increase the maximum capacity of NPP, since such an increase would inevitably lead to an increase in water consumption in the circulating water supply system.

Nowadays, in addition to cooling towers, it is possible to use such devices as water storage, spray pools, «dry» cooling towers and others as NPP water cooling equipment. However, cooling efficiency of such devices depends substantially on temperature and humidity parameters of outdoor air, as well as on other meteorological factors (wind speed, etc.). In this regard, none of the mentioned devices is able to provide a constant temperature of water cooling while maintaining its flow rate in summer period, which is the most heat-stressed. In addition, a use of atmospheric coolers of any type has a serious impact on the environment. There is an increase in ambient temperature and changes in air regime at locations of NPP. For example, a heat-and-humidity plume of a spray pool affects the environment as follows: there is fogging, dropping of droplets condensed in the direction of wind, moistening of a territory, and more rarely, formation of clouds.

Therefore, the development of new devices and circuits that will provide additional water cooling after a cooling tower, or – full water cooling instead is an expedient problem.

The problem can be solved by the inclusion of a heat pump plant in the technological scheme of the second circuit at NPP. That will make it possible to cool water from 30 °C to 25 °C after a cooling tower, regardless to ambient temperature. The use of a heat pump plant (HPP) in a cooling system of the second circuit of NPP will significantly reduce loss of water due to evaporation. It can be from 2.5 % to 10 % of the total volume of circulating water during operation of a cooling tower. The economic effect of water cooling with cool produced by HPP instead of a cooling tower increases significantly as cooling water temperature approaches the lower temperature limit of an evaporative cooling tower. The proposed cooling technology is quite safe for the environment.

A vacuum-evaporative HPP is the most attractive option among the existing types of thermotransformers (absorption, vapor-ejector, and vapor-compression) for the solution of the problem of cooling of the second circuit of NPP. We can attribute such plant to the class of vapor-compressor refrigerators by its main functional purpose (water cooling). However, we can consider it as a heat pump due to its operation temperature range (evaporation temperature above 10 °C and condensation temperature almost 80 °C). It is possible to use heated water after a condenser for hot water supply due to the high temperature range. In addition, the temperature range of a plant affects the type of heat exchange and compressor equipment of a technological circuit significantly when using water as a refrigerant. For example, when the evaporation temperature is below 0 °C, the thermotransformer is actually an ice generator with a screw compressor. When the evaporation temperature increases to 10 °C and above, it is necessary to use a high-speed turbocompressor in the circuit, as the specific volume of absorbed vapor increases significantly.

The main advantages of a vacuum-evaporative HPP over the existing systems of cooling of the second circuit of NPP are possibility of efficient cooling at high atmospheric air temperatures and absence of water losses due to evaporation. At the same time, there is a decrease of thermal influence on the environment. Advantages include possibility of utilization of part of thermal energy discharged from HPP for municipal needs. Using of HPP provides also reduction in the amount of dissolved gases in the water, which circulates in the second circuit, reducing a risk of radioactive leakage, and possibility of complex operation together with a cooling tower during peak loads.

A disadvantage is the relative complexity of a HPP circuit, as well as significant power consumption comparing to a cooling tower.

The use of water as a refrigerant for HPP creates preconditions for a significant simplification of the circuit of a thermotransformer. In practice, this means renunciation of the use of several hydraulic circuits in HPP design (refrigerant circuit, low-potential coolant circuit, and coolant condenser circuit) and switching to a simpler single or double circuit. The difference between such circuit and a traditional vapor compressor HPP is in the process of self-evaporation. Boiling in the evaporator occurs not due to the supply of liquid with higher pressure from the outside to boiling of liquid, but due to the flow of liquid with higher pressure into the evaporator, which maintains a low pressure, and instant boiling. Thus, the plant takes the heat required for evaporation from water, which needs cooling. The water obtains a lower temperature corresponding to the boiling point at the pressure in the evaporator due to the process of self-evaporation, and it goes into the main line, and vapor pumped by the compressor goes into the circuit of NPP.

In this regard, the use of contact heat exchangers instead of surface type devices in HPP makes possible to reduce its metal content and to increase energy efficiency by reducing the minimum temperature difference in the processes of heat exchange in evaporator and condenser. Meanwhile, the design dimensions of the evaporator must also be large to make possible phase transition of operating substance because of water vapor with a large specific volume pumped into the compressor.

There are other advantages of a vacuum-evaporative HPP. Water as a refrigerant is an environmentally friendly substance. It does not pollute the environment and does not contribute to the effect of global warming. From a technical and economic point of view, water is a relatively cheap operating substance; it is affordable and safe for use compared to freons, ammonia, and other refrigerants used in traditional heat pumps.

Thus, the expedient task is to develop a vacuum-evaporative HPP, which uses water as a refrigerant (R718) to provide the required level of water cooling in the second circuit of NPP under the specified economic and technical conditions of operation.

2. Literature review and problem statement

Studies on creation of HPP with R718 refrigerant are under development at present, so we have limited data on their actual efficiency.

We analyzed the current state of the problem, and we can note that the focus in the creation of vacuum-evaporative HPP is on the development of compressor equipment.

It is impossible to use an existing fleet of industrial compressors directly, due to specific properties of water as a refrigerant in vacuum-evaporative cooling plants. Existing turbocompressors do not have such high rate characteristics. Authors of paper [1] noted that a multi-stage axial turbocompressor is best for compressing of water vapor.

Authors of work [2] analyzed ways of creation of high-efficiency vapor-powered refrigerators with a turbocompressor. They proposed an engineering methodology for calculation of a turbocompressor on R718. They presented the results of comparison of its characteristics, such as a Mach number, circumferential speed, an impeller diameter, a number of rotations with characteristics of other refrigerants. However, they obtained certain operating and design characteristics without taking into account technical and economic indicators.

Paper [3] proposed a 3D model of a high-speed multistage axial vapor compressor based on the finite element method. Creators used composite materials for the compressor. They investigated strength and aerodynamic characteristics of the compressor using a 3D model. The material of an impeller of the axial compressor was epoxy. The axial turbocompressor had a high cost, due to the need to manufacture its blades of high-strength materials such as titanium and fiber composites. It was a major obstacle to the use of water vapor as operating substance. The introduction of epoxy manufacture technology opens wide prospects for large-scale use of turbocompressors with R718.

Authors of paper [4] developed a dynamic model of characteristics of a vacuum-evaporative heat pump to study the problem of launching of a plant under vacuum. A plant can suck air into the system during the initial period of plant operation (up to 1,000 s). As the authors noted, the development of the model is the solution, which gave the most complete understanding of characteristics of the cycle at the moment of launching. In addition, modeling makes it possible to determine ways to improve the system further by varying the geometric dimensions of elements, as well as changing the configuration of the circuit of the refrigerant circulation and its parameters. One of the advantages of dynamic modeling is possibility of improvement of energy efficiency by selection of the best strategies for management and control of the cleaning of the system from non-condensed gases. Researchers carried out verification of the dynamic model based on comparison with the results of an experimental study of the process of launching of a laboratory vacuum-evaporative HPP. The experimental HPP contained an axial air compressor adapted for vapor operation, a contact evaporator and a plate condenser [5]. We should note that there was a rather large discrepancy (more than 15 %) between the experimental data and the results of numerical simulations in the initial period of plant operation (up to 1,000 s). The results of the calculation of evaporation and condensation temperatures were satisfactorily consistent with the experimental data after an increase in operating time of the plant.

It is advisable to use a twin-screw air compressor to overcome the launching problem [6]. Adaptation of the air compressor for vapor operation is possible provided improvement of hermeticity of the sealing system.

Authors of work [7] analyzed energy efficiency of one-stage and two-stage refrigerating machines with R718 refrigerant containing a centrifugal compressor. They concluded that a combination circuit with an additional two-phase ejector is necessary to reduce irreversible losses due to throttling. In addition, the use of a contact type condenser in combination with a cooling tower in HPP circuit will make it possible to absorb air, which appears in the system during circulation through a cooling tower, from water.

Paper [8] proposed to use air-cooled condensers, which gives plants with them certain advantages in increasing of competitiveness of vacuum-evaporative thermotransformers with R718. However, replacement of a water condenser with air one causes an increase in discharge pressure by 30–50 %, and as a consequence, a decrease in the compressor volume flow ratio and cooling capacity. It is advisable to use a vacuum pump compressor with a discharge vapor injection mechanism to eliminate this disadvantage. Thus, we can reduce the flow of vapor from pumping to suction, which occurs in plants with air-cooled condensers. The use of such circuit reduces energy consumption by at least 10 % in vacuum evaporation systems. Capital costs also decrease.

In order to increase efficiency of the vacuum-evaporative thermotransformer, authors of paper [9] proposed to install an ejector and injector before an evaporator and before a deaerator heater, respectively, in the water circulation circuit. The receiving chamber of the ejector is connected to this circuit after a low-potential water source, and the receiving chamber of the injector is connected to the circuit of water vapor circulation after the compressor.

Monograph [10] considered the design of refrigeration and heat pump plants with R718 refrigerant, including a centrifugal compressor and a two-phase ejector [10]. The authors considered possibility of a use of a two-phase ejector instead of a centrifugal compressor and its use for additional compression in the second stage. The monograph evaluated an influence of the thermophysical properties of a refrigerant on design characteristics of an ejector. However, it did not cover issues of creation of heat exchange equipment for vacuum-evaporative plants.

We can summarize the analysis of research publications and note one important disadvantage. It is a lack of in-depth pre-project analysis of circuit-cycle solutions for vacuum-evaporative thermotransformers. As a consequence of invalidity of this approach, many authors attempt to confine themselves to consideration of the task of design of a single, even an important, element of a circuit without taking into account its interconnections with other elements.

We need a new approach to solve the problems of design of heat pump plants. The new approach should rely on modern methods, which ensure making appropriate decisions. These are methods of analysis and synthesis of thermal engineering systems based on application of the theoretical apparatus of technical thermodynamics, thermoeconomics, a theory of heat and mass transfer, and a theory of system engineering. In recent years, their development went in the field of analysis of structures of technological circuits, studies on relationships between elements, and evaluation of integration properties of a system. Application of these methods will eliminate possible design errors at the initial stages of development of a thermal transformation system, providing a choice of the optimal technological circuit and design characteristics of heat exchange and compressor equipment.

3. The aim and objectives of the study

The aim of this study is to develop a method for optimal design of vacuum-evaporative heat pump plants for a cooling system of technological equipment of the second circuit at NPP using modern methods of thermodynamic analysis and thermoeconomic optimization. Application of the method in practice will reduce financial costs for creation and operation of a vacuum-evaporative HPP, which operate on R718, compared with freon HPP.

We set the following tasks to achieve the objective:

– proposal of circuits for inclusion of a vacuum-evaporative heat pump plant into a cooling system of the second circuit of NPP;

– development of a thermodynamic model to forecast static characteristics of HPP, analysis of thermodynamic properties of water as a refrigerant, evaluation of their influence on mode parameters and energy efficiency of a vapor compression cycle;

– search for a rational technological circuit for a vacuum-evaporative HPP;

– thermoeconomic optimization based on mathematical modeling of thermal hydraulic processes in HPP and determination of mode-design characteristics, which correspond to the minimum of resulting costs.

4. Technological aspects of the development of cooling circuits of the second circuit of a nuclear power plant using a vacuum-evaporative heat pump plant

There are two possible circuits for the inclusion of a vacuum-evaporative HPP in a cooling system of NPP. The first circuit (Fig. 1) provides «removal» of peak load in a summer season due to cooling of water from 30 °C to 25 °C after a cooling tower in HPP. The second circuit (Fig. 2) provides «removal» of full heat load. There is cooling of water from 40 °C to 25 °C.

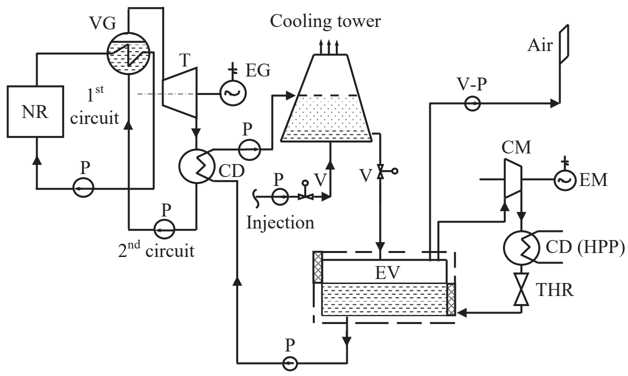


Fig. 1. Circuit of the cooling system of NPP with the combined use of the heat pump and the evaporative cooling tower: NR — nuclear reactor; VG — vapor generator; T — turbine; EG — electric generator; CD — condenser; P — pump; V— valve; EV — evaporator; V-P — vacuum pump; CM — compressor; EM— electric motor; CD (HPP) — condenser of HPP; THR — throttle

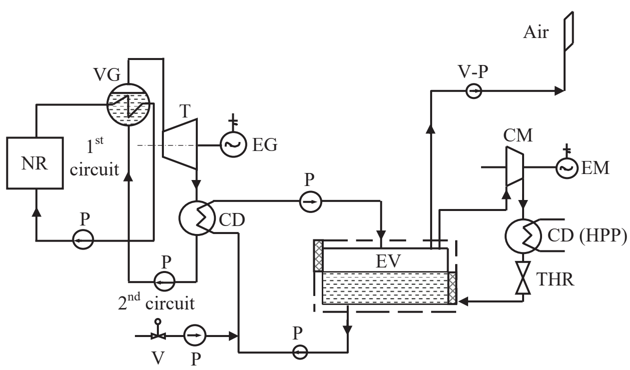


Fig. 2. Circuit of the cooling system of NPP with a heat pump

Operation of HPP occurs according to the following scheme. The cooled water enters the vacuum expander-evaporator, where the compressor maintains a pressure corresponding to the boiling of water at a temperature of 25 °C. The water, which enters the evaporator and boils at such pressure, and the compressor sucks off the generated vapor. The water cooled in the expander-evaporator to the boiling point enters the cooling system again. The compressor compresses the vapor to a pressure corresponding to a temperature of 50–70 °C. The vapor goes into the condenser, that is, the cooler, which also operates under vacuum, but at the higher temperature than the evaporator (pressure 0.015–0.02 MPa).

We should also note that this approach can simplify the circuit of the thermotransformer (Fig. 1, 2) greatly, increase its energy efficiency (by 20–40 %) and provide the required level of water cooling in comparison with other types of HPP.

The implementation of the described circuit of HPP requires additional work for compression of air released during boiling and dissolved in water. This work may be considerable, because the specific volume of water increases sharply with a decrease in the boiling point. It will decrease the efficiency of HPP. We also should note one more feature. Air suction into the system is possible in operation under vacuum due to incomplete tightness of heat exchanger seals. Leakage of uncondensed gases into the system worsens heat engineering efficiency of the condenser and reduces the compressor performance. Therefore, to avoid this, it is necessary to remove a considerable amount of air at the launching of the plant by means of a special blowing and removal system. We can eliminate many of disadvantages of the circuit described above by inclusion of a mixing heater-deaerator in the circuit of such HPP.

The compressor is the most complex unit of such system. It has to operate at a considerable degree of compression $(\bar{\pi}_{cm}=10)$, absorbing large volumes of vapor at sufficiently deep vacuum. As an example, the specific volume of water vapor is about 100 times higher than for ammonia under all other equal conditions. We have to have a compressor with about 1,000 rotations per minute to ensure suction of the volume of vapor generated in the evaporator at a temperature of about 25 °C. A use of a turbocompressor is prevalent among all known types of compressors, although a number of authors propose to use a screw compressor in HPP [11].

A small pressure difference in the vapor cycle makes possible creation of a turbocompressor using lightweight structural materials, since both axial and radial loads on bearings associated with aerodynamic forces are relatively small. The compressor will be relatively compact.

As the flow of cooling water at the outlet of the condenser of the second circuit is quite large (from 1,200 t/h for HPP of low power, up to 80,000 t/h for HPP of high power), it is necessary to create heat pump stations. Addition of vacuum-evaporative HPP of different design capacity will make it possible to carry out more flexible regulation of cold production of the plant by switching off part of plants depending on the ambient temperature. In addition, this reduces a cost for creation of turbocompressors by reducing of their design unit capacity.

5. Thermodynamic analysis of the cycle of a vacuum-evaporative heat pump plant

We can identify possibilities of increasing of energy efficiency of HPP due to improvement of the design of its elements in thermodynamic analysis of the plant, taking into account its mode characteristics.

To determine mode characteristics of HPP, it is necessary to have a model, which makes possible to forecast characteristics of HPP depending on the main determining factors, such as temperature of the low-potential coolant at the inlet of the evaporator – $T_{lpc1,}$ and temperature of the coolant at the outlet of the condenser – T_{w2}, at the design stage.

We based construction of characteristics of HPP on the equation of heat balance, which reflects a change in thermal loads associated with a change in the heat capacity of coolants located on hot and cold sides, as well as the machine itself.

We presented the main calculation relationships of the thermodynamic model of static characteristics of HPP below.

We determined Q_0 cooling capacity of the evaporator by the combined solution to the following equations:

$$Q_0 = m_{ref}\left[\left(1-x_{in}\right)r+\left(i_{cm}^{in}-i_{ev}^{out}\right)\right], \quad (1)$$

$$Q_0 = G_{lpc}c_{lpc}\left(T_{lpc1}-T_{lpc2}\right)=G_{lpc}c_{lpc}\bar{\eta}_{ev}\left(T_{lpc1}-T_0\right), \quad (2)$$

$$\bar{\eta}_{ev} = 1-e^{-\frac{k_{ev}A_{ev}}{G_{lpc}c_{lpc}}}, \quad (3)$$

$$x_{in} = 1-\frac{G_{lpc}c_p'}{m_{ref}r}\bar{\eta}_{ev}\left(T_{lpc1}-T_0\right)+\frac{c_p''}{r}\Delta T_{oh}, \quad (4)$$

$$m_{ref} = \lambda_{cm}\bar{V}_T/v_{cm}, \quad (5)$$

where m_{ref} is the mass flow rate of refrigerant in the cycle, kg/s; x_{in} is the mass vapor content at the inlet into the evaporator; r is the latent heat of vaporization, kJ/kg; i_{cm}^{in} is the enthalpy of refrigerant vapor at the compressor inlet, kJ/kg; i_{ev}^{out} is the enthalpy of vapor at the outlet of the evaporator, kJ/kg; G_{lpc} is the mass flow of low-potential coolant through the evaporator, kg/s; c_{lpc} is the heat capacity of low-potential coolant, kJ/(kg·K); $\bar{\eta}_{ev}$ is the coefficient of thermal-technical efficiency of the evaporator; $k_{ev}A_{ev}$ is the heat transfer intensity in the evaporator, kW/K; c_p', c_p'' is the heat capacity of liquid and saturated refrigerant vapor, kJ/(kg·K); ΔT_{oh} is the value of overheating in the evaporator, K; λ_{cm} is the coefficient of compressor feed; $\bar{V}_T$ is the theoretical volumetric capacity of the compressor, m^3/s; v_{cm} is the specific volume of vapor at suction into the compressor, m^3/kg.

We found λ_{cm} coefficient of compressor feed according to [12]:

$$\lambda_{cm} = 1-0.05\left[\bar{\pi}_{cm}^{0.869}-1\right], \quad (6)$$

where $\bar{\pi}_{cm}$ is the degree of compression in the compressor.

We found thermal load of the condenser from equations:

$$Q_h = m_{ref}\left(i_{cd}^{in}-i_{cd}^{out}\right), \quad (7)$$

$$Q_h = G_w c_w\left(T_{w2}-T_{w1}\right)=G_w c_w\bar{\eta}_{cd}\left(T_{cond}-T_{w1}\right), \quad (8)$$

$$\bar{\eta}_{cd} = 1-e^{-\frac{k_{cd}A_{cd}}{G_w c_w}}, \quad (9)$$

where i_{cd}^{in}, i_{cd}^{out} is the enthalpy of refrigerant at the inlet and outlet of the condenser, kJ/kg; G_w is the mass flow rate of coolant through the condenser, kg/s; c_w is the heat capacity of coolant, which cools the condenser, kJ/(kg·K); $\bar{\eta}_{cd}$ is the coefficient of thermal-technical efficiency of the condenser; $k_{cd}A_{cd}$ is the heat transfer intensity in the condenser, kW/K.

We defined evaporation and condensation temperatures in the second and subsequent approximations as the solution to:

$$T_0 = T_{lpc1}-\frac{Q_0}{G_{lpc}c_{lpc}\bar{\eta}_{ev}}, \quad (10)$$

$$T_{cond} = T_{w1}+\frac{Q_h}{G_w c_w\bar{\eta}_{cd}}. \quad (11)$$

We established temperatures of refrigerant at the outlet of the condenser – T_{w2}, and of the evaporator – T_{lpc2} using the following expressions:

$$T_{w2} = T_{cond}-\left(T_{cond}-T_{w1}\right)\bar{\eta}_{cd}, \quad (12)$$

$$T_{lpc2} = T_{lpc1}-\frac{Q_0}{G_{lpc}c_{lpc}}. \quad (13)$$

Temperature of overcooling of the condensate of refrigerant:

$$T_{oc} = T_{cond}-\frac{m_{ref}c_p'}{G_w c_w}\bar{\eta}_{cd}\left(T_2-T_{w1}\right)+\frac{c_p'}{r}. \quad (14)$$

We defined T_2 vapor temperature at the end of the compression process as:

$$T_2 = T_1\left[\frac{1}{\eta_{is}}\left(\bar{\pi}_{cm}^{\frac{k-1}{k}}-1\right)+1\right], \quad (15)$$

where $T_1 = T_0+\Delta T_{oh}$; η_{is} is the isentropic efficiency of the compressor; k is the adiabatic coefficient of refrigerant.

Compressor power consumption:

$$N_{cm} = \frac{\lambda_{cm}\bar{V}_T}{\nu_{cm}}\frac{l}{\eta_{is}}, \quad (16)$$

where l is the work of compression in the cycle.

Due to the isenthalpic nature of the process in the throttle valve of HPP, we determined the flow of refrigerant through the throttle and evaporator in the second and subsequent approximations by formula:

$$m_{ref} = \varpi\cdot f\sqrt{2\rho_{in}\left(P_{cond}-P_0\right)}, \quad (17)$$

where ϖ is the coefficient of leakage; f is the throttle cross-section, m^2; ρ_{in} is the density of liquid in front of the control valve, kg/m^3; P_{cond}, P_0 are the condensation and evaporation pressures, respectively, Pa.

We determined P_{suc} suction and P_{dis} discharge pressures in the compressor taking into account pressure losses in ΔP_{suc} suction and ΔP_{dis} discharge pipelines and valves of the compressor as:

$$P_{suc} = P_0-\Delta P_{suc} \quad (18)$$

and

$$P_{dis} = P_{cond}+\Delta P_{dis}. \quad (19)$$

According to the recommendations from paper [13], we performed calculation of thermal-hydraulic processes in the circuit of HPP to determine ΔP_{suc} and ΔP_{dis}.

The solution to the problem in question consisted in the iterative process, which consisted in the cyclic use of equations (1) to (19).

We carried out calculations for the 10 MW VVR-SM nuclear reactor, which operated under the climatic conditions of Central Asia. We determined mode characteristics of the vacuum-evaporative HPP with a turbocompressor with a theoretical volume flow of 5.2 m^3/s, which provided the design cold capacity of Q_0=300 kW, as an example of application of the proposed method (Fig. 3).

We constructed a nomogram for the graph-analytic determination of evaporation and condensation temperatures in the cycle at different Q_0, and variations of T_{lpc1} and T_{w2} (Fig. 3).

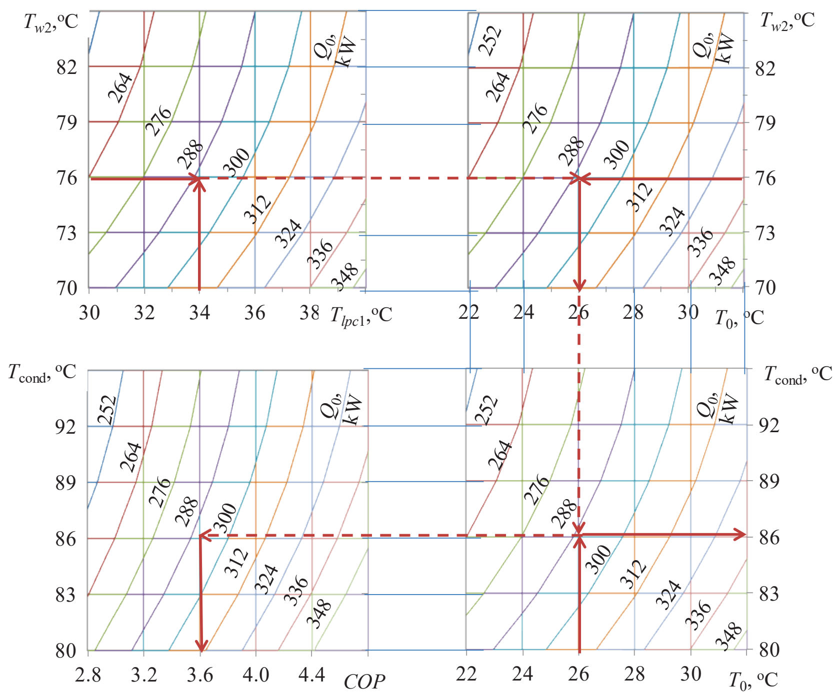


Fig. 3. Nomogram for determining the mode characteristics of HPP with the turbocompressor with a theoretical volume flow of 5.2 m^3/s

We used the theory of experiment planning in its construction [14]. We obtained regression models for the following characteristics: $Q_0=f(T_{lpc1}, T_{w2})$; $T_0=f(Q_0, T_{lpc1}, T_{w2})$; $T_{cond}=f(Q_0, T_0, T_{w2})$; $N_{cm}=f(T_0, T_{cond})$. We verified the adequacy of the regression model using the Fisher's test. We evaluated the significance of the regression coefficients using the Student's test [14]. Then, we found the correlation between parameters of each model using the obtained regression models.

The proposed nomogram (Fig. 3) shows clearly the relationship between the basic parameters, which gives a designer possibility to evaluate their effect on the plant efficiency without a need for multivariate calculations according to models (1) to (19).

There are the results of calculations of theoretical cycles of HPP with R718 refrigerant below. Fig. 4, *a* shows a change in $\bar{\pi}_{cm}$ degree of compression at changes in evaporation temperature in the cycle and at different temperature limits ($T_{cond}-T_0$).

Fig. 4, *b* shows the influence of T_0 evaporation temperature in the cycle and temperature limits of the cycle on T_2 temperature at the end of the compression process. As we can see in Fig. 4, *b*, R718 cycles have high temperatures at the end of the compression process, which requires consideration of possibility of reducing of irreversibility losses in the vapor overheating zone by application of a two-section condenser or cooling of this zone by injection of refrigerant into the injection line after throttling.

Fig. 5 shows the effect of pressure losses in suction and discharge pipelines on the cooling coefficient of the plant $COP=Q_0/N_{cm}$. We took the following parameters as fixed ones in calculations: evaporation temperature $T_0=20$ °C; condensation temperature $T_{cond}=50$ °C; isentropic efficiency of the compressor $\eta_{is}=0.9$; pressure loss in the evaporator $\Delta P_{ev}=0.01$ bar; condenser pressure loss $\Delta P_{cd}=0.01$ bar.

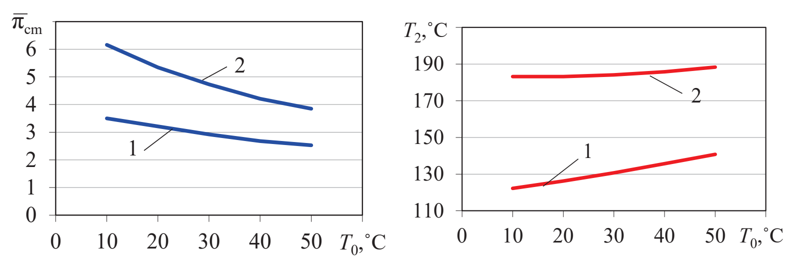
Fig. 6 illustrates the influence of the change in η_{is} isentropic efficiency of the compressor on *COP* at different evaporation temperatures in the cycle.

We can see in Fig. 6 a significant increase in *COP* (by 2.5 units at $\eta_{is}=0.5$, and by 4.4 units at $\eta_{is}=0.9$) at an increase in the evaporation temperature by 10 °C.

Analysis of a nature of inclination of boundary curves on *T-s* diagram made possible to conclude that there were irreversible losses in the thermodynamic cycle. Analysis of *T-s* diagram for different refrigerants (Fig. 7) showed what specific properties water has as a refrigerant compared to others. Thus, there was a sharp rise in the left boundary curve for R718 refrigerant. In practice, this will help to reduce irreversible losses during the throttling process.

The sharp inclination of isobars in the region of overheated vapor indicates possible large irreversible losses due to overheating of vapor during compression and high temperatures at the end of the compression process (Fig. 7).

Therefore, it is possible to use a two-section condenser containing a separately cooled section to remove overheating (pre-condensation) and a section of condensation in the circuit of HPP.



a *b*

Fig. 4. Dependence of parameters of HPP cycle with R718 refrigerant on T_0 evaporation temperature in the cycle at different temperature limits of the cycle ($T_{cond}-T_0$): *a* – $\bar{\pi}_{cm}$ degree of compression; *b* – T_2 temperature at the end of the compression process; 1 – ($T_{cond}-T_0$)=20 °C; 2 – ($T_{cond}-T_0$)=30 °C

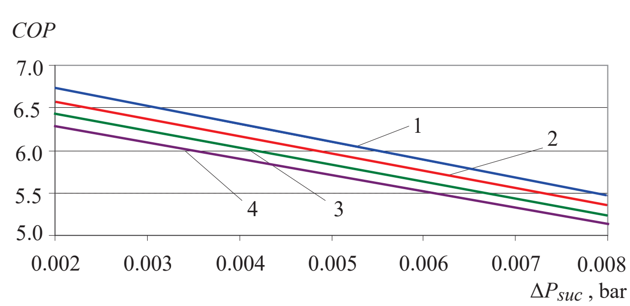


Fig. 5. Influence of hydraulic resistances in ΔP_{suc} suction and ΔP_{dis} discharge pipelines on the cooling coefficient *COP*: 1 – ΔP_{dis}=0.01 bar; 2 – ΔP_{dis}=0.015 bar; 3 – ΔP_{dis}=0.02 bar; 4 – ΔP_{dis}=0.025 bar

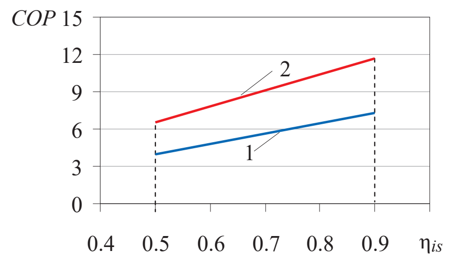


Fig. 6. Change in *COP* depending on η_{is} at different T_0 and T_{cond}=50 °C: 1 — T_0=20 °C; 2 — T_0=30 °C

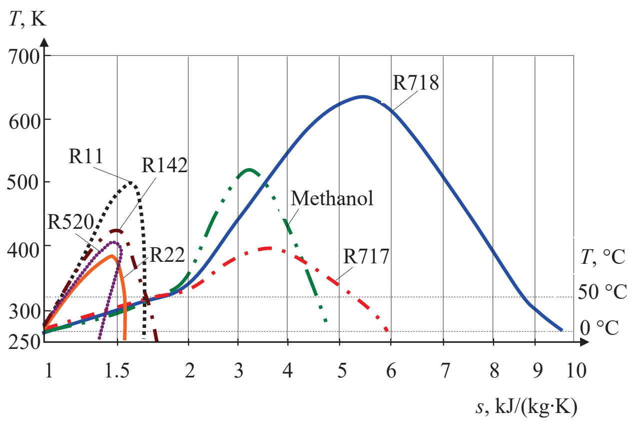


Fig. 7. Nature of the saturation curve for different refrigerants on *T-s* diagram

It is necessary to determine a need to overheat vapor before suction into the compressor for water vapor cycle separately for each compressor. On the one hand, overheating is necessary for safe operation of a compressor, since probability of condensation at the outlet of a nozzle is quite high for axial compressors [15]. On the other hand, calculations determined that vapor overheating before suction into the compressor only lowers energy performance of a plant in the cycle with R718. For example, the use of a regenerative heat exchanger in the cycle with R718 leads to a decrease in *COP* from 7.24 (without regeneration) to 6.85 (with regeneration) even with a sufficiently high regeneration factor η_{rhe}=0.8. There is a linear decrease of *COP* by 0.036 for each degree of increase in overheating in the cycle.

Analyzing thermodynamic properties of R718 as a refrigerant, we established that the pressure level is low in the compressor due to the low molecular weight of R718, so it is extremely problematic to ensure that the temperature difference in the cycle (between evaporation and condensation temperature) is over 40 °C at one-stage compression. At the same time, we can note that the implementation of intermediate cooling for the two-stage cycle provides a significant increase in *COP* by 20 % on average compared with a simple one-stage circuit, but it leads to a complication of HPP circuit.

6. Methodology for choosing a technological circuit of a vacuum-evaporative heat pump plant

We have to ground the decision of a choice of technological circuit of HPP on the analysis of a large number of factors. This is always a compromise between energy, economic and environmental performance.

It is expedient to use the method of graph-analytical thermoeconomic analysis based on construction of C-curves to choose the rational configuration of the cycle and the corresponding structural-topological circuit of HPP [16, 17]. Construction of C-curves makes design decisions easier because it provides a clear picture of the relationship between exergy costs and other optimization factors, such as capital and operating costs of a plant. An important element of the methodology is estimation of the level of structural complexity of the technological circuit. With respect to circuit-cycle solutions of heat pump plants, we can estimate a level of structural complexity of the circuit using D_{circ} complexity criterion. The most universal is the criterion of complexity of the following form $D_{circ} = D_i^{/}(2\bar{m}+\bar{p})$. It takes into account $\bar{p}$ total number of interactions of the thermal engineering system with the environment, $\bar{m}$ total number of technological connections between elements of the system and $D_i^{/}$ total complexity of all elements. Paper [16] presents the technique of graph-analytic optimization of circuit solutions by the criterion of structural complexity.

We considered technological circuits of vacuum-evaporative HPP of different levels of complexity, which operated by one-stage and two-stage cycles (Fig. 8), as competing options.

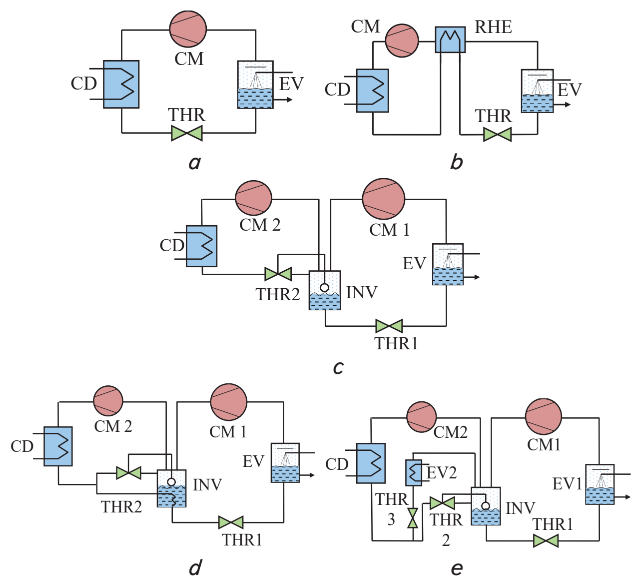


Fig. 8. HPP circuits of D_{circ} different complexity: *a* — a one-stage circuit (D_{circ}=11); *b* — a one-stage circuit with regenerative heat exchanger (RHE) (D_{circ}=15); *c* — a two-stage circuit with an open intermediate vessel (INV) (D_{circ}=20); *d* — a two-stage circuit with a closed intermediate vessel (D_{circ}=24); *e* — a two-stage circuit with a closed intermediate vessel and an additional evaporator (D_{circ}=27)

There are following designations adopted in Fig. 8: CD – condenser; CM – compressor; EV – evaporator; THR – throttle.

We performed the analysis under the following conditions. All HPP operated within the temperature limits of the cycle T_0=10 °C, T_{cond}=35 °C and T_0=25 °C, T_{cond}=50 °C. The calculation cooling capacity of plants was Q_0=300 kW, 600 kW, and 900 kW.

We should note that the calculation performance of a plant is an important factor, which affects internal energy dissipation in the HPP cycle and its structural characteristics (type of equipment). Therefore, creation of a single high-capacity HPP is not always expedient by technical and economic reasons. In addition, there is a high likelihood that a single high-capacity plant will operate in the underloading mode with an increase in energy dissipation in the cycle in practice [17]. Thus, to cover the required refrigeration load, it is necessary to use several units of different refrigerating capacity operating in parallel. In this case, it is possible to provide a more flexible regulation of cooling capacity during the entire period of operation.

A vacuum-evaporative HPP uses circulating water from the cooling system of the second circuit of NPP as a source of low-potential heat. That is, water, which comes from the condenser of a turbine with G_{lpc} constant calculation flow and fixed temperatures at the inlet T_{lpc1} and the output $T_{lpc2}=T_0$ of the HPP evaporator. The values of the mentioned parameters correspond to the conditions of technological processes at NPP. We took temperatures of the low-potential coolant (water) at the inlet of the evaporator T_{lpc1} and the water at the inlet to the condenser T_{w1} at T_0=10 °C and T_{cond}=35 °C as equal to 20 °C, and at T_0=25 °C and T_{cond}=50 °C – 35 °C. There is cooling of water in the evaporator by 5 °C and heating of water in the condenser by 10 °C.

We carried out estimation of the market value of elements of the plant using mathematical models developed based on the method of correlation-regression analysis. The regression dependence contained the parameters, which influence the capital cost of an element mostly. The selection criteria for pricing parameters were several factors: «purpose – quality – productivity».

Table 1 shows the dependencies for calculation of the capital cost of the basic equipment of HPP depending on the selected parameter.

When we calculated the volume of a contact-type vacuum evaporator, we assumed that water entered the evaporator from the cooling system of the second circuit of NPP with V_{lpc1}^{SCIR} volume. Part of water, when it entered a vacuum, transformed into vapor of $V_{lpc1}^{SCIR(1)}$ volume (Fig. 9). $V_{lpc1}^{SCIR(2)}$ remaining volume of water, cooled to T_0, fall to the bottom of the evaporator. In addition, a moist saturated refrigerant vapor (a mixture of water and saturated vapor) entered the evaporator from the throttle valve with T_0 temperature and V_{ref}^{thr} volume. A part of it expanded to $V_{ref}^{thr(1)}$ volume, replenished $V_{ref}^{cm}=V_{lpc1}^{SCIR(1)}+V_{ref}^{thr(1)}$ volume of vapor and $V_{ref}^{thr(2)}$ part replenished of $V_{lpc2}^{SCIR}=V_{lpc1}^{SCIR(2)}+V_{ref}^{thr(2)}$ volume of liquid at the bottom of the evaporator. The compressor removed V_{ref}^{cm} volume of vapor from the evaporator, and a water pump removed V_{lpc2}^{SCIR} volume of liquid from the cooling system of the second circuit of NPP (P) to the turbine condenser of NPP.

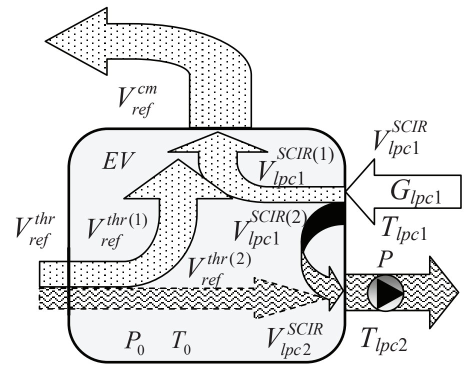


Fig. 9. Circuit for calculation of the volume of a vacuum evaporator of the contact type

Then, we assumed that there was vapor of V_{ref}^{cm} volume and liquid of V_{lpc2}^{SCIR} volume in the evaporator every second and they were, respectively, equal to the second volumetric flow rate of the refrigerant vapor through the compressor (volumetric compressor capacity, m^3/s) and the second volumetric flow rate of water through P, so we could calculate the volume of the evaporator as:

$$V_{ev}=V_{ref}^{cm}+V_{lpc2}^{SCIR}.$$

Since temperature at the inlet and outlet of the heat exchanger equipment, and condensation temperature were specified in the pre-design calculation provided the construction of the thermodynamic cycle of HPP, we determined $\bar{\eta}_{cd}$ thermal-technical efficiency of the shell-tube condenser for the calculation of its cost as:

$$\bar{\eta}_{cd}=\frac{T_{w2}-T_{w1}}{T_{cond}-T_{w1}}.$$

We presented the results of calculations of exergy and technical-and-economic parameters in the form of C-curves in the coordinates of «destruction of E_D exergy – $z^{/}$ specific capital value» to choose the technological circuit (Fig. 10). We defined the total destruction of exergy in the plant as:

$$E_D=Q_0\left(1/COP-1/COP_C\right),$$

and $z^{/}$ specific capital cost is a ratio of $\sum C_i$ total capital cost of the plant to τ_{oper}=5,000 h the number of hours of operation of HPP per year. COP_C is the refrigeration coefficient of the ideal Carnot cycle in the equation.

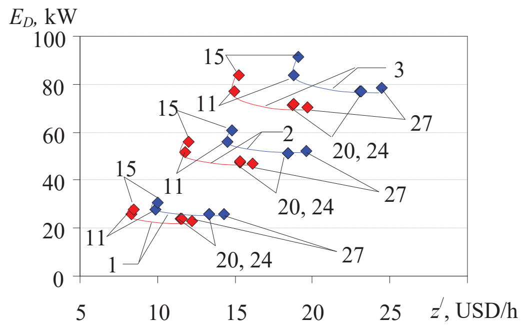
Table 1

Functions of the capital cost of the basic plant equipment

Basic equipment	X determining factor	C_i cost function (USD)	Literature source
Electric motor	Power of the motor (kW)	$110\cdot X+5{,}000$	[18]
Turbocompressor	Power on the shaft (kW)	$798.71124\cdot X^{0.592}$	[15]
Shell-tube condenser	Thermal technical efficiency, mass flow G_w (kg/s)	$1{,}000\cdot G_w\sqrt{\dfrac{X}{1-X}}$	[19]
Vacuum evaporator and intermediate vessel	Volume (m^3)	$600\cdot X^{0.78}+8{,}000$	[18]

Figures indicated the complexity of D_{circ} circuit in Fig. 10. Fig. 10 shows that one-stage HPP circuits, which operate within the temperature limits of the cycle T_0=25 °C and T_{cond}=50 °C, with complexity D_{circ}=11, have the lowest $z^{/}$ specific capital cost, and HPP circuits, which operate within the temperature limits of the cycle T_0=10 °C and T_{cond}=35 °C, with complexity D_{circ}=27, have the highest $z^{/}$.

All HPP circuits (Fig. 10), which operate within temperature limits of the cycle T_0=25 °C and T_{cond}=50 °C, are more attractive both in price and by lower values of E_D than those which operate at T_0=10 °C and T_{cond}=35 °C.



◆ – T_0=25 °C and T_{cond}=50 °C; ◆ – T_0 =10 °C and T_{cond}=35 °C

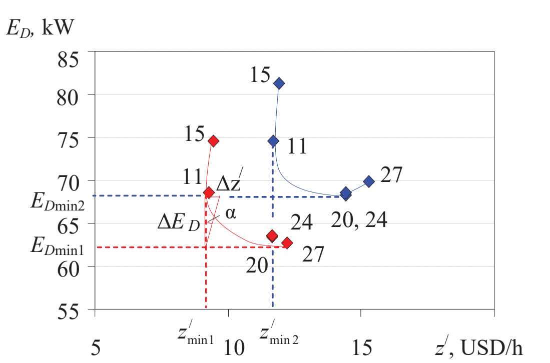
Fig. 10. Dependence of E_D on z' for circuits of a vacuum-evaporative HPP of different complexity with cold producing capacity: 1 — Q_0=300 kW; 2 — Q_0=600 kW; 3 — Q_0=900 kW

Fig. 11 shows the dependence of E_D on z' for circuits of a vacuum-evaporative HPP of different complexity and cold producing capacity of Q_0=800 kW, which operate at different temperature limits of the cycle. Fig. 11 illustrates a graphical method of choosing a circuit with the minimum total costs at the current tariffs for electric power and taking into account investments for its creation. Fig. 11 shows that the circuit with D_{circ}=27 shows the least destruction of $E_{D\min 1}$ exergy at T_0=25 °C and T_{cond}=50 °C, and the circuit with D_{circ}=24 – the smallest value of $E_{D\min 2}$ at T_0=10 °C and T_{cond}=35 °C (Fig. 8). In both cases, the smallest capital costs for creation of a plant $z'_{\min 1}$ and $z'_{\min 2}$ correspond to the circuit with D_{circ}=11.

We found the agreed optimum by assumption of a linear relationship between ΔE_D exergy overcosts and $\Delta z'$ costs [20]:

$$\Delta z' = \mathrm{tg}\,\alpha \cdot \Delta E_D,$$

where we assumed $\mathrm{tg}\,\alpha$ tangent of an angle of inclination equal to the cost of fuel for the system.



◆ – T_0=25 °C and T_{cond}=50 °C; ◆ – T_0 =10 °C and T_{cond}=35 °C

Fig. 11. Dependence of E_D on z' for circuits of a vacuum-evaporative HPP of different complexity at Q_0=800 kW, τ_{oper}=8,000 h

If we have an electric drive in a plant, it is the cost of electricity, and for gas-powered plants, it is the cost of gas. Authors of paper [20] considered $\mathrm{tg}\,\alpha$ equal to the cost of conventional fuel on the world market.

The α angle increases as the cost of fuel increases. However, the issue of fuel costs is not a priority one with regard to application of HPP at NPP, since an enterprise produces electricity itself and other tariffs determine its price. Electricity costs for the compressor and pumps of HPP relate to the expenses for needs of an enterprise. We can estimate them at the cost price of production. But α angle will not exceed 6° even at the cost of electricity at existing tariffs for electricity generating enterprises.

Thus, the analysis of C-curves is the first stage of thermoeconomic optimization, which makes it possible to choose a circuit by energy (destruction of exergy) and economic (capital and operating costs) indicators. The complexity criterion plays here the role of «a navigator» for choosing a circuit.

Thermoeconomic analysis with the use of a graphical apparatus for construction of C-curves showed that it is better to prefer the choice of a simple one-stage HPP circuit for a low cost of electricity. However, we could make the choice in favor of another, more complex circuit at any other combination of economic indicators.

7. Thermoeconomic optimization of vacuum-evaporative heat pump plants with R718 refrigerant

After choosing a circuit for a vacuum-evaporative HPP, we performed thermoeconomic optimization of its mode-design characteristics. The optimization reduces capital costs for creation of a plant and operational costs during its operation.

We developed a thermoeconomic model of HPP based on the autonomous method of thermoeconomic optimization. It makes possible to take into account both thermodynamic and economic parameters in optimization of the design and selection of economic modes of the plant operation [21].

We took the vacuum-evaporative HPP, which operated according to a one-stage circuit, as an object for optimization. The plant had a turbocompressor, a contact type evaporator, a horizontal smooth-tube shell-tube condenser with intertubular boiling of the refrigerant, a throttle valve and piston pumps for pumping of coolants through HPP heat exchangers. We performed modeling of thermohydraulic processes in the circuit of refrigerant circulation by the methodology described in a paper [21]. We revealed the following parameters of the original HPP: temperature and pressure of refrigerant at nodal points of the cycle, volumetric and mass flows and rates of refrigerant and coolants, a coefficient of heat transfer of the condenser, geometric characteristics and hydraulic resistance of heat exchangers and pipelines.

We took ΔT_{cd} average logarithmic temperature head and ΔT_w^{cd} heating of the coolant (water) in the condenser of the heat pump as the optimizing variables for solution of the optimization problem. The objective function was resulting costs determined by the expression:

$$PZ = \begin{bmatrix} c_{EE}\left(e_{cm} + e_p + e_{cdp}\right) + z_{cm} + \\ + z_{ev} + z_p + z_{cd} + z_{cdp} + z_{thr} \end{bmatrix} \cdot \tau_{oper} \cdot n_s + $$

$$+ C_{cm} + C_{ev} + C_p + C_{cd} + C_{cdp} + C_{thr}, \qquad (20)$$

where c_{EE} is the cost of electricity; e_{cm}, e_p, e_{cdp} is the exergy supplied to the compressor and pumps; n_s is the number of seasons of plant operation; τ_{oper} is the duration of operation of the system per year; C_{cm}, C_{ev}, C_p, C_{cd}, C_{cdp} and C_{thr} are the costs of a compressor, an evaporator, a water pump from the cooling system of the second circuit of NPP, a condenser, a pump for the water heated in condenser (CDP), and a throttle valve; z_{cm}, z_{ev}, z_p, z_{cd}, z_{cdp} and z_{thr} are the annual total deductions from the cost.

We expressed the economic indicators and exergy flows included in the objective function (20) in the form of functional dependences on the given cooling capacity of the evaporator Q_0=const and on the selected optimized variables:

$$e_{cm}, C_{cm}, z_{cm}, C_{ev}, z_{ev}, C_{cd}, z_{cd}, e_{cdp}, C_{cdp}, z_{cdp} = \\ = f\left(Q_0, \delta T_{cd}, \Delta T_w^{cd}\right). \quad (21)$$

Due to the constant cooling capacity of the evaporator, we assumed that under the given conditions:

$$e_p, C_p, z_p, C_{thr}, z_{thr} = f\left(Q_0\right) = \text{const}.$$

In order to solve the optimization problem, we represented functional expressions (21) present in the objective function of resulting costs (20) as detailed analytical dependencies, which described energy processes that occur in individual elements of a given heat pump [21].

We took the following initial data for the solution of the optimization problem: cold capacity of HPP Q_0=800 kW; volumetric flow rate of vapor for suction into the compressor 14.4 m^3/s; temperature of the cooling water at the inlet and outlet of the evaporator, respectively T_{lpc1}=40 °C and $T_{lpc2}=T_0$=25 °C; water pressure in the reverse pipeline (after P) P_{lpc2}=4 bar; temperature of the coolant (water) at the inlet into the condenser T_{w1}=25 °C; τ_{oper}=8,000 h; analyzed n_s=25 years; electricity tariff c_{EE}=0.0727 USD/(kW·h), normative coefficient of deductions from the cost of equipment k_{ni}=0.15; ambient temperature T_{am}=32 °C.

We also took the following geometrical characteristics of the condenser: inner and outer diameter of tubes, respectively, d_{in}=0.016 m and d_{out}=0.020 m; a number of tubes in the casing n_{tub}=100; tube material was brass with a coefficient of thermal conductivity λ_{tub}=93 W/(m·K); a number of CD moves n_{mov}=2; a number of partitions n_{part}=12. The internal diameter of the inlet and outlet pipelines of the condenser was 0.12 m; length of pipelines – 10 m; drop in height of the outlet pipeline – 5 m.

The optimization decreased the drive capacity of the compressor by 44 kW and the capacity of the water pump in the condenser increased by 8 kW. But still, the total power consumption of HPP decreased by 36 kW with constant cold capacity of Q_0=800 W.

Fig. 12, 13 show the results of thermoeconomic optimization of the vacuum-evaporative HPP.

In Fig. 12, 13, ORIG – shows HPP under optimization, OPT – HPP obtained as a result of optimization.

Fig. 12 shows that the condensation temperature decreased by 9 °C due to optimization. The evaporation temperature remained unchanged because of operational requirements.

Fig. 13 shows changes in PZ resulting costs over 25 years of HPP operation. The figure shows that the PZ decreased by USD 535.917 thousand, including capital costs – by USD 9.534 thousand, and operating costs – by USD 526.383 thousand due to optimization. The economic impact of optimization reached almost 35 %.

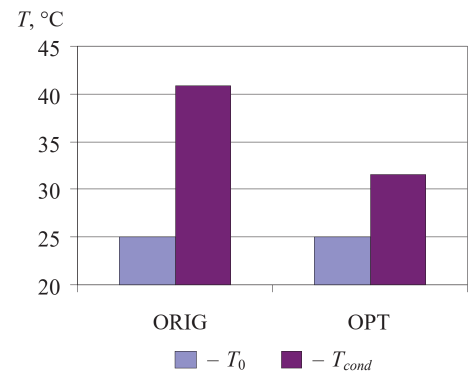


Fig. 12. Changes in evaporation and condensation temperature due to optimization

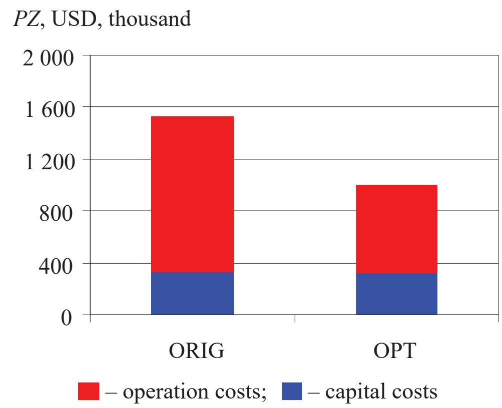


Fig. 13. Resulting costs of HPP

8. Discussion of results from solving a problem on the creation of a system of vacuum-evaporative cooling for NPP technological equipment

The study proposes the method for optimal design of a vacuum-evaporative HPP. We carried out the solution of the problem of optimal design of HPP at several levels. The first level involved the thermodynamic analysis of a vacuum-evaporative HPP cycle, taking into account its mode characteristics. We proposed a graph-analytic nomogram (Fig. 3) to forecast mode characteristics of a vacuum-evaporative HPP with R718 refrigerant (Fig. 3) constructed using the apparatus of the experiment planning theory. At the second level, there was choosing a technological circuit and cycle of HPP taking into account both energy and technical and economic indicators. We used the system approach to the design of thermotransformers in the study, instead of the traditional approach, which considers plant elements in isolation from the intended technological circuit. The system approach assumes taking into account relationships between the main elements conditioned by structural and topological features of a technological circuit. We used a graph-analytical apparatus of thermoeconomic optimization based on construction of C-curves for the search for a technological circuit. It made it possible to offer the most rational layout of a technological circuit of a vacuum-evaporative HPP for the given operating conditions of the second circuit of HPP (Fig. 11). At the third level, there was thermoeconomic optimization of mode-design parameters of HPP performed under condition of minimization of resulting costs. The developed thermoeconomic model of a vacuum-evaporative HPP (item 7) makes it possible to forecast a change in the cost of plant elements depending on a change in economic conditions, namely in the cost of electricity and a number of hours of plant operation per year.

Possible limitations of the proposed method include neglecting of environmental factors in design. Today, it is important to assess the environmental impact of the manufacturing process of production of elements of energy technology systems (for example, using ECO INDICATOR 99). It is advisable to consider the task of optimal design in combination with economic, environmental and energy indicators in further development of the research.

The practical application of the method is not limited to vacuum-evaporative plants. It is possible to use the basic concept of the proposed approach to the design of thermotransformers and the methods of analysis and optimization in the development of any other type of thermotransformers (absorption, air-compression).

9. Conclusions

1. We proposed two circuits for inclusion of a vacuum-evaporative HPP in the system of cooling of the second circuit of NPP. The first circuit involves the use of HPP in combination with an existing cooling tower and makes possible to cool water from 30 °C to 25 °C after a cooling tower. Only HPP cools water to the required parameters according to the second circuit.

2. We developed a thermodynamic model to forecast static characteristics of HPP. We analyzed thermodynamic properties of water as a refrigerant and evaluated their influence on mode parameters and energy efficiency of a vapor cycle. We established that water complies with all the environmental safety requirements for refrigerants of heat pumps fully. Its use makes possible to provide high cycle energy performance in comparison with synthetic refrigerants. The problematic aspects of the use include high temperature of a vapor cycle at the end of the compression process. It is possible to level the effect of the temperature on energy and operating performance of a plant by the use of a two-section condenser with utilization of heat to remove vapor overheating. It is necessary to consider possibility of the use of a circuit of injection of refrigerant into the discharge pipeline before the condenser after throttling at high temperatures of evaporation and condensation in a cycle (T_0=70 °C; T_{cond}=150 °C) to reduce losses from irreversibility in the zone of removal of vapor overheating. We can eliminate the significant influence of hydraulic resistance along the refrigerant circulation path on cycle efficiency by refusing to use surface evaporators and reducing a length of the suction pipeline. The difference between evaporation and condensation temperatures should not exceed 30 °C at low evaporation temperatures (up to 10 °C), otherwise the compression ratio will be significant (the compression ratio greater than 7).

3. We chose rational circuit-cycle solutions for a vacuum-evaporative heat pump plant using the graph-analytic apparatus for construction of C-curves. They ensure efficient operation of the plant for cooling of the technological equipment of the second circuit of NPP. We evaluated the capital capacity of the system. We established that the smallest capital expenditures for creation of a plant correspond to the circuit with structural complexity of D_{circ}=11.

4. We performed thermoeconomic optimization based on modeling of thermohydraulic processes in the circuit of the refrigerant circulation and determined mode-design characteristics of the plant, which correspond to the minimum of resulting costs during its operation. The condensation temperature decreased by 9 °C compared to the original HPP version due to optimization. The compressors drive power decreased by 44 kW. The overall economic impact of the optimization was 35 %.

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