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INFLUENCE OF PRESSURE IN THE TURBINE CONDENSER ON HEAT SUPPLY EFFICIENCY OF NPP WITH HEAT PUMP

В. Кравченко, А. Оверченко. **Вплив тиску в конденсаторі турбіни на ефективність теплопостачання АЕС з тепловим насосом.** Використання теплового насосу для теплопостачання дозволяє практично припинити теплове забруднення довкілля взимку при роботі теплових та атомних електростанцій. Якщо у якості низько потенційного джерела енергії теплового насосу (ТН) використати конденсатор парової турбіни, кількість відпущеної теплової енергії буде дорівнювати сумі теплової потужності конденсатора АЕС та потужності компресорів теплового насосу. З точки зору екологічної безпеки теплопостачання за рахунок комбінування електростанцій з тепловим насосом є актуальною задачею. Але відомо, що додаткова електрична потужність ТЕЦ через відсутність теплофікаційних відборів буде менша за потужність компресорів ТН. Таким чином з точки зору термодинамічної ефективності використання теплового насосу програє традиційній теплофікаційній установці (ТФУ). Метою роботи є визначення впливу кінцевого тиску в конденсаторі на термодинамічну ефективність атомної ТЕЦ з тепловим насосом. Розроблено математичну модель теплової схеми турбоустановки АЕС К-1000-5,8/1500 при роботі літом та взимку з ТФУ. При відпуску теплоти взимку у кількості 230 МВт електрична потужність блоку знижується на 43,5 МВт. Розроблено математичну модель теплового насосу, низько потенційним джерелом енергії для якого використовується конденсатор парової турбіни. Для забезпечення відпуску 230 МВт теплоти потужність компресору ТН має бути 48,4 МВт. Таким чином, якщо замінити ТФУ на ТН тієї ж потужності, електрична потужність знизиться на 4,8 МВт. Проведено розрахунки відносно впливу кінцевого тиску в конденсаторі на ексергетичний ККД АТЕЦ з ТН, який використовує всю потужність конденсатору турбіни. Аналіз отриманих результатів показав, що з підвищенням кінцевого тиску в конденсаторі ексергетичний ККД через підвищення відпущеної електричної потужності збільшується. Це пояснюється збільшенням коефіцієнта перетворення енергії теплового насосу.
Ключові слова: АЕС, тепловий насос, теплофікаційна установка, ексергетичний ККД

V. Kravchenko, A. Overchenko. **Influence of pressure in the turbine condenser on heat supply efficiency of NPP with heat pump**. The use of a heat pump for heat supply makes it possible to practically stop thermal pollution of the environment during the operation of thermal and nuclear power plants in winter. If a steam turbine condenser is used as a low-potential energy source for heat pump, the amount of released thermal energy will be equal to the sum of the thermal power of the NPP condenser and the power of the heat pump compressors. From the point of view of environmental safety, heat supply by combining power plant with a heat pump is an urgent task. But it is known that the due to the lack of steam extraction for water heating, the additional electrical power of the cogeneration heat and power plant will be less than the capacity of the heat pump compressors. Thus, in terms of thermodynamic efficiency, the use of a heat pump loses to a traditional cogeneration plant. The purpose of the work is to determine the influence of the final pressure in the turbine condenser on the thermodynamic efficiency of a nuclear power plant with a heat pump. A mathematical model of the thermal scheme of the K-1000-5.8/1500 NPP turbo-plant during summer and winter operation with heating plant has been developed. With the heating plant capacity of 230 MW, the electric capacity of NPP unit decreases by 43.5 MW. A mathematical model of a heat pump has been developed, for which a steam turbine condenser is used as a low-potential energy source. To ensure the release of 230 MW of heat, the power of the heat pump compressor must be 48.4 MW. Thus, if the heating plant is replaced with a heat pump of the same capacity, the electric power will decrease by 4.8 MW. Calculations were made regarding the influence of the final pressure in the condenser on the exergetic efficiency of the NPP with heat pump, which uses the entire capacity of the turbine condenser. The analysis of the obtained results showed that the exergetic efficiency due to the increase in electric power released in winter increases with the increase of the final pressure in the condenser. This is explained by an increase in the heat pump coefficient of performance.
Keywords: nuclear power plant, heat pump, heating plant, exergy efficiency

1. Introduction

Recently, thermal power plants and nuclear power plants have reached the limit of their thermodynamic perfection based on the accepted initial parameters, and in order to further increase their technical and economic indicators, it is necessary to look for new unconventional ways. One of the possible options for solving this problem is the use of heat pump technologies to recover the energy potential of discharge low-temperature flows [1, 2]. With a NPP efficiency of 32%, only a third of the useful heat is converted into electricity, which is sent to the consumer. The other two-thirds of the heat is irretrievably lost in the cooling system of the steam turbine condenser and is discharged with heated technical water into the environment. Such a significant loss is associated with the potential of this water (does not exceed 30 °C), which is too low for useful utilization. Thermal pollution is one of the significant disadvantages of thermal power plants and nuclear power plants. However, with the help of a heat pump (HP), this low potential heat can be transformed, increasing its temperature to 80…90 °C and making it sufficient for use. Thus, the study of combining nuclear power plants with heat pumps is an urgent task from the point of view of ecology and economy.

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2. Analysis of the latest research

Heat pumps are developing and improving. The number of heat pumps operating in Japan, Europe and the USA is estimated at tens of millions. Work is also being carried out in the direction of the combination of thermal power plants and nuclear power plants with heat pumps [3].

Depending on the type of heat pump, the work is performed either by a compressor (in compression HPs) or by thermochemical transformation (in absorption HPs).

The main value for evaluating the efficiency of these devices is the coefficient of performance (COP), which exceeds one. This means that the consumer receives several times more thermal energy than the electrical energy used to operate the heat pump compressor [1]:

$$\mathrm{COP} = \frac{Q_{\text{thermal}}}{N_{\text{compressor}}}, \tag{1}$$

where Q_{thermal} – thermal power released,

$N_{\text{compressor}}$ – electric power of the heat pump compressor.

The heat loss in the condensers of combined heat and power plant reaches values of 20...60% depending on the mode and work schedule. This means that in the summer, when there is no heating mode, the heat loss in the condenser will be equal to 60%. In the presence of heating, the capacity of the condenser decreases, and at the maximum heat capacity of the heating installation, the loss of heat in the condenser will be minimal (20%).

A very large amount of thermal energy can be obtained in heat pumps with a COP of at least 3.5. However, at the same time, it is necessary to spend some part of the generated electrical energy in the heat pump. The resulting heat can be sent to the regeneration system of the steam power plant of the power plant, thereby increasing the electrical efficiency. Or direct it to the heating of internal and external consumers, thus reducing the specific fuel consumption for heating and, as a result, the emission of greenhouse and toxic combustion products at thermal power plants.

Note that the circulating water is completely or partially cooled in the evaporator of the heat pump. Therefore, due to the reduction of its temperature in the turbine condenser, the vacuum improves and the possibility of generating additional electrical energy appears, which can be used to drive the heat pump compressor.

In order to increase the efficiency of thermal power plants and nuclear power plants at nominal and variable modes, various options for the use of heat pump equipment are proposed [4]. The most common option is the use of a compression type heat pump (CHP), which is characterized by high transformation coefficients. Another option is absorption heat pumps (AHP). They are characterized by the lowest energy consumption for their own needs, but they cost more. To date, a large theoretical and practical base has been accumulated on the use of heat pumps. Large-scale research was conducted in Ukraine and abroad, and schemes using heat pumps were developed in large quantities [1, 3].

According to theoretical studies [3], the thermal power received by the final consumer exceeds the thermal power of the source by 1.3 times. The use of a heat pump for heating consumers at a distance from the power plant using the heat of a low potential source covers all losses of the power plant, including losses in the condenser. If the heat pump is installed on the condenser of the steam turbine, in order to return the heat back into the cycle through the regeneration system, it is possible to significantly increase the efficiency of TPPs and NPPs. Unfortunately, this statement given in [3] is not confirmed by calculations.

Next, it is shown that the use of a heat pump at a nuclear power plant to dispose of condenser cooling system losses is economically feasible for two reasons:

1. In the condenser of the NPP turbine, the heat of vaporization that is lost makes up the largest part relative to the enthalpy difference of the TPP, so the use of a heat pump at the NPP will lead to a greater increase in its efficiency than at the TPP;

2. The final stages of the NPP steam turbine operate at high steam humidity, which causes additional losses in the turbine and increased erosive wear of the blades. Due to the heat pump, it is possible to return part of the heat of the condenser with increased potential to the last stages of the turbine, while increasing the dryness of the steam.

In [5], the basic thermal scheme of a nuclear power plant using a compression heat pump to return the heat of the condenser to the regeneration system and the last stages of the steam turbine is considered. At the same time, the heat pump condenser is connected to the regenerative heating system and replaces the first low-pressure heater.

With such a connection scheme, the design of the NPP steam turbine installation requires minimal changes. In this scheme, the heat pump evaporator is installed in the inlet or outlet channel of the cooling water, which greatly simplifies its installation.

Connecting the evaporator of the heat pump to the cooling water at the inlet to the condenser will allow to increase the generated power of the turbine due to two factors:

– the steam consumption in the last stages of the turbine will increase due to the fact that the heating of the condensate in the low-pressure heater 1 is carried out by a heat pump, and not by the extracted steam;

– there will be an opportunity to increase the activated heat transfer in the last stages of the turbine due to the reduction of the saturation pressure in the condenser as a result of heat removal by the evaporator of the heat pump from the cooling water and a decrease in its temperature.

However, the results of mathematical modeling [5] show that the power required for the operation of the heat pump compressor is higher than the electrical power additionally produced by the power unit, when the evaporator is placed both at the inlet and at the outlet of the cooling water of the main turbine condenser.

This conclusion is very important and practically cancels the expediency of using a heat pump in a complex with thermal power plants and nuclear power plants. That is, the improvement of the characteristics of the nuclear power plant due to the deterioration of the characteristics of the heat pump leads to the deterioration of the general characteristics of the power plant-heat pump complex.

However, from an environmental point of view, we can consider the possibility of using a compression heat pump in the main condenser circuit of the turbine to reduce the thermal discharges of the NPP to the environment.

Another option for the utilization of low-potential heat with the help of heat pumps is the heating of administrative, domestic and industrial buildings of the NPP and the nearest city. Currently, the heat for these purposes is taken from the selection of the working steam of the turbine, which reduces the production of electricity at the NPP. The use of heat produced by heat pumps with a coefficient of performance COP=4 for heating purposes allows:

– reduce heat emissions to the environment;

– eliminate (or reduce) steam withdrawals for heating needs.

Despite all the positive aspects of heat pumps, they are less common in the heat-cold supply systems of industrial facilities. The main reasons are the limitation of the temperature of the high-temperature heat carrier up to 55 °C and problems in creating heat pumps with a large unit capacity. Such limitations are laid down in the principle of operation of the heat pump and the Carnot cycle. It is impractical to heat the coolant to a temperature above 65...70°C, as the heat pump begins to consume too much energy.

These problems have been partially solved. High-temperature industrial heat pumps have already been developed and even put into operation. For example, a project was proposed to utilize the waste heat of the Leningrad NPP-2 steam turbine [6] with the help of a THCO2-2300 heat pump for heating mains water from 28 to 80 °C with a conversion factor of 5. Modern high-temperature heat pumps of the IWHSS series are capable of heating the coolant up to +95 °C, making it suitable for use in local heating and hot water systems.

To achieve an increase in temperature, the OCHSNER company developed a two-phase circular process [7]. High-temperature heat pumps are produced with a power from 190 to 750 kW in a single product, while the temperature of the heating water at the outlet of the condenser reaches +100 °C at a heat source temperature of +10 °C.

In [8], systems with large-sized LiBr-water absorption heat pumps for preheating circulating water in the district heating network from 45 to 80 °C are studied. A theoretical analysis of the system is presented, showing the advantages compared to the conventional absorption heat pump and the original heat supply system.

As it was indicated earlier, according to the results of calculations, it was found that when using heat pumps in a complex with a nuclear power plant (or TPP), the total thermodynamic efficiency of the complex decreases.

The purpose of this article is to determine the numerical value of this decrease and to determine the effect of increasing the final pressure in the turbine on the efficiency of the NPP + heat pump complex.

To achieve the set goal, it is necessary:

– to develop a mathematical model of a turbocharged nuclear power plant when operating in condensation and heating modes;

– develop a mathematical model of a heat pump;

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– conduct a study of the effect of pressure in the condenser on the overall efficiency of the NPP with a heat pump.

3. Calculation of the thermal scheme of the K-1000-5.8/25-2 turbine installation in condensation mode

To solve the given problem, a mathematical model of the thermal circuit of the K-1000-5.8/25-2 turbine installed on the power units of the Zaporizhzhya NPP was developed [9].

The purpose of calculating the thermal circuit is to determine the parameters and consumption of the working fluid at all calculation points of the circuit, indicators of thermodynamic efficiency. Some initial data are given in the Table 1.

The calculation is carried out according to the method [10]. A peculiarity of the calculation is the assumed temperature of the cooling water at the inlet to the turbine condenser. The pressure in the condenser is determined by the formula given in [9]. The initial data for the calculation and design of the K-33160 capacitor are given in [10]. The calculation sequence is as follows:

A. Calculation of the steam expansion process in a high-pressure cylinder (HPC) ($\eta_{oi}^{\text{HPC}} = 0.83$);

B. The pressure behind the low-type cylinder (LTC)(in the condenser) in this problem is determined by the temperature of the circulating water at the inlet to the condenser t_{cw1}.

The steam load of the condenser is determined by the ratio of steam flow into the condenser (D_k, kg/s)to the heat exchange surface:

$$d_k = D_k/F_k = 318/33160 = 0.00959 \text{ kg/(s}\times\text{m}^2).$$

Empirical coefficient x, which depends on the coefficient a=0.85 and t_{cw1}:

$$x = 0.12a(1+0.15t_{\text{cw1}}) = 0.12\times0.85(1+0.15\times18) = 0.3774.$$

Empirical coefficient b, which depends on d_k:

$$b = 0.52 – 7.2\times d_k = 0.52 – 7.2\times0.0095899 = 0.45095.$$

The number of tubes of one run of the condenser is equal to the total number of tubes of the condenser divided by the number of runs z=2.

The speed of circulating water in the condenser tubes in one stroke:

$$W_{\text{cw1}} = \frac{mD_k v_{\text{cw1}}}{0.785 d_{\text{tube}}^2 n_{\text{tube1}}} = \frac{54\times318\times.001}{0.785\times0.026^2\times\dfrac{27940}{2}} = 4.63 \text{ m/s.}$$

The multiplier Φ_z, which takes into account the effect of the number of water strokes z in the condenser:

$$\Phi_z = 1 + \frac{z-2}{10}\left(1 - \frac{t_{\text{cw1}}}{35}\right) = 1 + \frac{2-2}{10}\left(1 - \frac{18}{35}\right) = 1$$

Multiplier Φ_d, which takes into account the steam load in the condenser:

$$d_k^{\text{lim}} = (0.9 – 0.012\cdot t_{\text{cw1}})\times d_k = (0.9 – 0.012\times14)\times0.00958 = 0.00656,$$

since $d_k > d_k^{\text{lim}}$, then Φ_d=1

The average heat transfer coefficient in the condenser:

$$K = 4.07a\left(\frac{1.1\times w_{\text{cw1}}}{d_{\text{tube}}^{0.25}}\right)^x \left(1 - \frac{b\sqrt{a}}{1000}\times(35 - t_{\text{cw1}})\right)^2 \Phi_Z\Phi_d =$$

$$= 4.07\times0.85\times\left(\frac{1.1\times4.63}{0.026^{0\cdot25}}\right)^{0.2856}\left(1 - \frac{0.45095\times\sqrt{0.85}}{1000}\times(35-18)\right)\times1\times1 =$$

$$= 4.138 \text{ kWt/(m}^2\text{K).}$$

Condensation temperature:

$$t_c = t_{\text{cw1}} + \frac{r}{mc_p}\times\frac{\exp\left(\dfrac{K}{mc_p d_k}\right)}{\exp\left(\dfrac{K}{mc_p d_k}\right) - 1} = 18 + \frac{2400}{54\times4.19}\frac{\exp\left(\dfrac{4.13834}{54\times4.19\times0.00958}\right)}{\exp\left(\dfrac{4.13834}{54\times4.19\times0.00958}\right) - 1} = 30.46\ ^\circ\text{C}.$$

Corresponding condensing pressure P_k=4.36 kPa.

C. Calculation of the steam expansion process in low pressure cylinder ($\eta_{oi}^{\text{HPC}} = 0.82$).

D. Construction of the steam expansion process in the drive turbine of the feed pump.

E. Determination of parameters of heating steam in sampling and regenerative heaters, drains of heating steam, feed water and main condensate.

F. Determination of the flow rate of the working fluid at the nodal points of the scheme.

Steam consumption from a steam generator:

$$D_{\text{SG}} = \frac{Q_{\text{SG}}}{h_{\text{o}} + 0.005h_{\text{purging}} - 1.015h_{\text{FW}}} = \frac{3021.2 \times 1000}{2777 + 0.005 \times 1286 - 1.015 \times 973.14} = 1694.6 \text{ kg/s}$$

where Q_{SG} – the steam generator capacity is equal to the power of the reactor plus the power of the main circulation pumps:

$$Q_{\text{SG}} = 3000 + 4 \times 5.3 = 3021.2 \text{ MW}.$$

The results of the calculation of the parameters of the heating steam, the main condensate (MC) and the feed water (FW) are summarized in the Table 1.

Table 1

Summary table of bled steam, drain, main condensate (MC) and feed water (FW) parameters
(R1, R2 – reheater 1 and 2 stage, HPH – high pressure heater, LPH – low pressure heater,
D – deaerator, TFWP – turbine of feed water pump)

Extraction points	Heating steam and drain				
	0 R2	I R1	I HPH7	II HPH6	III HPH5
P_{ext}, MPa	5.82	2.87	2.87	1.82	1.20
P_{heat},MPa	5.42	2.84	2.76	1.73	1.13
h_{heat}, кJ/kg	2777.93	2673.79	2673.79	2606.07	2546.54
t_{heat},°C	261.61	231.41	231.41	207.72	187.96
$t_{\text{dr,}}$ °C	261.61	231.41	202.04	195.04	188.04
h_{dr},кJ/kg	1142.82	994.27	861.60	830.10	798.84

Extraction points	III D	III TFWP	IV LPH4	V LPH3	VI LPH2	VII LPH1
P_{ext}, MPa	1.20	1.03	0.58	0.31	0.08	0.021
P_{heat}, MPa	1.13	1.03	0.54	0.29	0.07	0.020
h_{heat}, кJ/kg	2546.54	2938.97	2827.55	2731.62	2547.65	2392.67
t_{in}, °C	187.96	250.00	157.65	137.05	93.49	61.12
$t_{\text{dr,}}$ °C	185.19	24.19	154.85	132.03	90.97	58.81
h_{dr}, кJ/kg	786.16	101.45	653.22	555.04	381.04	246.31

Steam, feed water, main condensate

Extraction points	0 R2	I R1	I HPH7	II HPH6	III HPH5
P_{in}, MPa	1.19	1.15	7.57	7.82	8.07
P_{out}, MPa	1.15	1.11	7.32	7.57	7.82
t_{kin}, °C	187.53	210.00	203.72	183.93	167.55
t_{kout}, °C	210.00	250.00	227.41	203.72	183.9
h_{in}, kJ/kg	2803.14	2844.07	869.20	780.73	708.47
h_{out}, kJ/kg	2844.07	2938.97	973.140	869,20	780.73
Extraction points	III D	IV TFWP	V LPH4	VI LPH3	VII LPH2
P_{in}, MPa	0.54	1.28	1.43	1.58	1.73
P_{out}, MPa	0.54	1.13	1.28	1.43	1.58
$t_{\text{c.in}}$, °C	154.65	134.05	90.49	59.12	32.46
$t_{\text{c.out}}$, °C	167.55	154.65	134.05	90.49	59.12
h_{in}, kJ/kg	652.34	563.69	379.01	247.46	118.10
h_{out}, kJ/kg	708.47	652.34	563.69	379.01	247.46

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According to the result of the solution of the system of equations of the material and energy balance of the heat exchange equipment of the turbo installation, the consumption of the working fluid in the selections of the turbine is determined (Table 2).

Table 2

Consumption of the working fluid at the points of the scheme of the turbine plant K-1000-60/1500-2

D_{FW}	D_{R1}	D_{R2}	D_{D}	D_{MSR}	D_{MS}	G_{MC}
1710,0	55.50	66.95	22.53	1298.4	138.26	1206.3
D_1	D_2	D_3	D_4	D_5	D_6	D_7
83.64	77.01	66.19	49.19	88.46	69.0	53.34

FW – feed water; R1, R2 – reheaters 1 and 2; MSR – moisture separator reheater; MS – moisture separator; MC – main condensate

G. Calculation of turbine capacity. Determining performance indicators of the turbo installation.

The consumption values of the main flows:

– steam consumption before a stop and a check valve: D=1603.3 kg/s;

– steam leakage through the sealing of turbine rods and valves: D_{Valv}=1.8 kg/s;

– steam consumption at the exit from the high-pressure cylinder: D_{MSR}=1242.62 kg/s;

– steam flow through the sealing of the high-pressure cylinder and the low-pressure cylinder: D_{seal}=2.4 kg/s;

Steam consumption for the drive turbine of the feed pump:

$$D_{\text{т fwp}} = D_{\text{fw}} \times \frac{\dfrac{\dfrac{v'(P_{\text{D}})\cdot DP_{\text{FP}}}{h_{\text{R2}} - h_k^{\text{TFWP}}}}{h^{\text{tfwp}}}}{h_{\text{м}}} == 1710 \times \frac{\dfrac{\dfrac{0.0011\cdot 7.86}{2938.97 - 2282.03}}{0.82}}{0.98} = 28.3 \text{ kg/s}.$$

The internal power of the turbine is determined in Table 3.

Table 3

Calculation of the internal work of the turbine

Compartment number	Compartment consumption, $D_{comp.i}$, kg/s	Enthalpy difference, $\Delta h_{comp.i}$, kJ/kg	$D_{comp.i}\times\Delta h_{comp.i}$, kW
1	1649.92	96.71	159563.76
2	1510.78	68.02	102763.25
3	1433.76	59.78	85710.17
4	1176.07	111.42	131037.72
5	1126.89	95.93	108102.56
6	1036.03	183.97	190598.44
7	967.02	154.98	149868.76
8	913.67	132.92	121445.02
Sum, W			1049089.68

Electric power of the turbogenerator:

$$N_{\text{e.br}} = W \times \eta_{\text{mech}} \times \eta_{\text{gen}} = 1049 \times 0.988 \times 0.99 = 1026 \text{ MW},$$

where η_{mech}=0.988 – mechanical efficiency of the turbine;

η_{gen}=0.99 – generator efficiency.

4. Calculation of the K-1000-60/1500-2 turbo installation scheme in heating mode.

The purpose of this part is to determine the electric capacity of the turbo installation when operating in the heating mode. The heating installation is presented in Fig. 1.

Mains water consumption G_w=2455t/h=687.5 kg/c. With the accepted values of the hot (forward) t_h and cold (return) t_c water temperatures, the thermal power supplied to the consumer from the NPP is equal:

$$Q_{HI}=G_w c_p(t_h – t_c)=687.5\times4.19\times(150–70)=230.4 \text{ MW},$$

where c_p – thermal conductivity of water.

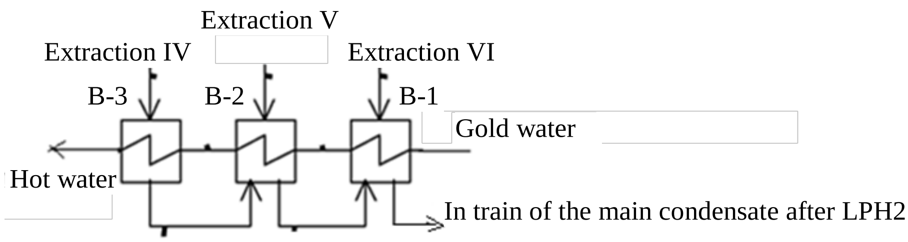

Fig. 1. Thermal scheme of the heating installation (HI)

From the equations of the material and energy balance of the boilers, we determine the steam consumption for them:

$$G_{B\text{-}3}=36.277 \text{ kg/c};\ G_{B\text{-}2}=47.23 \text{ kg/c};\ G_{B\text{-}1}=13.636 \text{ kg/c}.$$

Condensate consumption of heating steam from the heating installation:

$$G_{HI}=G_{B\text{-}3}+G_{B\text{-}2}+G_{B\text{-}1}=36.277+47.23+13.636 = 97.143 \text{ kg/s}.$$

As a result of the recalculation of the balances of LPH2 and LPH1, we will obtain steam consumption for them: D_6=62.45 kg/s; D_7=50.54 kg/s. The internal capacity of the turbine will be 1004.6 MW.

Electric power of the turbogenerator:

$$N_{e.br}=W\eta_{mech}\eta_{gen}=1004.6\times0.99\times0.988=982.6 \text{ MW}.$$

5. Use of a heat pump for heat supply from the NPP

When using a heat pump (HP), the heating installation with steam extraction does not work and the production of electricity increases, but the compressors of the heat pump consume electricity and therefore the supply to the consumer decreases. The purpose of this section is to calculate the power of HP compressors to determine the power released to the consumer during NPP operation with HP. In [1], the dependence of COP on the temperature of network water is given. The results of processing these data for a network water temperature of 65 °C are shown in the Table 4.

We will analyze the economic result of using a heat pump integrated in the condenser of the turbo installation. When releasing thermal energy in the amount of Q_{usful}=230.4 MW, the underproduction of electricity is: 1026.14–982.53=43.57 MW. During the operation of a heat pump installation of the same thermal power, the compressor power will be:

$$N_{compr}=Q_{usful}/\text{COP}=230.4/4.76=48.4 \text{ MW}.$$

Thus, during the operation of the NPP with heat pumps, the electricity output will decrease by 48.4–43.57=4.83 MW. Due to the small difference, this calls into question the conclusion [5] about the thermodynamic inexpediency of using a heat pump under the accepted conditions.

Table 4

The effect of steam condensation temperature as a low-temperature heat source for vaporizing freon in the heat pump cycle on its efficiency

t_{cond}, °C	29	30	31	32	33	34
COP	4.5896	4.793	4.9915	5.1936	5.4248	5.6648

Let's consider how the final pressure of the turbine installation will affect the performance of the NPP+heat pump complex. As the steam condensation temperature in the condenser increases, the efficiency of the heat pump increases, i.e. the COP increases.

In the Table 5 shows the results of calculating the electric power of the NPP in condensation mode when using a heat pump depending on the final pressure of the turbine.

Table 5

The influence of the final pressure of the turbine on the released electric power at a thermal power of 230 MW

t_{cond}, °C	Nel of NPP, MW	COP	Ncompr, MW	Nel – Ncompr, MW
32.28	1017	4.993	46.14	970.85
34	1008.7	5.226	44.09	964.61
36.8	995.5	5.576	41.32	954.18
39.66	982	5.935	38.82	943.18

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From the given data, it can be concluded that with an increase in the condensation temperature, the power of the power unit decreases faster than the COP of the heat pump increases. The total electrical power output is also reduced. That is, the conclusion given in [5] is confirmed.

We will calculate the option if the consumer is not supplied with fixed thermal power, but with all the thermal power that is discharged into the environment. That is, the entire heat capacity of the condenser is used. In this case, the useful thermal power released to the user is:

$$Q_{\text{usful}}=Q_{\text{condenser}}+N_{\text{compr}}. \quad (2)$$

The calculation results are given in the Table. 6. From the analysis of the given data, it can be seen that as the condensation temperature increases, the thermal power of the condenser Q_{cond} increases and the electric power of the turbine N_{el} decreases. At the same time, the COP increases and, accordingly, the required power of the compressor decreases:

$$N_{\text{compr}}=\frac{Q_{\text{condenser}}}{\text{COP}-1}. \quad (3)$$

Table 6

Results of calculating the influence of the final pressure
in the turbine condenser on the efficiency of NPP with HP

t_{cond}, °C	Q_{cond}, МВт	N_{el}, МВт	COP	N_{compr}, МВт	N_{el}–N_{compr}, МВт	Q_{usful}, МВт	E_Q, МВт	η_{ex}, %
32.28	2004.2	1017	4.993	494.7	522.3	2470	387.7	30.19
34	2012.5	1008.7	5.226	469.5	539.2	2454	385.2	30.6
36.8	2025.7	995.5	5.576	436.6	558.9	2434.7	382	31.2
39.66	2039.2	982	5.935	407.7	574.3	2419	379.7	31.6

The electric power supplied to the consumer increases. The overall efficiency of the APEC is estimated by the exergetic efficiency [11], which also increases with the increase in the final temperature (Table 6).

It should also be noted that one of the disadvantages of thermal power plants and nuclear power plants is heat pollution, which is of significant importance due to the relatively low efficiency. Currently, heat pollution charges are not collected, but given the great attention paid to the environment in recent times, it must be emphasized that it is precisely through the use of heat pumps that this heat pollution can be completely avoided. At the same time, a powerful source of cheap thermal energy appears.

Conclusion

1. A mathematical model of the thermal circuit of the K-1000-5.8/25-2 NPP turbine plant was developed, which takes into account the dependence of the steam condensation temperature in the condenser on the temperature of the circulating water.

2. As a result of the calculation, it was found that the power of the NPP power unit at a circulating water temperature of 18 °C in the condensation mode is equal to 1026 MW. When operating in winter and the power of the heating plant is 230.4 MW, the electric power of the power unit decreases to 982.6 MW (by 43.4 MW).

3. It was determined that when switching to heat supply from a heat pump and using a condenser as a low-potential energy source for a heat pump, the compressor power under these conditions (thermal power 230.4 MW, t_{cond}=30.5 °C, network water temperature 65 °C) is equal to 48.4 MW. That is, when comparing the operation of a nuclear power plant with a heating plant, which is provided by steam extracted from a turbine, the use of a heat pump leads to a decrease in electric power by 4.83 MW. The small value of this difference is explained by the relatively low value of the network water temperature (65 °C).

4. The influence of the final pressure on the efficiency of the TPP with a heat pump is analyzed. For the first time, it was found that with an increase in the final pressure, the increase in the efficiency of the heat pump precedes the decrease in the efficiency of the turbo installation, and the overall efficiency of the complex increases.

5. Considering that one of the disadvantages of TPPs and NPPs is heat pollution of the environment, it should be emphasized that the use of heat pumps, for which the turbine condenser will be a low-temperature source, allows you to completely avoid this pollution during winter operation.

Література

1. Increasing the Efficiency of NPP by Using the Heat Pump for Heat Supply / Kravchenko V., Kolykhanov M., Muromsky E., Vysotsky Yu., Galatsan M., Overchenko A. *Proceedings of the 13th International Conference of the Croatian Nuclear Society*. Zadar, Croatia, June 5–8, 2022. P. 120-1–120-10. URL: https://nuclear-option.org/wp-content/uploads/2022/11/S3-120.pdf.
2. Milana Treshcheva, Irina Anikina, Dmitry Treshchev, Sergey Skulkin. Heat Pump Capacity Selection for TPPs with Various Efficiency Levels. *Energies*. 2022. *15*(12), 4445. DOI: https://doi.org/10.3390/en15124445.
3. Ефимов Н.Н., Папин В.В., Малышев П.А., Безуглов Р.В. Анализ использования тепловых насосов на тепловых и атомных электростанциях. *Технические науки*. 2010. № 4. С. 35–39.
4. Чиркин Н. Б., Кузнецов М. А., Шерстов М. А., Стенников В. Н. Потенциальная возможность и техническая рациональность применения теплонасосных технологий при комбинированном производстве электрической и тепловой энергии. *Проблемы машиностроения*. 2014. Т. 17, № 1. С. 11–20.
5. Щеклеин С.Е., Ташлыков О.Л., Дубинин А.М. Повышение энергоэффективности АЭС. *Ядерная энергетика*. 2015. №4. С. 15–25.
6. Радечко Е.Н. Возможности использования турбонасосной установки путем утилизации низкопотенциального промышленного тепла турбин. Актуальные проблемы энергетики : *материалы 72-й научно-технической конференции студентов и аспирантов*. Минск : БНТУ, 2016. С. 530–533. URL: https://rep.bntu.by/handle/data/29235.
7. OCHSNER energy technology! Heat pumps for high outputs. *OCHSNER HEAT PUMPS*. URL: https://www.ochsner.com/en/ochsner-products/high-capacity-heat-pumps/.
8. Xu Z.Y., Mao H.C., Liu D.S., Wang R.Z. Waste heat recovery of power plant with large scale serial absorption heat pumps. *Energy*. 2018. Vol. 165, Part B. P. 1097–1105. DOI: https://doi.org/10.1016/j.energy.2018.10.052.
9. Паровые и газовые турбины для электростанций / А.Г. Костюк, В.В. Фролов, А.Е. Булкин и др. М. : Издательский дом МЭИ, 2008. 556 с.
10. Кіров В.С. Теплові схеми турбоустановок АЕС і їх розрахунок. Київ : Астропрінт, 2004. 212 с.
11. Дубковський В.О. Термодинамічний аналіз та оптимізація АЕС. Одеса : Наука і техніка, 2004. 48 с.

References

1. Kravchenko, V., Kolykhanov, M., Muromsky, E., Vysotsky, Yu., Galatsan, M., & Overchenko, A. (2022). Increasing the Efficiency of NPP by Using the Heat Pump for Heat Supply. *Proceedings of the 13th International Conference of the Croatian Nuclear Society*. Zadar, Croatia, June 5–8, 120-1–120-10. Retrieved from: https://nuclear-option.org/wp-content/uploads/2022/11/S3-120.pdf.
2. Milana Treshcheva, Irina Anikina, Dmitry Treshchev, & Sergey Skulkin. (2022). Heat Pump Capacity Selection for TPPs with Various Efficiency Levels. *Energies, 15*(12), 4445. DOI: https://doi.org/10.3390/en15124445.
3. Efimov, N.N., Papin, V.V., Malyshev, P.A., & Bezuglov, R.V. (2010). Analysis of the use of heat pumps in thermal and nuclear power plants. *Technical science*, *4*, 35–39.
4. Chirkin, N.B., Kuznetsov, M.A., Sherstov, M.A., & Stennikov, V.N. (2014). Potential possibility and technical rationality of using heat pump technologies in the combined production of electrical and thermal energy. *Problems of mechanical engineering*, *17*, 1, 11–20.
5. Shcheklein, S.E., Tashlykov, O.L., & Dubinin, A.M. (2015). Increasing the energy efficiency of nuclear power plants. *Nuclear energy, 4*, 15–25.
6. Radechko, E.N. (2016). Possibility of using a turbopump unit by utilizing low-grade industrial heat from turbines. Current problems of energy: materials of the 72nd scientific and technical conference of undergraduate and graduate students. Minsk: BNTU, pp. 530–533. Retrieved from: https://rep.bntu.by/handle/data/29235.
7. OCHSNER energy technology! Heat pumps for high outputs. *OCHSNER HEAT PUMPS*. Retrieved from: https://www.ochsner.com/en/ochsner-products/high-capacity-heat-pumps/.
8. Xu, Z.Y.; Mao, H.C.; Liu, D.S.; & Wang, R.Z. (2018). Waste heat recovery of power plant with large scale serial absorption heat pumps. *Energy, 165,* Part B, 1097–1105. DOI: https://doi.org/10.1016/j.energy.2018.10.052.

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

9. Kostyuk, A.G., Frolov, V.V., & Bulkin, A.E. et al. (2008). *Steam and gas turbines for power plants*. Moskow: MPEI Publishing House.
10. Kirov, V.S. (2004). *Thermal diagrams of AES turbine units and their design*. Kiev: Astroprint.
11. Dubkovsky, V.O. (2004). *Thermodynamic analysis and optimization of AES.* Odesa: Science and Technology.

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