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SHUBENKO, O. L.[1] (https://orcid.org/0000-0001-9014-1357),
BABAK, M. Yu.[1] (https://orcid.org/0000-0001-8146-2562),
and SENETSKYI, O. V.[1, 2] (https://orcid.org/0000-0002-2355-6664)

[1] A. Podgorny Institute of Mechanical Engineering Problems
of the National Academy of Sciences of Ukraine,
2/10, Pozharskogo St., Kharkiv, Ukraine, 61046,
+380 57 294 5514, admi@ipmach.kharkov.ua
[2] O. M. Beketov National University of Urban Economy in Kharkiv,
17, Marshal Bazhanov St., Kharkiv, 61002, Ukraine,
+380 57 707 3109, office@kname.edu.ua

APPROXIMATE MATHEMATICAL MODEL OF AN ABSORPTION HEAT PUMP WITH STEAM HEATING FOR INTEGRATION IN THE STEAM TURBINE THERMAL SCHEME

Introduction. *The results of theoretical and experimental studies presented in the literature have shown that CHPPs have a significant potential for energy savings when operating on a heat load by improving thermal schemes and operating characteristics. Solving the problem of improving the power plant turbine generator thermal scheme by implementing an absorption heat pump (AHP) improves the efficiency of the use of fuel and energy resources in the heat and electricity production.*

Problem Statement. *An analysis of literary sources has shown that in recent years, more and more attention has been paid to utilizing secondary sources of powerful power units operating in cogeneration mode with a significant supply of thermal energy to consumers. The presence of waste non-utilizable heat leads to a decrease in the efficiency of the use of initial fuel resources. This negatively affects the cost of heat and electricity and has a negative impact on the environment.*

Purpose. *The purpose of this research is to develop a fairly simple approximate mathematical model of AHP with steam heating (conversion coefficient $\mu = 1.71$), which is based on the real thermal transformers characteristics and is applicable in solving problems of its integration and to study the level of changes in the material flows of a powerful steam turbine with an integrated AHP with steam heating during a heating season.*

Material and Methods. *A simple approximation mathematical model based on the real characteristics of thermotransformer has been proposed to determine the AHP characteristics. It can be used to solve the problems of its integration into the thermal schemes of CHPP cogeneration units. The considered algorithm serves as a basis for creating software modules for determining the characteristics of AHP.*

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Results. *An approximate mathematical model of AHP with steam heating for solving the problems of integrating a heat pump in the cogeneration plant thermal schemes on the basis of the interpolation dependences of pump characteristics and conservation equations has been proposed and designed.*

Conclusions. *The proposed approximation mathematical model of AHP makes it possible to evaluate the performance of a cogeneration plant for heating season when AHP of the corresponding thermal power is integrated into the thermal scheme of steam turbine under a significant heat supply load.*

Keywords: energy saving, absorption heat pump, thermal scheme, heat supply, and cogeneration.

The turbine-based power plant turbogenerator has several heat flows that are dissipated in the environment. These are the flows of flue gases and cooling water from the condenser (steam turbine), the generator, and the lubrication system.

So for the steam turbine PT-60/70-130/13 [1], the flow rate of the condenser cooling water is ~6.3 Gcal/h (at a steam flow rate of ~12 t/h); the total capacity of the lubricant cooling systems and the generator at nominal load is ~0.47 MW. That is, in the cooling tower, when the systems of this turbine cool down, a heat of ~6.7 Gcal/h is released into the atmosphere.

In the situation of growing demand and prices for energy carriers, given the improving quality of heat pump technology, the heat of the turbogenerator cooling streams is utilized by integrating the lithium bromide absorption heat pump (AHP) into the thermal scheme (TS), for example, in Lithuania [2—5], which provides benefits through fuel savings, process water and environmental improvements.

The energy and environmental efficiency of recycling AHP is quite high, therefore, in China, the construction of new thermal power plants without these machines is prohibited at the legislative level [3].

Many researchers, including the Ukrainian ones [12–14, have been paying much attention to the study of energy saving when integrating AHP with steam heating in TS steam turbines: [6—12]. Both in our works [13], and in many others, the researchers, while considering the integration of AHP into the structure of turbine plants TS, have understudied the problems of simplified, but weighted, modeling of pump characteristics, especially given the need to solve optimization problems for choosing: pump power and operating modes.

General characteristics of AHP. In the simplest case, a steam heated AHP with single-stage regeneration (or a step-down thermal transformer) is a combination of four heat exchangers placed in one integrated body.

The two heat exchangers (generator and condenser) operate at a higher pressure, their purpose is to simply boil a liquid (water); the two other heat exchangers (evaporator and absorber) work at a reduced pressure, remove thermal energy from the source and convert the resulting vapor into a component of liquid solution [15—17].

The efficiency of AHP is estimated by the coefficient of performance μ (*COP*) determined as

$$\mu = Q_{AHP} / Q_h,$$

where Q_{AHP} and Q_h are the amount of heat supplied to the pump and the amount of heat that warms it.

The main advantages and disadvantages of AHP have been presented in [13].

AHP mathematical model based on the factory characteristics of pumps. Modeling of heat exchange processes in AHP is difficult, since the pump includes heat exchangers, where absorption-desorption processes take place. It requires knowledge of the characteristics of each machine element, thermodynamic properties of heat carriers, and their flow rates [17].

AHP works with three flowss of energy carriers at power plant with steam heating of the pump:

- water vapor that heats the pump (taken from the turbine outtake), with the initial parameters: pressure P_{h1} 0.14—0.6 MPa, temperature t_{h1} 130—165 °C;
- water with an initial temperature t_{s1} +7—35 °C, the heat of which is utilized (water of the three

cooling systems: condenser (CCS), lubrication (LCS), and generator (GCS));

- water with an initial temperature t_{w1} which is heated (return network water (RNW), feed water of the turbine unit and heating system), the temperature of which at the pump outlet t_{w2} does not exceed 90 °C.

For simulating the AHP characteristics, the following data have been used: the pumps performance curves of Broad Corporation, China (Fig. 1), the nomograms of SKB Teplosibmash, RF (Fig. 2), and the general characteristics of the pumps.

For each pressure P_{h1i} = 0.1, 0.2, 0.3, 0.4, 0.5 and 0.6 MPa (Fig. 1) of vapor that heats the pump, for known temperature of cooled circulating water (CW) at the pump outlet t_{s2j} (15; 20; 25; 30 and 35 °C) and temperature of RNW that heats the pump inlet t_{w1k} (40; 50; 60 and 70 °C), the values of $t_{w2i,j,k}$ are determined. The results underlie the interpolation algorithm that implements the dependence

$$t_{w2} = F_b\,(P_{h1}, t_{s2}, t_{w1}). \qquad (1)$$

The dependences on nomograms (Fig. 2) are linear, therefore, to approximate 4 lines on the right side and 5 ones on the left side, it is enough to specify the coordinates of two points.

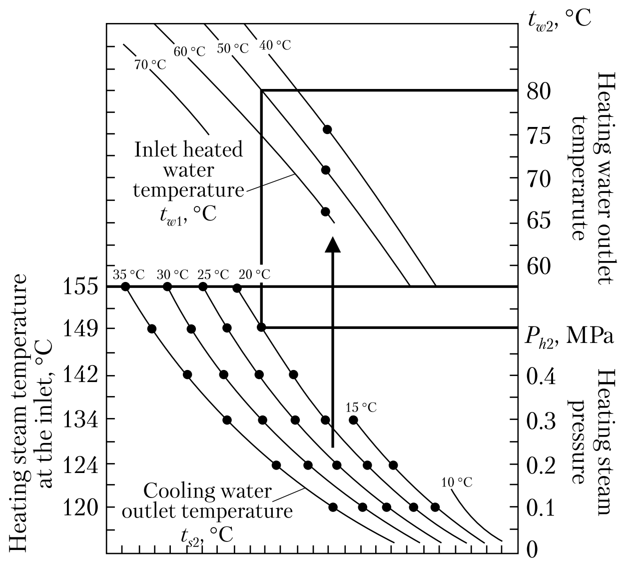

Fig. 1. AHP Performance curves Broad Corporation showing the parameters of the pump operation [3]: —— — AHP is heated by steam with standard parameters: 0.5 MPa, 149 °C

In accordance with the vertical breakdown, let us take the value of auxiliary function Y from 1 to 13; for the right nomogram the coordinates of 4

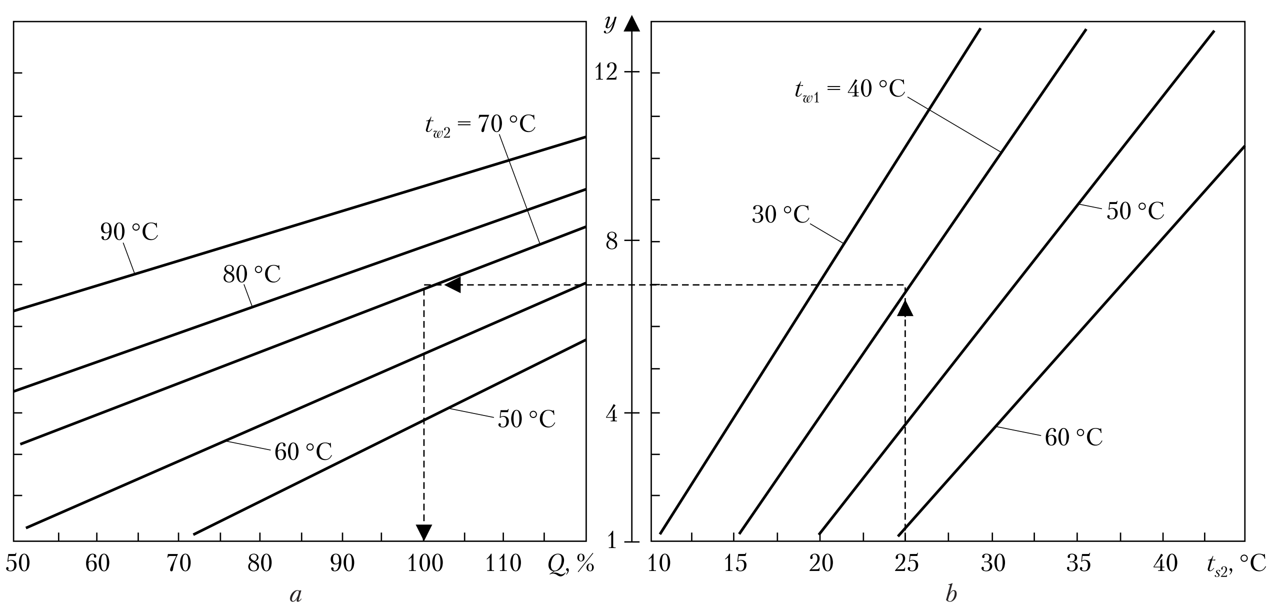

Fig. 2. Change in AHP relative thermal performance $\bar{Q}$ (a) and the temperature of the cooled water after the pump t_{s2} (b) depending on the temperature of the heated water: t_{w1} — at the inlet; t_{w2} — at the outlet of the pump [17]

base points on the line $y = 1$ at the points of intersection with the lines t_{s21}, t_{s22}, t_{s23}, and t_{s24} on the line $y = 1$ at the points of intersection with the lines let us take: $t_{w1} = 30$ °C, $t_{w1} = 40$ °C, $t_{w1} = 50$ °C, $t_{w1} = 60$ °C. As the further 4 basic straight points of this nomogram, we take the coordinates of the points y_1, y_2, y_3, and y_4 of the section of the straight line $t_{s1} = 30$ °C (the vertical bold line in Fig. 2, *b*) with the straight lines: $t_{w1} = 30$ °C, $t_{w1} = 40$ °C, $t_{w1} = 50$ °C, and $t_{w1} = 60$ °C.

For the left nomogram, there are coordinates of 5 base points of $\bar{Q} = 70$ % vertical line section (the bold left line in Fig. 2, *a*) with the lines: $t_{w2} = 90$ °C, $t_{w2} = 80$ °C, $t_{w2} = 70$ °C, $t_{w2} = 60$ °C, and $t_{w2} = 50$ °C. As the next 5 base points of intersection of this nomogram, let us take the coordinates of the points of intersection of line $\bar{Q} = 120\%$ (the right bold vertical line in Fig. 2, *a*) with the lines: $t_{w2} = 90$ °C, $t_{w2} = 80$ °C, $t_{w2} = 70$ °C, $t_{w2} = 60$ °C, and $t_{w2} = 50$ °C.

These base points values underlie the interpolation algorithm that implements the dependence

$$\bar{Q} = F_t\,(t_{s2}, t_{w1}, t_{w2}). \qquad (2)$$

For interpolation by base points, when determining t_{w2} and $\bar{Q}$, we use the one-dimensional cubic spline interpolation [18].

In addition, using data on the characteristics of AHP Broad Corporation with $\mu_n = 1.71$ (Table 1), we have obtain the approximation expressions depending on the normal pump thermal power $Q_{\text{n AHP}}$, in kW:

- for determining the pressure loss of the pump coolants in MPa: cooled ΔP_s and heated ΔP_w

$$\Delta P_s = a_{ps} \cdot Q_{\text{n AHP}} + b_{ps},$$
$$\Delta P_w = a_{pw} \cdot Q_{\text{n AHP}} + b_{pw}, \qquad (3)$$

where a_{ps}, b_{ps}, a_{pw}, b_{pw} are the approximation coefficients;

- for determining the normal flow rates of heat carriers AHP, in kg/h: $G_{h\,\text{n}}$, $G_{\text{s n}}$, $G_{w\,\text{n}}$ (heating, cooled and heated); and

$$G_{h\,\text{n}} = 0.8963 \cdot Q_{\text{n AHP}},$$
$$G_{\text{s n}} = 0.070811 \cdot Q_{\text{n AHP}},$$
$$G_{w\,\text{n}} = 0.02855 \cdot Q_{\text{n AHP}}; \qquad (4)$$

- for determining the electric power consumption by AHP $N_{e\,\text{AHP}}$, in kW

$$N_{e\,\text{AHP}} = 0.00225 \cdot Q_{\text{n AHP}}. \qquad (5)$$

The nominal parameters of heat carriers are as follows:

- the water temperature, inlet/outlet: cooled 30/25 °C ($\Delta ts = 5$ °C), heated 50/80 °C;
- the pressure of high-potential heating steam 0.5 MPa abs. (upper limit 110%).

Using interpolation dependencies (1), (2), data from Table 1 and the conservation equations, we have built an algorithm and software tools for determining the pump characteristics.

The initial data are the following characteristics of the pump:

- $\mu_n = 1.71$ is the rated transformation ratio;
- $Q_{\text{n AHP}}$ is nominal thermal power;

Table 1. **Characteristics of Powerful AHP with Steam Heating According to a Single-Stage Regeneration Scheme of the Chinese Broad Corporation, BDS [16] ($\Delta t_w = t_{w2} - t_{w1} = 30$ °C)**

Heat capacity $Q_{\text{n AHP}} / Q_s$, kW	CW		RNW		Costs	
	G_s, m³/h	ΔP_s, kPa	G_w, m³/h	ΔP_w, kPa	G_h, kg/h	*Ne* AHP, kW
16947 / 6980	15188	58	1200	83	486	38.8
22595 / 9304	20240	58	1600	83	648	50.4
28244 / 11630	25315	58	2000	83	810	52.4
33893 / 13956	30374	60	2400	85	971	75.6
45191 / 18608	40481	60	3200	85	1295	100.8
56489 / 23260	50631	60	4000	85	1619	104.8

- $\Delta t_s = t_{s1} - t_{s2} = 5$ °C;
- P_{h0}, t_{h0} is the heating steam pressure and temperature;
- the pressure and the temperature of water at the inlet to AHP: P_{s1}, t_{s1} — CW, P_{w1}, t_{w1} — RNW.

The sequence for calculating the characteristics of AHP is as follows.

To check if the input data belong to the permissible range of fluctuations (if not, then the limit values are accepted).

Based on known Δt_s, t_{s1}, $t_{s2} = t_{s1} - \Delta t_s$ is determined.

The steam pressure at AHP inlet is calculated as

$$P_{h1} = P_{h0} - \Delta P_h,$$

where $\Delta P_h = 0.05 P_{h0}$ is AHP heating steam pressure loss in the steam pipeline (similarly to the pressure loss in the steam pipelines before the turbine unit heaters, $t_{h1} = t_{h0}$).

Using the equations of steam and water state [19], we have calculated the specific enthalpies: $i_{h1} = f_{\mathrm{par}}(P_{h1}, t_{h1})$, $i_{s1} = f_{\mathrm{h2o}}(P_{s1}, t_{s1})$, $i_{w1} = f_{\mathrm{h2o}}(P_{w1}, t_{w1})$.

With the use of interpolation dependencies (1)—(4), t_{w2}, ΔP_s, ΔP_w, $G_{h\,\mathrm{n}}$, $G_{s\,\mathrm{n}}$, $G_{w\,\mathrm{n}}$, and $\bar{Q}$ have been determined.

If $\bar{Q}_T > 100$ %, it is necessary to reduce t_{s2} (Fig. 2) and to return to the beginning of the algorithm. Let us calculate $\mu = \mu_n \cdot \bar{Q}$.

The amount of heat is determined as:

- Q_h that heats AHP remains unchanged (rated): $Q_h = Q_{h\,\mathrm{n}}$;
- Q_s that is removed from CW (cooling):

$$Q_s = Q_h \cdot (\mu - 1);$$

- $Q_{r\,\mathrm{AHP}}$ that is supplied (real thermal power of the pump at a given mode):

$$Q_{r\,\mathrm{AHP}} = Q_w = Q_h + Q_s.$$

The pressure of water cooled P_{s2} and heated P_{w2} at the exit from AHP is calculated as:

$$P_{s2} = P_{s1} - \Delta P_s = P_{w1} - \Delta P_w.$$

Using the equations of water state [19], we determine the specific enthalpies: $i_{s2} = f_{h2o}(P_{s2}, t_{s2})$, $i_{w2} = f_{\mathrm{h2o}}(P_{w2}, t_{w2})$.

The consumption is calculated as follows:

- cooling medium (CW):

$$G_s = Q_s / (i_{s2} - i_{s1});$$

- water heated in AHP:

$$G_w = Q_w / (i_{w2} - i_{w1}).$$

The heated steam leaves AHP in the form of condensate with pressure P_{h2}. This allows us to determine the temperature and specific enthalpy of the condensate [20]: $t_{h2} = f_{t\ x=1}(P_{h2})$, $i_{h2} = f_{i\ x=1}(P_{h2})$ from the equations of the state of steam and water on the saturation line (degree of dryness $x = 1$). This condensate is used to heat water from AHP.

With the use of approximation dependence (5), electric power $N_{e\,\mathrm{AHP}}$ consumed by AHP is determined. It should be noted that when constructing the mathematical model of AHP, we limit variation of of t_{w1} to 60 °C. Such a temperature of RNW t_{RNW} with a heat supply schedule of 150/70 °C [20] is reached at frost of t_{oa} ~13.5 °C. This does not interfere with calculations for the integration of AHP into the turbine TS based on average monthly outdoor air temperature. The corresponding technical problem of AHP operation may be partially solved, for example, by cooling RNW at the pump inlet through heating the feed water.

The considered algorithm serves as the basis for creating software modules (implemented in Fortran G95) to determine the AHP characteristics.

The choice of the study object for integration of AHP into the turbine TS. The analysis of the problem has shown that powerful “T” and “PT” turbines operating at a large heat load (as soon as at a frost of a few degrees, it is necessary to turn on the hot water boiler, with the steam turbine operating at a fixed electric power, which is favorable for the pump) are the most promising for the introduction of AHP into the steam turbine TS [13].

Rather powerful steam turbines PT-60/70-130/13 and T-110/120-130 have been widespread in Ukraine: there are three former ones, seven fairly similar PT-60-90/13 turbines, and eight latter ones).

In this research, PT-60/70-130/13 (PT-60) turbine, the most widely used among steam turbines

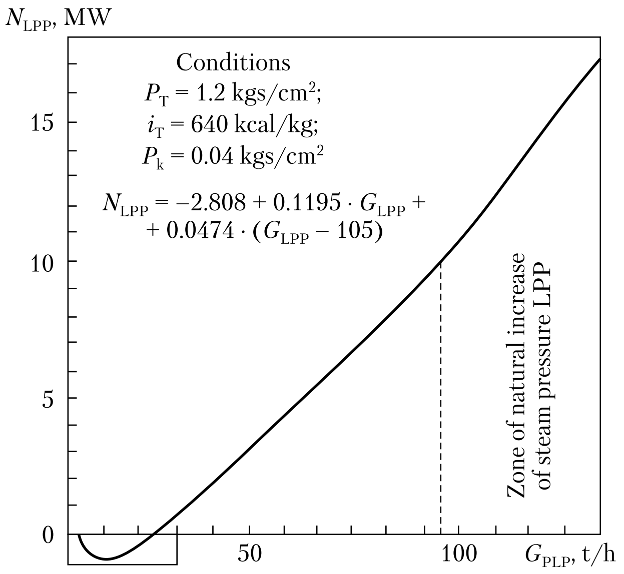

Fig. 3. The internal capacity of PT-60 LPP depending on the steam consumption according to factory data [22]

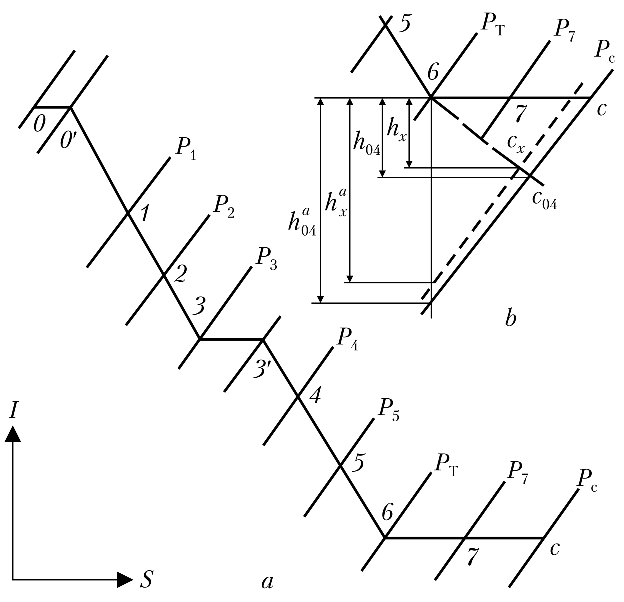

Fig. 4. The steam expansion process in PT-60 turbine in the *IS* diagram: P_1, P_2, ... — isobars of steam pressure in turbine outtakes; *0* — point of initial steam parameters; *1*, *2*, ..., *7* — steam outtakes; *3'* — production outtake chamber; *c* — condenser

manufactured in the Soviet era (more than 100 units) has been chosen as such an object.

The steam turbine PT-60 with a condensing unit and two controlled steam outtakes is a two-cylinder single-shaft unit (high + low pressure cylinders, the latter includes the two parts: the middle- and the low-pressure ones, respectively, MPP and LPP) with the following main characteristics [21]:

- the rated power of the turbine is 60 MW;
- the number of revolutions is 3000 rpm;
- the fresh steam parameters in front of the stop valve are as follows: the pressure is 12.75 MPa (130 kg · s/cm^2) and the temperature is 565 °C.
- the pressure in the condenser is 0.0034 MPa;
- the maximum parameters are as follows: the steam consumption through the turbine is 387 t/h, the rate of steam supply into the condenser is 160 t/h;
- the vapor pressure of controllable outtakes: the industrial one is 0.686—1.666 MPa and the cogeneration one is 0.0294—0.147 MPa.

The minimum steam supply rate in LPP (beyond the 27th stage) with a closed rotary control diaphragm (RCD), at a pressure in the outtake chamber of 0.0196 MPa (0.2 kg/cm^2) is 10 t/h with closed diaphragm. PT-60 in Ukraine has been produced for a long time, most of them do not have compacted RCD. PT-60 has 7 steam outtakes, two of which are controllable, 3 are the high pressure heaters (HPH) and 4 of them are the low pressure heaters (LPH). The feed water deaerator is connected to the same outtake as the lower HPH. The rated consumption of CW is 8000 m^3/h.

Modeling of PT-60 turbine TS. The most common method for modeling the steam turbine TS is the energy method [19] and we have used it in our research.

The features of the mathematical model of PT-60 TS, associated with the desire to bring the results of calculations closer to the real ones, include the use of the factory approximation $N_{LPP} = F_N(G_{LPP})$ to determine N_{LPP} that is the power of the LPP depending on the steam flow rate at the inlet G_{LPP} (see the approximation formula in Fig. 3).

When integrating AHP into the turbine TS, it is usually necessary to increase the steam pressure in the condenser P_c as compared with the rated 0.04 atm, for which the approximation dependence is determined in Fig. 3. This leads to the need of adjusting N_{LPP} depending on P_c.

The process of steam expansion in the flow path of PT-60 turbine in the *IS* diagram with the RCD closed is shown with the solid lines in Fig. 4.

Between points 6 (controllable heat outtake) and c (condenser), steam is throttled (i = const) in a closed RCD (the steam flow consumption in the condenser with an unsealed RCD, at which there are no ventilation losses, is $G_{\text{c min}}$ = 24 t/h, Fig. 3). When AHP is integrated, in many cases of P_c, the steam flow to the condenser increases, and a heat drop is used in the LPP (such a case in Fig. 4, *b* is shown by the dashed lines, index 04 corresponds to the state according to the factory data).

From the similarity of triangles in Fig. 4, *b* (the isobars are considered parallel), we obtain ratio $h_x/h_{04} = h_x^a/h_{04}^a = k_p$ that defines multiplier to N_{LPP} that takes into account the variation of P_c.

The considered particular qualities of PT-60 simulation are reflected in the software package based on the calculation of the steam turbine TS, which has been developed at A. Podhornyi Institute of Mechanical Engineering Problems of the NAS of Ukraine.

Choosing the modes for determining heating loads. To determine the performance of PT-60 during the heating period (November—March), the corresponding average monthly temperature has been chosen as outdoor air temperature t_{oa}, average for Ukrainian cities, based on DSTU for climatology. We assume that during the transitional periods (half of October and half of April), t_{oa} = +5 °C.

The thermal load of the turbine PT-60 at an outtake rate of 80 t/h steam and a flow rate of RNW G_{RNW} = 1500 t/h have been chosen based on the prospect of introducing AHP.

Table 2 shows the calculated indicators of these certain basic modes of PT-60 turbogenerator without AHP integrated into TS when operating at a heating load, which are important for comparing the characteristics of a turbine plant with integrated AHP.

The following initial conditions have been assumed:

- the fuel is natural gas (calorific value ~8350 kcal/m^3 at a density of ~0.7 kg/m^3);
- the efficiency of the boiler plant as well as the peak hot water boiler (PHWB) is 90%;
- the relative effective efficiency of the flow path: HPC — 80%, MPC — 82%, and LPC — 55.5%;
- the nominal steam parameters at the turbine inlet: the pressure is 12.75 MPa, the temperature is 555 °C;
- the heat supply schedule is 150/70 °C;
- the steam consumption at the outtake is unchanged and amounts to 80 t/h at the following parameters: 1.296 MPa, 280 °C, and a condensate return of 75% at a temperature of 40 °C;
- the maximum throughput of steam outtake is 150 t/h (for production, at HPH-3 and feed water deaerator) [21];
- the steam parameters for the heat outtake are unchanged (RCD is closed): the pressure is 0.1869 MPa, the temperature is 118.1 °C;
- the maximum throughput capacity of heating steam outtake is 150 t/h (the boiler, LPH-2, the atmospheric and vacuum deaerators (VD)) [22];
- the feed of network water is 0.2% at a temperature of 20 °C;
- the steam parameters in the turbine condenser are as follows: the pressure is 0.00334 MPa, the temperature is 25.9 °C, and the minimum flow rate is 26 t/h (RCD is assumed to be unsealed);
- since the steam flow to the condenser is low, LPH-1 is disabled, the same is true for recirculation to the condensate collector;
- the electrical power for the system own needs consists of the capacity of the condensate, the drainage, the feed, and the circulation pumps;
- the circulating system with a constant steam flow to the condenser of 26 t/h operates at a water flow rate of 1300 t/h and a technical water feed of ~26 t/h; the electric power of the circulation pump is ~83 kW at a head of 20 m water column.

The pressure of steam outtake in PT-60 is close to the nominal values [21].

When calculating each mode of TS without AHP, we choose the steam flow rate at the turbine inlet G_t, which provides the specified flow rate and steam parameters in the production outtake, the

Table 2. **Characteristics of PT-60/70-130/13 TS at Average Monthly Load Conditions without AHP for Heat Supply and Outtake, at a Steam Outtake Rate of 80 t/h**

Characteristic	Month					
	I	II	III	IV, X	XI	XII
Average monthly temperature t_{oa}, °C	–5.4	–4.5	+0.9	+5	+1.5	–
Time of standing of this temperature, h	744	672	744	732	720	744
Boiler feed water:						
temperature, °C	250.4	250.4	248.2	239.2	247.7	250.4
flow rate, t/h	358.6	358.6	339.0	269.34	334.57	358.6
Steam flow rate for:						
production outtake of steam, t/h, where:	98.82	98.82	97.92	94.63	97.71	98.82
HPH-3 for regeneration, t/h	15.35	15.35	14.60	11.87	14.43	15.35
LPH-4 for regeneration, t/h	13.41	13.41	13.02	10.57	12.87	13.41
LPH-3 for regeneration, t/h	8.79	8.79	8.61	7.45	8.54	8.79
heating outtake, t/h, from where:	148.31	148.31	135.72	90.12	132.85	148.31
LPH-2 — for regeneration, t/h (heat, Gcal/h)	10.91 (5.79)	10.91 (5.79)	10.82 (5.75)	10.52 (5.58)	10.80 (5.73)	10.91 (5.79)
boiler — t/h (heat input, Gcal/h)	128.45 (67.14)	128.45 (67.14)	115.95 (60.60)	70.66 (36.93)	113.10 (59.14)	128.45 (67.14)
Heat input to PHWB, Gcal/h	12.04	6.93	0.0	0.0	0.0	3.17
Flow rate:						
Vapors to the condenser, t/h			26.0	(pressure 0.00334 MPa)		
CW of make-up during the standing temperature, th. tons	19.34	17.47	19.34	19.03	18.72	19.34
Water flow rate for feeding the turbine unit:						
hourly, t/h	35.10	35.10	34.68	32.0	34.5	35.10
for the time of standing temperature, th. tons	26.01	23.59	25.80	23.42	24.84	26.01
Network water:						
flow rate (feeding), m^3/h			1500	(30)		
RNW temperature, °C	52.06	51.15	45.65	52.24	45.03	49.62
DNW temperature, °C	103.5	101.13	86.87	78.0	85.28	97.17
Electric power:						
own needs, MW	1.613	1.613	1.530	1.235	1510	1.613
"useful", MW	61.570	61.570	58.297	46.448	57.550	61.570
"Useful" electric energy during the temperature standing time, GWh	45.81	41.38	43.37	34.00	41.44	45.81
Electric efficiency, %	0.27214	0.27214	0.27132	0.27177	0.27118	0.27214
Heat released in total, Gcal/h	76.33	74.07	60.60	36.93	59.11	70.31
Hourly fuel consumption						
boiler, t.r.f./h	31.68	31.68	30.09	24.33	29.72	31.68
PHWB, t.r.f./h	1.46	1.10	0	0	0	0.50
total, t.r.f./h	33.14	32.78	30.09	24.33	29.72	32.18
Consumption of reference fuel during the time of standing temperature, th. t.r.f.	24.66	22.03	22.39	17.81	21.39	23.04

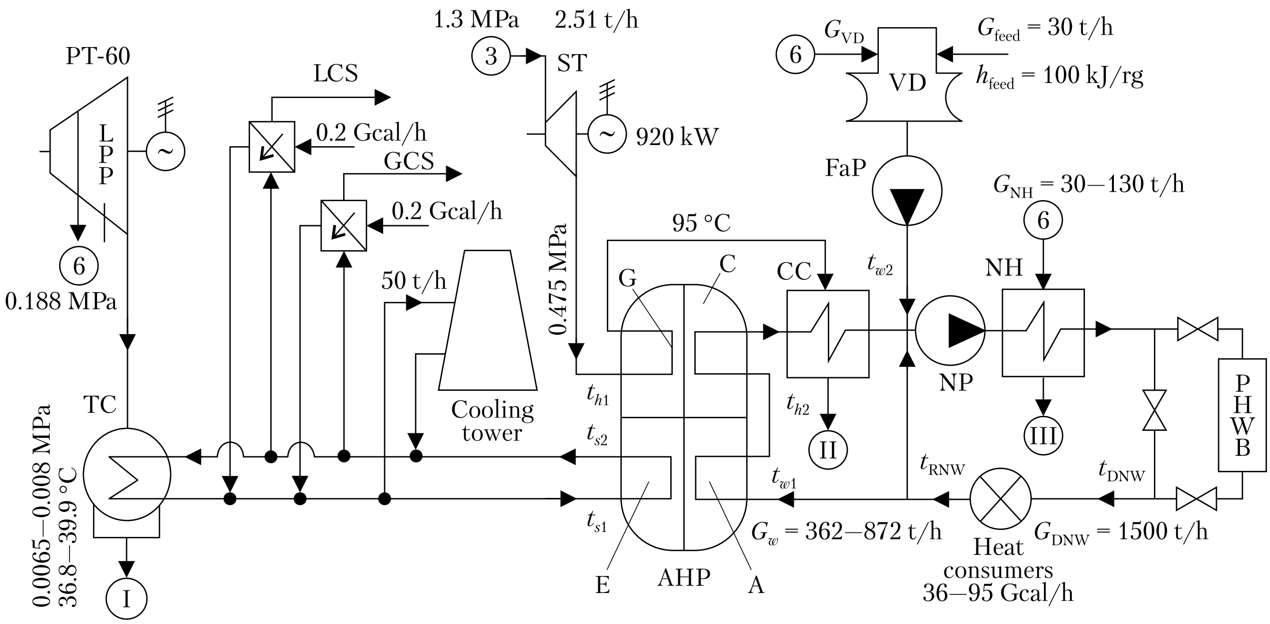

Fig. 5. Scheme of PT-60 turbine condenser cooling system when integrating AHP with 24 MW thermal power: AHP: A — absorber; G – generator; E — evaporator; C — condenser; CC — condensate cooler; VD — vacuum deaerator; TK — turbine condenser; ST — energy-saving low-power steam turbine with back pressure; pumps: FaP — for heating network feeding; NP — network; PHWB — peak hot water boiler; cooling systems: GCS — generator; LCS — lubricant; NH — network heater; LPP — low pressure part; controllable outtakes: 3 — industrial; 6 — heating; I, II, III — connections with elements of the thermal scheme PT-60

required temperature of the direct network water (DNW) and the known steam flow rate into the condenser G_c (takes the minimum allowable value of 26 t/h).

The highest fuel consumption (33.14 t.r.f./h) has been reported for the coldest month (January t_{oa} = –5.4 °C), while the lowest one has been recorded in the transition period.

As can be seen from the analysis of the data in Table 2, the resources consumed by PT-60 turbine unit during the heating season, which change with the integration of AHP have been determined: the electricity sold is 251.8 GWh, the fuel combusted is 131.32 thousand t.r.f., the technical water and softened water consumed for feeding is 113.24 thousand tons and 149.67 thousand tons, respectively.

The scheme of AHP integration into PT-60 TS, the feature of which is the presence of only one outtake of heating steam, is shown in Fig. 5 (see [21] for the basic turbine thermal scheme). As can be seen from this scheme, the steam for heating AHP is taken from the controllable outtake PT-60, operating at a pressure of 1.3 MPa. The pump is heated by steam at a pressure of 0.474 MPa, coming from the exhaust of an additional steam turbine with a back pressure of 1272 kW. It is installed for the purpose of energy saving (to utilize the excessive thermal difference).

The results of characteristics calculating of PT-60 TS with integrated AHP 24 MW at loads corresponding to the indicated average monthly outdoor air temperature at a heating steam pressure of the pump P_{h1} = 0.285 MPa (the steam pressure at the exhaust of an additional small steam turbine 0.3 MPa) are presented in Table 3.

When integrating AHP into steam turbine TS, it is necessary to decide. what heat output pump Q_{AHP} should be chosen, what steam parameters (pressure P_{h1}) should be used to heat it, what steam pressure in the condenser P_c should be chosen so that the CW is supplied to the pump at a required

Table 3. **Characteristics of PT-60 TS at Average Monthly Load Modes with Integrated AHP 24 MW at Heat Supply and Steam Outtake Rate of 80 t/h (the power of small steam turbine with back pressure is 1272 kW)**

Characteristic	Month					
	I	II	III	IV, X	XI	XII
Average monthly temperature t_{oa}, °C	–5.4	–4.5	+0.9	+5	+1.5	–
Turbine steam flow rate, t/h	359.3	353.8	316.92	238.14	312.87	343.6
Steam flow rate in turbine outtakess:						
production outtake of steam, t/h, where:	121.04	120.78	118.99	115.60	118.8	120.3
HPH-3 for regeneration, t/h	15.94	15.72	14.24	11.43	14.08	15.31
AHP, t/h				21.51		
LPH-4 for regeneration, t/h	14.23	14.03	12.70	10.18	12.55	13.66
LPH-3 for regeneration, t/h	12.56	12.86	13.65	10.58	13.65	13.23
heating outtake, t/h, from where:	125.80	121.27	95.32	50.14	92.45	114.0
LPH-2 for regeneration, t/h	12.77	12.97	13.65	12.36	13.72	13.19
boiler t/h (heat input, Gcal/h)	12.77	12.97	13.65	12.36	13.72	13.19
	12.77	12.97	13.65	12.36	13.72	13.19
VD NW, t/h	1.51	1.53	1.55	1.56	1.55	1.54
Absorption heat pump, Heating steam, inlet:	P_{h1} = 0.285 MPa, t_{h1} = 156 °C					
Absorption heat pump, Heating steam, outlet:	P_{h2} = 0.099 MPa, t_{h1} = 125 °C					
Absorption heat pump, Heating steam, heat for regeneration, Gcal/h	1.380	1.398	1.515	1.921	1.628	1.431
Absorption heat pump, Water that is cooled, inlet: P_{s1}=0.02 MPa, t_{s1}, °C	38.8	38.5	36.1	38.9	35.8	37.8
Absorption heat pump, Water that is cooled, outlet: P_{s2}=0.15 MPa, t_{s2}, °C	33.8	33.5	31.1	33.9	30.8	32.8
Absorption heat pump, Water that is cooled, flow rate G_s, t/h	1624	1645	1635	1636	1637	1640
Absorption heat pump, Water that is cooled, heat removed Q_s, MW	9.306	9.424	9.390	9.373	9.408	9.407
Absorption heat pump, heated water, inlet:	P_{w1} = 0.25 MPa, t_{w1} = t_{RNW}, °C					
Absorption heat pump, heated water, outlet: P_{w1} = 0.165 MPa, t_{w2}, °C	83.45	83.9	86.34	83.4	86.56	84.7
Absorption heat pump, heated water, flow rate, t/h	617	594	479	625	469	556
Absorption heat pump, Heat supply to the RNW, Gcal/h	19.136	19.237	19.207	19.194	19.223	19.22
Absorption heat pump, Pumps electric capacity, kW	49					
Absorption heat pump, Relative thermal power	0.994	0.999	0.998	0.997	0.998	0.998
Absorption heat pump, *COP*(μ)	1.700	1.708	1.706	1.705	1.707	1.707
Condenser:						
pressure P_c 10^3, MPa	7.24	7.12	6.26	7.27	6.17	6.88
flow rate, t/h	32.87	33.23	33.03	33.06	33.09	33.14
Network water	RNW and DNW flow rate (feeding) and temperature, see Table 2					
Flow rate:						
CW per cooling tower, t/h	50 (feeding 1)					
water for turbine unit feeding, t/h	36.03	35.81	34.35	31.606	34.19	35.40
Electric power:						
own needs, MW	1.659	1.636	1.474	1.170	1.456	1.594
"useful", MW	64.568	63.606	57.161	44.960	56.448	61.828
Electric efficiency, %	0.26923	0.26892	0.26690	0.26165	0.26670	0.2682
Heat supplied to NW, Gcal/h	76.330	74.07	60.60	36.93	59.115	70.30
Reference fuel consumption:						
boiler, t.r.f./h	32.922	32.459	29.327	23.395	28.976	31.602
PHWB, t.r.f./h	0					
total, t.r.f./h	32.922	32.459	29.327	23.395	28.976	31.602

temperature t_{s1}. Moreover, since it is assumed that AHP is heated by steam from the production outtake PT-60 with a low-power steam turbine with back pressure for energy saving (the pump is heated by its exhaust), the turbines can be supplied with different P_{h1} and t_{s1} in different modes.

Given the above, for the qualified solution of the problem of integrating AHP into the turbine TS, an optimization problem should be solved.

Firstly, for calculating each mode of PT-60 TS with AHP, the steam pressure in the condenser P_c, is determined. The steam pressure ensures the level μ ~1.65—1.71. Then, the steam flow rate at the turbine inlet G_t, which provides the specified production outtake (steam flow rate and parameters), the required temperature of DNW, and the steam flow rate to the condenser G_c are determined. The third is known, since it is based on the flow rate of cooled water in AHP, i.e. G_s, given the fact that 1 ton steam is left to keep the cooling tower in working order (the CW flow rate is 50 t/h), with the total capacity of lubricant and generator cooling systems at nominal load being ~0.47 MW.

As can be seen from Table 3, when integrating AHP (with 24 MW thermal power) into PT-60 TS we obtain as follows:

- in the pump, only a part of the RNW G_w (368—472 t/h) is heated, at G_{RNW} = 1500 t/h, so direct heating of feed water is impossible;
- up to the assumed outdoor air temperature of the coldest month (minus 5.4 °C), there is no need to use PHWB;
- the steam flow rate into the condenser G_c in all modes differs little, since G_s is determined by AHP in modes close to μ = 1.71.

The results of calculation of an improvement in the performance of PT-60 turbine unit after the integration of AHP into its TS are presented in Table 4. To determine the change in the amount of harmful emissions of CO_2 and NO_x into the atmosphere [22], it has been shown that when burning 1 ton natural gas, the following amount of harmful emissions is formed: H_{CO2} = 2.726 t/t, H_{NOx} = 0.0143 t/t.

Table 4. **Change in the PT-60 Indicators After AHP (a thermal power of 24 MW) Integration**

Characteristic	Month					
	I	II	III	IV, X	XI	XII
Average monthly temperature t_{oa}, ºC	–5.4	–4.5	+0.9	+5	+1.5	–3
Time of standing of this temperature, h	744	672	744	732	720	744
Hour difference:						
“useful” electric power, MW	2.998	2.036	–1.136	–1.488	–1.103	0.259
total reference fuel consumption, t.r.f./h	0.218	0.321	0.763	0.935	0.744	0.578
in natural gas consumption, t/h	0.128	0.188	0.447	0.548	0.436	0.339
feeding consumption:						
CW per cooling tower, t/h	25					
softened water, t/h	–0.93	–0.71	0.33	0.394	0.31	–0.3
in CO_2 emissions, t/h	0.348	0.513	1.219	1.494	1.189	0.924
in $NO_x \cdot 10^3$ emissions, t/h	1.83	2.69	6.40	7.84	6.24	4.84
Changes during standing time:						
electricity for sale, MWh	2.231	1.368	–0.845	–1.089	–0.794	0.193
total reference fuel consumption, th. t.r.f.	0.162	0.216	0.568	0.684	0.536	0.430
feeding consumption:						
CW per cooling tower, th. ton	18.6	16.8	18.6	18.3	18	18.6
softened water, th. ton	–0.692	–0.477	0.246	0.288	0.223	–0.223
in CO_2 emissions, t/h	259.2	344.7	907.1	1093.6	856.0	687.2
in $NO_x \cdot 10^3$ emissions, t/h	1.359	1.808	4.758	5.737	4.490	3.604

The results of the calculation of the change in the turbine unit parameters for AHP integrated into PT-60 TS are presented in Table 4.

The data in Table 4 show that when AHP is integrated, the consumption of fuel and technical water for feeding of the circulation system has a steady downward trend. The difference between the electric power generation with integrated AHP into PT-60 from the non-integrated case depends on the operating mode. The PT-60 modes without a heat pump, where PHWB works, have lesser generation. The PT-60 with an integrated AHP operates with a larger steam flow to the condenser as compared with the option without a heat pump, i.e. with steam consumption G_t, so it needs to be fed with more softened water. In modes with high t_{oa} (PHWB does not work), the increase in G_t with AHP integrated into PT-60 is not enough to compensate for the undergeneration due to heating steam outtake by AHP.

Based on the data of Table 4, the changes in the resources consumed during the heating season by conventional PT-60 turbine unit and integrated AHP (a thermal power of 4 MW) have been determined.

Conclusions. The approximation mathematical model of AHP with steam heating has been proposed for solving problems of pump integration. The mathematical model has been constructed with the use of interpolation dependences of pump characteristics and storage equations.

It has been used to assess the performance indicators of the steam turbine PT-60/70-130/13, in which TS an AHP with 24 MW thermal power is integrated and which has a significant heat supply load, for a heating season.

The obtained results of modeling the characteristics of AHP should be considered satisfactory, since they are built on the basis of the pumps characteristics from manufacturers.

According to the calculations results, the integration of AHP with a 24 MW thermal power into turbine PT-60/70-130/13 TS during a heating season leads to:

- an increase in the power generation by 0.4%;
- savings: fuel by 2%, technical water for feed to the circulation system by 96%;
- excessive consumption of softened water for feeding the turbine unit by 0.42%.

As a result of fuel savings, during heating season, the harmful emissions into the atmosphere decrease by 4148 tons for CO_2 and by 21.8 tons for NO_x, which, given savings of 108.9 thousand tons of water, is an important environmental effect from the integration of AHP.

The results obtained have indicated good prospects for energy saving with the use of AHP as part of TS on CHPP with powerful turbines, and the expediency of continuing research on this subject.

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О.Л. Шубенко (https://orcid.org/0000-0001-9014-1357),
М.Ю. Бабак (https://orcid.org/0000-0001-8146-2562),
О.В. Сенецький (https://orcid.org/0000-0001-8146-2562)

[1] Інститут проблем машинобудування ім. А.М. Підгорного
Національної академії наук України,
вул. Пожарського, 2/10, Харків, 61046, Україна,
+380 57 294 5514, admi@ipmach.kharkov.ua
[2] Харківський національний університет міського господарства імені О.М. Бекетова,
вул. Маршала Бажанова, 17, Харків, 61002, Україна,
+380 57 707 3109, office@kname.edu.ua

АПРОКСІМАЦІЙНА МАТЕМАТИЧНА МОДЕЛЬ АБСОРБЦІЙНОГО ТЕПЛОВОГО НАСОСУ З ПАРОВИМ ОБІГРІВОМ ДЛЯ ІНТЕГРАЦІЇ У ТЕПЛОВУ СХЕМУ ПАРОВОЇ ТУРБІНИ

Вступ. ТЕЦ мають значний потенціал впровадження енергозбереження при експлуатації на тепловому навантаженні шляхом вдосконалення теплових схем й режимних характеристик. Розв’язання задачі з вдосконалення теплової схеми турбогенератора електростанції шляхом реалізації абсорбційного теплового насосу (АБТН) дасть можливість підвищити ефективність використання паливно-енергетичних ресурсів (ПЕР) при виробництві тепло- та електроенергії.

Проблематика. Наразі значну увагу приділяють утилізації вторинних джерел енергії потужних енергоблоків, які працюють у когенераційному режимі. Наявність скидної теплоти, яка не утилізується, призводить до зниження ефективності використання ПЕР, що збільшує вартість тепло- та електроенергії й має негативний вплив на довкілля.

Мета. Розробка апроксимаційної математичної моделі АБТН з паровим обігрівом (μ = 1,71), що базується на характеристиках реальних термотрансформаторів, і може бути застосована при вирішенні задач з його інтеграції. Дослідити рівень змін матеріальних потоків потужної парової турбіни з інтегрованим АБТН з паровим обігрівом за опалювальний сезон.

Матеріали й методи. Використано методи математичного моделювання енергетичного устаткування з використанням інтерполяційних алгоритмів. Розглянутий алгоритм став базою для створення програмних модулів з визначення характеристик АБТН.

Результати. Розроблено апроксимаційну математичну модель АБТН з паровим обігрівом для вирішення завдань з інтеграції теплового насосу теплової схеми когенераційних установок. Її побудовано з використанням інтерполяційних залежностей характеристик насосів і рівнянь збереження.

Висновки. Запропонована модель АБТН дозволяє оцінити показники роботи когенераційної установки за опалювальний сезон при інтеграції до теплової схеми парової турбіни АБТН відповідної теплової потужності і яка має значне навантаження теплопостачання.

Ключові слова: енергозбереження, абсорбційний тепловий насос, теплова схема, теплопостачання, когенерація.