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CHOICE OF BASIC CONSTRUCTION PARAMETERS OF STEAM GENERATORS FOR NPP OF LOW POWER

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Designs and methods for calculating a steam generator (SG) for low power NPPs are considered. As a prototype adopted SG nuclear power plant type KLT-40C (RF). The general list of work in the design of SG is given. A methodology for the structural calculation of SG with a coil heating surface, as well as a mathematical model of SG, have been developed, including structural, thermal, hydraulic, strength and economic calculations. The derivation of formulas for determining the wall thickness of heat transfer tubes (HTT) and the number of concentric layers of coils is described. The methodology for determining the main dimensions of the vessel, shaft and flat cover. Using the developed computer program, the optimal diameter of the HTT, the distances between the heat transfer tubes in the layer and between the layers, the feedwater speed at the inlet to the heat transfer tubes were determined.

INTRODUCTION

The main task of the further development of nuclear energy – ensuring the safety of the population in any operation NPP including emergency situations, reliable management of spent fuel and radioactive waste. Therefore, the development of nuclear energy in the 21st century is associated with the development of a new generation of nuclear power plants (NPPs) [1].

One of such promising areas is low-power NPPs. [2–4]. First at all, the licensing, design, manufacture and installation of such NPP is a simpler task than for large NPPs. Secondly, the disposal of such an installation is greatly simplified, especially in underground locations. The disadvantage is the high unit cost. Such plant is not an alternative to other nuclear energy projects, and its specificity lies in technical, environmental and economic indicators that are not available for other projects not only today, but also in the near future.

Low power NPPs are now receiving worldwide attention. Projects from Russia, the USA, Japan and other countries are presented in the international market. Recently, the construction of the first floating nuclear power plant has been completed, which is one of the options for this type of nuclear power plant. Another example is the use of ship nuclear power plants of the KLT-40C type, KN-3, which can be used for integrated electricity and heat supply (both domestic and industrial) consumers in remote areas. The prospect of using low power NPPs is also interesting for Ukraine, which can establish their production, use and sale. Therefore, in this paper, attention is paid to the corresponding nuclear power plant, namely, one of its main elements – a steam generator (SG). For further consideration, a nuclear power plant with a thermal capacity of 180 MW was adopted, the prototype of which is taken by the KLT-40C nuclear power plant [5].

1. CONSTRUCTION DIAGRAM ONCE-THROUGH STEAM GENERATOR

This is a vertical recuperative heat exchanger with a coil heat transfer surface made of titanium alloys and forced circulation (Fig. 1). The coolant of the 1st circuit moves from top to bottom in the annulus. The working fluid moves inside the tubes from the bottom to up. In the lid of the steam generator (SG), vertical steam channels are made, combined into a common toroidal collector with a tube for the removal of superheated steam. In the central part of the lid there is a feed collector with a branch pipe for supplying feedwater. Steam collectors are welded to the bottom of the lid. Feedwater collectors are welded to the tube plate of the feed water chamber. The SG cover is attached to the housing flange by welding. On the feed collector of the lid, there are mounted fittings for air removal, drainage of the cavity of the 1st circuit and sampling from the cavity of the 2nd circuit.

To exclude the flow of the coolant of the 1st circuit from the pressure cavity into the drain (except for the tube system), on the outer shell of the tube system, a bellows and disk spring packages are installed in the upper part. In the event of a coil leakage, any of the supply tubes can be detected and plugged. Possible replacement of the entire tube system.

The purpose of this article is to develop a general design algorithm for a steam generator and to determine some of its features.

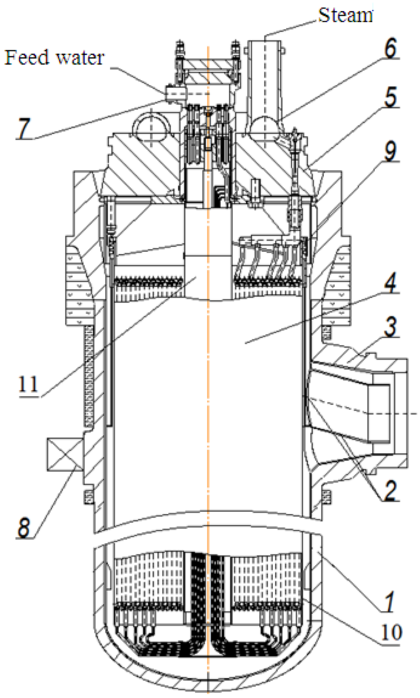

Fig. 1. Once-through steam generator:
1 – vessel; 2 – cabinet partition;
3 – pipe-to-pipe connection;
4 – coil heat transfer surface;
5 – flat cover; 6 – toroidal steam collector;
7 – branch pipe for feed water; 8 – support pin;
9 – bellows seals;
10 – shaft; 11 – central tube

2. GENERAL DESIGN ALGORITHM

Detailed calculation of SG, modeling of various load modes and other complex calculations are carried out for the selected SG design. Therefore, the first step is to choose the main design parameters: diameter of heat transfer tubes (HTT), the step between them and the feedwater speed at the inlet of the HTT. These parameters as a result of thermal calculation determine the dimensions of the heat exchange surface, on the basis of which, with the help of strength analysis, the wall thicknesses of the main structural elements and the dimensions of the entire SG are determined. For this design, the reduced costs are determined [6], which include the cost of the structure and hydraulic resistance on the sides of the coolant and the working fluid. Minimization of reduced costs determines optimal output parameters. The process of optimizing the main parameters requires the development of a mathematical model that includes structural, thermal, strength, hydraulic and economic calculations. The search for optimal parameters is carried out by enumerating all possible options.

After determining the design, a verification calculation is made, where the dimensions of the main structural elements based on modeling using computer codes are specified:

– determination of the thickness of the elements taking into account the cyclical loads during operation, hydraulic tests and stops;

– determination of hydraulic stability conditions;

– structural modeling and determination of current loads;

– development of working drawings;

– development of manufacturing technology, scope of production quality control;

– development of instructions for use, repair, maintenance.

Let us select some elements of the calculation.

3. DESIGN CALCULATION

The wall thickness of the tube is determined by the expression [7]:

$$s = s_{\mathrm{R}} + \mathrm{C} = \frac{\mathrm{P_p} \cdot D_0}{2\varphi[\sigma_{\mathrm{per}}] + \mathrm{P_p}} + \mathrm{C}, \qquad (1)$$

where $\mathrm{P_p}$– design pressure; D_0 – tube outer diameter; φ – attenuation coefficient, for tube φ = 1; $[\sigma_{\mathrm{per}}]$ – permissible load, defined as

$$[\sigma_{\mathrm{per}}] = \min\{\mathrm{R_m}/2{,}6;\ \mathrm{R_{0,2}}/1{,}5\}, \qquad (2)$$

$\mathrm{R_m}$, $\mathrm{R_{0,2}}$ – tensile strength and yield strength at design temperature; C – amendment, which is defined as

$$\mathrm{C} = \mathrm{C_1} + \mathrm{C_2} = \mathrm{C_{11}} + \mathrm{C_{12}} + \mathrm{C_2}, \qquad (3)$$

$\mathrm{C_{11}}$– equal to negative tolerance on wall thickness

$$\mathrm{C_{11}} = 0{,}15 \cdot \mathrm{s}, \qquad (4)$$

$\mathrm{C_{12}}$ – for bending, which are manufactured on pipe-lifting equipment by winding onto the sector, technological increase C_{12} accepted only for wall thickness S_{R1}. For pipes of heating surfaces:

$$\mathrm{C_{12}} = \left(\frac{1}{1 + 2{,}5 \cdot \frac{Rs}{D_0}}\right) \cdot s, \qquad (5)$$

Rs – bending radius; $\mathrm{C_2}$ – takes into account the corrosive effects of the working fluid on the material of structural elements in operating conditions. For zirconium alloys as well as austenitic grade stainless steel, it is recommended $\mathrm{C_2}$=0.1. HTT SG are made of titanium alloy. Given that the corrosion resistance of titanium alloys is even higher than that of stainless steel, we accept $\mathrm{C_2}$ = 0.1.

When $11 < D_0 < 30$ mm marginal deviation of wall thickness at normal manufacturing accuracy ±15%.

Ovality of pipes should not lead pipes with maximum deviations in pipe diameter +0.45 mm.

Substitute (3)–(5) in (1):

$$s = s_\mathrm{R} + 0{,}15 \cdot s + \left(\frac{1}{1+2{,}5\cdot\frac{Rs}{D_0}}\right) \cdot s + 0{,}1, \qquad (6)$$

As a result of solution (6) with respect to s, we obtain the expression

$$s = \frac{s_R + 0{,}1}{0{,}85 - \frac{1}{1+2{,}5\cdot\frac{Rs}{D_0}}}. \qquad (7)$$

Minimum winding diameter for HTT recommended (10…12)·D_0. If we take the minimum permissible winding radius Rs = 10·D_0/2 = 5·D_0, then (7) becomes:

$$s = \frac{s_R + 0{,}1}{0{,}85 - \frac{1}{1+2{,}5\cdot\frac{5D_0}{D_0}}} = \frac{s_R + 0{,}1}{0{,}776}. \qquad (8)$$

Using the formula (8), the HTT wall thickness was determined for the design conditions depending on the outer diameter for the ПТ-7М titanium alloy (P_p = 17.2 MPa, t_p = 350 °C, R_m = 250 MPa, $R_{0.2}$ = 180 MPa) (Table).

The results of calculating the wall thickness of the HTT with ПТ-7М for the design parameters of the SG

Parametr	External diameter HTT, mm					
	16	18	20	22	24	27
s_R, mm	1.4	1.6	1.7	1.9	2.1	2.3
Estimated s, mm	2.1	2.1	2.4	2.8	2.8	3.1
Accepted s, mm	2.2	2.2	2.5	2.8	3.0	3.2

Let us dwell on determining the number of concentric layers (rows) of HTT.

From the continuity equation, we find the total amount of HTT (feedwater speed at the inlet of HTT is taken from 0.5 m/s):

$$N_{htt} = G_{fw} \cdot v_{fw} / w_{fw} \, / \, f_p, \qquad (9)$$

where G_{fw},·v_{fw}, w_{fw} – flow rate, specific volume, feedwater speed; f_p – living section area of one HTT.

The average speed of the coolant in the pipe space is accepted (w_c = 1…3 m/s). Then the area for the passage of coolant:

$$F_c = G_c \cdot v_{c.cp} / w_c. \qquad (10)$$

The living cross-sectional area for the coolant passage for a given type of heat exchange surface is determined provided that the coil gives a ring with a horizontal cross-section. Then the minimum area for the passage of coolant is defined as the area of space between the shaft and the central pipe minus the area of all layers of the coils:

$$F_c = \frac{\pi}{4}\left(Д_{sh}^2 - Д_{ct}^2\right) - Д_{av} \cdot z \cdot d_p \, \pi, \qquad (11)$$

where $Д_{sh}$ – inner diameter of shaft, $Д_{ct}$ – outer diameter of the central tube.

Or as the sum of all the gaps between the coils (layers):

$$F_c = \pi \cdot Д_{avr} \cdot z \cdot (S - d_p). \qquad (12)$$

Given that the average diameter of the layers

$$Д_{avr} = (Д_1 + Дz)/2, \qquad (13)$$

$Д_1$, Дz – diameters of the first and last layers:

$$Д_1 = Д_{ct} + S, \qquad (14)$$

$$Д_z = Д_1 + 2\cdot(z-1)\cdot S = Д_{ct} + (2\cdot z-1)\cdot S; \qquad (15)$$

$$Д_{sh} = Д_{ct} + 2\cdot z\cdot S, \qquad (16)$$

z – number of layers of coils, S – distance between layers, d_p – diameter HTT.

After substituting (13)–(16) into (11) and solving the resulting equation with respect to z, we get an expression for determining the number of layers z:

$$z = \frac{Д_{ct} - \sqrt{Д_{ct}^2 - 4\cdot s\cdot\frac{F_c}{\pi(d_p - S)}}}{-2\cdot s}. \qquad (17)$$

To satisfy the condition of a constant length of tubes in the layers, the number of coils in the layers varies. The distribution of the number of coils in the layers is as follows. The average number of tubes in the layer:

$$N_{p.avr.} = N_p / z. \qquad (18)$$

The number of tubes in the i-th layer:

$$N_{p.i} = N_{p.avr.} \cdot Д_i / Д_{avr}. \qquad (19)$$

Thermal calculation performed using ASPEN-TECH computer code, which for given initial data determines the characteristics of the heat transfer surface: dimensions, heat exchange area, number of tubes, distribution of tubes in layers, length of HTT, mass of the heat transfer surface, hydraulic resistance on the side of the coolant and the working fluid.

4. OPTIMIZATION CRITERION

The optimal parameters are usually those that correspond to the minimum of annual costs [6, 8]. Thus, the optimization criterion is the annual expenses, are determined by the expression:

$$Z = E_н \cdot К + I, \qquad (20)$$

where К – cost of SG; $E_н$ – normative coefficient, $E_н$ = 0.12; I – annual expenses.

Annual production expenses represent the sum of the component of capital costs and operating expenses:

$$I = I_c + I_{oper}. \qquad (21)$$

Operating expenses are defined as pump drive costs:

$$I_{oper} = [\Delta P_c \cdot v_c \cdot G_c / (\eta_p \cdot \eta_m) + \\ + \Delta P_{wf} \cdot v_{fw} \cdot G_{fw} / (\eta_p \cdot \eta_m)] \cdot C_{el} \cdot \tau_{an}, \qquad (22)$$

where ΔP_c, ΔP_{wf} – hydraulic resistance between pipe (coolant) and pipe spaces (working fluid) in accordance; v_c, v_{fw} – specific volume of coolant and feedwater at the inlet to the SG, m^3/kg. With the parameters of the coolant at the inlet 270 °C and 15 MPa v_c = 0.00128 m^3/kg. With parameters of feed water at the inlet 65 °C and 4.0 Mpa v_{fw} = 0.00102 m^3/kg.

η_p, η_m – efficiency of the pump and electric motor, respectively, for the circulation pump of the 1st circuit and feedwater pump. Accepted for the circulation pump of the 1st circuit $\eta_p = 0.76$; $\eta_m = 0.98$. For feedwater pump $\eta_p = 0.8$; $\eta_m = 0.98$; G_c, G_{fw} – coolant and feedwater consumption, kg/s. G_c = 314.34 кг/с, G_{fw} = 16.96 kg/s; C_{el}, τ_{an} – electricity cost and annual operation time. Accepted for industrial consumers on January 2018: C_{el} = 1.67485 UAH/(кW· hour) = 0.062 $/(кW· hour), τ_{an} = 7000 hours.

DEFINITION ALGORITHM OF SG COST

Cost of SG is defined as the sum of the cost of materials for each part, multiplied by the coefficient of accounting for the cost of manufacture:

$$К_{пг} = \sum_i^n m_i c_i k_i, \quad (23)$$

where m_i – mass of the i-th component; c_i – unit cost of the i-th material; k_i – production cost factor. Given the design experience in production, k = 2.

When determining the cost of SG, its elements are taken into account: vessel (cylindrical shell); vessel bottom head cap, central tube, shaft, heat exchange surface, closure head cap.

Cost HTT from Ti alloy is equal to 60.55 $/kg [9].

The central tube is made of stainless steel. Cost depends on diameter. As a result of data processing in the literature, an approximation expression of the dependence of the pipe cost on the diameter was obtained:

$$С_{цт} = \frac{158146\cdot d^3 - 70166\cdot d^2 + 12066\cdot d - 453.93}{26.7}, \text{\$/м}. \quad (24)$$

Shaft is made of stainless steel, cost 3700 $/т.

The vessel is made of steel 10ГН2МФА at cost 956.27 $/т [10].

Shaft outer diameter

$$D_{out}^{sh} = D_{in}^{sh} + 2\delta_{sh},$$

δ_{sh} – shaft wall thickness.

The thickness of the annular gap between the shaft and the vessel is determined on the basis of a given area of the annular gap, which is determined from the continuity equation to ensure the optimal coolant speed in the annular gap:

$$F_{ag} = \frac{G_c \cdot v_c}{w_c},$$

where G_c – coolant flow rate, kg/s; v_c – specific volume of coolant at the inlet to the steam generator; w_c – the coolant speed in the annular gap is accepted.

Ring Gap Area

$$F_{ag} = (D_{out}^{sh} + (D_{out}^{sh} + \delta_{ag}))/2 \cdot \pi \cdot \delta_{ag} . \quad (25)$$

From here

$$\delta_{ag} = \sqrt{(\pi D_{out}^{sh})^2 + 2 \cdot F_{ag}/\pi} - \pi D_{out}^{sh} . \quad (26)$$

Inner diameter of the vessel

$$D_{vi} = D_{out}^{sh} + 2 \cdot \delta_{ag}.$$

Considering surfacing 5 mm, the inner diameter of the vessel

$$D_{vi}^{ws} = D_{vi} + 2 \cdot \delta_{surfacingи}.$$

Wall thickness of the vessel [7]

$$\delta_v = \frac{P_p \cdot D_{vi}^{ws}}{2[\sigma]\cdot\varphi - P_p} + C, \quad (27)$$

where P_p – design pressure of coolant, P_p= 17.2 МПа; $[\sigma]$ – permissible load for vessel material at design temperature $t_p = 350$ ℃.

For steel 10ГН2МФА [7] $R_m^t(350\ ℃) = 491$ Mpa, $R_{P0,2}^t(350\ ℃) = 294$ Mpa.

permissible load

$$[\sigma] = \min\begin{cases} 491/2.6 = 188.8\ \text{Mpa} \\ 294/1.5 = 196\ \text{Mpa} \end{cases} = 188.8\ \text{MPa},$$

φ – hole attenuation coefficient. The annular nozzle has amplifications, therefore, for the main part of the vessel $\varphi = 1$.

C – coefficient in the formula (1). C_2 – for pearlitic steel when working in water to 350 ℃, pH=8...10 equivalent to 1 mm. Accept C = 2 mm.

Outer diameter of the vessel: $D_{vo} = D_{vi}^{ws} + 2\delta_v$.

Bottom head cap of vessel: provided that the height of the bottom $h_в = 0.3D_{vi}^{ws}$, bottom thickness equal to wall thickness of vessel [11] $\delta_b = \delta_v$. At lower bottom heights, wall thickness $\delta_b > \delta_v$, which causes technological difficulties in the manufacture.

Theoretical blank diameter for the bottom [11]

$$D_{t.b} =$$

$$= 2\sqrt{(D_{vo} - \delta_v)h_1 + 0{,}345 \cdot \xi_H \cdot (D_{vo} - \delta_v)},$$

where h_1 – height of the cylindrical shell of the bottom, accepted h_1 = 0.1 м [11].

The coefficient ξ_H is determined by the expression

$$\xi = 0{,}725\left(1 + \frac{K^2}{2\sqrt{1-K^2}} \ln\frac{1+\sqrt{1-K^2}}{1-\sqrt{1-K^2}}\right),$$

$$K = \frac{\frac{D_{tb}}{\delta_b} - 4}{2(\frac{D_{tb}}{\delta_b} - 2)}.$$

Volume of the bottom metal

$$V_b = \pi \cdot D_{tb.}^2 \cdot b.$$

When the density of steel $\rho_{ст}$ the mass of the bottom

$$m_b = V_b \cdot \rho_{st}.$$

Closure head cap (CHC) of vessel. CHC thickness without loosening holes (weld thickness of CHC with vessel):

$$\delta_{c1} = K_4 \cdot D_{vi}^{ws} \cdot \sqrt{\frac{P_p}{\varphi[\sigma]}}, \quad (28)$$

$K_4 = K_0 \cdot x = K_0$ (for cover x = 1), $K_0 = 0.53$,
φ = 1 (there are no holes in the weld).

In this way,

$$\delta_{c1} = 0.53 \cdot D_{vi}^{ws} \cdot \sqrt{\frac{17.2}{188.8}}.$$

For flat caps with multiple holes,

$$\varphi_c = \frac{1}{1+\frac{\Sigma d_i}{D_R}+\left(\frac{\Sigma d_i}{D_R}\right)^2}, \quad (29)$$

where Σd_i – sum of hole diameters, lying on the same diameter of the CHC.

The diameter of the holes in the cover for the passage of steam $D_{sc} = 40$ mm, then

$$\Sigma d_i = 2D_{sc} + D_{ct},$$

where D_{ct} – diameter of the central pipe.

Full CHC thickness

$$\delta_{c_2} = K_0 \cdot D_{vi}^{ws} \sqrt{\frac{P_p}{\varphi_c \cdot [\sigma]}}. \quad (30)$$

Weight of blank for CHC:

$$m_c = V_c \cdot \rho_{st} = \pi\left(D_{vi}^{ws}\right)^2 \cdot \delta_{c_2} \cdot \rho_{st}.$$

The height of the cylindrical part of the vessel:

$$H_v = H_{HES} + \delta_{c_1} + 0.5 - 0.135 + 0.15 =$$

$$= H_{HTS} + \delta_{c_1} + 0.515, \quad (31)$$

where H_{HES} – heat exchange surface (HES) height; 0,515 – design size for placement in the upper part of the case of steam collectors that divert a steam from the coils in the upper part under the CHC, and spaces for welding pipes from coils in the lower part above the bottom, feed water is supplied.

The mass of the cylindrical part of the housing

$$m_{cv} = H_v \cdot \pi\left(D_{vi}^{ws} + \delta_v\right) \cdot \delta_v \cdot \rho_{st}.$$

5. HYDRAULIC CALCULATION ALGORITHM

Four programs have been developed. (in Visual Basic Application) for calculation of hydraulic resistance:

1) for the coolant from the entrance to the steam generator to the entrance to the heat exchange surface;

2) for the coolant from the exit from the heat exchange surface to the exit from the SG;

3) for feedwater from the entrance to the SG to the entrance to the heat exchange surface;

4) for steam from the exit from the heat exchange surface to the exit from the SG.

6. RESULTS OF OPTIMIZATION OF THE MAIN STRUCTURAL PARAMETERS OF SG

Based on the above materials, the main structural parameters of SG were optimized: diameter of coils, location distance (between coils in a layer and between layers), feedwater speed at the inlet to the coils.

Optimization criterion – a minimum of reduced annual costs. To carry out variant calculations, a mathematical model of the SG in EXCEL was developed, into which the matrix of the results of calculations according to the program was inserted ASPEN-Tech.

In Fig. 2, 3 for example, the dependencies of the main indicators: K is cost SG, operating costs and the given costs Z of the diameter of the coils and water velocities at the inlet of the HTT. The search method was used to find the optimal values.

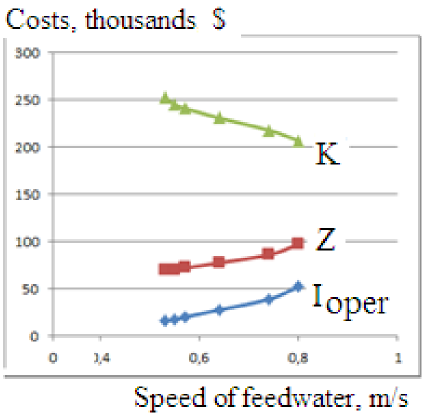

Fig. 2. Dependence of cost SG K, operating costs and reduced annual costs Z of the water speed at the inlet to the coils

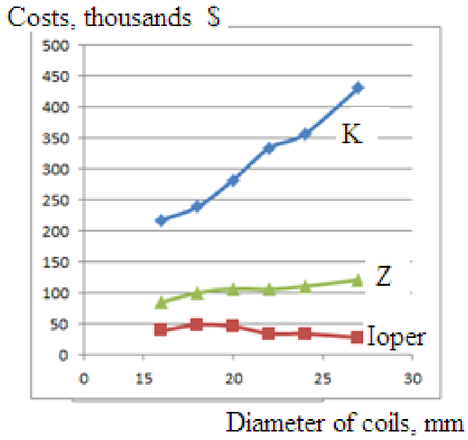

Fig. 3. Dependence of cost SG K, operating costs Ioper and reduced annual costs Z from the diameter of the coils

Ranges of change: diameter 16…27 mm, distance s_1=18…29, s_2=18…32 mm, water speed 0.3…0.97 m/s.

As a result of calculation and analysis 142 options, it was found that the minimum value of the reduced costs correspond to the following design parameters:

1) minimum outer diameter of the coil 16 mm;

2) the distance between the coils in the layer (vertically) 20 mm;

3) distance between layers (horizontally) 20 mm;

4) feedwater speed at the inlet to the coils 0.61 m/s.

The main characteristics of the best option are as follows: the number of coils – 252, coil tube length – 36.6 m; mass of heat exchange surface – 3.655 t; number of rows of coils (layers) – 28; number of pipes in the 1st row – 3; the number of pipes in the last row – 15; diameter of heat exchange surface – 1.35 m; height of heat exchange surface – 2.55 m; coolant speed in between pipe space – 1.58 m; feedwater velocity at the inlet to the tubes – 0.61 m/s; feedwater speed at the outlet of the tubes – 37.1 m/s; hydraulic resistance of the heat exchange surface with the working fluid – 3.5 bar; hydraulic resistance of the heat exchange surface with the coolant – 0.0037 bar; heat transfer surface margin – 15%; thickness of the annular gap between the vessel and the shaft – 60.8 mm; inner diameter of vessel – 1.51 m; wall thickness of vessel – 74 mm; outer diameter of vessel – 1.66 mm; height of vessel – 3.3 m; closure head cap of vessel thickness for welding – 242 mm; full closure head cap thickness – 272 mm; vessel weight – 9.644 t; cost of steam generator – 252650 doll.; fluid resistance on the coolant side – 0.068 bar; resistance on the side of the working fluid – 6.16 bar.

CONCLUSION

1. A methodology for the structural calculation of a coil heating surface has been developed.
2. A technique, algorithm and program for calculating the basic geometric characteristics of the main elements of the SG has been developed (steam, closure head cap of vessel, shaft, coils) and cost SG depending on dimensions heat exchange surface.
3. Developed a mathematical model SG, with the help of which the optimization of the design parameters of a 45 MW SG for low-power NPPs was carried out (the design of the KLT-40C type nuclear power plant, which has many years of successful experience in operating as a ship plants, was taken as a prototype).
4. As a result of using the method of enumerating possible options, optimal values were selected with respect to the minimum of the reduced annual costs for the longitudinal and transverse location of the coils, their diameter and feedwater speed at the inlet to the coils.

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ВЫБОР ОСНОВНЫХ КОНСТРУКТИВНЫХ ПАРАМЕТРОВ ПАРОГЕНЕРАТОРА ДЛЯ АЭС МАЛОЙ МОЩНОСТИ

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Рассмотрены конструкция и методика расчета парогенератора (ПГ) для АЭС малой мощности. В качестве прототипа принят ПГ ядерной энергетической установки типа КЛТ-40С (РФ). Приведен общий перечень работ при проектировании ПГ. Разработаны методика конструкционного расчета ПГ со змеевиковой поверхностью нагрева, а также математическая модель ПГ, включающая конструкционный, тепловой, гидравлический, прочностной и экономический расчеты. Описано выведение формул для определения толщины стенки теплообменных трубок (ТОТ) и количества концентрических слоев змеевиков. Приведена методика определения основных размеров корпуса, шахты и плоской крышки. С помощью разработанной компьютерной программы определены оптимальные диаметр ТОТ, шаги расположения ТОТ в слое и между слоями и скорость питательной воды на входе в ТОТ.

ВИБІР ОСНОВНИХ КОНСТРУКТИВНИХ ПАРАМЕТРІВ ПАРОГЕНЕРАТОРА ДЛЯ АЕС МАЛОЇ ПОТУЖНОСТІ

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Розглянуто конструкцію та методику розрахунку парогенератора (ПГ) для АЕС малої потужності. В якості прототипу прийнято ПГ ядерної енергетичної установки типу КЛТ-40С (РФ). Наведено загальний перелік робіт при проектуванні ПГ. Розроблено методику конструкційного розрахунку ПГ зі змієвиковою поверхнею нагріву, а також математичну модель ПГ, що включає конструкційний, тепловий, гідравлічний, міцностний та економічний розрахунки. Описано виведення формул для визначення товщини стінки теплообмінних трубок (ТОТ) та кількості концентричних шарів змійовиків. Наведено методику визначення основних розмірів корпусу, шахти та плоскої кришки. За допомогою розробленої комп'ютерної програми визначено оптимальні діаметр ТОТ, кроки розташування ТОТ у шарі та між шарами, швидкість живильної води на вході в ТОТ.