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RECALCULATING TECHNIQUE OF THE THERMO-HYDRAULIC PARAMETERS OF STRAIGHT-FLOW SMOOTH-WALLED STEAM GENERATORS FROM BOUNDARY CONDITIONS OF THE SECOND KIND FOR BOUNDARY CONDITIONS OF THE THIRD KIND

The article is devoted to the development of a method for recalculating the thermo-hydraulic characteristics of straight-through cylindrical steam-generating channels with boundary conditions of the second kind for boundary conditions of the third kind. The need for the development of this technique is due to the presence in the literature of a large number of calculated dependences describing the heat transfer during boiling and evaporation of the coolant in smooth-walled tubes for boundary conditions of the second kind, while the practical plan problems are often conditioned by other boundary conditions, in particular boundary conditions of the third kind. The ultimate goal of the recalculation method was to create a program for calculating the thermo-hydraulic efficiency of straight-through steam generators. The proposed recalculation technique makes it possible to calculate, in the straight-through tubular steam generators, for boundary conditions of the third kind, such thermal-hydraulic characteristics as the length of the channel required for complete evaporation of the heat carrier, the power required to pump the coolant in this case, and the total amount of heat transferred to the heat carrier during evaporation.

Keywords: *straight-through tubular steam generators, evaporation zone, recalculation of thermo-hydraulic characteristics from boundary conditions of the second kind for boundary conditions of the third kind.*

Стаття присвячена розробці методу перерахунку теплогідравлічних характеристик прямоточних циліндричних парогенеруючих каналів з граничних умов другого роду для граничних умов третього роду. Необхідність розробки даної методики обумовлена наявністю в літературі великої кількості розрахункових залежностей, що описують теплообмін при кипінні і випаровуванні теплоносія в гладкостінних трубах для граничних умов другого роду, в той час, як задачі практичного плану часто обумовлені іншими граничними умовами, зокрема третього роду. Кінцевою метою методики перерахунку було створення програми для розрахунку теплогідравлічної ефективності прямоточних парогенераторів. Пропонована методика перерахунку дозволяє обчислювати в прямоточних трубчастих парогенераторах для граничних умов третього роду такі теплогдравлічні характеристики, як довжина, каналу, необхідна для повного випаровування теплоностеля, потужність, необхідна для прокачування теплоносія в цьому випадку і загальна кількість тепла, яке передається теплоносію в процесі випаровування.

Ключові слова: *прямоточні трубчасті парогенератори, зона випаровування, перерахунок теплогідравлічних характеристик з граничних умов другого роду для граничних умов третього роду.*

Статья посвящена разработке метода пересчёта теплогидравлических характеристик прямоточных цилиндрических парогенерирующих каналов с граничных условий второго рода для граничных условий третьего рода. Необходимость разработки данной методики обусловлена наличием в литературе большого количества расчётных зависимостей, описывающих теплообмен при кипении и испарении теплоносителя в гладкостенных трубах для граничных условий второго рода, в то время, как задачи практического плана часто обусловлены другими граничными условиями, в частности граничными условиями третьего рода. Конечной целью методики пересчёта являлось создание программы по расчёту теплогидравлической эффективности прямоточных парогенераторов. Предлагаемая методика пересчёта позволяет вычислять в прямоточных трубчатых парогенераторах для граничных условий третьего рода такие теплогдравлические характеристики, как длина, канала, необходимая для полного испарения теплоностеля, мощность, необходимая для прокачки теплоносителя в этом случае и полное количество тепла, передаваемое теплоносителю в процессе испарения.

Ключевые слова: *прямоточные трубчатые парогенераторы, зона испарения, пересчёт теплогидравлических характеристик с граничных условий второго рода для граничных условий третьего рода.*

Introduction. In the process of creating of devices that simulate in the terrestrial conditions various heat exchange processes on-board of spacecraft and that use phase transition of the evaporation-condensation in their thermodynamic cycle, quite important is the heat-hydraulic calculation of the evaporation section. The analytical dependencies available in the literature for the thermal calculation of straight-through tube steam generators were mainly obtained for boundary conditions of the second kind. However, in the process of creating various engineering structures, problems arise for solving similar problems for other boundary conditions. Let’s consider a method of recalculating the characteristics of straight-flow smooth-walled steam generators from boundary conditions of the second kind for boundary conditions of the third kind. The purpose of this calculation is to determination the length of the channel necessary for complete evaporation of the coolant, the power expended for pumping of the evaporating heat carrier and the amount of heat necessary for evaporation of the coolant. Let's cause a brief list of calculation formulas describing heat exchange and hydrodynamics in a zone of vaporization of cylindrical smooth-walled straight-flow steam generators.

Boiling into smooth-walled cylindrical channels (pipes).

Zone of nucleate boiling. At creation of methodology for calculating thermal characteristics of smooth and porous channels, we assume that the liquid at the inlet to the channel is on the saturation line. According to [1], in the zone of the nucleate boiling heat transfer coefficient is defined by the formula

$$\alpha = \alpha_1 \sqrt{1 + 7 \cdot 10^{-9} (\rho \mathrm{w}_{mix} r/q)^{3/2} (0{,}7\alpha_o/\alpha_1)^2}, \qquad (1)$$

where

$$\alpha_1 = \sqrt{\alpha_\kappa^2 + (0{,}7\alpha_o)^2}, \qquad (2)$$

here $\alpha_к$ – heat transfer coefficient in the water flow phase within a pipe or channel, which is calculated by the formula

$$Nu_d = \frac{\alpha_\kappa d}{\lambda} = \frac{(\xi\xi 8)\text{Re}\cdot \text{Pr}\cdot C_t}{k + 4{,}5\sqrt{\xi} \quad (\text{Pr}^{2/3} - 1)}, \tag{3}$$

where $$k = 1 + 900/\text{Re}; \tag{4}$$

$$\xi = (1{,}82 \lg \text{Re} - 1{,}64) \tag{5}$$

C_t – amendment on flow flow nonisothermality.
For drop liquids at $\mu_w/\mu_t = 0{,}08 \div 40$

$$C_t = (\mu_f / \mu_w)^n, \tag{6}$$

where n = 0,11 – while heating the liquid;
n = 0,25 – while cooling the liquid.

In formula (6) μ is coefficient of dynamic viscosity, indices *f* and *w* means relating to the mean temperature and the wall temperature, respectively. Value α_o in formula (1) is boiling heat transfer coefficient in a large volume. Calculating value α_o using formula

$$\alpha_o = 4{,}34 q^{0,7} (p^{0,14} + 1{,}35 \cdot 10^{-2}\, p^2), \tag{7}$$

where q is value of the specific heat flux in W/m^2; *p* is pressure in MPa; α_o is heat transfer coefficient in $W/(m^2K)$. Value w_{mix} in formula (1) - the average speed of the water-steam mixture, m/s. Calculated by formula

$$w_{mix} = w_o[1 + x(\rho'/\rho'' - 1)], \tag{8}$$

where w_o - flow speed, m/s; ρ' и ρ'' - respectively, the density of the liquid and vapor on the saturation line; x - expendable mass vapour content.

As it is seen, the formula (1), that used for determine heat transfer coefficient α, contains the heat flux q. Let's substitute in equation (1) the value of the heat flux q, written for the boundary conditions of third kind. According to [2], for boundary conditions of the third kind can be written the following relationship:

$$\frac{1}{\kappa} = \frac{1}{\alpha} + \frac{1}{\kappa'} \tag{9}$$

or

$$\kappa = \frac{\alpha\kappa'}{\kappa' + \alpha}, \text{ where } \kappa' = \frac{Bi\lambda}{d}; \tag{10}$$

so we can write

$$q = \kappa(T_{1\infty} - \overline{T}_s) = \left(\frac{\frac{\alpha \cdot Bi \cdot \lambda}{d}}{\frac{Bi \cdot \lambda}{d} + \alpha} \right) (T_{1\infty} - \overline{T}_s) \tag{11}$$

In the relations (9) - (11) are used the following notations:

α – local heat transfer coefficient from the fluid flowing in the pipe to an inner wall surface; κ' – local heat transfer coefficient from the inner wall to the environment; κ – local heat transfer coefficient from the fluid flowing in the pipe to the environment; $t_{1\infty}$ – ambient temperature; $\bar{t}_{(s)}$ – (mass-average) temperature of the liquid at the saturation line in this cross section.

For a round pipe, for the parameter κ', according to [2] , we can write the next relationship:

$$\frac{1}{\kappa'} = \frac{d}{2\lambda_c} \ell n \frac{d_1}{d} + \frac{d}{\alpha_1 d_1} \tag{12}$$

where d and d_1 - inner and outer diameters of the tube; δ – wall thickness; λ_c – thermal conductivity of the wall material.

At substituting in equation (1) value of $q = \kappa(t_{1\infty} - \bar{t}_{(s)}) = f(\alpha)$ from equation (11), we can notice that in this case the value α is contained as in the left so in right side of the equation. Therefore, the equation (1) is non-linear algebraic equation regarding to the value the heat-transfer coefficient α, that can't be resolved analytically in explicit view. When calculating the value α according equation (1) should use the method of successive approximations. The value of the specific heat flux recorded by the formula (11) for the boundary conditions of the 3rd kind should be also inserted in the formula (7) for heat transfer coefficient α_o (at the boil in a large volume).

Calculation of boiling heat transfer in cylindrical smooth wall channels at dryout zone. As it is known [1], [3], [4], at boiling liquid in the smooth-walled pipe, with an increase in the value of mass consumable steam content, starting with a certain value, called the boundary vapor content, dispersed flow regime occurs, when starting the drying of the liquid film on the wall. Vapor-liquid stream begins to move in a vaporized mixture of steam and liquid droplets. Thus there is a significant deterioration in the heat exchange channel. According to [5] the relation for the calculation of the boundary vapour content between pre-crisis zone and supercritical zone can be represented as follows.

$$x_{bond.}=0{,}3+0{,}7\exp(-45\tilde{w}) \quad (13)$$

where $\tilde{w}=[\rho w\mu'/(\sigma\rho')](\rho/\rho'')$; $\rho w = w_g$ – mass velocity.

The heat transfer coefficient in the dryout zone was calculated from the equation proposed by [6]:

$$Nu''=0{,}028\,\mathrm{Re}''^{0,8}\,\mathrm{Pr}''^{0,4}(\rho'/\rho'')^{1,15}\ , \quad (14)$$

where $\mathrm{Nu}''=\alpha d/\lambda''$; $\mathrm{Re}''=w''d/\nu''$ - the Reynolds number is calculated using the reduced speed of steam $w_r=(\overline{\rho w})x/\rho''$. Here d –diameter of the channel; λ'' - thermal conductivity of a vapour.

In this case, for calculating of the specific heat flow transferred into the evaporation zone in the supercritical part of the channel (zone of dryout), as well, as for the zone of advanced nucleate boiling, it is necessary to use a formula similar to the expression (11), in which, however, instead of the heat transfer coefficient for the pre-crisis areas should be used the heat transfer coefficient for the supercritical zone (dryout zone).

$$q=\kappa(T_{1\infty}-\overline{T}_s)=\left(\frac{\dfrac{\alpha_{sc.z}\cdot Bi\cdot\lambda}{\alpha}}{\dfrac{Bi\cdot\lambda}{\alpha_{sc.z.}}+\alpha_{sc.z}}\right)(T_{1\infty}-\overline{T}_s)\ , \quad (15)$$

where $\alpha_{sc.z}$ – the heat transfer coefficient in the supercritical zone (dryout zone)

Calculation of hydraulic resistance at the motion of two-phase vapour-liquid flow in smooth-wall channels. When calculating the hydraulic resistance at the motion of two-phase vapour-liquid flow in the pipes and channels, the method developed by Lockhart, R. W. and Martinelli R.C. [7] was used. The essence of it is that the pressure gradient due to friction in the two-phase flow is usually expressed in terms of coefficients, which are multiplied by the respective gradients in the single-phase flows, i.e

$$(dp/dz)_{TP}=\Phi_L^2(dp/dz)_L \quad \text{or} \quad (dp/dz)_{TP}=\Phi_G^2(dp/dz)_G \quad (16)$$

where $(dp/dz)_{TP}$ – is pressure drop due to friction in the two-phase flow; $(dp/dz)_L$ and $(dp/dz)_G$ - are respectively, the pressure drop for a liquid or gas, if a liquid or gas occupied the entire cross section of the pipe; Φ_L^2 и Φ_G^2 - empirically determined coefficients; z is a coordinate.

Lockhart and Martinelli [6] found that the coefficients Φ_L^2 and Φ_G^2 are the function of the parameter X^2, which is determined as follows

$$X^2 = (dp/dz)_L / (dp/dz)_G \tag{17}$$

According Chisholm, D., Sutherland L. [8]:

$$\Phi_L^2 = 1 + C/X + 1/X^2 , \tag{18}$$

$$\Phi_G^2 = 1 + CX + X^2 \tag{19}$$

where
$$C = (1/K)(\sqrt{\rho_L/\rho_G}) + (K)(\sqrt{\rho_G/\rho_L}) \tag{20}$$

Herein K is the slide factor, that according Zivi, S. M. [9] for vapour-water flow is determined as:

$$K = (\rho_L/\rho_G)^{1/3} \tag{21}$$

When calculating values $(dp/dz)_L$ and $(dp/dz)_G$ in formula (6)), the following values of Reynolds numbers have to be used:

$$Re_L = Re_o (1 - x) ; \tag{22}$$

$$Re_G = Re_o\, x\, (\mu'/\mu'') \tag{23}$$

where Re_o is Reynolds number at the channel inlet; x is the mass expendable vapour content of the flow; μ′ and μ″ are, correspondingly, coefficients of dynamic viscosity of liquid and vapour at saturation line.

The scheme for calculating the thermo-physical parameters of a cylindrical channel when two-phase vapour-liquid flow passes through it and under boundary conditions of the third kind. Let's depict schematically the order of calculations.

At the inlet to the channel of diameter d, the flow parameters on the saturation line are set - temperature T_{so} and pressure P_{so}. Also, the Reynolds number Re_0 at the entrance of the channel is set (mass flow rate ṁ, kg / c). On the channel surface, the ambient temperature $T_{1\infty}$ and the law of heat exchange between the channel wall and the environment (criterion Bi) are set. In this case, the temperature drop along channel wall thickness is not taken into account, since it is assumed that the wall thickness of the channel is rather small.

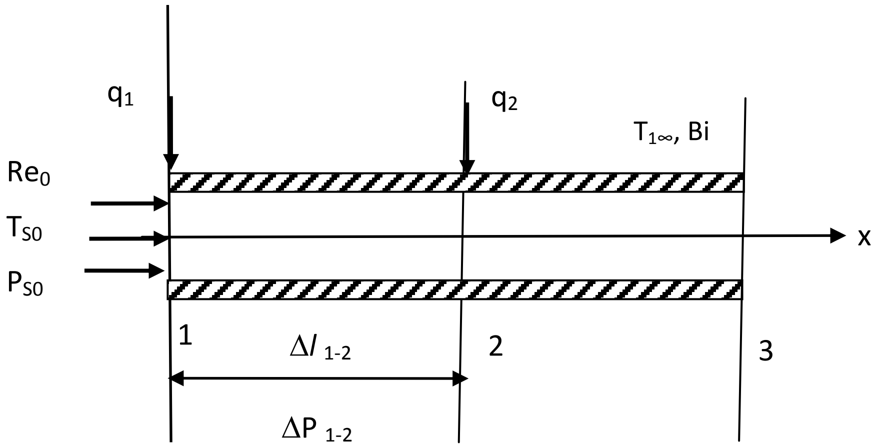

Fig. 1. Calculation scheme

Due to the fact that for beginning of calculations of thermal characteristics in a smooth-wall steam-generating channel (formulas (1) - (12)), it is necessary to know the value of the vapour content, then we assume that at the channel entrance this value is equal to 0.001 (x_0 = 0.001). As is known [1], the zone of heat exchange in cylindrical channels in general is characterized by two sections - zone of nucleate boiling and dryout zone (respectively pre-crisis zone and supercritical zone). The boundary between these zones is determined by the crisis of heat exchange of the second kind. The parameter that characterizes the end of one section and the beginning of the second section is the boundary vapour content. Let's describe firstly the calculation procedure for the zone of nucleate boiling.

Since the calculated values of the heat flux and the heat transfer coefficient in formula (1) are written for boundary conditions of the second kind, we stake out the channel into small sections Δl, within which we will assume that the value of the heat flux is constant. The parameter characterizing the end of one section and the beginning of the second section is the boundary vapour content. Within these sections, the calculation of the thermo-physical characteristics of the channel was carried out by the method of successive approximations.

In the first step of the calculations in section 1 (input section) (Fig. 1), calculate the specific heat flux

$$q_1^{(1)} = f\,(T_{1\infty},\ Bi,\ T_{so},\ P_{so},\ x_o,\ \mathrm{Re}_o,\ d)\ , \tag{24}$$

from equation (1) and equation (11) by the half-dividing method in the region $0 \le \alpha \le 10^7$, since according to [4] the heat transfer coefficient at boiling of the moving fluid in the pipes varies within the limits of $\alpha \approx 10^3$-10^5 W / m^2 K.

Next, we calculate the total amount of heat, which is transferred to the evaporating heat carrier between sections 1 and 2 by formula

$$Q_{1-2}^{(1)} = q_1 \cdot F_{lat.surf.1-2} \;, \tag{25}$$

where
$$F_{lat.surf.1-2} = \pi d \Delta\ell_{1-2} \;. \tag{26}$$

After that, we find the increment of the vapour content in the section 1-2:

$$\Delta x_{1-2}^{(1)} = \frac{Q_{1-2}}{\dot{m} \cdot r(1-x_0)} \;, \tag{27}$$

and the value of the vapor content in section 2 in the first step of the iteration:

$$\mathrm{x}_2^{(1)} = \mathrm{x}_o + \Delta \mathrm{x}_{1-2} \;. \tag{28}$$

where $\dot{m}$ – - the total mass flow rate of the heat carrier through the pipe (channel), kg / s; r - is the heat of vaporization, J / kg; x – is the mass steam content of flow.

Note, that the lower indices in the above expressions refer to the cross section number of channel, and the upper indices refer to the sequence number of the iteration.

Knowing the values of x_o and $\mathrm{x}_2^{(1)}$, we find the average value of the vapour content between cross sections 1 and 2 in the first step of the iteration

$$x_{1-2\,\mathrm{av.}}^{(1)} = (x_o + x_2^{(1)})/2 \tag{29}$$

Based on the thermo-physical properties of the liquid in the inlet cross section of the pipe, the calculated vapor content, the relative length of the section $x_{1-2\,\mathrm{av.}}^{(1)}$, the relative length of the section $\Delta\xi = \Delta\ell / d$, the Reynolds number at the entrance of the channel Re_o and the diameter of the channel d, we calculate the pressure drop in section 1-2 (formulas (16) - (23))

$$\Delta P_{1-2}^{(1)} = f\,(\Delta\xi,\; x_{1\text{-}2\,\mathrm{av.}}^{(1)},\; \mathrm{Re}_o,\; d) \tag{30}$$

and the value of saturation pressure in the cross section 2:

$$P_{s2}^{(1)} = P_{so} - \Delta P_{1-2}^{(1)} \;. \tag{31}$$

Using the pressure value $\mathrm{P}_{s2}^{(1)}$, we find the temperature value on the saturation line in cross section 2 in the first step of the iteration

$$T_{s2}^{(1)} = f(P_{s2}^{(1)}) \ . \tag{32}$$

Values of thermo-physical properties of liquid and vapour on the saturation line are usually found using reference data by interpolation between discrete points in steps of 10 ^{0}C. In the course of numerical realization of this calculation procedure, interpolation was carried out using the Lagrange interpolation formula. The discrete values of the thermo-physical properties of liquid and vapour were taken from [10].

The average temperature of the liquid in the section 1-2 is calculated from the temperature values T_{so} *и* $\mathrm{T}_{s2}^{(1)}$

$$T_{s1-2\ \mathrm{av}}^{(1)} = \frac{T_{so} + T_{s2}^{1}}{2} \ . \tag{33}$$

By the value of this temperature, using the Lagrange interpolation formula, the average value of the saturation pressure in the 1-2 section is calculated.

$$P_{s1-2\ \mathrm{av}}^{(1)} = f(T_{s1-2\ \mathrm{av}}^{(1)}) \tag{34}$$

Further, proceeding from the previously calculated values $x_2^{(1)}$ $\mathrm{T}_{s2}^{(1)}$, $\mathrm{P}_{s2}^{(1)}$ (formulas (28), (31), (32)), we calculate the value of the specific heat flux in cross section 2 in the first step of the iteration.

$$q_1^{(1)} = f(T_{1\infty},\ Bi,\ T_{s2}^{(1)},\ P_{s2}^{(1)},\ x_2^{(1)},\ \mathrm{Re}_o\ ,\ d) \tag{35}$$

Then we calculate the value of the average specific heat flux in the section 1-2 in the first step of the iteration, assuming this value is constant within the section 1-2

$$q_{1-2\ \mathrm{av}}^{(1)} = \frac{q_1^{(1)} + q_2^{(1)}}{2} \ . \tag{36}$$

After this, we go to the second step of the iteration and again calculate the value of the total amount of heat which is transferred to the evaporating heat carrier between sections 1 and 2

$$Q_{1-2}^{(1)} = q_{1-2\,\mathrm{av}}^{(1)} \ \pi d \Delta \ell_{1-2} \tag{37}$$

Further, according to the above scheme, we calculate the next set of thermo-hydraulic characteristics of the evaporating flow in section 1-2 in the second step of the iteration.

$$\Delta x_{1-2\,\mathrm{av}}^{(1)} = \frac{Q_{1-2}^{(2)}}{\dot{m} \cdot r(1-x)} \tag{38}$$

We note that at this step of the iteration the value of the heat of evaporation r in formula (38) is calculated from the mean values of the saturation temperature $T^{(1)}_{s1-2\,\mathrm{av}}$ and the saturation pressure $P^{(1)}_{s1-2\,\mathrm{av}}$ obtained in the previous iteration step (formulas (33) and (34), respectively).

$$x_2^{(2)} = x_o + \Delta x_{1-2\ av}^{(2)}\ , \tag{39}$$

$$x_{1-2\ av}^{(2)} = \frac{x_o + x_2^{(2)}}{2}\ , \tag{40}$$

$$\Delta P_{1-2}^{(2)} = f(\Delta\xi,\ x_{1-2\,\mathrm{av}}^{(2)},\ \mathrm{Re}_o,\ d), \tag{41}$$

$$P_{s2}^{(2)} = P_{so} - \Delta P_{1-2}^{(2)}\ , \tag{42}$$

$$T_{s2}^{(2)} = f(P_{s2}^{(2)}), \tag{43}$$

$$T_{s1-2\ av}^{(2)} = \frac{T_{so} + T_{s2}^{(2)}}{2}\ , \tag{44}$$

$$P_{s1-2\ av}^{(2)} = f(T_{s1-2\ av}^{(2)})\ , \tag{45}$$

$$q_2^{(2)} = f(T_{1\infty},\ Bi\ ,\ T_{s2}^{(2)},\ P_{s2}^{(2)},\ x_2^{(2)},\ \mathrm{Re}_o\ ,\ d)\ , \tag{46}$$

$$q_{1-2\ av}^{(2)} = \frac{q_1 + q_2^{(2)}}{2} \tag{47}$$

Having the values of $q^{(1)}_{1-2\ \mathrm{av}}$ and $q^{2}_{1-2\ \mathrm{av}}$, calculated at the first and second step of the iteration, we compare them with each other and calculate the relative magnitude of the discrepancy between these quantities.

$$\Delta C = \frac{q^{(2)}_{1-2\,\mathrm{av}} - q^{(1)}_{1-2\ \mathrm{av}}}{q^{(2)}_{1-2\ \mathrm{av}}} \cdot 100\,\% \tag{48}$$

If the value of ΔC is greater than 1%, then the average value of the specific heat flux in the distance between the cross sections 1-2 obtained in the second iteration step ($q^{(2)}_{1-2\ av}$) is substituted into the formula (37) instead of the same value that was obtained in the first step of the iteration ($q^{(1)}_{1-2\ \mathrm{av}}$) and calculations of the thermo-hydraulic values, starting with formula (37) to formula (48) are carried out again. In such way is organized the third step of the iteration (the third successive approximation). The calculation of thermal-hydraulic characteristics by the method of the successive approximations in the initial section 1-2 continues until the condition ΔC≤1% is satisfied. After this condition is fulfilled, the final value of the total amount of heat is calculated on the interval between cross sections 1 and 2

$$Q^{(I)}_{1-2\ \text{оконч.}} = q^{(i)}_{1-2\ \text{оконч.}} \pi d \Delta\ell \ . \qquad (49)$$

After the condition ΔC≤1% has been fulfilled, the hydraulic power required to pump the coolant in the 1-2 section is calculated too.

$$N^{(I)}_{1-2\ \text{оконч.}} = \Delta P^{(i)}_{1-2\ \text{оконч.}} \cdot \frac{\dot{m}}{\rho^{(i)}_{1-2}} \ , \qquad (50)$$

where $\dot{m} = \frac{\mathrm{Re}_o\, \mu_o \pi d}{4}$ - is the total mass flow rate of the heat carrier through the pipe (channel), kg / s; Re_0 and μ_0 are the Reynolds number and the dynamic viscosity of the fluid in the inlet cross section of the channel; $\rho^{(i)}_{1-2}$ - the density of the liquid, calculated from the value of temperature $T^{(i)}_{s1-2}$.

After calculating of the values $Q^{(I)}_{1-2\ \text{fin.}}$ and $N^{(I)}_{1-2\,\text{fin.}}$, these values as well as the value $\Delta\ell_{1-2} = const$ are entered in the database for their subsequent use when finding the total amount of heat which is transferred to the evaporating heat carrier, the total power expended for pumping the cooler and the total length of the channel at which the evaporation of the coolant is complete.

After the calculation of the thermo-hydraulic parameters of the evaporating flow in the section 1-2 was completed, the calculated values in cross section 2 were taken as the initial values for the next interval 2-3 (Fig. 1) and the calculations in this section were carried out according to the calculation scheme in the interval 1-2. After the completion of calculations in section 2-3, the summation of the values $Q^{(I)}_{1-2\ \text{fin.}} + Q^{(II)}_{2-3\ \text{fin.}}$; $N^{(I)}_{1-2\ \text{fin.}} + N^{(II)}_{2-3\ \text{fin.}}$; $\Delta\ell_{1-2} + \Delta\ell_{2-3}$, the calculations were transferred to the interval 3-4, etc.

The calculation was carried out until the vapor content with an accuracy of 1% became equal to the boundary vapour content x_{bond} (formula (5)).

Next, we briefly describe the order of calculations in the dryout zone ($x_{\text{bond.}} \le x \le 1$). When performing calculations in this zone, the computation of the heat flux and the heat transfer coefficient should be performed using the dependence (15), (14), (12) that are typical for the dryout zone. In the rest, the calculation procedure coincides with the order of calculation for the pre-critical zone. Calculation in the dryout(supercritical) zone should be carried out until, with an accuracy of 1%, the vapour content will not differ from 1.

Conclusion. The above procedure allows to do the recalculation of the thermo-hydraulic characteristics of straight-flow smooth-walled steam generators from boundary conditions of the second kind for boundary conditions of the first kind. This calculation method was created with the purpose of subsequent calculation of the thermo-hydraulic efficiency of straight-through cylindrical steam generators and was implemented in the form of a computational program.

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