ТЕРМОДИНАМІЧНИЙ АНАЛІЗ ТА МОДЕЛЮВАННЯ

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Methodology for recalculating the thermal-hydraulic characteristics of straight-through cylindrical steam generators operating on freon coolant, from boundary conditions of the second kind to third kind

Anatolii Lukisha
N. S. Polyakov Institute of Geotechnical Mechanics, National Academy of Sciences of Ukraine, 14 Matroska Str., Dnipro, 49033, Ukraine
🖂 e-mail: lukisha@ukr.net
ORCID: http://orcid.org/0000-0002-9535-4228

The methodology presented in the article allows calculating the thermal-hydraulic characteristics of direct-flow cylindrical steam generators operating on a freon coolant for boundary conditions of the third kind. The purpose of this article is a calculation to determine the length of the channel required for complete evaporation of the coolant, the power expended on pumping the evaporating coolant and the amount of heat required for evaporation of the coolant. In addition, of significant interest is the search for design parameters of steam generators with heat exchange intensifiers, with which it is possible to obtain an energy gain compared to smooth-walled steam generators. This gain can be obtained only under boundary conditions of the first or third kind. In the case of boundary conditions of the second kind, according to the law of conservation of energy, with the same heat flows on the surface, for a channel with a heat exchange intensifier and for a smooth-walled channel, the same length of these channels is required for the evaporation of the same amount of liquid. Considering that the hydraulic resistance of channels with a heat intensifier is higher than the hydraulic resistance of smooth-walled channels, it is impossible to obtain an energy gain in this case. It should be noted that since the analytical dependencies available in the literature for the thermal calculation of the evaporation sections of direct-flow steam generators were mainly obtained for the boundary conditions of the second kind, it is necessary to create a method for recalculating the thermal-hydraulic characteristics of direct-flow steam generators from the boundary conditions of the second kind, in particular, for the boundary conditions of the third kind. This is ultimately necessary for conducting a comparative analysis of direct-flow freon steam generators with heat exchange intensifiers and smooth-walled freon direct-flow steam generators.

Keywords: *Steam generators; Freon coolant; Boundary conditions; Heat exchange; Thermal-hydraulic characteristics*

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1. Introduction

An important task of industrial development in modern conditions is the creation of highly efficient refrigeration and air conditioning equipment operating on freon coolants and using various types of heat exchange intensifiers, such as porous high-heat-conducting inserts. At the same time, in the process of design development of such systems, the thermohydraulic calculation of the evaporation or vaporization section is of significant importance. Let us consider the method of recalculating the characteristics of straight cylindrical smooth-walled freon steam-generating channels from the boundary conditions of the second kind to the boundary conditions of the third kind. We will provide a short list of calculation formulas characterizing heat exchange and hydraulics in the evaporation zone of direct-flow steam generators.

2. Heat exchange in a cylindrical smooth-walled channel

According to [1], the average heat transfer coefficient during boiling of Freon in the channels can be calculated using the formula of Bogdanov S.N.:

$$\alpha = Aq^{0,6}(w\rho)^{0,2}d_{in}^{-0,2} \qquad (1)$$

where α [kW / m^{2}K], q [kW / m^2]; $w\rho$ is the specific mass flow rate of the cooler, [kg / m^2 · s]; d_{in} – internal diameter of the channel [m].

To calculate the coefficient A, depending on the type of Freon and boiling temperature, the following reference table is recommended in [1]:

***Table** – Data for calculate the coefficient A[1]*

Refrigerant	Boiling temperature t_0, ^{0}C				
	-30	-10	0	10	30
R11	0,0208	0,0300	0,0341	0,0382	0,0498
R142	0,0372	0,0461	0,0514	0,0568	0,0710
R12	0,0536	0,0659	0,0719	0,0776	0,0928
R22	0,0599	0,0738	0,0833	0,0928	0,1170

Note: Here R11, R142, R12, R22 are respectively the names of the refrigerants Freon 11, Freon 142, Freon 12, Freon 22

Formula (1) takes into account the various flow regimes that occur when the vapour content varies along the length of the evaporator from $x = 0$ to $x = 1$. The range of application of the formula (1) is: $q = 0.6… 25$ kW / m^2; $w\rho = 50…600$ kg / m^2 · s.

The formula (1) for determining the heat transfer coefficient α includes the value of the heat flow q. Let us substitute into equation (1) the value of the heat flow q, written for the boundary conditions of the 3rd kind. According to [2], for the boundary conditions of the third kind, the following relationship can be written:

$$\frac{1}{\kappa} = \frac{1}{\alpha} + \frac{1}{\kappa'} \qquad (2)$$

or

$$\kappa = \frac{\alpha\kappa'}{\kappa' + \alpha}, \text{ where } \kappa' = \frac{\mathrm{Bi}\lambda}{d} \qquad (2)$$

That's why

$$q = \kappa(T_{1\infty} - \overline{T}_s) = \left(\frac{\dfrac{\alpha \cdot \mathrm{Bi} \cdot \lambda}{d}}{\dfrac{\mathrm{Bi} \cdot \lambda}{d} + \alpha} \right)(T_{1\infty} - \overline{T}_s) \qquad (4)$$

In relations (2)-(4) the following notations are adopted: α – local coefficient of heat transfer from the liquid flowing in the pipe to the inner surface of the wall; κ' – local coefficient of heat transfer from the inner surface of the wall to the environment; κ – local coefficient of heat transfer from the liquid flowing in the pipe to the environment; $T_{1\infty}$ – ambient temperature; $\overline{T}_s$ – (average mass) temperature of the liquid on the saturation line in a given section.

For a round pipe, for the parameter κ', according to [2], the following relationship can be written:

$$\frac{1}{\kappa'} = \frac{d}{2\lambda_c} \ell n \frac{d}{d_{in}} + \frac{d_{in}}{\alpha_1 d} \qquad (5)$$

where d_{in} and d – are the inner and outer diameters of the pipe; λ_c is the thermal conductivity coefficient of the wall material.

When substituting the value $q = \kappa(T_{1\infty} - \overline{T}_s) = f(\alpha)$ from equation (4) into equation (1), we obtain the following expression

$$\alpha = Aq^{0,6}(w\rho)^{0,2}d_{in}^{-0,2}\kappa(T_{1\infty} - \overline{T}_s) =$$
$$= A(\frac{\mathrm{Re}_0\,\mu}{d_{in}})^{0,2}d_{in}^{-0,2}\left[\left(\frac{\dfrac{\alpha \cdot \mathrm{Bi} \cdot \lambda}{d_{in}}}{\dfrac{\mathrm{Bi} \cdot \lambda}{d_{in}} + \alpha} \right)(T_{1\infty} - \overline{T}_s)\right]^{0,6} \qquad (6)$$

It was taken into account that since the Reynolds number of the flow at the entrance to the channel is

$$\mathrm{Re}_0 = \frac{w\rho d_{in}}{\mu}, \text{ then } w\rho = \frac{\mathrm{Re}_0\,\mu}{d_{in}} \qquad (7)$$

It can be noted that the value of α is included in the left and right parts of equation (6). Therefore, equation (6) is a nonlinear algebraic equation with respect to the value of the heat transfer coefficient α, which cannot be solved analytically in an explicit form. When calculating the value of α using equation (1), the method of successive approximations should be used.

3. Calculation of the pressure drop during evaporation of freon in a cylindrical smooth-walled channel

When freon evaporates in a cylindrical smooth-walled channel, the pressure drop is calculated, according to [1], [3], using the simplified Beau-Pierre equation for complete evaporation of the refrigerant:

$$\Delta P^{(0)} = f v'' \cdot x_{av}\,(l / d_{in})\,(\rho w)^2 \tag{8}$$

where ΔP_0 [kPa]; f is the coefficient of total resistance, taking into account friction losses, acceleration and flow turns. For pure refrigerant $f = 1{,}5 \cdot 10^{-5}$, and in the presence of oil $f = 3{,}5 \cdot 10^{-5}$. v'' is the specific volume of saturated refrigerant vapor, m^3/kg; x_{av} is the average vapor content (consumption one). Since the calculations assume complete evaporation of freon, we take $x_{av} = 0.5$.

4. Calculation scheme for calculating the thermophysical and hydraulic characteristics of a freon direct-flow evaporator for boundary conditions of the 3rd kind

Let us consider the calculation procedure for a smooth-walled direct-flow steam generator with freon as the working coolant. Since for this case the average heat transfer coefficient along the length of the evaporation zone is specified in the literature, the calculation procedure for the channel length required for evaporation, the amount of heat absorbed in the process, and the pressure drop applied to pump the coolant will look as follows:

1. Based on the data on the flow of liquid at the channel inlet available before the start of the calculation, namely: the Reynolds number Re_0, the dynamic viscosity and thermal conductivity coefficient of the liquid on the saturation line (μ and λ), as well as the ambient temperature and the temperature of the liquid at the channel inlet, and specified the Biot criterion (Bi) and the channel diameter, we determine the average heat transfer coefficient along the length of the channel (evaporation zone) using formula (6):

$$\alpha_{\mathrm{av}}^{(0)} = A q^{0,6} (w\rho)^{0,2} d_{in}^{-0,2} \kappa_{\mathrm{av}}^{(0)} (T_{1\infty} - \bar{T}_s) =$$
$$= A\left(\frac{\mathrm{Re}_0\,\mu}{d_{in}}\right)^{0,2} d_{in}^{-0,2} \left[\left(\frac{\frac{\alpha_{\mathrm{av}}^{0} \cdot \mathrm{Bi} \cdot \lambda}{d_{in}}}{\frac{\mathrm{Bi} \cdot \lambda}{d_{in}} + \alpha_{\mathrm{av}}^{0}} \right) (T_{1\infty} - \bar{T}_{1s}) \right]^{0,6} \tag{9}$$

Since the channel wall thickness is neglected during the calculations, we assume that $d_{in} = d$. The value of A at the zero iteration step is calculated based on the value of the saturation temperature of the coolant t_0 at the channel inlet. The value of the dynamic viscosity μ and thermal conductivity of the liquid λ at the zero iteration step is found based on the values of the temperature and pressure of the coolant on the saturation line at the channel inlet (t_0, P_0).

2. We calculate the average specific heat flow along the length of the evaporator at the zero step of the iteration, using the formula:

$$q_{av}^{(0)} = \kappa_{\mathrm{av}}^{(0)} \Delta t_{\mathrm{av}}^{0} = \left(\frac{\frac{\alpha_{\mathrm{av}}^{(0)} \cdot \mathrm{Bi} \cdot \lambda}{d_{in}}}{\frac{\mathrm{Bi} \cdot \lambda}{d_{in}} + \alpha_{\mathrm{av}}^{(0)}} \right) \Delta t_{av}^{0} \tag{10}$$

Here the superscript (0) indicates the iteration step.

3. Based on the average value of the specific heat flow $q^{(0)}_{av}$ calculated at the zero step of the iteration, we calculate the length of the smooth-walled evaporation tube from the following relationships:

$$Q^{(0)} = q_{av}^{(0)} \cdot \pi d l^{(0)} \tag{11}$$

$$Q^{(0)} = \dot{m} \cdot r \tag{12}$$

$$l^{(0)} = \frac{\mathrm{Q}^{(0)}}{\pi \mathrm{d} \mathrm{q}_{\mathrm{av}}^{(0)}} = \frac{\dot{\mathrm{m}} \cdot \mathrm{r}}{\pi \mathrm{d} \mathrm{q}_{\mathrm{av}}^{(0)}} \tag{13}$$

Since $\dot{m} = \rho w F_{\text{cross section}} = \rho w \cdot (\pi d^2/4)$ and taking into account (7), we obtain:

$$\dot{\mathrm{m}} = \left(\frac{\mathrm{Re}_0\,\mu_0}{\mathrm{d}}\right) \cdot \frac{\pi \mathrm{d}^2}{4} \tag{14}$$

$$l^{(0)} = \frac{\mathrm{Re}_0\,\mu_0 \mathrm{r}}{4 \mathrm{q}_{\mathrm{av}}^{0}} \tag{15}$$

Here $\dot{m}$ – is the mass flow rate of the coolant, kg/s; r is the specific heat of vaporization, J/kg.

4. Using the simplified Bo-Pierre equation [1], [3], we calculate the pressure drop for complete evaporation of the coolant:

$$\Delta P^{(0)} = fv''\cdot x_{av}\,(l\,/\,d)\,(\rho w)^2 \tag{16}$$

The value of the parameter f, according to the data [1], for pure Freon, without oil, is taken to be f = $1{,}5\cdot 10^{-5}$. The value of v" = (1/ρ") – the specific volume of steam on the saturation line is taken at the temperature and pressure at the entrance to the channel. The value of the average mass flow rate of steam content along the length of the channel is taken to be equal to x_{av} = 0,5.

5. We calculate the pressure at the outlet of the channel at the zero step of the iteration:

$$P^{(0)}_{(S)2} = P_0 - \Delta P^{(0)} \tag{17}$$

The subscript (2) refers to the outlet cross-section.

6. Based on the pressure value $P^0_{2(S)}$, we determine the saturation temperature in the outlet section at the zero iteration step:

$$T^0_{(S)2} = f(P^0_{(S)2}) \tag{18}$$

For this purpose, you can use, for example, the data from the reference book [4]

7. We calculate the average temperature head (average temperature difference) based on the values of the temperature head at the inlet and at the outlet of the steam-generating channel:

$$\Delta t_1^{(0)} = T_w - T^0_{(S)1}; \qquad \Delta t_2^{(0)} = T_w - T^{(0)}_{(S)2};$$

$$\Delta t^0_{av} = \frac{\Delta t_1^{(0)} + \Delta t_2^{(0)}}{2} \tag{19}$$

Subscripts 1 and 2 refer to the input and output cross-sections, respectively.

8. We calculate the value of the average specific heat flow at the zero iteration step, based on the value of the average temperature difference at the zero iteration step $\Delta t_{av}^{(0)}$:

$$q_{av}^{(0-1)} = \kappa^{(0)}_{av}\Delta t^0_{av} = \left(\frac{\dfrac{\alpha^0_{av}\cdot \mathrm{Bi}\cdot\lambda}{d_{in}}}{\dfrac{\mathrm{Bi}\cdot\lambda}{d_{in}} + \alpha^0_{av}}\right)\Delta t^0_{av} \tag{20}$$

9. We compare the value $q_{av}^{(0\text{-}1)}$ with the value $q_{av}^{(0)}$ and calculate the value of the relative error of mismatch:

$$\Delta^{(0)} = \frac{q_{av}^{(0-1)} - q_{av}^{(0)}}{q_{av}^{(0-1)}}\cdot 100\ \% \tag{21}$$

10. If the value Δ(0) ≤ 1%, we accept that the size of the smooth-walled steam-generating channel l_{sm}, the average value of the specific heat flow q_{av} and the pressure drop in the channel ΔP, as found.

11. If the value Δ(0) > 1%, we accept

$$q_{av}^{(1)} = \frac{q_{av}^{(0-1)} + q_{av}^{(0)}}{2} \tag{22}$$

12. We calculate the average temperature in the channel

$$t_{av}^{(1)} = \frac{t_{(0)} + T^{(0)}_{(S)2}}{2} \tag{23}$$

13. We determine the value of the coefficient A for calculation using formula (8) at the first step of the iteration, based on the value of $t_{av}^{(1)}$, i.e.

$$A^{(1)} = f(t_{av}^{(1)}) \tag{24}$$

The superscript (1) denotes the iteration number.

14. We calculate the average value of the heat transfer coefficient in the evaporation channel at the first step of the iteration

$$\alpha^{(1)}_{av} = A^{(1)}\cdot q^{(1)}_{av}\left(\frac{\mathrm{Re}_0\,\mu}{d}\right)^{0,2}\cdot d^{-0,2} \tag{25}$$

Note that the calculation of all thermo physical quantities in the channel (except of the complex $\left(\frac{\mathrm{Re}_0\,\mu_0}{d}\right)^{0,2}$) at the first step of the iteration is performed based on the value of the average temperature of the cooler, which is on the saturation line.

15. Next we calculate the length of the smooth-walled evaporation channel at the first step of the iteration

$$l^{(1)} = \frac{\mathrm{Re}_0\cdot\mu_0\cdot r}{4q^{(1)}_{av}} \tag{26}$$

16. After that we find the pressure drop at the first step of iteration

$$\Delta P^{(1)} = fv'' \cdot x_{av} (l^{(1)}/d)(\rho w)^2 \qquad (27)$$

or

$$\Delta P^{(1)} = fv'' \cdot x_{av} (l^{(1)}/d) (\frac{Re_0 \mu_0}{d})^2 \qquad (28)$$

17. We determine the pressure and temperature on the saturation line in the outlet section of the evaporation channel

$$P^{(1)}_{(S)2} = P_0 - \Delta P^{(1)}; \quad T^{(1)}_{(S)2} = f(P^{(1)}_{(S)2}) \qquad (29)$$

18. Next we calculate the temperature heads (the temperature differences) in the inlet and outlet cross-sections of the channel, as well as the average temperature head (average temperature difference) along the length of the channel:

$$\Delta t^{(1)}{}_1 = T_w - T^{(0)}_{(S)1}\, t_0; \qquad \Delta t^{(1)}{}_2 = T_w - T^{(1)}_{(S)2} \qquad (30)$$

$$\Delta t^{(1)}_{av} = \frac{\Delta t^{(1)}_1 + \Delta t^{(1)}_2}{2} \qquad (31)$$

19. After that we calculate the value of the average specific heat flux in the channel at the first iteration step based on the value of $\Delta t_{av}^{(1)}$:

$$q^{(1-1)}_{av} = \kappa^{(1)}_{av} \Delta t^{(1)}_{av} = \left(\frac{\dfrac{\alpha^{(1)}_{av} \cdot Bi \cdot \lambda}{d_{in}}}{\dfrac{Bi \cdot \lambda}{d_{in}} + \alpha^{(1)}_{av}} \right) \Delta t^{(1)}_{av} \qquad (32)$$

20. Compare the valuesof average heat flows $q_{av}^{(1-1)}$ and $q_{av}^{(1)}$ calculated using formulas (32) and (22) at the first step of the iteration.

$$\Delta^{(1)} = \frac{q^{(1-1)}_{av} - q^{(1)}_{av}}{q^{(1-1)}_{av}} \cdot 100\% \qquad (33)$$

21. If the value $\Delta^{(1)} \leq 1\%$, we stop the calculation process and accept the values $l^{(1)}$, and $\Delta P^{(1)}$ as final.

22. If the value $\Delta^{(1)} > 1\%$, we organize the calculation at the second iteration step similarly to the first iteration step, starting with formula (21) and to formula (33), etc.

5. Conclusions

The above method allows calculating the thermal-hydraulic characteristics of direct-flow cylindrical steam generators operating on a freon coolant for boundary conditions of the third kind. This method was created for the purpose of subsequent calculations of the thermal-hydraulic efficiency of direct-flow cylindrical freon steam generators under boundary conditions of the third kind and was implemented in the form of a calculation program.

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Методика перерахунку теплогідравлічних характеристик прямоточних циліндричних парогенераторів, що працюють на фреоновому теплоносії, з граничних умов другого роду для граничних умов третього роду

А.П. Лукіша
Інститут геотехнічної механіки імені Н. С. Полякова Національної академії наук України, вул. Матросська, 14, Дніпро, 49033, Україна
✉ e-mail: lukisha@ukr.net
ORCID: http://orcid.org/0000-0002-9535-4228

Наведена у статті методика дозволяє проводити розрахунок теплогідравлічних характеристик прямоточних циліндричних парогенераторів, що працюють на фреоновому теплоносії, для граничних умов третього роду. Метою даного розрахунку є визначення довжини каналу, необхідної для повного випаровування теплоносія, потужності, що витрачається на прокачування теплоносія, що випаровується, і кількості тепла, необхідного для випаровування теплоносія. Крім того, значний інтерес представляє пошук розрахунково-конструктивних параметрів парогенераторів з інтенсифікаторами теплообміну, при яких можна отримати енергетичний виграш у порівнянні з гладкостінними парогенераторами. Цей виграш можна отримати лише за граничних умов першого чи третього роду. У разі граничних умов другого роду, згідно із законом збереження енергії, при однакових теплових потоках на поверхні, для каналу з інтенсифікатором теплообміну та для гладкостінного каналу, для випаровування однакової кількості рідини, необхідна однакова довжина цих каналів. Враховуючи, що гідроопір каналів з інтенсифікатором тепла вищий, ніж гідроопір гладкостінних каналів, отримати енергетичний виграш у цьому випадку неможливо. При цьому слід зазначити, що оскільки аналітичні залежності для теплового розрахунку ділянок випаровування прямоточних парогенераторів в основному були отримані для граничних умов другого роду, необхідно створити методику перерахунку теплогідравлічних характеристик прямоточних парогенераторів з граничних умов другого роду, зокрема, для граничних умов третього роду. Це, зрештою, необхідно щодо порівняльного аналізу прямоточних фреонових парогенераторів з інтенсифікаторами теплообміну і гладкостінних фреонових прямоточних парогенераторів.

Ключові слова: *Парогенератори; Фреоновий теплоносій; Граничні умови; Теплообмін; Теплогідравлічні характеристики*

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