INFLUENCE OF WEATHER CONDITIONS ON THE POWER OF HEAT REMOVAL BY THE EMERGENCY COOLING SYSTEM OF A REACTOR

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An analytical and numerical analysis is made of the influence of the slot sizes between gate plates and weather conditions on total heat losses in the hot stand-by mode and emergency operating conditions of the passive safety system of an APS.

The present-day level of development of science and technology in nuclear power engineering allows a design of an APS with a vanishingly small probability of accidents accompanied by impermissible radionuclide discharge into the environment. In the best projects (AR-600, AÉS-92, etc.) the possibility of such an accident is evaluated as $10^{-8}-10^{-9}$ reactor/year against a value of $10^{-6}-10^{-7}$ recommended by the IAEA and national normative technical documents, i.e., no more than one accident per thousand reactors per $10^{3}-10^{4}$ years. This result stems from the perfection and reliability of the safety systems and equipment of the station as a whole, which ensure failure of the fission reaction with subsequent removal of residual heat at a prescribed rate to prevent overheating of fuel elements and depressurization of their jackets.

The main problem lies in safe emergency cooling of a reactor plant under mostly unfavorable possible combinations of the postulated initial conditions and accident development scenarios, including long-term failure of all sources of electric power supply and control and monitoring under extreme weather conditions for the region where an APS is located. It is evident that reliable shut-down cooling of a reactor under the conditions mentioned can be accomplished only with the use of so-called passive systems, the functioning of which does not demand any external power sources, control, or monitoring but rests on simple physical processes of heat removal by natural convective circulation, phase transitions, and so on.

To the largest extent the requirements above are met, apparently, by the station "AÉS-92" of the new generation with a water-moderated water-cooled power reactor VVÉR-1000 developed by the Moscow Institute "Atoménergoproekt" and the design office "Gidropress." Removal of residual heat released to the environment is accomplished by vapor condensation of the secondary circuit in condensers with natural circulation of cooling air, thus providing infinitely prolonged operation of the passive system [1].

The system consists of four independent circuits with three condensers in each connected to the vapor main of the corresponding steam generators of four loops of the first circuit. Each condenser is located in an individual air pit with two gates installed at the condenser inlet and outlet. The amount of gate opening (an angle of rotation of gate plates) determines the flow rate of the cooling air and the thermal capacity of the condenser. The gates are actuated by the mechanical action of the vapor pressure in the steam generator. Under normal operating conditions, the gates are shut and the entire system is in the "hot stand-by" mode of operation.

When the pressure exceeds the prescribed value, the gates open and the system spontaneously starts shut-down cooling of the reactor by maintaining the prescribed pressure in the secondary circuit independently of the ambient temperature, power output, and the other parameters. Four groups of condensers are arranged over

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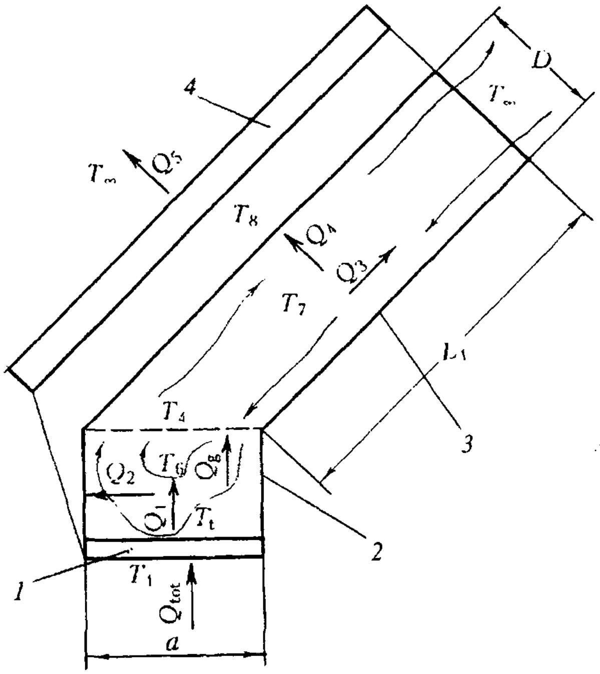

Fig. 1. Schematic of the air pit of the PHRS: 1) top gate; 2) joint zone; 3) air pit; 4) protective dome of the APS.

the perimeter of the upper part of the cylindrical protective envelope (PE) with a spherical dome. Thrust sections of twelve air pots are located along the vertical generators of the spherical PE dome and are integrated by a common deflecter [2].

Air flowing around the PE changes the static pressure distribution over the envelope surface. At wind velocities of about 10 m/sec, the pressure at the inlet of some condensers can exceed that of the natural thrust, which leads either to a decrease in or complete cessation of heat removal by these condensers and to instability of the entire system. Therefore, the input chambers of the condensers are integrated by a common-ring collector (with a cross section of about 20 m^2), which has a 1100-mm-wide slot along its bottom.

As a result of the investigations carried out by the Institute of Power Engineering Problems (IPEP) of the National Academy of Sciences (NAS) of Belarus in collaboration with the Russian organizations mentioned above [3-6], the initial data and closing equations for the thermohydrodynamic characteristics of the equipment and shut-down cooling system are obtained, which make it possible to develop a mathematical model and a program to calculate heat and mass transfer by natural and combined convection in a multichannel air cooling system for an analytical study of the influence of weather factors (wind velocity and direction, air temperature) and some design parameters on heat losses in the hot stand-by mode of operation and on the emergency cooling power. When creating an emergency cooling system, it is necessary to minimize heat losses in the hot stand-by mode of operation and to provide sufficient heat power at maximum ambient temperature. This requires the appropriate computer programs.

Mathematical Modeling of Convective Heat Losses in Hot Stand-by Mode of Operation. To evaluate free-convective heat losses in an individual air pit of the passive heat removal system (PHRS) with allowance for air leakage via leaky spots in the gate, the actual construction of the air pit is replaced by a simplified scheme (Fig. 1). The air pit is modeled by two sections: 1) vertical (the joint zone of a square gate and a circular pipe), in which the flow undergoes hydrodynamical restructurization, and 2) descending (the section of inclined pipes of an actual plant), where heat transfer occurs on the pipe ends. The part of the spherical APS dome through which heat transfer to the environment occurs in the radial direction is modeled by a plane rectangular surface with a slope of 45°. A gate with nine plates and different heat-conduction sections is modeled by an equivalent homogeneous system with the effective coefficient of convective heat transfer $h_g = (h_{t1}S_1 + h_{t2}S_2)/(S_1 + S_2)$ and a temperature of the upper surface of $T_t = (T_2S_1 + T_3S_2)/(S_1 + S_2)$. It is assumed that the extra heat power Q_g due to mass overflow via leaky

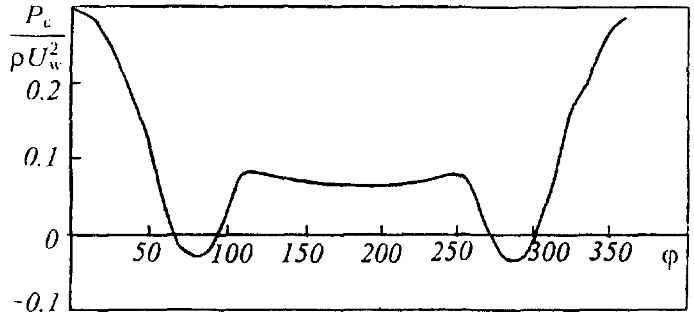

Fig. 2. Pressure drop distribution $P_c/\rho U_w^2$ in the air channel of the PHRS created by near-ground wind.

spots in the gate causes an increase in the temperature at the boundary of the heat transfer zone on the pipe ends and does not change the dimensionless heat transfer relations.

The amount of heat-carrier overflow under free convection conditions in an air pit considerably depends on the extra hydraulic resistance of the upper and lower gates and on the influence of the retarding pressure head under conditions of uniform wind blowing around the PE in the near ground layer of the atmosphere.

With air flowing around the cylindrical part of the protective envelope of a reactor at the velocity U_w, the distribution of the excess retarding head can be calculated by the formula [7, 8]

$$\Delta P_w = \frac{\rho U_w^2}{2} - \frac{1}{2}\rho\,(2U_w \sin\varphi)^2 \tag{1}$$

or approximated by a polynomial dependence [7, 9] obtained experimentally. The mean velocity of free-convective motion also depends on the pressure drop between the inlet slot of the circular collector and the deflector outlet $\Delta P_w = P_{col}(\varphi) - P_d$

$$u = \sqrt{\left(2g\beta\,(T_1 - T_\infty)\,H + \frac{2\Delta P_w}{\rho}\right)}\,. \tag{2}$$

To calculate the flow velocity along the collector channel, it is necessary to take into account the pressure losses that occur on passing through the slot (ΔP^{σ}) and on turning of the flow (ΔP_y), which are calculated by known formulas [10, 11] with due regard for the configuration of the passive cooling system. The pressure at any point at the collector inlet is given by the expression

$$P(\varphi) = P_w(\varphi)\left[0.9 - \left(\frac{1}{\varepsilon} - 1\right)^2\right]. \tag{3}$$

The average pressure with regard for its drop to overcome the hydraulic resistance of the channel walls can be found by integration of its local value, which is dependent on the angle of flow, over the entire collector contour with allowance for hydraulic losses

$$P_{col}(\varphi) = \frac{1}{S}\oint\left[P(\varphi') - \xi_{col}\frac{l_{col}(\varphi,\varphi')}{d_{col}}\frac{\rho U_{col}^2(\varphi,\varphi')}{2}\right]dS\,, \tag{4}$$

where $S = \pi\sigma(2R + \sigma)$ is the inlet slot area; $d_{col} = (2l_{col}L_{col})/(l_{col} + L_{col})$ is the hydraulic diameter; $\xi_{col} = (1.821 \log \mathrm{Re} - 1.64)^{-2}$ is the hydraulic resistance coefficient of the channel for isothermal turbulent flow [12]. Re values are calculated using the hydraulic diameter and physical parameter values at the ambient temperature.

The pressure distribution at the deflector outlet (Fig. 2) can be calculated by the empirical relation [2]

$$P_d = -\,0.665\,\frac{\rho U_w^2}{2}\,. \tag{5}$$

Thus, with all the pressure losses taken into consideration, the equation of momentum conservation for the cooling loop of the air pit in the presence of mass overflow via the upper and lower gates plates can be written in general form as

$$\frac{\rho Q_v^2}{2(\delta a)^2} = \rho g \beta (T_1 - T_\infty) H + \Delta P_w - \frac{1}{12}[\Delta P_{f1} + \Delta P_{f2} + 2\Delta P^{\Delta}_{dif} + 2\Delta P^{\delta}_{dif} + 2\Delta P^{\Delta}_{con} + 2\Delta P^{\delta}_{con} + 4\Delta P_{\gamma}], \quad (6)$$

where ΔP_{f1}, ΔP_{f2}, ΔP^{Δ}_{con}, ΔP^{δ}_{con}, ΔP^{Δ}_{dif}, ΔP^{δ}_{dif}, and ΔP_{γ} are the pressure losses caused by friction forces in the slots of the upper and lower gates, by flow narrowing at the gap inlet between the gate plates and on passing between membranes, by flow broadening at the gap outlet between the gate plates and of the slot between the membranes, and by flow turning, respectively. On deriving Eq. (6) it was assumed that the leakages via the ten main longitudinal slots between the nine plates and the two additional slots on the ends were the same in value.

Substituting the values of all pressure drops calculated by the formulas from [10] into equation (6), we can calculate the mass flow rate of air flowing through the gate slots with allowance for the pressure drop between the inlet slot of the circular collector and the deflector outlet

$$Q_m = \left\{ (288 g \beta (T_1 - T_\infty) H \rho^2 + 288 \Delta P_w \rho) \Big/ \left[\frac{12}{(\delta a)^2} + \frac{2}{(\Delta a)^2} \left(1 - \frac{\delta}{\Delta}\right)^2 + \frac{2}{a^4}\left(1 - \frac{\Delta}{a}\right)^2 + \right. \right.$$

$$\left. \left. + \frac{1}{(\delta a)^2}\left(1 - \frac{\delta}{\Delta}\right) + \frac{1}{(\Delta a)^2}\left(1 - \frac{\Delta}{a}\right) + \frac{0.4}{(\Delta a)^2} + \frac{1}{(\delta a)^2}\frac{l}{d}(\xi_1 + \xi_2) \right] \right\}^{1/2}. \quad (7)$$

The heat power due to mass overflow via the gate plates Q_{lid} is determined by using the air mass flow rates calculated by formula (7), and its values will be different for different positions of the group of pits relative to the wind direction. The slot temperature, which is required for calculation of the mass flow rate and heat input to the boundary of the heat transfer zone on the ends due to leakages, is assumed to be the mean temperature of the upper gate surface without heat insulation T_3 and of the heat exchanger zone T_1. With these assumptions taken into consideration, conjugated heat transfer in this construction will be described by 16 equations with 16 unknown quantities:

$$Q_{tot} = h_g (T_1 - T_t) S_t + Q_g, \quad Q_t = \alpha_1 (T_t - T_4) S_t,$$

$$Q_2 = h_{ven} [T_6 - T_{11}] S_3 = \alpha_2 (T_{ven1} - T_8) S_3 = \alpha_6 (T_6 - T_{11}) S_3,$$

$$Q_3 = \alpha (T_4 - T_\infty),$$

$$Q_4 = h_{ven} [T_{12} - T_{ven2}] S_4 = \alpha_3 (T_{ven2} - T_8) S_4 = \alpha_7 (T_7 - T_{12}) S_4,$$

$$Q_5 = \alpha_4 (T_8 - T_9) S_5 = h_{dom} (T_9 - T_{10}) S_5 = \alpha_5 (T_{10} - T_\infty) S_5,$$

$$Q_{tot} = Q_1 + Q_2 + Q_g,$$

$$Q_1 = Q_g = Q_3 + Q_4, \quad Q_5 = Q_2 + Q_4, \quad Q_g = Q_m c_p \left(\frac{T_1 + T_3}{2} - T_4\right). \quad (8)$$

The convective heat transfer coefficients are determined from the dimensionless equattions for the corresponding surfaces [7-12], and the coefficient of convective heat transfer on the ends is calculated by the criterial relations from [13, 14].

Total heat losses of the PHRS in the hot stand-by mode of operation will be determined by the heat losses in each group of pits. The heat losses of the air pits in the same group are considered to be equal. Thus, for different wind directions, the total heat losses are determined by the following expressions:

$$Q_{\mathrm{I}} = 3Q_{\mathrm{tot},\varphi=0} + 6Q_{\mathrm{tot},\varphi=90} + 3Q_{\mathrm{tot},\varphi=180}\,, \quad Q_{\mathrm{II}} = 6Q_{\mathrm{tot},\varphi=45} + 6Q_{\mathrm{tot},\varphi=135}\,, \tag{9}$$

where Q_{I} and Q_{II} are the heat losses at angles of attack of 0 and 45° relative to the group of condensers, respectively; $Q_{\mathrm{tot}\varphi}$ the heat loss of an individual air pit at the point characterized by the angle of flow φ reckoned from the direction of wind.

Convective Heat Removal in Condensers of a Multichannel Air Cooling System under Emergency Conditions. The heat power dissipated by a passive safety system under emergency conditions is determined by the dependence of the heat power released by the condensers on the mass flow rate of air and the temperature difference. According to the designs of the PHRS condensers at the IPEP of the NAS of Belarus, this dependence is as follows:

$$Q = (0.24207 + 0.729446G)\,(550.98 - T_{\infty})\,. \tag{10}$$

Thus, calculation of the active regime reduces to determination of the mass flow rate for different groups of pits under different weather conditions.

To calculate the air flow rate in a single pit, we use the equation of motion in the form

$$\frac{\rho u^2}{2} = \rho g \beta \Delta T H + \Delta P_{\mathrm{w}} - \sum_i \Delta P_i\,, \tag{11}$$

where $\sum_i \Delta P_i$ are the pressure drops due to the hydraulic resistances of the channel of the air pit. On using Eq. (11), the following hydraulic resistances are implied: of flow turning at the collector inlet, of the collector slot, of flow narrowing in the zone from the collector to the condenser, of the turn by an angle of $\gamma = 45^{\circ}$ in the pit channel, of the friction of the pit channel, and of the condenser.

Proceeding from the experimental data of [5], we neglect the change in rarefaction in the deflector on passing of the system from the hot stand-by mode of operation to the emergency mode. To model the operation of the system in the emergency mode, we use the pressure drop values calculated for the hot stand-by mode of operation.

To calculate the hydraulic resistances, we used the standard relations [15] and an equation derived at the IPEP of the NAS of Belarus from the design of the PHRS condensers [5]:

$$\Delta P_{\mathrm{cond}} = [0.05844 + 1.0859 \cdot 10^{-4}\,(T_{\infty} - 273.16)]\,G^{1.92}\,. \tag{12}$$

Determination of the decisive temperatures entering the formulas for calculation of the hydraulic resistances required for solution of the conjugated heat transfer problem for an air pit and an APS dome, which was carried out by an iteration technique similar to the method adopted in calculation of the hot stand-by mode of operation.

Method and Algorithms of the Program for Heat Loss Calculation in PHRS Condensers. Using the analogy between heat conduction and electrical conduction we can give an equivalent circuit diagram for the present problem [16] (Fig. 3), where the electrical resistances correspond to the given thermal resistances, the potentials correspond to the temperatures, and the currents are appropriate for the heat powers delivered. Resistances R_{1-R_6} are determined by the following relations

$$R_1 = \frac{1}{h_{\mathrm{g}}\,S_{\mathrm{t}}} + \frac{1}{2\alpha_1\,S_{\mathrm{t}}}\,; \quad R_2 = \frac{1}{h_{\mathrm{ven}}S_3} + \frac{1}{\alpha_2 S_3} + \frac{1}{\alpha_6 S_3}\,; \quad R_3 = \frac{1}{2\alpha_1\,S_{\mathrm{t}}}\,;$$
$$R_4 = \frac{1}{2\mathrm{Nu}\,\lambda D}\,; \quad R_5 = \frac{1}{h_{\mathrm{ven}}S_4} + \frac{1}{\alpha_3 S_4} + \frac{1}{\alpha_7 S_4}\,; \quad R_6 = \frac{1}{h_{\mathrm{dom}}S_5} + \frac{1}{\alpha_4 S_5} + \frac{1}{\alpha_5 S_5}\,. \tag{13}$$

The heat flux Q_{tot} is the sought quantity in this calculation, but for determination of all temperatures entering the system of equations (8) it is necessary to determine heat fluxes Q_1, Q_2, Q_3, and Q_4. Performing twice the star-to-delta conversion, which is well known in electrical engineering, one can obtain a simplified circuit, proceeding from which Q_{tot} can be represented by the following expression:

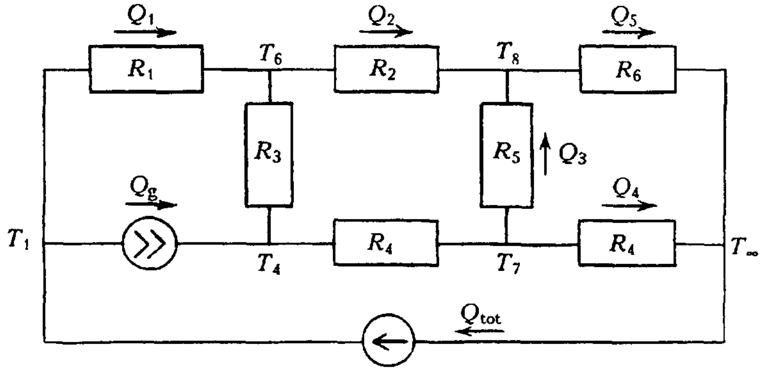

Fig. 3. Equivalent circuit for calculation of conjugated heat transfer in an air

TABLE 1. Total Heat Losses of PHRS in Hot Stand-by Mode of Operation

U m/sec	T, °C						T, °C					
	−40			+30			−40			+30		
	δ, mm						δ, mm					
	0.5	1	2	0.5	1	2	0.5	1	2	0.5	1	2
	Q_I, kW						Q_{II}, kW					
5	899.6	1197.9	1607.2	659.5	919.5	1285.6	899.7	1198.0	1607.3	659.6	919.7	1285.8
10	905.2	1206.6	1620.0	665.0	928.3	1299.0	905.6	1207.2	1620.9	665.4	929.0	1300.0
20	925.3	1238.0	1666.8	684.4	960.0	1347.4	927.7	1241.8	1672.6	687.0	964.3	1354.1
30	954.9	1284.4	1735.9	712.5	1005.7	1417.2	962.2	1296.0	1753.3	720.5	1018.9	1437.7

$$Q_{\mathrm{tot}} = \frac{(T_1 - T_\infty) + Q_g R_{16}}{R_{15} + R_{16}}, \tag{14}$$

where resistances R_{15} and R_{16} are determined in terms of the initial resistances $R_1 - R_6$ by the known conversion rules [17].

Since resistances $R_1 - R_6$ are variables that are dependent on the temperatures T_t, T_{ven1}, T_{ven2}, T_∞, $T_4 - T_{12}$, the problem stated is solved by the method of iterations over arbitrarily prescribed initial temperatures.

In calculation, the initial data corresponding to the PHRS were used, and the thermophysical characteristics of air were considered to be functions of the mean temperature of each zone and approximated within a temperature range of 233–670 K by polynomials to the seventh degree.

Data from the handbook [7] were used to construct the approximation dependence.

Calculated Results and Their Analysis. Depending on external metereological conditions (ambient temperature, wind velocity and directions) and the slot sizes in the gate in the hot stand-by mode of operation, the total heat losses of the system are from 659.5 to 1753.3 kW, and their value depends mainly on the slot size (Table 1). The increase in the total heat losses with a change in slot size of from 0.5 to 1 mm is 35–40%, and with a change in slot size of from 0.5 to 2 mm, it is 80–100% in relation to weather conditions.

The influence of wind velocity on total heat losses is less pronounced. Its increase from 5 to 30 m/sec results in an increase of 6–9 and 8–12%, depending on slot size, at an ambient temperature of −40°C and +30°C, respectively. A change in the wind direction (angle of attack) of from 0 to 45° gives an insignificant increase in heat losses of about 1–1.5%.

The angle of attack exerts a more pronounced influence on the distribution of heat losses relative to different groups of air pits (Fig. 4). At an angle of attack of 0°, the difference in heat losses is especially noticeable between the pits located at the points $\varphi = 90^\circ$ and $\varphi = 0^\circ$. While at small wind velocities ($U = 5-10$ m/sec) it is no more than 4.5%, at U =30 m/sec the heat losses in the pit located at the head point ($\varphi = 0^\circ$) is larger by 20–33% as compared to the pit located at the point $\varphi = 90^\circ$ in relation to slot size. At an angle of attack of 45°, the difference

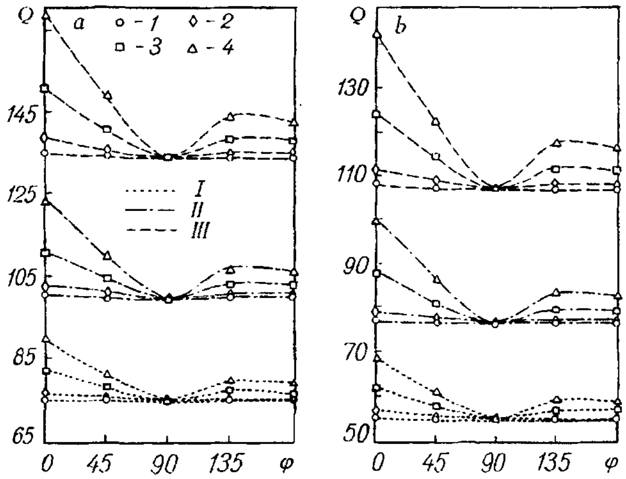

Fig. 4. Heat losses Q in an individual pit as a function of the slot size δ, wind velocity U_w, and position in relation to the wind direction φ at the ambient temperature $T_\infty = -40^\circ C$ (a) and $+30^\circ C$ (b); I) $\delta = 0.5$ mm; II) 1; III) 2 mm; 1) $U_w = 5$ m/sec; 2) 10; 3) 20; 4) 30. Q, kW; φ, deg.

TABLE 2. Total Heat Power Dissipated by PHRS under Emergency Cooling Conditions

U, m/sec	T, °C			
	−40	+30	−40	+30
	Q_I, kW		Q_{II}, kW	
5	146.77	81.46	146.80	81.48
10	148.37	82.66	148.49	82.75
20	154.36	87.05	155.07	87.65
30	163.31	93.44	165.47	95.26

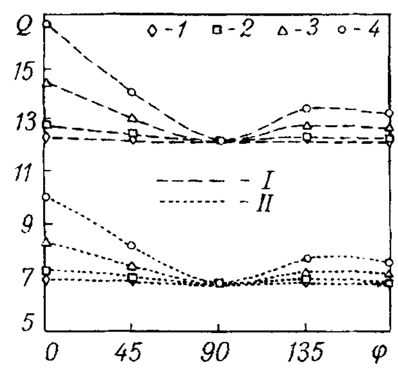

Fig. 5. Heat power Q dissipated by an individual pit as a function of the wind velocity U_w, position in relation to the wind direction φ, and ambient temperature T_∞: I) $T_\infty = -40^\circ C$; II) $+30^\circ C$; 1) $U_w = 5$ m/sec; 2) 10; 3) 20; 4) 30. Q , MW.

in heat losses in different groups of pits is considerably smaller and does not exceed 4.5%, regardless of the slot size and external meteorological conditions.

Under emergency cooling conditions, the total heat power dissipated by the PHRS considerably depends on weather conditions and is 81.5–95.3 MW at an ambient temperature of $+30^\circ C$ and 146.8–165.5 MW at a temperature of $-40^\circ C$ (Table 2). With increasing temperature in the mentioned range, it increases by ~75% and practically does not depend on the angle of attack (the difference is ~0.03–0.38%) or the wind velocity (the difference is ~1.5–1.9%). As in the hot stand-by mode of operation, wind velocity influences the dissipated power to a greater extent than the ambient temperature does. An increase in wind velocity of from 5 to 30 m/sec entails an increase in the total dissipated power by 11.3–14.7% at an angle of attack of 0° and by 12.7–16.9% at an angle of attack of 45°.

Thus, the angle of attack under emergency cooling conditions exerts a more substantial influence on the total dissipated power as compared to the hot stand-by mode of operation. This is particularly noticeable for some groups of pits (Fig. 5). Thus, at an angle of attack of 0° the difference between the pits with minimum and maximum dissipated powers can reach 48%, while under the same conditions and an angle of attack of 45° this difference is 13.3%.

The calculations revealed a substantial influence of weather conditions on the operation of the PHRS in the hot stand-by mode of operation and in the case of emergency cooling, when the difference in dissipated powers can attain 100%.

CONCLUSIONS

1. As a result of calculations, the heat losses in the hot stand-by mode of operation and the maximum heat power dissipated under emergency cooling conditions are determined. The total heat losses of the system in the hot stand-by mode of operation are found to be from 659.5 to 1753.3 kW. The heat losses are mainly affected by the size of the slots between the plates of the closed gates. The next, in importance, factor is the ambient temperature. When it decreases from +30 to -40°C, the total heat losses increase by 22–36% in relation to the wind velocity. A comparative analysis of the influence of the wind velocity and its direction on the heat loss distribution for different air pits has shown the stronger influence of the angle of attack of the external air flow.

2. In the regime of emergency cooling the maximum total heat power dissipated by the PHRS considerably depends on weather conditions and is 81.5–95.3 MW at an ambient temperature of $+30^{\circ}$C and 146.8–165.5 MW at a temperature of -40°C. The increase in the total heat power with decreasing temperature in the indicated range is ~75% and is practically independent of wind direction. Wind velocity exerts a less pronounced influence, as compared to the ambient temperature, on the dissipated power.

3. In the hot stand-by mode of operation the heat losses are less than 0.05% of the heat power of the reactor, but in absolute value they are rather significant. Therefore, further improvement of the gate design with allowance for possible thermal deformations, manufacture errors, and so on is advisable for minimization of air leakage.

In the emergency regime the slope of the gate plates and the air flow rate via the condensers are changed automatically with the purpose of maintaining the pressure values in the steam generators close to the nominal value irrespective of the environment parameters.

In an analysis and calculation of the stability of the emergency cooling system of a reactor it is necessary to take into consideration the difference in the power of the steam generators and the nonsynchronous operation of the gates over the PE perimeter.

NOTATION

T_1, T_4, T_6, T_7, T_8, T_∞, mean temperatures, respectively, of the heat exchanger zone, of the beginning of the zone of heat transfer on the ends of the joint zone, of the heat transfer section on the ends, of the gap between the ventillating pipe and the dome, of the surrounding medium; T_t, α_1, S_t; T_{ven1}, α_2, S_3; T_{ven2}, α_3, S_4; T_9, α_4, S_5; T_{10}, α_5, S_5; T_{11}, α_6, S_3; T_{12}, α_7, S_4, temperature, convective heat transfer coefficient, and area, respectively, of the upper gate surface, of the outside surface of the joint zone, of the outside surface of the ventillating pipe over the heat transfer on the ends, of the inside surface of the heat-transfer section of the dome, of the outside surface of the heat-transfer section of the dome, of the inside surface of the joint zone, of the inside surface of the ventillating pipe over the heat-transfer section on the ends; U_w, wind velocity; h_g, h_{ven}, h_{dom}, convective heat transfer coefficients of the gate, of the ventillating pipe walls, and the dome, respectively; α, convective heat transfer coefficient of the section of face heat transfer; T_2, S_1, h_{t1}, temperature of the upper surface, area, and heat transfer coefficient of the gate sections with heat insulation, respectively; T_3, S_2, h_{t2}, temperature of the upper surface area and heat transfer coefficient of the gate sections without heat insulation, respectively; Q_g, heat power generated by mass overflow via leaky spots in the gate $P_{col}(\varphi)$, averaged pressure in the collector channel; Q_v, volume flow rate of air via the gate plates; P_d, pressure at the deflector outlet; H, height difference between the surface of the top gate and the open end of the air pit; a, plate length; Δ, gap width between the gate plates; δ, slot width between membranes of the gate plates; σ, inlet slot width of the collector; Q_{tot}, total heat losses of an individual air pit; D, diameter of the ventillating pipe; L_{col}, R, height and curvature radius of the collector channel; l_{col}, coordinate along the collector channel; λ, ρ, u, thermal conductivity, mean density and mean velocity of air in the pit channel; ε, parameter accounting for the system configuration; c_p, heat capacity of air at a constant pressure; ξ_1 and ξ_2, friction loss coefficients; ΔT, temperature drop between the condenser outlet and the surrounding medium; ΔP_w, pressure drop due to wind between the collector and the deflector of the PHRS ; ΔP_i, pressure drops due to friction losses in the PHRS channel; Q, heat power of the PHRS collector; G, mass flow rate of air via the condenser; Nu, Nusselt number characterizing heat transfer between air layers in an air pit. Subscripts: w, wind; g, gate; c, channel; d, deflector; i, current index; ∞, initial parameters; f, friction loss; m, mass flow rate; v, volumetric flow rate; cond, condenser; tot, total; γ, angle of flow turn; σ, inlet slot of the collector; φ, angle characterizing the position of air pit relative to wind direction; t, upper surface of gate; ven, ventillating pipe surface; dif, diffusor; con, confusor; col, collector; dom, dome.

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