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Effect of the Fill Ventilation Window on Performance of a Natural Draft Cooling Tower Subjected to Cross-Winds

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Abstract. Various aerodynamic design elements and technics (wind deflectors, wind walls, etc.) are utilized for improvement of the thermal efficiency of the natural draft cooling towers, particularly in conditions of cross wind. One of the technical methods, proposed by engineers of Belarus Academy of Sciences, is installation of the ventilation window in the center of the fill. This method is substantiated by the fact that the flow of cooling gas obtains maximum temperature and humidity near the center of the under-fill space of cooling tower and, as a consequence, performs minimal heat exchange. The influence of the fill ventilation window and wind deflectors in the inlet windows of the cooling tower on its thermal performance in condition of cross-wind is investigated in the paper numerically. The cooling tower of the “Woo-Jin” power plant (China) 150 m of the height and 114 m of the base diameter was taken as a prototype. The analogy (equivalence) between the heat and mass transfer was taken into consideration, which enabled us to consider single-phase flow and perform complicated 3D simulation by using modern personal computers. Heat transfer coefficient for the fill and its hydrodynamic resistance were defined by using actual data on total flow rate in the cooling tower. The numerical model and computational methods were tested and verified in numerous previous works. The non-linear dependence of the thermal performance of the cooling tower on wind velocity (with the minimum in vicinity of $U_{cr} \sim 8$ m/s for the simulated system) was demonstrated. Calculations show that in the condition of the average wind speed the fill ventilation window doesn’t improve, but slightly decrease (by 3–7 %) performance of the cooling tower. Situation changes in the condition of strong winds $U_{cw} > 12$ m/s, which are not typical for Belarus. Utilization of airflow deflectors at the inlet windows of cooling tower, conversely, increases thermal performance of the cooling tower on 10–12 % for the wide range of winds exceeding 2 m/s. Results of the investigation may be helpful for optimal design of natural draft and perspective “hybrid” cooling towers.

Keywords: cooling tower, numerical simulation, thermal efficiency, cross wind

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Влияние вентиляционного отверстия в оросителе на тепловую работу башенной испарительной градирни в условиях ветра

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Реферат. Для улучшения тепловой работы башенных испарительных градирен, особенно в условиях ветровых нагрузок, могут использоваться разнообразные аэродинамические элементы и конструктивные решения (дефлекторы, ветроперегородки и т. п.). Одним из предложений, впервые сформулированных сотрудниками Национальной академии наук Беларуси, было сооружение центрального вентиляционного отверстия в оросителе градирни. Данное конструктивное решение обосновывалось тем, что паровоздушный поток вблизи центра подоросительного пространства градирни имеет максимум температуры и влагонасыщенности и вследствие этого слабо участвует в теплообмене. В статье численно исследуется влияние центрального вентиляционного отверстия в оросителе совместно с ветронаправляющими дефлекторами во входном окне градирни на ее тепловую работу в условиях ветра. В качестве прототипа взята градирня ТЭС «Ву-Джин», Китай, высотой 150 м и диаметром основания 114 м. При расчетах используется аналогия тепло- и массопереноса, что позволяет моделировать аэродинамику однофазного потока и выполнять детальные трехмерные расчеты на современных персональных вычислительных машинах. Коэффициент теплоотдачи оросителя и его гидродинамическое сопротивление устанавливаются в соответствии с экспериментальными данными по общему расходу воздуха в градирне. Численная модель протестирована и верифицирована в ряде предыдущих исследований. Продemonстрирована нелинейная зависимость тепловой производительности БИГ от скорости ветра с минимумом при $U_{cr} \sim 8$ м/с для моделируемой системы. Расчет показал, что при средних скоростях ветра устройство центрального отверстия не увеличивает, а наоборот, несколько уменьшает (на 3–7 %) эффективность работы градирни. Ситуация изменяется при очень сильных ветрах, превышающих 12 м/с, не характерных для условий Беларуси. Использование ветронаправляющих дефлекторов во входном окне градирни, напротив, увеличивает производительность градирни на 10–12 % в широком диапазоне скоростей ветра, превышающих 2 м/с. Результаты исследования могут быть полезны для оптимизации конструкции градирен, в том числе и перспективных градирен гибридного типа.

Ключевые слова: градирня, математическое моделирование, тепловая эффективность, ветровой поток

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Natural draft cooling tower (NDCT) is the important element of the recirculation water cooling system for Condensation Power Plants and Nuclear Power Plants. Both processes of heat and mass transfer (latent heat of evaporation) play essential role for water cooling. Hot water is sprayed over the fill – the highly porous system with expanded surface area, – and flowing down it exchanges heat with opposite gas flow. Meanwhile some fraction of water evaporates and humidity of the gaseous flow approaches 100 %. Latent heat of evaporation reduces internal energy (and temperature) of the water. Following that, water drips down to the pool and directed back to condenser (Fig. 1). Total flow rate of the water may be as much as 100 thousand tones per hour at modern power plants and up to 2 % of that may be evaporated and released to atmosphere.

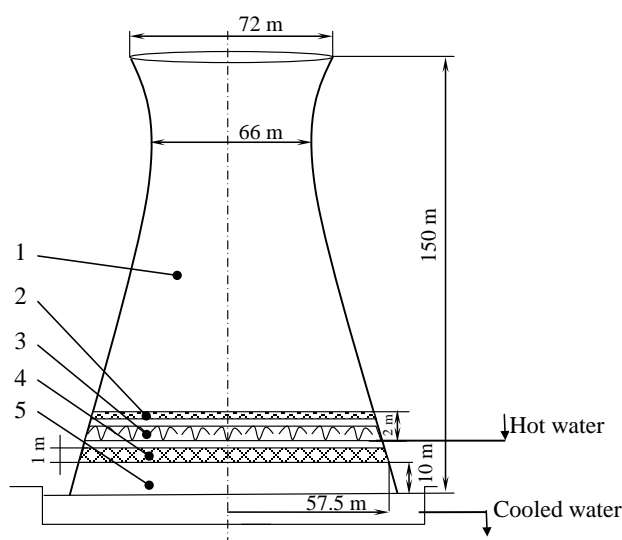


Fig. 1. Natural draft cooling tower scheme: 1 – overshower zone; 2 – drift eliminators; 3 – sprinklers; 4 – fill; 5 – rain zone

Convection in NDCT is defined with internal heat and mass transfer processes as well as with external meteorological conditions: wind speed, temperature, humidity. The effect of the wind is of complicated nature. The total air flow rate and NDCT efficiency is insignificantly effected by weak winds, $U_{cw} \leq 1\text{--}2$ m/s. With wind speed growth the total air flow rate decreases due to throttling of the NDCT outflow face with vortices and throughout flows in the rain zone. Decrease of the NDCT performance continues until the wind speed reaches its “critical” value U_{cr} (within the range from 4 to 10 m/s) and with the further growth of the wind velocity NDCT performance grows.

Numerous papers dedicated to investigation of the wind speed influence on NDCT performance and development of methods of performance improvement in windy conditions [1–9]. D. Derksen and T. Bender [3–5] investigated NDCT performance in windy conditions in their experiments in wind tunnel and by numerical simulation. Among the methods of NDCT performance improvement is installation of wind walls in the rain zone and wind deflectors in the inlet windows of NDCT [6–8]. The mentioned technical solutions were investigated experimentally [6, 8] and by means of CFD tools [7].

Note that experimental study of the innovative designs with the bench scale models is not adequate because of impossibility to satisfy all necessary similarity criterions of the system. Experimentation on real NDCT is very expensive and it is complicated with instability of the ambience meteorological factors. Analytical investigations are limited with simplified and schematic problem statements. State-of-the-art computational experiment let one perform intensive parametric study of NDCT with high reliability and specification of the system [9–12].

Numerous methods for improvement of NDCT performance are discussed in the literature. Among them variation of the sprinkling density, variation of the

design and height of fill, aerodynamic control of the flow, onset of the big scale vortex inside NDCT [13] etc. It is shown in [11] that flow rotation in the over-shower zone intensifies heat and mass transfer in NDCT and shifts condensation zone to periphery of the shell. Technical solutions for rotation onset are discussed in [14]. The paper [15] is dedicated to numerical investigation of influence of wind walls and inlet window deflectors on NDCT performance in conditions of the side wind.

Patent [16] describes technical proposal of a ventilation window creation in the center of fill for the aim of reduction of total aerodynamic resistance of the NDCT, improvement of the rain zone performance, and possible creation of a vortex movement in the central part of a NDCT shell. Experimental data obtained on the bench-scale model confirm bell-type profile of temperature and humidity in the rain zone which substantiates the idea. Basing on the patent, modernization of the Grodno Power Plant-2 was implemented and positive effect obtained. To note, together with the fill ventilation window, inlet window deflectors were installed and this fact makes it impossible to define a specific separate effect of the central window on the NDCT performance.

In this paper investigation of the NDCT equipped with the fill ventilation window as well as with inlet window deflectors was performed numerically. 3D modeling with standard CFD tools [17] was tested and verified in previous works by the author. Total gas mass flow rate G was calculated as a function of the wind speed for different diameters of the fill ventilation window. The performance efficiency parameter η , directly defined via NDCT input and output temperatures T_{wi} , T_{wo} and “wet bulb” temperature was also estimated. Simulation results are discussed. The obtained data may be utilised for optimization of working NDCTs and construction of new ones.

Problem statement

The counter flow natural draft wet cooling tower of Woo-Jin Power Plant, Shanghai, China was selected as a prototype of the simulated system. Its height is 150 m, basement diameter is 114 m and inlet window height – 10 m. Computational domain was a cylinder with the radius and height equal to 500 m (Fig. 2). The domain consisted of 1.8 million structured and unstructured (hybrid) mesh elements. The mesh size was enough for reliable convergence of iterations and it was kept constant for all cases under investigation. The mesh element size was smoothly stretched from NDCT walls to periphery to resolve the high gradient regions.

We considered NDCT equipped with deflectors in inlet windows and ventilation window in fill (Fig 3). Diameter of the ventilation window was varied $D_0 = 2, 4, 8$ и 12 m. Inlet windows deflectors were set at an angle of $\alpha = 72^\circ$ to tangent line, according to recommendations [18]. A standard NDCT not equipped with the mentioned elements (Fig. 3b) was simulated as a reference case.

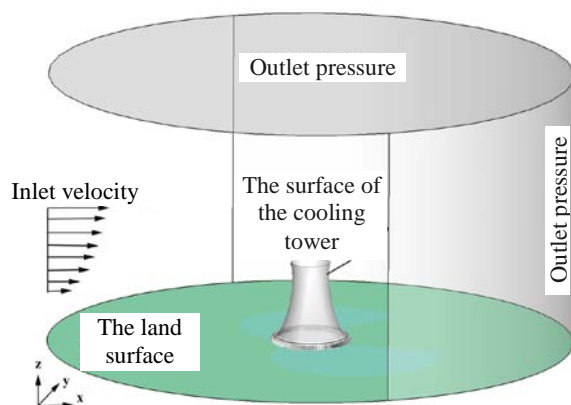


Fig. 2. Computational domain and boundary conditions

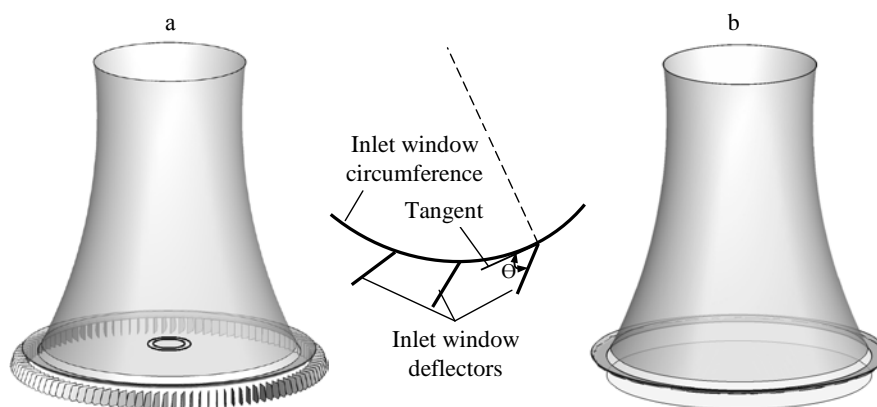


Fig. 3. Scheme of NDCT equipped with deflectors in inlet windows and ventilation window in fill (a) and a standard NDCT (b)

The wind speed vertical distribution was simulated according to the formula [19]

$$\frac{U_{CW}}{U_{ref}} = \left(\frac{z}{z_{ref}} \right)^{0.2}. \quad (1)$$

The reference value of velocity U_{ref} corresponds to the velocity at the height of 10 m over the land [19]. It is used as an independent parameter for graphs. Simulations were performed for the U_{ref} values laying in the range from 0 to 12 m/s.

The fill of the NDCT together with complimenting elements such as shell supports, spray piping network, fill supports and drift eliminators was simulated by using the Fluent^(R) model of “Radiator”. The temperature assigned to the “radiator” was fixed. The radiator is characterized with two fundamental parame-

ters, – pressure loss coefficient k_f and heat transfer coefficient h_{th} . The pressure gap on “radiator” was expressed in terms of the pressure loss coefficient, air density, and perpendicular velocity component across the surface [20]

$$\Delta p = k_f \frac{1}{2} \rho v^2. \quad (2)$$

The effective pressure loss coefficients of the tower shell supports, the spray piping network, the fill supports, and the drift eliminators were set to $k_{tsh} = 0.5$, $k_{spn} = 0.5$, $k_{fs} = 0.5$, $k_{de} = 3.0$ correspondingly, according to D. Kröger [21]. The total pressure loss coefficient was obtained as a sum of the particular coefficients $k_f = k_{tsh} + k_{spn} + k_{fs} + k_{de} = 4.5$. The value of the heat transfer coefficient for the “radiator” was defined by tuning of h_{th} to the experimentally measured value of the total air mass flow rate in [22].

The rain zone of NDCT was simulated as a highly porous media by Fluent^(R) tools. The porosity value was determined according to estimated volume of droplets in the zone, the average velocity of droplets and total volume of the sprinkled water being known and comprised $m = 0.998$ for the standard parameters of the task. The pressure loss in the rain zone was calculated according to internal procedures of the porous media module of Fluent^(R).

Mathematical Model

In accordance with the stated problem, gas flow was simulated within steady-state Eulerian approach [17]. A single-phase flow is considered, standard k – ε -model of turbulence is adopted. The governing equations for the problem are the Navier – Stokes equations for an incompressible viscous fluid flow, continuity and energy conservation equations:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \mathbf{v}) = 0; \quad (3)$$

$$\frac{\partial}{\partial t}(\rho \mathbf{v}) + \nabla(\rho \mathbf{v} \mathbf{v}) = -\nabla p + \nabla \bar{\tau} + \rho \mathbf{g}; \quad (4)$$

$$\bar{\tau} = \mu \left[\left(\nabla \mathbf{v} + \nabla \mathbf{v}^T \right) - \frac{2}{3} \nabla \mathbf{v} \mathbf{I} \right]; \quad (5)$$

$$\frac{\partial(\rho E)}{\partial t} + \nabla(\mathbf{v}(\rho E + p)) = 0. \quad (6)$$

Here \mathbf{I} – unit tensor, the second term in the right hand part of (5) represents volumetric viscosity effect; E – internal energy; p – pressure.

Note that the model doesn't consider latent heat of phase transformations. To take into consideration latent heat of phase transformation we accept hy-

pothesis of heat and mass transfer equivalence (analogy), formally written in a following form

$$Q^{th} + Q^{lat} = Q^{tot} = \kappa Q^{th}. \quad (7)$$

Here Q^{th} – convection and conduction heat flow; Q^{lat} – heat flow defined by phase transformation (evaporation); κ – constant coefficient.

The equivalence hypothesis is commonly used in the similar cases if evaporation rate is low (the system is far from boiling parameters). The criterion of applicability of the hypothesis may be written as follows [23]

$$b = \frac{j_{ci}}{\rho \omega_0} \frac{1}{St_0} < 0.1, \quad (8)$$

where j_{ci} – mass flow rate normal to the phase transformation surface; ρ – liquid density; ω_0 – gas flow velocity far from surface (out of boundary layer); St_0 – Stanton number calculated ignoring mass transfer.

Estimates show that $b \approx 10^{-3} \ll 0.1$ for the considered case, which justify application of the named hypothesis. The constant parameter in (7) is estimated as $\kappa = 2.86$ basing on the data [24].

The influence of the variable humidity on the density of the gaseous flow is neglected. The boundary conditions are applied at the boundaries of the computational domain (Fig. 2) and at solid surfaces of the system. The wind velocity component collinear to wind direction is set according to (1) at the inlet part the boundary.

Air temperature at inlet is set to $T_a = 297.15$ K. The pressure outlet boundary condition is used to define the air static pressure at the domain outlets (top and right-hand side). In calm conditions $U_{ref} = 0$ m/s the boundary conditions are defined by pressure.

B.C. at the entrance surface:

$$v_x|_{inlet} = U_{cw}; \quad v_y|_{inlet} = 0; \quad v_z|_{inlet} = 0; \quad T|_{inlet} = T_{amb} = 297.15 \text{ K}. \quad (9)$$

At the top surface ($H = 500$ m):

$$\frac{\partial p}{\partial z} = 0; \quad \frac{\partial v_x}{\partial z} = 0; \quad \frac{\partial v_y}{\partial z} = 0; \quad T = T_{amb} = 297.15 \text{ K}. \quad (10)$$

At the exit surface of computational domain:

$$\frac{\partial p}{\partial z} = 0; \quad \frac{\partial v_z}{\partial z} = 0; \quad T = T_{amb} = 297.5 \text{ K}. \quad (11)$$

For the solid surfaces (land, NDCT shell, deflectors) no-slip conditions are applied. Described model was used in [11, 15, 18] and was verified basing on the experimental data on total air flow rate for Minsk Power Plant-4 [25].

Results and discussion

Thermal efficiency of NDCT may be characterized with the value of the total mass flow rate of the cooling air G , as far as G is defined by the heat transferred to cooling air. Functional dependence of $G = G(Q^{tot})$ is not linear in general case. Nevertheless parameter G is a suitable parameter for parametric study and comparison of different regimes of NDCT thermal work.

Total mass flow rate of the cooling air was calculated as a function of the wind speed for the cases of ventilation windows radiuses $R = 2, 4, 6, 8, 10, 12$ m and for the case of absence of ventilation window (Fig. 4). Corresponding area of the window related to total area of fill is:

$$S^0/S = 1.23 \cdot 10^{-3}; 4.9 \cdot 10^{-3}; 1.11 \cdot 10^{-2}; 1.97 \cdot 10^{-2}; 3.1 \cdot 10^{-2} \text{ and } 4.4 \cdot 10^{-2}.$$

The case marked as “standard” on the graph corresponds to the simplest (without ventilation window and deflectors) design of the NDCT.

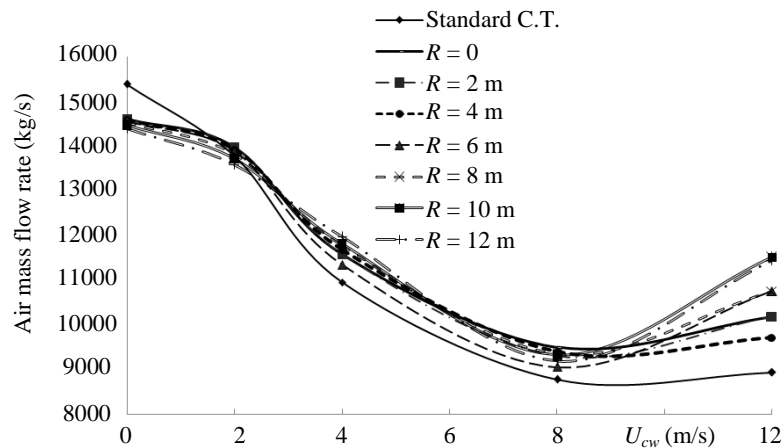


Fig. 4. Total mass flow rate of cooling air G as a function of wind speed U_{cw} . Radius of the ventilation window is presented on the graph field. Standard C.T. – standard design of NDCT

Computations show that input window deflectors improve thermal performance of NDCT at any practical wind condition as far as wind calm $U_{cw} < 1$ m/s is very rare case in Belarus. The effect of decrease of the total mass flow rate of cooling air at calm is explained with some aerodynamic resistance of the inlet window deflectors. Deflectors provide increase of total mass flow rate of cooling air by approximately 10 % for the typical wind speed, and more in the case of strong air.

Note that the cases with small radius of the ventilation window ($R = 2.4$ m) are quite similar to the case of absence of window. The differences in the total mass flow rates for the mentioned cases are of the order of computational mistakes and time averaging mistakes. More profound effect of the central window one can observe at stronger wind speed $U_{cw} \sim 10$ m/s and more. In this condition

relatively better performance is ensured by bigger windows with area $S^0/S \sim 3\%$ and more.

The commonly used parameter for characterization of NDCT performance is efficiency parameter η , defined via the temperatures of a water at inlet and outlet of the NDCT – T_{wi} and T_{wo} , and the temperature of a wet bulb T_{wb} [24]

$$\eta = \frac{T_{wi} - T_{wo}}{T_{wi} - T_{wb}}. \quad (12)$$

Parameter η characterizes the final aim functionality of cooling tower – temperature reduction of the circulating water taking into consideration physical limitations on performance efficiency connected with ambient air temperature and humidity.

Simulation of the efficiency parameter η is presented in the Fig. 5. The temperature of the wet bulb was considered a constant for all computational cases.

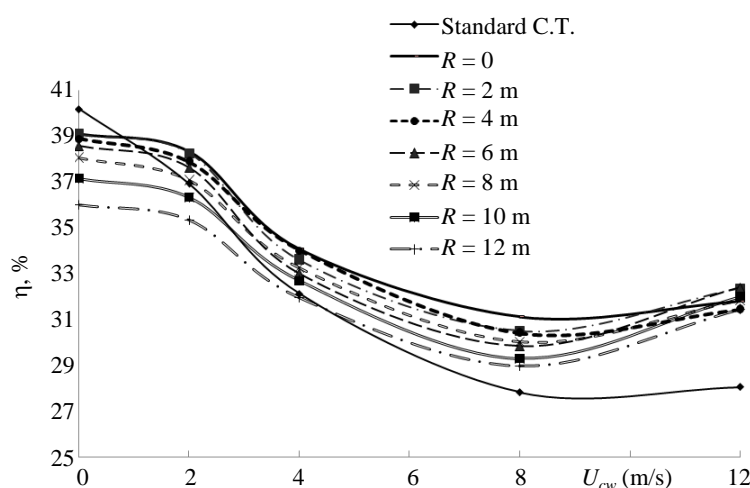


Fig. 5. The efficiency parameter η as a function of a cross-wind speed U_{cw} . Radius of the ventilation window is presented on the graph field. Standard C.T. – standard design of NDCT

The graphs $\eta(U_{cw})$ presented on Fig. 5 let one quantitatively characterize an influence of the ventilation window on the NDCT performance. Computations show that at typical for the central Europe region wind conditions installation of the ventilation window in NDCT fill doesn't improve, but rather impede general thermal performance of NDCT. Situation changes for very strong winds exceeding 12 m/s.

Note that humidity spatial distribution, which may give some advantage to the fill ventilation window technology, was not considered in the model. Taking into account that 1–2 % variability of the relative humidity (order of 50 Pa absolute humidity) may result in only 0.2–0.3 K variability of the recirculation water, general results and conclusions of the modeling would be generally the same.

CONCLUSIONS

Thus, creation of the ventilation windows in fill is not rational for considered in the paper big scale NDCT. The task may be further approached for smaller scale cooling towers and cooling towers operating in conditions of strong cross winds. But for the later case installation of the wind walls, and special inlet window deflectors, NDCT mouth directors etc. should be considered. Creation of the big scale vortex by using oriented sprinkler nozzles [14] also remains one of the perspective technical solution for NDCT performance improvement at average wind velocity conditions.

In view of development of the NDCT simulation program it would be interesting to take into consideration humidity distribution in the system.

Nomenclature

G_a – air mass flow rate, kg/s; $G_{a,0}$ – mass flow rate of air in windless condition, kg/s; g – gravity acceleration, m/s²; H – height of inlet window of NDCT, m; j_{ci} – mass flow rate normal to interface surface, kg/m²/s; κ – constant parameter; m – porosity; p – pressure, Pa; Q – heat power, kW; Q^h – heat power provided by convective heat transfer, kW; Q^{lat} – latent heat power provided by evaporation of water, kW; St_0 – Stanton number in the absence of mass transfer; S – total fill area, m²; S^0 – fill ventilation window area, m²; T – temperature, K; $U_{c,w}$ – cross-wind velocity, m/s; U_{ref} – wind velocity reference speed for (1), m/s; U_{cr} – critical velocity of cross wind, m/s; v – скорость, м/с. **Greek Symbols:** μ – dynamic viscosity, Pa·s; η – NDCT efficiency parameter; ρ – density, kg/m³; ω_0 – stream velocity out of boundary layer, m/s. **Subscripts:** ave – average value; a – air; tot – total or integral value; i – initial value; w – water; o – outlet face value; cw – cross wind.

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