EFFECT OF THE PHYSICAL PROPERTIES OF A WALL ON HEAT TRANSFER IN TURBULENT NATURAL CONVECTION. 2. ANALYTICAL STUDY

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The experimental heat transfer coefficient is calculated from data on the structure of the turbulent boundary layer and assumptions on heat-transfer mechanism.

In a rigorous approach to the solution of the problem of heat transfer in the turbulent flow of a liquid near a vertical wall the coupling between the flow and the heat transfer must be taken into account. Indeed, because of the significant space–time inhomogeneities of turbulent flow, the thermal boundary conditions at each point of the surface are time dependent and one cannot assume steady boundary conditions (Dirichlet and Neumann conditions), as considered in numerous papers [1, 2, 3] on turbulent boundary layers. Turbulent flow near a wall (such as in natural convection) cannot be treated mathematically in a rigorous way. Hence the effect of the wall on the temperature of the fluid, as well as the effect of the flow on the wall, cannot be analyzed rigorously by solving the unsteady equations of motion for the liquid and the heat equation for the wall. On the other hand, studies of turbulent convection [4] give the dependence of the experimentally defined heat transfer coefficient α on the physical properties and thickness of the wall. The temperature fluctuations in the wall observed in [4], which cause α for a steel cylinder in the turbulent flow of liquid nitrogen to deviate from the calculated value and from the corresponding quantities for a copper cylinder, show that the temperature field in the wall can depend significantly on fluctuations in the flow. The thermal effect of a wall on turbulent natural convection has evidently not been studied. In the case of forced turbulent flow, the effect of the physical parameters of the wall on the turbulent Prandtl number and on the spectral distribution and intensity of temperature fluctuations in the viscous sublayer was observed experimentally [7], on the basis of the standard analysis of [5, 6]. It is difficult to determine from these results the extent to which the local heat transfer coefficient α and its average value $\langle\alpha\rangle$ are affected by fluctuations in the temperature and heat flux at the wall. The assumption of [8] seems convincing, however. According to this assumption, the local heat transfer coefficient of liquids with $Pr \gtrsim 1$ in turbulent flow is determined only by the hydrodynamic parameters (and also the time-averaged heat flux density on the wall in the case of natural convection) and does not depend on the physical properties and thickness of the wall and hence can be taken as the true heat transfer coefficient. Indeed, most of the thermal resistance of the flow for $Pr \gtrsim 1$ is localized within a thin boundary layer (including a viscous sublayer) and heat within the boundary layer is transported mainly by conduction. Hence the effect of the wall on the turbulent characteristics of the viscous sublayer should not have a dominating effect on heat transport within the sublayer. The model of [8], which is based on the same assumptions, predicts that the heat transfer coefficient is inversely proportional to the thickness of the viscous sublayer.

Experimental studies show that velocity and temperature fluctuations (which are highly correlated for fully turbulent natural convection [9]) are typically periodic in the direction of the average flow (x) and the motion of coherent structures (turbulence with "memory") damps out more slowly for natural convective flow near a wall than for isotropic turbulence in forced flow [9]. The phase velocity of propagation of temperature disturbances near the maximum average velocity is approximately equal to the maximum velocity ($U_{ph,x} \approx U_m$) and differs only slightly from U_m for other values of y [9]. According to the data of [10], $U_{ph,x} > 0.9U_m$ for the minimum scale of the temperature fluctuations in the x direction.

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It was shown in [11, 12] that in forced turbulent flow, hydrodynamic disturbances in the boundary layer have a three-dimensional band structure and there is a certain regularity in the space and time distribution of the "bands" [11]. This structure is associated with bursts in the boundary layer. The periodicity in time of "bursts" has been studied in numerous experimental papers (see [13]) for forced convection. Similar results were obtained in [14] for turbulent natural convection of water and air near a vertical plate with q_w = const. Different experimental methods have established an essentially periodic space–time structure of the boundary layer associated with the motion of large vortices and characterized by injection and ejection of liquid near the wall. The temporal τ_0 and spatial z_0 periods determining the motion of the bands of hydrodynamic disturbances and the associated temperature fluctuations of the liquid in the boundary layer were given in [14] in dimensionless form using the units of frequency $f_T = (g\beta_f q_w/\lambda_f)^{1/2}$ and length $z_T = (\lambda_f \nu_f^2/g\beta_f q_w)^{1/4}$:

$$f_0^+ = 1/(\tau_0 f_T) = 0{,}03;\ z_0^+ = z_0/z_T = 50, \tag{1}$$

where f_0^+ and z_0^+ do not depend on the Rayleigh number Ra_x^* in the region 10^{13}-10^{17}.

Measurements of the instantaneous velocity distributions in the boundary layer in forced turbulent flow by stroboscopic visualization [15, 16] indicate the presence of tangential stress fluctuations τ_w on the surface and characterize the time variation of the thickness η of the viscous sublayer. It is natural to associate these phenomena with the existence of large vortex structures in the flow and to assume (following [8]) that fluctuations in τ_w and η lead to fluctuations in the true heat transfer coefficient α, where $\alpha \sim \lambda_f/\eta$ (for $Pr \gtrsim 1$). Measurements of this kind for natural convection have apparently not been carried out.

Since the temperature disturbance waves in natural convection have spatial scales in the x [9] and z [14] directions, the true heat transfer coefficient can be written as a periodic function $\alpha(x, z, \tau)$. Hence in a direction making an angle $\phi = \arccos(x_0/\sqrt{x_0^2 + z_0^2})$ with the z axis, a plane temperature wave propagates with a spatial period of $x_0 z_0/\sqrt{x_0^2 + z_0^2}$. We estimate the ratio of the phase velocities of the temperature waves in the x and z directions, or, equivalently, the ratio x_0/z_0. The only available information on z_0 is the experimental results of [14]. Putting $x_0 \approx U_m \tau_0$ and $U_m = 2{,}1\{g\beta_f q_w[x - x_{cr}(1 - \alpha_{\ell}/\langle\alpha\rangle)]/\rho_f C_{p,f}\}^{1/3}$ [15], τ_0 can be introduced either according to (1) or in the form $\tau_0^+ = \tau_0 U_*^2/\nu_f = 325$ [4]. In the first case, if we also assume that $Nu_x = C(Gr_x^* Pr)^{1/4}$ for turbulent natural convection with q_w = const and that $Nu_x = 0.6163\ (Gr_x^*)^{1/5} Pr^{2/5}/(Pr + 0.8)^{1/5}$ for laminar natural convection [18], we obtain

$$x_0 = 70 Pr^{-1/3}\{x^3/Gr_x^{*1/2} - x_{cr}[1 - 0{,}77\, Pr^{3/20}/Gr_{cr}^{*1/20} C (Pr + 0{,}8)^{1/5}]/Gr_{cr}^{*1/2}\}^{1/3}, \tag{2}$$

$$(x_0/z_0) = 1{,}4\, Pr^{-1/3}\{Gr_{cr}^{*1/4} - Gr_{cr}^{*1/4}[1 - 0{,}77\, Pr^{3/20}/C (Pr + 0{,}8)^{1/5} Gr_{cr}^{*1/20}]\}^{1/3}. \tag{3}$$

In the second case $(U_* = [5{,}08(\beta_f g \nu_f \langle\vartheta\rangle)^{2/3} Pr^{-1/6}]^{0/5}$ [17]) we obtain the corresponding results

$$x_0 = 134{,}35 C^{2/3}\{x^3/Gr_x^{*1/2} - x_{cr}^3[1 - 0{,}77\, Pr^{3/20}/Gr_{cr}^{*1/20} C (Pr + 0{,}8)^{1/5}]/Gr_{cr}^{*\frac{1}{2}}\}^{\frac{1}{3}}, \tag{4}$$

$$(x_0/z_0) = 2{,}687 C^{2/3}\{Gr_x^{*1/4} - Gr_{cr}^{*1/4}[1 - 0{,}77\, Pr^{3/20}/C (Pr + 0{,}8)^{1/5} Gr_{cr}^{*1/20}]\}^{1/3}. \tag{5}$$

We see that (2), (4) and (3), (5) differ by constant factors and by the power of Pr. According to (4) and (5), x_0 and x_0/z_0 depend on Pr very weakly, whereas (2) and (3) predict that x_0 and x_0/z_0 vary approximately as $Pr^{-1/3}$. For liquid nitrogen (Pr = 2.4, $Gr^*_{cr} = 1.4 \cdot 10^{12}$, C = 0.2227) (2), (4) and (3), (5) give very similar results. In general, (2) and (3) are preferred, since the relation for τ_0 in these equations has been obtained experimentally for turbulent natural convection of water and air and accurately predicts τ_0 for convection of liquid nitrogen, while the dimensionless quantity $\tau_0^+ = 325$ is based only on experiments with liquid nitrogen. Our discussion will be based on (2) and (3) with the above values of Gr^*_{cr} and C. It follows from (3) that for fully turbulent natural convection ($Gr^*_x > 5 \cdot 10^{12}$) $(x_0/z_0) \gg 1$ for liquids with moderate Pr. For example, in the case of liquid nitrogen in the interval $Gr^*_x = 5 \cdot 10^{12}$-10^{14} (which corresponds to the interval of Gr^*_x studied experimentally in [4]) the quantity x_0/z_0 increases from 11.3 to 14.9. Then the values of z_0 calculated from (1) and corresponding to the experimental values of z_0 decrease with increasing q_w from 7.5 to 5.2 mm.

To calculate the experimentally defined heat transfer coefficient in terms of the true heat transfer coefficient one must solve the three-dimensional heat equation with a bound-

ary condition in the form of a progressive wave $\alpha(x, z, \tau)$ (for a plane wall) on the heat-transfer surface. Introducing the generalized variable $\xi = (\tau/\tau_0 \pm z/z_0 \pm x/x_0)$ the equation for the dimensionless temperature fluctuation θ in the wall can be transformed to

$$\partial^2\theta/\partial\bar{y}^2 = m\partial\theta/\partial\xi - \partial^2\theta/\partial\xi^2, \quad (6)$$

where $m = (x_0 z_0)^2/\sqrt{x_0^2+z_0^2}(a_w\tau_0)$; $\bar{y} = y\sqrt{x_0^2+z_0^2}/x_0 z_0$. This equation is formally identical to the two-dimensional heat equation considered in [8]. The only difference is that in [8] $m = z_0^2/a_w\tau_0$; $\bar{y} = y/z_0$, and $\xi = (\tau/\tau_0 \pm z/z_0)$. Hence the results of [8] can be carried over directly, replacing the one-dimensional variables and parameters by the corresponding two-dimensional quantities. Next, the function $\alpha(\xi)$ must be determined. The measurements of [16, 18] indicate that the dimensionless thickness of the viscous sublayer varies within the limits η^+ = 2-15 and the average boundary corresponds to the approximate value $\langle\eta^+\rangle = 6$. Assuming that η varies as $\eta = \langle\eta\rangle\sqrt{1-b^2}/(1+b\cos\xi)$, where $\xi = (\tau/\tau_0 \pm z/z_0 \pm x/x_0)$, we obtain $\langle\eta^+\rangle = 5.48$ and b = 13/17. The thickness of the viscous sublayer in turbulent natural convection near a flat plate was estimated in [3] using the data of [2] and [19] as $\langle\eta^+\rangle = 4$. Since there are no accurate data on the variation η in natural convection, we assume that η is given by the above inverse cosine dependence, where $\langle\eta^+\rangle = 5.48$ and b = 13/17 and does not depend on Gr^*_x, and that τ_0 and z_0 are given by (1) and x_0 is given by (2). Putting $\alpha(\xi) \sim \lambda_f/\eta(\xi)$, we obtain $\alpha(\xi) = \langle\alpha\rangle(1 + b\cos\xi)$ for the true heat transfer coefficient. We next consider the calculation of the experimental coefficient $\bar{\alpha}$ in terms of the average heat transfer in natural convection of liquid nitrogen and freon-113 (Pr = 6.7) [4]. Let $\langle\alpha\rangle$ be defined by the equation $Nu_x = 0.2227\,(Gr^*_x Pr)^{1/4}$. Using the boundary condition q_w = const on the inner surface of the wall, we have all of the necessary data in the framework of our model.

After introducing the appropriate two-dimensional parameters, the solution of [8] for $\varepsilon = \bar{\alpha}/\langle\alpha\rangle$ (where $\alpha(\xi)$ is in the form of a cosine) shows the effect of λ_w, ρ_w, C_w, $\langle\alpha\rangle$, τ_0, z_0, x_0 and the wall thickness δ on ε. Since ξ_0, z_0, and x_0 are in general unknown in turbulent natural convection near a vertical cylinder ((1) and (2) are for a flat plate) the use of the solution of [8] for a plate is an approximation in the case considered here (the ratio of the outer and inner diameters of the cylinder is 1.06). The expression for ε involves the sum of two complex-conjugate continued fractions. The quantity ε can be obtained to at least two decimal places by keeping the first five terms in the expansion. The calculation shows that ε varies from 0.822 to 0.842 for liquid nitrogen in the internal $Gr^*_x = 5\cdot10^{12}$-10^{14} for a 12Kh18N10T stainless steel wall with δ = 0.35 mm. The corresponding results for ε are 0.981 and 0.984 for an M-3 copper wall with δ = 0.35 mm. For turbulent convection of liquid freon-113 in the interval $Gr^*_x = 1.2\cdot10^{12}$-$2.2\cdot10^{13}$, ε increases from 0.956 to 0.965 for steel and from 0.978 to 0.981 for a copper wall (of the same thickness). The physical properties of M-3 copper and 12Kh18N10T steel were taken according to [20] from the saturation temperatures of liquid nitrogen and freon-113 (the physical properties of the liquids were taken from [21]). Hence deviations between the experimental heat transfer coefficient and $\varepsilon\langle\alpha\rangle$ larger than the experimental errors should be expected only for the steel wall for turbulent natural convection of liquid nitrogen, which was observed in the experiments of [4]. Some of the calculated results are compared to the experimental data in Tables 1 and 2. As shown above, the spatial dimensions of the temperature waves in turbulent natural convection of liquids with moderate Pr satisfy the condition $(x_0/z_0) \gg 1$. It is obvious that in this case all of the variables and parameters of the three-dimensional heat equation ($\alpha(x, z, \tau)$) approach the corresponding quantities of the two-dimensional

TABLE 1. Turbulent Natural Convection of Liquid Freon-113 ($t_\infty \approx t_s$ = 47.68°C): Comparison of the Experimental and Calculated Results

	M-3 copper cylinder, δ = 0.35 mm						12Kh18N10T steel cylinder, δ = 0.35 mm					
$Gr^*_x \cdot 10^{-12}$	1,257	3,73	11,3	2,411	7,155	21,68	1,312	3,892	11,52	2,541	7,453	22,05
q_w, W/м²	431,9			828,5			450,7			863,1		
$\langle\alpha\rangle$, W/(м²·K)	141,4			166,4			142,9			168,1		
$\varepsilon\langle\alpha\rangle$, W/(м²·K)	138,2			162,2			136,6			162,1		
$\bar{\alpha}$, W/(м²·K)	128,4	130,7	137,7	148,2	161,7	165,6	123,0	125,5	134,5	152,8	152,8	153,4
$(\varepsilon\langle\alpha\rangle-\bar{\alpha})/\bar{\alpha}\cdot100\%$	7,7	5,7	0,4	10,1	0,9	1,5	11,0	8,0	1,6	6,1	6,1	5,7

TABLE 2. Turbulent Natural Convection of Liquid Nitrogen ($T_\infty \approx T_s$ = 77.35 K): Comparison of the Experimental and Calculated Results

	M-3 copper cylinder, δ = 0.35 mm						12Kh18N10T steel cylinder δ = 0.35 mm					
$Gr_x^* \cdot 10^{-12}$	4,983	14,78	44,82	10,04	30,69	93,01	4,824	14,32	42,36	9,631	28,58	84,56
q_w, W/м²	376,0			780,3			364,0			726,7		
$\langle\alpha\rangle$ W/(м²·K)	295,3			354,5			292,9			348,2		
$\varepsilon\langle\alpha\rangle$, W/(м²·K)	289,7			348,8			240,9			293,1		
$\bar{\alpha}$, W/(м²·K)	242,5	301,3	301,3	322,9	345,6	364,3	232,8	221,0	226,7	269,9	243,4	282,8
$(\varepsilon\langle\alpha\rangle-\bar{\alpha})/\bar{\alpha}\cdot 100\%$	19,5	—4,0	—4,0	8,0	0,9	—4,3	3,4	9,0	6,3	8,6	10,4	3,6

problem ($\alpha(z, \tau)$). The results for $\varepsilon\langle\alpha\rangle$ calculated according to the three-dimensional and two-dimensional models are the same, to within the accuracy of the values listed in the tables. We note that from (1) the parameters of the two-dimensional model take the simple form: $m = 75\,\nu_f/a_w$; $\langle\bar{\alpha}\rangle = 11{,}135\ \mathrm{Pr}^{1/4}\lambda_f/\lambda_w$; $(\delta/z_0) = (\lambda_f\nu_f^2/g\beta_f q_w\delta^4)^{-1/4} = \mathrm{Gr}_\delta^{*1/4}$ and depend only on the physical properties of the liquid and the wall and on δ and q_w.

One of our fundamental assumptions was that the temperature waves and the functions $\alpha(\xi)$ and $\eta(\xi)$ in natural convective flow near the wall are periodic, whereas this is true only in a statistical sense [4, 14]. Therefore (and also in view of the other assumptions made above) the calculated results are approximate. Using the concepts of viscous and heat-conduction sublayers for natural convective flow of liquid nitrogen and freon-113 and the true heat transfer coefficient (independent of the physical properties and the thickness of the wall) and assuming a simple (cosine) dependence $\alpha(\xi)$, analysis of the experimental data on the thermal and hydrodynamic characteristics of the flow yields not only qualitative agreement but also a certain quantitative agreement between the calculated and experimental results for the average heat transfer.

Finally we note that when the results of the standard analysis of [22] are used to estimate α for the experimental conditions of [4], there are no significant differences in the average heat transfer of the copper and steel rods. This is evidently because the model of [22] does not contain the wall thickness or the spatial characteristics of the turbulent flow near the wall as parameters.

NOTATION

$\bar{\alpha} = q_w/(\bar{\vartheta})_{y=0}$, experimentally defined heat transfer coefficient; q_w, time-averaged local heat flux density on the surface; $\bar{\vartheta} = (\bar{\vartheta})_{y=0}$, time-averaged local temperature excess on the surface; $\vartheta(y, \xi)$, temperature excess in the wall (as a function of depth); $\alpha(\xi) = -(\lambda_w/\vartheta)\partial\vartheta/\partial y)_{y=0}$, $\alpha(x, z, \tau)$, $\alpha(z, \tau)$; α, true heat transfer coefficient of the liquid in turbulent flow; $\langle\alpha\rangle$, time-averaged value of α; $\langle\vartheta\rangle = q_w/\langle\alpha\rangle$; ϑ, fluctuation in ϑ; x, y, z, coordinates defined along the direction of average flow (along the plate), along the normal to the surface, and in the transverse direction in the plate; τ_0, x_0, z_0, temporal and spatial scales of the temperature disturbance; f_T, z_T, frequency and length scales; $f_0^+ = 1/(\tau_0 f_T)$; $z_0 = z_0/z_T$; $U_* = \sqrt{\tau_w/\rho_f}$; τ_w, tangential stress on the surface; $U_{ph,z} = x_0/\tau_0$; η, $\eta(\xi)$, thickness of the viscous sublayer; $\eta = yU_*/\nu_f$; $\langle\eta\rangle$; $\langle\eta^*\rangle$, average values of η and η^*; b, amplitude of the fluctuations in α and η; U_m, maximum average flow velocity in the turbulent boundary layer; $\bar{\alpha}_\ell$; heat transfer coefficient for laminar flow, average along the surface; $\xi = (\tau/\tau_0 \pm x/x_0 \pm z/z_0)$; λ_f, a_f, ν_f, β_f, c_{pf}, ρ_f, thermal conductivity, thermal diffusivity, kinematic viscosity, coefficient of volume expansion, specific heat, and density of the liquid; λ_w, a_w, C_w, ρ_w, thermal concuctivity, thermal diffusivity, specific heat, and density of the wall; $\mathrm{Gr}_x^* = g\beta_f q_w x^4/\nu_f^2\lambda_f$; Gr^*_{cr} and x_{cr}, critical Grashof number and the corresponding value of x for a given q_w; $\varepsilon = \alpha/\langle\alpha\rangle$, $\mathrm{Pr} = \nu_f/a_f$; $\langle\bar{\alpha}\rangle = \langle\alpha\rangle z_0/\lambda_w$; $\mathrm{Ra}_x^* = \mathrm{Gr}_x^*\mathrm{Pr}$; $\mathrm{Nu}_x = \langle\alpha\rangle x/\lambda_f$; $\theta = \hat{\vartheta}/\bar{\vartheta}$.

LITERATURE CITED

1. E. R. G. Eckert and T. W. Jakson, NACA Rep., No. 1015 (1951).
2. W. K. George and S. P. Capp, Int. J. Heat Mass Transf., 22, No. 6, 813-826 (1979).
3. W. M. To and J. A. C. Humphrey, Int. J. Heat Mass Transfer, 29, No. 4, 573-591 (1986).

4. G. I. Abramov and K. A. Ivanov, Inzh.-fiz. Zh., 60, No. 3, 379-385 (1991).
5. P. I. Geshev, Inzh.-fiz. Zh., 35, No. 2, 292-296 (1978).
6. A. F. Polyakov, Teplofiz. Vys. Temp., 12, No. 2, 328-336 (1974).
7. B. S. Petukhov, A. F. Polyakov, Yu. L. Shekhter, and Yu. V. Tsypulev, Turbulent Boundary-Layer Flow [in Russian], Novosibirsk (1975), Part 2, pp. 162-177.
8. D. A. Labuntsov and Yu. B. Zudin, Periodic Heat-Transfer Processes [in Russian], Moscow (1984).
9. R. Cheesewright and K. S. Doan, Int. J. Heat Mass Transfer, 21, No. 7, 911-921 (1978).
10. V. P. Ivakin, A. G. Kirdyashkin, and L. I. Chernyavskii, Turbulent Boundary-Layer Flow [in Russian], Novosibirsk (1975); Part 2, pp. 256-269.
11. S. Klein, W. Reynolds, F. Shroub, et al., Mekhanika, No. 4, 42-78 (1969).
12. E. Corino and R. Brodky, Mekhanika, No. 1, 56-82 (1971).
13. L. C. Thomas, Int. J. Heat Mass Transfer, 25, No. 8, 1127-1136 (1982).
14. K. Kutamura, M. Koike, I. Fukuoka, and T. Saito, Int. J. Heat Mass Transfer, 28, No. 4, 837-850 (1985).
15. E. M. Khabakhpasheva, Problems of Heat Physics and Hydrodynamics [in Russian], Novosibirsk (1974), pp. 223-235.
16. S. S. Kutateladze, B. P. Mironov, V. E. Nakoryakov, and E. M. Khabakhpasheva, Experimental Studies of Turbulent Boundary-Layer Flow [in Russian], Novosibirsk (1975).
17. S. S. Kutateladze, A. G. Kirdyashkin, and V. P. Ivakin, Dokl. Akad. Nauk SSSR, 214, No. 6, 1270-1273 (1974).
18. S. S. Kutateladze, E. M. Khabakhpasheva, and B. V. Perepelitsa, Heat Transfer, 1978. Soviet Research [in Russian], Moscow (1980), pp. 5-13.
19. R. Cheesewright and E. Ierokipiotis, Proc. 7th Int. Heat Trans. Conf., Vol. 2 (1982), pp. 305-309.
20. I. G. Kozhevnikov and L. A. Novitskii, Thermal and Physical Properties of Materials at Low Temperatures [in Russian], Moscow (1982).
21. N. B. Vargaftik, Thermal and Physical Properties of Gases and Liquids [in Russian], Moscow (1972).
22. P. L. Kirillov, Inzh.-fiz. Zh., 50, No. 3, 501-512 (1986).

DISSOCIATION OF GASEOUS HYDRATES IN BEDS

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A mathematical model which accounts for the mobility of the liquid phase is constructed to describe the dissociation of gaseous hydrates in beds.

1. The extraction of gaseous hydrates from beds presumes that these compounds break down in the beds. The dissociation process, accompanied by the evolution of substantial volumes of gas and absorbed heat, is controlled to a considerable extent by the initial state of the bed system. Three different mathematical models corresponding to different states have been constructed to describe the breakdown of a gaseous hydrate in a porous medium. The model in [1] is based on the proposition that the pores in the bed are completely saturated with gaseous hydrate. The authors of [2] examined the case when the gas in the porous medium is in the two-phase state at the initial moment of time. In [3], it was assumed that the bed initially contains the gaseous hydrate, gas, and water in a state of thermodynamic equilibrium. These models make it possible to obtain quantitative estimates of processes occurring in beds when gas is being extracted from a gas–hydrate deposit by the pressure-reduction method in combination with heating of the well region.

The above three models are all based on the assumption that the effect of water on the given physical process (dissociation of hydrates) is small enough to be ignored. In those cases in which the level of hydrate saturation is high and, thus, a proportionately large

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