CALCULATION OF THE TEMPERATURE FIELD IN A GAS VENTILATED CHARGE OF FINE-POROSITY MATERIAL

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The temperature distribution is obtained for a gas-ventilated layer of fine-porosity material containing internal heat sources and cooled with a turbulent stream of liquid. Both parallel flow and counterflow of the two heat carriers are analyzed.

The problem of temperature distribution in a gas-ventilated charge, taking into account the heat transfer between the container walls and the cooling stream, has been considered mainly in studies in which the presence of internal heat sources is disregarded or an approximate solution is given [1-3]. All these studies are concerned with the case in which a stationary gas-ventilated charge of comminuted material is cooled with a stream of liquid, the temperature of which does not vary along the height. In heat-engineering practice, however, it is much more often found that there is a simultaneous temperature variation in the ventilated layer and in the coolant. The gas and the coolant may be flowing in the same direction or in opposite directions.

We will consider the temperature field in a gas-ventilated charge of finely dispersed material containing uniformly distributed bulk-heat sources and the temperature field in the turbulent cooling liquid.

For this case the temperature distribution is found from the solution of a system of differential equations set up under the following assumptions: 1) the ratio of charge diameter to particle diameter D/d, the charge being contained in a cylinder, is such that the mass rate of gas flow is constant over the cross section; 2) the thermal resistance of the container walls is small; 3) the mass rates of gas and liquid flow as well as the thermophysical properties are invariable; 4) the coefficients of heat transfer between the walls and the coolant are constant over the height; 5) the effective thermal conductivity of a layer is the same in the axial and in the radial direction; 6) the heat generated in the charge is immediately dissipated in raising the temperature of the mainstream.

With these assumptions, the system of equations is:

$$\lambda\left(\frac{\partial^2 t}{\partial r^2}+\frac{1}{r}\cdot\frac{\partial t}{\partial r}+\frac{\partial^2 t}{\partial z^2}\right)-w\gamma c_p\frac{\partial t}{\partial z}+q_v=0,$$

$$\pm w_1\gamma_1 c_{p1}\frac{\partial t_1}{\partial z}=\alpha\,\frac{\Pi}{S}\,[t(R,\ z)-t_1(z)]. \qquad (1)$$

The problem will be solved by the Fourier method.

In the second of Eqs. (1) the plus sign refers to a parallel flow of heat carriers and the minus sign refers to a counterflow of heat carriers.

The boundary conditions are the same in both cases considered here:

$$\left.\frac{\partial t}{\partial r}\right|_{r=0}=0;\qquad \left.\frac{\partial t}{\partial r}\right|_{r=R}=-\frac{\alpha}{\lambda}\,[t(R,\ z)-t_1(z)]; \qquad (2)$$

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the initial conditions are, for parallel flow,

$$t|_{z=0} = t_0; \quad t_1|_{z=0} = t_{10}; \quad t|_{z\to\infty} \text{ limit}; \tag{3a}$$

and, for counterflow,

$$t|_{z=0} = t_0; \quad t_1|_{z=L} = t_{10}; \quad t|_{z\to\infty} \text{ limit}, \tag{3b}$$

where L is the channel height.

We introduce the following designations:

$$a = \alpha \cdot \frac{\Pi}{S}; \quad b = \frac{q_v}{\lambda}; \quad c = \frac{a}{\beta_1}; \quad d = \frac{\beta a}{\beta_1 \lambda}; \quad \varepsilon = \frac{\beta}{\lambda}.$$

We first extract the singularities in the solution, writing the latter in the form:

$$\begin{gathered} t(r, z) = \tau(r, z) + Az + Br^2 + F, \\ t_1(z) = \tau_1(z) + Ez + D. \end{gathered} \tag{4}$$

The coefficients in Eqs. (4) are determined so that the particular solutions will satisfy Eqs. (1) and the boundary conditions (2). Then,

$$\begin{gathered} -\varepsilon A + 4B + b = 0, \\ E = \pm c\,(Az + BR^2 + F - Ez - D), \\ 2BR = -\frac{\alpha}{\lambda}(BR^2 + F - D). \end{gathered} \tag{5}$$

From this we have

$$A = E = \pm \frac{bR^2 d}{\varepsilon\,(2\mathrm{Bi} \pm dR^2)};$$

for counterflow when $2\mathrm{Bi} \neq dR^2$:

$$\begin{gathered} B = -\frac{b}{4\left(1 \pm \dfrac{dR^2}{2\mathrm{Bi}}\right)}; \\ F - D = \frac{bR^2\left(\dfrac{2}{\mathrm{Bi}} + 1\right)}{4\left(1 \pm \dfrac{dR^2}{2\mathrm{Bi}}\right)}. \end{gathered} \tag{5a}$$

For the unknown functions τ and τ_1 we obtain the following system of equations:

$$\begin{gathered} \frac{\partial^2\tau}{\partial r^2} + \frac{1}{r}\cdot\frac{\partial\tau}{\partial r} + \frac{\partial^2\tau}{\partial z^2} - \varepsilon\frac{\partial\tau}{\partial z} = 0, \\ \frac{\partial\tau_1}{\partial z} = \pm c\,[\tau(R, z) - \tau_1(z)], \end{gathered} \tag{6}$$

$$\left.\frac{\partial\tau}{\partial r}\right|_{r=0} = 0; \quad \left.\frac{\partial\tau}{\partial r}\right|_{r=R} = -\frac{\alpha}{\lambda}\,[\tau(R, z) - \tau_1(z)], \tag{7}$$

with the initial conditions for parallel flow

$$\tau|_{z=0} = t_0 + \frac{br^2}{4\left(1 + \dfrac{dR^2}{2\mathrm{Bi}}\right)} - F, \tag{7a}$$

$$\tau|_{z\to\infty} = 0 \quad \text{for} \quad E - D = \frac{bR^2\left(1 + \dfrac{2}{\mathrm{Bi}}\right)}{4\left(1 + \dfrac{dR^2}{2\mathrm{Bi}}\right)},$$

$$\tau_1|_{z=0} = t_{10} - D,$$

and for counterflow

$$\tau|_{z=0} = t_0 + \frac{br^2}{4\left(1 - \frac{dR^2}{2\mathrm{Bi}}\right)} - F, \tag{7b}$$

$$\tau|_{r\to\infty} = 0 \quad \text{for} \quad F - D = \frac{bR^2\left(1 + \frac{2}{\mathrm{Bi}}\right)}{4\left(1 - \frac{dR^2}{2\mathrm{Bi}}\right)},$$

$$\tau_1|_{z=L} = t_{10} - D + \frac{bR^2 dL}{\varepsilon(2\mathrm{Bi} - dR^2)}.$$

The solution to system (6) will be sought in the form of a series:

$$\tau(r, z) = \sum_{n=1}^{\infty} A_n \varphi_n(r) f_n(z). \tag{8}$$

In accordance with the constraints at r = 0 and z → ∞, we have chosen the following solutions:

$$\varphi_n(r) = J_0(k_n r),$$
$$f_n(z) = \exp\left[\left(\frac{\varepsilon}{2} - \sqrt{\frac{\varepsilon^2}{4} + k_n^2}\right) z\right]. \tag{8a}$$

The solution

$$\varphi_n(r) = I_0(k_n r),$$
$$f_n(z) = \exp\left[\left(\frac{\varepsilon}{2} - \sqrt{\frac{\varepsilon^2}{4} - k_n^2}\right) z\right]$$

has no physical meaning, since there can be no unlimited temperature rise with increasing r in the presence of internal heat sources.

From the second equation in (6) we determine $\tau_1(z)$:

$$\tau_1(z) = \pm c \sum_{n=1}^{\infty} J_0(k_n R) \frac{\exp\left[\left(\frac{\varepsilon}{2} - \sqrt{\frac{\varepsilon^2}{4} + k_n^2}\right) z\right]}{\pm c + \frac{\varepsilon}{2} - \sqrt{\frac{\varepsilon^2}{4} + k_n^2}}. \tag{9}$$

Inserting this solution into the boundary condition at r = R, we obtain the characteristic equation for determining the values of k_n:

$$-J_0(k_n R) + \frac{k_n^2 \mp d\left(\frac{1}{2} + \sqrt{\frac{1}{4} + \frac{k_n^2}{\varepsilon^2}}\right)}{k_n \mathrm{Bi}} R J_1(k_n R) = 0. \tag{10}$$

Since often in practice $\varepsilon R \simeq 10^3$, Eq. (10) becomes

$$-J_0(\mu_n) + \frac{\mu_n^2 \mp dR^2}{\mu_n \mathrm{Bi}} J_1(\mu_n) = 0 \tag{10a}$$

for not very high values of $\mu_n = k_n R$. The first five roots of Eq. (10a) for parallel flow and for counterflow with $dR^2 = 6.72$ are listed in Table 1.

The largest difference between values of the roots is observed in the fundamental mode. At high values of n the values of the roots will be slightly lower than the corresponding roots of equation $J_1(\mu_n) = 0$ for both types of flow. It is to be noted that at low values of Bi ($\mathrm{Bi} < dR^2/2$) the values of the first roots of Eq. (10a) increase much more in the case of counterflow than in the case of parallel flow.

The coefficients A_n are determined from the initial conditions (7) and (7a) at z = 0:

$$\sum_{n=1}^{\infty} A_n J_0(k_n r) = t_0 + \frac{br^2}{4\left(1 \pm \frac{dR^2}{2\mathrm{Bi}}\right)} - F. \tag{11}$$

TABLE 1. Values of the First Five Roots of the Characteristic Equation (dR^2 = 6.72)

Bi	μ_n				
	μ_1	μ_2	μ_3	μ_4	μ_5
For parallel flow					
1	2,530	4,214	7,176	10,277	13,401
3	2,480	4,645	7,457	10,470	13,551
5	2,459	4,878	7,674	10,646	13,690
8	2,445	5,065	7,893	10,856	13,872
10	2,438	5,141	8,010	10,965	13,975
For counterflow					
1	4,009	7,140	10,265	13,396	16,53
3	4,330	7,371	10,441	13,536	16,65
5	1,174	4,575	7,568	10,600	13,667
8	1,680	4,794	7,793	10,800	13,842
10	1,835	4,926	7,907	10,909	13,941

After necessary transformations, we have

$$A_n = \frac{2(t_0 - F)}{k_n R J_1(k_n R)\left\{\dfrac{\lambda^2}{\alpha^2}\dfrac{\left[k_n^2 \mp d\left(\dfrac{1}{2} + \sqrt{\dfrac{1}{4} + k_n^4/\varepsilon^2}\right)\right]^2}{k_n^2} + 1\right\}} + \frac{b}{2\left(1 \pm \dfrac{dR^2}{2\mathrm{Bi}}\right)} \cdot \frac{\dfrac{R}{k_n} + 2\dfrac{1}{k_n^2}\cdot\dfrac{J_0(k_n R)}{J_1(k_n R)} - 4\dfrac{1}{R k_n^3}}{\left\{\dfrac{\lambda^2}{\alpha^2}\cdot\dfrac{\left[k_n^2 \mp d\left(\dfrac{1}{2} + \sqrt{\dfrac{1}{4} + k_n^2/\varepsilon^2}\right)\right]^2}{k_n^2} + 1\right\}}. \tag{12}$$

For finding the constant term F we use the boundary conditions for $\tau_1(z)$. We now consider parallel flow and counterflow separately.

For parallel flow at z = 0 the equality

$$t_{10} - F + \frac{bR^2\left(\dfrac{2}{\mathrm{Bi}} + 1\right)}{4\left(1 + \dfrac{dR^2}{2\mathrm{Bi}}\right)} = \sum_{n=1}^{\infty} \frac{A_n (k_n R)\, d J_1(k_n R)}{\mathrm{Bi}\, \varepsilon s_n}$$

together with expression (12) yields, after necessary transformations:

$$A_n = \frac{2\Delta t_0}{(1+\sigma_1)(k_n R)\left[J_1(k_n R) + \dfrac{J_0^2(k_n R)}{J_1(k_n R)}\right]} + \frac{bR^2\left[2\mu_n J_0(\mu_n) - 4J_1(\mu_n) + \dfrac{\sigma_1 - \sigma_2 - \dfrac{2}{\mathrm{Bi}}}{1+\sigma_1} J_1(\mu_n)\mu_n^2\right]}{\dfrac{1 + dR^2}{\mathrm{Bi}}\mu_n^3\left[J_0^2(\mu_n) + J_1^2(\mu_n)\right]}. \tag{12a}$$

Analogously, for counterflow at z = L the equality

$$t_{10} - F + \frac{bR^2\left(\dfrac{2}{\mathrm{Bi}} + 1\right)}{4\left(1 - \dfrac{dR^2}{2\mathrm{Bi}}\right)} + \frac{bR^2 dL}{\varepsilon(2\mathrm{Bi} - dR^2)} = \sum_{n=1}^{\infty} \frac{A_n(\mu_n)\, d J_1(\mu_n)}{\mathrm{Bi}\, \varepsilon s_n} \exp(s_n L)$$

yields

$$A_n = \frac{2\Delta t_0}{(1-\sigma_1)\mu_n\left[J_1(\mu_n) + \dfrac{J_0^2(\mu_n)}{J_1(\mu_n)}\right]} + \frac{bR^2\left[2\mu_n\dfrac{J_0(\mu_n)}{J_1(\mu_n)} - 4 - \dfrac{\sigma_1 - \sigma_2 + \dfrac{2}{\mathrm{Bi}} + \dfrac{2}{\mathrm{Bi}}\cdot\dfrac{dL}{\varepsilon}}{1-\sigma_1}\mu_n^2\right]}{\left(\dfrac{1 - dR^2}{\mathrm{Bi}}\right)\mu_n^3\left[\dfrac{J_0^2(\mu_n)}{J_1(\mu_n)} + J_1(\mu_n)\right]}. \tag{12b}$$

Taking into account expressions (5a), (8), and (12), we finally obtain from (4) the temperature distribution in the ventilated layer:

$$t(r,\ z) = t_0 - \frac{\Delta t_0}{1 \pm \sigma_1} \pm \frac{bR^2dz}{\varepsilon\,(2\mathrm{Bi} \pm dR^2)} + \frac{bR^2}{4\left(1 \pm \dfrac{dR^2}{2\mathrm{Bi}}\right)}\left(K - \frac{r^2}{R^2}\right)$$

$$+ \frac{\Delta t_0}{1 \pm \sigma_1}\sum_{n=1}^{\infty}\frac{2J_0(\mu_n r/R)\exp(s_n z)}{\mu_n\left[J_1(\mu_n) + \dfrac{J_0^2(\mu_n)}{J_1(\mu_n)}\right]}$$

$$+ \frac{bR^2}{2\left(1 \pm \dfrac{dR^2}{2\mathrm{Bi}}\right)}\sum_{n=1}^{\infty}\frac{2\mu_n\dfrac{J_0(\mu_n)}{J_1(\mu_n)} - 4 - (K-1)\,\mu_n^2}{\mu_n^3\left[J_1(\mu_n) + \dfrac{J_0^2(\mu_n)}{J_1(\mu_n)}\right]}$$

$$\times J_0\left(\mu_n\frac{r}{R}\right)\exp(s_n z). \qquad (13)$$

The values of K and σ_1 are determined from the relations

$$\sigma_1 = \frac{dR^2}{\mathrm{Bi}}\sum_{n=1}^{\infty}\frac{1 + \sqrt{1 + \left(\dfrac{2\mu_n}{\varepsilon R}\right)^2}}{\mu_n^2\left[1 + \dfrac{J_0^2(\mu_n)}{J_1^2(\mu_n)}\right]}\exp\left(\frac{s_n L}{2} \mp \frac{s_n L}{2}\right),$$

$$K = \frac{\dfrac{2}{\mathrm{Bi}} + 1 \pm \sigma_2 + \dfrac{dL}{\mathrm{Bi}\,\varepsilon} \mp \dfrac{dL}{\mathrm{Bi}\,\varepsilon}}{1 \pm \sigma_1},$$

where

$$\sigma_2 = \frac{dR^2}{\mathrm{Bi}}\sum_{n=1}^{\infty}\frac{\left(1 + \sqrt{1 + \left(\dfrac{2\mu_n}{\varepsilon R}\right)^2}\right)[(\mu_n^2 - 4)\,J_1^2(\mu_n) + 2\mu_n J_1(\mu_n)\,J_0(\mu_n)]}{\mu_n^4\,[J_0^2(\mu_n) + J_1^2(\mu_n)]\exp\left(-\dfrac{s_n L}{2} \pm \dfrac{s_n L}{2}\right)}$$

and

$$s_n = \frac{\varepsilon}{2} - \sqrt{\left(\frac{\varepsilon}{2}\right)^2 + \left(\frac{\mu_n}{R}\right)^2}.$$

The temperature distribution in the coolant along the channel height is, for parallel flow and for counterflow respectively,

$$t_1(z) = t_0 - \frac{\Delta t_0}{1 \pm \sigma_1} \pm \frac{bR^2dz}{\varepsilon\,(2\mathrm{Bi} \pm dR^2)} + \frac{bR^2(K-1)}{4\left(1 \pm \dfrac{dR^2}{2\mathrm{Bi}}\right)}$$

$$\pm \frac{\Delta t_0}{1 \pm \sigma_1}\cdot\frac{2d}{\mathrm{Bi}\,\varepsilon}\sum_{n=1}^{\infty}\frac{J_1(\mu_n)\exp(s_n z)}{s_n\left[J_1(\mu_n) + \dfrac{J_0^2(\mu_n)}{J_1(\mu_n)}\right]}$$

$$\pm \frac{bR^2}{\left(1 \pm \dfrac{dR^2}{2\mathrm{Bi}}\right)}\,\frac{d}{\mathrm{Bi}\,\varepsilon}\sum_{n=1}^{\infty}\frac{2\mu_n\dfrac{J_0(\mu_n)}{J_1(\mu_n)} - 4 + (1-K)\,\mu_n^2}{\mu_n^2 s_n\left[J_1(\mu_n) + \dfrac{J_0^2(\mu_n)}{J_1(\mu_n)}\right]}\,J_1(\mu_n)\exp(s_n z), \qquad (14)$$

where

$$\Delta t = t_0 - t_{10}.$$

In expressions (13) and (14) the upper sign applies to parallel flow and the lower sign applies to counterflow.

The results which have been obtained here for counterflow are valid when $dR^2 \neq 2\mathrm{Bi}$. In order to obtain a solution when $dR^2 = 2\mathrm{Bi}$, we must revert to Eqs. (1), (2), (3) and insert there $dR^2 = 2\mathrm{Bi}$. The solution can then be sought directly in the form of a sum:

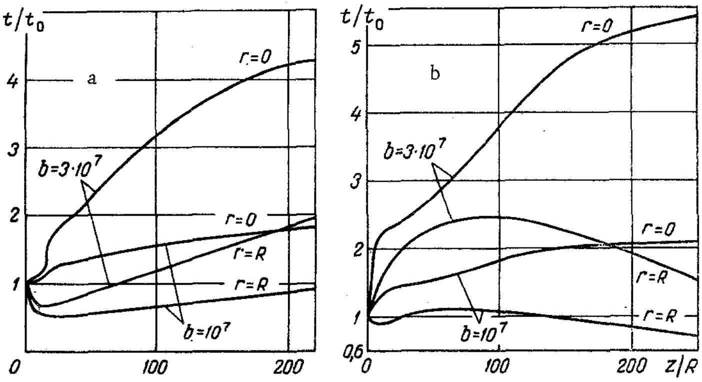

Fig. 1. Temperature distribution in a ventilated layer: a) with parallel flow cooling; b) with counterflow cooling; $dR^2 = 6.72$; Bi = 10; $\varepsilon R = 550$; $t_0 = 200°C$; $t_{10} = 40°C$.

$$t(r,\ z)=\sum_{n=1}^{\infty} A_n \varphi_n(r) f_n(z), \tag{15}$$

where the values of $\varphi_n(r)$ and $f_n(r)$ are determined according to Eq. (8a). The value of μ_n here will be the solution of the transcendental equation

$$-J_0(\mu_n)+\frac{\mu_n^2+2\mathrm{Bi}}{\mu_n\,\mathrm{Bi}} J_1(\mu_n)=0.$$

It is evident from Eq. (10a) that the first root of systems (1), (2), (3) is equal to zero and, therefore, the first term of sum (15) is a constant. At large z values the temperature t(r, z) is independent of the channel height.

When $dR^2 > 2\mathrm{Bi}$, the first root near unity vanishes in the sums of expression (13). The values of several successive roots are comparable and, therefore, must all be taken into account. Along the asymptotic portion of expression (13), where only one or two terms of the sums need to be considered, it is always the case that $t_{\mathrm{counter}}(r,\ z) < t_{\mathrm{parallel}}(r,\ z)$.

The solutions for both t(r, z) and $t_1(r,\ z)$ depend strongly on the parameter dR^2, which is proportional to the ratio of water equivalents β/β_1. A decrease of this ratio down to $\beta/\beta_1 \to 0$ will cause the temperature distribution in the coolant to approach $t_1(z) = \mathrm{const}$ in both cases considered here and, consequently, the difference between parallel flow and counterflow will be erased. In the absence of heat sources, the expression for t(r, z) corresponds now to the earlier case of a ventilated layer cooled with a liquid at constant temperature [2].

A comparison of the solutions for the case without a heat source (b = 0) will show that, for a more effective cooling of a ventilated layer, parallel flow is needed when $\mathrm{Bi} \lessdot dR^2/2$ ($\sigma_1 > 1$) and counterflow is needed when $\mathrm{Bi} \gtrdot dR^2/2$ ($\sigma_1 < 1$), if $\Delta t > 0$.

When a heat source is present ($b \neq 0$) and the ratio of water equivalents β/β_1 is smaller than 2Bi, then the parallel flow arrangement is more effective than the counterflow arrangement up to moderate heights z; but as z increases, the difference in temperatures $t_{\mathrm{parallel}}(r,\ z)-t_{\mathrm{counter}}(r,\ z)$ changes sign. A typical temperature distribution in a ventilated layer with parallel flow and with counterflow cooling is shown in Fig. 1a, b. The following data were used for the calculations here: $dR^2 = 6.72$, Bi = 10, $t_0 = 200°C$, and $t_{10} = 40°C$; the values of parameters b and εR were varied from 10^7 to $3 \cdot 10^7$ and from 550 to 1100, respectively. Evidently, the asymptotic range with a linear dependence on z is reached sooner as less heat is generated in the bulk. The radial temperature gradient is in this range proportional to the heat generated in the bulk. A characteristic peculiarity of the temperature distribution in a counterflow arrangement is the occurrence of a peak, while the temperature continues to rise in a parallel flow arrangement. Based on a comparison between these results, there follow two criteria choosing the appropriate cooling arrangement: 1) if lower mean-over-the-height temperatures in the charge are desired, then parallel flow is preferable; 2) if a more uniform temperature distribution over the height of a charge is desired, then counterflow is preferable.

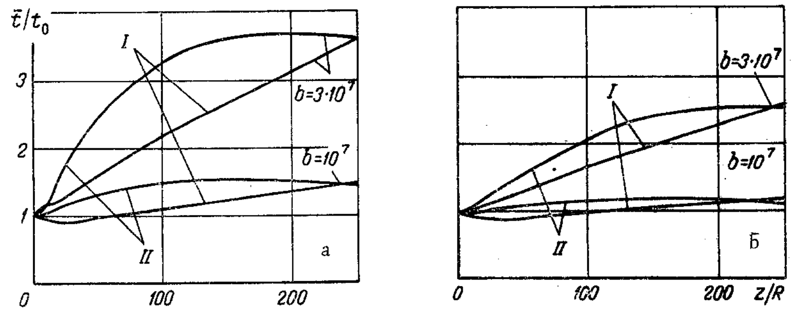

Fig. 2. Mean-over-the-radius temperature as a function of the layer height: a) $\varepsilon R = 550$; $\varepsilon R = 1100$. Parallel flow (I) and counterflow (II).

The basic difference characterizing the peripheral regions of a charge is retained when one considers the mean-over-the-radius temperatures described by the following relation:

$$t(z) = t_0 - \frac{\Delta t_0}{1 \pm \sigma_1} \pm \frac{bR^2 dz}{\varepsilon(2\mathrm{Bi} \pm dR^2)}$$

$$- \frac{bR^2\left(\frac{2}{\mathrm{Bi}} + \frac{1}{2} \mp \frac{\sigma_1}{2} \pm \sigma_2 + \frac{dL}{\mathrm{Bi}\,\varepsilon} \mp \frac{dL}{\mathrm{Bi}\,\varepsilon}\right)}{4\left(1 \pm \frac{dR^2}{2\mathrm{Bi}}\right)(1 \pm \sigma_1)}$$

$$+ \frac{\Delta t_0}{1 \pm \sigma_1} \sum_{n=1}^{\infty} \frac{4\exp(s_n z)}{\mu_n^2\left[1 + \frac{J_0^2(\mu_n)}{J_1^2(\mu_n)}\right]}$$

$$+ \frac{bR^2}{1 \pm \frac{dR^2}{2\mathrm{Bi}}} \sum_{n=1}^{\infty} \frac{2\mu_n \frac{J_0(\mu_n)}{J_1(\mu_n)} - 4 + \frac{\pm\sigma_1 \mp \sigma_2 - \frac{2}{\mathrm{Bi}} - \frac{dL}{\mathrm{Bi}\,\varepsilon} \pm \frac{dL}{\mathrm{Bi}\,\varepsilon}}{1 \pm \sigma_1}}{\mu_n^4\left[1 + \frac{J_0^2(\mu_n)}{J_1^2(\mu_n)}\right]\exp(-s_n z)}$$

and represented graphically in Fig. 2a.

It is evident that the difference between parallel flow and counterflow in the degrees of temperature uniformity over the height of a charge increases as $b = q_v/\lambda$ becomes larger. We note that along a definite distance the temperatures tend to equalize and the point where the curves for parallel flow and for counterflow intersect depends on the magnitudes of b and εR. Increasing the flow rate of the ventilating gas causes a shift of the intersection point to the left, i.e., a drop of the mean-over-the-length temperature in the case of counterflow cooling. Thus, as the value of εR increases, the counterflow arrangement yields a more uniform and intensive cooling of a gas ventilated fine-dispersion layer (Fig. 2b). With all the other conditions unchanged, the same effect will be observed also when dR^2 increases.

It is to be noted that a special case of the solution shown here is the temperature field in a solid cylinder with heat sources uniformly distributed over its volume, corresponding to the value $\beta = 0$ in Eq. (1).

NOTATION

t, t_1	are the temperature of the charge layer and of the cooling liquid, respectively;
R, r	are the outer radius and radius at any point of a cylindrical charge;
λ	is the effective thermal conductivity of a layer;
w, γ, c_p	are the velocity, specific gravity, and specific heat of the ventilating gas, $w\gamma c_p = \beta$;
w_1, γ_1, c_{p1}	are the velocity, specific gravity, and specific heat of the coolant, $w_1\gamma_1 c_{p1} = \beta_1$;
q_v	is the heat generated in the bulk;
α	is the mean coefficient of heat transfer between wall and coolant;
Π/S	is the ratio of the perimeter of the layer to the area occupied by the coolant.

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