BOUNDARY CONDITIONS FOR A FORCED TRANSPIRATION COOLING SYSTEM

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UDC 536.24:532.685

A study is made of the boundary conditions, and results are presented on the effects from the convective heat-transfer rate at the inner surface of a permeable wall on the temperature state in it.

One of the most effective methods of thermal protection is cooling by transpiration for any structure subject to external convective or radiative heat fluxes of high density [1]. Permeable components with bulk heat production may also be cooled in this way, and the heat can arise from various sources: ohmic heating, fission in nuclear fuel, bulk absorption of penetrating radiation, or radiative heat fluxes within a porous semitransparent matrix. The temperature patterns in such systems have been examined on many occasions [2-25]. However, it is difficult to compare the results because of the substantial arbitrary element in formulating the boundary conditions, which sometimes also contain errors.

The distribution of the temperatures in the matrix T and coolant t is described by the following system of equations for a planar permeable wall subject to a high-density heat input and cooling from a counter current of single-phase coolant passed through it with a mass flow rate G (Fig. 1):

$$\lambda \frac{d^2T}{dZ^2} = h_v (T - t) = Gc \frac{dt}{dZ}, \tag{1}$$

which can be written by means of dimensionless quantities as

$$\frac{d^3t}{dz^3} + A\frac{d^2t}{dz^2} - AB\frac{dt}{dz} = 0,\ T = t + \frac{1}{A}\frac{dt}{dz}, \tag{2}$$

whereupon the solution takes the form

$$t = C_1 + C_2 \exp D_1 z + C_3 \exp D_2 z, \tag{3}$$

$$T = C_1 + C_2 (1 + D_1/A) \exp D_1 z + C_3 (1 + D_2/A) \exp D_2 z, \tag{4}$$

where

$$z = Z/\delta;\ A = h_v\delta/Gc;\ B = G\delta c/\lambda;\ D_{1,2} = \frac{A}{2}[-1 \pm (1 + 4B/A)^{1/2}]. \tag{5}$$

A survey has been made [26] of the experimental data on the rates h_v of bulk convective heat transfer in permeable materials of various structures, while in [27] there is a survey of the thermal conductivities λ of porous metals.

To determine the coefficients C_1, C_2, and C_3 one needs to specify three boundary conditions, but only the following two have a clear-cut physical meaning in this process.

The outer surface of the wall is heated by a radiative heat flux of density q_r or by a flow of hot gas with temperature t_∞, or by both together:

$$Z = \delta\ \lambda dT/dZ = q_r + \alpha'' (t_\infty - T''). \tag{6}$$

One of the following heat-balance equations is obeyed at the inner surface of the permeable wall in accordance with the coolant supply conditions: a) when a gas with initial temperature t_0 is supplied along the normal to the surface,

$$Z = 0\ \lambda dT/dZ = Gc (t' - t_0), \tag{7}$$

b) on flow along the porous wall

$$Z = 0\ \lambda dT/dZ = \alpha' (T' - t_0), \tag{8}$$

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 47, No. 4, pp. 587-594, October, 1984. Original article submitted March 18, 1983.

0022-0841/84/4704-1170$08.50

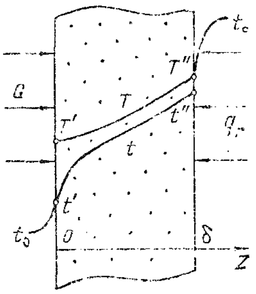

Fig. 1. Physical model

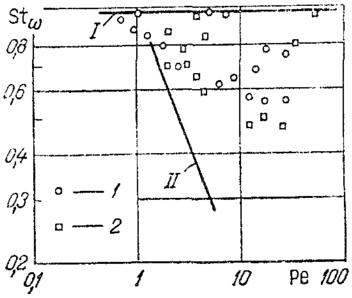

Fig. 2. Experimental data on the heat-transfer rates at the input surfaces of porous powdered metals: 1) results of [22] for d_p = 100 μm; 2) 350 μm; empirical correlations: I) [22]; II) [23].

TABLE 1. Experimental Conditions and Results on Heat Transfer at the Inlet to a Permeable Wall

Dimensionless heat-transfer equation	Representation		Range in Pe	Coolant	Porous material	Porosity	Mean particle size d_p, μm	Reference
	Nu	Pe						
$Nu=Pe$	$\frac{\alpha_w d_p}{\lambda_0}$	$\frac{Gcd_p}{\lambda_0}$	0,5—100	Argon, helium	Stainless steel	0,31 0,28	100 350	[22]
$Nu=Pe^{0,235}$	$\frac{\alpha_w d_p}{\lambda_0}$	$\frac{Gcd_p}{\lambda_0}$	1,2—5,0	Air	Stainless steel	0,583	80	[23]

where α' is the heat-transfer coefficient when the flow moves in a channel having a porous wall and the coolant is sucked off through it with a mass flow rate G, and t_0 is the mean temperature over the cross section of the flow.

The third boundary condition for (2)-(5) does not have an obvious physical significance, and therefore there is a considerable arbitrary element in formulating it. We note the following major forms. In the foreign literature, an erroneous assumption (see below) is made that the temperatures are equal at the outer surface $T'' = t''$ for convective heating [1, 2], which can lead to an underestimate in the temperature of the porous material. An interesting point is that use of this assumption in [3] led to considerable difficulties even at the stage of formulation for a system of evaporative liquid porous cooling under conditions where a two-phase mixture was involved.

In [4], another condition was used, which has been shown to be erroneous [5]: the equality of the temperature gradients in the flow of coolant on both sides of the internal surface $dt/dZ|_{Z=0-0} = dt/dZ|_{Z=0+0}$. In [6], the relation $h_v(T'-t') = Gcdt/dZ$ was adopted for the internal surface, while in [7] use was made of $h_v(T'-t') = \lambda d^2T/dZ^2$. It follows from (1) that both of these are obeyed identically, and therefore they do not determine the coefficients C_1-C_3.

Often, an adiabatic condition is applied to the case of coolant supplied along the normal to the inner surface: $t' = t_0$, $dT/dZ = 0$ at $Z = 0$, and as a result one neglects the heating of the coolant before entry into the permeable wall [8-10]. However, it has been shown [11] that the rising gas temperature before entry can in some cases be a substantial fraction of the total. Even cruder assumptions were used in [12] at the inner surface ($Z = 0$ $t' = T' = \text{const}$).

An unusual pair of boundary conditions at the inner surface was proposed in [13] for longitudinal coolant supply: $\lambda dT/dZ = \alpha'(T'-t_0)$, $t' = t_0$ at $Z = 0$. Here there is no heating of the coolant withdrawn by suction until it enters into the wall, whereas heat is transferred convectively into the core of the flow from the wall. With analogous assumptions, $\lambda dt/dZ = \alpha'(T'-t_0)$, $t = t'$ at $Z = 0$ in [25], and then the gas temperature t' at the inlet differed from the initial value t_0, but no recommendations are made on determining it.

In some cases [14-16], the temperatures of the porous material at the outer and inner surfaces are specified. However, this requires preliminary measurements, which may actually be carried out for example to determine h_v, the rate of the convective heat transfer within the pores, but which are completely superfluous in preliminary calculations in order to select the optimum formal system.

Very often, two conditions are applied at the inner surface without relation to the mode of coolant supplied:

$$Z = 0 \quad \lambda dT/dZ = Gc\,(t' - t_0) = \alpha_w\,(T' - t_0). \tag{9}$$

Here one introduces the heat-transfer coefficient α_w between the coolant and the permeable matrix at the inlet to the wall, this being taken as given, although methods of calculating α_w are not stated [17-21].

We consider the justification for using condition (9) separately for the two forms of transverse and longitudinal gas supply. When the coolant moves along the normal to the porous wall, simultaneous use of both equations in (9) is equivalent to using not only (7), which has physical significance, but also a second condition, which may be put conveniently in the form

$$Z = 0 \quad (t' - t_0)/(T' - t_0) = \alpha_w/Gc = \mathrm{St}_w. \tag{10}$$

This defines the relative temperature rise in the coolant $(t'-t_0)/(T'-t_0)$ before arrival at the inner surface, which is determined by the so far unknown quantity α_w. There is an obvious constraint: $\mathrm{St}_w \leqslant 1$, $\alpha_w \leqslant Gc$.

We note at once that at present there are no theoretical methods of calculating α_w. When the coolant is supplied along the normal, it begins to accelerate at a distance of several times the pore diameter from the input surface in the direction of the nearest pore, which causes small-scale convective heat transfer. Therefore, α_w at the input surface incorporates the effects from this mechanism and those from thermal conduction in the incident flow.

Table 1 collates the published data on heat transfer at the input surface to a porous powdered metal. Figure 2 gives the reduction to the form of the boundary condition of (10). The experimental data are extremely scanty and show a considerable spread, so it is not possible to make any generalization from them. Nevertheless, it is evident that there is a reduction in the relative heating $(t'-t_0)/(T'-t_0) = \mathrm{St}_w$ as the mass flow rate increases.

When the coolant flows along the surface, it is possible to use (8), but it is erroneous to use both conditions of (9) simultaneously, since in that case the heat-balance equation of (8) can be put in the form $\lambda dT/dZ = \alpha'(T'-t_0) = Gc(t'-t_0) + \alpha^0(T'-t_0)$, i.e., the heat given out by the permeable surface $\alpha'(T'-t_0)$ is accounted for not only by the amount $Gc(t'-t_0)$ involved in heating the coolant entering it but also in raising the mean temperature t_0 (heating) by $\alpha^0(T'-t_0)$ for the entire longitudinal flow. Here α^0 is the heat-transfer coefficient from the porous wall to the remaining flow in the channel. The relation between these components varies with the parameters of the flow and suction, and the error from assuming that $Gc(t'-t_0) = \alpha'(T'-t_0)$ or $\alpha' = \alpha_w$ increases as the suction decreases and becomes particularly large for $G \to 0$, when $\alpha' \to \alpha$, where α_0' is the heat-transfer coefficient for an impermeable wall. Then the ratio $\alpha'/Gc = (t'-t_0)/(T'-t_0)$ may become much larger than 1, which is without physical significance. Any rise in t_0 along the permeable wall results in a reduction in the cooling performance (as has been observed for example with porous cooling of gas-turbine blades [24] and in heated tubes [28]), and this point must be borne in mind.

Therefore, in longitudinal flow one should use condition (8) at the inner surface along with (10), in which α_w is the effective heat-transfer coefficient from the permeable surface to the coolant entering it. At present, we do not have any information on the division of the overall heat-transfer coefficient α' into the components α^0 and α_w. The only dispersion of this type [28] is of qualitative character. To a first approximation, α_w can be taken as the heat-transfer coefficient from the permeable surface to the coolant moving normal to it (Fig. 2). However, there is additional heat transfer from the surface to the core of the longitudinal flow, and there is also the gradient in the longitudinal velocity near the permeable surface, which makes the analogy between the two processes very nominal, and it is difficult to say anything definite about the accuracy of this approximation.

We now examine the effects of boundary condition (10) on the temperature distribution in the porous wall. For this purpose, we consider the simple case of gas supplied along the normal to the surface (condition (7)), when the increment in the coolant temperature is

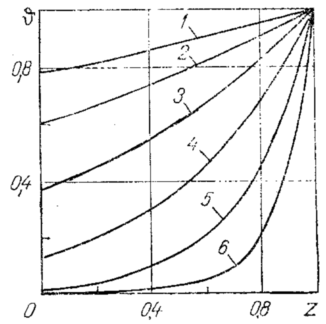

Fig. 3

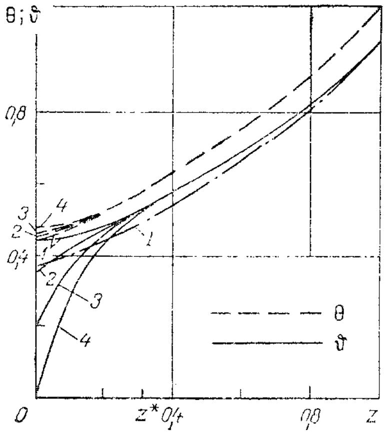

Fig. 4

Fig. 3. Effects of parameter B on the temperature distribution in a porous wall under conditions of local thermal equilibrium T = t ($A \to \infty$, $St_w \to 1$): 1) B = 0.25; 2) 0.5; 3) 1; 4) 2; 5) 4; 6) 8.

Fig. 4. Effects of heat-transfer rate at the input surface (parameter St_w) on the temperature distribution in a permeable wall for B = 1, A = 10: 1) St_w = 1.0; 2) 0.8; 3) 0.4; 4) 0.0; I) T = t (B = 1, $A \to \infty$, St_w = 1).

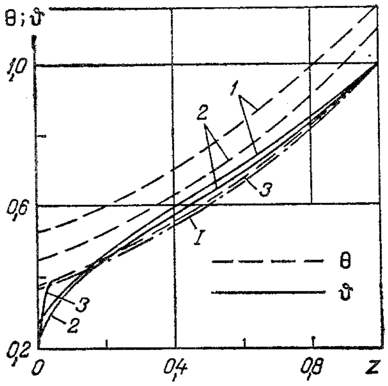

Fig. 5

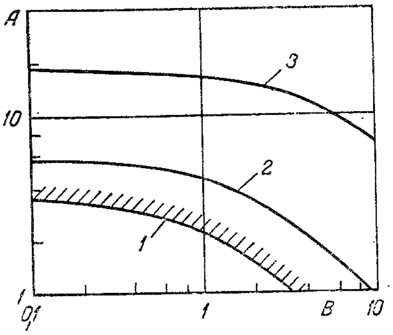

Fig. 6

Fig. 5. Effects of heat-transfer rate within pores (parameter A) on the temperature distribution in a permeable wall for B = 1, St_w = 0.5: 1) A = 5; 2) 10; 3) 100; I) see Fig. 4.

Fig. 6. Relation between parameters A and B such that the length z* of the inlet part showing thermal stabilization has a fixed value (ε = 0.025): 1) z* = 1; 2) 0.6; 3) 0.2.

determined by the heat-balance equation at the outer surface even for compound radiative and convective wall heating:

$$Z = \delta\ \lambda dT/dZ = q_r + \alpha''(t_\infty - T'') = Gc(t'' - t_0). \tag{11}$$

The solution to (3)-(4) is solved in the form

$$\vartheta = (t - t_0)/(t'' - t_0),\quad \Theta = (T - t_0)/(t'' - t_0) \tag{12}$$

with boundary conditions (7), (10), and (11), which can be written as

$$z = 0\quad d\Theta/dz = B\vartheta,\quad \vartheta = \mathrm{St}_w\Theta;$$
$$z = 1\quad d\Theta/dz = B. \tag{13}$$

Then the solution to (12) will be

$$\vartheta = (P\exp D_1 z - Q\exp D_2 z)/(P\exp D_1 - Q\exp D_2), \tag{14}$$

$$\Theta = [(1 + D_1/A) P \exp D_1 z - (1 + D_2/A) Q \exp D_2 z]/(P \exp D_1 - Q \exp D_2). \tag{15}$$

Here

$$P = 1 - \mathrm{St}_w (1 + D_2/A); \; Q = 1 - \mathrm{St}_w (1 + D_1/A). \tag{16}$$

The results of (14)-(16) are dependent on three parameters: B, A, and $\mathrm{St_w}$. Figures 3-5 show data enabling one to evaluate the effects of each.

The role of B is most evident in the limiting state of local thermal equilibrium T = t throughout the wall thickness. The solution

$$\Theta = \vartheta = \exp B (z - 1) \tag{17}$$

is derived from (14)-(16) as a particular case when two conditions are obeyed: $A \to \infty (D_1 \to B, D_2 \to -A, P \to 1)$; $\mathrm{St_w} \to 1$ $(Q \to 0)$. Figure 3 shows graphs derived from (17) for various values of B, while the dot-dash curve I in Figs. 4 and 5 applies for B = 1.

For these values of A and B, any change in $\mathrm{St_w}$ is hardly reflected in the distribution of the temperature Θ of the matrix, but there is a very large effect on the distribution of the temperature ϑ in the coolant (Fig. 4), although this occurs only within the inlet cold part of the wall, namely the zone of effect of the heat transfer at the inner surface or in the region of thermal stabilization (for $z < z^* = 0.31$ in Fig. 4). Outside this, the behavior of Θ and ϑ is determined only by the heat transfer within the pores (parameters A and B), which is the zone of stabilized heat transfer (for $z > z^*$). The main effect of $\mathrm{St_w}$ on Θ in the inlet part is due to the term $Q \exp D_2 z$ (14) because Q is a sign-varying function in (16) of $\mathrm{St_w}$. An interesting point is that Θ for the entire wall and ϑ in the stabilized heat-transfer zone will always be higher than the value corresponding to the limiting state of thermal equilibrium (curve I).

The role of parameter A becomes clear if (2) is written in the dimensionless form $\Theta - \vartheta = \frac{1}{A} \frac{d\vartheta}{dz}$, which implies that the temperature difference $\Theta - \vartheta$ decreases as A increases, with both temperatures tending upwards simultaneously towards curve I (Fig. 5). The inlet zone then becomes narrower (this is due to the increased role of heat transfer within the pores relative to heat transfer at the inelt), and there is increasingly sharp change in ϑ in it. The magnitude of the change decreases as $\mathrm{St_w}$ increases (Fig. 4) and in the limit $A \to \infty$, $\mathrm{St_w} \to 1$, the temperatures Θ and ϑ coincide with each other $\Theta = \vartheta$ and with curve I throughout the wall thickness, including at the inlet surface.

It is of interest to determine the length z^* of the thermal-stabilization zone. We estimate this form (14) as the distance from the inlet at which the perturbation in the coolant temperature ϑ due to heat transfer at the inner surface becomes small:

$$[(Q \exp D_2 z)/(P \exp D_1 z)]_{z=z^*} \leqslant \varepsilon. \tag{18}$$

Here ε is a preset small quantity. The coordinate z^* is maximal for $\mathrm{St_w} = 0$, when ϑ shows the largest change in the inlet region. In that case, we have P = Q = 1 from (16), and then (18) gives

$$z^* = \ln (1/\varepsilon) [A (1 + 4B/A)^{1/2}]^{-1}. \tag{19}$$

We may note curve 1 for $z^* = 1$ (Fig. 6), since it is the lower bound to the region denoted by the hatching of parameters A and B with which $Z^* < 1$. In that case, $\Theta - \vartheta$ at the outer surface (the temperature rise in the material) is independent of the heat-transfer rate at the inlet surface and is determined only by the heat transfer within the pores (by parameters A and B), and it takes the minimum value, which can be calculated from an equation derived from (14) and (15):

$$(\Theta - \vartheta)_{z=1} = D_1/A = \frac{1}{2} [-1 + (1 + 4B/A)^{1/2}]. \tag{20}$$

Then it follows that $(\Theta - \vartheta)_{z=1} = B/A$ for small B/A.

The data of Fig. 4 give rise to the following comments on the permissible values of $\mathrm{St_w}$ in this form of supplying the coolant along the normal to the inlet surface. An unrealistic case (for example, for $\mathrm{St_w} = 1$) is when the heat-transfer rate at the inlet is higher than the heat-transfer rate within the pores (if one can speak in this way of quantities differing in physical nature). There is a fixed value of St_w^0 at which the heat transfer at the input surface has the same effect on the temperature distribution in the stabilization zone as does heat transfer within the pores. This corresponds to the case where the trend in Θ and ϑ in the region of stabilized heat flow ($z > z^*$) as determined only by the heat transfer within the pores can be extended back to the input surface. This is so for $Q = 0$, so we have

$$St_w \leqslant St_w^0 = 2[1 + (1 + 4B/A)^{1/2}]^{-1}. \tag{21}$$

On this basis, we can recommend (7) and (10) or (8) and (10) for St_w in the range $St_w < St_w^0$ as boundary conditions at the inner surface of a porous wall, in accordance with the mode of coolant supply; these conditions are also applicable to a permeable component with bulk heat production.

Also, we note that when the coolant is supplied along the normal to the inner surface, (11) for heat balance at the outer surface enables one to eliminate the dependence of the temperature distribution within the porous wall on the physical nature of the external heat flux. As there is always a finite temperature difference $T''-t' > 0$ at the outer surface in this case (for example, from (20)), this indicates that the temperature equality $T'' = t''$ used in [1, 2] as the third boundary condition for convective heating is erroneous. This can also be demonstrated more rigorously. For example, if we use $T'' = t''$ together with (7) and (11), we get the physically unreal result $\Theta - \vartheta < 0$ for all points within the permeable wall apart from the outer surface.

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GENERALIZATION OF DATA ON THE SECOND VIRIAL COEFFICIENT OF GASES

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UDC 536.71

Generalization of empirical data for more than 30 substances (see Table 1) resulted in a relation for the second virial coefficient of normal substances. A single-parameter generalized law for the respective states is discussed.

Knowledge of general principles characterizing the behavior of the second virial coefficient of a large class of substances is necessary to obtain a single equation of state, to substantiate model potentials for the interaction of polyatomic molecules, and to determine the role of multiple-particle reactions in fluids. Two studies which have generalized data on the second virial coefficient are known. In [1], such a generalization was made for the class of normal (nonassociated) substances using a single-parameter generalized law for the corresponding states:

$$\frac{B}{V_B} = F_0\left(\frac{T_B}{T}\right) + \omega F_1\left(\frac{T_B}{T}\right), \tag{1}$$

where T_B and V_B are correction parameters; T_B is Boyle's temperature $(B(T_B) = 0)$; V_B is a characteristic volume and is determined by the formula

$$V_B = T_B\left(\frac{\partial B}{\partial T}\right)_{T=T_B}. \tag{2}$$

The quantity ω in Eq. (1) is a unique characteristic of the substance — Pitzer's governing similitude criterion ("the acentric factor"). In [2], the generalization was done on the basis of broader empirical data: the author used new data for Ar and Kr ($\omega = 0$), n-butane

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Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 47, No. 4, pp. 595-601, October, 1984. Original article submitted July 25, 1983.

0022-0841/84/4704-1176$08.50