HYDROMECHANICS OF THE INLET OF HEAT-RELEASING APPARATUS

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The article investigates the motion of liquid in an inlet with spiral channel, and a method is devised for calculating the design parameters of the inlet.

Improving the efficiency of heat-releasing apparatus used in power engineering, heat engineering, and chemical technology entails the use of granular material with an internal heat source (catalyst) arranged in the form of annular layers, with the heat carrier radially pumped through. A diagram of such an apparatus is shown in Fig. 1.

The existence of several pseudostable states of gas filtration in the heat-releasing layer [1] and possible uncontrolled transitions from one such state to another at the time of operation [1-3] hinder the organization of uniform distribution of the heat carrier over the circumference of the layer which would correspond to optimal heat removal.

If instead of a layer, a special inlet with a spiral channel is used for distributing the gas stream, the problem of uniform gas distribution may be solved; in that case the apparatus remains compact, and its reliability is improved.

We will examine the operation of the inlet whose porous wall has an outer radius that is much larger than the maximum equivalent hydraulic diameter of the channel. This constraint enables us to neglect the hydrodynamic effects connected with the curvature of the main line, and justifies the utilization of extreme value results obtained for straight penetrable channels [4].

We write the equation of motion in the projection onto the S-axis in integral form which is expedient for further transformations:

$$\oint_{\Sigma}\rho V_j\left(\frac{1}{2}V_s^2\right)d\Sigma_j=-\int_{\tau}V_s\frac{\partial}{\partial x_j}(P\delta_{sj}-\sigma_{sj})\,d\tau. \tag{1}$$

As the closed surface Σ we select the full surface of an elementary volume $\tau = Fds$. If we average the flow parameters in expression (1) with respect to F, we obtain the linear equation

$$-\frac{1}{\rho}\frac{dP}{ds}=\frac{1}{G}\frac{dG}{ds}\mathrm{K}\langle V_s\rangle^2+\mathrm{K}\frac{d}{ds}\langle V_s\rangle^2+\langle V_s\rangle^2\frac{d}{ds}\mathrm{K}-$$
$$-\frac{1}{2G}\frac{dG}{ds}(V_s^{*2})+\frac{\xi\langle V_s\rangle^2}{2D};\quad \mathrm{K}=\frac{1}{2}+\frac{3}{2}\langle[f(r)-1]^2\rangle, \tag{2}$$

which describes the gas motion with variable flow rate along the path in a channel with arbitrary cross section.

In deriving Eq. (2) it was assumed that the change of the normalized profile of the longitudinal component of the velocity vector along the channel is slight, i.e., that $V_S = \langle V_S\rangle f(r)$. The correctness of this assumption was demonstrated in [5-7]. The change of pressure, associated with dissipative losses, is taken into account by the term with the coefficient ξ.

With developed turbulent flow of an incompressible liquid and the dominant value of the radial component of the pressure gradient on the penetrable wall ($\partial P^*/\partial r \gg dP/ds$) Eq. (2) is greatly simplified; in polar coordinates it has the form

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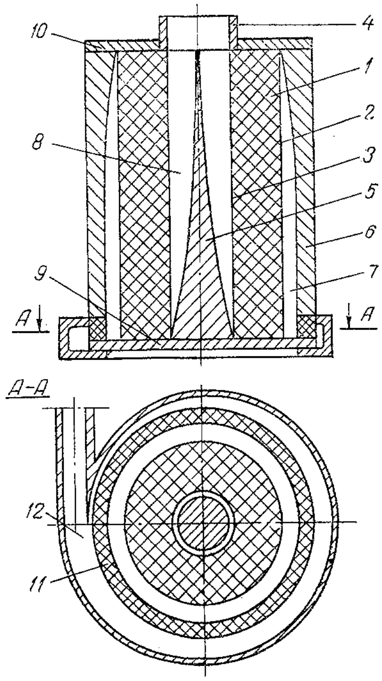

Fig. 1. Diagram of apparatus with heat releasing granular layer (longitudinal and cross sections): 1) heat-releasing layer; 2) outer guarding grid; 3) inner guarding grid; 4) outlet pipe; 5) displacer rod; 6) cowl; 7) distributor channel; 8) discharge channel; 9) protective disk; 10) protective washer; 11) porous wall; 12) inlet main line.

$$\frac{3}{2}\frac{G}{\rho F^2}\frac{dG}{d\varphi}-\frac{G^2}{\rho F^3}\frac{dF}{d\varphi}+\xi\frac{G^2R}{2\rho F^2D}=-\frac{dP}{d\varphi}. \tag{3}$$

We take it that the motion of the heat carrier in the porous wall is close to being radial, and that it obeys the square law of resistance

$$\nabla P=-\rho\frac{1.7(1-\varepsilon)}{\varepsilon^3 d}|\mathbf{V}|\mathbf{V}. \tag{4}$$

If we integrate (4), we obtain the formula for determining pressure losses in the porous wall:

$$\Delta P=\frac{1}{\rho}\left(\frac{dG}{d\varphi}\right)^2\Phi;\quad \Phi=\frac{1.7(1-\varepsilon)}{\varepsilon^3 dl^2}\left(\frac{1}{R_1}-\frac{1}{R}\right). \tag{5}$$

The condition of dynamic matching of the system inlet main line–porous wall–distributor channel is expressed by the equation

$$\frac{dP}{d\varphi}-\frac{dP_1}{d\varphi}=\frac{d}{d\varphi}\Delta P, \tag{6}$$

which in kinematic variables [in accordance with (3) and (5)] has the form

$$-\frac{3}{2}\frac{G}{\rho F^2}\frac{dG}{d\varphi}+\frac{G^2}{\rho F^3}\frac{dF}{d\varphi}-\frac{\xi}{D}\frac{G^2R}{2\rho F^2}-\frac{2}{\rho}\left(\frac{dG}{d\varphi}\right)\frac{d^2G}{d\varphi^2}\Phi=\frac{dP_1}{d\varphi}. \tag{7}$$

When the stream is uniformly distributed, the flow rate of the heat carrier in the inlet main line is described by the linear dependence

$$G = G_0\left(B - \frac{\varphi}{2\pi}\right), \tag{8}$$

and the pressure gradient in the porous wall is constant (for brevity, the parameter B is called the circulation frequency). In that case the gradientless flow in the main line is a condition of uniform distribution of the heat carrier; it shows that the approximation of an incompressible liquid is correct for the calculation of the shape of the inlet channel when a gaseous heat carrier is introduced.

If we substitute expression (8) into (7) and put P_1 = const, we obtain the regularity of the change of the useful main line section of the inlet:

$$\frac{\xi}{D}\,\frac{(B-\varphi/2\pi)\,R}{2} - \frac{(B-\varphi/2\pi)}{F}\,\frac{dF}{d\varphi} = \frac{3}{4\pi};\quad F|_{\varphi=0} = F_0. \tag{9}$$

If the distribution of heat carrier is to be uniform while the apparatus operates at different levels of power and flow rate, self-similar flow [$\xi \neq f(\mathrm{Re})$] in the inlet main line is indispensible. It is attained by artificially roughing the wetted surfaces and by the turbulizing effect on the exhaust stream [8, 9]. With such flow, the circulation frequency depends only on the shape of the main line, and therefore in inlets with large values of the parameter B, self-similar flow exists even when G_0 is small.

We determined experimentally the coefficient of hydraulic resistance

$$\xi = 10{**}\,[-1.41 - 0.97\log_2(D/60\Delta)] \tag{10}$$

and the lowest flow rate of heat carrier at which the noted flow regime still exists [4]:

$$\inf G_0 \leqslant \frac{\nu\rho F_0}{BD}\,[10\,{**}\,(5.27 - 0.35\lg 100\,\xi)]|_{\varphi=0}. \tag{11}$$

The solution of (9) for the case $\xi = \xi(\varphi)$ has the form

$$F = F_0 \exp\int_0^{\varphi}\left[\frac{\xi R}{2D} - \frac{3}{4\pi}\,\frac{1}{(B-\varphi/2\pi)}\right]d\varphi; \tag{12}$$

from below it is bounded by the function

$$\inf F = F_0\left(1 - \frac{\varphi}{2B\pi}\right)^{3/2}.$$

To avoid the negative effects associated with the detachment of flow in the inlet main line, its useful section has to become narrower in the direction in which the heat carrier flows. This requirement imposes constraints on the design parameters of the inlet:

$$0 < \frac{\xi R}{D} \leqslant \frac{3}{2\pi}\,\frac{1}{(B-\varphi/2\pi)}. \tag{13}$$

We substitute expression (12) into Eq. (7) and put P_1 = const. Let us analyze the obtained equation with the conditions on the boundary:

$$G|_{\varphi=0} = BG_0;\quad G|_{\varphi=2\pi} = (B-1)\,G_0. \tag{14}$$

It is simple to prove that there exists a solution of the boundary problem and that it is unique, and also that it is stable, i.e., that the function $\delta G/\delta F$ is bounded. This testifies to the correctness of the statement of the problem of seeking the shape of the inlet main line.

Random perturbations in energy release cause local fluctuations of the mass velocity of filtration [1, 2] and disturb the uniform distribution of the stream over the circumference of the heat-releasing layer, i.e., $\partial V_r/\partial\varphi \neq 0$; in this case the tangential component of the pressure gradient in the distributor channel is also nonzero, and this leads to curving of the stream.

It is of practical importance that the dynamic perturbations connected with fluctuations of energy release, propagating upstream, be attenuated under the effect of the inlet.

Within the framework of the linear theory of perturbations we investigate the effect of dynamic oscillations with infinite period, originating in the distributor channel, on the operation of the inlet. For this purpose we transform Eq. (7) and the boundary conditions (14) for perturbed flow into dimensionless form, and then we linearize them, bearing (8) and (12) in mind. (As scale we chose the parameters of unperturbed flow.) As a result of this operation we obtain the differential equation

$$\frac{d\delta z}{d\eta} = -\frac{a}{\psi^2}\frac{d}{d\eta}\left(\frac{\delta g}{B-\eta}\right) + b\frac{d^2\delta g}{d\eta^2};$$
$$a = \frac{3G_0^2}{2\rho P_1 F_0^2}; \quad b = \frac{G_0^2\Phi}{2\pi^2\rho P_1}; \quad z = \frac{P_1^+}{P_r}; \quad g = \frac{G}{G_0}; \tag{15}$$
$$\eta = \frac{\varphi}{2\pi}; \; \psi = \frac{1}{(B-\eta)}\exp\int_0^\eta\left[\frac{\xi R}{2D} - \frac{3}{4\pi}\frac{1}{(B-\eta)}\right]d\eta$$

with the boundary conditions

$$\delta g|_{\eta=0} = B\delta g_0; \quad \delta g|_{\eta=1} = (B-1)\,\delta g_0, \tag{16}$$

describing the reaction of the stream δg to the dynamic perturbation δz.

The solution of (15) and (16) has the form

$$\delta g = (B-\eta)\left\{\int\left[C_1A^+ + A^+\int\frac{A^-}{b(B-\eta)}\frac{d\delta z}{d\eta}d\eta\right]d\eta + C_2\right\}; \tag{17}$$

here,

$$A^+ = \exp\int\left(2 + \frac{a}{\psi^2 b}\right)\frac{d\eta}{(B-\eta)}; \quad A^- = 1/A^+; \quad N(n) = \int A^+ d\eta|_{\eta=n};$$
$$M(n) = \int A^+\left(\int\frac{A^-}{b(B-\eta)}\frac{d\delta z}{d\eta}d\eta\right)d\eta\Big|_{\eta=n}; \quad n = (0 \vee 1); \tag{18}$$
$$C_1 = \frac{M(1)-M(0)}{N(0)-N(1)}; \quad C_2 = \delta g_0 - C_1N(0) - M(0).$$

We add to relation (18) the condition correlating the variation of the flow rate of the heat carrier supplied to the apparatus with the dynamic perturbations δz:

$$\delta g_0 = \frac{P_1}{U}\left[\delta z(0) - b\frac{d\delta g}{d\eta}\Big|_{\eta=0}\right]. \tag{19}$$

Relation (19) was obtained by linearization of the equation of dynamic matching in integral form and of the formula for determining the power of the pump required for pumping the heat carrier through: $Q = UG_0/\rho$. (It is understood that the pump has an independent drive, and therefore $\delta Q = 0$.)

We differentiate expression (17) with respect to the dimensionless coordinate η, and the obtained result

$$\frac{d\delta g}{d\eta} = -C_1N(\eta) - M(\eta) - \delta g_0 + C_1N(0) + M(0) + (B-\eta)K(\eta);$$
$$K(\eta) = A^+\left(C_1 + \int\frac{A^-}{b(B-\eta)}\frac{d\delta z}{d\eta}d\eta\right) \tag{20}$$

for $\eta = 0$ is substituted into (19) and resolved with respect to δg_0; as a result we obtain

$$\delta g_0 = \frac{P_1}{U-bP_1}[\delta z(0) + bBK(0)]. \tag{21}$$

Equations (17), (18), (20), and (21) describe fully the behavior of a stream of heat carrier in the inlet when there are dynamic perturbations.

We now investigate the dynamic stability of the inlet. For that purpose we compile and analyze the system of characteristic equations

$$\int_0^1 \frac{d\delta z}{d\eta} d\eta = 0; \quad \int_0^1 \frac{1}{P_1} \frac{d\delta P^+}{d\eta} d\eta = 0, \tag{22}$$

which expresses the continuity of the pressure fields in the distributor channel and in the main line of the inlet. Basing ourselves on expressions (5) and (15), and on the variation of Eq. (6), we transform the system of equations (22) into the form

$$\int_0^1 \frac{d\delta z}{d\eta} d\eta = 0; \quad \left.\frac{d\delta g}{d\eta}\right|_0^1 = 0. \tag{23}$$

From (22) and (23) we obtain that the requirement of continuity of the pressure field is equivalent to the condition of continuity of the velocity field in the porous wall.

In accordance with (18), (20), and (21), the system (23) is equivalent to

$$\int_0^1 \frac{d\delta z}{d\eta} d\eta = 0; \quad (B-1)K(1) = BK(0). \tag{24}$$

The intersection of the sets of solutions satisfying separately each of the integral equations of system (24) is the trivial solution $d\delta z/d\eta = 0$; this follows from the properties of the integral equations, the determination of the value of K [cf. (20)], and expressions (18).

Thus the inlet damps the dynamic perturbations of the variable ($\delta z \neq$ const); their origin becomes improbable. An ordinary receiver is a passive object and does not hinder the origin and propagation of perturbations. Therefore a heat-releasing apparatus with an inlet whose main line has spiral shape is of increased reliability in operation.

When the installation is in operation for a long time, the roughness of the main line walls may be worn off, and the result would be distortion of the uniform distribution of heat carrier. The problem arises: At what function $\Delta = \Delta(\varphi)$ does the wearing off of roughness cause minimum distortion of the uniform distribution of heat carrier? Let us solve it.

We vary Eq. (7) and the boundary conditions (14) with respect to flow rate, pressure, and the function $\xi R/D$; when we substitute expressions (8) and (12) into the result and go over to a dimensionless magnitudes, we obtain the boundary problem

$$\frac{F_0^2 P_1 \rho}{G_0^2} \frac{d\delta z}{d\eta} + \frac{\frac{\xi R}{D}\delta m}{\psi^2} = -\frac{a}{\psi^2} \frac{d}{d\eta}\left(\frac{\delta g}{B-\eta}\right) + b\frac{d^2\delta g}{d\eta^2}; \tag{25}$$
$$\delta g|_{\eta=1} = Bg_0; \quad \delta g|_{\eta=1} = (B-1)\,\delta g_0.$$

In approximation to creeping flow in the distributor channel near the porous wall, the value of $d\delta z/d\eta$ is determined from the equation

$$\frac{d\delta z}{d\eta} = T\frac{d^2\delta g}{d\eta^2}; \quad T = \frac{\nu G_0}{\pi R_1^2 l P_1}. \tag{26}$$

When we substitute expression (26) into (25), we obtain an equation whose solution is analogous to the problem (15), (16).

We maintain the supply of heat carrier to the apparatus constant (by changing the power on the pump shaft); the requirements as to inertia of the system ensuring $\delta g_0 = 0$ are not stringent because the process of wearing off of roughness is not a short-lived one.

We differentiate the solution of the system (25), (26) $\delta g = \delta g(\eta)$ for $\delta g_0 = 0$ with respect to the coordinate η; in the limit transition, when $\delta m \to 0$, we obtain the relation

$$\left|\frac{d\delta g}{d\eta}\right| \leqslant \left|E^+ \int \frac{E^- W(\eta)}{Y(B-\eta)} d\eta\right| \left|\frac{\delta m}{\ln|\delta m|}\right|; \tag{27}$$

$$E^{+}=\exp\int\left(2+\frac{a}{\psi^{2}Y}\right)\frac{d\eta}{(B-\eta)};\ E^{-}=1/E^{+};\ W(\eta)=\frac{\xi R}{D}/\psi^{2};$$

$$Y=b-\frac{TF_{0}^{2}P_{1}\rho}{G_{0}^{2}}.$$

The value of $\xi R/D$ is determined as a function minimizing the functional:

$$\int_{0}^{1}\left(E^{+}\int\frac{E^{-}W(\eta)}{(B-\eta)Y}d\eta\right)^{2}d\eta=\min; \tag{28}$$

in this case it is to be expected that wearing off of roughness will lead to minimal distortion of the uniform distribution of heat carrier.

The method of calculating the design parameters of the inlet reduces to the solution of the system of transcendental equations:

$$\begin{aligned}
&\inf G_{0}=\frac{\nu\rho(l^{2}+F_{0})}{2Bl}\left[10{*}{*}\left(5.27-0.35\lg\frac{300BF_{0}l}{\pi R(l^{2}+F_{0})}\right)\right];\\
&\Delta P=\frac{1.7(1-\varepsilon)}{\varepsilon^{3}\rho dl^{2}}\left(\frac{\sup G_{0}}{2\pi}\right)^{2}\left(\frac{1}{R_{1}}-\frac{1}{R}\right);\quad\frac{0.85(1-\varepsilon)}{\varepsilon^{3}d(\pi Rl)^{2}\Lambda}=14.3;\\
&B=\frac{1}{1-\exp\int_{0}^{1}\left[\frac{\xi R}{2D}-\frac{3}{4\pi(B-\eta)}\right]d\eta};\\
&\frac{\inf G_{0}C}{2\pi\rho Rl\nu}=3.35\cdot10^{4};\quad C=1\,\mathrm{m},
\end{aligned} \tag{29}$$

supplemented by relations (10), (12) jointly with the variational problem (28) with natural conditions at the boundary and the existence of the constraint (13); the equations of system (29) express the condition of self-similar flow in the main line of the inlet, the requirement to ensure the specified pressure losses, the absence of negative effects caused by vortex formation in the distributor channel [4, 10, 11], the condition of optimum merging of the streams ($\langle V_s\rangle|_{\varphi=0}=\langle V_s\rangle|_{\varphi=2\pi}$), the endeavor to attain the smallest possible contamination with erosion products of the porous wall; all this in the present case is equivalent to the requirement of minimum volume fraction of dead zones in the porous wall [12].

The initial data for the calculation are: nominal flow rate of heat carrier (G_0) and the range of its change ($\inf G_0$, $\sup G_0$), pressure losses (ΔP), the profiling function (Λ), the properties of the heat carrier (ρ, ν), the radius of the inner surface of the porous wall (R_1) (it is equated to the inner radius of the cowl at the entrance to the distributor channel).

As a result of the calculation we obtain: the outer radius of the porous wall (R), its height (l) and equivalent pore diameter (d), the useful section of the inlet main line [$F=F(\varphi)$], the height of the protuberances of artifical roughness [$\Delta=\Delta(\varphi)$], and the circulation frequency B.

If the roughness is specified, the method of calculating the inlet is considerably simplified, the equations can be solved with the aid of desk calculators. The correctness of the results obtained in this article was confirmed experimentally by the authors.

NOTATION

ρ, density of the heat carrier; V, velocity vector; P, static pressure; δ_{ij}, Kronecker delta; σ, tensor of viscous stresses; x_j, coordinate; F, useful section; G, flow rate of heat carrier; ξ, coefficient of hydraulic losses; D, equivalent hydraulic diameter; φ, polar angle; ε, porosity of the wall; d, equivalent pore diameter; l, height of the porous wall; R, R_1, outer and inner radius, respectively, of the porous wall; ΔP, pressure gradient in the porous wall; G_0, flow rate of heat carrier supplied to the apparatus; B, circulation frequency; Re, Reynolds number; Δ, height of protrusions (surface roughness); Q, required pump power; P_1^+, P_1, static pressure of the perturbed and unperturbed flows, respectively, in the distributor channel; δm, relative variation of the function $\xi R/D$; Λ, profiling function; $**$, sign of raising to a power; ν, kinematic viscosity; U, gradient of pressure induced by the pump; $*$, magnitudes on the penetrated main line wall; +, parameters of perturbed flow; l, magnitude relating to the distributor channel; l, to the magnitudes relating to the porous wall and the inlet main line a subscript was not assigned.

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EFFECT OF THE INLET TEMPERATURE ON ENERGY DISTRIBUTION IN EDDY THERMOTRANSFORMERS

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The article presents the results of the experimental investigation of the effect of energy separation in eddy thermotransformers with inlet temperatures of the working substance $500 \leq T_1^* \leq 2000°K$.

Rank's eddy thermotransformers, operating at inlet temperatures $T_1^* > 500°K$, are used in aircraft, rocket, and power engineering for solving a number of specific problems [1]. At such temperatures the magnitude of the effect of thermotransformation may be greatly influenced by the variability of the thermophysical properties of the working substance. To find more accurate characteristics of high-temperature eddy thermotransformers, experimental investigations were carried out with a carefully insulated cylindrical pipe with diameter $d_{pi} = 20$ mm and relative length $l = L/d_{pi}$ equal to 9 times the bore, and with a rectification cross at the hot end.

The experimental setup was a gas generator which made it possible to ascertain the flow rate of the working substance with specified parameters being fed to the inlet of the swirler of the eddy thermotransformer.

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