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A RECUPERATOR WITH A MAGNETORHEOLOGICAL COOLANT

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We have evaluated the effectiveness of a "tube-within-a-tube"-type recuperator with the coolant based on a magnetorheological suspension, and we have performed the calculations.

It had earlier been established [1] that when a weakly concentrated magnetorheological suspension (MRS) of noncolloidal particles of a ferromagnetic carbonyl iron is involved in turbulent flow through a tube, and acted upon by a uniform magnetic field oriented perpendicular to the mean-velocity vector, significant intensification of heat transfer to the wall of the tube occurs, as well as an increase in the hydraulic resistance that is a function of the magnitude of field strength. The derived empirical relationships linking the heat-transfer coefficient and the coefficient of hydraulic resistance to the flow parameters of the MRS in the field are of the following form:

$$\alpha_H = \alpha_0 (1 + 7.5\mathrm{Al}'),\ \xi_H = \xi_0 (1 + 13.5\mathrm{Al}'). \tag{1}$$

Here $\mathrm{Al}' = \mu_0 H^2 \varphi / \rho w^2$ is the modified Alfvén number [2]. Formulas (1) are valid for the interval $0 \le \mathrm{Al}' \le 1.2$, which corresponds to the range of variations $0 \le H \le 3.2 \cdot 10^5$ A/m; 1 m/sec $\le w \le 2.4$ m/sec; $0 \le \varphi \le 0.01$.

Application of the observed effect offers the possibility of developing special heat exchangers with capabilities impossible in traditional equipment or, at least, difficult to achieve, and namely, the operational control of the basic working characteristics (the temperature difference in the heat carriers, the output of heat) without altering the flow rate of the heat carrier through a regulated increase in the heat-transfer coefficient.

Below we present the results from an estimate of the efficiency of a recuperator with a magnetorheological heat coolant (MRC). In this case, the basis for the comparison was pro-

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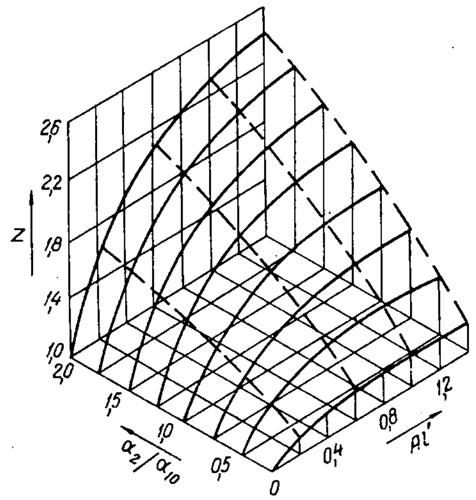

Fig. 1. Diagram showing change in the quantity Z as a function of Al' and α_2/α_{10}.

vided by the characteristics of the most simple "tube-within-a-tube"-type heat exchanger. To design such a heat exchanger with an ordinary coolant such as, for example, water, we might make use of the relationships (2.12), (2.19)-(2.24), presented in [3].

Let us assume that the initial coolant was an MRS and that the exchange of heat is taking place within a magnetic field. When we take (1) into consideration, expression (2.12a) from [3] assumes the form

$$k_H = \frac{1}{\dfrac{d_{av}}{d_1\alpha_{10}(1+7.5\mathrm{Al}')} + \dfrac{d_{av}}{2\lambda_w}\ln\dfrac{d_2}{d_1} + \dfrac{d_{av}}{d_2\alpha_2}}. \tag{2}$$

Let us introduce some function $Z(\mathrm{Al}', \alpha_2/\alpha_{10})$, representing the ratio of the heat-transfer coefficients in the case of heat exchange, both with and without a field:

$$Z = k_H/k_0. \tag{3}$$

It was demonstrated in [1] that α_{10}, k_0, and λ_0 are equal in magnitude to the coefficients under the conditions in which the two water coolants are used.

Let us replace the k in expression (2.20a) from [3] by k_H and let us substitute the resulting expressions for v_{xH} and v_H into Eqs. (2.19), (2.20), (2.21), (2.22), and (2.24a) [3], which assume the following form:

a) direct flow

$$\frac{T_1 - T_1'}{T_2' - T_1'} = \frac{1 - \exp[-Zv_x(1+R_{12})]}{1+R_{12}}, \tag{4}$$

$$\frac{T_2' - T_2}{T_2' - T_1'} = \frac{R_{12}}{1+R_{12}}\{1 - \exp[-Zv_x(1+R_{12})]\}, \tag{5}$$

b) counterflow

$$\frac{T_1 - T_2'}{T_2' - T_1'} = \frac{1 - \exp[-Zv_x(1-R_{12})]}{1 - R_{12}\exp[-Zv(1-R_{12})]}, \tag{6}$$

$$\frac{T_2 - T_2'}{T_2' - T_1'} = \frac{R_{12}\{1 - \exp[-Zv_x(1-R_{12})]\}}{1 - R_{12}\exp[-Zv(1-R_{12})]}, \tag{7}$$

$$Q = Zk_0F_\Sigma\Delta T_{av} \tag{8}$$

Figure 1 shows a diagram constructed on the basis of relationship (3) to reflect the change in Z in terms of the parameters which determine the process of heat exchange within a recuperator using MRC. The greatest intensification in heat exchange should be expected with regimes with $\alpha_2 \gg \alpha_{10}$, when the principal component out of three is the thermal resistance that is due to the transfer of heat from the first coolant to the separating wall.

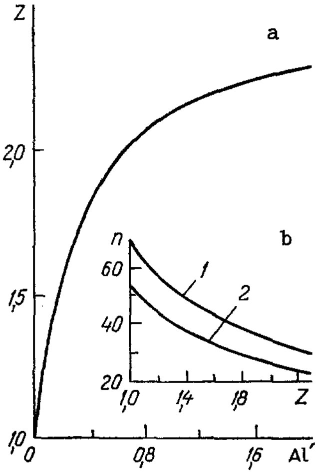

Fig. 2

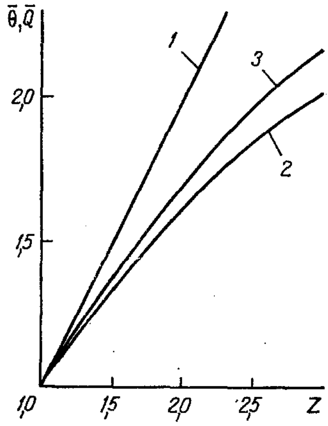

Fig. 3

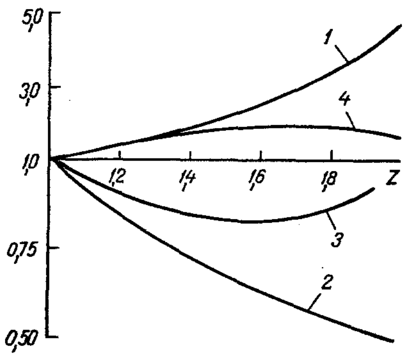

Fig. 4

Fig. 2. Intensification of heat transfer in a magnetic field (a); change in the quantity of individual heat-exchange sections providing the required output of heat (b): 1) direct flow; 2) counterflow. n, piece.

Fig. 3. Rise in heat output (1) and heat drops in the magnetic field (2, 3): 2) direct flow; 3) counterflow.

Fig. 4. Effective operation of the recuperator with MRC: 1) $\Delta\bar{P}_1$; 2) $\Delta\bar{P}_2$; 3) $\bar{W}$; 4) E.

As an example, let us take an industrial heat-exchange section of the "tube-within-a-tube" type exhibiting the following structural dimensions: $d_1 = 57 \cdot 10^{-3}$ m, $d_2 = 64 \cdot 10^{-3}$ m, $d_3 = 89 \cdot 10^{-3}$ m, $d_{av} = 60.5 \cdot 10^{-3}$ m, L = 6 m, F = 0.114 m³, $\lambda_w = 85$ W/(m·deg).

Let us assume that this section is fabricated out of nonmagnetic material (brass), with a pressure head adequate to develop the required rate of flow, and let us further assume that the losses of heat are insignificant and that there is no change in the overall state of the heat carriers.

Structural Design. Let us specify the required initial data: $w_1 = 1.0$ m/sec, $\alpha_{10} = 2440$ W/(m²·deg) ($\alpha_{10} = 0.023\,\mathrm{Re}^{0.8}\mathrm{Pr}^{0.43}\lambda_0/d_1$ [4]), $T_1' = 20°C$, $T_1'' = 85°C$, $T_2' = 150°C$, $T_2'' = 110°C$. From the familiar relationships from [3, 4] we find that $Q = 7 \cdot 10^5$ W, $\alpha_2 = 4030$ W/(m²·deg), $k_0 = 1380$ W/(m²·deg), $\Delta T_{av\uparrow\uparrow} = 63.7°C$, $\Delta T_{av\uparrow\downarrow} = 76.8°C$, $F_{\Sigma\uparrow\uparrow} = 8.0$ m², $F_{\Sigma\uparrow\downarrow} = 6.6$ m², $n_{\uparrow\uparrow} = 68$, $n_{\uparrow\downarrow} = 58$.

For the case for which the exchange of heat occurs within a magnetic field, we can use (2) and (3) to determine the relationship Z = Z(Al') for the quantities of α_2 and α_{10}. Graphically, we can find a representation of this relationship in Fig. 2a. Figure 2b shows the change in the number of required heat-exchange sections to ensure the rigid output of heat for both of the heat-carrier circuits.

Verification. In this case we know T_1' and T_2', k_0, and we have also obtained T_{10}'', T_{20}'' and Q_0. Let us trace the manner in which the field affects the calculated quantities. In order to accomplish this, we will separate relationships (4)-(7) into the corresponding quantities (2.19)-(2.22) [3], replacing T_1, T_2 in these by T_1'', T_2'', v_x by v and denoting $(T_1'' - T_1')/(T_2' - T_1') = \theta_1$, $(T_2' - T_2'')/(T_2' - T_1') = \theta_2$. We obtain the following expressions:

$$\bar{\theta}_{\uparrow\uparrow} = \frac{\theta_{1H}}{\theta_{10}} = \frac{\theta_{2H}}{\theta_{20}} = \frac{1-\exp[-Zv(1+R_{12})]}{1-\exp[-v(1+R_{12})]},$$

$$\bar{\theta}_{\uparrow\downarrow} = \frac{\{1-\exp[-Zv(1-R_{12})]\}\{1-R_{12}\exp[-v(1-R_{12})]\}}{\{1-R_{12}\exp[-Zv(1-R_{12})]\}\{1-\exp[-v(1-R_{12})]\}}, \tag{9}$$

$$\bar{Q} = Q_H/Q_0 = Z.$$

From (2.20a) [3], with consideration of the aforementioned technical parameters of the heat-exchange section and the data from [1] we will calculate the values of the complexes in the equations and we will find the relationship by which the ratios of the dimensionless temperature differences and the heat flow are dependent on Z, as shown in Fig. 3. With a fixed heat-exchange area in the field, the heat output of the recuperator increases with a corresponding increase in the temperature differences between the heat carriers.

Operational Efficiency. We will evaluate the efficiency of the heat-exchange recuperative sections by comparing the intensification of the heat transfer within the magnetic field as the energy changes to the pumping of the heat carriers. (The expenditures on the development of the magnetic field in this stage have not been examined.)

The work performed by the external forces in unit time spent in overcoming the hydraulic resistance of the recuperator to the MRC flow is written in the form $W_1 = G_1\Delta P_1$.

The total energy required for the pumping of both of the heat carriers is equal to $W = G_1\Delta P_1 + G_2\Delta P_2$, while the relative change of the energy in the magnetic field

$$\bar{W} = \frac{G_1\Delta P_{1H} + G_2\Delta P_{2H}}{G_1\Delta P_{10} + G_2\Delta P_{20}}. \tag{10}$$

The energy will undergo change both as a consequence of the increase in the MRC hydraulic resistance within the field, as well as because of the drop in the hydraulic resistance of the nonmagnetic second heat carrier, which comes about because of the reduction in the number of recuperation sections (in accordance with Fig. 2b).

The sought estimate of the efficiency can be found from the expression

$$\mathrm{E} = \frac{k_H(G_1\Delta P_{10} + G_2\Delta P_{20})}{k_0(G_1\Delta P_{1H} + G_2\Delta P_{2H})} = \frac{Z}{\bar{W}}. \tag{11}$$

Figure 4 illustrates how the quantity E changes with the increase in Z, i.e., with a strengthening of the field effect and with intensification of heat transfer. We examine the optimum regime of MRC motion for the case in which Z = 1.8, which corresponds to Al' = 0.4. Here we also find a relationship $\bar{W}(Z)$, $\overline{\Delta P_1}(Z)$, and $\overline{\Delta P_2}(Z)$, where $\overline{\Delta P_1} = \Delta P_{1H}/\Delta P_{10}$, $\overline{\Delta P_2} = \Delta P_{2H}/\Delta P_{20}$, with the quantity ΔP_{1H} determined through consideration of (1).

In conclusion, let us note that the synthesis of the MRC, apparently, should be carried out immediately in front of the heat-exchange zone by installing an iron-powder metering device at the inlet to the recuperator, and a separator at the outlet from the recuperator, so as to produce recirculation of the particles (as this was done in the experiments [1]). As regards the large-scale industrial application of this heat-exchange process, it is hardly of any use at the present time; however, in certain special cases such as, for example, in the chemical industry, it is our opinion that the utilization of MRC to raise the efficiency of the technological cycle is exceedingly promising.

NOTATION

α, the coefficient of heat transfer; ξ, the coefficient of hydraulic resistance; $\mu_0 = 1.256\cdot10^{-6}$ H/m is the magnetic constant; H, the magnetic field strength; φ, the volumetric concentration of particles in the MRS; ρ, the density of the MRS; w, the mean flow velocity; k, coefficient of heat transfer; λ, coefficient of heat-carrier thermal conductivity; λ_w, the coefficient of thermal conductivity for the wall; d_1, d_2, d_{av}, the inside, outside, and mean diameters of the inner recuperator tube; d_3, the inside diameter of the outer recuperator tube; L, the length of the heat-exchange section; F, the area of the heat-exchange surface; T, the temperature of the heat carrier; ΔT_{av}, the average temperature head; Q, the heat output; n, the number of heat-exchange sections; Re = wd/ν, the Reynolds number; Pr = ν/a, the Prandtl number; G, the heat-carrier flow rate; ΔP, the pressure drop. Subscripts: H, the magnetic field; 0, the absence of a magnetic field; 1, the first heat carrier; 2, the second heat carrier; a prime indicates the inlet to the recuperator; double prime indicates the outlet from the recuperator; $\uparrow\uparrow$, direct motion of the heat carrier; $\uparrow\downarrow$, counterflow of the coolant.

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CLASSIFICATION OF THERMAL MODELS OF FLOWTHROUGH SYSTEMS OF THERMOSTATICALLY CONTROLLED OBJECTS AT VARIOUS TEMPERATURE LEVELS

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We describe an approach to the selection of simplified methods of designing flowthrough systems of thermostatically controlled objects at various temperature levels; this approach is based on the classification of thermal models of the system, in terms of the nature of the thermal links.

Frequent use is made in devices and industrial installations containing thermostatically controlled units of flowthrough thermostatic-control systems (STC) in which the coolant for heat carrier flows through heat exchangers in thermostatically controlled objects (OTC). Where necessary to maintain various temperature levels in OTC in close proximity to each other it is advisable to use sequential separation of the conduit with the coolant to ensure minimum coolant consumption.

In determining the requirements to be imposed on the parameters of such STC, we must take into consideration the mutual thermal effect of the OTC, thus limiting the applicability of the calculation method developed for thermostats designed to stabilize the temperature of a single object [1, 2].

It is the purpose of this paper to undertake the classification of the thermal models of systems to reflect the structural features of the indicated STC, and this classification is based on determination of the criteria of maximum and minimum coolant flow rate, as well as of the strong and weak thermal links between the elements, thus allowing us to make recommendations with regard to the simplification of the calculation methods. The research was carried out for the case of thermostatic control of three sequentially cooled objects; however, it is not difficult to extend both the results and conclusions to STC with an arbitrary number of OTC.

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