Also interesting are the results of experiments (Figs. 2 and 3) with electromagnetic high-frequency heating and delayed acoustic treatment (f = 16 kHz). According to the graphs here, the heating rate increased sharply immediately after the acoustic field had been turned on. The rate of increase depended on the distance from the acoustic radiator and from the electromagnetic high-frequency radiator.

These changes in the space–time distribution of the temperature in our model of a bituminous bed can be explained by an increase in the thermal diffusivity of sandstone due to acoustic treatment. As a consequence, the electromagnetic high-frequency heating depends more strongly on the thermal diffusivity and contributes more to the temperature distribution during high-frequency heating.

One must also consider a possible contribution to the intensification of high-frequency heating by a combination of lower viscosity of heated bitumen in an acoustic field and filtration flow due to an acoustic-pulse pressure drop.

NOTATION

f (kHz) is the frequency of the acoustic field;
ΔT (°C) is the temperature rise;
t is the time;
t_{ac} is the time of turning on the acoustic field;
R (cm) is the distance from the wall.

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PERFORMANCE OF A HEAT-EXCHANGE TURBULIZER

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UDC 536.24

Experimental evidence is surveyed for the performance of various types of turbulizers in convective heat transfer. An optimum range in Reynolds number and optimum ranges of geometrical parameters are identified.

There are two major ways of accelerating heat transfer in tubes; the first involves producing spiralling with strip and plate devices, which influence the entire flow. The second involves modifying the flow region near the wall by means of artificial roughnesses such as grooves on the inside wall, wire spirals, etc.

A large volume of experimental evidence has been accumulated on heat transfer in tubes with various devices covering wide ranges in heat load and physical parameters [1-15]. Some methods based on strip devices form the subject of interesting surveys [4, 16]. Table 1 gives the major results. No systematic survey has previously been published on the experimental data for the various types of systems in a form that could be used in comparative evaluation.

The ultimate purpose of any method of accelerating convective heat transfer is to provide a basis for designing equipment with the minimum transfer surface or minimum temperature difference subject to minimum power consumption in fluid pumping. Any of the methods of accelerating heat transfer increases the hydraulic resistance and thus increases the pumping power, so a major parameter must be the performance of the convecting surfaces. The following performance factor (the energy factor) is the one usually employed [2, 14, 17, 18]:

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TABLE 1. Measurements on Accelerated Heat Transfer in Tubes

Method of acceleration	Relative pipe length	Range Re	Medium	Resultant equation	Ref.
Wire spiral $0{,}35 \leqslant D/S \leqslant 1{,}76$	35,3D	$1{,}7\cdot10^3$—$20\cdot10^3$	Water	$Nu=0{,}3Re^{0{,}6}Pr^{0{,}43}(d/D)^{0{,}135}$	[5]
Strip device $0{,}13 \leqslant D/S \leqslant 0{,}31$	35,3D	$1{,}7\cdot10^3$—$20\cdot10^3$	Water	$Nu=1{,}84Re^{0{,}44}Pr^{0{,}36}(D/S)^{0{,}33}$	[5]
Strip device $0 \leqslant D/S \leqslant 0{,}25$	56,7D	10^4—$4\cdot10^4$	Water	$Nu=0{,}021Re^{0{,}8}Pr^{0{,}43}\left(\frac{Pr_w}{Pr_l}\right)^{0{,}25}K_T$ $K_T=1+1{,}13\cdot10^{-5}(D/SRe^{1{,}2})$	[6]
Wire spiral ($d=0{,}052$, 0,063 и 0,072)	68D	$3\cdot10^3$—$3\cdot10^5$	Water	$Nu=0{,}175Re^{0{,}7}Pr^{1/3}(d/D)^{-0{,}35}$	[7]
Strip device $2{,}5 \leqslant S/D \leqslant 11$	20D	160—5000	Air	$Nu=0{,}3Re^{0{,}6}Pr^{0{,}43}(d/D)^{0{,}135}$ (derived by V. K. Shchukin)	[8]
Strip device $3{,}16 \leqslant S/D \leqslant 9{,}5$	19D	120—5500	Air	$Nu=0{,}3Re^{0{,}6}Pr^{0{,}43}(d/D)^{0{,}135}$ V.K. Shchukin	[9]
Strip device $1{,}81 \leqslant S/D \leqslant 11$	20D	$6\cdot10^3$—$1\cdot10^5$	Air Water	$Nu=\dfrac{RePr}{\dfrac{D}{D_e}+\dfrac{1400}{\xi Re}\dfrac{D}{S}Pr^{0{,}731}}\times$ $\times\left\{\dfrac{50{,}9D/S}{Re\sqrt{\xi}}+0{,}023\dfrac{D}{D_э}Re^{-0{,}2}Pr^{-2/3}\left[1+\dfrac{0{,}0219}{(S/D)^2\xi}\right]^{1/2}\right\}$	[10]
Internal tube $S=27{,}6$—102) ($\varphi=45$—75°,	60D	10^3—$6\cdot10^4$	Water	$Nu=0{,}021Re^{0{,}8}Pr^{0{,}43}\left(\frac{Pr_w}{Pr_l}\right)^{0{,}25}\left(1+0{,}092\varphi^{*1{,}73}\right),\ \varphi^*=\frac{\varphi}{15°}$	[11]
Strip device $0 \leqslant S/D \leqslant 8{,}24$	—	$5\cdot10^3$—10^5	Air	$Nu=RePr\sqrt{\xi/2}\left\{\lambda\left[5Pr+d\ln\left(\dfrac{\dfrac{30}{d}-1+\dfrac{1}{Pr}}{\dfrac{5}{d}-1+\dfrac{1}{Pr}}\right)+d\ln(Re\sqrt{\xi/2}/60)\right]\right\}^{-1}$	[12]

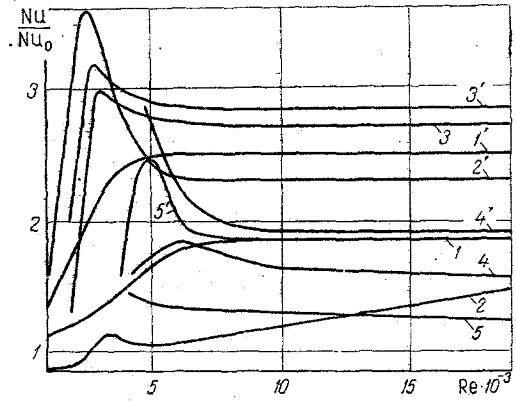

Fig. 1. Comparison of measurements on heat transfer in tubes: 1, 1') internal spiral tubes, φ of 45 and 75° [11]; 2, 2') transverse grooves, d/D of 0.983 and 0.875 [13]; 3, 3') spiral grooves, S/D of 3.25 and 1.0 [14]; 4, 4') wire spirals, S/D of 2.17 and 0.724 [15]; 5, 5') strip devices, S/D of 19.0 and 3.16 [9].

$$\alpha = f(N_0). \tag{1}$$

However, this factor must be determined for each surface type for identical mean temperatures [17], which means that it is virtually impossible to use in comparative evaluation of methods if the original data are employed.

We have employed a standard and reasonably general criterion in comparative evaluation of the various devices which serves to define the preferred types and the areas of application.

The performance may be evaluated from experiments performed by different workers with different average temperatures and different ranges in the Reynolds and Prandtl numbers if one uses the relation

$$(\mathrm{Nu}/\mathrm{Nu}_0) = f(\mathrm{Re}). \tag{2}$$

Then (2) characterizes the increase in the heat-transfer coefficient relative to the value for a smooth tube.

Figure 1 shows results from processing various bodies of data [9, 11, 13-15] with $\mathrm{Nu}/\mathrm{Nu}_0$ as a function of Re, where the values of the Nusselt number are referred to Reynolds numbers identical with those for smooth tubes. This indicates that methods acting at the periphery are the most promising for viscous media; these include transverse and spiral grooves on the inner surface and wire-spiral devices. Also, these plots define the best ranges in Reynolds number. Figure 1 shows that the largest improvement in heat transfer arises for Reynolds numbers between 2000 and 8000, i.e., in the laminar and weak-turbulence ranges.

If artificial periodic roughness is employed, eddies develop behind each projecting element; turbulence then develops gradually, and $\mathrm{Nu}/\mathrm{Nu}_0$ falls somewhat, but still remains substantially larger than 1. At Re > 8000, the main effect on the heat transfer comes from the turbulence, whereas the effect from the eddies gradually decreases.

Devices that influence the entire flow (strips, internal spiral tubes, etc.) are indicated as being more promising for media of high viscosity and at high Prandtl numbers.

Any of these techniques increase the hydrodynamic resistance and therefore the pumping power. Therefore, a proper comparison of the performance from all aspects is best based on the following relation:

$$(\mathrm{Nu}/\mathrm{Nu}_0)/(\xi/\xi_0) = f(\mathrm{Re}), \tag{3}$$

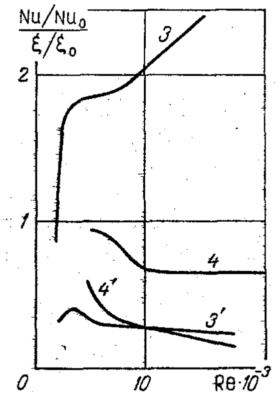

Fig. 2. Performance comparison for various device types (symbols as in Fig. 1).

which characterizes the relative increase in the heat transfer per unit additional energy consumption.

Evaluation via (3) differs from that by the method of [1] in that it enables one to compare various designs, not only for turbulent flow but also for laminar flow or weak turbulence. It also defines the best range in Reynolds number. The disadvantage of (3) include the need to process the data for identical major dimensions for the smooth and other cases.

Figure 2 shows data processed in accordance with (3); unfortunately, some workers give no data on the hydraulic resistance, so it is not possible to evaluate the performance of all the devices covered by Fig. 1. Figure 2 shows that artificial periodic roughnesses can provide high performance, and that the performance increases with the Reynolds number. The increase in performance relative to the increase in hydraulic loss may be as much as a factor of 2.5.

NOTATION

α	is the heat-transfer coefficient of surface;
N_0	is the energy consumed per unit time per m^2 by flowing medium;
Nu, ξ	are the Nusselt number and hydraulic resistance for tube with turbulizer;
Nu_0, ξ_0	are the values for a smooth tube;
Re	is the Reynolds number.

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GENERALIZED DIFFUSION THEORY OF MULTITEMPERATURE HOMOGENEOUS MIXTURES

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UDC 532.529.5

A generalized diffusion theory is constructed for motion of homogeneous mixtures consisting of components at different temperatures.

The theory of motion of mutually penetrable media, e.g., the motion of various materials and phases with consideration of phase transitions, chemical reactions, heat transfer, and radiation; the motion of liquids and gases with various particles; and the motion of liquids and gases in soils, is important in solution of numerous practical problems, in particular, those related to chemical production, hydroenergetics, hydromelioration, and the development of the petroleum and gas industry.

There exist various approaches to the problem of study of the complex motion and varied processes in interpenetrating media. One of these is diffusion theory.

We construct below a generalized diffusion theory of motion of multitemperature homogeneous mixtures. In contrast to classical diffusion theory [1-4] here we introduce a certain general characteristic velocity of the mixture as a whole, and diffusion flows are defined relative to this velocity. Using general integral laws of the mechanics of mixtures, a new system of equations is obtained for determination of the unknown values, generalizing Prigogine's theorem [2, 3]. This system permits transforming from one characteristic velocity to another in the general case of motion of different-temperature homogeneous mixtures. It is shown that with consideration of the contribution of diffusion flows to the energy of the total mixture the equations for determination of the diffusion flows are differential, and not algebraic, equations. The equations for change in component concentration are of the hyperbolic type, not parabolic, as in classical theory, i.e., generalized diffusion laws are obtained, as were proposed in [5, 6] in analogy to the rheology of viscoelastic media.

1. Basic Integral Laws of the Mechanics of Mixtures. We will consider the motion of mixtures consisting of n components. We will assume that all these components fill one and the same volume, occupied by the mixture. Let ρ_k, u_k be the density and velocity of the k-th component. It is known that in the theory of homogeneous mixtures, together with the density ρ_k it is necessary to consider other quantities characterizing the inertness of each component. These quantities are M_k, the molar mass; V_k, the molar partial volume; and N_k, the number of moles of the k-th component per unit volume of mixture.

Let u_a be some characteristic velocity of the mixture. We will assume that u_a may be expressed as a linear combination of u_k with the aid of some system of normalizing weights a_k, i.e.,

$$u_a = \sum_{k=1}^{n} a_k u_k; \quad \sum_{k=1}^{n} a_k = 1, \tag{1.1}$$

where a_k may depend only on ρ_k, M_k, N_k and V_k.

The motion of the k-th component relative to an observer moving with a velocity u_a is determined with the aid of a generalized diffusion current J_k^a, equal to

$$J_k^a = \rho_k (u_k - u_a); \quad \sum_{k=1}^{n} \frac{a_k}{\rho_k} J_k^a = 0. \tag{1.2}$$

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