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HYDRODYNAMIC THEORY OF HEAT TRANSFER BETWEEN A STABILIZED GAS SUSPENSION FLOW AND CHANNEL WALLS

Z. R. GORBIS and F. E. SPOKOINII

Odessa Technological Institute for Refrigerator Industry

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Abstract—The theoretical study is made of heat transfer between dilute gas–solid flows ($\beta < 3$ per cent) at a stabilized section and channel walls at $q = \text{const}$. For the conditions under consideration the energy equation is reduced to the form of the modified Lyon integral. The solution is obtained for developed turbulent transfer with regard for the effect of solid particles upon hydrodynamical characteristics of a carrying agent. The relations are established which according to pressure loss data allow calculation of the rate of heat transfer with a flow as well as anlysis of the effect of the main factors upon the integral result. The relations are obtained for negligibly small and essential temperature slip of components. In the latter case the additional thermal resistance due to the finite rate of intercomponent heat transfer is taken into consideration.

The calculation results and the available experimental data are compared. Their satisfactory agreement is found, and the effect of individual thermal resistances upon total heat transfer is shown.

NOMENCLATURE

a,	thermal diffusivity;
c,	heat capacity;
$D = 2r_0$,	channel diameter;
d,	particle diameter;
q,	specific heat flux;
R,	dimensionless radius;
G,	mass flow rate;
r, x,	cylindrical coordinates;
t,	temperature;
v,	velocity;
Z,	ratio of water numbers of components;
Pr,	Prandtl number;
Nu,	Nusselt number;
Re,	Reynolds number;
α,	heat-transfer coefficient;
β,	true volumetric concentration;
δ,	viscous sub-layer thickness;
χ,	Kármán constant;
ξ,	drag coefficient;
ζ,	longitudinal velocity loss coefficient with impact;
μ,	mass flow rate concentration;
ν,	kinematic viscosity;
η,	dynamic viscosity;
Ω,	channel cross-section;
ρ,	density;
φ_t,	temperature slip coefficient of components.

Subscripts

f,	disperse flow;
s,	solid particles;
$*$,	turbulent analog;
w,	wall value;
im,	impact;
○,	flow without particles;
$+$,	transverse direction;
0,	dynamic velocity;
c,	flow core;
b,	viscous sublayer.

INTRODUCTION

THE GAS flows with suspended solid particles are a perspective mode of heat agent since these may ensure intensive heat removal with high proper heat capacity. For this reason a number of studies on heat transfer with such flows have been recently carried out. The review and comparative analysis of such investigations are given in [1]. For concentrations within a range ($2 < \mu < \mu_{cr}$) the essential increase in the heat transfer coefficient proportional to μ^n where $n \leqslant 1$ is observed. In this case the increase in Nu_f appeared to be essentially different under various conditions. For small concentrations qualitative changes are revealed, i.e. heat transfer with a flow at $\mu \simeq 1$–3 may be less intensive than that with a pure gas.

So much essential difference in the nature of the effect of particle concentration upon the rate of heat transfer with a flow cannot be explained by the known solutions. This is due to the use of two extremely essential assumptions in the analysis.

1. Temperature and velocity slip of components is considered to be negligibly small [1, 4, 5]. As will be shown below, in some cases (coarse particles) this may lead to essential overestimation of the heat transfer rate.

2. It is considered that particles appear not to influence a velocity structure of a carrying agent (pulsation velocities and viscous sublayer thickness), [2–5]. If it is taken into account that according to [1] a decrease in the thickness of a viscous sublayer due to particles is the main mechanism to intensify heat transfer with flow, then it is clear that the use of the second assumption leads to an essential underestimation of Nu_f.

The present paper gives an attempt to solve a thermal energy equation for a disperse flow without introducing so much essential assumption.

STATEMENT OF PROBLEM

Heat transfer with a gas suspension flow at a thermal stabilization section at $q_w = \text{const}$ is considered. According to [1]

$$\beta\rho_s c_s v_s \frac{\partial t_s}{\partial x} + (1-\beta)\rho c v \frac{\partial t}{\partial x} = \lambda_s \beta_F \left(\frac{\partial^2 t_s}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r}\right) + \lambda(1-\beta_F)\left(\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r}\right). \qquad (1)$$

Neglecting thermal conduction in the axial direction, conductive transfer at the points of contact between particles and a wall ($\beta_F = 0$) and using turbulent conduction analogs

$$\lambda_s^* = \beta\rho_s c_s a_s^*; \qquad \lambda^* = (1-\beta)\rho c a^* \qquad (2)$$

it is possible to arrive at

$$(1-\beta)c\rho v\left(\frac{\partial t}{\partial x} + r\frac{\partial t_s}{\partial x}\right) = \frac{1}{r}\frac{\partial}{\partial r}\left[(\lambda + \lambda^* + \lambda_s^*) r \frac{\partial t}{\partial r}\right]. \qquad (3)$$

In a developed turbulent flow temperature and velocity non-uniformities are localized in a thin layer adjacent to a wall. Therefore, when considering balance expressions it is possible, with a high accuracy, to assume

$$\mu = \bar{\mu}; \qquad v = \bar{v}; \qquad t = \bar{t}. \qquad (4)$$

The bar in expression (4) means averaging of a quantity with respect to a cross-section. Owing to the mixing effect of particles for gas disperse flow, expression (4) is still more applicable than for continuous ones. Then, from the heat balance equation for an element dx it follows

$$\frac{\partial t}{\partial x} + Z\frac{\partial t_s}{\partial x} = \frac{4q_w}{Dv\rho c(1-\beta)}. \qquad (5)$$

From expression (3) with regard for expression (5) and $R = r/r_0$ it is possible to obtain

$$2q_w R r_0 = \frac{\mathrm{d}}{\mathrm{d}R}\left[(\lambda + \lambda^* + \lambda_s^*) R \frac{\mathrm{d}t}{\mathrm{d}R}\right]. \qquad (6)$$

Hence

$$\mathrm{d}t = Dq_w \frac{\int_0^R R\,\mathrm{d}R}{(\lambda + \lambda^* + \lambda_s^*)R}\,\mathrm{d}R. \qquad (7)$$

The mean calorimetric flow temperature in a cross-section X

$$\bar{t}_f = \frac{c\rho(1-\beta)\int\limits_{\Omega}(1+Z\varphi_t)vt\,\mathrm{d}\Omega}{c\rho(1-\beta)\int\limits_{\Omega}(1+Z)v\,\mathrm{d}\Omega}$$

$$= \frac{1+Z\varphi_t}{1+Z}\,2\int\limits_0^1 tR\mathrm{d}R$$

$$= \frac{1+Z\varphi_t}{1+Z}\left[t_w - 2\int\limits_0^1\left(\int\limits_0^R R\,\mathrm{d}R\right)\mathrm{d}t\right]$$

$$= \frac{1+Z\varphi_t}{1+Z}\left[t_w - Dq_w\left(\frac{1}{2}\int\limits_0^1\frac{R^3\,\mathrm{d}R}{\lambda+\lambda^*+\lambda_s^*}\right)\right]. \quad (8)$$

Finding a temperature drop $(t_w - \bar{t}_f)$ from equation (8), we have

$$Nu_f^{-1} = \frac{\lambda(t_w - \bar{t}_f)}{q_w D} = \frac{1}{2}\int\limits_0^1\frac{\lambda R^3\,\mathrm{d}R}{\lambda+\lambda^*+\lambda_s^*} + \frac{\lambda\bar{t}_f}{q_w D}\,\frac{Z(1-\varphi_t)}{1+Z\varphi_t}. \quad (9)$$

Formula (9) may be considered as some extension of the integral Lyon expression to the case of disperse flow with essential temperature slip of components. In this case the account for temperature slip of components leads to the fact that additional thermal resistance (second addend in formula (9)) appears. Hence, the inverse dependence of the heat transfer rate (Nu_f) upon temperature flow non-equilibrium conditions $(1 - \varphi_t)$ follows uniquely.

The physical reason of the onset of additional thermal resistance at $\varphi_t \neq 1$ is the finite rate of an intercomponent heat transfer process being a part of total heat transfer between the whole flow and wall.

FLOW WITHOUT TEMPERATURE SLIP OF COMPONENTS

For practically equilibrium gas suspension flow $(\varphi \to 1)$ when a two-layer flow model is used, then expression (9) reduces to the form:

$$(Nu_f^{-1})_{\varphi_t=1} = \frac{1}{2}\int\limits_0^{R_1}\frac{\lambda R^3\,\mathrm{d}R}{\lambda^*+\lambda_s^*} + \frac{1}{2}\int\limits_{R_1}^1\mathrm{d}R. \quad (10)$$

Taking into account a unique (turbulent) transfer mechanism of heat and impulse by components in a flow core, take as a first approximation

$$\frac{a_s^*}{a^*} = \frac{v_s^*}{v^*}; \qquad Pr_s^* = Pr^*. \quad (11)$$

This assumption is at least more substantiated than those in [2, 3] $a_s^*/a^* = 0$ and $a_s^*/a^* = 1$ in [1,4].

Equality (11) does not establish a similar a priori relationship between molar and a molecular transfer coefficients, but corresponds to the statement on the similarity of different turbulent transfer processes for each component individually. In this case, with regard for expression (2)

$$\frac{\lambda^*+\lambda_s^*}{\lambda} = \frac{a^*}{a}\left[1 + \frac{c_s\rho_s\beta}{c\rho(1-\beta)}\cdot\frac{a_s^*}{a^*}\right]$$

$$= \frac{Pr}{Pr^*}\cdot\frac{v^*}{v}\left[1 + \frac{c_s\eta_s^*}{c\eta^*}\right]. \quad (12)$$

For turbulent flows it may be considered that $Pr^* = 1$. The ratio of turbulent analogs of the dynamic viscosity coefficient is determined in terms of the effective von Kármán constant according to the approximation of the data given in [6].

$$\frac{\eta_f^*}{\eta^*} = \frac{\eta^*+\eta_s^*}{\eta^*} = (\chi/\chi_0)^2 = (1 + 0{\cdot}16\,\mu^{0\cdot 9})^2. \quad (13)$$

In this case

$$\frac{\lambda^*+\lambda_s^*}{\lambda} = Pr\frac{v^*}{v}\left\{1 + \frac{c_s}{c}\left[(\chi/\chi_0)^2 - 1\right]\right\}$$

$$= Pr\frac{v^*}{v}\varkappa. \quad (14)$$

Hence

$$(Nu_f^{-1})_{\varphi_t=1} = \frac{1}{2}\int_0^{R_1} \frac{R^3 \mathrm{d}R}{Pr\frac{v^*}{v}\varkappa} + \frac{1}{2}\int_{R_1}^{1} \mathrm{d}R. \quad (15)$$

Thus, formula (15) differs from the known one for uniform flows [7] only by a multiplier $\varkappa^{-1}$ in the first integral which does not depend upon a radial coordinate. Then, from formula (7) it follows

$$\begin{aligned}(Nu_f^{-1})_{\varphi_t=1} &= \frac{1}{2}\frac{1}{\varkappa Pr Re\,\chi\sqrt{(\xi/32)}} \\ &\quad\times\left[\ln\frac{Rev}{v^0\delta}\sqrt{\left(\frac{\xi}{32}\right)}-\frac{3}{2}\right] \\ &\quad+\frac{1}{2}\frac{v^0\delta}{vRe}\sqrt{\left(\frac{32}{\xi}\right)} = \frac{1}{2}\left[\frac{1}{\varkappa\chi Pr Re\sqrt{(\xi/32)}}\right. \\ &\quad\left.\times\left(\ln\frac{r_0}{\delta}-\frac{3}{2}\right)+\frac{\delta}{r_0}\right]. \quad (16)\end{aligned}$$

For making calculations by means of relation (16) it is necessary to know the friction coefficient ξ and viscous sublayer thickness δ of a carrying agent involving particles in it.

The quantity ξ characterizes friction stresses only of a liquid which are defined in terms of total resistance of the whole disperse flow with no regard for the effect of direct interaction between particles and a wall. The hypothesis on equal probability for collision of all particles with a wall is made, whose application is mostly substantiated for vertical flows. In this case $\xi \neq \xi_0$ and is determined by the expression

$$\begin{aligned}\Delta P_{\mathrm{im}} &= n_{\mathrm{im}}\pi DL\,.\,m_s v_{sw}\,.\,\zeta\frac{4}{\pi D^2} \\ &= \frac{\pi\zeta}{2}\cdot\frac{L}{D}\rho_s v_{sw} v'_{s+}\beta \\ \xi &= \xi_f - \xi_{\mathrm{im}} = \xi_f - \frac{\Delta P_{\mathrm{im}}}{\frac{1}{2}\frac{L}{D}\rho v^2} \\ &= \xi_f - \pi\zeta\varphi^1_+\frac{v'_+}{v'_{0+}}\sqrt{\left(\frac{\xi_0}{8}\right)}\frac{v_{sw}}{v_s}\mu. \quad (17)\end{aligned}$$

The quantity $\varphi'_+ = v'_+/v'_+$ takes into account component slip by a fluctuation velocity and is found by the relations obtained in [8], v'_+/v'_{0+} characterizes the influence of particles upon carrying agent turbulence [9]. The axial particle velocity at an impact moment v_{sw} is determined by a liquid one at a distance of a particle diameter from a wall minus a relative particle velocity.

Not introducing the additional assumptions, calculate a thickness of a viscous carrying-agent layer adjacent to a wall in a suspension flow. The integral identity

$$\frac{\bar{v}}{v^0} = \sqrt{\left(\frac{8}{\xi}\right)} = \frac{2}{r_0^2}\int_0^{r_0}\frac{v}{v^0}r\,\mathrm{d}r \quad (18)$$

may be revealed using the two-layer flow model earlier accepted. Neglecting small-order terms in the expression obtained by integration allows determination of δ by solving a simple transcendental equation

$$\ln\frac{r_0}{\delta} \simeq \chi\sqrt{\left(\frac{8}{\xi}\right)}+\frac{3}{2}-\chi Re\sqrt{\left(\frac{\xi}{32}\right)}\cdot\frac{\delta}{r_0}. \quad (19)$$

With regard for formula (19) expression (16) may be written as follows

$$(Nu_f^{-1})_{\varphi_t=1} = \frac{1}{2}\left[\frac{1}{Pr\varkappa}\left(\frac{16}{Re\xi}-\frac{\delta}{r_0}\right)+\frac{\delta}{r_0}\right]. \quad (20)$$

Here the addends correspond to thermal resistances of a core and viscous sublayer in a gas suspension flow

$$\begin{aligned}\left(\frac{Nu_f}{Nu_0}\right)_{\varphi_t=1} &= \frac{R_{0c}+R_{0b}}{R_{fc}+R_{fb}} \\ &= \frac{\frac{1}{Pr}\left(\frac{16}{Re\xi_0}-\frac{\delta_0}{r_0}\right)+\frac{\delta_0}{r_0}}{\frac{1}{Pr\varkappa}\left(\frac{16}{Re\xi}-\frac{\delta}{r_0}\right)+\frac{\delta}{r_0}}. \quad (21)\end{aligned}$$

TEMPERATURE NON-EQUILIBRIUM FLOW

In order to determine the rate of heat transfer with gas suspension flows at $\varphi_t \neq 1$, return to the analysis of relation (9). Use the formula from [1] on the basis of [4] for temperature slip of components

$$\varphi_t = \frac{t_s}{t} = 1$$

$$\frac{q_w \pi D(1 + Z)[1 - \exp(-px)]}{Gc(1 + Z)pt_f + q_w \pi DZ[1 - \exp(-px)]} \tag{22}$$

Here $p = 6\alpha_s(1 + Z/(c_s \rho_s v_s d_s)$; α_s is the intercomponent heat-transfer coefficient. With regard for formula (22) the correction for $\varphi_t \neq 1$ is of the form:

$$\frac{\lambda t_f}{q_w D} \cdot \frac{Z(1 - \varphi_t)}{1 + r\varphi_t} = \frac{\lambda \pi Z[1 - \exp(-px)]}{Gc(1 + z)p}$$

$$= \frac{2}{3} \frac{Z}{(1 + Z)^2} \times \left(\frac{d_s}{D}\right)^2 \frac{c_s \rho_s v_s}{c \rho v} \frac{1}{Nu_s} \left\{1 - \exp\left[-\frac{6\alpha_s(1 + Z)x}{c_s \rho_s v_s d_s}\right]\right\}. \tag{23}$$

According to formulae (9) and (23) the relative rate of heat transfer with gas suspension flows at a stabilization section may be expressed by the following formula

$$\frac{Nu_f}{Nu_0} = \frac{\frac{1}{2}\left[\frac{1}{Pr}\left(\frac{16}{Re\xi_0} - \frac{\delta_0}{r_0}\right) + \frac{\delta_0}{r_0}\right]}{\frac{1}{2}\left[\frac{1}{Pr\varkappa}\left(\frac{16}{Re\xi} - \frac{\delta}{r_0}\right) + \frac{\delta}{r_0}\right] + \frac{2}{3}\frac{Z}{(1 + Z)^2} \cdot \frac{d_s^2 c_s \rho_s v_s}{D^2 c \rho v Nu_s}}. \tag{24}$$

Formula (24) differs from formula (21), i.e. in the denominator there appears the last addend expressing the effect of the rate of intercomponent heat transfer (correction for $\varphi_t \neq 1$). Relation (23) passes over into the known expression obtained in [3] only if $x \to \infty$. In this case the factor in the braces of formula (23) tends to unity. From this it follows that at the initial section, thermal resistance due to the non-equilibrium conditions of a flow grows along a length, achieving a maximum at the end of this section. In this case the heat transfer rate decreases accordingly. The dependence of the correction for $\varphi_t \neq 1$ upon the water number ratio for components Z in equation (23) has a maximum at $Z = 1$ and shows a decrease in the heat-transfer rate over a range $0 < Z < 2$, that was mentioned in the literature. Formula (24) written in terms of thermal flow core resistances R_c, viscous sublayer R_b and intercomponent heat-transfer resistance R_t assumes the form

$$\frac{Nu_f}{Nu_0} = \frac{R_0}{R_f} = \frac{R_{0c} + R_{0b}}{(R_{fc} + R_{fb})_{\varphi_t = 1} + R_{ft}}. \tag{25}$$

For gases formula (24) is essentially simplified ($Pr \simeq 1$)

$$\frac{Nu_f}{Nu_0} = \left[\frac{1}{\varkappa}\left(\frac{\xi_0}{\xi} - \frac{Re\xi_0}{16}\frac{\delta}{r_0}\right) + \frac{Re\xi_0}{16}\frac{\delta}{r_0} + \frac{1}{12}\frac{Z}{(1 + Z)^2} \cdot \frac{d_s^2 c_s \rho_s v_s \xi_0 Re}{D^2 c \rho v Nu_s}\right]^{-1}. \tag{26}$$

COMPARISON WITH EXPERIMENTAL DATA

In order to compare the relation obtained with direct test data the experimental works on pressure losses and heat transfer are used. As the preliminary analysis has shown, the condition $\varphi_t = 1$ is satisfied in experiments with very fine graphite particles ($d_s \sim 1\ \mu$), [10, 11] over a wide range of the number Re and mass flow rate concentration since R_{fb} exceeds R_{ft} by more than three orders.

Experimental data of [10] are obtained over a range of mass flow rate concentrations at $\mu < 12$ and those of [11], at $\mu > 10$. Some disagreement in the experimental data of these investigators over a range of $10 < \mu < 15$ may be apparently explained by a certain deviation of a particle diameter and by the difference in the experimental methods used. The calculations made by expressions (13), (14), (17) and

(19) have shown that the contribution of particles to turbulent thermal conductivity of a flow and the viscous sublayer thickness essentially differ from their limiting values a priori introduced into known solutions [2–5].

$$\frac{\xi_f}{\xi_0} > \frac{\delta_0}{\delta} > 1; \qquad 1 + Z > \varkappa > 1.$$

Figure 1 gives the nature of the dependence of relative thermal resistances upon concentration. From Fig. 1 it is seen that basic thermal resistance is within a viscous layer adjacent to a wall, and heat transfer intensification is mainly achieved due to a decrease in δ when introducing particles into the flow.

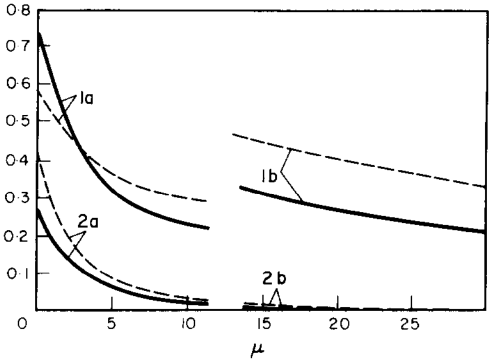

Fig. 1. Plot of thermal resistance vs. concentration $\varphi_t = 1$ ———, $Re = 10^4$; - - - - -, $Re = 2{\cdot}5.10^4$; curves a, calculation for experimental conditions [10]; curves b, calculation for experimental conditions [11]; 1, R_{fb}/R_0; 2, R_{fc}/R_0.

Figure 2 gives the comparison of the calculation results obtained by the theoretical expressions in [1–4] and by relation (21) with the experimental data of [10, 11]. This comparison shows that the regard for a real turbulent flow structure at $\varphi_t = 1$ (relation (21)) allows a good agreement between the experimental and calculation results to be achieved.

To verify relations (23)–(26) obtained assuming $\varphi_t \neq 1$ the calculation results are compared with the experimental ones [4, 5] on gas graphite flows ($d_s = 70$, 100, 200, 250 μ) in circular tubes ($D = 5{\cdot}33$, $8{\cdot}16$, $18{\cdot}8$ mm). The calculation results show that with an increase in a particle size the value of thermal resistance due to the non-equilibrium conditions of a flow acquires rather essential importance: for particles ($d_s = 70\,\mu$) the quantity R_{ft} exceeds flow core resistance (Fig. 3) and for particles ($d_s = 200\,\mu$ and more) it becomes commensurable even with thermal resistance of a boundary layer (Fig. 4). Consequently, in the experiments [4, 5] there

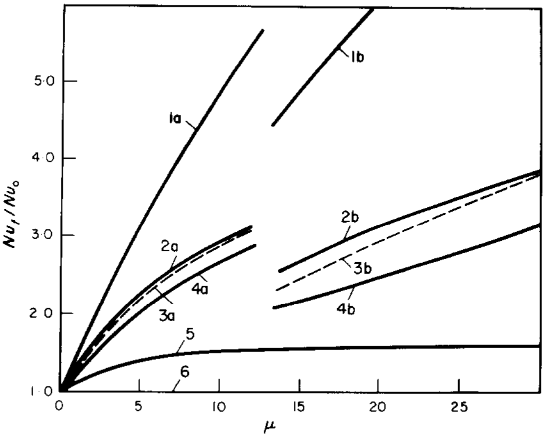

Fig. 2. Dependence of rate of heat transfer with gas suspension flow at $\varphi_t = 1$ ($Re = 2{\cdot}5.10^4$); curves a, b, calculation for experimental conditions [10, 11]; 1, calculation according to [1]; 2, calculation according to [1] with regard for $\xi \neq \xi_f$; 3, experimental data [10, 11]; 4, calculation according to (21); 5, calculation according to [4]; 6, calculation according to [2, 3].

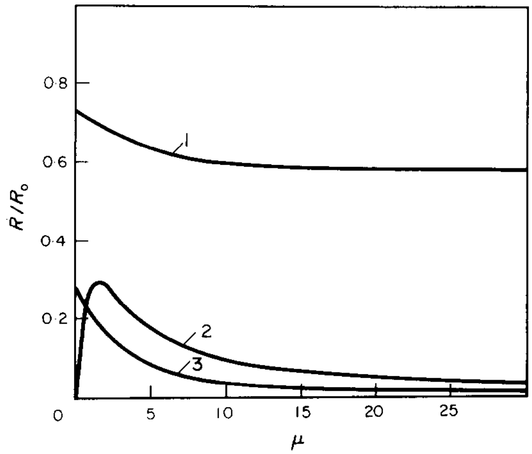

Fig. 3. Plot of thermal resistances vs. concentration ($d_s = 70\,\mu$, $Re = 10^4$). 1, R_{fb}/R_0; 2, R_{ft}/R_0; 3, R_{fc}/R_0.

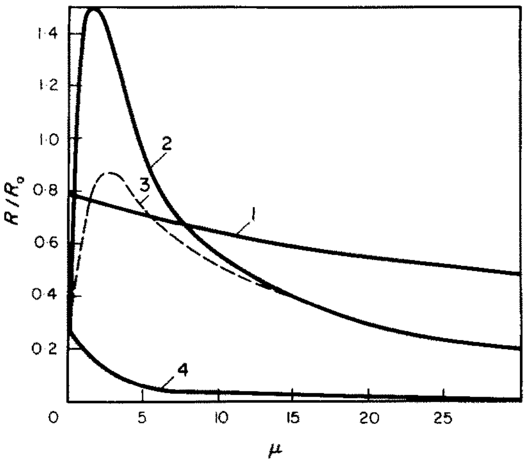

FIG. 4. Plot of thermal resistances vs. concentration ($d_s = 200\,\mu$ $Re = 10^4$); 1, R_{fb}/R_0; 2, R_{ft}/R_0; 3, R_{ft}/R_0 with regard for expression (23); 4, R_{fc}/R_0.

occurred essential temperature slip of components. Note that in accordance with expression (24) the dependence of R_{ft} upon mass flow rate concentration is not monotonic (see Figs. 3 and 4). Taking into account in expression (24) not $x = \infty$ but a finite length of an experimental section [4] a maximum R_{ft} removes into the domain $\mu > 1$ and its value essentially decreases.

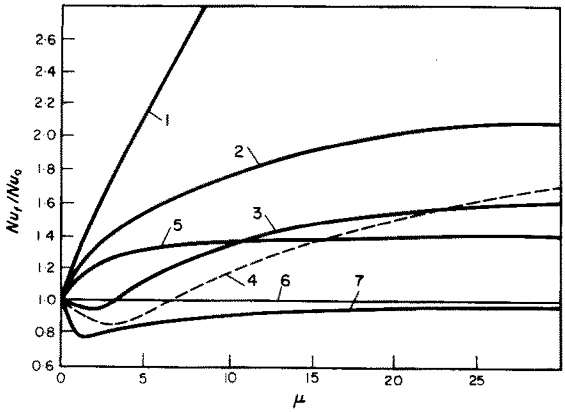

FIG. 5. Dependence of rate of heat transfer with a gas suspension flow at $\varphi_t \neq 1$ ($d_s = 70\ \mu$ $Re = 10^4$); 1, calculation according to [1]; 2, calculation according to [1] with regard for $\varphi_t \neq 1$; 3, calculation according to expression (26); 4, experimental data [4]; 5, calculation according to [4]; 6, calculation according to [2]; 7, calculation according to [3].

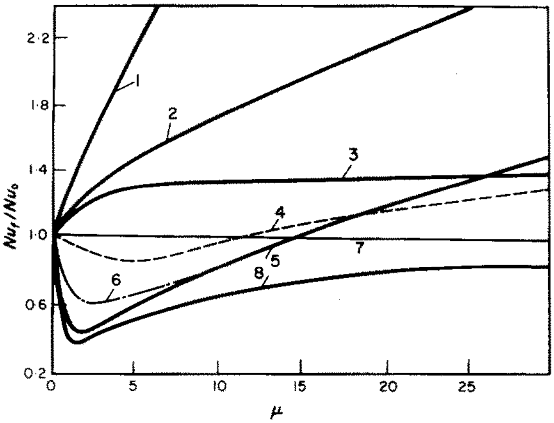

FIG. 6. Dependence of rate of heat transfer with a gas suspension flow at $\varphi_t \neq 1$ ($d_s = 200\ \mu$ $Re = 10^4$); 1, calculation according to [1]; 2, calculation according to [1] with regard for $\xi \neq \xi_f$; 3, calculation according to [4]; 4, experimental data of [4]; 5, calculation according to expression (26); 6, calculation according to expression (26 with regard for expression (23); 7, calculation according to [2]; 8, calculation according to [3].

In Figs. 5 and 6 is presented the comparison of the experimental relations for Nu_f/Nu_0 according to [4, 5] with the theoretical expressions obtained in [1–4] and in the present paper. According to Fig. 5 ($d_s = 70\,\mu$, $Re = 10^4$ only relation (26) deviating from the experimental data with a mean error of order of 15 per cent is similar to the experimental curve. It also allows us to explain some decrease in the heat-transfer rate at $\mu \simeq 1$ and a relatively weak dependence of Nu_f/Nu_0 upon concentration. The same conclusions may be made by analysing Fig. 6 ($d_s = 200\,\mu$, $Re = 10^4$). This confirms the assumption that the effect of temperature slip of components is a basis of such an abnormal dependence of the heat transfer rate upon μ. A noticeable discrepancy of the experimental data and those calculated by expression (26) at small μ (Fig. 6) may be evidently explained by the fact that in [4, 5] the conditions of a stabilized thermal process were not ensured. Then, thermal resistance of intercomponent heat transfer, being of importance namely at small μ, calculated by equation (26) is essentially overestimated. Correct account of this resistance according to expression (23) gives a considerable decrease in the discrepancy of the experimental and calculation data (Fig. 6).

CONCLUSION

Expressions (21)–(26) obtained in the present paper allow, according to the known hydromechanical characteristics of a gas suspension flow, estimation of the heat transfer rate, elucidation and analysis of the effect of individual thermal resistances upon the integral result. It is obvious that formula (21) is a particular case of expression (24) since the latter is transformed into (21) under zero temperature non-equilibrium conditions of flow components, that corresponds to $\varphi_t = 1$ and is valid for fine dispersed gas suspensions.

It should be noted that the theoretical analysis of the non-stabilized heat transfer conditions at short sections is of interest in a number of cases and is therefore one of the trends of further investigations in the filed under consideration.

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THEORIE HYDRODYNAMIQUE DU TRANSFERT THERMIQUE ENTRE UN ECOULEMENT D'UNE SUSPENSION GAZEUSE STABILISEE ET LES PAROIS D'UN CANAL

Résumé—On a fait l'étude théorique du transfert thermique à une section fixe entre des écoulements dilués gaz-solide ($\beta < 3\%$) et les parois d'un canal chauffé à q = constante. Sous les conditions considérées, l'équation d'énergie est réduite à la forme de l'intégrale de Lyon modifiée. La solution est obtenue pour un transfert turbulent établi, en considérant les effets de particules solides sur les caractéristiques hydrodynamiques d'un fluide de transport. On a établi les relations qui, selon les résultats de perte de pression, permettent le calcul de flux thermique aussi bien qu'une analyse de l'effet des facteurs principaux sur le résultat de l'intégrale. Les relations sont obtenues pour un saut de température petit ou essentiel en des composantes. Dans le dernier cas la résistance thermique additionnelle due au flux fini de chaleur entre les composants est prise en considération.

Les résultats du calcul sont comparés aux résultats expérimentaux utilisables. On a trouvé leur accord satisfaisant et on a montré l'effet des résistances thermiques individuelles sur le transfert thermique total.

EINE HYDRODYNAMISCHE THEORIE DER WÄRMEÜBERTRAGUNG ZWISCHEN EINER STABILEN GASSUSPENSIONSSTRÖMUNG UND DEN KANALWÄNDEN

Zusammenfassung—Eine theoretische Untersuchung der Wärmeübertragung zwischen verdünnten Gas-Fest-Strömen ($\beta < 3\%$) und den Kanalwänden wurde in einem konstant gehaltenen Abschnitt für q = const. durchgeführt. Unter Berücksichtigung der Voraussetzungen wurde die Energiegleichung auf die Form des modifizierten Lyon-Integrals reduziert. Unter Beachtung des Effektes von den festen Partikeln auf die hydrodynamischen Eigenschaften des Transportmittels wurde eine Lösung für die ausgebildete turbulente Übertragung gewonnen. Die eingeführten Beziehungen ermöglichen sowohl die Berechnung des Grades der Wärmeübertragung mit einer Strömung gemäss den Dreieckverlustdaten, wie die Analyse des Effektes der Hauptfaktoren auf das integrale Ergebnis. Die Beziehungen wurden gewonnen für einen sehr kleinen und auch für einen grösseren Temperaturschlupf der Komponenten. Für den letzteren Fall wurde der zusätzliche thermische Widerstand, aufgrund der endlichen Wärmeübertragung zwischen den Komponenten, in die Betrachtung einbezogen. Die berechneten Ergebnisse wurden mit den verfügbaren Versuchswerten verglichen. Dabei wurde eine zufriedenstellende Übereinstimmung gefunden und der Effekt des einzelnen thermischen Widerstandes auf die Gesamtwärmeübertragung gezeigt.

ГИДРОДИНАМИЧЕСКАЯ ТЕОРИЯ ТЕПЛООБМЕНА СТАБИЛИЗИРОВАННОГО ПОТОКА ГАЗОВЗВЕСИ

Аннотация—Проведено теоретическое исследование процесса теплообмена на стабилизированном участке разреженных газодисперсных потоков ($\beta < 3\%$) со стенками канала при q = const. Для рассматриваемых условий уравнение энергии приведено к виду модифцированного интеграла Лайона. Решение получено для режима развитого турбулентного транспорта с учетом влияния твердых частиц на гидромеханические характеристики несущей среды. Установлены зависимости, которые позволяют по сведениям о потерях давления рассчитать интенсивность теплообмена с потоком, а текже анализировать влияние основных факторов на интегральный результат. Зависимости получены для случаев пренебрежимо малого и существенного температурного скольжения компонентов. В последнем случае учитывается дополнительное термическое сопротивление, обязанное конечной интенсивности межкомпонентного теплообмена.

Проведено сопоставление расчетных результатов с экспериментальными данными, известными в литературе. Обнаружено их удовлетворительное согласие, а также показано влияние отдельных термических сопротивлений на общий теплоперенос.