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Fluid–solids flow with thermal and hydrodynamic non-equilibrium

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Abstract—A model is developed to describe the behaviour of particles in air streams. The equations of particle flow and heat transfer are given in dimensionless form. Two practical applications for suspension flows are solved: (a) the flow past a temperature step and (b) the injection of hot particles in a pipe carrying a gas. In both cases instantaneous velocity and temperature differences for the gas and solids are calculated and the effect of several dimensionless groups (such as Reynolds numbers, loading and dimension ratios) on these two quantities is determined. It was found that non-equilibrium effects are accentuated when bigger particles are in the mixture.

INTRODUCTION

THE SUBJECT of heat transfer in suspension flows was examined experimentally in the 1960s, when suspensions were considered for heat transfer augmentation. During that time the experimental work by Farbar and co-workers [1, 2], Tien and Quan [3], Schludersberg *et al.* [4], Wachtell *et al.* [5] and Danziger [6], among others, provided the experimental data and design correlations for the convective coefficient (film coefficient) of a suspension. A review of the experimental data and of the correlations that have resulted is given by Pfeffer *et al.* [7]. An analytical study by Tien [8] for homogeneous suspensions with low solids content and more recently studies by Ozbelge and Somer [9] and Michaelides [10] have added to the knowledge of homogeneous suspensions.

When non-equilibrium flows are concerned, a study by Whalley *et al.* [11] treats the subject of droplet flows in a stream. Very little is known about the flow of suspensions with hydrodynamic and thermodynamic non-equilibrium. Experimental data are non-existent because of the difficulty of measuring the temperature of solids and we could not find any analytical approach to the subject other than the treatment of homogeneous suspensions, which precludes non-equilibrium. This, despite the fact that heat transfer in suspensions is very important in pneumatic conveying, drying of solids and chemical flows with catalysts.

This work aims at the development of a model, which describes the behaviour of solid particles in gaseous streams when the velocities and temperatures of the two phases are considerably different. The equations developed are solved for the following cases: (a) pipe flow of particles with a gas, through a temperature step on the wall temperature and (b) the injection of hot particles in a colder pipeline carrying a gas.

The emphasis of this study is in the mechanical and thermal non-equilibrium that exists between the two phases. The temperatures and velocities of the two phases are obtained and parameters that affect the lack of equilibrium are pointed out. A limited number of cases where heat transfer from the wall to the suspension is calculated are also presented in this study.

THE SET OF EQUATIONS FOR A SUSPENSION

A one-dimensional model is developed for the flow of the air–particles mixture in a cylindrical domain of diameter D. The diameter of the particles is assumed to be low enough ($d/D < 0.05$) for the particles to be considered much smaller than the pipe diameter. Also the Reynolds numbers of the flow are high enough for the particles to be in a state of nearly homogeneous suspension within the gas. The concentration of the suspension is always low (< 0.06), which ensures that interparticle effects are negligible. The temperature differences considered are low enough for radiation not to play an important role in the heat transfer between the particles, the gas and the pipe walls.

Under these conditions the conservation equations may be written as follows.

Continuity equations for gas and particles

$$\frac{\mathrm{d}}{\mathrm{d}x}(\rho_g U_g) = \frac{4}{D} E_s \tag{1a}$$

and

$$\frac{\mathrm{d}}{\mathrm{d}x}(N m_p U_p) = -\frac{4}{D} E_s \tag{1b}$$

where E_s is the sublimation rate of the particles expressed per unit area of the pipe wall. For particles which do not split or agglomerate N is constant. Then

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the last equation together with the momentum equation for a particle will yield the rate of change of the particle radius (dr/dx) when sublimation or condensation occurs. It is evident that in the absence of phase change the continuity equations are reduced to two conservation of species equations.

The momentum equation for the particles

The complete momentum equation for the solid particles is actually a force balance, which in its complete form may be written as

$$\frac{4}{3}\pi a^3 \rho_p \frac{dU_p}{dt} = F_D - \frac{4}{3}\pi a^3 \frac{dP}{dx} + \frac{2}{3}\pi a^3 \rho_g \left(\frac{dU_g}{dt} - \frac{dU_p}{dt}\right) + 3a\rho_g^2 \sqrt{(\pi \nu)} \times \int_0^t \frac{\frac{dU_g}{dt} - \frac{dU_p}{dt}}{\sqrt{(t-t')}} dt' + \frac{4}{3}\pi a^3 (\rho_p - \rho_g) g. \quad (2a)$$

The left-hand side of equation (2a) represents the acceleration of the particle. Of the terms on the right-hand side the first accounts for the viscous interaction with the fluid; the second term is due to the pressure gradient and is usually very small; the third term represents the 'added mass' force and accounts for the volume of fluid the particle carries with it as it accelerates (this is about one-half the volume of the particle); the fourth term is the Basset force, which represents the influence of the history of motion of the particle in its trajectory and is usually very small; finally the last term is the force due to the gravitational field.

When the density of the particles is much higher than the fluid density ($\rho_p/\rho_g \gg 1$), the second, third and fourth terms on the right-hand side become negligibly small. This is usually the case with gas–solid flows where the density ratio is of the order of 1000. Thus, for such flows the equation of motion for the particle reduces to

$$\frac{4}{3}\pi a^3 \rho_p \frac{dU_p}{dt} = F_D + \frac{4}{3}\pi a^3 \rho_p g. \quad (2b)$$

The viscous drag force F_D is given by an empirical relation as follows:

$$F_D = \frac{1}{2} c_D \pi a^2 \rho_g |U_g - U_p| (U_g - U_p) \quad (3)$$

where the coefficient c_D is usually given as a function of the Reynolds number for the particle, Re_p

$$c_D = \frac{24}{Re_p} f(Re_p) \quad (4a)$$

where the function $f(Re_p)$ approaches asymptotically the value 1 as Re_p becomes very small (Stokes law). Boothroyd [12] and Sharma and Crowe [13] suggest the following expression to be used with small to medium size particles:

$$c_D = \frac{24}{Re_p}(1 + Re_p^{0.687}). \quad (4b)$$

This expression has been used successfully in the past for the prediction of particle trajectories and has been verified by comparison with experimental data [12]. The Reynolds number for the particles, which appears in the above equations is

$$Re_p = \frac{2a|U_g - U_p|\rho_g}{\mu}. \quad (5)$$

In pipe flows the momentum equation for the gas is reduced to a pressure loss equation, which may be taken into account with an appropriate friction factor [12] as follows:

$$\frac{dP}{dx} = \frac{1}{2D} f \rho_g U_g^2. \quad (6)$$

The energy equations

For the applications examined in this paper, convection is the predominant mode of heat transfer between the fluid and the wall and also between the fluid and the particles. The fluid interacts thermally with both the pipe and the particles. However, the particles interact only with the fluid. This happens in suspension flows for two reasons: first, the time of collisions of particles with the walls of the pipe are extremely small for any thermal interaction to take place during this time. Second, the mixtures considered here are of small concentrations for an appreciable amount of particles to be in close proximity to the wall of the pipe, where any thermal interaction has to take place. The relatively low temperature differences considered here exclude the possibility of radiation interaction as mentioned before and the absence of any phase change is assumed.

The energy equation for the particles may be reduced to an expression for the change of temperature of the particles

$$\frac{dT_p}{dx} = \frac{4\pi h_p a^2 (T_g - T_p)}{m_p c_p U_p} = \frac{3h_p(T_g - T_p)}{a c_p U_p \rho_p}. \quad (7)$$

Similarly the energy equation for the fluid may be written as an expression for the change of its temperature as follows:

$$\frac{dT_g}{dx} = \frac{\pi D h_g (T_w - T_g)}{\dot{m}_g c_g} + \frac{4\pi \dot{m}_p h_p a^2 (T_p - T_g)}{\dot{m}_g m_p U_p c_g} = \frac{\pi D h_g (T_w - T_g)}{\dot{m}_g c_g} + \frac{3\dot{m}^* h_p (T_p - T_g)}{a c_g U_p \rho_p}. \quad (8)$$

In the last equation the first term emanates from the heat transfer from the particles to the fluid. The second is from the convection through the wall.

The two film coefficients h_p and h_g must be given by closure equations. For spherical particles in a fluid the standard expression for the heat transfer from spheres is taken as follows [15]:

$$h_p = \frac{k}{2a}(2+0.6Re_p^{0.5}\,Pr^{0.33}). \tag{9}$$

The heat transfer coefficient h_g in the absence of particles is given by the Dittus–Boelter expression

$$h_g = 0.023\frac{k}{D}Re^{0.8}\,Pr^{0.4}. \tag{10}$$

This expression is modified by the presence of the particles in the flow. Of all the modifications to the empirical equation suggested in the literature the one derived by Pfeffer *et al.* [7] from a variety of other expressions, seems to represent more adequately the effect of the solids in the increase of the wall to gas film coefficient. This expression is

$$h_g = 0.023\frac{k}{D}Re^{0.8}\,Pr^{0.4}\left(1+4Re^{-0.32}\dot{m}^*\frac{c_p}{c_g}\right). \tag{11}$$

In the above expression the increase of h_g is due to the modification of the flow variables for the gas and not to any particle with wall interactions.

DIMENSIONLESS FORM OF EQUATIONS

The conservation equations may be written in dimensionless form by dividing them with appropriate variables. For the latter, the fluid properties (μ, k, c_g, p_g), the initial temperature difference ΔT and the fluid velocity U based on the final equilibrium density, have been chosen here. This fluid velocity is given by

$$\bar{U} = \frac{4\dot{m}_g}{\pi D^2 \rho_g} \tag{12}$$

where ρ_g is evaluated at the final (equilibrium) fluid temperature.

Thus, equations (1a), (1b), (2b), (7) and (8) become

$$\frac{d}{dx^*}(\rho_g^* U_g^*) = 4E_s^* \tag{13}$$

$$\frac{d}{dx^*}(\rho_s^* U_p^*) = -4E_s^* \tag{14}$$

$$\frac{dU_p^*}{dx^*} = \frac{18}{4}\frac{1}{Re}\left(\frac{D}{a}\right)^2\left(\frac{\rho_g}{\rho_p}\right)\left(\frac{U_g^*}{U_p^*}-1\right) \times (1+0.15Re_p^{0.687}) \tag{15}$$

$$\frac{dT_p^*}{dx^*} = \frac{3}{2}Nu_p\frac{1}{Re\,Pr}\frac{c_g}{c_p}\frac{\rho_g}{\rho_p}\left(\frac{D}{a}\right)^2\left(\frac{U_g^*}{U_p^*}\right)(T_g^*-T_p^*) \tag{16}$$

and

$$\frac{dT_g^*}{dx^*} = \frac{4Nu}{Re\,Pr}(T_w^*-T_g^*)+\frac{3}{2}\frac{\dot{m}^*\,Nu_p}{Re\,Pr} \times\left(\frac{U_g^*}{U_p^*}\right)\frac{\rho_g}{\rho_p}\left(\frac{D}{a}\right)^2(T_p^*-T_g^*). \tag{17}$$

In the above equations the dimensionless groups Nu, Re and Pr refer to the fluid phase and have their usual meaning. Re_p and Nu_p refer to the particles and are defined according to equations (5) and (9). The longitudinal distance x is made dimensionless with respect to the pipe diameter D. Equations (13)–(17) highlight the dimensionless groups that play important roles in the convective heat transfer of suspension flows. Among these groups are the property ratios (c_p/c_g, p_p/p_g, etc.), the slip ratio (U_g^*/U_p^* or U_g/U_p) and the ratio of characteristic dimensions of the pipe to particle ($D/2a$).

If the fluid is a gas then its density may be given by an equation of state. In particular if the gas is considered an ideal gas its density will be inversely proportional to the temperature (given that pressure changes are extremely small in pipe flows, when compared with the absolute pressure of the gas). Then, the ratio of densities may be written as

$$\frac{\rho_p}{\rho_g} = \left(\frac{\rho_p}{\rho_g}\right)^* T_g^* \tag{18}$$

where the term in parenthesis with the asterisk is the density ratio at the final gas temperature and T_g^* corresponds to the thermodynamic temperature of the gas made dimensionless.

In the absence of sublimation, condensation or any other form of phase change, equations (13) and (14) are considerably simplified and yield the velocity of the fluid and the concentration of solids. Also, when the coefficients of equations (15)–(17) are constant (flow of a particle in an infinite domain where transport processes do not affect the gas properties) one may integrate analytically these equations. The solutions obtained are relaxation solutions for the velocity and the temperature of the particles which approach asymptotically the velocity and temperature of the carrier fluid.

SUSPENSION FLOW PAST A TEMPERATURE STEP

As the first application of the subject of heat transfer in suspension flows we consider the flow where the wall temperature undergoes a step from T_1 to T_2, where $T_2 > T_1$. According to the assumption about radiation heat transfer, the temperature difference $\Delta T = T_2 - T_1$ cannot be extremely high. We also assume that there is no phase change in the flow and that the fluid is an ideal gas the density of which behaves in the manner described by equation (18). The initial conditions of the flow are that gas and solid particles are in thermal and mechanical equilibrium.

As the suspension enters the domain of the higher

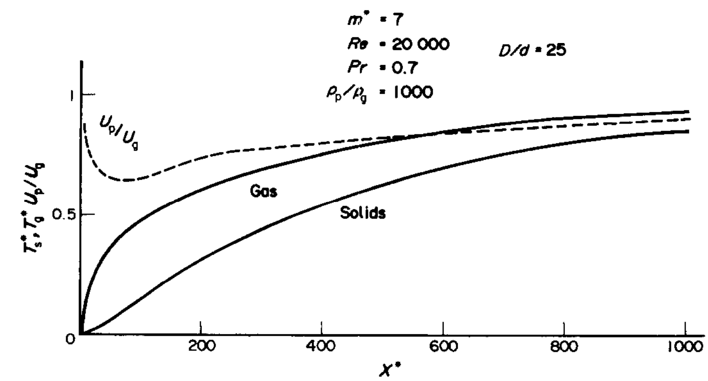

FIG. 1. Temperatures of gas and solid particles and velocity ratio vs the distance from the origin.

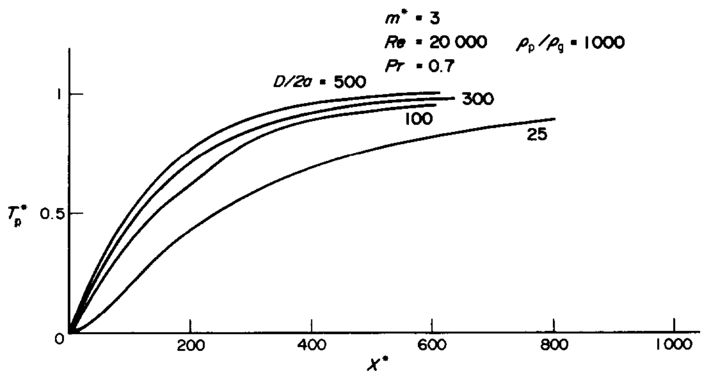

FIG. 2. Particle temperature for different sizes.

temperature T_2, the gas is heated up and accelerates. The solid particles are in turn heated by the gas and they are accelerated. The temperature and velocity of the particles are always lagging behind those variables of the gas (for $T_2 > T_1$). Equations (15)–(17) describe the behaviour of the gas stream and the solid particles. These equations are solved numerically by a step-by-step method. The computational results are shown in the following figures.

Figure 1 shows the evolution of temperatures of the gas and solid particles in the pipe after the step of temperature has been applied. The results are for a Reynolds number of the gas of 20 000, Prandtl number of 0.7, loading of 7 and ratio of pipe to particle diameters of 25. The temperatures are made dimensionless by subtracting T_1 and dividing by ΔT. Thus

$$T_g^* = \frac{T_g - T_1}{T_2 - T_1}. \quad (19)$$

The low value of the $D/2a$ ratio was chosen in order to highlight the thermal and mechanical non-equilibrium which characterizes the flow of the two phases. It may be seen that for the first 400 diameters of the flow the temperatures of the two phases differ by more than 20% and that the slip ratio is greater than 1.2. It is obvious that as the flow progresses the mixture will arrive asymptotically at an equilibrium position with the solids having obtained the temperature and velocity of the gas.

Figure 2 shows the effect of the dimension ratio $D/2a$ on the temperature of the solids T_p^*. It is evident that as the particle diameter decreases the solid particles attain thermal equilibrium with the gas much faster. Actually, when $D/2a = 500$ the values T_g^* and T_p^* are always within 4% and the slip ratio is never more than 1.05. In this case the gas–solid mixture flows in relative thermal and mechanical equilibrium. This figure is drawn for $Re = 20\,000$, $m^* = 3$, $Pr = 0.7$ and $\rho_p/\rho_g = 1000$.

The effect of the loading ratio m^* on the evolution of the solids temperature is shown in Fig. 3. The flow conditions are $Re = 20\,000$, $Pr = 0.7$, $\rho_p/\rho_g = 1000$ and $c_g/c_p = 2$. The pipe to particles diameter ratio is 25, because this case presents more interest (since the gas and particle temperatures differ considerably). The suspensions of higher loading take a longer time to heat up. This is expected, given that these suspensions contain a great deal more material, which is to be heated.

If one were to use suspensions as a heat transfer medium the total transfer per unit length is of importance. For this purpose we estimate the heat transferred to the suspension when the solid particles have reached $T_p^* = 0.8$ and define the following figure of merit for the pipe:

$$\frac{Q^*}{x^*} = \frac{1}{x^*}\frac{c_g \dot{m}_g \Delta T_g + c_p \dot{m}_p \Delta T_p}{c_g \dot{m}_g \Delta T}$$

$$= \frac{1}{x^*}\left(T_g^* + \dot{m}^* \frac{c_p}{c_g} T_p^*\right). \quad (20)$$

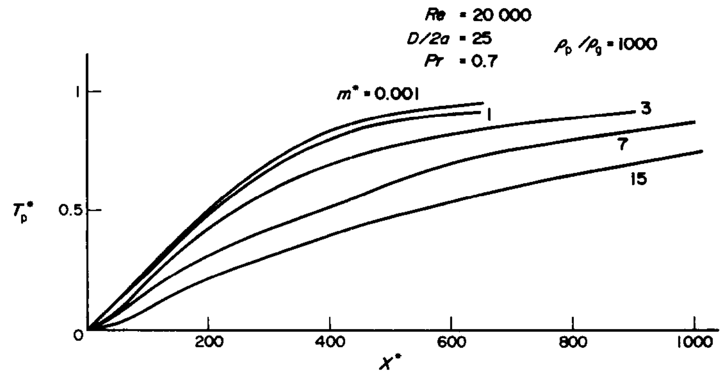

FIG. 3. Particle temperatures for different loadings (coarse size particles).

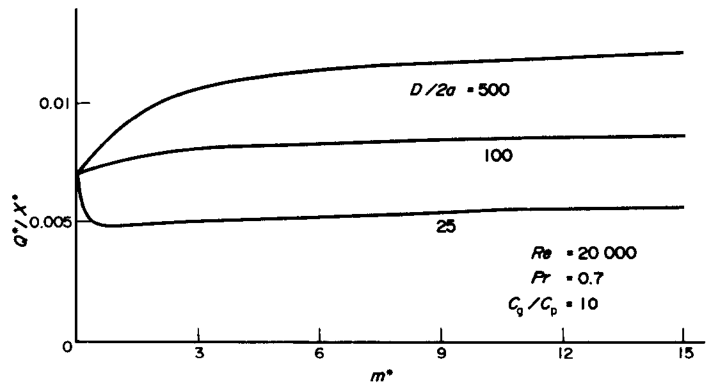

FIG. 4. Heat convected vs loading for different particle sizes.

This quantity is an expression of the heat transferred to the gas–solids mixture until the point when the solids obtain 80% of the imposed temperature difference. Figure 4 depicts Q^*/x^* for various loading ratios and pipe to particle diameter ratios. It is evident that at low $D/2a$ the heat transfer characteristics of the mixture are poor mainly because the large spheres take a long time to be heated. However, there is an improvement in this figure of merit when $D/2a$ is high. This is because the particles absorb some of the heat from the gas and, therefore, keep it cooler; thus the temperature difference between the wall and the gas is high and heat enters more readily in the pipe. A mixture of gas and small particles is obviously a good heat transfer medium, its other disadvantages not withstanding.

Figure 5 shows the effect of the Reynolds number on the heat transferred per unit length of pipe. This figure is for $c_g/c_p = 2$ and $Pr = 0.7$. It can be seen that Q^*/x^* decreases with increasing Re. This may be misleading because of the way Q becomes dimensionless (by dividing with $m_g\, c_p\, \Delta T$). When everything else is equal, a Reynolds number of 50 000 signifies 2.5 times more mass of gas than a Reynolds number of 20 000. Thus, if one were to use a common denominator for both sets of curves, those for $Re = 50\,000$ would have been multiplied by a factor of 2.5 and, hence, they would be above the corresponding curves for $Re = 20\,000$. A glance at both Figs. 5 and 4 also proves that Q^*/x^* is higher when the specific heat of the solids is lower, a fact which can be deduced from equation (16).

INTRODUCTION OF HOT PARTICLES IN A COLD GASEOUS STREAM

The second application we consider is the injection of solid particles at a high temperature T_2 into a gas stream at temperature T, with the walls of the pipe maintained at constant temperature T_1. The solids are injected at lower velocity (slip = 5) into the pipe, mix with the gas and get cooler. The gas is heated up by the solids initially and subsequently cools off because it is in contact with the colder wall.

This physical situation is depicted in Fig. 6 which is drawn for $Re = 20\,000$, $Pr = 0.7$ and $c_g/c_p = 2$. It can be seen that initially the temperature of the gas increases substantially, while that of the particles drops fast. This happens because at the injection point of the solids, gas and particles mix under almost adiabatic conditions (very short length of pipe for the heat to be convected out). At later stages, when the length of the pipe is sufficient to allow enough heat to escape,

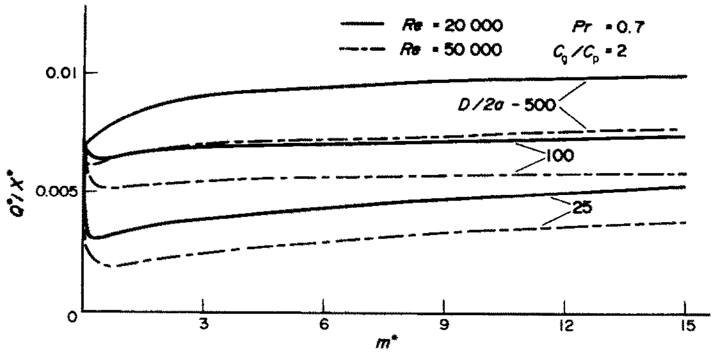

FIG. 5. Heat convected vs loading for two Reynolds numbers.

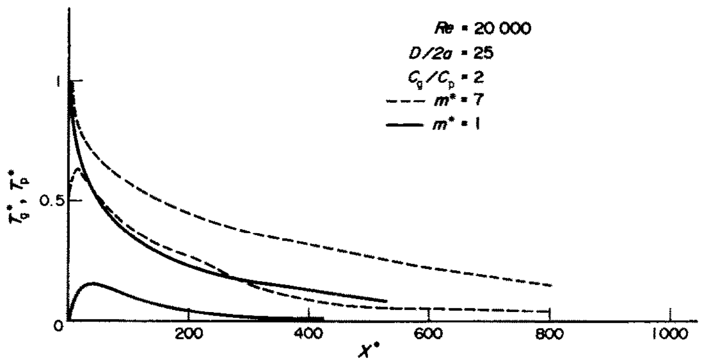

FIG. 6. Particle and gas temperatures vs distance from the origin for the case of cooling of particles.

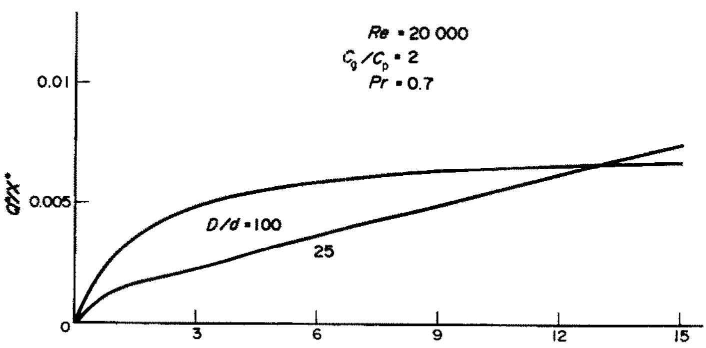

FIG. 7. Heat convected vs loading for two sizes of particles.

the two phases approach thermal equilibrium. The results for the two loadings indicate that the initial rise in the gas temperature is more pronounced when the loading is higher, for reasons which become apparent.

The total heat conveyed from the walls of the pipe per unit length of pipe is depicted in Fig. 7. Again the ratio Q^*/x^* is evaluated when the temperature of the particles reaches 80% of its final value. This figure of merit increases with m^* since the gas becomes hotter.

CONCLUSION

A model of fluid–solid suspension flow with heat transfer has been presented. The model treats the two phases separately and allows for thermal and mechanical non-equilibrium. The dimensionless form of the equations is derived and the groups that influence the heat and momentum exchange between the two phases are pointed out.

The flow and heat transfer equations are solved for two cases: (a) the flow of the suspension past a temperature step and (b) the injection of hot particles into a cold stream. In both cases it is seen that the thermal and mechanical non-equilibrium between the phases becomes more pronounced when larger particles are involved or when the loading is higher.

Among the other parameters of the suspensions examined are the heat transfer per unit length and the effect on it of Reynolds numbers, specific heat ratios, diameter ratios and loadings.

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ECOULEMENTS FLUIDE–SOLIDE HORS EQUILIBRE THERMIQUE ET HYDRODYNAMIQUE

Résumé—Un modèle est développé pour décrire le comportement des particules dans des écoulements d'air. Les équations de l'écoulement des particules et du transfert thermique sont données sous forme adimensionnelle. On traite deux applications pratiques: (a) l'écoulement subit un échelon de température et (b) l'injection de particules chaudes dans un tube canalisant un gaz. On calcule dans les deux cas les différences de vitesse et de température pour le gaz et le solide et on détermine l'influence de plusieurs groupes adimensionnels (comme les nombres de Reynolds, les rapports de charge et de dimension) sur ces deux grandeurs. On trouve que les effets de non-équilibre sont accentués lorsque des plus grosses particules sont dans le mélange.

FLUID-FESTKÖRPER-STRÖMUNG BEI THERMISCHEM UND HYDRODYNAMISCHEM UNGLEICHGEWICHT

Zusammenfassung—Es wird ein Modell entwickelt, um das Verhalten von Partikeln in Luftströmen zu beschreiben. Die Gleichungen für die Partikelströmung und den Wärmetransport werden in dimensionsloser Form angegeben. Es werden zwei praktische Anwendungsfälle für Suspensionsströmungen untersucht: (a) sprunghafte Temperaturerhöhung und (b) Einbringen von heißen Partikeln in eine von Gas durchströmte Rohrleitung. In beiden Fällen werden die momentanen Geschwindigkeits- und Temperaturunterschiede zwischen Gas und Partikeln berechnet und der Einfluß verschiedener dimensionsloser Kennzahlen (wie Reynolds-Zahl, Belastungs- und Geometrie-Verhältnissen) auf diese beiden Größen untersucht. Es ergibt sich, daß Ungleichgewichtseffekte mit wachsender Partikelgröße in der Mischung zunehemen.

ТЕЧЕНИЕ ВЗВЕСЕЙ В УСЛОВИЯХ ТЕПЛОВОЙ И ГИДРОДИНАМИЧЕСКОЙ НЕРАВНОВЕСНОСТИ

Аннотация—Предложена модель для описания поведения частиц в потоке воздуха. Уравнения движения частиц и теплообмена приводятся в безразмерном виде. Даны решения для двух случаев практического использования течения взвесей: а, течение за температурным скачком и б, вдув горячих частиц в трубу с газом. В обоих случаях рассчитываются изменения локальной скорости и температуры газа и твердых частиц, а также влияние нескольких безразмерных критериев (таких как числа Рейнольдса, отношение нагрузок и размеров). Найдено, что с увеличением размера частиц влияние неравновесных эффектов в смеси усиливается.