Pergamon

0301-9322(93)E0025-Z

Int. J. Multiphase Flow Vol. 20, No. 3, pp. 493–504, 1994

0301-9322/94 $7.00 + 0.00

LOCAL ENHANCEMENT OF HEAT TRANSFER IN A PARTICULATE CROSS FLOW—I

HEAT TRANSFER MECHANISMS

D. B. MURRAY

Department of Mechanical Engineering, Trinity College, University of Dublin, Dublin 2, Ireland

(Received 5 October 1992; in revised form 30 November 1993)

Abstract—The mechanisms associated with the enhancement of heat transfer over the front of a tube in a gas–particle cross flow are investigated. A correlation derived in a previous study of enhancement from the increased thermal capacity of the suspension is modified to take account of local particle residence times and the calculations of gas-to-particle heat transfer are refined. The influence on the enhancement of conduction between an impacting particle and the tube wall is analysed and the correlation is modified to include the effect of energy transport by rebounding particles. In addition, the effects of changes in the gas flow structure and boundary-layer characteristics on local heat transfer are explored.

Key Words: gas–particle cross flow, heat transfer enhancement, particle rebounds, increased thermal capacity, turbulence modification

1. INTRODUCTION

The effect of solid particles in suspension on convective heat transfer has been the subject of many experimental investigations, covering a wide range of applications. The possibility of using a graphite dust suspension in gas as a nuclear reactor coolant was examined by Schluderberg *et al.* (1961) and more recently by Hasegawa *et al.* (1983) for higher temperature suspensions. The heat transfer performance of gas–particle suspensions in pipe or duct flows is relevant for processes, such as catalytic reaction or drying, which can be carried out in a transport line. As a consequence, suspension heat transfer in pipelines has been investigated in numerous experimental studies, such as those of Jepson *et al.* (1963) and Wilkinson & Norman (1967). Theoretical studies of the same problem have been made by Michaelides (1986), Kuo & Chiou (1988) and Han *et al.* (1991). For impinging jet flows, experimental and analytical studies have been conducted by Yoshida *et al.* (1990) and by Kurosaki *et al.* (1990). The effect of solid particles on heat transfer for a tube or tube array in cross flow has been investigated primarily for fluidized bed applications (Saxena *et al.* 1978; George & Grace 1982). However, suspension cross-flow heat transfer has also been studied by Woodcock & Worley (1966) for an in-line tube bank and by Murray & Fitzpatrick (1991) for a staggered array of tubes. Although most of these studies found that particles enhanced the convective heat transfer, there were also a number of cases where the Nusselt number was reduced by the presence of a dilute concentration of particles in the flow.

From these investigations, it has been established that the modification of heat transfer in suspension flows is primarily a consequence of the altered thermal capacity of the mixture, along with changes in the flow characteristics of the gas phase. The latter can take the form of a change in the boundary-layer velocity profiles, an increase in the turbulence intensity associated with wake formation of individual particles or turbulence suppression as a result of eddy–particle interactions. For pipe flows, Han *et al.* (1991) found that at low mass loadings the thickness of the viscous sublayer is the most significant parameter, whereas at higher particle concentrations the thermal properties of the suspension are more important. For impinging jet flows, Kurosaki *et al.* (1990) established that conduction between impacting particles and the impingement surface could account for a significant proportion of the measured enhancement of heat transfer. Although several mechanisms of suspension heat transfer have been identified, the relative significance of these separate effects for tubes in cross flow has not been established. In a study by Murray &

Fitzpatrick (1991), the manner by which heat transfer is both enhanced and reduced at different locations within a staggered tube array subject to a gas–particle cross flow was investigated. Although a correlation was developed to predict the enhancement based on the increased thermal capacity of the mixture, no attempt was made to account for heat transfer by conduction between impacting particles and the tube wall. Furthermore, the effect of energy transport by particle rebounds from the heated zone around the tube was not addressed, although a similar mechanism has been identified by Depew & Kramer (1972) as significant for suspension pipe flows.

In this paper, a detailed analysis of the local enhancement of heat transfer over the front of a tube in a gas–particle cross flow is described. A correlation developed previously to account for the increased thermal capacity mechanism is modified to account for the effects of wall-to-particle conduction and the energy transport by particle rebounds. In addition, the potential effect on heat transfer of changes in the gas flow characteristics due to the presence of the particles is examined.

2. SUSPENSION HEAT TRANSFER MECHANISMS

The main mechanisms by which suspended particles affect heat transfer between a flowing gas and a surface may be identified as the following:

(1) Increased thermal capacity of the suspension.
(2) Conduction between impacting particles and the tube wall.
(3) Thermal energy transport by rebounding particles.
(4) Changes in the boundary-layer characteristics and turbulent structure of the gas flow.

The present study is restricted to cases where temperature levels are close to ambient and in which the temperature difference between the suspension and surface is small. As a consequence, it is probable that any changes in heat transfer result primarily from the effect of the particles on convective rather than radiative heat transfer. Other factors, such as deposition or thermophoresis, can influence the heat transfer performance of a suspension under certain conditions. The remainder of this paper will investigate the physical basis for the mechanisms listed above and the manner in which they may contribute to the enhancement of heat transfer over the front of a cylinder in cross flow. Note that, although treated separately in the first instance, mechanisms (1)–(3) above are interlinked and will be drawn together in a single calculation encompassing all three aspects.

3. INCREASED THERMAL CAPACITY MECHANISM

3.1. *Existing correlation*

A correlation for the enhancement of heat transfer from the increased thermal capacity of the suspension was developed by Murray & Fitzpatrick (1991) for gas–particle cross flows. The Nusselt number of the suspension relative to that of the single-phase flow was given by

$$\frac{\mathrm{Nu_{su}}}{\mathrm{Nu_a}} = [1 + \eta_t S_L c_{pr}]^{0.37}, \qquad [1]$$

where S_L is the solids mass loading ratio, c_{pr} is the ratio of the specific heat of the solid to that of the gas phase and the parameter η_t, defined as the thermal effectiveness factor, represents the ratio of a characteristic residence time for the solid (t_{res}) to the particle thermal response time (t_{rel}). The subscripts su and a refer to the suspension and air flows, respectively.

The justification for [1] is as follows. Although the specific heat (c_p) of solid particles is often similar to that of air, their presence in an air flow will increase the volumetric heat capacity as a result of the high density of the solid. This increase in thermal capacity depends on the solids concentration and on the specific heat ratio of the two phases but the resultant enhancement of heat transfer may also vary with particle size and flow Reynolds number. This is because an increase in Nusselt number can only result if the particles have sufficient time in the heated or cooled zone to exchange heat with the surrounding fluid. Consequently, the greatest enhancement of heat transfer due to the increased thermal capacity effect is anticipated for suspensions of small particles and low velocities. Equation [1] was developed from an empirically based relationship between the Nusselt and Prandtl numbers for a single-phase cross flow. In this analysis, it was assumed that any change in the effective viscosity of the suspension is small, as indicated by Jepson *et al.* (1963). In addition, a simplified calculation by Murray (1989) of one-dimensional conduction through a dilute air–solid mixture showed that the dependence on thermal conductivity is small relative to the specific heat effects. The gas–particle mixture was treated initially as a homogeneous fluid with the equivalent properties of the suspension but the thermal effectiveness factor, η_t, was introduced to account for the effects of particle size and Reynolds number.

Although [1] provides trends which are broadly similar to the measured variation in mean Nusselt number at the first row of a model tube bank (Murray & Fitzpatrick, 1991), this analysis cannot be used to predict the local enhancement over the front of a tube in cross flow. The reasons for this are as follows. Firstly, the original calculation of particle residence time was based on the total time taken by particles in passing over the tube. However, the time available to a particle to absorb heat from the surrounding fluid at a given angular location will be much less than that calculated from the global estimate just described, with a consequent reduction in the local enhancement. In addition, the original definition of the thermal effectiveness factor as the ratio of the particle residence to thermal response times neglects the fact that the thermal response time is a first-order time constant and represents the time taken for 63% (rather than 100%) of the potential heat transfer between the particle and the surrounding fluid to occur. Thus, all estimates of the enhancement based on this definition of η_t will be excessive.

3.2. *Local residence time*

Equation [1] was based originally on the limiting condition of a suspension of infinitesimally small particles which follow the fluid streamlines (i.e. a homogeneous fluid). This model is adequate for suspension of very fine dust particles ($\simeq 1\ \mu$m dia) and, in this situation, no correction for particle response times is necessary. For most gas–solid cross flows, however, particles do not follow fluid streamlines and instead travel across the boundary layer as a result of their own inertia. This is illustrated in figure 1. It is likely that many particles rebounding from the tube surface will enter the boundary-layer region for a second time, resulting in an apparent increase in either the local particle concentration or in the particle residence times. However, as this effect is difficult to quantify, the minimum distance moved by the particles within the heated fluid zone (twice the thermal boundary-layer thickness) is used in the present analysis. The thermal boundary-layer thickness, δ_t, is taken as midway between the value calculated for air flow over a cylinder (Schlichting 1979) and that estimated for a homogeneous fluid with the equivalent thermal properties of the suspension under consideration.

The particle velocity will lag behind the upstream air velocity by an amount related to the terminal velocity, with an additional difference linked to the particle rebound process. The residence time calculation may be adjusted to account for this difference, using a modifying factor chosen to match the calculated particle velocities with measured values from experimental studies of suspension cross flows. Thus, the local residence time necessary to estimate the enhancement from the increased thermal capacity effect is given by

$$t_{res} = \frac{2\delta_t}{V_p} = \frac{2\delta_t}{A(V_a - V_{ter})}, \qquad [2]$$

where V_a is the upstream air velocity, V_{ter} is the particle terminal velocity, V_p is the average particle velocity in the boundary layer and A is a modifying factor chosen to match the calculated particle velocities with measured values for equivalent conditions. [A value of 0.75 is used in all the present calculations, as this gives velocities close to those measured by Fitzpatrick *et al.* (1992) using a laser Doppler anemometer in a small triangular tube array for similar suspension flow conditions.]

This method of estimating the particle residence time, although appproximate, has a strong physical basis and should yield the correct order of magnitude for this parameter. Although it may seem inconsistent to use time periods based on particle trajectories across the boundary layer in conjunction with a correlation derived for homogeneous flow, this approach leads to an initial estimate of the heat transfer enhancement due to the increased thermal capacity of the suspension.

The specific effect on heat transfer of the energy transport associated with particle rebounds will be addressed in section 5.

3.3. Thermal effectiveness factor

The thermal effectiveness factor, η_t, previously defined as the ratio of the particle residence and thermal response times, can be modified to represent the ratio of the actual heat absorbed by a particle to the maximum possible heat transfer. Analysis of the convective heat transfer process between a particle and a surrounding fluid leads to the following expression for the particle temperature variation with time:

$$(\rho c_p)_p V \frac{dT_p}{dt} = hA_s[T_a - T_p]; \qquad [3]$$

T_p is the particle temperature, A_s and V are the surface area and volume of the particle, respectively, and $(\rho c_p)_p$ denotes the density × specific heat of the solid material; h is the convective heat transfer

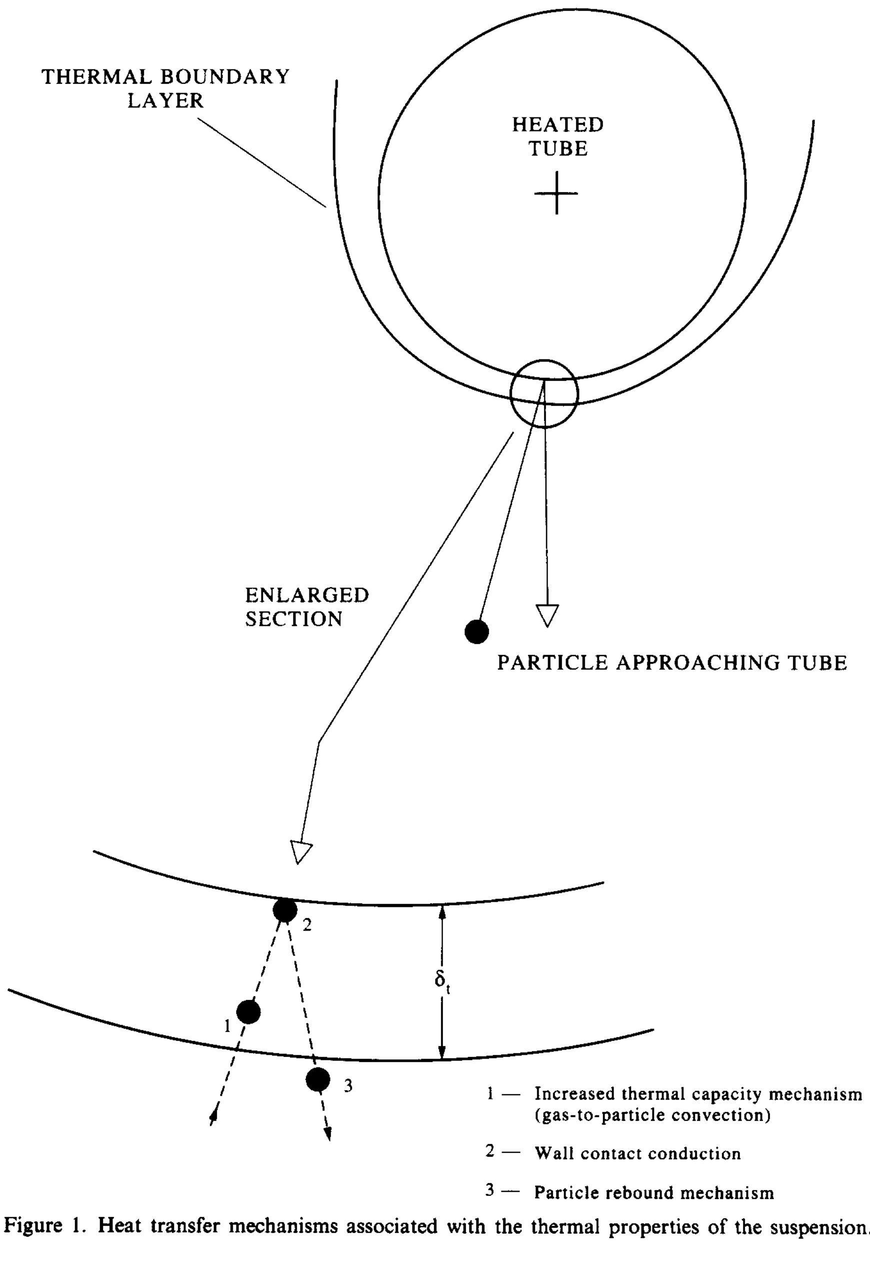

Figure 1. Heat transfer mechanisms associated with the thermal properties of the suspension.

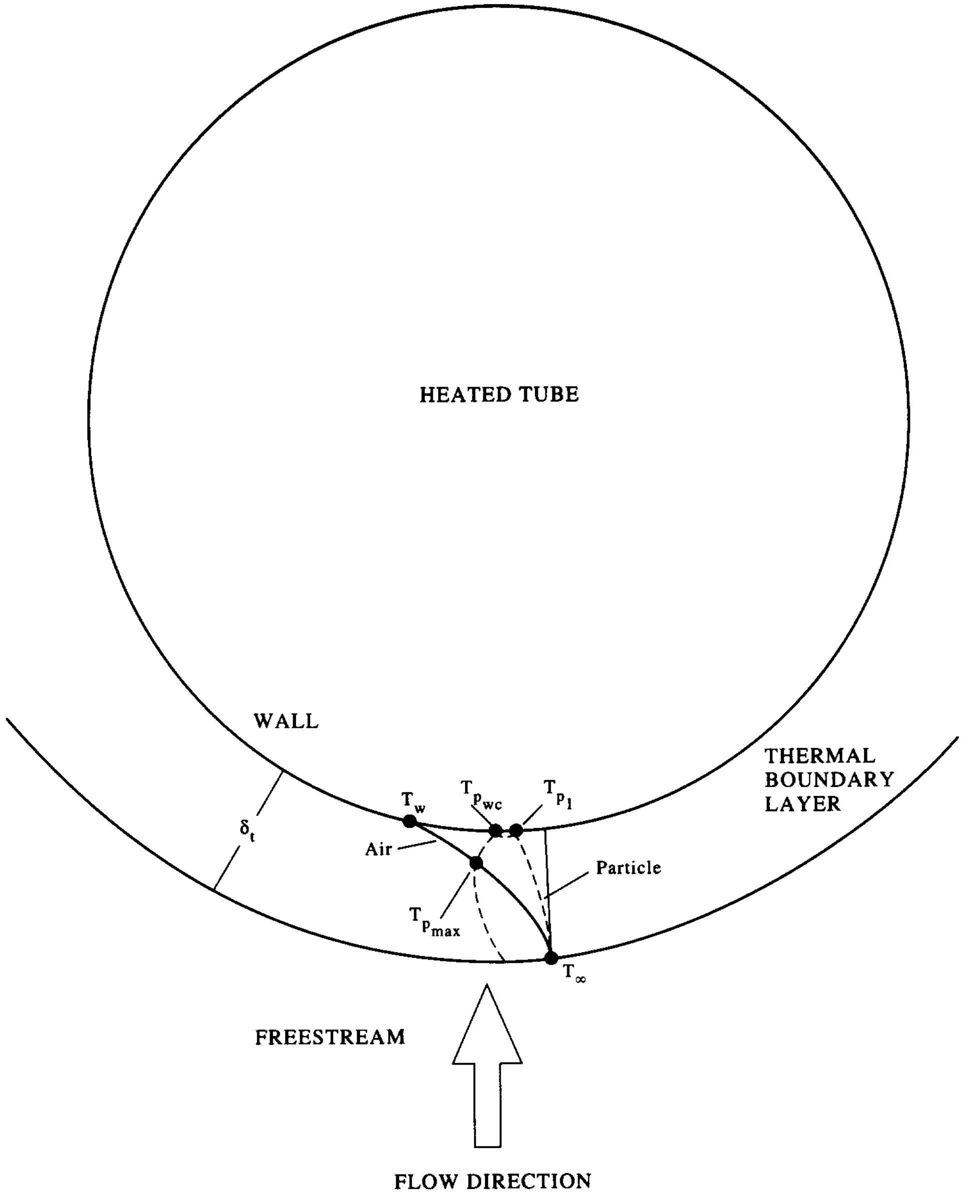

Figure 2. Air and particle temperature profiles in the near-wall region.

coefficient between a spherical particle and air, estimated from the Nusselt number correlation of Hughmark (1967). For a surrounding fluid of uniform temperature, evaluation of the particle/air temperature difference as a function of time leads to an expression for the thermal response or relaxation time as

$$t_{rel} = \frac{V(\rho c_p)_p}{hA_s}. \qquad [4]$$

In the present case, the air temperature varies parabolically with distance from the tube wall, as shown in figure 2, and the rise in temperature of the particle as it passes through the boundary layer will reflect this. The air temperature profile for the laminar boundary layer over the front of the tube may be represented as follows (Eckert & Drake 1959), where y is the distance measured normal to the tube wall and δ_t is an average value for the zone in question:

$$T_a = T_w + (T_\infty - T_w)\left[\frac{3}{2}\left(\frac{y}{\delta_t}\right) - \frac{1}{2}\left(\frac{y}{\delta_t}\right)^3\right]. \qquad [5]$$

This variation in air temperature with position can be translated into a temporal variation, from the viewpoint of the particle passing through the boundary layer. Thus, the change in the particle temperature as it moves from the outer edge of the thermal boundary layer up to the tube surface is determined from [3], where the variable y used in calculating the local air temperature is taken as $V_p(t_{res_l} - t)$, t_{res_l} being the time taken for a single boundary-layer traversal, i.e. $0.5t_{res}$. Solution of [3] for this case leads to an expression for the particle temperature as it contacts the tube surface for the first time:

$$T_{p_l} = T_\infty + (T_\infty - T_w)\left[\frac{3}{2}\frac{t_{rel}}{t_{res_l}} - 3\left(\frac{t_{rel}}{t_{res_l}}\right)^3 - 1\right] + \frac{(T_\infty - T_w)}{\exp\left(\dfrac{t_{res_l}}{t_{rel}}\right)}\left[3\left(\frac{t_{rel}}{t_{res_l}}\right)^3 + 3\left(\frac{t_{rel}}{t_{res_l}}\right)^2\right]. \qquad [6]$$

The change in the particle temperature associated with movement back from the tube surface is obtained by defining the variable y as $V_p t$, resulting in the following equation for the particle temperature variation with time:

$$T_p = T_w + (T_\infty - T_w)\left[\frac{3}{2}\frac{t}{t_{res_l}} - \frac{3}{2}\frac{t_{rel}}{t_{res_l}} - \frac{1}{2}\left(\frac{t}{t_{res_l}}\right)^3 + \frac{3}{2}\frac{t^2 t_{rel}}{t_{res_l}^3} - 3\frac{t\,t_{rel}^2}{t_{res_l}^3} + 3\left(\frac{t_{rel}}{t_{res_l}}\right)^3\right]$$
$$+\exp\left(\frac{-t}{t_{rel}}\right)\left\{(T_{p_l} - T_w) + (T_\infty - T_w)\left[\frac{3}{2}\frac{t_{rel}}{t_{res_l}} - 3\left(\frac{t_{rel}}{t_{res_l}}\right)^3\right]\right\}. \qquad [7]$$

The maximum particle temperature reached on traversal through the thermal boundary layer, determined from [7], may now be used to define the modified thermal effectiveness factor:

$$\eta_t = \frac{q_p}{q_{p_{max}}} = \frac{T_{p_{max}} - T_\infty}{T_w - T_\infty}. \qquad [8]$$

Equation [8] yields a more conservative estimate of the enhancement from the increased thermal capacity effect than obtained from the original correlation based on the residence and response time ratio. The present approach reflects the fact that an infinite time period is strictly required for the particles to participate fully in the convective heat transfer process between the surface and suspension.

4. WALL CONDUCTION MECHANISM

The analysis of Murray & Fitzpatrick (1991) assumed that all of the improvement in heat transfer performance resulted from convective heat transfer between the air and the particles. A number of studies, such as that of Depew & Kramer (1972), have suggested that the conduction between impinging particles and a solid wall will normally be insignificant due to the small area and short time of contact. However, it has been shown by Kurosaki *et al.* (1990) that this mechanism can contribute a significant proportion of the heat transfer enhancement for impinging jet flows. As a consequence, the possibility of additional enhancement due to wall contact conduction must be explored.

An analysis of heat transfer by conduction between impinging particles and a heated tube has been performed by Murray (1992). Estimates of the contact radius and contact time were initially based on fully elastic collisions between a particle and the tube but later calculations investigated the effect of plastic deformation of the tube surface. The particle and tube were treated as two semi-infinite solids with no thermal resistance between the two contacting surfaces. It has been shown by Sun & Chen (1988) that, for cases where the contact time is short, it is sufficient to use a simple one-dimensional calculation for this problem. The total energy exchange on impact (q_{imp}) is then given by

$$q_{imp} = \frac{0.87(T_w - T_{p_l})\pi r_c^2 t_c^{1/2}}{(\rho c_p k)_p^{-1/2} + (\rho c_p k)_w^{-1/2}}, \qquad [9]$$

where t_c is the contact time, r_c is the contact radius, $(T_w - T_{p_l})$ is the initial temperature difference between the two bodies and k is the thermal conductivity. Subscripts p and w refer to the particle and wall materials, respectively. Details of the wall conduction calculations are given by Murray (1992).

The potential enhancement from wall conduction can be included in the heat transfer correlation as an integral component of the increased thermal capacity mechanism. This is achieved by using the estimate of heat transfer to the particle when in contact with the wall, q_{imp}, to calculate the particle temperature following wall contact:

$$T_{\text{p}_{\text{wc}}} = T_{\text{p}_{\text{i}}} + \frac{q_{\text{imp}}}{m_{\text{p}} c_{\text{pp}}}; \qquad [10]$$

m_{p} is the mass of the particle. This temperature may then be used in place of $T_{\text{p}_{\text{i}}}$ in [7] as the initial particle temperature for the rebound process. Equation [1] is still used to evaluate the enhancement due to the increased thermal capacity effect but the conduction to the particle during wall contact has now been incorporated in the calculation of the thermal effectiveness factor, η_{t}.

5. PARTICLE REBOUND MECHANISM

Another mechanism of potential heat transfer enhancement is linked to the particle rebound process. Particles which have absorbed heat in the thermal boundary layer over the front of the tube will rebound out of this region, thereby transporting the stored thermal energy with them. In single-phase cross flows with high freestream turbulence, the energy transport associated with the motion of small turbulent eddies across a nominally laminar boundary layer contributes to the high heat transfer coefficients observed. In the same way, it might be expected that the energy transport associated with the particle rebound mechanism would give rise to an increase in heat transfer for suspension cross flows.

The heat transfer enhancement from the particle rebound mechanism can be estimated from the rate at which thermal energy is carried out of the boundary layer by rebounding particles. This can be written as

$$q''_{\text{pr}} = \dot{n} m_{\text{p}} c_{\text{pp}} (\eta_{\text{t}}) [T_{\text{w}} - T_{\infty}], \qquad [11]$$

where $\dot{n}$ is the number of particle rebounds per m^2 per s. This equation, which is based on the maximum temperature reached by the particle, represents an upper limit to this effect.

Comparison of this particulate heat flux with the single-phase heat flux leads to an expression for the heat transfer enhancement due to particle rebounds:

$$\left(\frac{\text{Nu}_{\text{su}}}{\text{Nu}_{\text{a}}}\right)_{\text{pr}} = 1 + \frac{q''_{\text{pr}}}{q''_{\text{a}}}, \qquad [12]$$

where the single-phase heat flux, q''_{a}, is the product of the single-phase convective heat transfer coefficient and the wall/freestream temperature difference.

An alternate approach to assessing the potential enhancement from the particle rebound mechanism is based on the analogy with freestream turbulence effects. The net energy transfer across the boundary layer associated with a turbulent air flow can be approximated as

$$q''_{\text{turb fluid}} = \left(\frac{\dot{m}_{\text{turb fluid}}}{A_{\text{f}}}\right) c_{\text{pa}} [T_{\text{w}} - T_{\infty}], \qquad [13]$$

where $\dot{m}_{\text{turb fluid}}$ is the average mass flow rate associated with the turbulent velocity and A_{f} is the flow area. Equating the total rate of energy transport by rebounding particles, from [11], to that for turbulent air flow, from [13], leads, following some simplification, to an expression for the ratio of an equivalent turbulent air velocity (V_{turb}) to the upstream mean air velocity (V_{a}):

$$\frac{V_{\text{turb}}}{V_{\text{a}}} = Tu_{\text{eq}} = S_{\text{L}} c_{\text{pr}} \eta_{\text{t}}. \qquad [14]$$

A correlation suggested by Dyban & Epick (1970) for the effect of turbulence intensity on the Nusselt number over the front of a tube in cross flow may now be used to predict the local increase in heat transfer:

$$\left(\frac{\text{Nu}_{\text{su}}}{\text{Nu}_{\text{a}}}\right)_{\text{pr}} = \frac{1 + 0.01(\text{Re}\, Tu_{\text{su}})^{0.5}}{1 + 0.01(\text{Re}\, Tu_{\text{a}})^{0.5}}, \qquad [15]$$

where Tu_{su} represents the sum of the equivalent turbulence intensity for particle rebounds and the actual turbulence intensity of the gas flow.

It is evident that the calculation of the equivalent turbulence intensity is speculative, so that an exact estimate of the heat transfer enhancement cannot be expected. Nevertheless, the level of enhancement to be anticipated from the particle rebound mechanism may be estimated by comparison of the results from the two alternate calculations proposed to quantify this effect.

In this analysis, all of the energy absorbed by the particles, via gas convection and wall conduction, has been assumed to be carried away by the rebounding particles. As a consequence, the particle transport mechanism will encompass the enhancement associated with the increased thermal capacity of the suspension. This issue of the interdependence of the various heat transfer mechanisms will be explored later.

6. FLOW STRUCTURE EFFECTS

The influence of the physical properties of a suspension on cross-flow heat transfer is not well-defined. However, as a first approximation, the potential enhancement resulting from changes in the boundary-layer characteristics due to the particles can be estimated from

$$\left(\frac{\mathrm{Nu}_{su}}{\mathrm{Nu}_a}\right)_{\mathrm{Re}} = (1 + S_L)^{0.5}. \qquad [16]$$

This relationship originates in the dependence of the single-phase Nusselt number on $(\mathrm{Re})^{0.5}$ for the front stagnation point of a tube in cross flow. The fluid density has been replaced with an effective density based on a homogeneous model of the suspension flow.

Such enhancement of heat transfer might be linked to a change in the local velocity field or a reduction in the local boundary-layer thickness. Boundary-layer thinning has been observed over the sides of a tube in a suspension cross flow (Fitzpatrick *et al.* 1992), in regions where the momentum transfer from particles to gas contributes to an increase in the wall velocity gradients. This concept is illustrated in figure 3. For a suspension of infinitesimally small particles, a reduction in the boundary-layer thickness may take place over the front of the tube, leading to an increase in the local Nusselt number, as estimated in [16]. However, heat transfer enhancement in this region may also result from a local increase in the turbulence brought about by the movement of particles through the boundary layer, as illustrated in figure 3. For suspension flows with low particle Reynolds numbers, $\sim O(10)$, the suspended solids will not shed vortices as they move through the flow, although the possibility exists that rebounding particles may induce some gas-phase flow reversal, as described in Yoshida *et al.* (1990). Where particle Reynolds numbers are high, vortex shedding by individual particles may lead to small-scale turbulence generation, with a consequent increase in local heat transfer.

The present study is mainly directed towards fine particle suspensions with particle Reynolds numbers of order 10 or less. As a consequence, specific turbulence generation due to vortex shedding by particles is not anticipated. [For cases where the particle Reynolds numbers are of a higher order, or where the ratio of the particle diameter to typical eddy sizes suggests that turbulence enhancement is likely (Gore & Crowe 1989), further specific investigations may be necessary to estimate the potential effect on heat transfer.] As changes in the local flow structure are linked to the higher effective density for fine particle suspensions, [16] will be used to estimate the enhancement of heat transfer from this effect.

It is also possible for the presence of solid particles in a gas flow to reduce the level of turbulence as a result of eddy–particle interactions (Hetsroni 1989). Analysis of the work done by an eddy on the particles it encounters led Owen (1969) to suggest that the ratio of the particle relaxation time, t^*, to the characteristic time scale of eddies, t_e, be used for assessing turbulence modification trends. (The relaxation time, t^*, is the time taken for a particle at rest to be accelerated to within 63% of the fluid velocity and can be written as $d_p^2\rho_p/18\nu\rho_a$; the characteristic time scale for the eddies is their characteristic size divided by some characteristic velocity, l_e/V_e.) For very small particles, where $t^* \ll t_e$, Owen (1969) suggested that the rate of turbulent energy dissipation is increased, compared to single-phase flow, by a ratio of $(1 + S_L)^{1/2}$, where S_L is the solids mass loading ratio. For larger particles with $t^* > t_e$, Owen (1969) suggested that the turbulent fluctuations in the presence of the particles should decrease as $[1 + (S_L)(t_e/t^*)]^{-1/2}$, compared with the particle freestream. (For very large particles, where $t^* \gg t_e$, the particles were considered to make little contribution to turbulence dissipation and may generate turbulence instead, as discussed above.)

Once the altered turbulence intensity is predicted from the analysis of Owen (1969), the local change in the Nusselt number as a consequence of turbulence modification can be estimated from the correlation of Dynan & Epick (1970), as shown in [15]. Note that [15] relates to an equivalent (imaginary) turbulence intensity representing the effect of particle rebounds, whereas the present calculation refers to a change in the actual turbulence intensity of the gas flow.

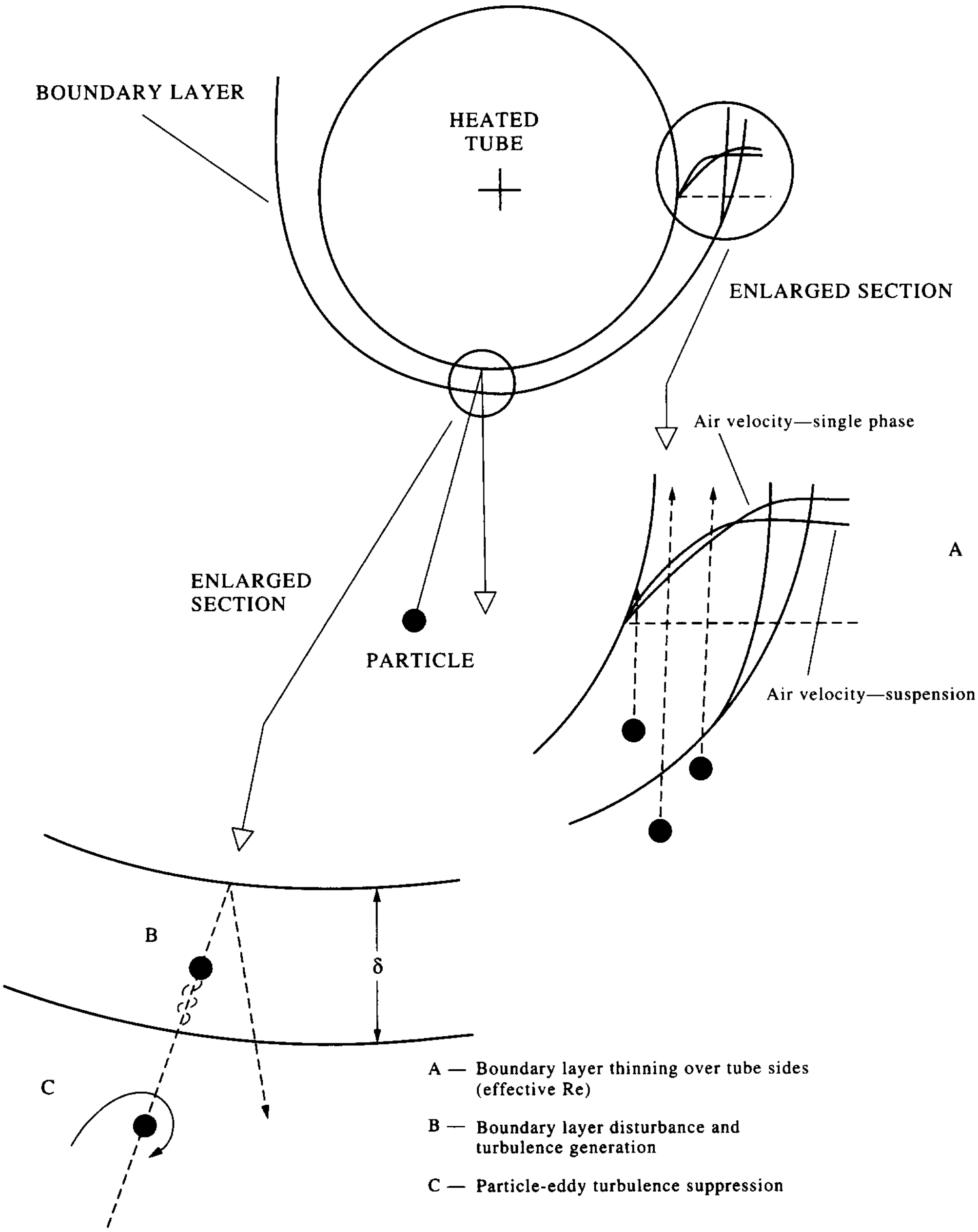

Figure 3. Heat transfer mechanisms associated with the physical properties of the suspension.

Turbulence suppression by particles will reduce rather than enhance heat transfer. Nevertheless, this must be taken into consideration as any measured change in heat transfer may result from a combination of both positive and negative effects. It is evident that significant scope for turbulence suppression exists in the highly turbulent flow encountered within a tube array. In contrast, changes in heat transfer as a result of turbulence suppression are likely to be less significant for a single tube in cross flow or for a tube in the first row of a tube bank.

7. SUMMARY OF THE LOCAL ENHANCEMENT MECHANISMS

The main mechanisms of heat transfer enhancement associated with the thermal properties of a suspension cross flow are illustrated in figure 1 and those linked to the physical properties and flow structure are shown in figure 3. Before estimating the overall effect of particles on cross-flow heat transfer, two aspects of this analysis merit further discussion. Firstly, as mentioned in section 5, the particle rebound calculations are based on the assumption that all of the energy absorbed by the particles is convected away in the rebound process. Thus, the inclusion of both increased thermal capacity and particle rebound terms in the assessment of the total enhancement would represent a "double accounting" or overestimate of the heat transfer from this source. Secondly, in this analysis all of the heat transfer mechanisms have been treated as independent, whereas there are reasons to believe that the effects are linked. In particular, the increased thermal capacity and effective Reynolds number terms both originate from the same correlation for single-phase heat transfer, suggesting a multiplicative rather than an additive relationship.

As the relative significance of the heat transfer mechanisms is strongly dependent on the type of suspension flow, it is appropriate to consider two limiting cases. For very fine particle suspensions that approximate a homogeneous fluid of higher density and heat capacity, the particle rebound effect will be insignificant and the remaining terms are likely to be linked in a multiplicative manner, resulting in the following equation for the total enhancement in such flows:

$$\left(\frac{\mathrm{Nu_{su}}}{\mathrm{Nu_a}}\right)_{\mathrm{total_{fp}}} = \left(\frac{\mathrm{Nu_{su}}}{\mathrm{Nu_a}}\right)_{\mathrm{itc}} \times \left(\frac{\mathrm{Nu_{su}}}{\mathrm{Nu_a}}\right)_{\mathrm{Re}} \times \left(\frac{\mathrm{Nu_{su}}}{\mathrm{Nu_a}}\right)_{Tu}. \qquad [17]$$

The subscript fp denotes a fine particle suspension. Note that the thermal effectiveness factor used in the increased thermal capacity calculation would strictly have a value of 1 for a homogeneous model of a suspension flow.

In contrast, for large particle suspensions the particle rebound mechanism is most significant and encompasses the increased thermal capacity effect. It is considered unlikely that a large particle suspension will cause the reduction in the boundary-layer thickness anticipated for a homogeneous fluid of higher effective density, i.e. the effective Reynolds number term can be neglected. Modification of the gas-phase turbulence is anticipated but the change in heat transfer resulting from this effect is not directly linked to the energy transfer by particle rebounds. As a consequence, these effects will be assumed to be independent and additive, leading to the following equation for the enhancement of cross-flow heat transfer in large particle suspensions:

$$\left(\frac{\mathrm{Nu_{su}}}{\mathrm{Nu_a}}\right)_{\mathrm{total_{lp}}} = 1 + \left[\left(\frac{\mathrm{Nu_{su}}}{\mathrm{Nu_a}}\right)_{\mathrm{pr}} - 1\right] + \left[\left(\frac{\mathrm{Nu_{su}}}{\mathrm{Nu_a}}\right)_{Tu} - 1\right]. \qquad [18]$$

Two separate calculations, given by [12] and [15], can be used to quantify the particle rebound effect. The subscript lp refers to a large particle suspension.

8. CONCLUSION

The mechanisms by which suspended particles can influence convective heat transfer over the front of a tube in cross flow have been investigated. Analysis has shown that the increased thermal capacity of the suspension will lead to higher Nusselt numbers, provided that the time available for heat transfer is of the same order as the particle thermal response times. This effect will be assisted by particle rebounds from the heated fluid zone and by wall-to-particle conduction. Additional enhancement of heat transfer may result from the higher effective density and Reynolds number of the suspension and from local turbulence generation by particles. Finally, the suppression of turbulence due to eddy–particle interactions may cause a local reduction in heat transfer. The relative significance of each of these mechanisms will depend on the suspension flow parameters and two limiting cases—a large particle suspension and a flow with infinitesimally small particles—have been identified. Comparison of the trends predicted from this analysis with specific experimental data is necessary for the validity of the proposed correlations to be assessed.

Acknowledgement—The author would like to acknowledge the useful advice and encouragement of Professor J. A. Fitzpatrick in the preparation of this paper.

REFERENCES

DEPEW, C. A. & KRAMER, T. J. 1972 Heat transfer to flowing gas–solid mixtures. *Adv. Heat Transfer* **9,** 113–180.

DYBAN, E. & EPICK, E. 1970 Some heat transfer features in the air flow of intensified turbulence. In *Proc. 4th Int. Heat Transfer Conf.*, Paris, Vol. 2, paper FC 5.7.

ECKERT, E. R. G. & DRAKE, R. M. 1959 *Heat and Mass Transfer*, 2nd edn. McGraw–Hill, New York.

FITZPATRICK, J. A., LAMBERT, B. & MURRAY, D. B. 1992 Measurements in the separation region of a gas–particle cross flow. *Expts Fluids* **12,** 329–341.

GEORGE, S. E. & GRACE, J. R. 1982 Heat transfer to horizontal tubes in the freeboard region of a gas fluidized bed. *AIChE Jl* **28,** 759–765.

GORE, R. A. & CROWE, C. T. 1989 Effect of particle size on modulating turbulence intensity. *Int. J. Multiphase Flow* **15,** 279–285.

HAN, K. S., SUNG, H. J. & CHUNG, M. K. 1991 Analysis of heat transfer in a pipe carrying two-phase gas–particle suspension. *Int. J. Heat Mass Transfer* **34,** 69–78.

HASEGAWA, S., ECHIGO, R., KANEMARU, K., ICHIMIYA, K. & SANUI, M. 1983 Experimental study on forced convective heat transfer of flowing gaseous solid suspension at high temperature. *Int. J. Multiphase Flow* **9,** 131–145.

HETSRONI, G. 1989 Particle–turbulence interaction. *Int. J. Multiphase Flow* **15,** 735–746.

HUGHMARK, G. A. 1967 Mass and heat transfer from rigid spheres. *AIChE Jl* **13,** 1219–1221.

JEPSON, G., POLL, A. & SMITH, W. 1963 Heat transfer from gas to wall in a gas–solids transport line. *Trans. Instn Chem. Engrs* 207–211.

KUO, J. T. & CHIOU, C. H. 1988 Momentum and heat transfer of gas–solids suspensions in vertical pipes. *AIChE Symp. Ser.* **84**(263), 31–37.

KUROSAKI, Y., SATOH, I., KAMEOKA, Y. & ANNMO, Y. 1990 Mechanisms of heat transfer enhancement around the stagnation point of an impinging air jet laden with solid particles. In *Proc. 9th Int. Heat Transfer Conf.*, Jerusalem, Vol. 4, pp. 99–104.

MICHAELIDES, E. E. 1986 Heat transfer in particulate flows. *Int. J. Heat Mass Transfer* **29,** 265–273.

MURRAY, D. B. 1989 The effect of solid particles on cross flow heat transfer. Ph.D. Thesis, Trinity College, Univ. of Dublin.

MURRAY, D. B. 1992 Cross flow heat transfer with impinging particles. In *Recent Advances in Heat Transfer*, pp. 134–148. Elsevier, Amsterdam.

MURRAY, D. B. & FITZPATRICK, J. A. 1991 Heat transfer in a staggered tube array for a gas–solid suspension flow. *Trans. ASME J. Heat Transfer* **113**, 865–873.

OWEN, P. R. 1969 Pneumatic transport. *J. Fluid Mech.* **39,** 407–432.

SAXENA, S. C., GREWAL, N. S. & GABOR, J. D. 1978 Heat transfer between a gas fluidized bed and immersed tubes. *Adv. Heat Transfer* **14,** 149–247.

SCHLICHTING, H. 1979 *Boundary-layer Theory*, 7th edn, Chap. 10. McGraw–Hill, New York.

SCHLUDERBERG, D. C., WHITELAW, R. L. & CARLSON, R. W. 1961 Gaseous suspensions—a new reactor coolant. *Nucleonics* **19,** 67–76.

SUN, J. & CHEN, M. M. 1988 A theoretical analysis of heat transfer due to particle impact. *Int. J. Heat Mass Transfer* **31,** 969–975.

WILKINSON, G. T. & NORMAN, J. R. 1967 Heat transfer to a suspension of solids in a gas. *Trans. Instn Chem. Engrs* **45,** 314–318.

WOODCOCK, M. T. & WORLEY, N. G. 1966 Gas–solid suspensions as heat transfer media. *Proc. Instn Mech. Engrs* **181**(31), 17–33.

YOSHIDA, H., SUENAGA, K. & ECHIGO, R. 1990 Turbulence structure and heat transfer of a two-dimensional impinging jet with gas–solid suspensions. *Int. J. Heat Mass Transfer* **33**, 859–867.