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The measurement of instantaneous heat transfer coefficients around the circumference of a tube immersed in a high temperature fluidized bed

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Abstract—Instantaneous and time-averaged local heat transfer coefficients are obtained for an immersed heat transfer tube in a high temperature (900 K) fluidized bed. Surface temperature measurements are made by very fine ribbon thermocouples circumferentially located 120° around the surface of the immersed tube. Surface temperature measurements are converted to local instantaneous heat transfer coefficients by solving the conduction equation for the tube wall. Results from this work are compared to previous work in fluidized beds.

INTRODUCTION

HEAT TRANSFER coefficients between the vessel walls or immersed surfaces and gas fluidized beds have been measured for a wide variety of systems in the past 40 years. Most measurements have concentrated on obtaining time averaged local and overall coefficients in beds at ambient temperature and relatively low pressure. The measurement of instantaneous coefficients is much less common and all but one such study [1] has been performed at near ambient conditions. The purpose of this work was to develop a heat transfer probe capable of measuring instantaneous local heat transfer coefficients around the circumference of a tube immersed in a fluidized bed at a temperature of approximately 900 K. The information that such a probe provides would give insight into the temperature and heat flux variations that exist in industrial fluidized bed coal combustors and similar equipment and may well be useful in predicting the corrosion that occurs in boiler tube bundles due to the non-homogeneous temperature and concentration profiles which almost certainly exist in the vicinity of banks of heat exchange tube.

BACKGROUND

Previous measurements of time varying heat transfer coefficients between gas fluidized beds and surfaces in contact with them have used one of four types of heat transfer probes.

Catipovic [2] and Fitzgerald *et al.* [3] have reported instantaneous heat transfer coefficients for fluidized beds at ambient conditions using electrically heated metallic film heat sensors. These probes consisted of a thin platinum foil deposited on a glass-like substrate and are based on similar probes developed for boiling heat transfer measurements [4] and later modified for fluidized beds by Tuot and Clift [5]. By simultaneously measuring the power dissipated by the foil and the temperature of the foil these researchers were able to calculate the instantaneous heat transfer coefficients at various locations around the circumference of a tube immersed in a fluidized bed. Due to the very thin foil used in the probe, which was exposed to the bed, the erosion of the platinum was excessive and a mylar sheath had to be added to protect the platinum. This sheath increased the response time of the probe to approximately 20 ms. The response time defined here and by the authors [3] refers to the time required for the probe and electronics to restore the power dissipated by the probe to 98% of its original value prior to a step change in heat transfer coefficient. Although this probe gave excellent results for low temperature systems it was unsuitable for high temperature fluidized beds.

George and Welty [6] and Goshayeshi *et al.* [7, 8] used thermopile type gauges to determine local heat transfer coefficients in high temperature fluidized beds. These gauges consisted of a thin layer of insulating material with arrays of thermocouples or resistance temperature detectors placed on either side of the material. The gauges were calibrated by measuring the heat flux through the layer of insulating material as a function of the temperature difference across the wafer. When placed in contact with the fluidized bed the temperature across the wafer along with the outer surface temperature, in contact with the bed, were measured and from these the local time-averaged heat transfer coefficients could be calculated. Again these probes were very susceptible to erosion and a protective sheath of stainless steel shim was wrapped around the probes. The resulting response time was

NOMENCLATURE

Ar	Archimedes number, $gd_p^3(\rho_s-\rho_f)\rho_f/\mu_f^2$
d_p	particle diameter [m]
g	acceleration due to gravity [m s^{-2}]
$h(t)$	local instantaneous heat transfer coefficient at time t [W m^{-2} K^{-1}]
$\langle h\rangle$	local time-averaged heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
Nu_{max}	maximum time-averaged spatial-averaged Nusselt number, $h_{max}d_p/k_f$
$q_w(t)$	local instantaneous surface heat flux at time t [W m^{-2}]
R	radius of heat transfer tube [m]
r	radial distance [m]
T	temperature [K]
t	time [s]
u	velocity [m s^{-1}].

Greek symbols

α	thermal diffusivity of tube material [m^2 s^{-1}]
θ	angular position on tube [deg]
μ	viscosity [kg m^{-1} s^{-1}]
ρ	density [kg m^{-3}].

Subscripts

bed	referring to fluidized bed
c	cooling water
f	fluid
i	at inner radius of tube
j	jth data point taken at t_j
max	maximum value
mf	at minimum fluidizing conditions
min	minimum value
n	value at the end of the nth time interval
o	at outer radius of tube
s	solid.

increased to approximately 1 s and thus only time averaged information on the heat transfer signal could be measured.

The third type of heat probe used was a circular foil heat flux gauge due to Gardon [9, 10], modified for high temperature fluidized beds by Gosmeyer [11]. The principle of operation of this gauge is that of a differential thermocouple formed at the surface in contact with the fluidized bed. The differential thermocouple is formed by a thin copper tube with constantan foil soldered to the top of the tube with a copper wire soldered to the middle of the constantan foil. The tube is filled with a ceramic material and the whole assembly placed in contact with the fluidized bed. The temperature difference between the center of the foil and the circumference of the copper tube and the radial heat flux can be measured and from these the heat flux normal to the foil can be calculated. Although this type of gauge has been very successful in the field of high flux radiant heat transfer, the non-uniform temperature distribution across the foil due to the constantly changing solid movement renders this type of gauge unsuitable for use in fluidized beds and the instantaneous heat transfer coefficients obtained by Gosmeyer [11] were unreliable.

The fourth probe used for instantaneous heat flux measurements in fluidized beds is one based on a very low response time temperature measuring element placed on the surface in contact with the fluidized bed. Such a probe was used by George [1] to successfully measure time varying heat transfer coefficients from the vessel wall to a high temperature fluidized bed. The principle of operation of this type of probe differs from the previous gauges in that only a single time varying surface temperature measurement is taken along with a steady temperature on the outer wall surface. From such data, steady state and instantaneous heat transfer coefficients are inferred by solving the conduction equation for the wall in contact with the bed. The probe developed by George [1] uses two very fine thermocouple ribbons insulated from each other except at the surface of the vessel wall where they form a junction flush with the wall's surface. The response time of the thermocouple junction is very small, the junction is erosion/abrasion resistant and due to the small size of the junction the probe is non-intrusive. The many advantages of this type of probe made it suitable for this work and it forms the basis of the probe developed in this paper and described below.

DESCRIPTION OF HEAT TRANSFER PROBE AND EXPERIMENTAL APPARATUS

Unlike the probe developed by George [1], the current work concentrated on developing an instrument to be used on the surface of a 50.8 mm diameter 304 stainless steel tube. A detailed diagram of the probe is shown in Fig. 1. The thermocouple junction is formed at the surface of the tube by welding alumel and chromel ribbons, both 0.0635 mm thick, together. The two ribbons are held in place by two split cylinders of 304 stainless steel which are force fit into a 304 stainless steel sleeve. The thermocouple ribbons are electrically isolated from the tube and each other by sandwiching them between thin, 0.0025 mm, sheets of mica. The whole assembly comprising of thermocouple ribbons, insulating mica wafers, stainless steel half cylinders and sleeve are then force fit into the 304 stainless steel tube. Three such assemblies are located

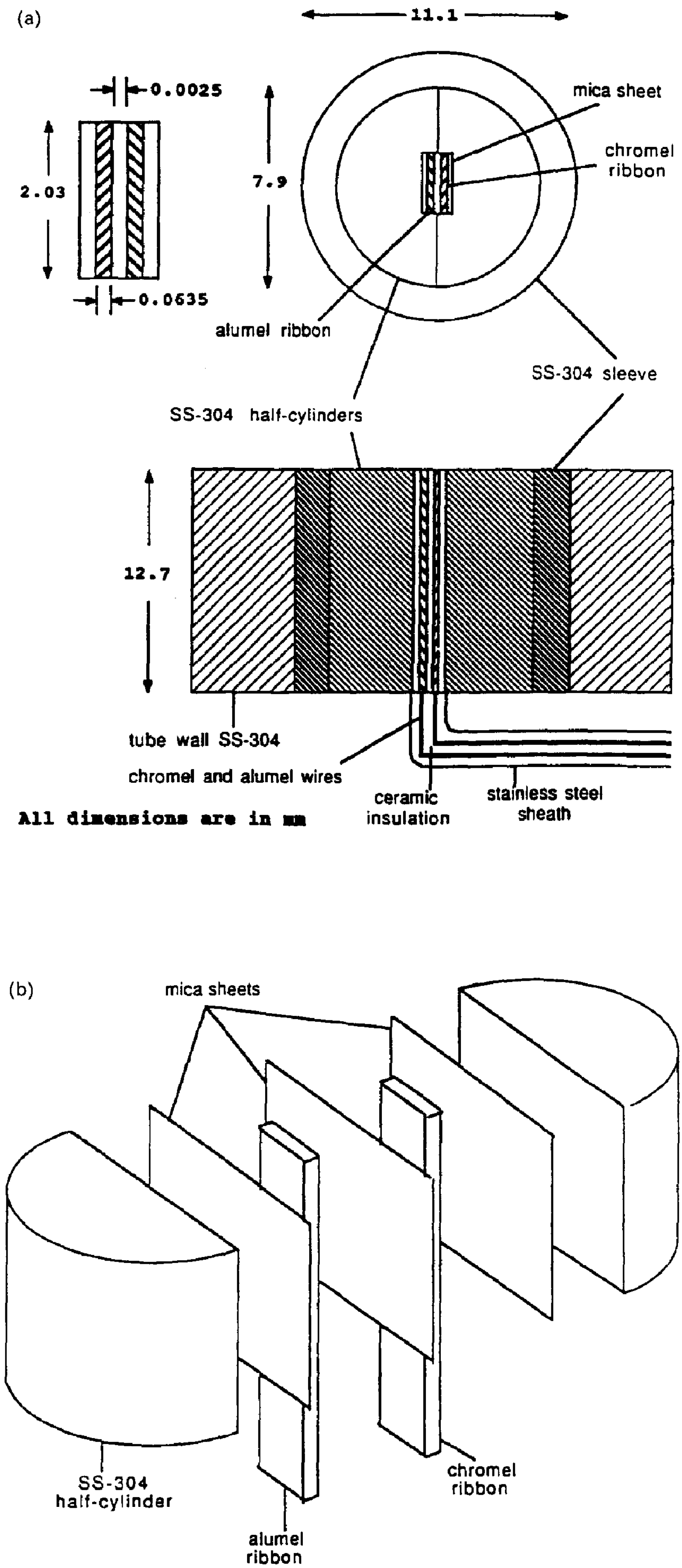

FIG. 1. (a) Detailed diagram of heat probe. (b) Exploded view of transducer.

at 120° circumferentially around the tube, as illustrated in Fig. 2.

The welding of the metal ribbons can be carried out by lightly rubbing the protruding ribbons with an abrasive paper, thus smearing the two materials together and forming a thermocouple junction at the surface of the tube. The junction so formed has the added advantage that as the tube erodes the thermocouple junction should continue to reform and should thus be reliable for long periods of time [1]. This claim, however, needs to be verified experimentally since runs of only a few hours were completed in this work, which are too short to verify the abrasion resistance of the probe.

The stainless steel tube has a 25.4 mm inside diameter through which water is continuously fed during operation of the probe. Thermocouples were placed at both ends of the tube to monitor the inlet and outlet temperatures of the water. The flow rate of water was so large that no difference in temperature between the inlet and outlet was observed. This fact, coupled with the very high inside heat transfer coefficient between the water and the tube wall, justifies the assumption used in the data analysis that the inside wall temperature is constant and equal to the cooling water temperature.

The heat transfer probes were calibrated and tested for both static and dynamic conditions. The three transducers were found to be accurate to within 0.6°C of each other at the operating temperature of the bed. The transient response of the transducers was tested by exposing them to a step change in surface temperature. The response time required for the transducers to indicate 98% of the change in temperature was found to be approximately 10 ms as recorded by the data acquisition system using a sampling frequency of 1 kHz. Subsequent experimental data were recorded at a sampling frequency of 100 Hz. Details of the calibration of the probe are given by Khan [12].

The experimental apparatus for the hot fluidized bed runs are shown in Figs. 3(a) and (b). The stainless steel tubular fluidized bed used was 203 mm in diameter and 711 mm high. Two sets of ports, 228 and 456 mm above the distributor, were installed to accommodate the probe. Two type-K thermocouples were placed 102 mm above and below the centerline of the probe to facilitate the measurement of the bed temperature. An expander section, 406 mm in diameter and 609 mm high, was placed on top of the bed. Fluidizing air was supplied via a control valve and rotameter to the plenum of the bed. Natural gas, used to heat the bed, was supplied at a point just above the distributor plate. An oxyacetylene torch, placed at the top of the bed, served to ensure that no unburnt gas accumulated in the vicinity, thus eliminating any explosion risk. Data were acquired from the probe via a Keithley 570 A-D system connected to a Zenith 248 PC.

From Fig. 3 it can be seen that the single heat transfer tube with the transducers occupies approximately 30% of the area of the fluidized bed. The presence of the tube will, to a certain extent, affect the local hydrodynamics and hence the heat transfer coefficients obtained in this research may differ from those in larger equipment in which edge effects are less important. Little work has been done in quantifying the effects of equipment size on the measurement of heat transfer coefficients in fluidized beds and thus no attempt has been made to estimate the magnitude of this effect.

THEORY AND ANALYSIS OF DATA

To infer heat fluxes and heat transfer coefficients from time varying surface temperature measurements it is necessary to solve the unsteady conduction equation within the tube.

In order to simplify the analysis it was assumed

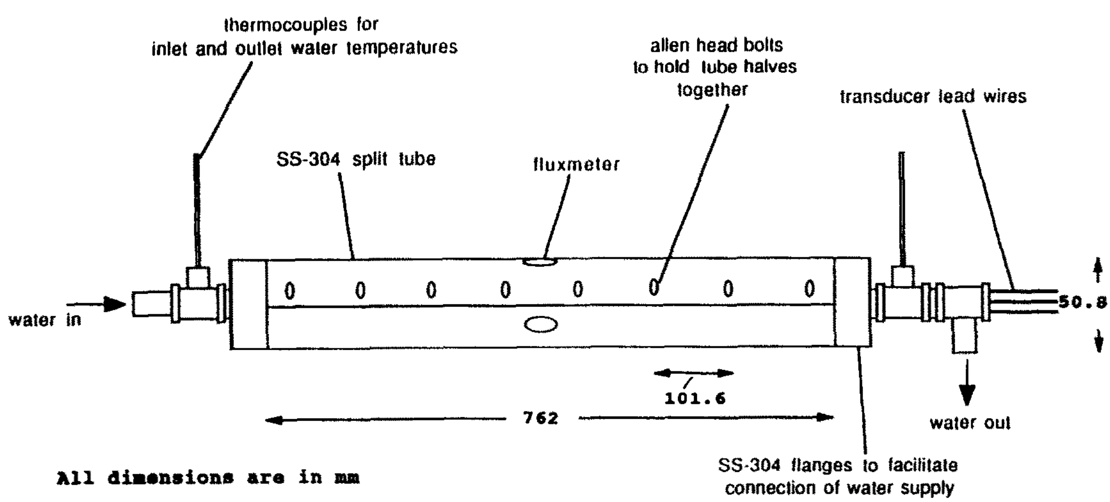

FIG. 2. Diagram of instrumented heat transfer tube.

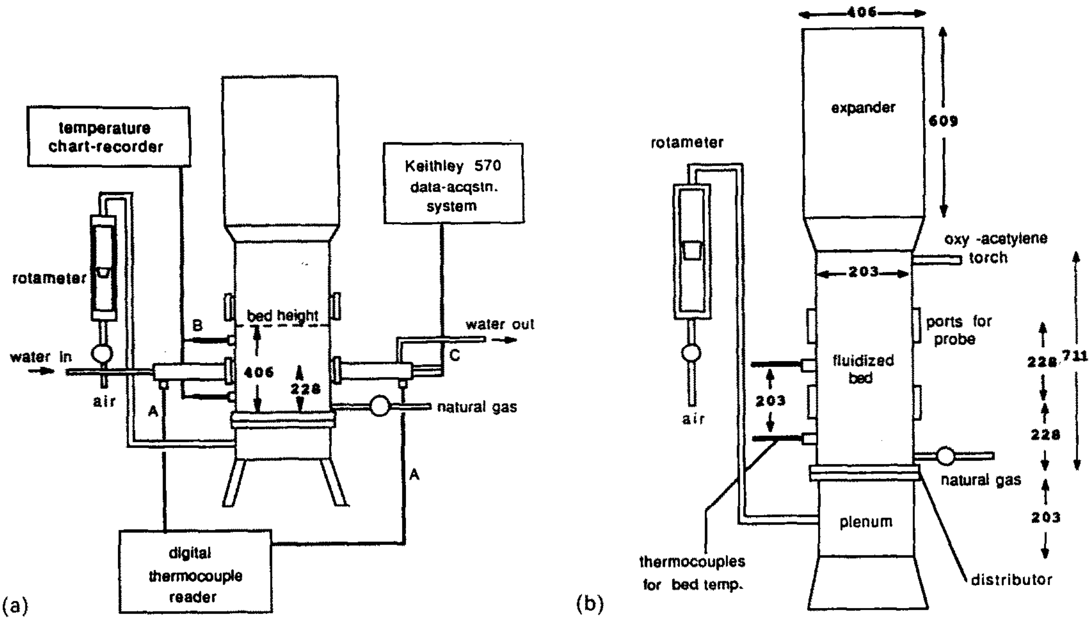

FIG. 3. (a) Diagram of fluidized bed. (b) Experimental set-up showing location of tube and other instrumentation.

that heat conduction in the axial and circumferential directions could be ignored when compared to the radial flux and that the inside tube wall temperature could be assumed to be constant and equal to that of the cooling water. With these simplifications the governing equations for conduction within the tube wall are as follows:

$$\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}=\frac{1}{\alpha}\frac{\partial T}{\partial t} \tag{1}$$

with the following boundary conditions:

$$T(R_o, t) = T_w(t)$$

$$T(R_i, t) = T_c = \text{cooling water temperature (constant).}$$

For the initial condition it is assumed that the temperature distribution is the steady state distribution which would occur using the time average of the wall temperature (T_w). The effect of using such an initial condition is to initially distort the time varying temperature profile in the tube. However, this effect is damped out relatively quickly such that after 10 s the profile is independent of the initial condition. In all the data analyses the results of the first 40 s of simulation were ignored, thus eliminating any biases caused by the initial condition.

Equation (1) was integrated numerically with the boundary and initial conditions shown above, using a Crank–Nicholson method, to yield the temperature profile in the tube wall. In order to obtain the heat flux at the surface of the tube the temperature profile was differentiated. Both two-point and three-point numerical differentiations were used with almost identical results (due to the narrow grid spacing used in the integration method). The instantaneous and time-averaged bed–wall heat transfer coefficients were then calculated as follows:

$$h_n = h(t_n) = \frac{q_w(t_n)}{T_{bed} - T_w(t_n)} \tag{2}$$

$$\langle h \rangle = \frac{\sum_{j=1}^{n} h_j}{n}. \tag{3}$$

Details of the calculation procedure and numerical algorithms are given by Khan [12].

The resolution of the data acquisition system allowed temperature measurements to be made within ∓ 0.6°C. The accuracy of the heat transfer coefficients obtained by solving equation (1) depends on the accuracy of the surface temperature measurements, the thermal properties of the tube material and the validity of using a one-dimensional unsteady state conduction model. The material properties are well known for this material (304 stainless steel) and vary less than 5% over the temperature range experienced by the tube wall. The effect of inaccuracies in the measurement of the surface temperature were minimized by smoothing the raw surface temperature–time data prior to using it as the boundary condition for equation (1). These inaccuracies were further investigated in a separate study [12], where a set of data was altered by ∓ 0.6°C at random and then re-entered into the computer program. The instantaneous heat transfer coefficients for the measured and 'tampered' surface temperature data were compared and seen to be accurate within 3% of each other.

Finally, the effects of radial and axial temperature gradients are small and estimated not to effect these results by more than 5%.

EXPERIMENTAL RESULTS

Wall surface temperatures for the heat transfer tube were obtained for two materials, sand ($d_p = 0.730$ mm) and limestone ($d_p = 1.215$ mm), at four and two fluidizing gas velocities, respectively, and at a bed temperature of 900 K. From these time varying surface temperatures instantaneous local heat transfer coefficients were calculated.

Instantaneous heat transfer coefficients

The local instantaneous heat transfer coefficients at different angular positions around the tube for sand and limestone at two superficial velocities are shown in Figs. 4 and 5 and Table 1. No similar data exist for the conditions used in this work. For comparison purposes, however, the results of Catipovic [2] giving instantaneous heat transfer coefficients around the circumference of a tube immersed in an ambient fluidized bed will be made. The magnitude of fluctuations in heat transfer coefficients shown in Figs. 4 and 5 increase with increasing superficial velocity. The degree of the increase in magnitude of these fluctuations varies with circumferential position. For the case of sand (Fig. 5), it can be seen that the magnitude of fluctuations increases dramatically with an increase in gas velocity at $\theta = 150°$ and $180°$. Similar behavior may be seen at $\theta = 0$ for limestone in Fig. 4. This behavior has also been noted by Catipovic [2], who attributes this phenomenon to the presence of stagnation points at the lower and upper points on the tube. At low gas velocities, there is a formation of a cap of relatively stagnant particles, the 'Lee Stack', at the top of the tube. Consequently, little fluctuation is seen in the local instantaneous heat transfer coefficient at this point. As the superficial gas velocity increases, however, the 'Lee Stack' breaks up, thereby allowing more interaction between the emulsion phase and the tube wall. This results in an increase in the magnitude of fluctuations of the local instantaneous heat transfer coefficient at the top of the tube. Catipovic [2] attributes the behavior at the bottom of the tube to a lower stagnation point caused by a void or air cushion at this point. At low gas velocities only a few particles can penetrate this void and come into contact with the tube wall. As the gas velocity is increased, however, more and more particles are able to penetrate the void, causing an increase in the magnitude of fluctuations of the local instantaneous heat transfer coefficient.

Table 1. Maximum, minimum and time-averaged instantaneous heat transfer coefficients for sand and limestone fluidized beds (data from Figs. 4 and 5)

Angular position	h_{max} ($W\,m^{-2}\,K^{-1}$)	h_{min} ($W\,m^{-2}\,K^{-1}$)	h_{ave} ($W\,m^{-2}\,K^{-1}$)
Limestone, d_p = 1.22 mm, u_o/u_{mf} = 1.20			
0	271	120	165
30	368	134	213
60	339	108	207
90	500	106	237
120	386	111	220
150	204	109	149
180	116	52	84
Limestone, d_p = 1.22 mm, u_o/u_{mf} = 1.43			
0	555	172	268
30	420	162	268
60	410	117	242
90	592	129	276
120	395	175	389
150	224	133	182
180	143	73	100
Sand, d_p = 0.73 mm, u_o/u_{mf} = 1.20			
0	652	217	350
30	608	212	377
60	619	179	361
90	773	135	421
120	637	178	440
150	563	220	350
180	279	129	196
Limestone, d_p = 0.73 mm, u_o/u_{mf} = 1.60			
0	787	175	402
30	754	212	429
60	787	178	431
90	804	144	464
120	935	243	550
150	816	202	539
180	820	198	443

Catipovic [2] has reported minimum values for the local instantaneous heat transfer coefficient for a bed of sand ($d_p = 0.8$ mm and $u_o/u_{mf} = 1.6$) at ambient conditions in the range of 50–75 $W\,m^{-2}\,K^{-1}$. The values reported here for sand and limestone (Table 1) lie well above this range and this can be attributed to a combination of radiation heat transfer and higher gas thermal conductivity. These minimum values correspond to the passage of the bubbles either engulfing or moving close to the transducers. During such events the heat transfer coefficients measured can be taken as the sum of the radiative and gas convective coefficients. For the bed temperature used in this work the radiative coefficient can be estimated [13] to be approximately 35 $W\,m^{-2}\,K^{-1}$. The change in the transport properties of the gas, especially the thermal conductivity, will significantly change the gas convective component of the heat transfer coefficient. Comparing the temperature range used here with that used by Catipovic [2], it can be shown that the gas convective component is increased by a factor of about 1.8. Thus combining these two effects and using Catipovic's [2] values for the minimum gas convective components at ambient conditions, the resulting range for the minimum coefficients for a bed tem-

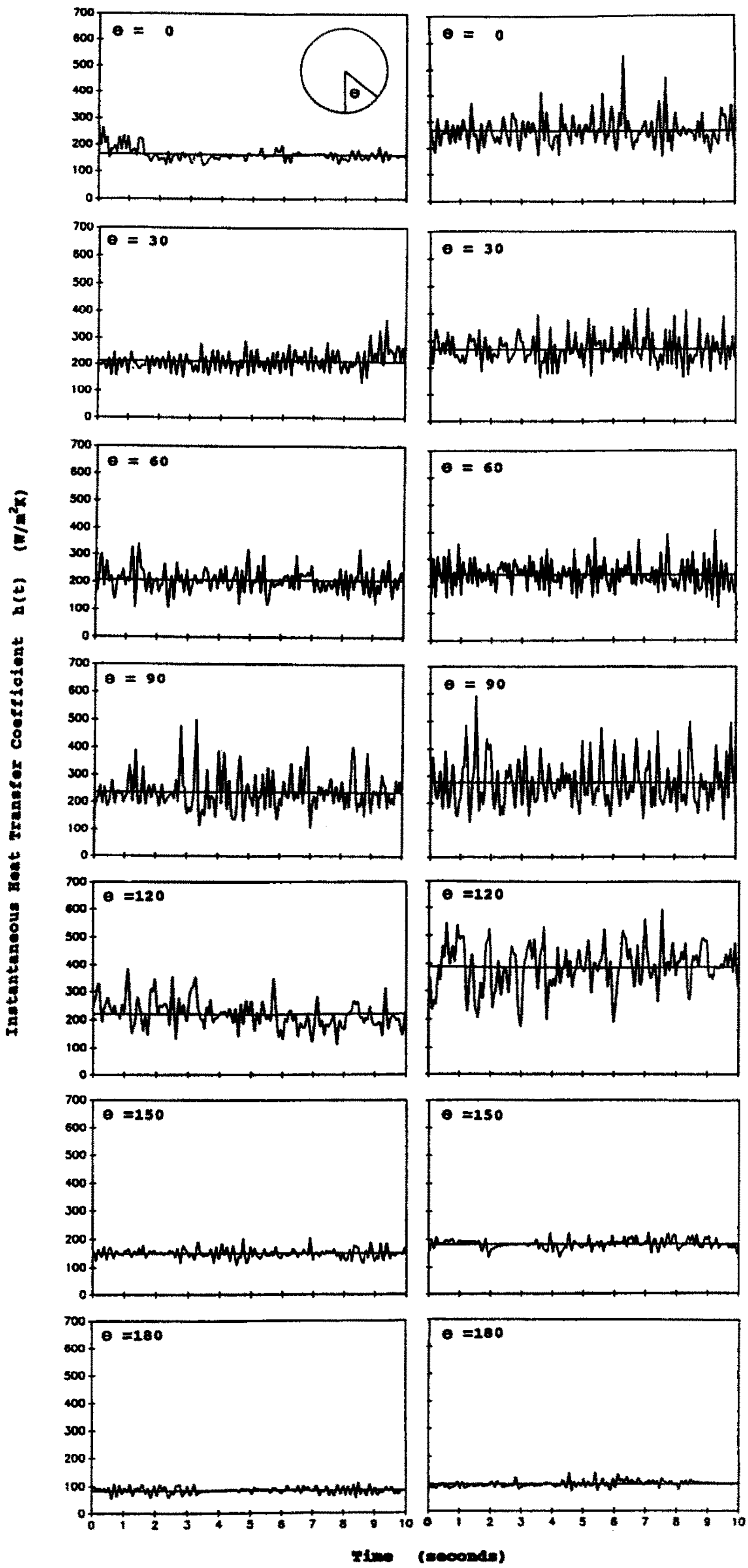

Fig. 4. Local instantaneous heat transfer coefficients vs angular position for limestone (d_p = 1.22 mm, T_{bed} = 900 K) at u_o/u_{mf} = 1.20 (left) and 1.43 (right).

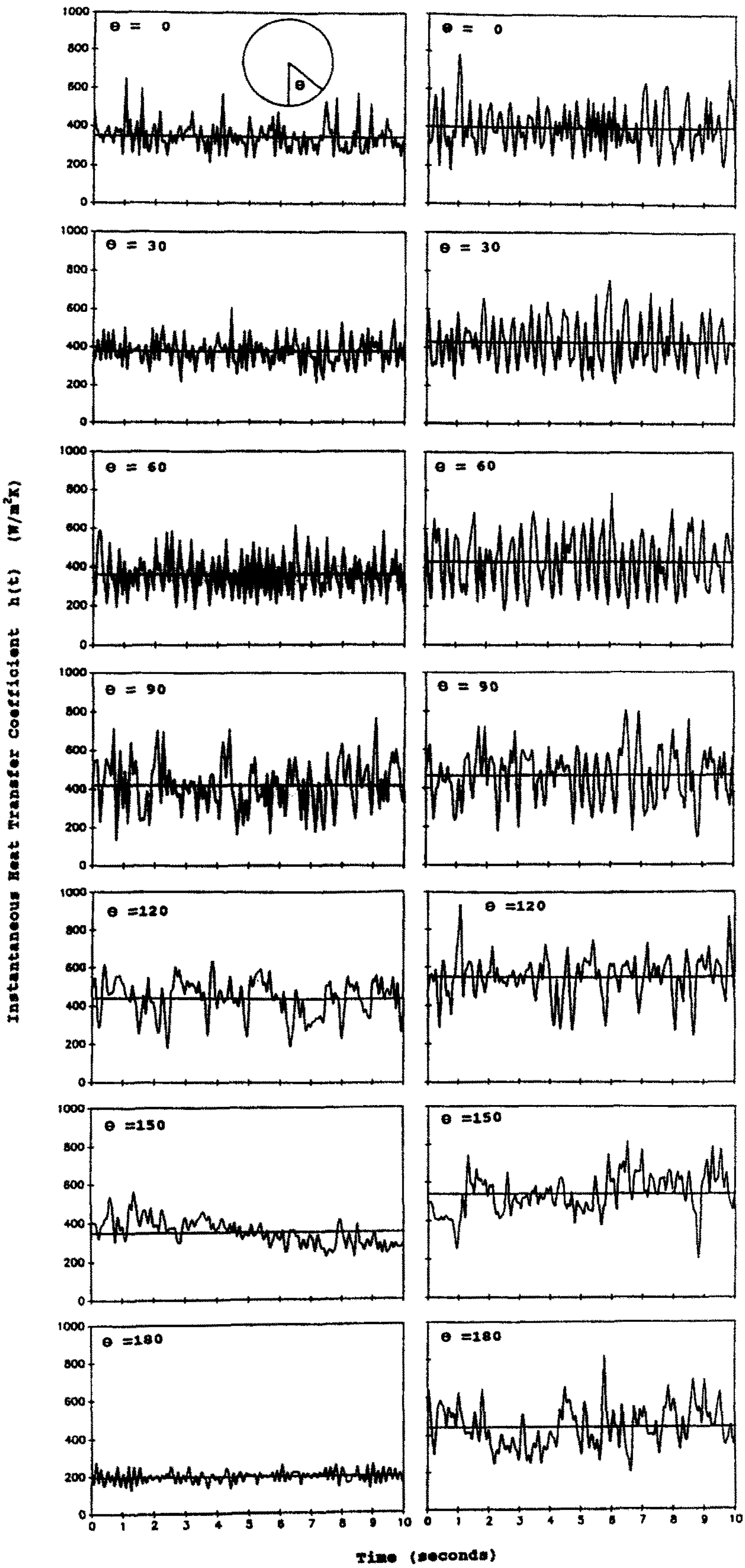

FIG. 5. Local instantaneous heat transfer coefficients vs angular position for sand ($d_p = 0.73$ mm, $T_{bed} = 900$ K) at $u_o/u_{mf} = 1.20$ (left) and 1.6 (right).

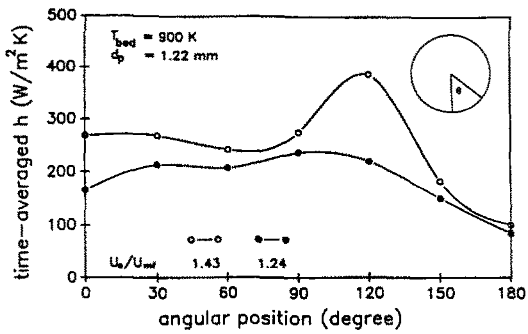

FIG. 6. Local time-averaged heat transfer coefficients vs angular position for limestone.

perature of 900 K is 125–170 W m^{-2} K^{-1}. This range is consistent with most of the data from this study.

The maximum instantaneous heat transfer coefficients from this work (935 W m^{-2} K^{-1} for sand and 592 W m^{-2} K^{-1} for limestone) are also much greater than those measured by Catipovic [2] for ambient beds with similar size particles (320 and 275 W m^{-2} K^{-1}). No explanation for this increase in the maximum fluctuations recorded can be found at this time. However, it is interesting to note that the only other data collected for instantaneous heat transfer coefficients in high temperature fluidized beds were by George [1] and he found maximum instantaneous values of 1285 W m^{-2} K^{-1}, albeit for much smaller particles ($d_p = 0.15$ mm).

Time-averaged heat transfer coefficients

Again, time-averaged local heat transfer coefficients for the same conditions employed in this work are not available for comparison. The data of George and Welty [6] for single tubes immersed in high temperature (810 and 1050 K) fluidized beds of large ($d_p = 2.14$ and 3.23 mm) particles come closest to the conditions used here and will be used to compare the trends in the data.

Figure 6 and 7 show the time-averaged local heat transfer coefficients as a function of circumferential position for the different fluidizing velocities used in the current work. The shapes of the curves in these figures are very similar to those reported by George

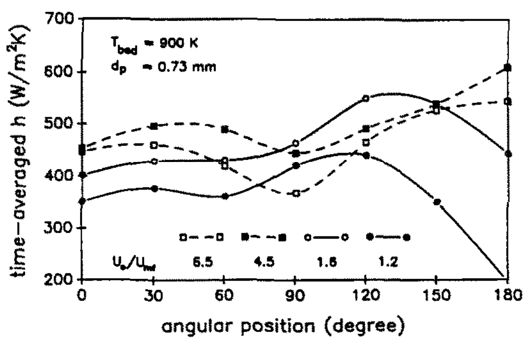

FIG. 7. Local time-averaged heat transfer coefficients vs angular position for sand.

and Welty [6]. Specifically, for low levels of fluidization, the local heat transfer coefficients at the top and bottom of the tubes are lower than those measured on the sides of the tube with the minimum values occurring at the top of the tube. The maximum local values are seen to occur at an angle of approximately 120° from the bottom of the tube. At much higher fluidization levels this trend is seen to change as shown in Fig. 7, where the maximum values shifts to the top of the tube and the minimum shifts to approximately the 90° location. This last trend is consistent with the findings of Noack [14] for a horizontal tube immersed in a low temperature bed.

The values of the local heat transfer coefficients reported here are significantly higher than those reported by George and Welty [6]; however, this is consistent with the fact that smaller particles were used in the current work.

As a final comparison, the results of the current work for the maximum time-averaged, spatially-averaged heat transfer coefficients are shown in Fig. 8. The results are plotted as Nusselt number vs Archimedes number. Also shown in this figure are the results from George and Welty [6] and Catipovic [2] and the correlation of Zabrodsky *et al.* [15]:

$$Nu_{max} = 0.88Ar^{0.213}.$$

The data from this work appear to agree well with the data from similar previous studies of heat transfer in fluidized beds and with the correlation of Zabrodsky *et al.* [15].

SUMMARY OF RESULTS

It is seen that on increasing the superficial gas velocity there is an increase in the magnitude of fluctuation of the local instantaneous heat transfer coefficient. This effect is most pronounced at the upper and lower stagnation points on the tube. The maximum value of the instantaneous heat transfer coefficient at any given point on the tube is seen to increase with an increase in the gas velocity. No uniform trend can be seen in the variation of the minimum local instantaneous heat transfer coefficient with increase in the gas velocity.

The local time-averaged heat transfer coefficient is seen to increase with an increase in the superficial gas velocity, the most pronounced effect being seen at the stagnation points. A maximum is seen in the distribution of time-averaged heat transfer coefficients in the top half of the tube. As the superficial gas velocity is increased, this maximum is seen to shift from $\theta = 120°$ to 180°.

The time-averaged and instantaneous heat transfer coefficients are seen to increase with a decrease in the particle size.

CONCLUSIONS

A cylindrical probe capable of measuring local instantaneous heat transfer coefficients in high tem-

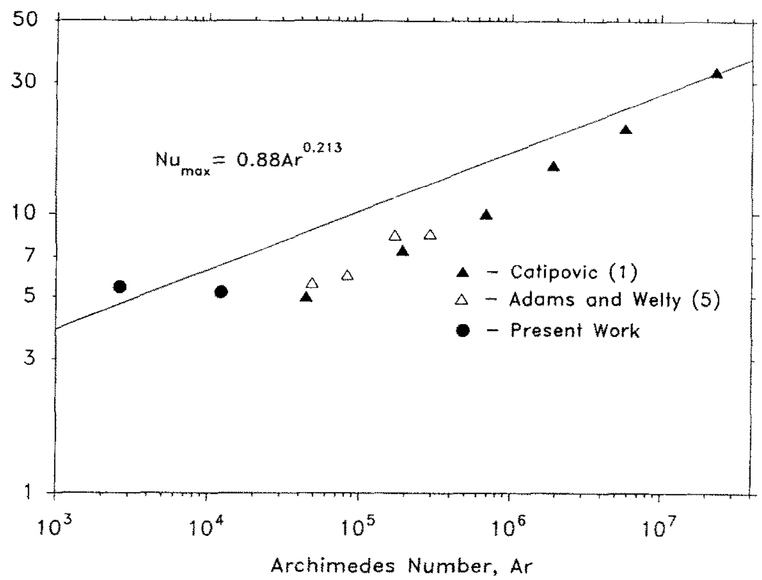

FIG. 8. Comparison of maximum time-averaged Nusselt numbers vs Archimedes number. Results of this study and previous work are compared to the correlation of Zabrodsky *et al.* [15].

perature fluidized beds has been successfully built and tested. Both instantaneous and time-averaged local heat transfer coefficients were obtained in the present work and these compare favorably with data obtained previously in low temperature systems.

The heat transfer probe gives reproducible results and is believed to be extremely abrasion resistant, although this has to be confirmed by long term runs in the future.

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