Basic Studies on Fluidized Bed Heat Exchanger*

(Heat Transfer from Single Row of Horizontal Rectangular Tubes Immersed in Floating Low Density Particles)

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Using the low density particles (polystyrene hollow spheres) described in the 1st report, mean heat transfer coefficients from a single row of horizontal cylindrical tubes immersed in floating particles were measured, with the aim of reducing pressure drop in a fluidized bed heat exchanger. In this study, furthermore, in order to make particles float stably around tubes, rectangular tubes (aspect ratio : 30/3) were used because the change in flow area between the tubes is small; that is, the change in flow velocity between the tubes is small compared with that of cylindrical tubes. The results show that : ① $L_0/L_h \geq 1$, and the mean heat transfer coefficients are independent of both L_0 and L_h , in the same way as with cylindrical tubes. ② Mean heat transfer coefficients decrease slightly more than those with cylindrical tubes. ③ However, since rectangular tubes can be arranged more densely than cylindrical tubes, their thermal performance under the same pumping power is improved to about three times that of cylindrical tubes.

Key Words: Heat Exchanger, Fluidized Bed, Convective Heat Transfer, Low Density Particle, Single Row of Horizontal Rectangular Tubes

1. Introduction

Intensive studies designed to realize a fluidized bed heat exchanger with excellent heat transfer characteristics have been conducted in recent years^{(1)~(3)}. One of the important subjects in this research is to lessen the pressure loss due to the insertion of particles. Concentrating on this point, the authors experimentally clarified, in a previous report⁽⁴⁾, the flow pattern characteristic and mean heat transfer characteristic of a single row of tubes placed horizontally in a floating fluidized bed, using particles of extremely low density (polystyrene hollow spheres) instead of the conventional bubbling fluidized bed. As a result, as far as the volume goodness factor with respect to pressure drop is concerned, it was determined that the best performance is obtained with the

narrowest tube pitch ($P=13$) in the experimental range. This seems to mean that performance is enhanced by an increased density of tube arrangement because, if the tube pitch is narrower, the degree of freedom of floating of particles is reduced and the heat transfer coefficient is slightly lowered, but the pressure drop due to the particles is predominant and the increase in pressure drop due to the narrowness of the tube pitch does not become apparent. Therefore, if the particles float stably around the heat transfer tubes and the density of the tube arrangement is increased at the same time, a further improvement in performance by an increase in the heat transfer coefficient is expected.

From this viewpoint, this study was designed to improve the heat transfer characteristic by increasing the density of tube arrangement and by using the low-density particles mentioned in the previous report, also by using rectangular tubes having a relatively large cross-sectional aspect ratio, and also by expecting that the particles will float stably around the tubes owing to the smallness of the changes in the cross-sectional area of flow passage between the tubes, because these tubes differ from circular tubes in that

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there are fewer change in local velocity around them.

Conventionally, in order to improve heat transfer characteristics, studies into finned tubes⁽⁵⁾ and attempts to roughen the tube surfaces⁽⁶⁾ have been made, but nothing has been reported, so far as the authors have determined from the literature, about the use of non-circular tubes for heat transfer in a fluidized bed heat exchanger. However, in the field of flow patterns making use of the gravitational downward flow of similar particles in fluidized beds, the local heat transfer characteristics of circular tubes, elliptical tubes and lenticular tubes, albeit single tubes, have been studied by Kurochkin⁽⁷⁾. According to this report, there were no practical differences between circular and elliptical tubes, and the lenticular tube exhibited the maximum heat transfer coefficient. This is because, in the case of circular and elliptical tubes, the appearance of motionless solid particles or air pockets occur on the upstream and downstream side of each tube and the thermal resistance increases, whereas, with lenticular tubes, solid particles move over the entire surface of the tube. His report concerns conditions which differ from those of the present experiment, but this result suggests the validity of this paper.

2. Nomenclature

- A : Cross-sectional area of flow passage (m^2)
 A_h : Heat transfer surface area per tube (m^2)
 Ar : Archimedes number
 C : Clearance of tubes (mm)
 D : Tube wall thickness (mm)
 d_p : Mean particle size (mm)
 E_{std} : Standard pumping power ($\text{N/m} \cdot \text{s}$)
 H : Tube height (mm)
 h : Mean heat transfer coefficient ($\text{W/m}^2 \cdot \text{K}$)
 h_{std} : Standard heat transfer coefficient ($\text{W/m}^2 \cdot \text{K}$)
 L_o : Height of static bed (mm)

Fig. 1 Experimental apparatus

- L_h : Distance from distributor to lower side of heat transfer tube (mm)
 L_f : Height of floating particles (mm)
 N : Number of heat transfer tubes
 Nu : Nusselt number
 P : Tube pitch (mm)
 p : Pressure in fluidized bed (Pa)
 Δp : Pressure drop in fluidized bed (Pa)
 Δp_t : Total pressure drop; sum of pressure drops in distributor and heat transfer tube (Pa)
 Re : Reynolds number
 T_w : Heat transfer tube surface temperature (K)
 T_{in} : Temperature in plenum chamber (K)
 T_{out} : Temperature 200 mm downstream of heat transfer tubes (K)
 ΔT : Temperature difference (K)
 u : Superficial gas velocity in fluidization conditions (m/s)
 u_{mf} : Superficial gas velocity at minimum fluidization (m/s)
 Y : Distance from distributor in downstream direction (mm)
 W : Total weight of solid particles (g)
 ε_{mf} : Void fraction at minimum fluidization
 ρ : Density of solid particles (kg/m^3)

Subscript

- c : Single-phase forced convective heat transfer coefficient

3. Experimental Apparatus and Method

The experimental apparatus is the same as that used in the experiment of the previous report⁽⁴⁾ and so it is not described here (See Fig. 1). The distributor is also the same as that used previously, that is, it is a commercial water filter (10-mm thick) which is sandwiched between 20- and 50-gauge mesh. The coordinate system and principal symbols used in this experiment are shown in Fig. 2. Figure 3 is a schematic

Fig. 2 Principal symbols and coordinate system

diagram of a rectangular tube used for measuring the mean heat transfer coefficient, in which a rectangular copper tube 3 mm in thickness, 30 mm in height and with an aspect ratio of 10 (corresponding to a circular tube of 20.2 mm in diameter, after the circumferential length is adjusted) has both ends formed into a semicircle 3 mm in diameter. Built-in nichrome wire was directly energized by a stabilized DC source, and the experiment was conducted at a uniform wall temperature. The heat loss in the axial direction was calculated from the temperature gradient measured by a thermocouple built into a bakelite plate (1). In the range of this experiment, the heat loss was within 1% of the current input. The temperature difference ΔT used in the definition of the mean heat transfer coefficient was similar to that given in the previous report, that is, the results of measurements of temperature distribution with respect to wall temperature of the heat transfer tube show that $\Delta T = T_w - (T_{in} + T_{out})/2$ at $u/u_{mf} < 1$ and $\Delta T = T_w - T_{out}$ at $u/u_{mf} \geq 1$. The tube pitch was varied from 6 mm to 23 mm and the number of heat transfer tubes was varied from 7 to 11.

The specifications of the polystyrene hollow spheres used in the experiment are given in Table 1. Particles of within ± 0.1 mm were used, selected by a gauge mesh, the same as in the previous report. In this table, the value of u_{mf} was determined from the turning point in the previous report's correlation diagram (corresponding to Figs. 5 and 6 in this experiment) of Δp with respect to the superficial gas velocity

Table 1 Specifications of Particles

Sample	S1	S3
Particles	Polystyrene Hollow Spheres	
Shape	Spherical	
d_p (mm)	1.0 (± 0.1)	1.5 (± 0.1)
ρ_p (kg/m ³)	742	322
u_{mf} (m/s)	0.29	0.26
ϵ_{mf}	0.47	0.36
Ar (10^4)	2.17	3.58

Fig. 3 Heat transfer tube

ity in the fluidization condition. Prior to the heat transfer experiments with the fluidized bed, experiments to determine the forced convective heat transfer coefficient of a single phase with both a single tube and bundle of tubes were conducted, the results of which being shown in Fig. 4. The curve is summarized by the Nusselt number and Reynolds number, using 1/2 of the circumferential length as the equivalent length and the superficial gas velocity in the fluidization condition as the equivalent velocity. At the flow velocity in this experimental range, the Nu of a single tube is, as shown in the diagram, summarized by $Re^{0.5}$ in the same way as in the result reported by Pohlhausen⁽⁸⁾ on a flat plate, but the absolute value is about 14% higher. In the case of a bundle of tubes, as the tube pitch becomes narrower, the heat transfer coefficient increases and the exponent of Re seems to decrease.

4. Experimental Results and Discussion

4.1 Pressure drop

Figure 5 shows examples of changes in the net pressure drop due to the Lo of particles, obtained by simply subtracting the pressure drops of the distributor and tubes ($S3, P=23, Lh=50$). In the same way as with the bundle of circular tubes quoted in the previous report, Δp increased together with the flow velocity and became constant at above u_{mf} . In addition, together with the increase of Lo (corresponding to amount of particles), Δp inevitably increased, and these curves agreed very well with the theoretical value (W/A) determined from the balance of forces indicated by the solid line in the diagram. Therefore, this is considered to indicate the uniform fluidized condition. Figure 6 shows the effects on Δp imposed by the particles, in which, as in the case of a circular

Fig. 4 Single-phase forced convective heat transfer of bundle of rectangular tubes

Fig. 5 Pressure drop of fluidized bed (effect of L_o)

Fig. 6 Pressure drop of fluidized bed (effect of particle size)

tube⁽⁴⁾, a difference was noted in the stage at which u_{mf} was reached, but there was no difference from the theoretical value after fluidization if W was matched. If the tube pitch was varied, the magnitude of the pressure drop was determined by the total mass of particles after fluidization ($u \geq u_{mf}$).

Figures 7 (a) to (c) show changes in the distribution of pressure p in the fluidized bed in the Y direction along the column. Specifically, Fig. 7 (a) relates to changes in the Y direction distribution of p due to the flow velocity ($S3, L_o=84, P=23, L_h=50$).

(a) Changes due to u (b) Changes due to P (c) Changes due to L_o

Fig. 7 Pressure distribution in fluidized bed

At $u=0.275$ m/s, since the particles are almost motionless, the changes in p are almost linear from the position of $Y=L_o$ (84 mm) to the area immediately above the distributor. In the so-called bubbling state when some of the particles have become fluidized ($u=0.493$ to 0.664 m/s), the gradient of p is small and there is no significant change. As u becomes larger, the gradient becomes much smaller, and the range of variation of p with respect to L_f increases in the Y direction. Finally, midway between tubes, the gradient is minimized; but it becomes large at the positions directly above the tubes. When the balance

(a) Effects of height of static bed on mean heat transfer coefficient

(b) Effects of particle size on mean heat transfer coefficient

of forces is considered, the gradient of p in the Y direction corresponds to the density of floating particles (void fraction). As mentioned in the previous report, it is clear that the heat transfer coefficient cannot be immediately determined from just the density of floating particles, but there is a very close relation between them, as shown later (Section 4.2). Figure 7 (b) shows $P=8$ and 23 as examples of the effect of tube pitch on the Y direction distribution of p ($S1, Lh=50, u \approx 1.82, L_o=65$ ($W/A = \text{constant}$)). At $P=23$, where the intertubular speed is low, the gradient of p is larger in the vicinity of the tubes than at $P=8$, and the density of floating particles is larger, as is clear from this diagram. Figure 7 (c) shows the effect of L_o on the p distribution at $u \approx 1.06$ m/s ($S1, P=23, Lh=50$). It is clear that the maximum value of Δp varies with respect to the total mass of particles. However, at $Lo/Lh \geq 1$, it is known that a similar p distribution is displayed in the area below the tubes.

4.2 Mean heat transfer coefficients

Changes in mean heat transfer coefficient with respect to u are shown in Figs. 8 (a) to (c). Specifically, Fig. 8 (a) shows examples of the effect of L_o ($S3, P=23, Lh=50$), in which as in the case of circular tubes⁽⁴⁾, when L_o is small, no increase can be expected from the single-phase forced convective heat transfer coefficient until the flow velocity at which floating particles can rise to the vicinity of the heat transfer tube is reached. However, as L_o increases, the flow velocity at which an increase above the single-phase forced convective heat transfer coefficient appears becomes gradually smaller; and at $Lo/Lh \geq 1$, the heat transfer coefficient is nearly uniform with respect to the flow velocity, regardless of

(c) Effects of tube pitch on mean heat transfer coefficient

Fig. 8

Lo . On the other hand, as the flow velocity increases, h shows a slight peak in the vicinity of $u = u_{mf}$, regardless of Lo , and thereafter it has an almost constant value.

Figure 8 (b) shows examples of the effect of particles ($Lo \approx 85$ mm constant, $Lo/Lh \geq 1$, $P=23$). In the same way as with circular tubes, the effect of particle size is not noted in the low-speed region, but, in the high-speed region, h is higher with S1 in which particle size is smaller. Incidentally, the flow velocity at which h is a maximum seems to be slightly smaller than $2u_{mf}$ with circular tubes.

Figure 8 (c) shows examples of the effect of tube pitch (S3, $Lh=50$, $Lo/Lh > 1$), in which there is almost no effect due to P at $u < 2u_{mf}$. However, at $u \geq 2u_{mf}$, the larger the value of P , the greater the degree of freedom of the floating particles, so that h becomes higher. However, as the tube pitch becomes narrower to $P=6$, channeling occurs in the range of $u=0.3$ to 0.8 m/s after minimum fluidization, and the heat transfer coefficients of individual tubes fluctuate significantly, although these fluctuations differ in degree, and the heat transfer coefficient is considerably reduced with certain tubes.

In the diagram, data with slash (/) marks refers to the average of the heat transfer coefficients of all tubes when channeling occurs. Such channeling occurs when the gas between tubes are narrow, the height H of tubes is sufficiently large compared with the tube gap C , Lo/Lh is near 1, and a "slender channel" (channel of air flow) is formed after the particles are elevated by local bubbling immediately after fluidization so as to make the bed height spatially non-uniform. In this example, not only is the flowing matter made non-uniform by the generation of channeling, but also a special geometrical flow passage is formed to encourage this phenomenon. However, when Lo/Lh is small or when the flow passage is greater, the imbalance of the bed height in the incipient fluidization is canceled as the flow velocity increases, so that channeling does not occur. This channeling phenomenon also takes place with S1 in which the particle size is small (see Fig. 10). However, there is almost no difference in the generated flow velocity range or degree in the resultant drop of h , within the range of particle sizes in the experiment, and the greatest drop is recorded at a flow velocity of slightly lower than $2u_{mf}$. This fact suggests that channeling may also be likely to occur immediately before the effective floating of particles in the entire tube as the floating bed of particles begins to rise above the tubes. Therefore, in order to avoid this channeling and maintain a sufficient heat-transfer effect, it is necessary to reduce the height H of the tubes and arrange the tubes in

multiple rows, or modify the structure of the distributor as reported in the study by Aihara et al.⁽⁹⁾, by a slit type distributor, which will be discussed in a subsequent report. Incidentally, this phenomenon was not observed in the case of circular tubes (tube diameter 10 mm) in the previous report.

Figure 9 shows the maximum value of h obtained in the vicinity of $u = 2u_{mf}$ at $Lo/Lh \geq 1$, with respect to the tube gap C . In the diagram, the results obtained with circular tubes are also plotted for reference, and CC and RC respectively stand for circular and rectangular tubes. At all tube gaps, h was higher at S1 where the particle size was smaller, regardless of the tube sectional shape. Furthermore, regardless of the particle size, as the tube pitch becomes narrower, h drops, and the single tube gives the highest value. Noting the relationship between h and the tube's sectional shape, h is generally high with a circular tube, regardless of the particle size, but with S1 when the tube pitch is narrower, the heat transfer coefficient of rectangular tubes is slightly higher than that of circular tubes. However, as mentioned above, in the case of rectangular tubes, since channeling occurs at $P=6$, the heat transfer coefficient of the entire bundle of tubes is considerably reduced. With rectangular tubes, the maximum value in the flow velocity range of this experiment only occurs at $P=6$. Therefore, the reason why the value for S1 is significantly lower than that for S3 is because the change with flow velocity of h is affected by the particle size or tube pitch, not by the channeling.

Figure 10 shows an example of the effect of tube pitch on rate of increase h/h_c with respect to the single-phase forced convective heat transfer coefficient (S1, $Lh=50$, $Lo/Lh \geq 1$). As predicted, the larger the tube pitch, the greater the rate of increase, the factor reaching as high as 6.5, if the factor of 9.0

Fig. 9 Changes of maximum value of mean heat transfer coefficient due to tube gap

of the single tube is excluded. This significant difference in factor between the single tube and bundle of tubes is due to the fact that the value of the single-phase forced convective heat transfer coefficient of the bundle of tubes is larger than that of the single tube in the case of empty columns, although this difference is not so clear in the case of a fluidized bed (Fig. 9). On the other hand, with the same tube gap, the rate of increase with rectangular tubes is slightly higher than that of the single tube in the previous report. Changes in h/h_c with respect to the tube pitch at $u < 2u_{mf}$ are not clear.

Next, in order to evaluate the heat transfer coefficient characteristic by taking into account the total pressure drop, including that due to the distributor, the relation between the standard heat transfer coefficient h_{std} and the "volume goodness factor"

used in the evaluation of the results with the circular tube of the previous report⁽⁴⁾, that is, the standard pumping power E_{std} , is discussed below. E_{std} is assumed to be a work factor per unit flow passage area, as follows:

$$E_{std} = u \Delta p_t \quad (1)$$

By contrast, h_{std} is defined as:

$$h_{std} = Nh(Ah/A) \quad (2)$$

Figure 11 shows examples ($Lo/Lh \geq 1$, $P=23$) of varying Lh and Lo while keeping the ratio Lo/Lh constant, and the effects of particle size. Referring first to the results obtained for $Lo/Lh=1.47$ and 1.68 using sample S1, as shown in Fig. 8 (a); at $Lo/Lh \geq 1$, since h shows a similar distribution with respect to u , regardless of Lo , the modes of changes of h_{std} with respect to E_{std} are nearly the same as those with the circular tube of the previous report. After it reaches a maximum value, the performance is represented by visually one curve, although there are slight variations, and the maximum value is obtained at a small value of E_{std} because the pressure drop is smaller when Lo is smaller. Furthermore, comparing the result with S1 at $Lo/Lh=1.68$ with that with S3 at $Lo/Lh=1.72$, the performance is clearly better with S1 in which the particle size is smaller. However, in the region with smaller values of E_{std} , H_{std} is larger with S3 than with S1. This is because the particle density is lower in S3 than in S1. This agrees with the tendency of the results for circular tubes ($P=20$) shown in the diagram. Anyway, with the particles used in the experiment, performance was better with rectangular tubes. Incidentally, the results obtained with the circular tubes of the previous report were compared with the results of the single-phase forced convective heat transfer coefficient, but, since the pressure drop is extremely small with rectangular tubes, E_{std} in the range established for a fluidized bed

Fig. 10 Rate of increase with respect to single-phase forced convective heat transfer

Fig. 11 Evaluation of thermal performance at different particle sizes

Fig. 12 Evaluation of thermal performance at different tube pitches

is not obtained by single-phase forced convective heat transfer, so it is not directly compared in this report.

Figure 12 shows examples for the effect of tube pitch on performance ($S1, Lh=50, Lo/Lh \geq 1$), in which the performance is also compared with that of circular tubes. On the whole, the performance is better with rectangular tubes than circular tubes. In the range of this experiment, at the narrowest tube pitch ($P=6$), channeling occurs in the vicinity of $2u_{mf}$ and h is lowered, but it is noticeable that the highest value of h_{std} is achieved. In particular, when compared at the same standard pumping power, it is interesting that h_{std} of rectangular tubes shows a thermal performance which is improved by about 3 times over that of circular tubes with the same tube gap. This is because the tube arrangement is denser with rectangular tubes than with circular tubes with the same flow passage sectional area, but the increase in pressure drop is not so significant. However, rectangular tubes raise the problem that channeling occurs at a narrower tube pitch, significantly decreasing the heat transfer coefficient, and thus increasing the fluctuations in the heat transfer coefficient in individual tubes. Further studies into this phenomenon are necessary.

5. Conclusions

Following the previous report, the mean heat transfer characteristic of a bundle of horizontal rectangular tubes in a single row with a large tube sectional aspect ratio (30/3) were studied, using two low-density particle samples, by systematically varying the height of the static bed, the height of the heat transfer tubes, and the tube pitch. At the same time, the thermal performance was evaluated, taking into

account pressure drop, and the results were compared with those of circular tubes. The following conclusions were obtained:

(1) The pressure drop of particles in the fluidized state is constant, regardless of the flow velocity, and is determined almost entirely by the total mass of particles, rather than by the shape of the heat transfer tubes. Therefore, if the density of particles is lower, the pressure drop can be decreased. This holds true with circular tubes.

(2) The mean heat transfer coefficient increases continuously with increasing u , and, after showing a slight peak in the vicinity of $2u_{mf}$, the curve becomes nearly flat.

(3) If $Lo/Lh \geq 1$, the change of mean heat transfer coefficient with respect to u does not depend on the height of the static bed Lo , in the same way as with circular tubes.

(4) As the tube pitch becomes narrower, the density of floating particles around the heat transfer tubes is lowered, so that the heat transfer coefficient drops, but, in contrast, since the pressure drop does not increase much, the tube arrangement becomes denser and the performance is improved. However, if the tube pitch is too narrow, channeling occurs in a specific range of flow velocities and the values for individual tubes fluctuate, so that the heat transfer coefficient is lowered overall. Therefore, it is suggested that the performance may be further improved if proper consideration is given to the shape of the heat transfer tubes and distributor.

(5) When compared with circular tubes, the mean heat transfer coefficient of rectangular tubes is lower, if only very slightly. However, as the tube gap becomes narrower, performance tends to become better with rectangular tubes.

(6) In contrast, when compared with circular tubes, the performance of rectangular tubes is significantly improved. For example, if the tube gap is the same, the value of h_{std} with respect to the same standard pumping power provided by a rectangular pump is about 3 times that of circular tubes. This is because the pressure drop due to the dense arrangement of tubes does not increase in practice, so that the drop in mean heat transfer coefficient may be compensated by the increased density of tubes.

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