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An experimental investigation into the effect of fins on heat transfer in circulating fluidized beds

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Abstract—An experimental investigation into the effect of projected fins on bed to wall heat transfer in circulating fluidized beds at room temperature is reported. Experiments are conducted in two different test rigs at two locations, one at Indian Institute of Technology, Kharagpur, India and the other at Technical University of Nova Scotia, Halifax, Canada, under different operating conditions. Experimental conditions include three sizes of sand (130, 280 and 310 μm), suspension densities in the range of 8–80 kg m^{-3} and velocities of 5–11 m s^{-1}. The overall heat transfer rate is found to increase in all cases within the above range of operating and design conditions due to the use of fins.

1. INTRODUCTION

Fuel flexibility and environmental compatability have spurred the development of circulating fluidized bed (CFB) technology for steam generation. To maintain the combustion temperature at an optimum level it is necessary for the walls of a CFB furnace to absorb a certain fraction of the heat input to the furnace. While the heat input is proportional to the bed cross-section, the heat absorption is proportional to the perimeter of the furnace (height remaining constant). Thus, with the increase in the capacity of the boiler, the heat input and, hence, the required amount of heat absorption increases; but the wall area available for heat extraction does not increase in the same proportion. So large CFB boilers are required to have either additional heating surfaces across the furnace or external heat exchangers. Both options are costly and increase the risk of tube erosion.

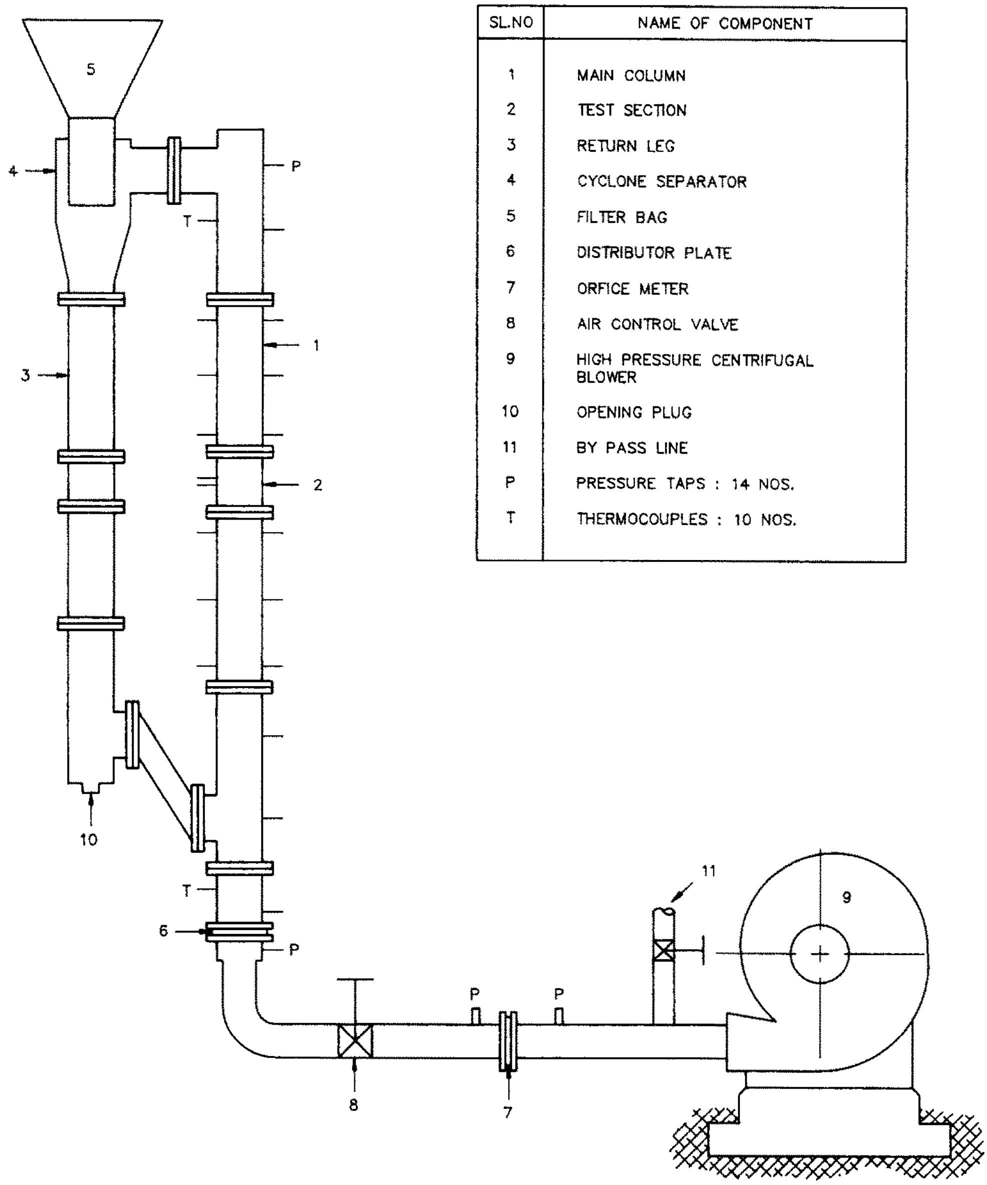
Finned tubes are widely used in heat exchangers including the economizers of boilers to enhance the total heat transfer rate. In modern boilers, panel wall tubes are welded to each other by flat fins. These fins are only partially effective, because only one side of the fin is available for heat transfer. The other side faces the insulated casing of the boiler. The heat absorption by each wall tube may be greatly increased if additional heating surfaces can be provided by welding to each tube vertical fins projecting into the furnace. The present exploratory work examines if the overall heat transfer can be increased by using such projected fins on the walls of a circulating fluidized bed. If so, by what amount. Experiments were carried out at room temperature, but covering other operating conditions relevant to commercial boilers such that at least the knowledge of the convective component of heat transfer in actual boilers can be gained through this work. In CFB boilers radiation may play an important role. However, as a first step in this exploration we examined the heat transfer in the absence of significant radiation. As far as known, no data on projected fins in circulating fluidized beds have been reported in the published literature.

2. EXPERIMENTS

The experimental work was carried out at two different laboratories. One unit was at the Steam Laboratory of the Indian Institute of Technology (IIT), Kharagpur, India (referred to as Column I). The other unit was the Department of Mechanical Engineering of the Technical University of Nova Scotia (TUNS), Halifax, Canada (referred to as Column II).

2.1. *The unit at IIT (Column I)*

The circulating fluidized bed test rig comprises of a 5150 mm tall and 100 mm diameter column with a return leg, a cyclone and a bag filter (Fig. 1(a)). Air is supplied by a high pressure blower. The air-flow rate is controlled by a valve and a by-pass arrangement. The air distributor plate used is of straight hole orifice type having 12.4% opening. Solids are returned to the main column at a height 500 mm above the distributor. Static pressures were measured at 500 mm intervals along the height of the column. Fine wire mesh (BS 400) and cigarette filters were used at the end of pressure taps to dampen pressure fluctuations in the water filled manometers. The 300 mm long test section was located 2750 mm above the distributor (Fig. 2(a)). A 1.25 kW tape heater was wrapped uniformly around it. It was then adequately insulated with glass wool and insulating rope. The temperature of the external surface was measured and found to be close to that of the surroundings thus minimizing the radiation and natural convection loss. To minimize any heat loss through conduction along the pipe wall, high temperature insulating gaskets of 10 mm thick-



LEGEND

SL.NO	NAME OF COMPONENT
1	MAIN COLUMN
2	TEST SECTION
3	RETURN LEG
4	CYCLONE SEPARATOR
5	FILTER BAG
6	DISTRIBUTOR PLATE
7	ORFICE METER
8	AIR CONTROL VALVE
9	HIGH PRESSURE CENTRIFUGAL BLOWER
10	OPENING PLUG
11	BY PASS LINE
P	PRESSURE TAPS : 14 NOS.
T	THERMOCOUPLES : 10 NOS.

Fig. 1(a). Schematic of Column I at Indian Institute of Technology.

ness were used at the flanges and guard tape heaters were provided above and below the test section. Electrical energy input to the heater was controlled by an autotransformer and measured with a voltmeter and an ammeter. Temperatures of the inner wall and the bed were measured with copper–constantan thermocouples. The thermocouples were connected to a digital multimeter through a multipoint switch.

Experiments on the test section were first conducted without fins. Then each of those experiments was repeated successively with two and four vertical rectangular fins (246 mm $\times$ 23 mm $\times$ 6 mm) under identical operating conditions. Fins were fitted inside the wall diametrically opposite to each other ensuring a near perfect thermal contact of the fin with its base. It was made of the same carbon steel as the tube having thermal conductivity of 67 W m^{-1} K^{-1}.

Sand of mean diameter 310 μm was used as the bed material in Column I. It was recycled to the main column through a cyclone. Six superficial air velocities ranging from 5.6 to 11.4 m s^{-1} were used. Three constant energy fluxes (3.58, 5.52 and 7.88 kW m^{-2}) were employed for each air velocity. Experiments were performed for three bed inventories of 20, 26 and 32 kg of sand. There is no separate means of control of circulation rate of solids. It was governed by the fluidizing velocity and the bed inventory. Bed temperatures varied from 71 to 91°C.

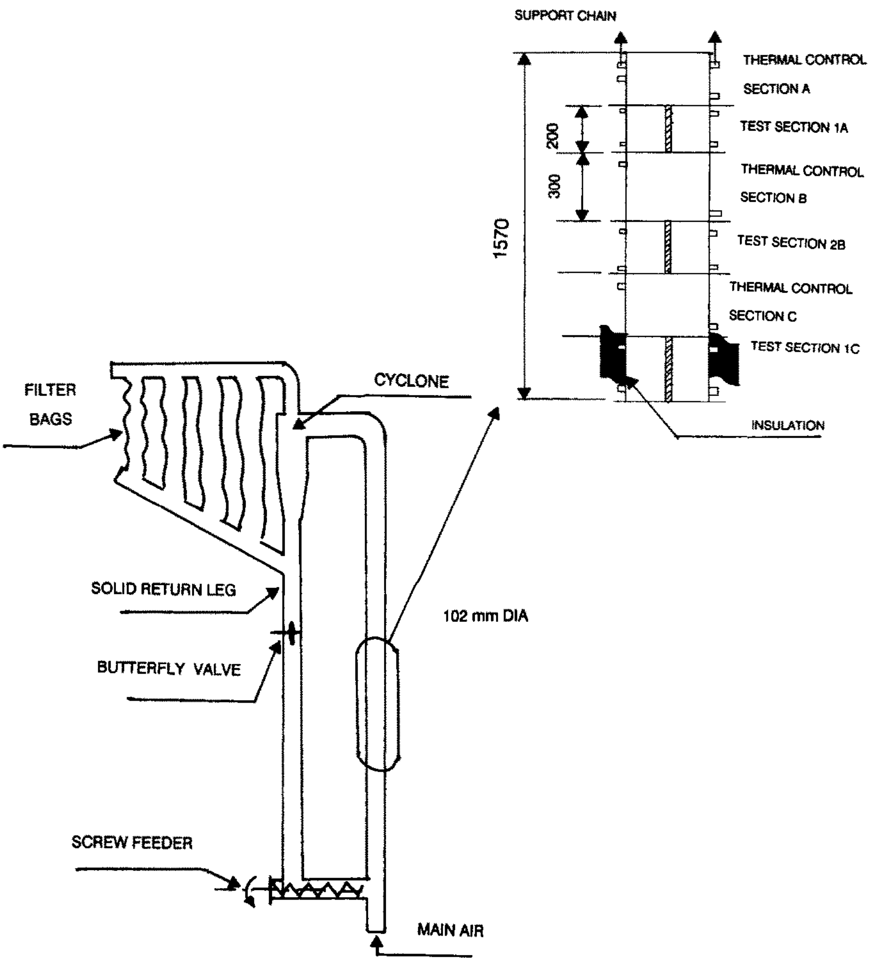


FIG. 1(b). Schematic of Column II at Technical University of Nova Scotia.

2.2. *Unit at TUNS (Column II)*

The circulating fluidized bed test rig, used at the Technical University of Nova Scotia, comprises of a 102 mm diameter, 5500 mm tall column, a cyclone and a bag filter (Fig. 1(b)). For precision control of the circulation rate of solids a screw feeder was used. Details of this test rig are given in ref. [1]. Test sections were 200 mm in length and were split in two halves. Annular water jackets were welded to both halves of the test section. Hot water from a thermostatically controlled constant head water tank was fed to each section. Three test sections were spaced 300 mm vertically apart. To reduce axial heat loss from the test sections, two 300 mm long jacketed sections cooled by the same water were placed above and below each test section. Copper–constantan thermocouples were located at the water inlet and outlet of each test section as well as within the bed. The output from each thermocouple was amplified and fed to a computer through a data interface board. The pressure drop across the test section was measured by a manometer. The suspension density was determined from the measured pressure drop.

Two sizes of sand of mean size 130 and 280 μm were used. Analyses of these sands are given elsewhere [2]. The fluidizing air, supplied by a compressor, was measured by an orifice plate.

The cross-section of a typical test section is shown in Fig. 2(b). The test section was split in two halves along its axis. An insulating gasket was provided to prevent heat exchange between two sections of the cylindrical test section. Vertical carbon steel fins (25 mm $\times$ 200 mm $\times$ 6 mm), having thermal conductivity of 67 W m^{-1} °C^{-1}, were welded to one section. This section provided data on finned tube heat transfer. Carbon steel fins of identical size were attached to the opposite half of the section with an insulating layer of silicone glue.

The fin effectiveness is defined as

$$\text{Fin effectiveness} = \frac{\text{Actual heat transfer through the fin}}{\text{Maximum heat transfer through the fin}}.$$

Maximum heat transfer will occur when all surfaces of the fin are at the same temperature as the base of the fin and the heat transfer coefficient over the entire surface of the fin will be the same as that over its base.

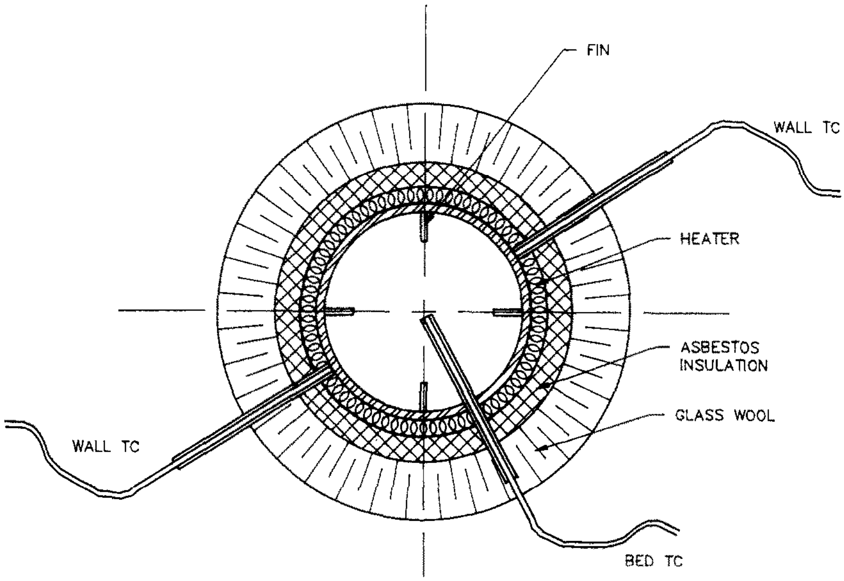


FIG. 2(a). Cross-section of the test section on Column I.

A reasonable approximation of the maximum heat transfer will be the product of the actual fin area, the heat transfer coefficient measured for the plane wall and the temperature difference between the suspension and the base wall of the fin. This was done in the experiments on Column I. A further refinement of the technique will be to measure the heat transfer coefficient on the base of the fin simultaneously with the measurement of heat transfer on the finned surface. To do this, dummy carbon steel fins of exactly the same dimensions as the conducting metallic fin are glued to the opposite half by silicone glue with insulation such that no heat can flow to the wall. The placement of such dummy fins ensures similar hydrodynamic conditions on two halves of the test section. Thus, the section with dummy fins provides the heat transfer coefficient on the base while the section with real fins gives the heat transfer on the finned surface.

The superficial velocity was changed from 3.5 to 5 m s^{-1}. The circulation rate was changed for each velocity to vary the suspension density near the test section in the range of 8–80 kg m^{-3}. The experiments on Column II were carried out at near isothermal conditions.

3. RESULTS AND DISCUSSIONS

Experimental conditions used for the results presented in this paper are given in Table 1.

3.1. *Heat transfer in the absence of fins*

To form a basis of comparison, as well as to check the reliability of the measuring techniques employed, heat transfer was measured on the plane wall of Column I and compared with the results of other workers measured on plane walls under similar conditions. It was further necessary to check if the presence of fins modified the local solid flow so radically as to affect the heat transfer to the plane wall (base) substantially. To explore this question, non-conducting fins were attached to the wall of Column II; heat transfer was measured on it. In both cases the suspension density was varied over a range of 8–80 kg m^{-3}.

Figure 3 shows that heat transfer coefficients measured on the plane wall, as well as on one with non-conducting (false) fins, increase with suspension density. It was interesting to note that these data do not differ substantially from each other. At lower density both values overlap each other while at higher density the wall with false fins shows marginally

Table 1. Experimental conditions

	Column I (IIT)	Column II (TUNS)
Fluidizing velocity [m s^{-1}]	5.6–11.4	3.5–8.0
Bed temperature [°C]	71–91	40–50
Suspension density [kg m^{-3}]	25–80	8–80
Mean particle diameter [μm]	310	130 and 280
Minimum fluidizing velocity [m s^{-1}]	0.075	0.065 and 0.025
Mode of heating	Electric heater	Hot water
Length of heated section [m]	0.9	1.87

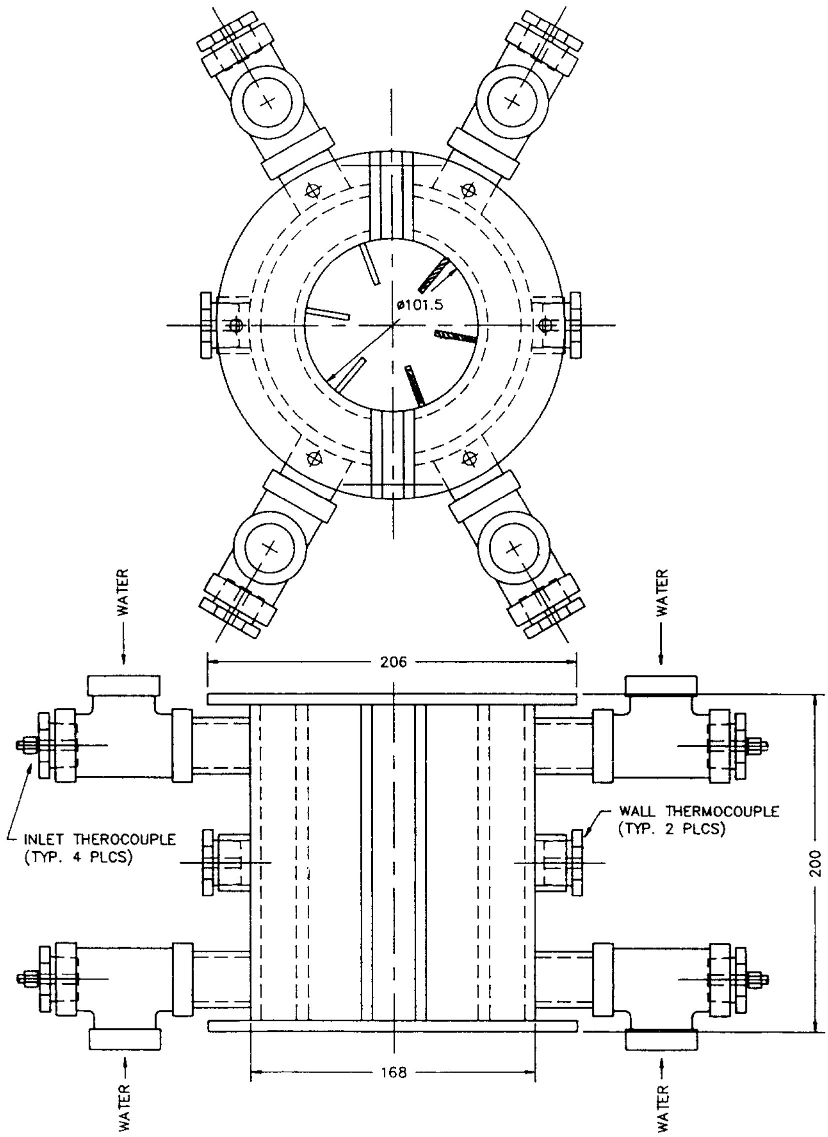


FIG. 2(b). Cross-section of the test section in Column II.

higher values of heat transfer. Inadequate data are available to substantiate this apparent difference at higher suspension densities. However, this relatively close agreement tends to preclude any drastic modification of particle dynamics by the presence of fins on the wall. As the present work is exploratory in nature, no further study on this aspect was made.

The heat transfer coefficient measured on plane walls by Basu and Nag [3], Kobro and Brereton [4] and Sekthira *et al.* [5] are also plotted on Fig. 3. The data of Wu *et al.*, shown by a straight line, is a fit of their data taken from ref. [6]. The increase in heat transfer coefficient with suspension density is observed for all data. The data of ref. [3] are higher than other values presumably due to their use of the smaller size of probe and finer particles. It has been shown [3, 7] that longer heat transfer surfaces experience lower heat transfer rates due to the cooling of solids. Also, as explained later in Section 3.3, the effect of particle size is muted for such long surfaces. This is apparent from the overlapping of data of 300 μm [5] with that of 241 μm [6], both of which used long heat transfer surfaces. Kobro and Brereton [4] used a smaller probe and therefore, report relatively high heat transfer coefficients.

Present data on plane walls fall in the same range of data as of other workers and show similar trends of variation with suspension density. This adds to the confidence on further experiments using fins on the present test rigs.

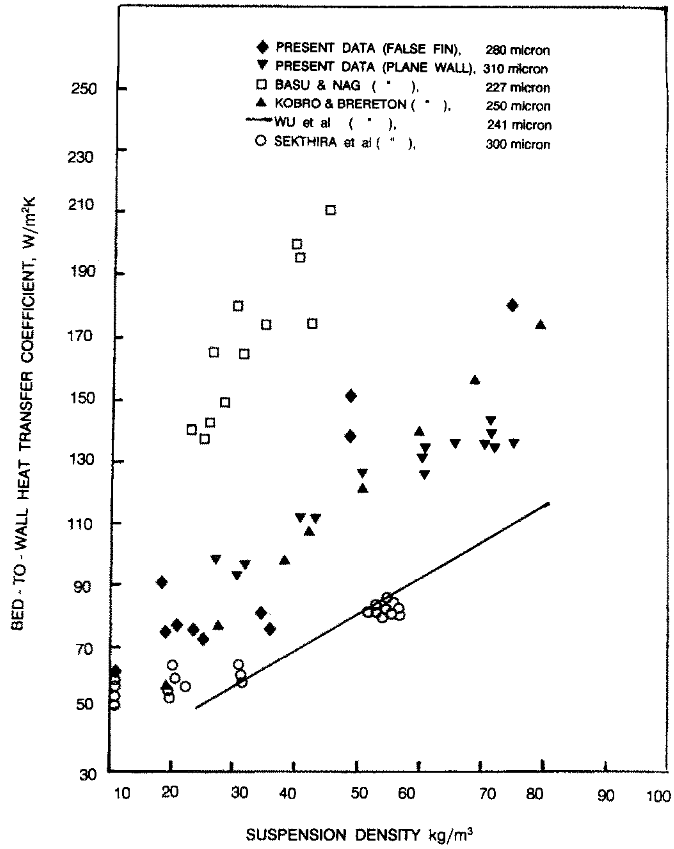


FIG. 3. Comparison of present results on plane walls with those of other workers.

3.2. *Effect of fins on the bed hydrodynamics*

The voidage profile measured along the height of Column I is shown in Fig. 4. Six profiles are shown for six superficial velocities, two rectangular fins and a bed inventory of 32 kg of sand. The bulk voidage, calculated from the static pressure drop across the vertical sections, was found to increase with superficial velocity at all heights. Although the voidage generally increased along the height of the combustor, a change in gradient of the axial voidage profile near the level of the finned section was noted at all velocities. The voidage tended to decrease near the finned section. The particles were hindered and decelerated by the presence of projected fins. This may have caused the decrease in bed voidage near the fins. Similar lowering of voidage just below a projected surface is also observed in commercial boilers. This inflection in the voidage profile is less pronounced at higher velocities due to the lower solid concentration in the column. In commercial situations such an effect on the hydrodynamics may not be present because the fins will be continuous and will run along the entire length of the wall. No sudden projection into the furnace, as in the present laboratory units, is expected in commercial boilers. However, for a rational comparison all data on the fin heat transfer coefficient are referred to the local suspension density measured across the fin section.

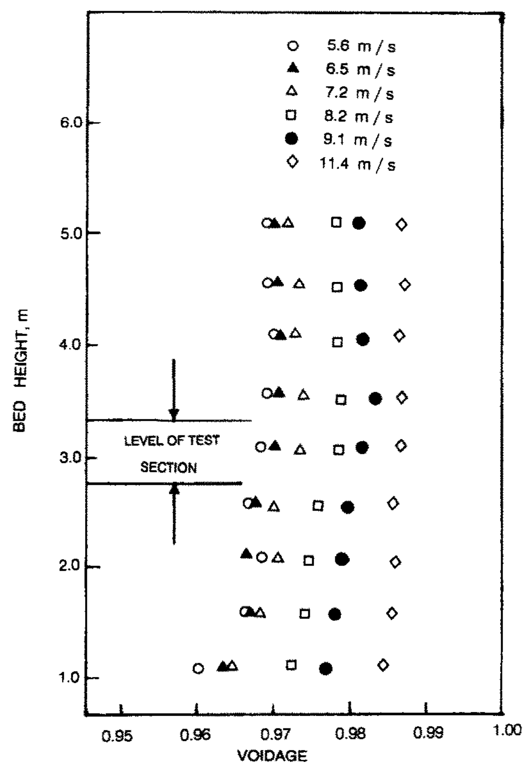


FIG. 4. Axial voidage profile measured in the presence of two fins on Column I for 310 μm sand particles.

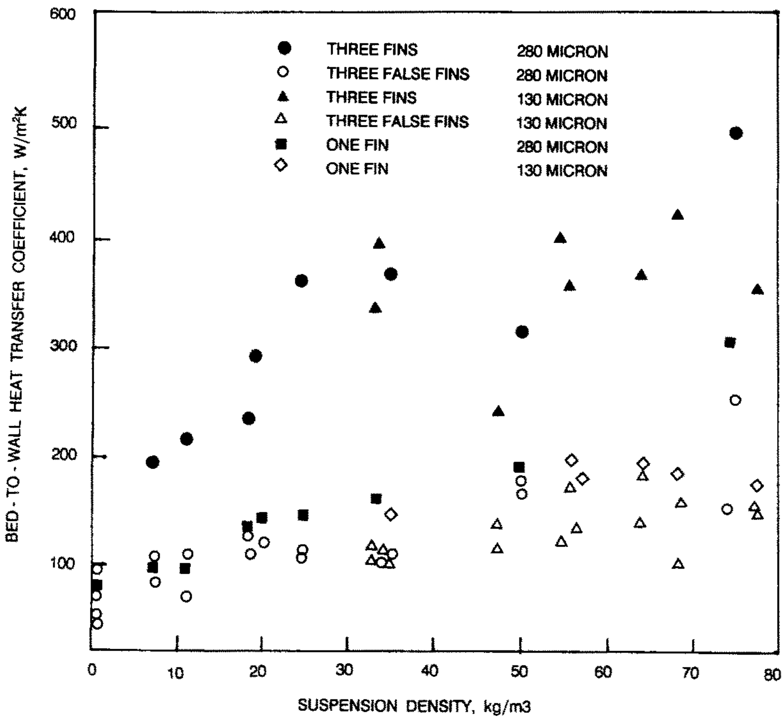


FIG. 5. Effect of fins on heat transfer coefficients on Column II.

3.3. *Effect of fins*

Heat transfer coefficients on finned surfaces are expressed on the basis of the plane area of the wall. These are then plotted on Figs. 5 and 6, against suspension density measured across the finned section. Data for two sets of finned surfaces (single and triple fins) and one non-conducting (false fin) finned surface for two particle sizes are given on Fig. 5. The heat transfer coefficient on two sets of finned surfaces (double and quadruple fins) and one non-finned plane surface are given for one particle size on Fig. 6. The heat transfer coefficient in all cases increased with suspension density in the same fashion as it did for plane walls (Fig. 3).

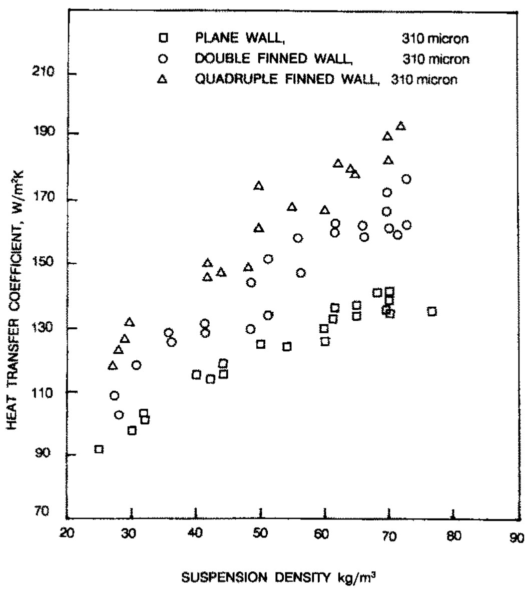


FIG. 6. Effect of heat transfer coefficients on Column I.

The heat transfer coefficient on the false fin represents the heat transferred to the base wall while it is subjected to the same hydrodynamic condition, due to the presence of fins, as the finned wall sector of the column on the opposite wall. The heat transfer coefficient calculated on the basis of base area, for all finned surfaces, is consistently higher than that for the plane wall or the false fin surface at all suspension densities. This demonstrates the enhancement of heat transfer through the use of extended surfaces of fins. The difference between heat transfer coefficients on finned and non-finned surfaces represents the gain in heat transfer due to the use of fins.

Difficulties in duplicating the hydrodynamic condition with and without fins in Column I and those with maintaining constant water flow rate in Column II resulted in some scatter in data. In spite of this scatter the enhancement of heat transfer with increasing number of fins is obvious from Fig. 6. A similar increase in heat transfer coefficients with the increase in the number of fins is also apparent in Fig. 5. The data for 130 μm particles on three as well as one fin are not very different from that for 280 μm particles (Fig. 5). These data were collected on Column II where test sections were sandwiched between two other annular jacketed sections which were also heated by the same water as that which heated the test section. This formed a continuous 1870 mm long cooled section. The thermal resistance of contact between the first layer of particles of clusters and the wall is directly proportional to the particle size [7]. If the heating surface is short, particle clusters will exchange heat with it for a very short period of time. So, the thermal resistance between the wall and the first layer of particles will govern heat transfer. Therefore, the particle size will play a dominant role on the heat transfer rate. In the case of long heating surfaces, the clusters exchange heat with the wall for long periods of time and therefore the heat conduction into the particle cluster, which is less sensitive to the particle size, dominates the process. The overlapping of data of 280 and 130 μm particles for all conducting, as well as false, fin surfaces suggests that the heat transfer to fins is governed by the same physical mechanism as on plane walls without fins.

3.4. *Fin effectiveness*

The heat transfer coefficient on the finned surfaces increases with suspension density for single, double, triple and quadruple fins as shown in Figs. 5 and 6. The greater number of fins yielded higher heat transfer. But the heat transfer does not increase exactly in proportion to the surface area added through fins. The actual gain is proportional to the additional surface area times the effectiveness of the fin. The effectiveness of the fin as defined in Section 2.2 is plotted against suspension density (Fig. 7). It is calculated for two fins and four fins on Column I and for three fins and one fin on Column II. The effectiveness is generally in the range of 85–95%. Results of one and three fins on Column II are more scattered than those for two and four fins on Column I.

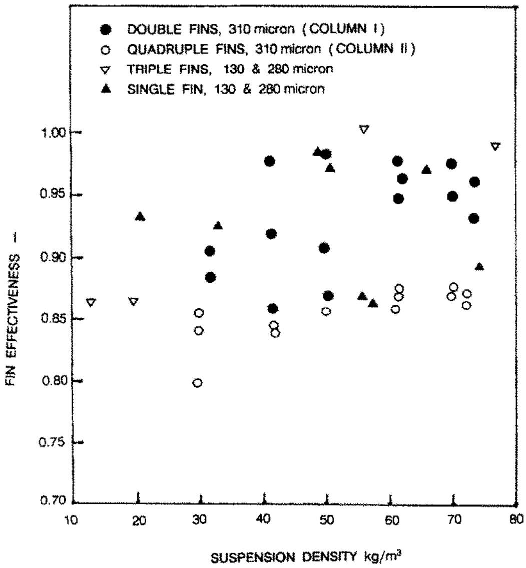


FIG. 7. Fin effectiveness as a function of suspension density.

The set of four fins on Column I clearly has lower effectiveness than that for two fins on the same column. A major difference between fins on a circulating fluidized bed and that on a conventional heat exchanger is that in the latter case the local heat transfer coefficient on the fin surface is not significantly different from that on the base, because both are exposed to similar hydrodynamic conditions. In a circulating fluidized bed a thin layer of solids frequently slides down the wall. The bulk density of the gas solid suspension is lower away from the wall. So, the fin extending away from the wall comes in contact with down-flowing solid suspension with reduced concentration. The higher the suspension density, the thicker the down-flowing layer of solids and hence, the greater portion of the fin is exposed to the higher concentration of solids. It is known [7] that the greater the solid concentration on the wall, the higher the heat transfer. So the fin should be more effective at a higher suspension density when a larger fraction of its surface is exposed to solids. But it will have a saturation limit, beyond which an increased solid concentration may not increase the heat transfer proportionately because at denser gas–solid suspension the mobility of the solid particles between adjacent fins is affected. This may be the reason why for the sets of two and four fins on Column I, the fin effectiveness initially increases with suspension density, but the increase tapers off beyond 70 kg m^{-3}. However, this trend was not traceable in the results on single and triple fins on the Column II unit where the results were too scattered for a definite conclusion on the effect of suspension density on the fin effectiveness. More data are required to elucidate this point. However, a mathematical model is being developed [8] to explain the nature of heat transfer from fins in greater detail.

3.5. *Effect of test rig*

At the time of writing, no data on fin tube heat transfer in a fast bed were available in the published literature. Although experiments were repeated and carried out for different particle sizes and number of fins on Column II, a need was felt to verify these results through another set of independent data. So, experiments were carried out on Column I which is similar in size but used slightly different design and measurement techniques.

Column I used a solid recycle system different from that of Column II. Here the fluidization velocity was varied to change the suspension density. In Column II the fluidization velocity was kept constant and the suspension density was changed through change in circulation rate. Wu et al. [6, 9] showed fluidization velocity and circulation rate have a marginal effect on the heat transfer coefficient compared to that of suspension density. So, both techniques for controlling the suspension density were valid for the present study of its effect on heat transfer. The test section on Column I was heated by electrical heaters. Since the electrical heaters provided constant heat flux and

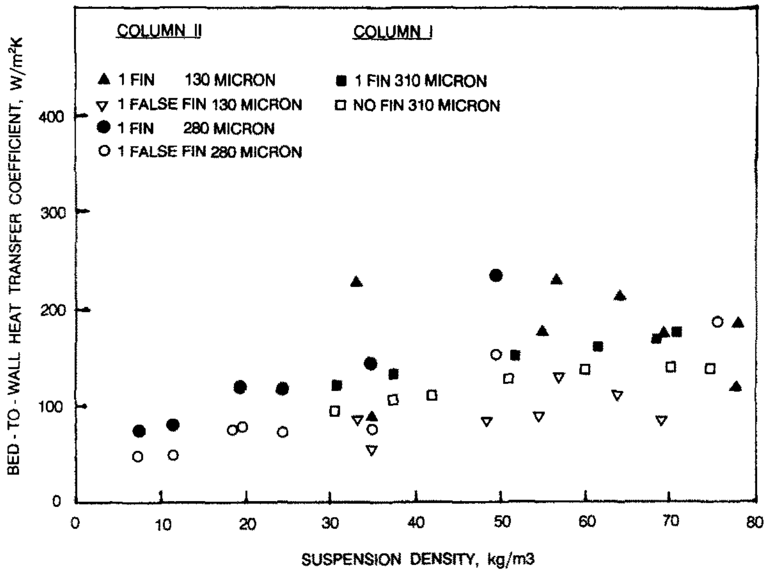


FIG. 8. Comparison of finned heat transfer data for a single fin collected on Columns I and II.

the heat transfer coefficient was determined by the measured temperature difference between the wall and the bed it may be considered to be a constant flux measurement condition. In Column II the temperature of the water heating the test section was always kept constant using a constant temperature bath. The change in heat absorption was reflected in the exit temperature of the water. Thus this condition was different from the constant flux condition of Column I.

Figure 8 compares data for single fins (one on each half of the test section) on Column II with that of Column I for 280 and 310 μm solids, respectively. It shows that over a range of suspension density of 8–80 kg m^{-3} the heat transfer coefficients are in reasonable agreement with each other. This figure also compares data of the plane wall on Column I with that of false fins on Column II. There is a close agreement between the heat transfer coefficient measured on the plane wall and that on the base of non-conducting fins both measured at the same local suspension density. This also suggests that although the presence of fins affected the flow pattern of solids in the nearby region, it did not influence the heat transfer coefficient as long as the local suspension density was the same.

The most important observation is that the heat transfer coefficient increased with installation of increasing number of fins and in both units the fin effectiveness was in the range of 85–95%. This further supports the validity of the present observations.

4. CONCLUSIONS

(1) The heat transfer coefficient improved with the installation of conducting fins on the walls of a circulating fluidized bed. The magnitude of this enhancement of heat transfer increased with the number of fins.

(2) The heat transfer coefficient on the finned surface increased with suspension density in the same fashion as on plane walls.

(3) The effectiveness of the fin surfaces (for up to four fins) was in the neighborhood of 90%.

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APPENDIX. SAMPLE CALCULATION

Unfinned surface

Heat transfer surface area

$$A_{uf} = 2\pi R L_t = 2\times 3.14\times 0.05\times 0.26 = 0.0816\ \mathrm{m}^2$$

Heat flux (measured from electrical heat input), $q'' = 5519\ \mathrm{W\ m^{-2}}$

Measured surface temperature of the wall, $T_s = 128.48°\mathrm{C}$

Bed temperature, $T_b = 77.95°\mathrm{C}$

$$\tilde{O} = T_s - T_b = 50.53°\mathrm{C}$$

Heat transfer coefficient, $h_f = q''/\tilde{O} = 5519/50.53 = 109.22\ \mathrm{W\ m^{-2}\ K^{-1}}$.

Finned surface

For two vertical rectangular fins of size 24.6 cm × 2.3 cm × 0.6 cm:

Measured $q'' = 5519\ \mathrm{W\ m^{-2}}$, $T_s = 133.65°\mathrm{C}$, $T_b = 79.36°\mathrm{C}$, $\tilde{O} = 54.29°\mathrm{C}$

Surface area of one fin, $A_f = 2\times 0.246\times 0.023 + 0.246\times 0.006 = 0.013\ \mathrm{m}^2$.

Heat transfer surface area when two fins are attached:

$$A_0 = (0.0816 + 2\times 0.013) - 2\times 0.006\times 0.246 = 0.1046\ \mathrm{m}^2$$

$$h_f = 5519/54.29 = 101.66\ \mathrm{W\ m^{-2}\ K^{-1}}$$

Fin effectiveness, $\ddot{y}_f = \tanh(ml)/ml$ (neglecting tip loss)

Thermal conductivity of fin material, $k = 67\ \mathrm{W\ m^{-1}\ K^{-1}}$

For rectangular fins, $ml = \ll 2h_f/kt \times l = (2\times 101.66/67\times 0.006)^{0.5}\times 0.023 = 0.5173$

$$\ddot{y}_f = \tanh 0.5173/0.5173 = 0.9194 \text{ or } 91.94\%.$$

ETUDE EXPERIMENTALE DE L'EFFET DES AILETTES SUR LE TRANSFERT THERMIQUE DANS DES LITS FLUIDISES CIRCULANTS

Résumé—On étude expérimentalement l'effet d'ailettes sur le transfert de chaleur entre le lit et la paroi dans des lits fluidisés circulants à des températures ambiantes. Des expériences ont été effectuées sur deux plateformes à deux endroits, l'une à l'Indian Institute of Technology, Kharagpur, Inde et l'autre a Technical University of Nova Scotia, Halifax, Canada, dans des conditions opératoires différentes. Les résultats concernent trois tailles de sable (130, 280 et 310 μm), des densités de suspension entre 8 et 80 kg m^{-3}, des vitesses entre 5 et 11 m s^{-1}. Le coefficient de transfert global est trouvé croître dans tous les cas grâce à l'utilisation des ailettes.

EINE EXPERIMENTELLE UNTERSUCHUNG ÜBER DEN EINFLUSS VON RIPPEN AUF DEN WÄRMEÜBERGANG IN EINEM ZIRKULIERENDEN WIRBELBETT

Zusammenfassung—Der Einfluß von Rippen auf den Wärmeübergang von einem zirkulierenden Wirbelbett bei Raumtemperatur an die Wand wird experimentell untersucht. Dabei werden zwei unterschiedliche Teststände an zwei Orten untersucht, einer am Indian Institute of Technology, Kharagpur/Indien, und der andere an der Technical University of Nova Scotia, Halifax/Kanada, wobei die Betriebsbedingungen unterschiedlich sind. Es wird Sand mit drei unterschiedlichen Körnungen verwendet (130; 280 und 310 μm), Suspensionsdichten im Bereich von 8 bis 80 kg m^{-3} und Geschwindigkeiten zwischen 5 und 11 m s^{-1}.

Der Gesamtwärmeübergang nimmt im gesamten untersuchten Bereich aufgrund der Berippung zu.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ВЛИЯНИЯ РЕБЕР НА ТЕПЛОПЕРЕНОС В ЦИРКУЛИРУЮЩИХ ПСЕВДООЖИЖЕННЫХ СЛОЯХ

Аннотация—Экспериментально исследуется влияние выступающих ребер на теплоперенос от циркулирующего псевдоожиженного слоя к стенке при комнатной температуре. Эксперименты проводились на двух испытательных стендах (один—в Индийском технологическом институте, Кхарагпур, Индия, второй—в Техническом университете, Халифакс, Канада) при разных режимных параметрах. Экспериментальные условия включали три размера частиц песка (130, 280 и 310 мкм), а также диапазон изменения плотности суспензии, составляющий 8–80 кг м$^{-3}$, и диапазон изменения скорости 5–11 м с$^{-1}$. Найдено что в указанных диапазонах режимных и конструктивных параметров скорость суммарного теплопереноса увеличивается во всех случаях за счет ребер.