



Heat transfer from a horizontal bundle of tubes in an air fluidized bed
by Stephen John Priebe

A thesis submitted to the Graduate Faculty in partial fulfillment of the requirements for the degree of
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Abstract:

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Results also showed that heat transfer was less in the center of a tube bundle, with a symmetrical bundle. Some indication was given that nearby heaters may have an adverse affect on the heat transfer. This was shown in asymmetrical bundles where the lowest heat transfer did not occur in the center.

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HEAT TRANSFER FROM A HORIZONTAL BUNDLE OF TUBES IN AN AIR FLUIDIZED BED

by

STEPHEN JOHN PRIEBE

A thesis submitted to the Graduate Faculty in partial
fulfillment of the requirements for the degree

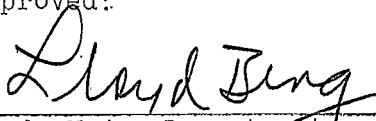
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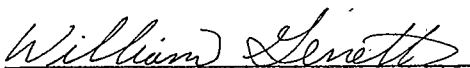
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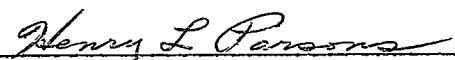
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ABSTRACT

Heat transfer from a horizontal bundle of bare tubes to a fluidized bed was measured to determine; 1) effect of number of tubes in a tube bundle, and 2) effect of heater position in a 19 tube bundle. Glass beads were used for the particle bed, and air was the fluidizing media.

Results indicated that, at higher air flowrates, the heat transfer increased with number of tubes up to about 7 tubes, and then decreased. At lower flowrates, the heat transfer increased monotonically over the range investigated.

Results also showed that heat transfer was less in the center of a tube bundle, with a symmetrical bundle. Some indication was given that nearby heaters may have an adverse affect on the heat transfer. This was shown in asymmetrical bundles where the lowest heat transfer did not occur in the center.

INTRODUCTION

Fluidization of a bed of particles implies that the particles are suspended in some fluid (gas or liquid) moving upward through the bed. The degree of fluidization depends upon the velocity of the fluidizing medium. When the velocity is such that the pressure drop across the bed is equal to the weight of the bed, fluidization occurs. This point is known as minimum fluidization. At minimum fluidization, particle circulation is limited, and the bed of particles acts like a highly viscous fluid.

As the velocity is increased, the particles begin to circulate in a regular manner, rising in the center, and falling at the sides of the column. Further increase in the velocity, causes bubbles to form. The size of the bubbles depends upon the nature of the distributor plate, and the gas velocity. This regime of flow is known as aggregative fluidization.

With increased velocity, the bubbles begin to coalesce until "slugs" of gas, occupying the entire cross-section of the column, pass through the bed. This is known as slugging. These three regimes of fluidization are sketched in Figure 1.

Fluidization has been used commercially for about thirty years. It was initiated primarily by the petroleum industry, in the fluidized catalytic cracking units. Two primary advantages over the former fixed bed reactors, are better contact and continuous regeneration of catalyst.

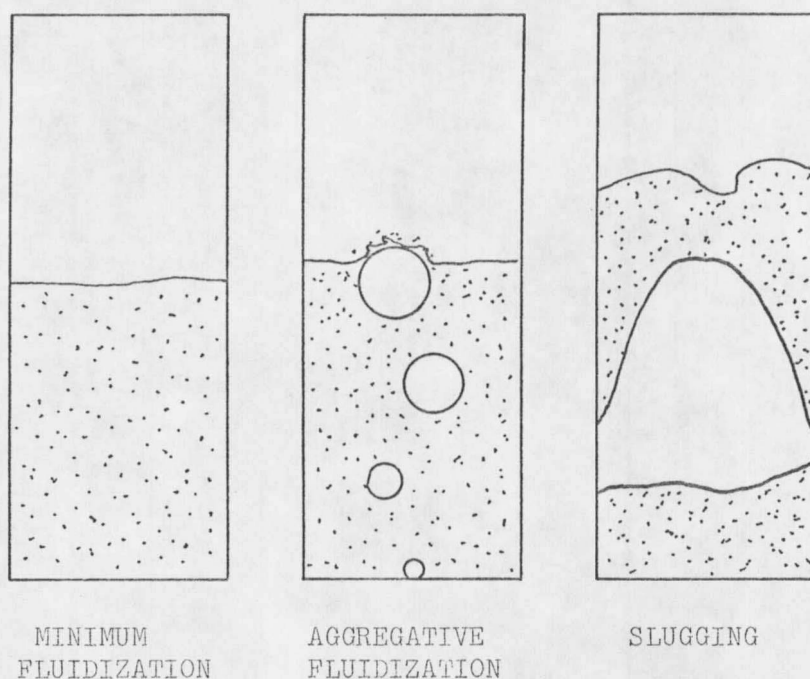


Figure 1. Regimes of Fluidization

From these beginnings, fluidization spread to many other areas such as roasting of metal ores, drying of some powders, and treating of radioactive wastes from nuclear reactors. In all of these applications, and many more, heat must be transferred to the bed.

Originally, this heat transfer was done simply by heating the walls of the column. As fluidized beds got larger, this became unsatisfactory, so people turned to internal heating. This is usually accomplished with vertical or horizontal tube bundles. The fluidized bed greatly increased the heat transfer coefficients over those obtained from forced convection alone. This is attributed to the much higher degree of turbulence

in the bed, and to the large amount of particle contact with the heat transfer surface.

Work has been done to determine the effect of various properties of the bed particles, and the fluidizing media on the heat transfer coefficients. The purpose of this study is to determine how the heat transfer coefficient in a horizontal bundle of tubes is affected by the size of the tube bundle, and by the position of a heater in the bundle.

THEORY AND RELATED STUDIES

The theory and related studies will be discussed in two parts as follows:

- 1) Mechanism of heat transfer
- 2) Effect of various parameters on heat transfer coefficients.

Mechanism of Heat Transfer.

Several models for heat transfer to a bed of fluidized particles have been proposed to account for the large heat transfer coefficients observed experimentally.

In 1949 and 1954, two similar models were proposed by Leva, Weintraub, and Brummer (5), and Levenspiel and Walton (6), respectively. This model proposed that the boundary layer on the surface of the heater is eroded away by the action of the particles moving past the surface. This reduction in film thickness would decrease the resistance, and so increase the heat transfer coefficient.

Another model, proposed by Mickley and Fairbanks (6) in 1955, suggests that a "packet" of particles from the bulk phase comes into contact with the heating surface. After remaining there only a short time, the "packet" moves back into the bulk phase where it is quickly dispersed, thus circulating the heat rapidly.

The third model, proposed by Ziegler, Koppel, and Brazelton (13), and modified by Genetti and Knudsen (3), and again by Bartel (1), views the mechanism as occurring with single particles. A particle moves

into the region near the heating surface, but rather than being heated by direct contact with the heater, it is surrounded by gas at the temperature of the heater. Because of the larger surface area exposed, the particle is rapidly heated and then moves back into the bulk phase, where the heat is dispersed.

Using this particle model, Ziegler, et al found that the particle Nusselt number can be expressed as:

$$Nu_p = \frac{7.2}{\left(1 + \frac{6k_{air}\bar{t}}{\rho_s C_{ps} D_p^2}\right)^2}$$

where,

k_{air} = thermal conductivity of fluidizing air, BTU/hr-ft-°F

ρ_s = density of particles, lb/ft³

$\bar{t}$ = mean residence time, hr

C_{ps} = heat capacity of solid particles, BTU/lb-°F

D_p = particle diameter, ft

Genetti and Knudsen found, from their study, that the constant 7.2, should be replaced by $5(1 - \epsilon)^{0.5}$, where $(1 - \epsilon)$ is the particle fraction in the bed. Bartel further extended the model to account for tube spacing in a bundle, and for fin height in cases where finned tubes are used. Zigler and Brazelton (14) showed, by comparing mass transfer and heat transfer rates, that a particle mode mechanism accounts for 80-95% of the total heat transfer from the heating surface to the bed.

Effect of Various Physical Parameters on Heat Transfer Coefficients

The physical parameters effecting heat transfer can be separated into three main areas: 1) fluidizing media, 2) particle bed, and 3) heater and tube assembly.

-Fluidizing media-

It has been shown by many investigators, including Davidson (2), Leva, et al, and Bartel, that the flowrate of the fluidizing gas has a pronounced effect on heat transfer coefficient. Depending upon the type and size of particles used, the heat transfer coefficient may increase monotonically, decrease monotonically, or pass through a maximum. These differences can be attributed to two opposing mechanisms occurring simultaneously. At lower flowrates, increasing the gas velocity enhances particle movement. This, in turn, increases the heat transfer coefficient. If the gas flowrate is increased further, the bed expands more, lowering the particle fraction. With fewer particles near the heaters, the heat transfer decreases.

This effect is determined somewhat by the particle size, and the point at which minimum fluidization occurs. That is, with large particles (0.015 inch diameter), the heat transfer coefficient decreases with increased velocity. With smaller particles (0.10 inch diameter), there is a maximum coefficient.

The other primary factor of the fluidizing gas, is the thermal conductivity of the gas. Ziegler, et al and Davidson, showed that

increased thermal conductivity enhances the heat transfer. The density of the fluidizing media also has some affect

-Particle Bed-

Several investigators, Davidson, Leva, et al, and Ziegler, et al, have shown that there are three major particle parameters which effect heat transfer in a fluidized bed. They are particle diameter, solids heat capacity, and particle shape.

Smaller particles have higher heat transfer coefficients in general. As particle size decreases, more particles can get into the heated region of the heaters. The additional area promotes heat transfer. This effect decreases at very small particles diameters (less than 0.01 inch).

The second effect, that of heat capacity, can also be explained. If more heat can be stored in a particle, then the heat transfer should be increased.

According to Bartel, particle shape can decrease the heat transfer coefficient, by reducing the overall surface area. Although the more jagged particles have more surface area per particle than spherical glass beads, the rough shaped particles can pack more closely together than can the spheres. Thus, the overall surface area is reduced.

Thermal conductivity and density of the solid particles does not seem to have an effect on the heat transfer. According to Davidson, the bed height also has little effect if the bed is more than a few centimeters high.

-Heater and Tube Assembly-

Two major factors have been found to have influence on heat transfer. Davidson suggests that the tube diameter has an effect for diameters increasing from about 0.1 mm to about 1 cm. Larger diameters, seem not to have any affect on heat transfer.

Also, according to Davidson, heat transfer can vary with angular position around a tube, because of the dynamics of flow around a horizontal cylinder in a fluidized bed. As shown in Figure 2, a stagnant cap of particles forms on top of the tube. In this cap, the particle motion is much less than in the bulk phase. On the bottom of the tube, a thin gas film forms, into which few particles can penetrate. The top and bottom of the tube are then effectively insulated and will have higher surface temperatures than will the sides of the tubes, where fluidization is much better. This higher temperature will cause lower coefficients to be observed on the top and bottom than on the sides. It is therefore important to locate the thermocouple in the same angular position on the tube for all experimental runs, to measure uniform heat transfer coefficients.

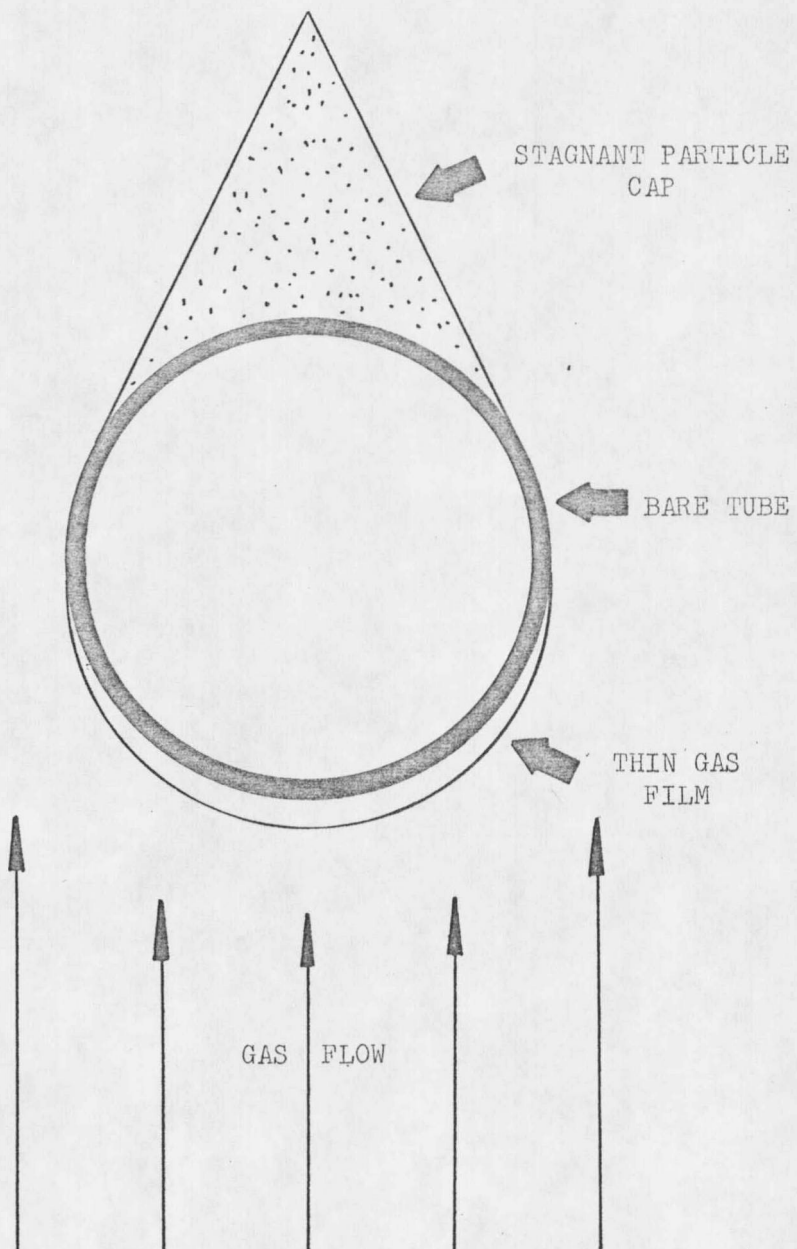


Figure 2. Bed Dynamics Around a Bare Tube

EXPERIMENTAL WORK

Objectives of This Study

This study was divided into two parts: 1) to determine the change in heat transfer coefficient with respect to the number of tubes in a horizontal tube bundle, and 2) to determine the effect of heater position in a 19 tube bundle. In addition, the air mass velocity was varied to obtain data on both sides of the maximum heat transfer as mentioned earlier. Preliminary runs were made to calibrate the instruments, and to determine if particle size had any effect other than that already mentioned.

Briefly then, the following experimental parameters were used in this investigation:

- Fluidizing Media: air, entering at room temperature. The inlet properties were assumed to be fairly constant.
- Bed Particles: spherical glass beads, manufactured by Minnesota Mining and Manufacturing Company. The final diameter selected was 0.011 inch, though, for preliminary runs, 0.008, and 0.0185 inch glass beads were also used.
- Tubes: carbon steel tubes, 5/8" outside diameter. Tubes in a bundle were located symmetrically about a center tube. Bundles of 1, 7, and 19 tubes were used.
- Heat Input: Two different levels of heat input were used as follows:
 - 1) single tubes and very low flowrates - 100-105 watts, 2) tube

bundles - 150-160 watts. The variations at each level were caused by inherent differences in the various heaters.

Experimental Equipment

The equipment used in this study, can be separated into three sections: 1) fluidizing column and air supply, 2) heater and tube assembly, and 3) related equipment.

-Fluidizing Column and Air Supply-

Figures 3,4, and 5 show the fluidizing column employed in this study. It was a rectangular column, 7.5 feet high, with inside dimensions of 14" by 6.625". It was constructed of 0.75" Plexiglas, and fastened together with screws and solvent to prevent air and particle leakage.

The top consisted of two pieces of Plexiglas with five 2" diameter holes in each. Sandwiched between the two pieces was 140 mesh brass wire cloth and a rubber gasket to prevent particle leakage. In all four sides at the top of the column, was another hole, 3" in diameter, with brass wire cloth covering them.

The distributor plate was constructed of two 1/8" steel plates perforated with 1/8" holes. Again, sandwiched between the plates, was brass wire cloth and a rubber gasket.

Two micarta plates, 15.5" by 18", were placed on opposite sides of the column, 5" above the distributor plate. In the front plate, 5/8" holes were drilled through to let the heaters pass through. At the corresponding points in the opposite plate, 1/2" holes were drilled

in 0.4" to accomodate the insulated ends of the heaters.

Air was pumped to the column from a Sutorbilt Blower run by a 7 1/2 Hp motor. Two valves, a main line valve and a bypass valve, were used to adjust the flowrate of air. To cut down on vibration, a section of rubber hose connected the steel pipe from the blower to the column. Eight stainless steel tubes located below the distributor plate, and perpendicular to the air flow, were used to straighten the flow as it entered the column.

-Tube and Heater Assembly-

Bare, carbon steel tubes with 5/8" outside diameter were used. A detailed drawing is given in Figure 6.

The heaters were Firerod cartridge electric heaters manufactured by Watlow Electric Manufacturing. They have a 6.5" heated section, and insulated ends of 0.4" and 3". The leads were connected through a Simpson wattmeter and a rheostat, to a 110 volt A.C. outlet. For runs in which a lower heat input was desired, a Variac was connected in the line. The Variac was used in some cases to maintain the input voltage, and at other times to prevent overheating in the column.

The blower described earlier, operated with a magnetic switch. Since it was necessary to have the blower on if the heaters were on, a safety switch for the heaters was connected through the blower switch. Thus, if the blower was accidentally shut off, the heaters would also shut off. A heater is shown in detail in Figure 7.

To promote contact between the tube and heater, the heater was coated with copper antisieze compound before inserting the heater into the tube.

An iron-constantan thermocouple was attached to each tube. Each thermocouple, located at the midpoint of the tube, was embedded $1/16$ " into the tube and silver soldered into place. Each thermocouple was checked before final assembly. The thermocouple wire was then passed out through a $1/16$ " diameter hole bored longitudinally in the 3" insulated end of the heater. This is shown in Figure 8.

-Related Equipment-

A thermocouple was located inside each of three, $1/16$ " diameter copper tubes. These tubes were located at three positions in the column to record the temperature at the top, bottom, and middle of the bed. The tubes were passed through the side of column. All the temperatures, both the tubes and bed, were recorded on a Honeywell Brown Elektronik chart recorder.

Two manometers were included in the system: one measured the pressure drop across an orifice in the main air line. It was used to determine the air mass velocity to the column. Backpressure from the column was accounted for with a Duraguage pressure guage. The other manometer measured the pressure drop across the tube bundle. Copper tubes, similar to those used for the bed thermocouples, were used for the pressure taps. The thermocouples and pressure taps in the bed are shown in Figure 9.

EXPERIMENTAL PROCEDURE

Preliminary Work

-Calibration of Bypass Valve-

Since several runs were to be made at each air flowrate, a reproducible method of setting the valves in the bypass and main air lines was needed. The orifice itself, was not suitable, because slugging in the fluidized bed caused large fluctuations in the manometer readings, thus making reproducible settings very difficult.

It was found that the main valve had to be kept fully opened to operate the bed. Then, by calibrating the number of turns of the bypass valve, to the air flowrate, a much improved system was obtained.

As mentioned earlier, the back pressure to the column was determined with a pressure gauge located between the orifice and the column. By running many air flowrates through the column, a calibration curve for the bypass valve was obtained as shown in Figure 10. This curve turned out to be a straight line in the region of interest.

-Calibration of Temperature Recorder-

The Brown Elektronik temperature recorder was calibrated using a mercury-in-glass thermometer immersed in either boiling water, or heated oil. These two fluids bracketed the temperature range of interest. This calibration was checked periodically.

-Normal Operating Procedure-

For a typical experimental run, the tube arrangement to be investigated, was set up on the micarta plate, with the longer

insulated end of the heater protruding through the Swagelock fittings. All the other holes were plugged with 1/2" plugs. The micarta plate was then positioned on the column and held in place by three trunk-lid clamps on each side. Latex sealant was used to seal the plate to the column and to prevent air and particle leakage.

The 0.4" insulated ends of the heaters were pressed into their appropriate holes in the opposite plate. The heaters were then rotated so that the thermocouple beads were on top the tube, and then the Swagelock fittings were tightened. The thermocouple and heater leads were plugged into their proper connectors.

The proper size particles were poured into the top of the column until a static bed of 23 inches was obtained. The top was replaced and held by four trunk-lid clamps. Since the overall size of all the tube bundles used was the same, except for single tubes, the pressure taps above and below the bundle were kept in the same position for all runs.

The air blower was turned on and the desired setting on the bypass valve was made. The column was operated until steady state conditions were obtained. Steady state was determined to be the time at which the bed temperature no longer changed with time. For the first run of a day, this was usually about 4 hours, and for subsequent runs, about 2 hours. When the first run achieved steady state, the temperatures, wattages, and manometer reading were recorded. The readings

were then repeated to obtain a better average. The flowrate was then changed for the second run.

When all runs of a given tube arrangement had been completed, the blower was shut off and the particles drained through a one inch pipe in the bottom of the column. The micarta was disassembled and a new tube configuration was arranged.

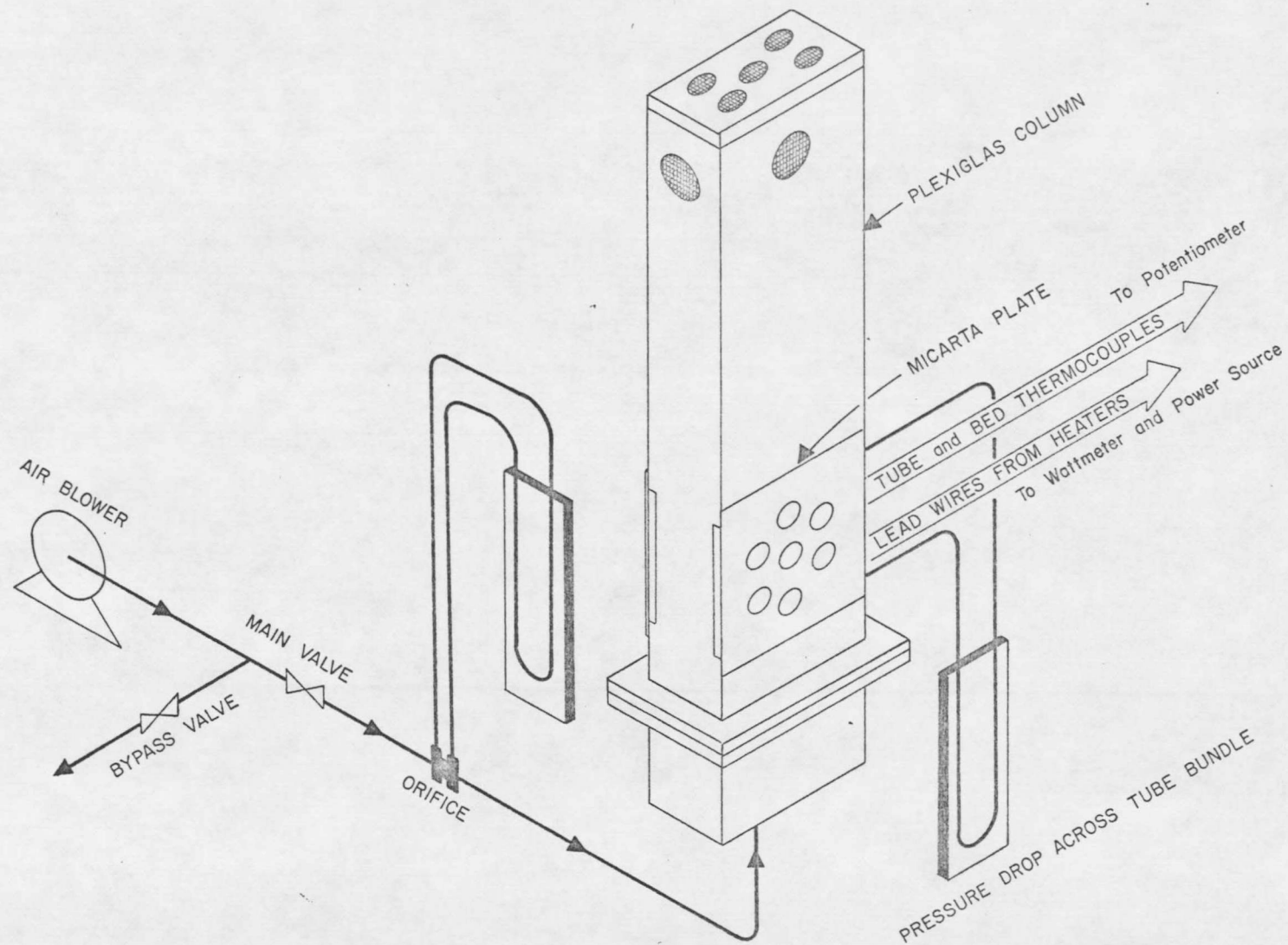


FIGURE 3. SCHEMATIC DRAWING OF EQUIPMENT

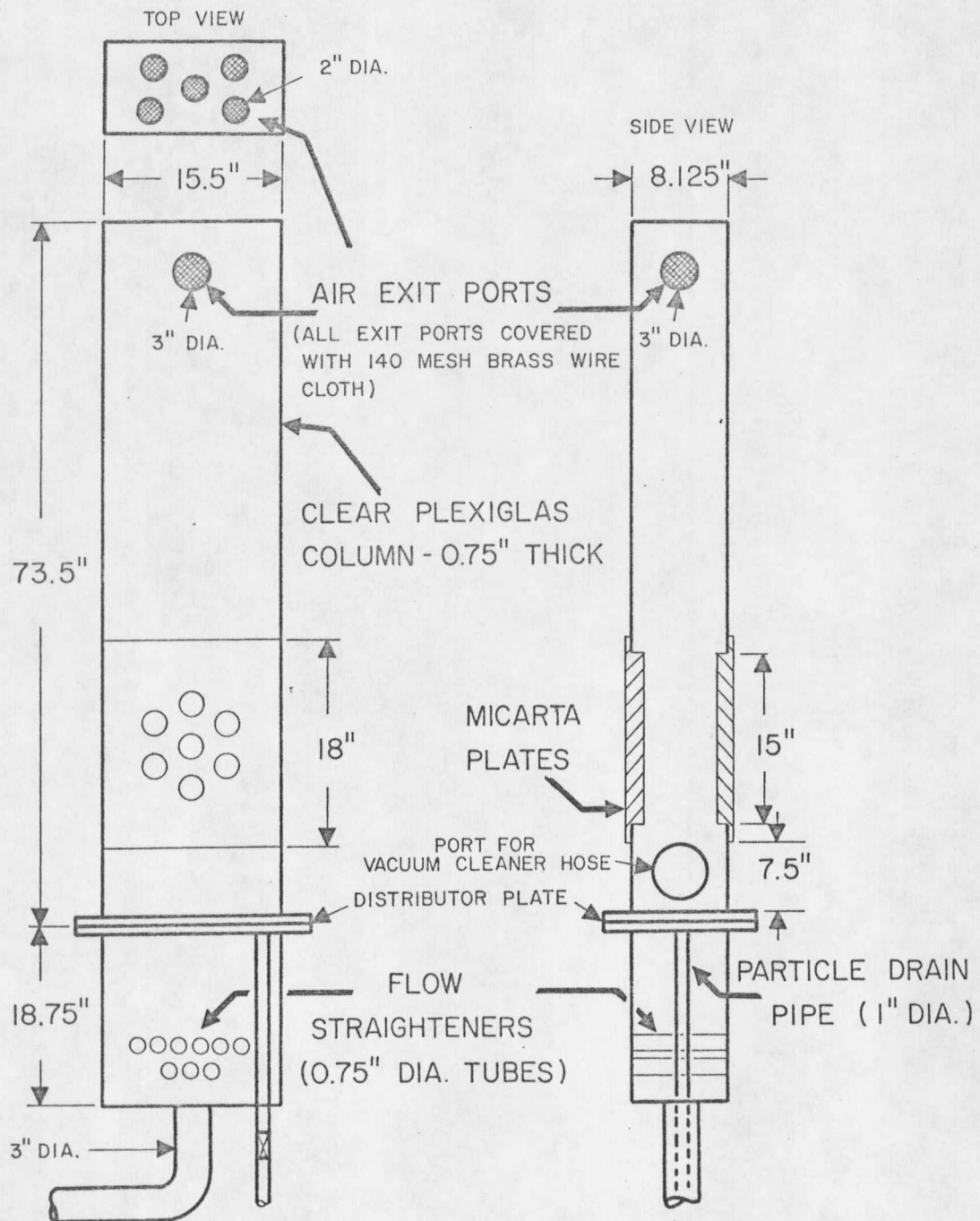


FIGURE 4. DETAILS OF FLUIDIZING COLUMN

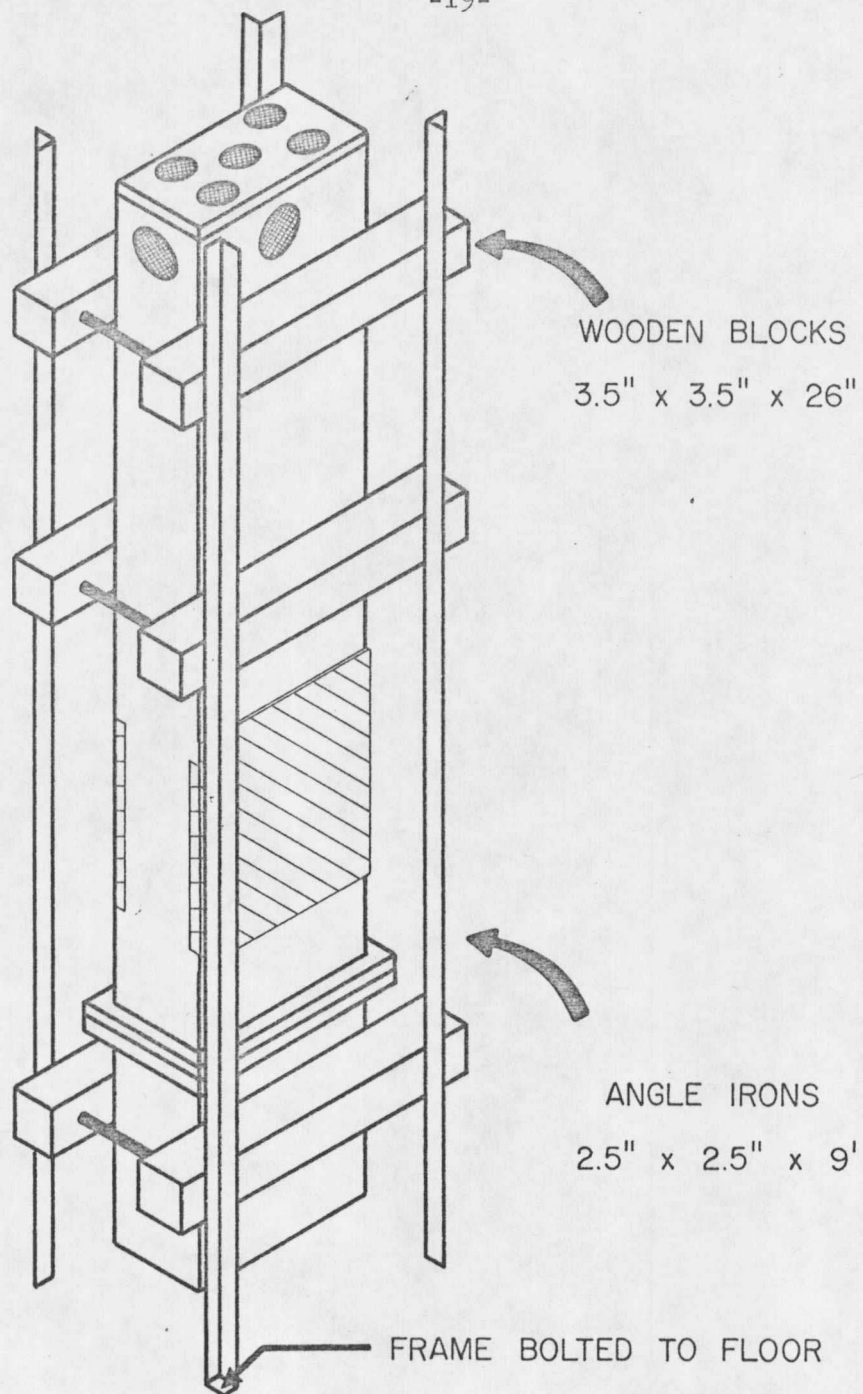
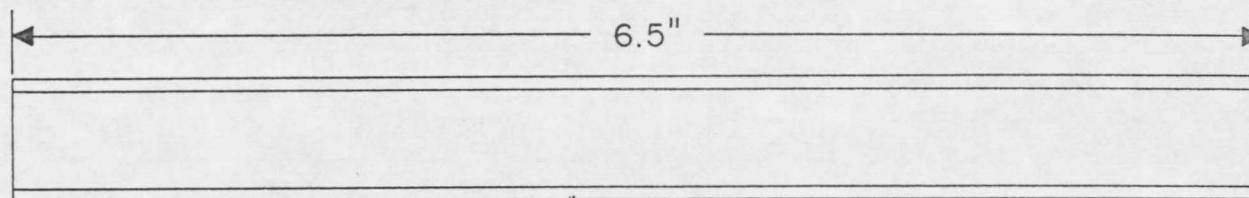


FIGURE 5. COLUMN AND SUPPORTING FRAMEWORK



TUBE OUTSIDE DIAMETER = 0.625"
TUBE INSIDE DIAMETER = 0.525"
TUBE WALL THICKNESS = 0.050"

MATERIAL
CARBON STEEL

SCALE: FULL

FIGURE 6. DETAILS OF A TYPICAL BARE TUBE

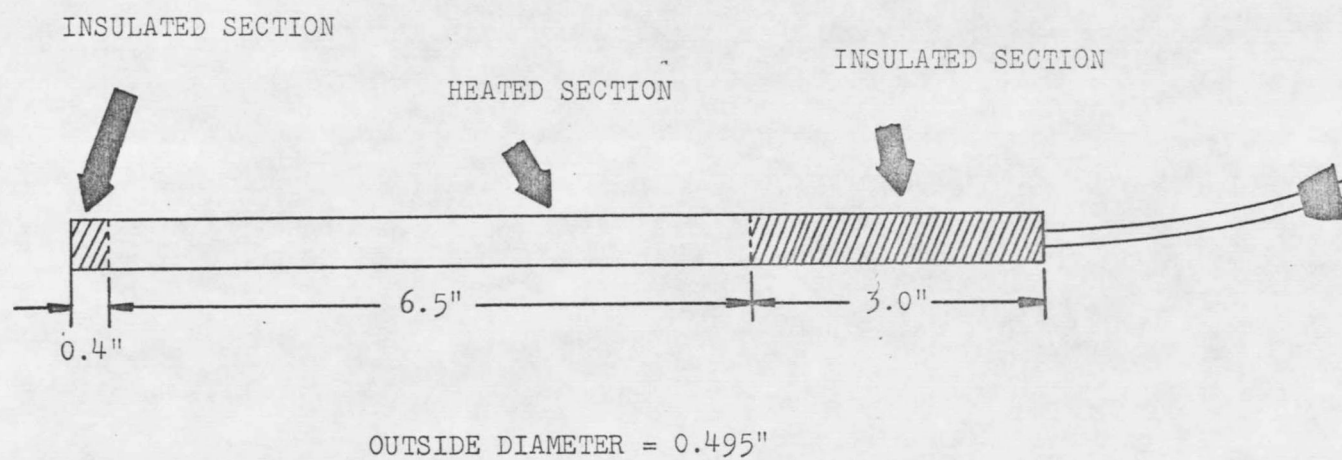
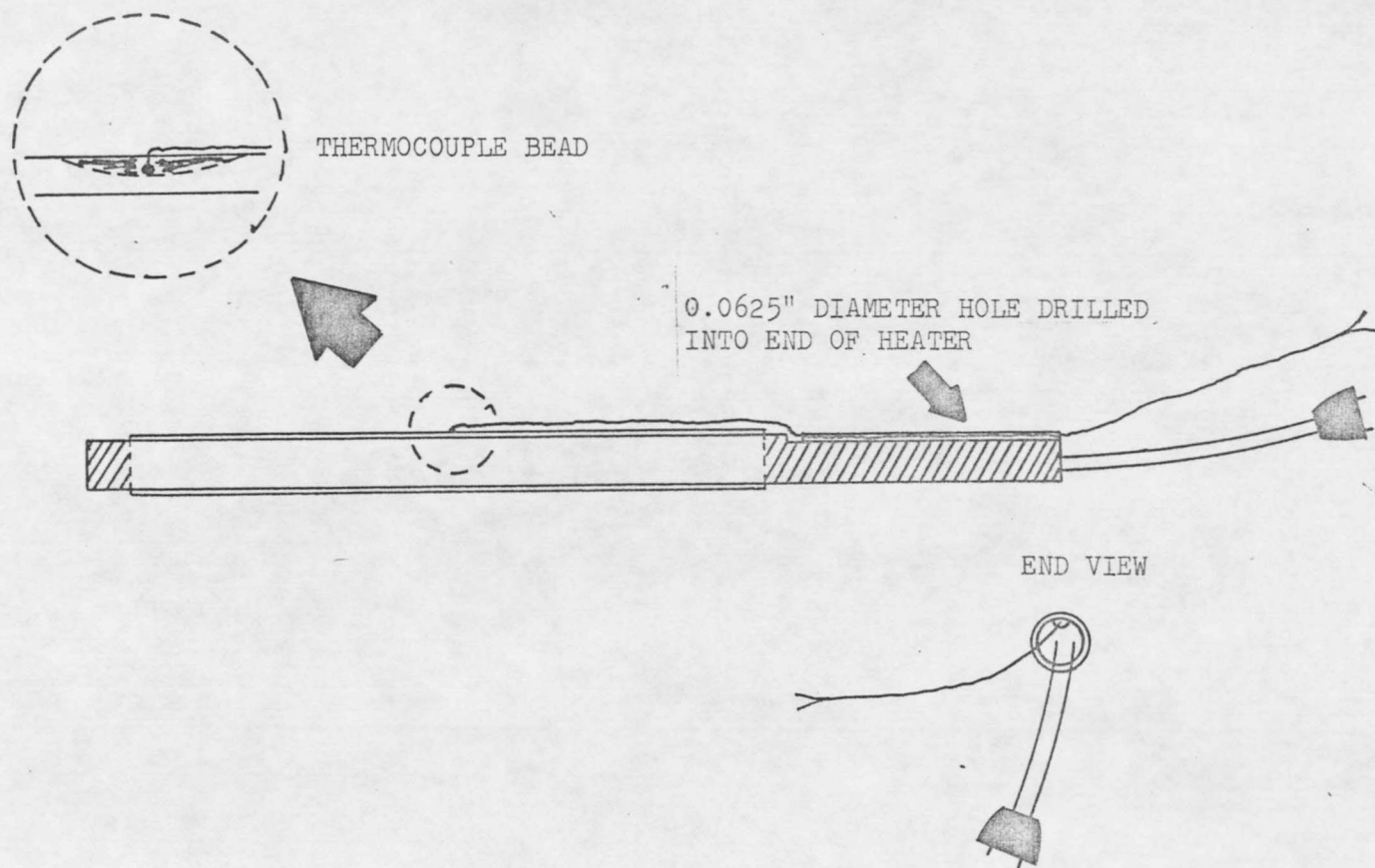


Figure 7. Details of Electrical Heater



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Figure 8. Tube and Heater Assembly

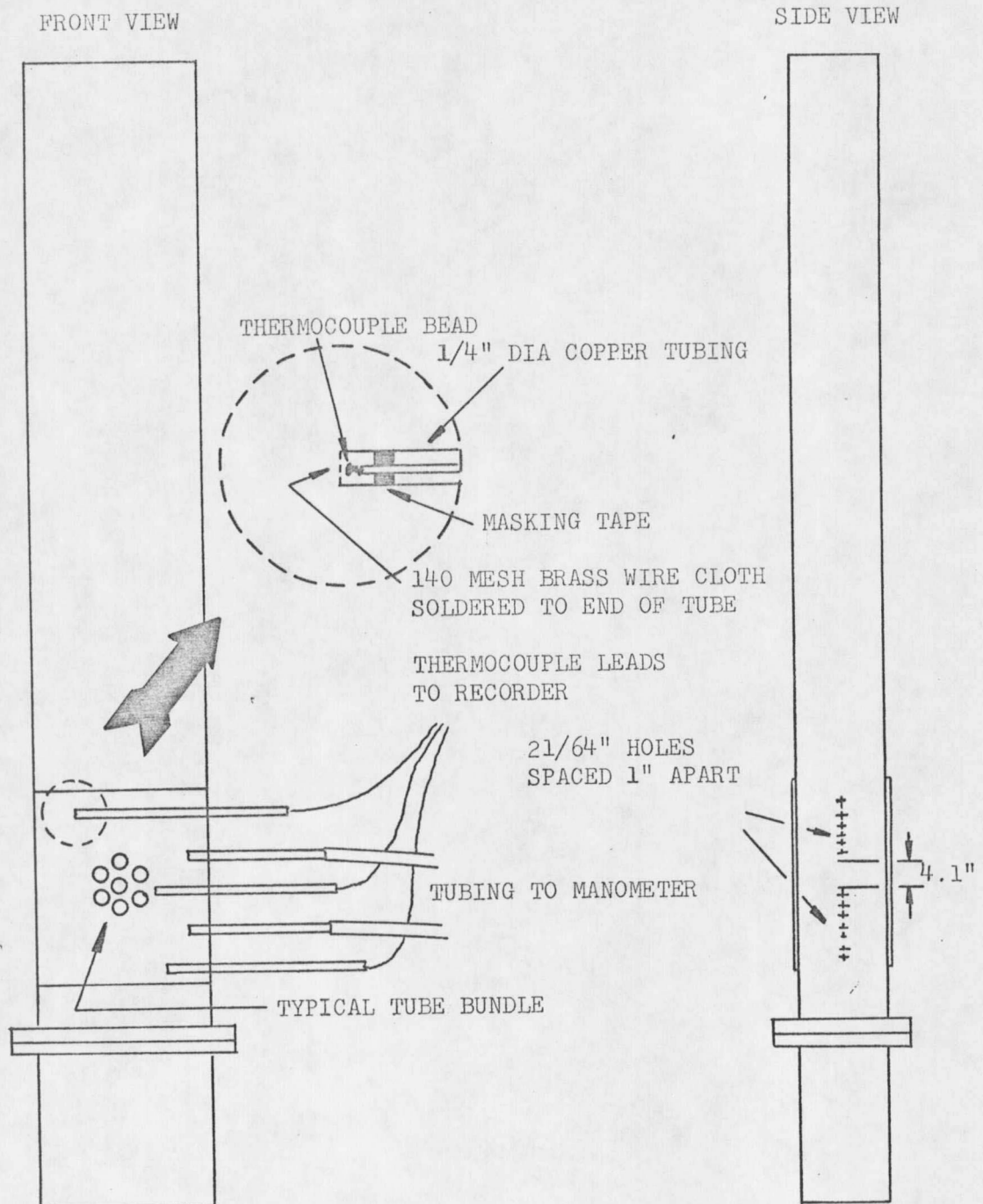


Figure 9. Details of Bed Thermocouples and Pressure Taps

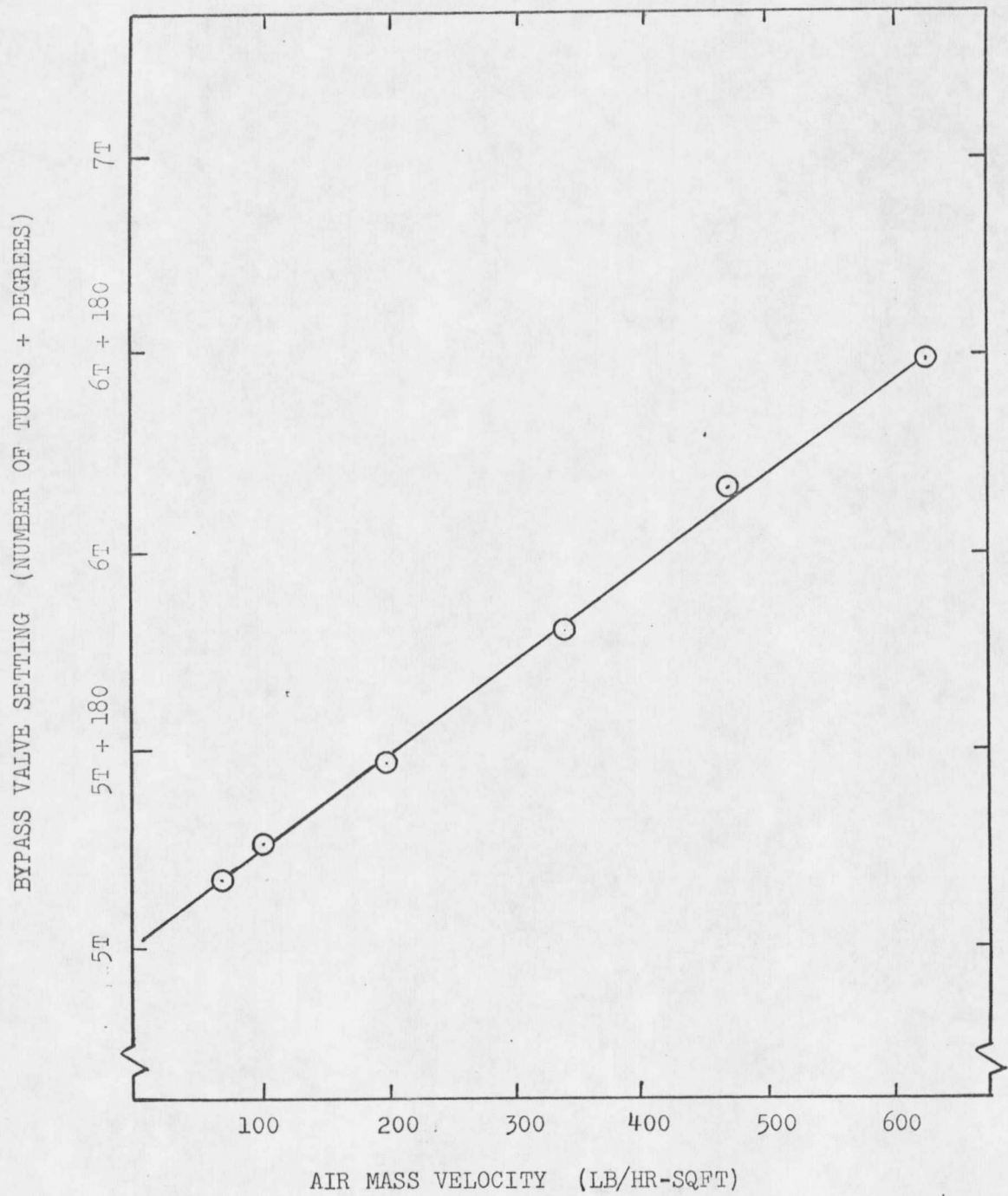


Figure 10. Calibration of Bypass Valve

CALCULATIONS

Air Flow Rate

The air mass velocity was calculated using the standard orifice equation (9):

$$W = \frac{3600 C Y S_c}{A_c} \sqrt{\frac{2 g_c (p_1 - p_2) \rho_1}{1 - \beta^4}} \quad ; \quad \text{lbm/ft}^2\text{-hr}$$

where,

C = orifice coefficient

Y = expansion factor

S_c = crosssectional area of orifice, ft

A_c = crosssectional area of column, ft

g_c = gravitational constant, lbm-ft/lbf-sec²

p_1 = pressure at upstream tap, lbf/ft²

p_2 = pressure at downstream tap, lbf/ft²

ρ_1 = upstream density, lbm/ft³

β = ratio of orifice diameter to inside column diameter

The expansion factor, Y , can be found from (9):

$$Y = 1 - \frac{(p_1 - p_2)}{p_1} \left(\frac{C_v}{C_p} \right) (0.41 + 0.35 \beta^4)$$

According to Sprenkle (11), the orifice coefficient for a sharp edged orifice under the conditions of this study, is, $C = 0.608$.

There is about 0.3% variation in this value over the entire flowrate range used in this study.

Heat Input

Power input to the heaters was measured with a wattmeter and converted to BTU/hr with the correct conversion factor:

$$q_i = \text{watts} \times 3.413 \text{ BTU/watt-hr}$$

Bed Temperature

As described, three thermocouples were used to measure the bed temperature. The arithmetic mean of these three was then taken to be the bed temperature:

$$T_{\text{Bed}} = \frac{T_{\text{Upper}} + T_{\text{Middle}} + T_{\text{Lower}}}{3} ; \text{ } ^\circ\text{F}$$

It was observed that the maximum difference between the three temperatures, was 2°F, with the upper bed being the lowest.

Tube Temperature

The tube temperatures were read directly from the chart recorder in degrees Fahrenheit.

Outside Tube Surface Area

The outside area of the tube was found from the standard equation for area of a cylinder:

$$A = \pi D_o L$$

where,

$$D_o = \text{outside diameter, ft}$$

$$L = \text{length of tube, ft}$$

The area of the tubes used in this study, was found to be 0.0870 ft².

Heat Transfer Coefficient for Individual Tubes

The standard heat transfer equation was used to find the heat transfer coefficient of each tube:

$$h_i = \frac{q_i}{A_i(T_{\text{Tube}} - T_{\text{Bed}})} \quad ; \quad \text{BTU/hr-sqft-}^\circ\text{F}$$

Heat Transfer Coefficient for Tube Bundles

The heat transfer coefficient for a tube bundle was found as the arithmetic mean of all the heated tubes:

$$\bar{h} = \frac{\sum_{i=1}^n h_i}{n} \quad ; \quad \text{BTU/hr-sqft-}^\circ\text{F}$$

Particle Nusselt Number

The nusselt number based on the particle diameter was found from:

$$Nu_p = \frac{\bar{h}D_p}{k_f}$$

where,

D_p = particle diameter, ft

k_f = thermal conductivity of air, BTU/hr-sqft- $^\circ\text{F}$

Particle Reynolds Number

The Reynold's number based on the particle diameter is:

$$Re_p = \frac{D_p G}{\mu}$$

where,

G = air mass velocity, lbm/hr-ft²

μ = air viscosity, lbm/hr-ft

Since the thermal conductivity and viscosity of air vary with temperature, a straight line was fit to data obtained in the literature, to obtain useful equations:

$$k_f = 0.0133 + 0.000024T_{\text{Bed}}$$

$$\mu = [2.45(T_{\text{Bed}} - 32) + 1538.1] 2.688 \times 10^{-5}$$

Particle Fraction

According to Parent, Yogel, and Stiener (8), pressure drop in a fluidized bed should be equal to the weight of solid per unit cross-sectional area of the column. This leads to the particle fraction:

$$1 - \epsilon = \frac{\Delta P}{L_t(\rho_s - \rho_f)}$$

where,

ΔP = pressure drop, lbf/ft²

L_t = distance between pressure taps, ft

ρ_s = solids density, lbm/ft³

ρ_f = air density, lbm/ft³

EFFECT OF NUMBER OF TUBES IN A TUBE BUNDLE

The first part of this study was concerned with the effect of the number of tubes in a tube bundle on the heat transfer coefficient.

Preliminary investigation indicated that the relative difference between a 7 tube bundle and a 19 tube bundle, was about the same for all particle sizes tried; 0.008, 0.011 and 0.0185 inch diameter. Furthermore there was no consistency to any differences observed.

The medium size particles, 0.011 inch, were chosen for this study, because previous work has shown that a plot of heat transfer coefficient versus air mass velocity passes through a maximum. This maximum is explained on the basis of two competing mechanisms, as explained earlier in this paper.

If both of the mechanisms discussed are important, the curve will pass through a maximum. On the basis of where this maximum occurred for the medium particles, flowrates were chosen to bracket the maximum.

Table 1 gives the range of experimental parameters over which this present study is applicable.

To study the effect of number of tubes in a tube bundle, three different size bundles were used; 1, 7, and 19 tubes. In addition, three flowrates were selected; 75, 200, and 400 lb/hr-sqft. The lowest of the flowrates was the lowest that could be obtained and still retain uniform fluidization. Even at this flowrate, the tube

TABLE 1. Range of Experimental Applicability

<u>Variable</u>	<u>Range</u>
Particle diameter	0.011 inch
Fluidizing velocity	75-500 lbm/hr-sqft
Tube diameter	0.625 inch
Bed temperature	95-200 °F
Tube surface temperature	150-280 °F
Center to center distance	1.625 inch
Number of tubes	1, 7, 19 tubes

surface temperatures were unstable. This effect was most noticeable with a single tube, where air bubbles were less likely to come into contact with the tube. Since at this flowrate, the particle motion is very limited, the tube surface temperature tends to rise sharply until a bubble passes and circulates the particles. In a tube bundle, there is more area for the bubbles to strike, and hitting any part of the bundle tends to disperse the air bubble and circulate particles in the bundle.

Only seven heaters were used in the 19 tube bundle, to prevent overheating. The remaining tubes had "dummy" rods in them. According to Lese and Hermode (4), unless a stagnant gas cavity forms between a heated tube and an unheated tube, no change in heat transfer coefficient occurs. It was assumed then that these seven tubes would

act as they would if all the tubes were heated, and that the heat transfer coefficient would be indicative of the entire bundle.

Since variations occurred between different heater and tube combinations, to obtain a good average of the single tube, each combination was run as a single tube. The seven coefficients were then averaged.

Some indication of the accuracy of the results can be seen by comparing the experimental data to correlations found the literature. Petrie, Freebie, and Buckham (10), correlated heat transfer coefficients for horizontal bundles of tubes heated by steam. Their correlation only predicts the rising portion of the curve. Bartel's correlation (1) involves coefficients based upon bundles of finned and bare tubes. His correlation predicts both the rising and falling portions of the curve. Vreedenberg (12) proposed a correlation for single tubes.

According to Petrie, et al., The Nusselt number based on the tube diameter is given by:

$$Nu_t = 14(G/G_{mf})^{0.33}(Pr)^{0.33}(D_t/D_p)^{0.667}$$

where,

$$Nu_t = hD_t/k$$

G/G_{mf} = reduced fluidizing air mass velocity ratio

Pr = Prandtl number of fluidizing air

D_t/D_p = Ratio of tube diameter to particle diameter

Fluid properties for Petrie's correlation are evaluated at a

film temperature defined as the arithmetic average between the bed temperature and the tube surface temperature.

According to Bartel, the particle Nusselt number can be expressed as:

$$Nu_p = \frac{10(1 - \epsilon)^{0.5} \left[1 - \frac{0.027 + 4.3L^{1.5}}{P(1.12 + 3.2L^{0.6})} \right]}{\left[1 + \frac{0.00102 + 0.047L^{0.8}}{Re_p(0.33 + 0.4L^{9.33})D_p(1.23 - 0.57L^{0.23})} \right]^2}$$

where,

$$Nu_p = hD_p/k$$

$1 - \epsilon$ = particle fraction

L = fin height, inches

P = center to center distance, inches

$$Re_p = D_p G / \mu$$

Fluid properties are evaluated at the bed temperature.

For a single horizontal tube, according to Vreedenberg,

$$Nu_t = 420 Pr^{0.33} (GD_t \mu / D_p^3 \rho_s \rho g)^{0.33}$$

where,

G = air mass velocity, lbm/hr-sqft

μ = air viscosity, lbm/hr-ft

ρ_s = apparent density of solids, lbm/ft³

ρ = air density, lbm/ft³

g = acceleration due to gravity, ft/hr²

Fluid properties are again evaluated at the bed temperature.

These three correlations and the experimental data are plotted

in Figure 11. The single tube data are well within the 29% error observed by Vreedenberg in his data. In the range of Petrie's correlation (59-283 lbm/hr-ft²) the data for tube bundles agree well within the 29% error of his data. The data also agree well within the 15% error observed by Bartel, in the range of his correlation (120-825 lbm/hr-ft²). This gives some indication of the validity of the experimental results obtained in this study.

In Figure 12, the heat transfer coefficient is plotted versus the number of tubes in the bundle. At higher flowrates, a maximum coefficient occurs at about 7 tubes. As the flowrate is decreased, the maximum gradually disappears until, at low air velocities, the curve shows no maximum, but is only monotonically increasing. In all cases, the curves tend toward a given value above 19 tubes. There is little difference also, in the coefficient for a single tube.

The explanation of this data is given with respect to the fluidization of the bed. At high flowrates, slugging occurs, in which large volumes are occupied by air. This slugging tends to reduce the particle fraction and so, the heat transfer coefficient decreases. A single tube in the bed does little to break up the slugging. A tube bundle, however, having more area, does tend to break up the slugging, at least in the vicinity of the heaters. This increases the particle fraction around the heaters, and hence the heat transfer coefficient is increased.

A second, opposing mechanism, also becomes important in larger tube bundles. Particles entering a large bundle with a relatively high velocity, are forced to alter their paths, thus slowing them down appreciably. This hinderance of particle motion lowers the heat transfer coefficient.

The breakup of the slugging is probably less pronounced between two large bundles, than between a single tube and a bundle. Since the column has only finite dimensions, there is a limited number of tubes which can be placed across the column, and still maintain a given tube spacing. Larger bundles must increase in number in the vertical direction which has a smaller effect on the slugging.

The particle hinderance, however, will be increased with more tubes in either a vertical or horizontal direction. As the first mechanism loses importance, the second mechanism begins to dominate, so the curve passes through a maximum.

At lower flowrates, the mechanism seems to be slightly different. Since the particles are already moving very slowly, they are not slowed appreciably within a tube bundle. Rather, the particles tend to be circulated more by being forced around tubes. This tends to increase particle circulation and enhances heat transfer. Thus, the coefficient increases monotonically with the number of tubes in a bundle.

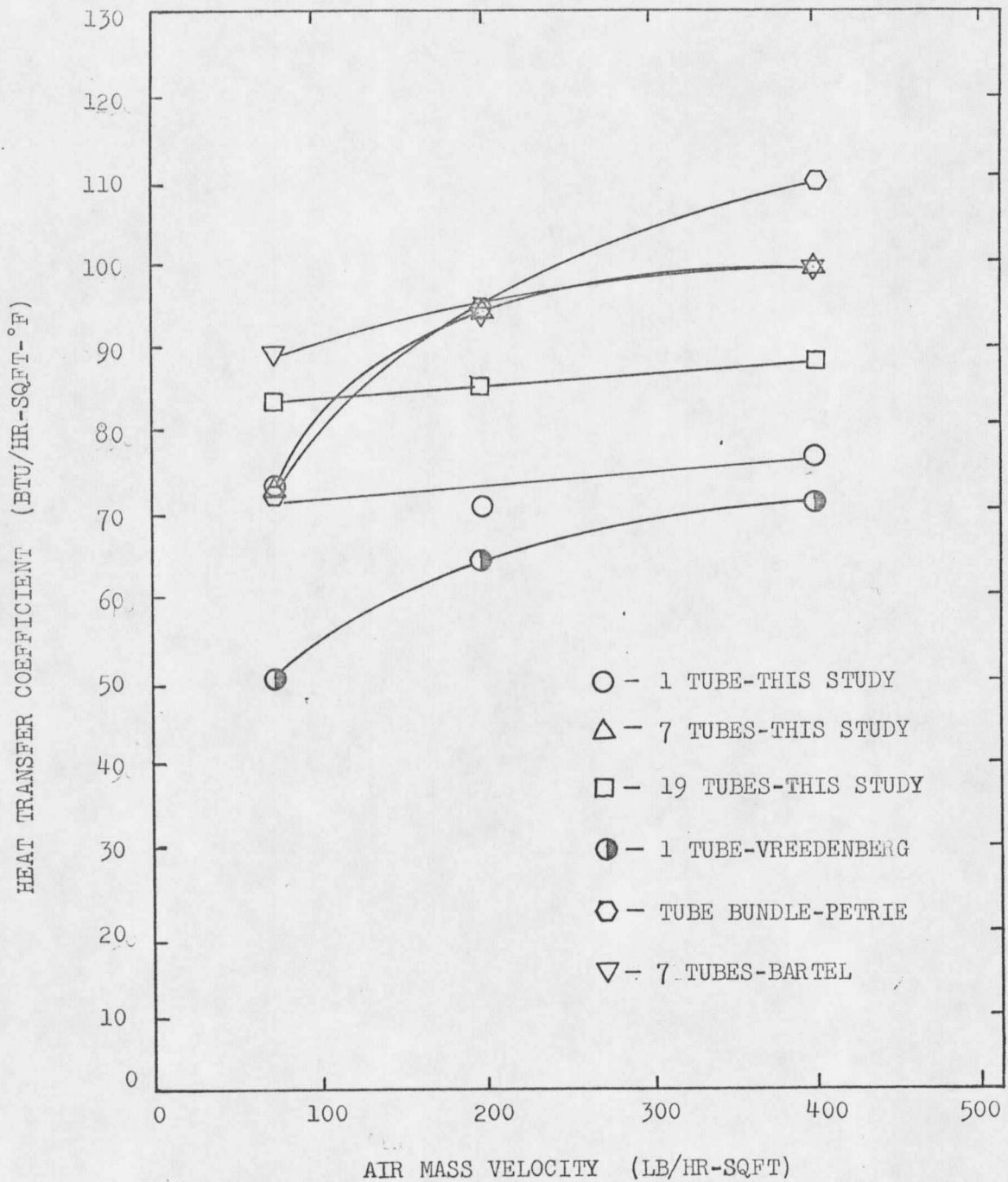


Figure 11. Comparison of Experimental Data with Literature Correlations

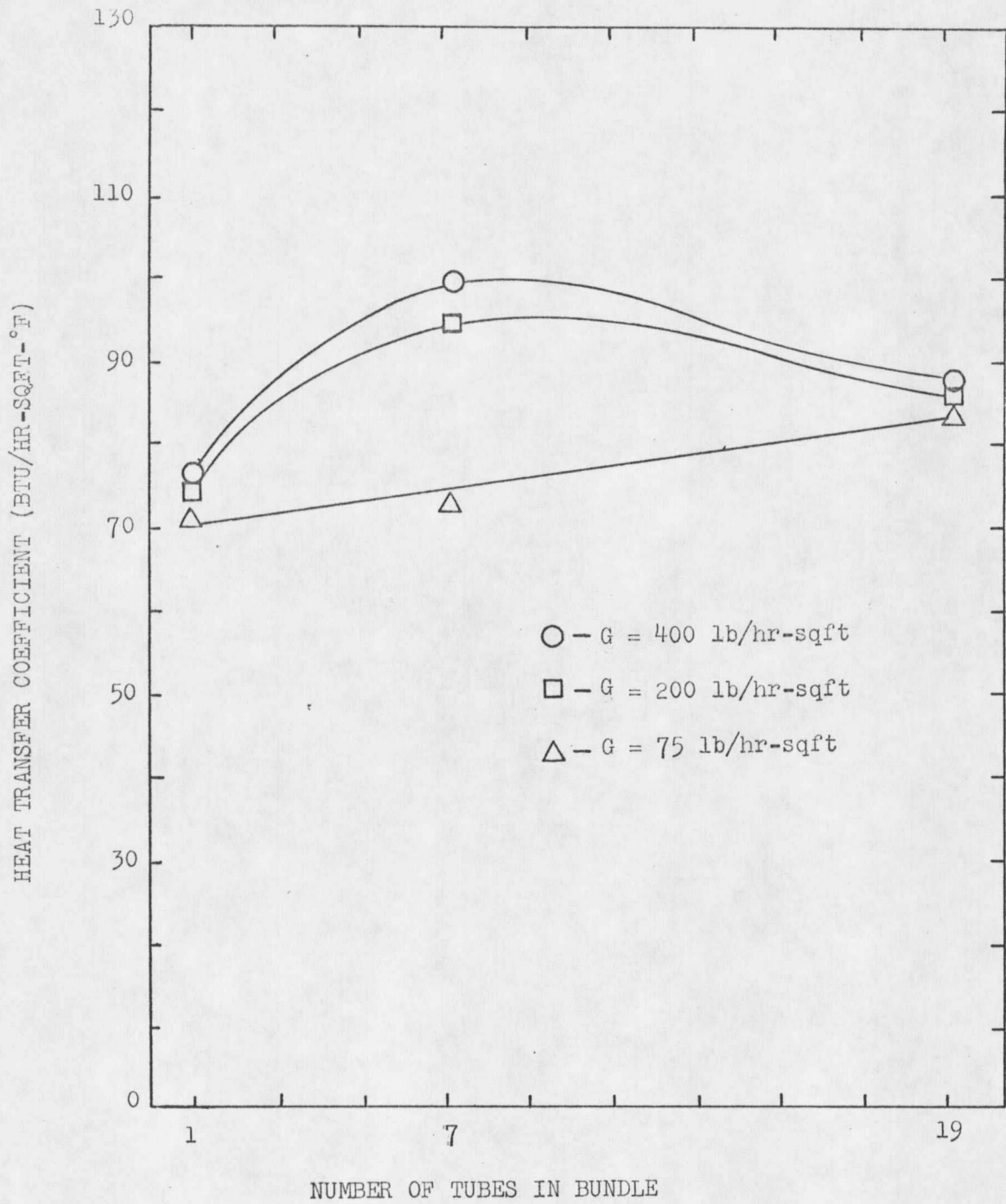


Figure 12. Performance of Tube Bundles

EFFECT OF POSITION IN A 19 TUBE BUNDLE

In determining the effect of position on heat transfer coefficient, it was necessary to correct for differences in the various heaters and thermocouples. In preliminary work, differences of up to 20% were observed between given heater and tube combinations. Repeated runs for the heater and tube assemblies, gave a maximum error of about 4%, indicating that something other than experimental error was causing the differences.

Resistances in the heaters and thermocouples were measured and found to vary as much as 7-9%. To account for these differences, each experiment was performed seven times, alternating the heater assemblies. The average at each position was then found so that the "real" differences could be seen. Only seven heaters were run, as in the previous section. The seven heaters were arranged in two configurations, as shown in Figures 13 and 14. The first configuration was chosen to show a symmetrical formation about the center tube, and the second, because it was asymmetric.

The results of this part of the study conflicted somewhat with anticipated results. It was felt that particle motion would be hindered somewhat in the interior of the tube bundle, thus lowering the transfer coefficient. For the symmetric configuration, this was indicated. The center tube had the lowest coefficient, and, in general, the

coefficients increased with distance from the center tube.

Since the two tubes on either side, horizontally, of the center tube were symmetric, it was expected that they would have the same coefficient. However at lower flowrates, this was not true, because the fluidization was not even across the bed. One side of the bed had considerably more particle circulation than did the other side. As the air velocity was increased, slugging began to even out the fluidization, and the two coefficients became equal.

The second configuration gave unexpected results. Rather than having the lowest coefficient in the center, the highest coefficient occurred in the center. Once more, the higher flowrates were considered more representative, because the fluidization was more uniform.

The lowest coefficient was observed in position 11, on the extreme right of the column, and nearly surrounded by heaters. No explanation could be given at this time concerning this effect, though an interesting trend was noticed. It seems that the presence of heaters around a heated tube may have some effect on the coefficient. This would also support evidence that the center tube in the bundle of heaters should be the lowest.

Plots of heat transfer coefficient versus air mass velocity are shown in Figures 15 and 16, for each heated tube in the two configurations.

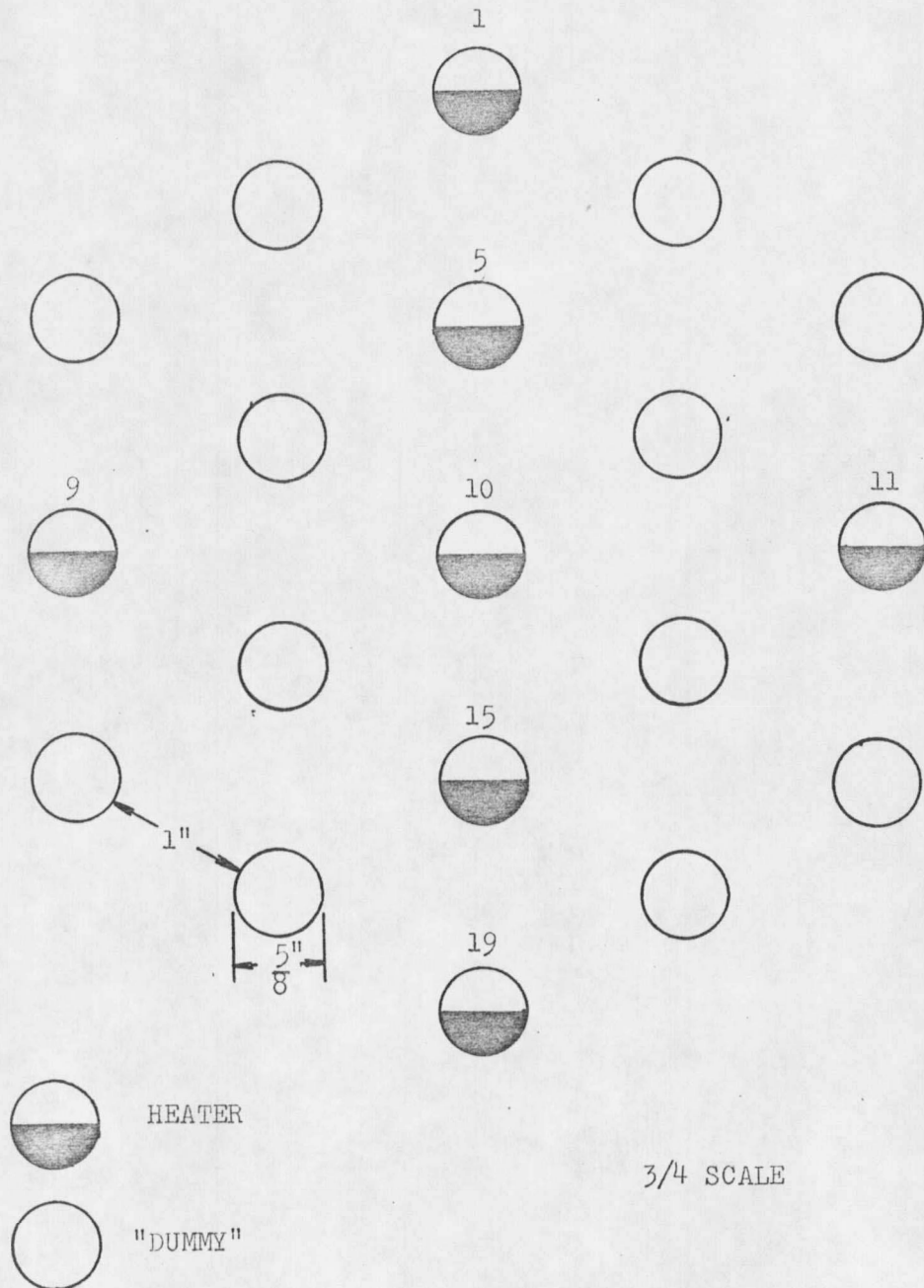


Figure 13. 19 Tube Bundle-Configuration One

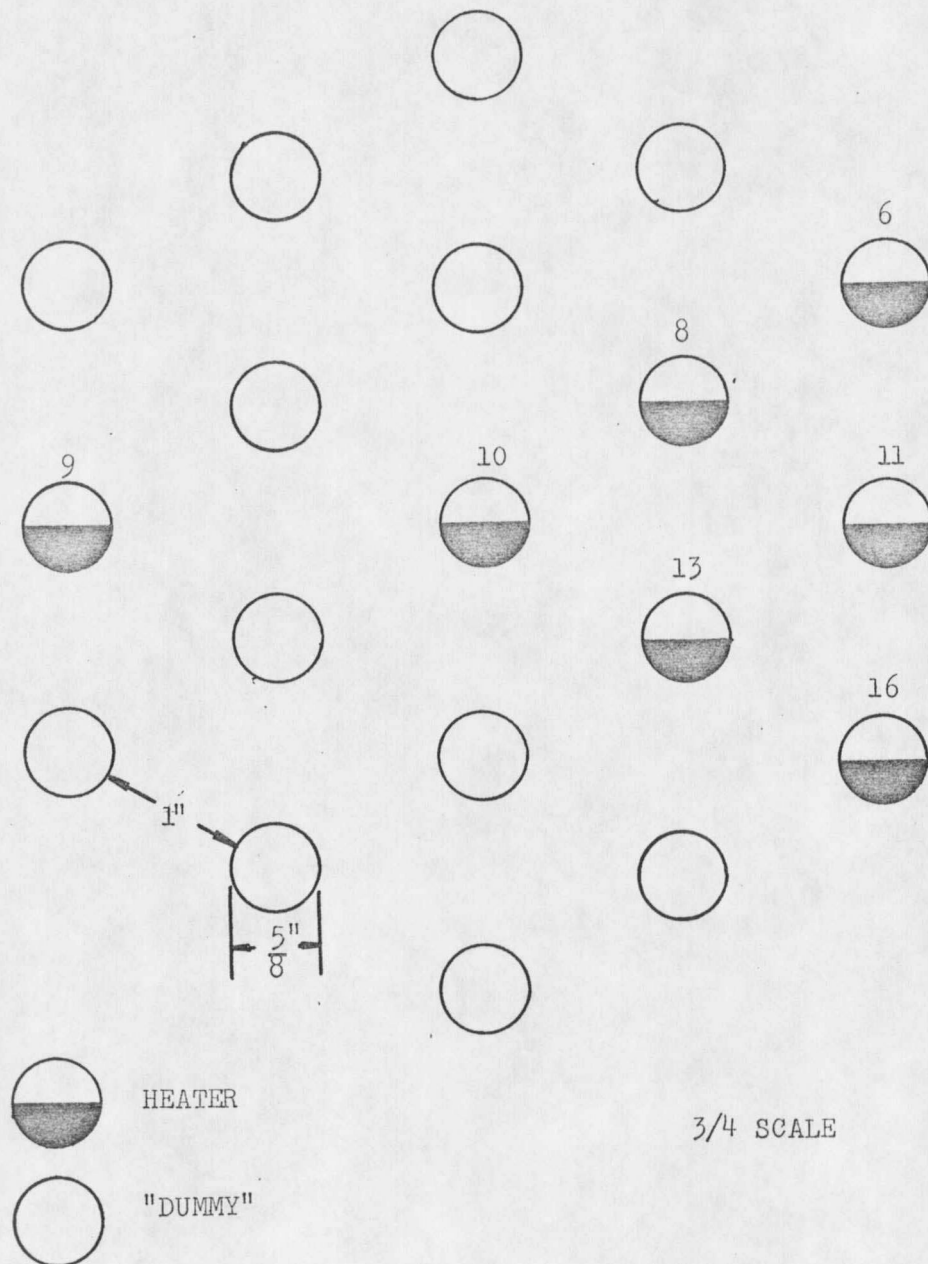


Figure 14. 19 Tube Bundle-Configuration Two

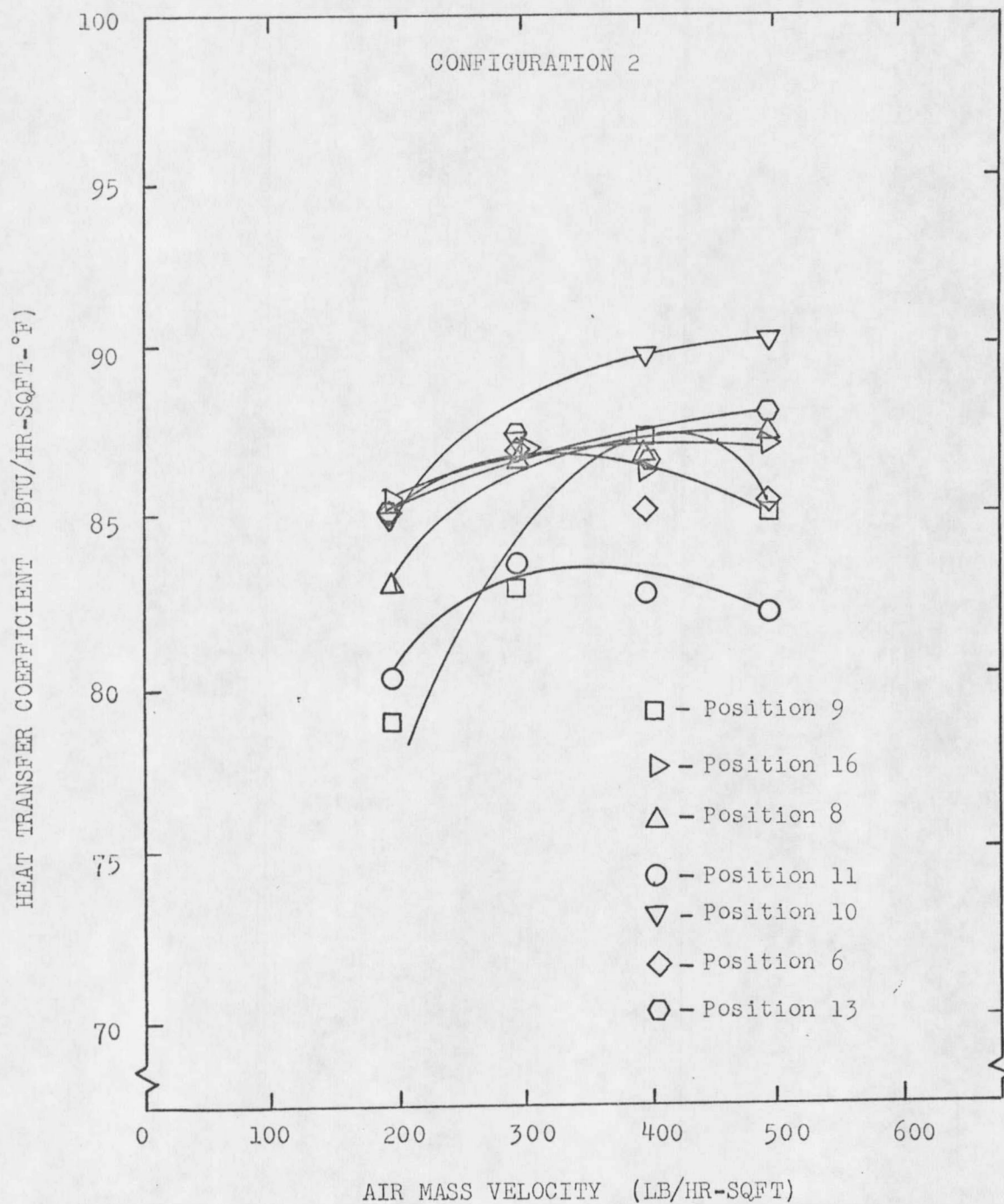


Figure 16. Performance Curves for Individual Bare Tubes

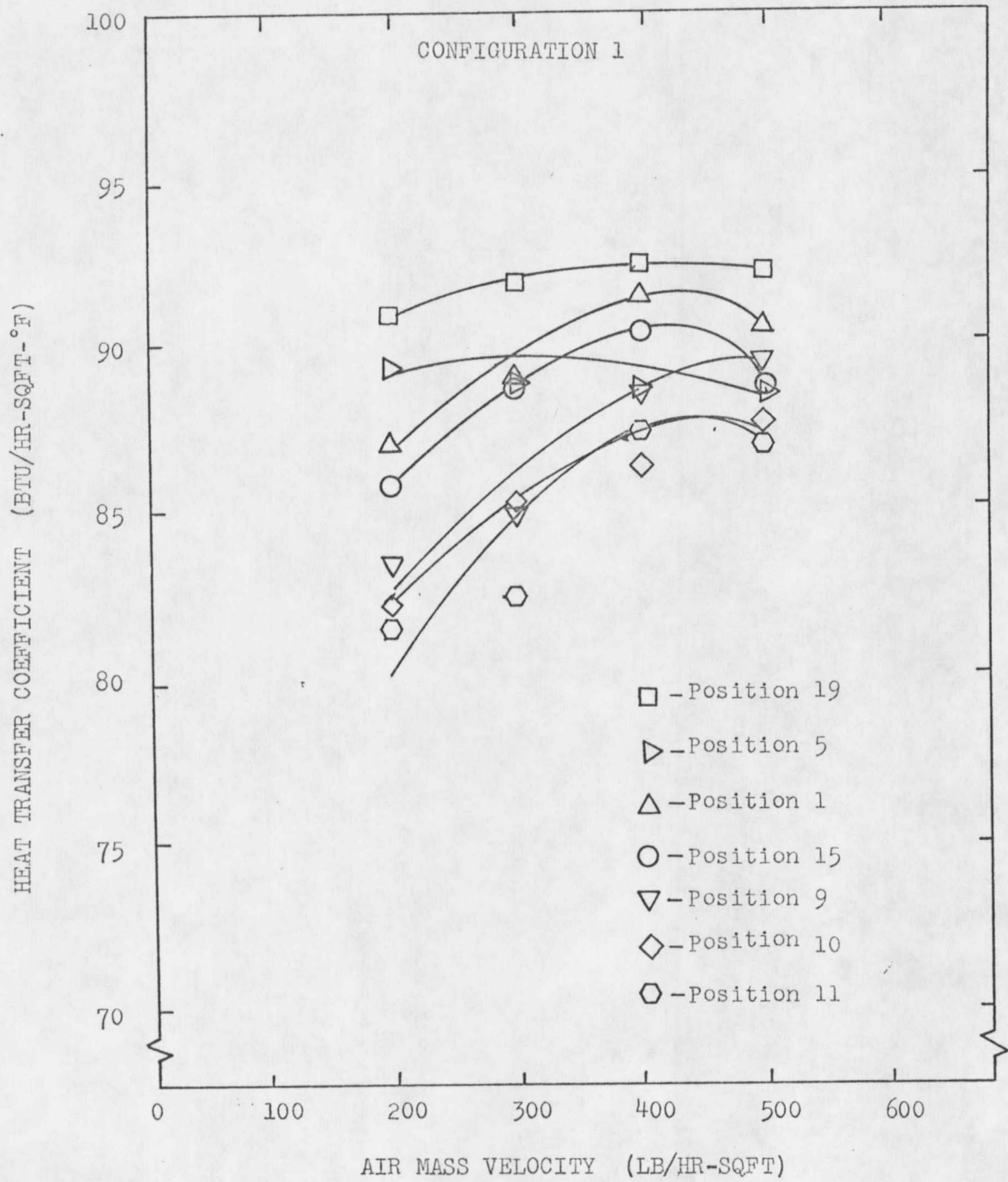


Figure 15. Performance Curves for Individual Bare Tubes

CONCLUSIONS

Part I

Based upon data obtained, two competing mechanisms seem to operate when determining how the number of tubes in a tube bundle affects the heat transfer coefficient, at higher flowrates:

- 1) Slugging in the column is broken, thus decreasing void fraction
- 2) Hindrance to particle motion increases with number of tubes.

At lower flowrates, larger bundles tend to increase particle motion, thus increasing the heat transfer coefficient.

Part II

Coefficients tend to be smaller in the center of tube bundles where particle motion is slowed somewhat. Some indication is given that the proximity of heated tubes to a given heated tube in question may affect the heat transfer.

RECOMMENDATIONS FOR FURTHER STUDY

Part I

Several larger bundles should be investigated to determine if some constant heat transfer coefficient results with large bundles. Bundles of five to ten tubes should be studied to determine where the maximum coefficient occurs.

Part II

Several more configurations should be studied to determine if the results obtained here are genuine, since only a small amount of data were taken.

NOMENCLATURE

<u>Symbol</u>	<u>Definition</u>	<u>Dimensions</u>
A	Area of bare tube	ft ²
A _c	Cross-sectional area of tube	ft ²
C	Orifice coefficient	dimensionless
C _p	Heat capacity	BTU/lb-°F
D _p	Particle diameter	ft
D _o	Orifice diameter	ft
D _t	Tube diameter	ft
G or W	Air mass velocity	lbm/hr-ft ²
G _{mf}	Air mass velocity at minimum fluidization	lbm/hr-ft ²
g	Acceleration due to gravity	ft/hr ²
g _c	Gravitational constant	lbm-ft/lbf-hr ²
h _i	Individual tube heat transfer coefficient	BTU/hr-ft ² -°F
h or $\bar{h}$	Average heat transfer coefficient	BTU/hr-ft ² -°F
k	Thermal conductivity	BTU/hr-ft-°F
L	Tube length	ft
L _t	Distance between pressure taps in bed	ft
Nu _p	Particle Nusselt number	dimensionless
Nu _p	Nusselt number based on tube diameter	dimensionless
P	Center to center spacing	inches

Pr	Prandtl number	dimensionless
p_1	Upstream pressure in pipe	lb_f/ft^2
p_2	Downstream pressure in pipe	lb_f/ft^2
q	Heat transfered	BTU/hr
Re_p	Reynold's number	dimensionless
S_c	Cross-sectional area of orifice	ft^2
T_{Bed}	Temperature of bed	$^{\circ}\text{F}$
T_{Tube}	Temperature of tube surface	$^{\circ}\text{F}$
$\bar{t}$	Mean residence time	hr
Y	Expansion factor in orifice	dimensionless
β	Ratio of orifice diameter to inside column diameter	dimensionless
ϵ	Void fraction in particle bed	dimensionless
μ	Viscosity of fluidizing air	$\text{lb}_m/\text{hr-ft}$
ρ or ρ_f	Density of fluidizing air	lb_m/ft^3
ρ_s	Density of solid particles	lb_m/ft^3

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