



Heat transfer from single, vertical, longitudinally finned tubes in an air fluidized bed
by Steven Daniel Hickel

A thesis submitted in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE
in Chemical Engineering
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Abstract:

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The Nusselt numbers for the finned tubes were correlated with two equations. The first equation was for the limestone data and the second was for the glass bead data.

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Date October 8, 1981

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FINNED TUBES IN AN AIR FLUIDIZED BED

by

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A thesis submitted in partial fulfillment
of the requirements for the degree

of

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ABSTRACT

Heat transfer coefficients were measured from vertical, bare and longitudinally finned tubes to a circular air fluidized bed. The fluidized particles were glass beads and limestone particles. Experimental parameters were fin height, bed particle diameter, bed particle type, and air fluidizing velocity.

Results for the finned tubes indicate that for increased air mass velocity the heat transfer coefficient rises. In several cases a maximum was obtained for the value of the heat transfer coefficient. The smaller fin height tubes generally gave higher values for the heat transfer coefficient. For a given tube, the glass beads gave a higher maximum heat transfer coefficient, but at a much lower air mass velocity than where the peak occurred for the limestone particles. The best performance was obtained with the 1/2 inch fin height tube using glass beads, and with the 3/8 inch fin height tube using limestone particles.

The Nusselt numbers for the finned tubes were correlated with two equations. The first equation was for the limestone data and the second was for the glass bead data.

INTRODUCTION

A fluidized bed is a column containing solid particles and having a porous bottom through which a fluid can be forced to flow. The motion of the fluid moving through the particles causes them to be lifted and mixed. The degree and dynamics of the fluidization are dependent upon the physical characteristics of the system and the rate at which the fluid flows through the system.

In a gas fluidized system, such as is used in this experimental program, several regimes exist during fluidization. Before the upward force of the air is equal to the weight of the bed material there is no fluidization and the bed remains fixed. The properties of the bed during this stage are essentially the same as those of a fixed bed. When the air velocity is increased above the point where the force of the air can support the weight of the bed, the bed begins to fluidize. This point is called the minimum fluidization velocity, or the incipient fluidization velocity. Near minimum fluidization there is only mild mixing of the particles with the air flowing between the particles and no bubbles.

being formed. Further increase in the fluid velocity causes larger and larger bubbles to form, increasing particle mixing. The bed expands during this stage of the fluidization. This regime of fluidization is called aggregative, because expansions and contractions in the bed volume result in a time dependent bed density. Bubble size increases with fluid velocity increase until these bubbles occupy the entire cross section of the column. We now lose the so called dense phase fluidization, where there is a clearly defined upper surface of the bed. At these high fluidization velocities the terminal velocity of some particles is reached and they are thrown high in the column. Intense mixing is occurring in the bed. The upper boundary of the bed becomes hard to distinguish and particles are carried out of the column. This regime is called the dilute-, or lean-phase fluidization region. These regimes are shown in Figure 1. For greater detail and terminology the reader is referred to numerous texts (1), (2), (3), (4).

Fluidized beds have many applications in modern

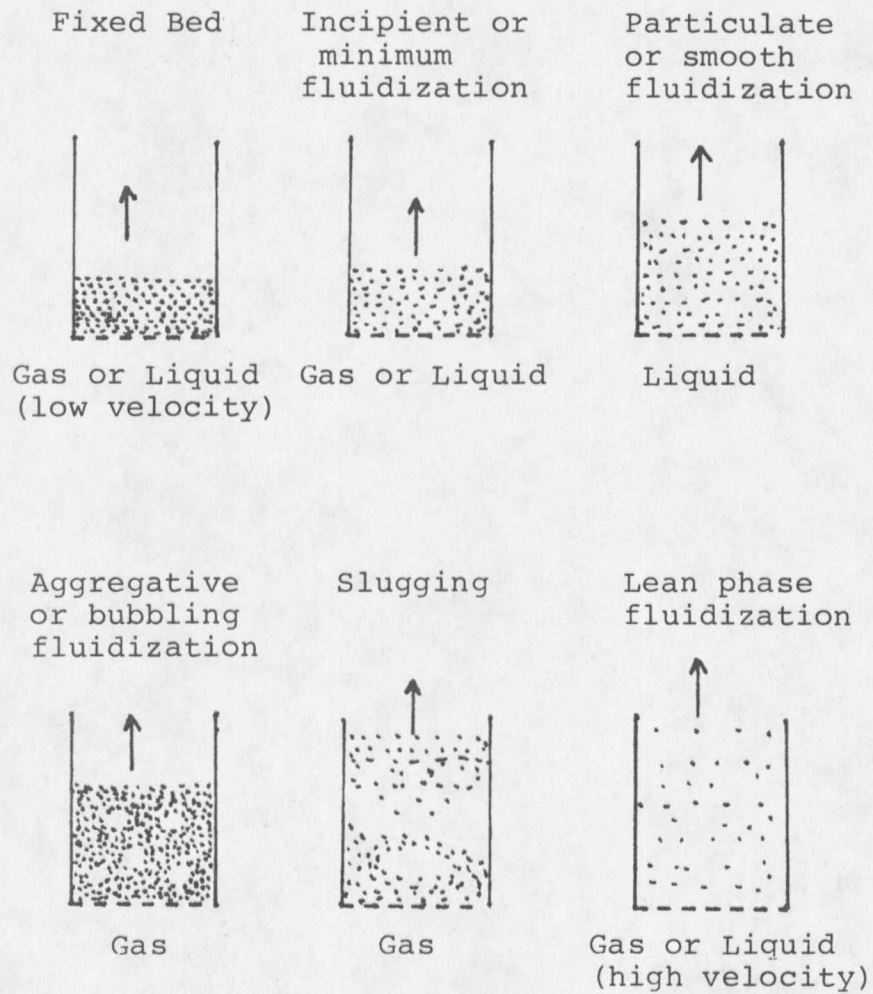


FIGURE 1. REGIMES OF FLUIDIZATION

industry. A recent development is the use of fluidized bed combustors to destroy toxic substances such as PCBs (5). New coal fired generators are also being designed using fluidized bed technology. Large particle limestone or dolomite fluidized bed combustors show great potential for reducing SO_2 emissions from a combustion chamber (6). There are currently a great number of commercial applications of fluidized bed technology (3).

The reason that fluidized beds have found such utility is that the operating characteristics are very well suited for certain processing needs. A number of the advantages are:

1. Extremely large areas of contact between solid and fluid which provide high overall rates of mass and heat transfer.
2. Easily handled fluidized solids.
3. High heat transfer rates to immersed surfaces.
4. High thermal inertia of solids.
5. Properties of a thermodynamic fluid with a low vapor pressure, even at high temperature.

This process system is not without certain disadvantages.

Drawbacks are:

1. Power is required for fluidization.
2. Particle attrition and agglomeration can be troublesome.
3. The operating rates are limited by the minimum fluidization velocity and the terminal velocity of the particles.
4. There is a certain range of particles that can be fluidized (.001-.25 Inch).
5. True counter current operation can not be obtained because of the high degree of mixing.
6. In certain applications thermal gradients may be necessary.
7. The scale up of fluidized beds is a difficult task without very accurate information and even then it is not without risk.

In spite of the various drawbacks there is considerable promise for new fluidized bed applications. This research is directed toward finding heat transfer coefficients that would be useful for someone designing a coal or other fossil fuel fired generator. I have investigated large, limestone particles and smaller

glass beads to determine heat transfer coefficients from vertically orientated, longitudinally finned heat transfer surfaces. The experimental variables were fin height, particle diameter, particle type, and fluidizing gas velocity. Experiments were also performed to determine local, along the fin, heat transfer coefficients from the largest fin height tube.

THEORY AND PREVIOUS RELATED RESEARCH

The theory and previous related research is divided into two parts. The first part deals with proposed heat transfer mechanisms. The second part deals with related research that has been done on heat transfer from immersed surfaces.

Mechanisms of fluidized bed heat transfer

Heat can be transferred from an immersed surface to a fluidized bed by a number of different mechanisms. Various forms of heat transport are radiation, gas convection and conduction, and particle convection. Depending upon the particulars of fluidization, one or more of these mechanisms can dominate, acting in a series or parallel manner. A fluidized bed is essentially isothermal in the bulk, because of good particle mixing, therefore the resistance to heat transfer is in a film surrounding the immersed surface. All modeling work is done in this boundary, which extends approximately one particle diameter out from the immersed heat transfer surface. The following discussion describes some of the types of models that have been proposed. Radiation is not considered in these

models, because most of the experimental work was done at low temperatures (less than 600°C) where its effect can be assumed to be negligible (10), and because in high temperature situations its effect can be estimated (1).

A number of phenomena can occur at the heat transfer surface. Bubbles contacting at or near the surface can change the heat transfer coefficient. Particles coming into contact with the surface can absorb heat and then by dispersing into the bulk, transport the heat into the bulk. The rate of heat transfer in this case will be dependent upon the residence time of the particles and other thermal-physical properties such as particle heat capacity and fluidizing medium thermal conductivity. Heat can also be transported by gas conduction from the surface. One of the first trends in modeling of the heat transfer coefficient from an immersed surface used a boundary layer approach. An illustration of this type of process is shown in Figure 2. In this model heat is conducted through a gas film. The heat transfer coefficient is defined as,

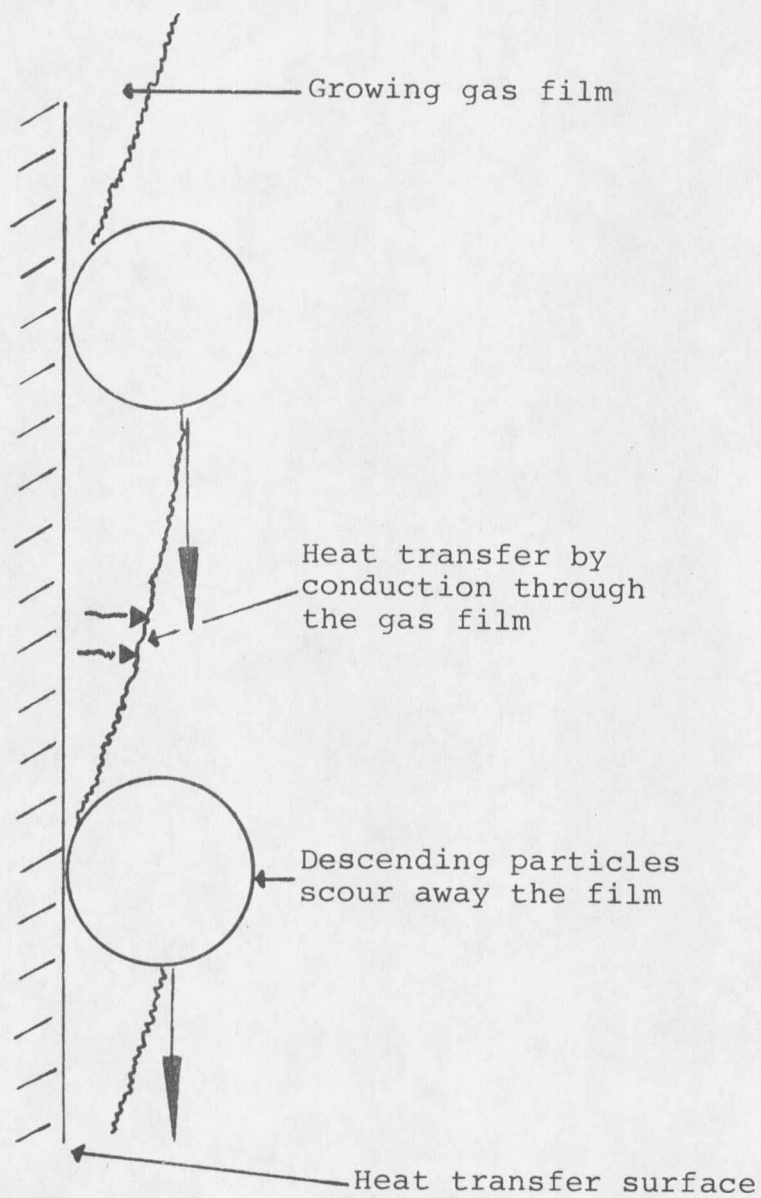


FIGURE 2. FILM MODEL FOR HEAT TRANSFER

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$$h = k_f / \delta'$$

In a fluidized bed the thickness of the boundary layer δ' depends not only on the velocity and properties of the fluid but also on the intensity of the motion of the solid particles, which "strip off" the boundary layer. In this model the maximum in the heat transfer coefficient (h) versus air mass velocity (G) curve is explained by an increased voidage at higher fluidizing mass velocities which results in fewer particles scrubbing the heat transfer surface. The principal defect of these types of boundary layer models is that they take no account of the influence on heat transfer of the thermal-physical properties of the solid material. Even subsequent attempts to include these factors have proven unsatisfactory. These correlations may be useful for assessing the convective component of heat transfer.

A second trend is to assume that the particles play a dominant part in the heat transfer process and also take into account the contribution of thermal conduction through the film layer near the surface. High thermal gradients between the surface and the particles gives rise to large h values. The thermal-physical properties

of the fluidized particles effect the rate of heat transfer in these situations. Various schemes are presented to model the transitory nature of particle contact and subsequent dispersion into the bed, based on measurable characteristics of the bed. In these models the maximum in the h vs. G curve is explained by the high temperature gradient dominance at low mass flow rates, increasing h with increasing G , and the increasing dominance of low particle density at higher flow rates, decreasing h with increasing G . An illustration of this process is shown in Figure 3.

A third trend of heat transfer that was introduced by Mickley and Fairbanks (21) is illustrated in Figure 4. In this process transient heat transfer by "packets" of solid particles (continuous phase) in contact with the heat transfer surface occurs. These packets are continuously being displaced from the heat transfer surface by gas bubbles (discrete phase). The heat transfer rate is dependent upon the rate at which the packets heat up and the frequency of their replacement from the surface. The maximum in the h vs. G curve is attributed to the simultaneous increase in the frequency

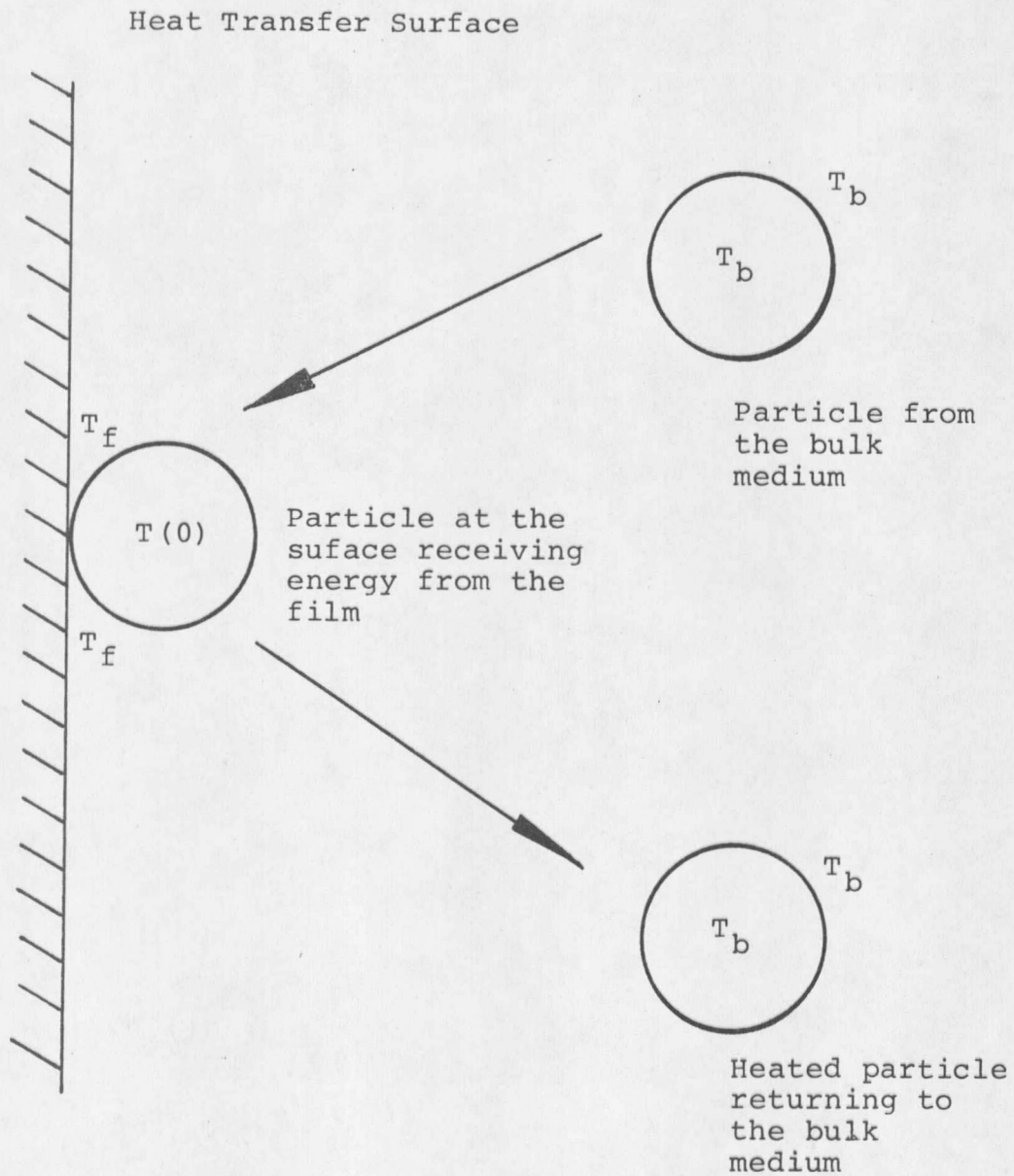


FIGURE 3. PARTICLE MODEL FOR HEAT TRANSFER

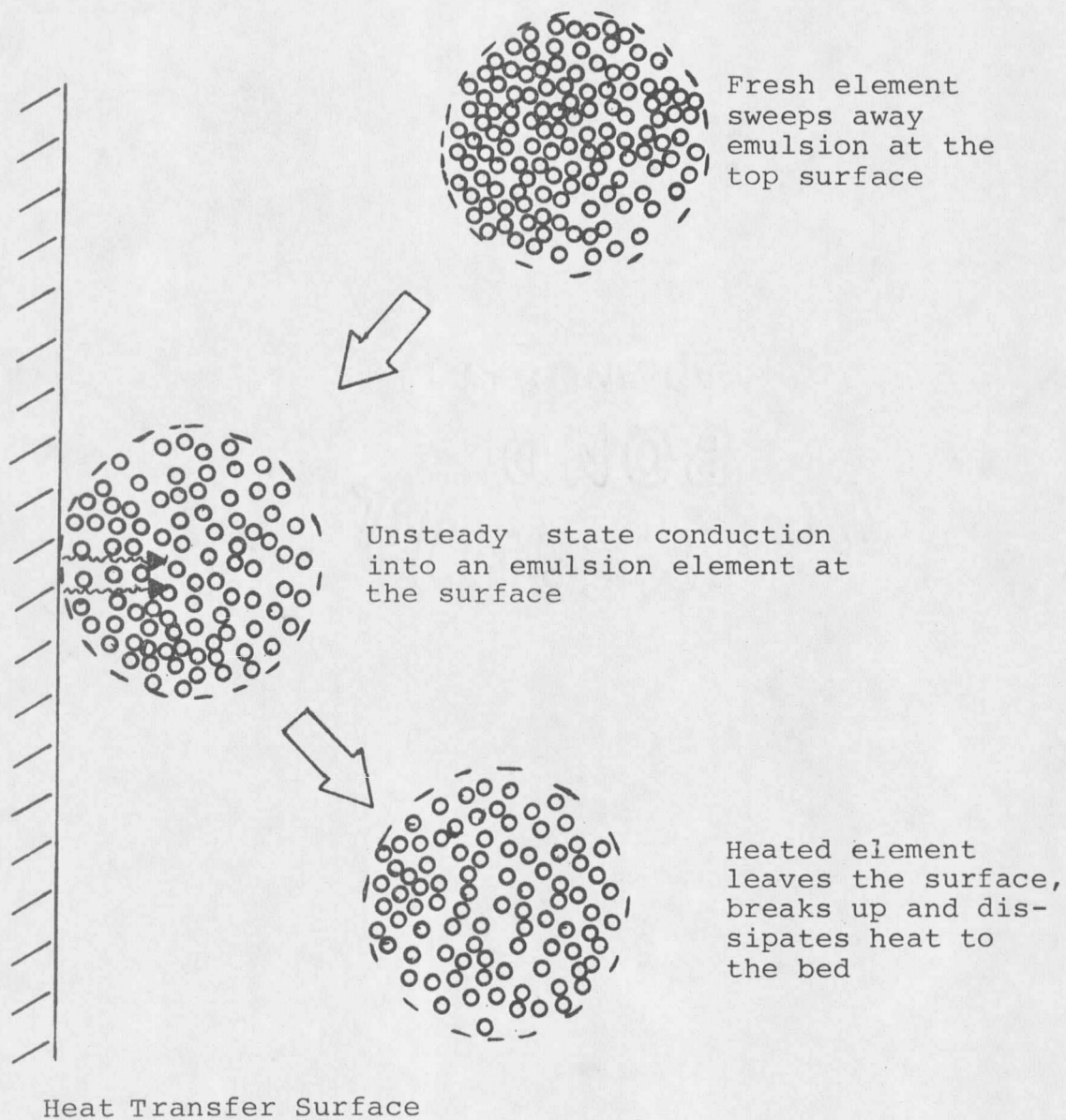


FIGURE 4. PACKET MODEL FOR HEAT TRANSFER

of particle packet replacement and the increase in the number of gas bubbles at the heat transfer surface. This heat transfer mechanism predominates in the dense phase at bed voidage $\epsilon < 0.7$ to 0.8 and breaks down upon further bed expansion.

When working with large particles, the gas convection component of heat transfer becomes important because large particles can not move fast enough to fully utilize their higher heat storage capacity. This transfer mechanism is similar to that of heat conduction through the boundary layer model. The total heat transfer in a large particle, bubbling or slugging fluidized bed is accomplished by a combination of packet heat transfer, gas percolating between the particles of the packet and the surface, and by gas bubbles or slugs. Modeling is therefore difficult because at least these three effects should be considered (14).

Previous Related Research

There has been some work done recently on fluidized bed heat transfer to beds of large particles.

With large particle fluidization the predominant mode of heat transfer is the gas convection mechanism (15,16,17). Gas convection models have recently been developed by Adams and Welty (18) and George et. al. (17). These models were developed to model heat transfer to immersed horizontal cylinders in large particle gas fluidized beds.

Adams and Weltly (18) investigated heat transfer from a horizontal tube in beds of large particles (3mm and 6mm mean diameter limestone). They developed computer programs to calculate the Nusselt number distributions around the cylinder surface and the total heat transfer coefficients. The assumptions of the model are expected to be valid for particle diameters of 2 to 3mm and larger.

Baskakov and Suprun (20) developed a dimensionless equation for the convective component of fluidized bed heat transfer from a vertical naphthalene surface. using a wide range of particle sizes (0.16, 0.32, and 0.5mm mean diameter corundum, and 0.8-1.25, 2.0-3.0, and 3 -5 mm mean diameter fireclay) they correlated data with the following equation:

$$Nu_{conv} = 0.0175 Ar^{0.46} Pr^{0.33}$$

The analogy between heat and mass transfer was used to separate the convective component of heat transfer from the other modes of transfer.

Botterill and Denloye (19) modeled heat transfer from a vertical cylinder in a 114 mm bed and correlated the Archimedes number with the quiescent bed heat transfer coefficient using the following equation:

$$\frac{h_{mf} \cdot d_p^{1/2}}{k_g} = 0.863 Ar^{0.39} \quad 10^3 < Ar < 2 \times 10^6$$

The value of the heat transfer coefficient was used as a measure of the gas convection heat transfer coefficient. Mean particle sizes ranged from 160 μm to 2400 μm , fluidized with air, argon, and carbon dioxide, operating at pressures up to 10 atm.

The packed-bed model of Gabor (25) can be used as an estimate of the gas convection portion of heat transfer if it is assumed that it is equal to the heat transfer before particle circulation begins, as suggested by

Botterill and Denloye (19). The model:

$$h_{gc} = \frac{(4 \cdot k_e \cdot \rho_g \cdot C_g \cdot \mu)^{1/2}}{(\pi L)^{1/2}}$$

showed good agreement with data up to a particle Reynolds number of about 250 (26).

Catipovic, et al (14) devised a model for large particle systems ($d_p > 1$ mm) assuming that the heat was transferred by the particles and by the gas percolating between the particles and the surface. They measured heat transfer in a cold bed with particles in the range 0.2 to 6.6 mm and obtained the following equation:

$$\frac{h d_p}{\lambda_g} = 6(1-\phi) + (0.0175 Ar^{0.46} Pr^{0.33})(1-\phi) + \frac{d_p}{D}(0.88 Re_{mf}^{0.5} + 0.0042 Re_{mf}) Pr^{0.33}$$

They suggest that expressions based upon an experimentally based mechanistic model have more predictive power and can be used with more confidence than empirical, dimensionless correlations.

Wender and Cooper (22) correlated data for vertical

tubes from a number of experimenters with the following equation:

$$\frac{h d_p}{k_g (1-\epsilon)} \left[\frac{k_g}{C_g \rho_g} \right]^{.43} = 3.5 \times 10^{-4} C_R \left[\frac{d_p G}{\mu_g} \right]^{.23} \left[\frac{C_p}{C_g} \right]^{.8} \left[\frac{\rho_p}{\rho_g} \right]^{.66}$$

$$10^{-2} < Re_p < 10^2$$

The factor C_R in the correlation is to allow for the effect of non-axial positioning of the tube.

Other correlations of data are provided by Gamson (23) and Vreedenberg (24) for vertical tubes.

Although detailed models which describe the means by which surface to bed heat transfer takes place have been proposed, they are of limited application because they require knowledge of parameters which are not generally available for practical situations. Similarly, although many empirical correlations relating bed to surface heat transfer coefficients to a range of operating variables have been proposed they are of doubtful validity because they can not make adequate allowance for the dynamic behavior of a gas-fluidized system. Zabrotsky has been very right to warn that any design correlation should only be used in cases very close to the original

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experimental conditions on which they were based (1).

EXPERIMENTAL PROGRAM

The objective of this experimental investigation was to gather information so that heat transfer coefficients could be calculated for several vertical, longitudinally finned tubes. The parameters involved in this experimental program can be divided into three groups. The first group involves parameters dealing with the fluidizing components, which are the particle size and type, and the physical properties of the air, the density, temperature, thermal conductivity, heat capacity, and viscosity. The second group of parameters are the operating condition parameters, which are the fluidizing medium mass velocity and the static height of the solid particles in the bed. The third set of parameters deal with the physical scale of the column and placement of the internal heat transfer surface. The four variables investigated during this experimental program were particle type, particle diameter, air mass velocity, and fin height. Inherent parameters of the particle type are heat capacity of the particles, shape factor of the particles, density of the particles, and thermal conductivity of the particles.

Air, with an inlet temperature in the range of 85 °F to 100 °F, was used as the fluidizing medium. The physical properties of the inlet air were considered constant. Spherical glass beads and irregular limestone rocks were used as the solid particles. The density of the glass beads is approximately 155 lb/ft³ and the density of the limestone particles is approximately 139 lb/ft³. Table I gives the range of the experimental variables.

TABLE I. RANGE OF EXPERIMENTAL VARIABLES

Variables	Range
Particle Diameter	0.0625 (In.) Limestone 0.0109 (In.) Glass Beads
Air Mass Velocity	94.0 to 858.0 lb/hr-ft ²
Fin Height	0.0, 0.25, 0.375, 0.5, 0.75 (In.)
Particle Types	Limestone, Glass

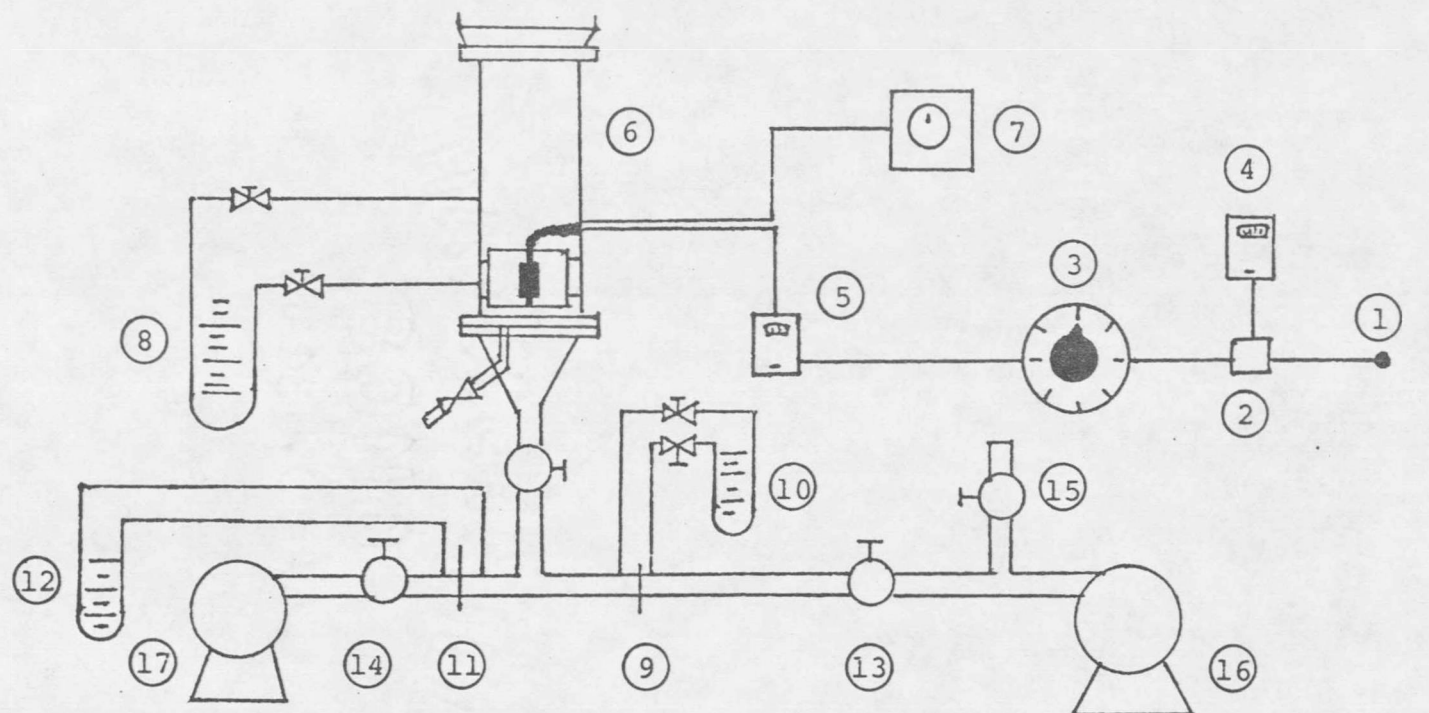
EXPERIMENTAL EQUIPMENT

Fluidizing System

The main parts of the fluidizing system are the column, glass and limestone particles, distributor plate, funnel and the two air blowers. A schematic drawing of the experimental equipment is shown in Figure 5.

The column was cylindrical and constructed of clear plexiglass that was 0.25 inches thick. The column was 13.5 inches inside diameter and 59 inches in height. A 0.75 inch thick flange was connected to the top and the bottom of the column. One access port, 4 inches in diameter was located with its center 6 inches from the bottom of the column. This access port was not used during the experimental runs and was plugged with a 4 inch diameter wooden piece to minimize the effect of this port on bed fluid dynamics. The column was supported by a wooden frame which was bolted to the floor.

Two sizes of particles were used in this investigation. One being glass beads and the second being limestone particles. The sizes and distributions of



(1) Power Supply; (2) Switch Box; (3) Powerstat; (4) High Limit Temperature Controller; (5) Wattmeter; (6) Plexiglass Column; (7) Potentiometer; (8) Bed Manometer; (9) Main Orifice; (10) Main Orifice Manometer; (11) Secondary Orifice; (12) Secondary Orifice Manometer; (13) Main Air Valve; (14) Secondary Air Valve; (15) Bypass Valve; (16) Main Air Blower; (17) Secondary Air Blower.

FIGURE 5. SCHEMATIC VIEW OF THE EXPERIMENTAL SYSTEM

the glass beads were determined by Everly (7). The size of the limestone particles was determined by measurement with a micro caliper. The particle size data is listed in Table II.

TABLE II. PARTICLE SIZES

Particle Type	Mean Particle Diameter (Inch)	Sieve Size Range (Inch)
Limestone	0.0109	0.0098-0.0160
Glass Beads	0.0625	0.0394-0.0787

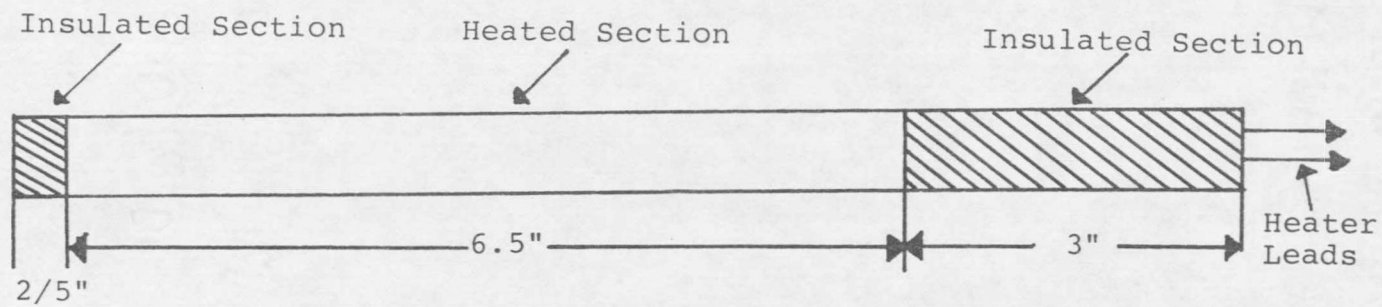
The distributor plate consisted of two layers of a lightweight cotton cloth sandwiched between two layers of 100 mesh stainless steel wire cloth which was placed between two pieces of 0.03125 inch thick steel perforated plates. The perforations were .25 inches in diameter and spaced .5 inches apart center to center distance. The funnel connecting the bottom of the column to the air flow line was 13.5 inches in diameter at the top and 2 inches in diameter at the

bottom. It was 12 inches high and was made of 16 gauge galvanized steel. At the lower end a spout 4 inches long and 2 inches in diameter was attached. The distributor plate was fitted with two rubber gaskets and together with the funnel section and a 13-1/2 inch diameter wooden reinforcement ring was bolted to the bottom of the column. A 1 inch diameter drain pipe was connected between a hole in the distributor plate and a hole in the side of the funnel. The drain pipe was fitted with a gate valve. The two blowers were connected in parallel to the fluidized bed. The first blower was a size 4L Sutorbilt Blower driven by a 3 H.P. electrical motor. The blower fed into a 2 inch, schedule 40 pipe which was connected to a T connection with a flexible rubber hose which linked the T connection and the bottom of the funnel section of the bed. The air flow was measured by using a 1.5 orifice with vena contracts taps and a water filled manometer. The second blower was a Sutorbilt Blower driven by a 7.5 H.P. electrical motor. The second blower fed into a 2.5 inch, schedule 40 steel pipe before reaching the T connection that

is connected to the bottom of the column. The air flow rate was measured using a 1.5 inch orifice with vena contracts taps and a water filled manometer. Although there were gate valves in each of the blower feed lines, the flow rate was controlled by adjusting the gate valve for the 2 inch bypass line of the larger, 7.5 H.P. blower motor, air supply line. For fluidization of the small particles, only the larger 7.5 H.P. blower was used. The smaller blower was disconnected and the line plugged. This was possible because smaller air flow rates were required.

Heater and Finned Tube Assemblies

Watlow Firerod Cartridge Heaters were used as the heat source in this investigation. As shown in Figure 6, each cartridge was 10 inches long, comprised of a heated section 6.5 inches long and two insulated ends, one 3 inches long and the other 0.5 inches long. The heater cartridges were 5/8 inches in diameter. The finned tubes were 12 inches long and the outside diameter was 1 inch. The iron fin material was removed from all but 6.5 inches of the 12 inch long tube. The



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FIGURE 6. DETAILS OF A CARTRIDGE HEATER

tubes were bared 3 inches in from one end and 2.5 inches in from the other end. The heaters were wrapped with 3M metal tape and coated with copper antiseize compound. This treatment provided for maximum contact between the heater cartridge and the finned portion of the tube. The heaters were then inserted into the finned tubes so that the 3 inch insulated end of the heater cartridge corresponded with the 3 inch bared end of the finned tube. One set screw in this end held the cartridge in place. The opposite end was sealed with permatex sealant. The tube assembly was then mounted onto a frame. This assembly is shown in Figure 7. The frame was then inserted into the opened bottom of the column and bolted to the insides of the column. The wires protruding from the top of the assembly were then run through a 8 inch long piece of conduit to a hole located 20 inches above the distributor plate. In this manner all of the tubes were mounted and investigated. A total of five different tubes were investigated. The tube surface areas and fin heights are given in Table III. These tubes were supplied by Tex Fin Company of Houston, Texas. The

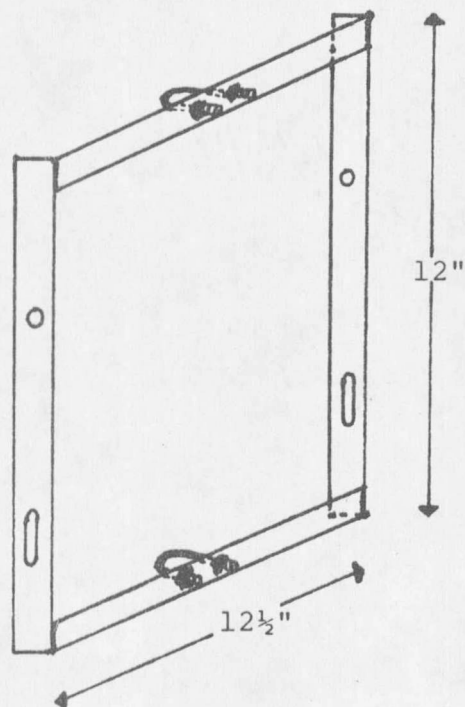
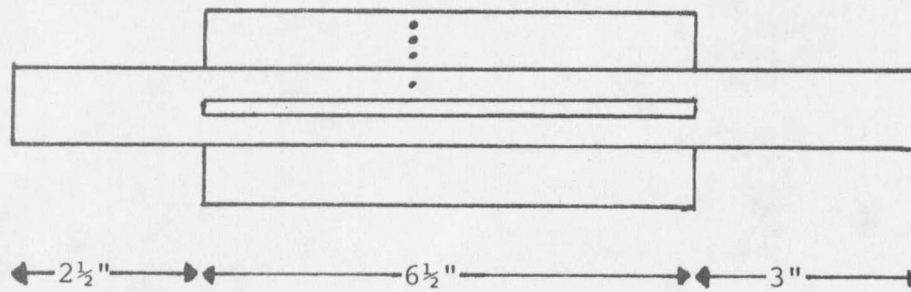


FIGURE 7. DETAILED VIEW OF A FINNED TUBE AND THE FRAME USED TO HOLD THE TUBE IN PLACE

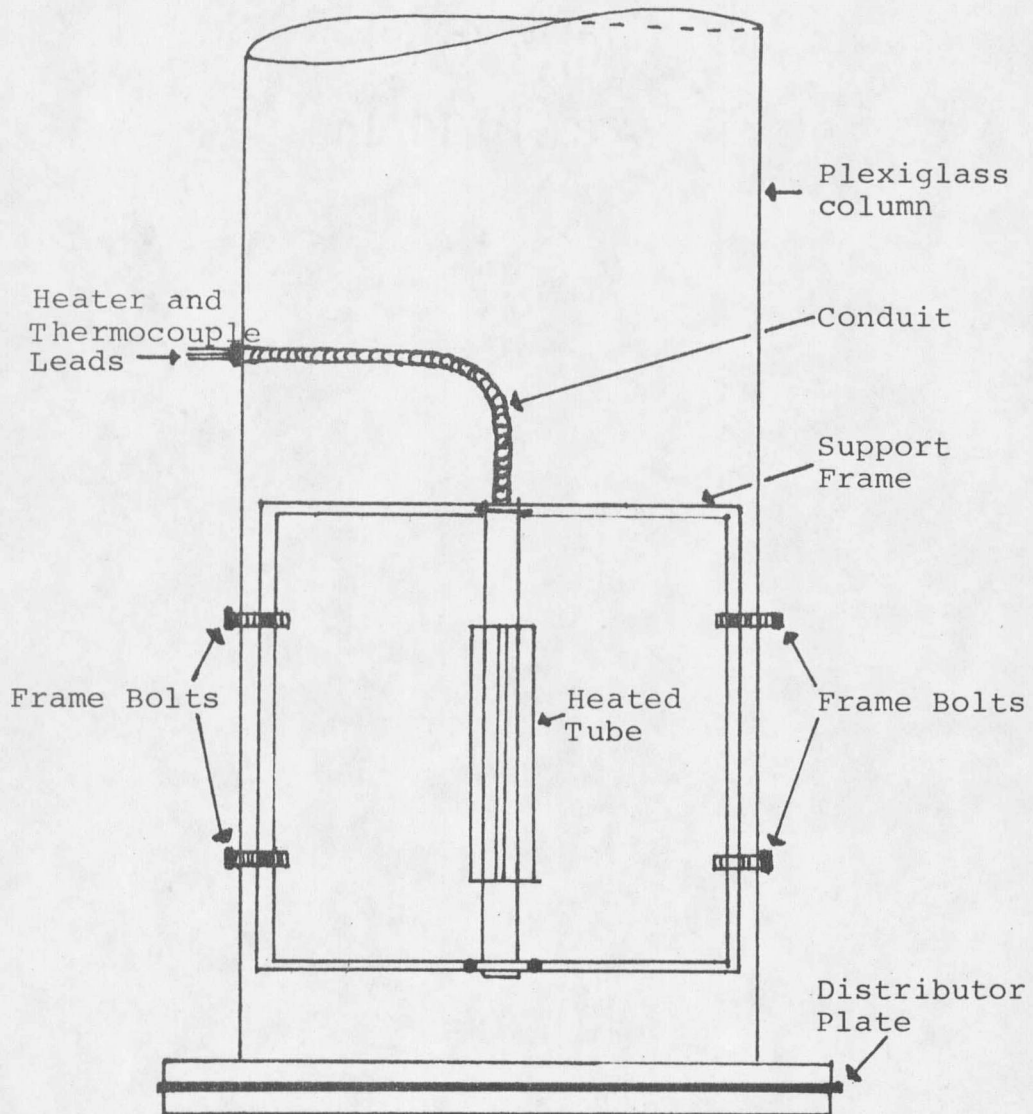


FIGURE 8. DETAILED VIEW OF THE COLUMN ASSEMBLY

completed column is shown in Figure 8.

TABLE III. FINNED TUBE DIMENSIONS

Tube No.	Tube OD. (in)	Fins per Tube	Fin Height (in.)	Fin Thickness (in.)	Total Surface Area (ft ²)
1	1.	0	0.	0.	.2836
2	1.	12	.25	.04	.4126
3	1.	12	.375	.04	.5481
4	1.	12	.5	.04	.6835
5	1.	12	.75	.04	.9543

Electrical System

The tube surface temperature was measured by one thermocouple located in the bare area between fins at a location midway between the ends of the fin material. The thermocouple lead was then run along the tube to the top of the tube and into the conduit leading to the hole in the side of the column. This thermocouple was then attached to a Varian G-2000 chart recorder and a Wheelco Instruments Model 330 Potentiometer. For temperatures lower than 600° F, the Wheelco Potentiometer was disconnected and a direct reading Model 156X15-P Brown Potentiometer was used. One thermocouple was used to measure the bed temperature. It was located in a thermowell that was 12 inches from the distributor plate and extended 3 inches in the bed. This thermocouple was connected to the Brown Potentiometer for temperature measurement. A third thermocouple was placed into a thermowell 8 inches from the distributor plate and extending 3 inches into the bed. This thermocouple was attached to a Fenwall Model 524 high limit controller to protect the plexiglass column from high bed temperatures which would

damage the column. A fourth thermocouple was inserted into a thermowell that was located 15 inches from the distributor plate and extended 3 inches into the bed. This thermocouple was attached to the Varian G-2000 chart recorder to provide an indication of how fast the bed temperature was approaching a steady value.

The electrical heater was connected to a Simpson Model 390 Wattmeter, which was connected to a Powerstat. A Fenwall Model 524 high limit controller was connected between the Powerstat and the 110 volt power box. In case of high bed temperature, this controller would turn off the power to the cartridge heater in the bed.

Three thermocouples were attached to the fins of the highest fin height tube (Tube #5) so that temperatures of the fin at discrete points could be measured. The thermocouples were attached at the longitudinal center of the fin material, $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$ of the fin height out from the base of the tube, each on a different fin. This was necessary because of the difficult nature of the soldering process. The leads exited the column through the same conduit used by the power leads. Temperatures were measured using a Brown Potentiometer.

EXPERIMENTAL PROCEDURE

Minimum Fluidization Determination

The first quantity obtained during the trial stages of this experimental work was the minimum fluidizing velocity for the limestone particles used in this research. The particles, mean diameter approximately 1.6 mm (.063 Inch), were poured into the top of the column until a static height of 18 inches above the distributor plate was obtained. The main and secondary blowers were then turned on and adjusted so that the bed was barely fluidized. The heater was then turned on and adjusted so that the temperature on the tube surface did not rise above 1000^o F. The bed was operated for 3-4 hours before normal bed temperatures were obtained. Then the main bypass valve was manipulated to control the air flow through the bed. Water filled manometers were used to measure pressure drops across the bed, main air flow line orifice, and the secondary air flow line orifice. Data was obtained that allowed a graph of pressure drop across the bed vs. mass air flow rate to be constructed. From this graph the minimum fluidization

velocity was obtained. The minimum fluidization velocity for the glass beads was obtained from Yurich's thesis (12).

Procedure for a typical run

The finned tube to be investigated was chosen, the tube was cut to a length of 12 inches, and the fin portion 3 inches in from one end and 2.5 inches in from the other end was removed. A typical finned tube is shown in Figure 7. A thermocouple was then silver soldered to the mid point between adjacent fins and the ends of the finned section. Aluminum tape and copper anti-sieze compound were put on a heater cartridge and this cartridge was inserted in the finned tube. The heated section of the cartridge was situated radially under the finned section of the tube. This finned tube was then clamped to a frame. The frame was placed into the column through the bottom and attached to the column walls by four bolts. The thermocouple and power leads were run through a piece of conduit from the top of the finned tube to a hole 23 inches above the distributor plate. The bed was

then bolted together at the bottom and the particles were added to the column until a static height of 18 inches had been obtained. The main and secondary blowers were turned on and a valve in the bypass line was used to adjust the flow rate to the desired fluidization velocity. The power to the heater was adjusted to 1000 watts. The column was allowed to operate for 4-5 hours to reach steady state operating conditions. The temperatures of the tube surface and bed were then recorded. The pressure drop across the bed, main line orifice, and secondary line orifice were recorded, and an estimate of the expanded bed height was made. The flow rate of the bypass valve was then adjusted to increase the flow through the bed and the column was operated for 1 hour for the limestone particles and 2 hours for the glass beads to reach a new steady state. Readings were again taken and the flow rate through the bed increased. This sequence was repeated until maximum flow through the column was reached.

The power to the heater was then turned off and then the blowers were turned off. The particles were

then removed through the 1 inch drain pipe, the bottom of the column was taken apart, and the frame was removed. A new tube was then prepared, placed on the frame, and inserted into the column. The column was then sealed and filled with particles in preparation for another run. This procedure was continued until all tubes and particle sizes had been investigated. Data were recorded in a form shown in Figure 29.

DEVELOPMENT OF THEORETICAL MODELS FOR LONGITUDINAL FINNED TUBES

This section is divided into three parts. The first part is the mathematical modeling of the single thermocouple finned tubes. The second part is the mathematical modeling of the multiple thermocouple tube. The third part is the presentation of the correlating equation.

Modeling of the single thermocouple finned tubes

In the cases where only one thermocouple is attached to a finned tube, it can not be assumed that the temperature along the fins are uniform. By calculating a heat transfer coefficient which takes into account the temperature profile along the fin, I can make this coefficient independent of the fin material thermoconductivity. In this section it is assumed that the heat transfer coefficient is constant for all surfaces at a given air flow rate.

Figure 9 shows a fin and the coordinate system used to analyze this problem. Heat enters the left end of the differential section. During steady state operation there is no accumulation of energy. All the

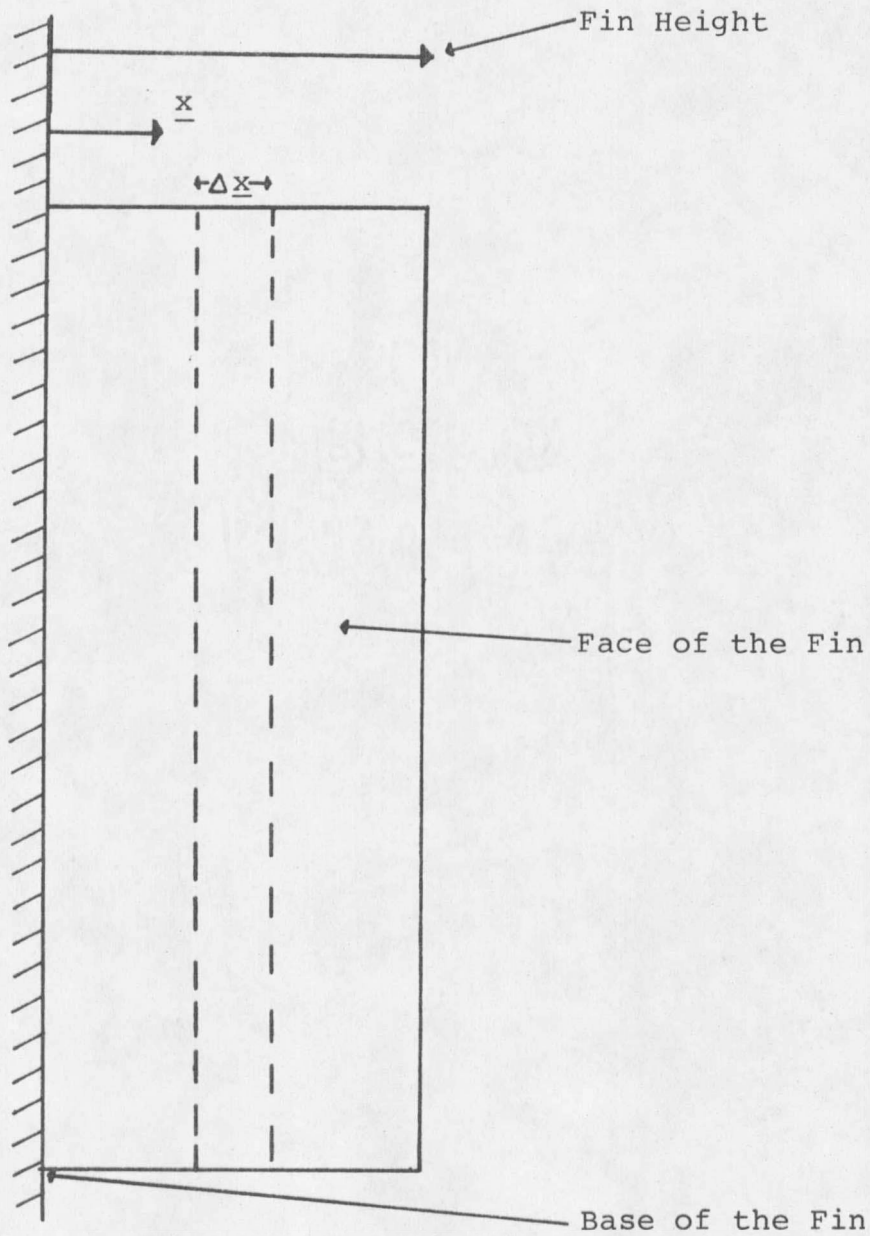


FIGURE 9. PROPOSED MODEL OF THE LONGITUDINAL FIN

energy therefore leaves this element by conduction out the right end or by convection from the faces of the element. In mathematical terms this element can be described as shown below.

$$-kB\delta \left. \frac{dT}{dx} \right|_{\underline{x}} + kB\delta \left. \frac{dT}{dx} \right|_{\underline{x}+\underline{\Delta x}} - 2B\Delta x h (T - T_b) = 0$$

-Rate of heat flow by con- duction into the element at point <u>x</u>	+	Rate of heat flow out of the element at point <u>x + Δx</u>	-	Rate of heat flow = 0 by convection from the surface between points <u>x</u> and <u>x + Δx</u>
---	---	---	---	--

where,

k = Thermal conductivity of the fin material,
BTU-ft/hr-ft²-°F

B = Longitudinal length of the fin, ft

δ = Thickness of the fin, ft

T = Fin temperature, °F

T_b = Bed temperature, °F

Δx = Size of the differential element, ft

h = Heat transfer coefficient, Btu/hr-ft²-°F

In this analysis it is assumed that end effects are negligible.

Dividing through by Δx and taking the limit as

$\Delta x \rightarrow 0$ and simplifying, the following differential equation for the temperature profile along the fin is obtained.

$$\frac{d^2 T}{dx^2} - \frac{2h}{k\delta} (T - T_b) = 0$$

Introducing the dimensionless variables,

$$\theta = \frac{T - T_b}{T_w - T_b} ; \quad x = \frac{x}{L} ; \quad m^2 = \frac{2hL^2}{k\delta}$$

where,

L = Fin Height, Ft

and simplifying yields the following differential equation,

$$\frac{d^2 \theta}{dx^2} - m^2 \theta = 0$$

The solution of this equation is the well know form shown below,

$$\theta = C_1 \sinh(mx) + C_2 \cosh(mx)$$

The arbitrary constants can be solved for using two boundary conditions. These boundary conditions are,

a) $\theta = 1, \quad @ \ x = 0$

b) $-\frac{k}{L} \frac{d\theta}{dx} \Big|_{x=1} = h \theta \Big|_{x=1} \quad @ \ x = 1$

Applying these two boundary conditions to equation (1) yields the following values for the constants C_1 and C_2 .

$$C_1 = - \frac{(h \cosh(m) + (km/L) \sinh(m))}{(h \sinh(m) + (km/L) \cosh(m))}$$

$$C_2 = 1$$

Substituting these equations for the arbitrary constants back into equation (1) yields the following equation describing the temperature profile along the fin.

$$\theta = \cosh(mx) - \frac{(h \cdot \cosh(m) + (km/L) \cdot \sinh(m))}{(h \cdot \sinh(m) + (km/L) \cdot \cosh(m))} \cdot \sinh(mx) \quad 2.$$

Using the Fourier law of heat conduction I can now calculate the amount of heat transferred to each fin.

$$Q_{Fin} = - \frac{kB\delta (T_w - T_b)}{L} \frac{d\theta}{dx} \Big|_{x=0} \quad 3.$$

Taking the first derivative of equation (2) with respect to x and substituting this into equation (3)

yields an expression for the amount of heat going into each fin.

$$\left. \frac{d\theta}{dx} \right|_{x=0} = - \frac{m \cdot (h \cdot \cosh(m) + (km/L) \cdot \sinh(m))}{(h \cdot \sinh(m) + (km/L) \cdot \cosh(m))}$$

$$Q_{Fin} = \frac{kB \delta (T_w - T_b)}{L} \frac{m(h \cosh(m) + (km/L) \sinh(m))}{(h \sinh(m) + (km/L) \cosh(m))} \quad 4.$$

An overall energy balance yields the following equation (5),

$$Q_{Total} = N_{Fin} Q_{Fin} + Q_{Base} \quad 5.$$

where,

$$Q_{Base} = hA_b (T_w - T_b) \quad 6.$$

and

$$A_b = \text{Area of the base} - \text{Area taken up by the base of the fins.}$$

$$A_b = \pi DB - N_{Fin} \delta B$$

Q_{Total} is a measured quantity. Equations (4) and (6) can be substituted into equation (5) to yield equation (7).

$$\frac{Q_{\text{Total}}}{(T_w - T_b)} = \frac{N_{\text{Fin}} k B \delta m}{L} \frac{(h \cdot \cosh(m) + (km/L) \cdot \sinh(m))}{(h \cdot \sinh(m) + (km/L) \cdot \cosh(m))} + hB(\pi D - N_{\text{Fin}} \delta) \quad 7.$$

This equation can now be solved to determine values of h vs. $Q_{\text{Total}}/(T_w - T_b)$. A computer program was written to find the heat transfer coefficients for specific data points. This program is shown in Figure 30.

This program uses a Wegstein convergence routine to find the value of the heat transfer coefficient.

Output from this computer program for all runs made is shown in Figure 32 through 41. This program uses a data file, as shown in Figure 31, to calculate the Nusselt number, Prandtl number, G/GMF , Reynolds number, and the Archimedes number for all data points.

A second computer program was written to illustrate the relationship between the heat transfer coefficient and the quantity $Q_{\text{Total}}/(T_s - T_b)$. Each finned tube yields a different graph. These graphs are shown in Figures 10 and 11. With a value of $Q_{\text{Total}}/(T_s - T_b)$ it is very simple to obtain a value of h .

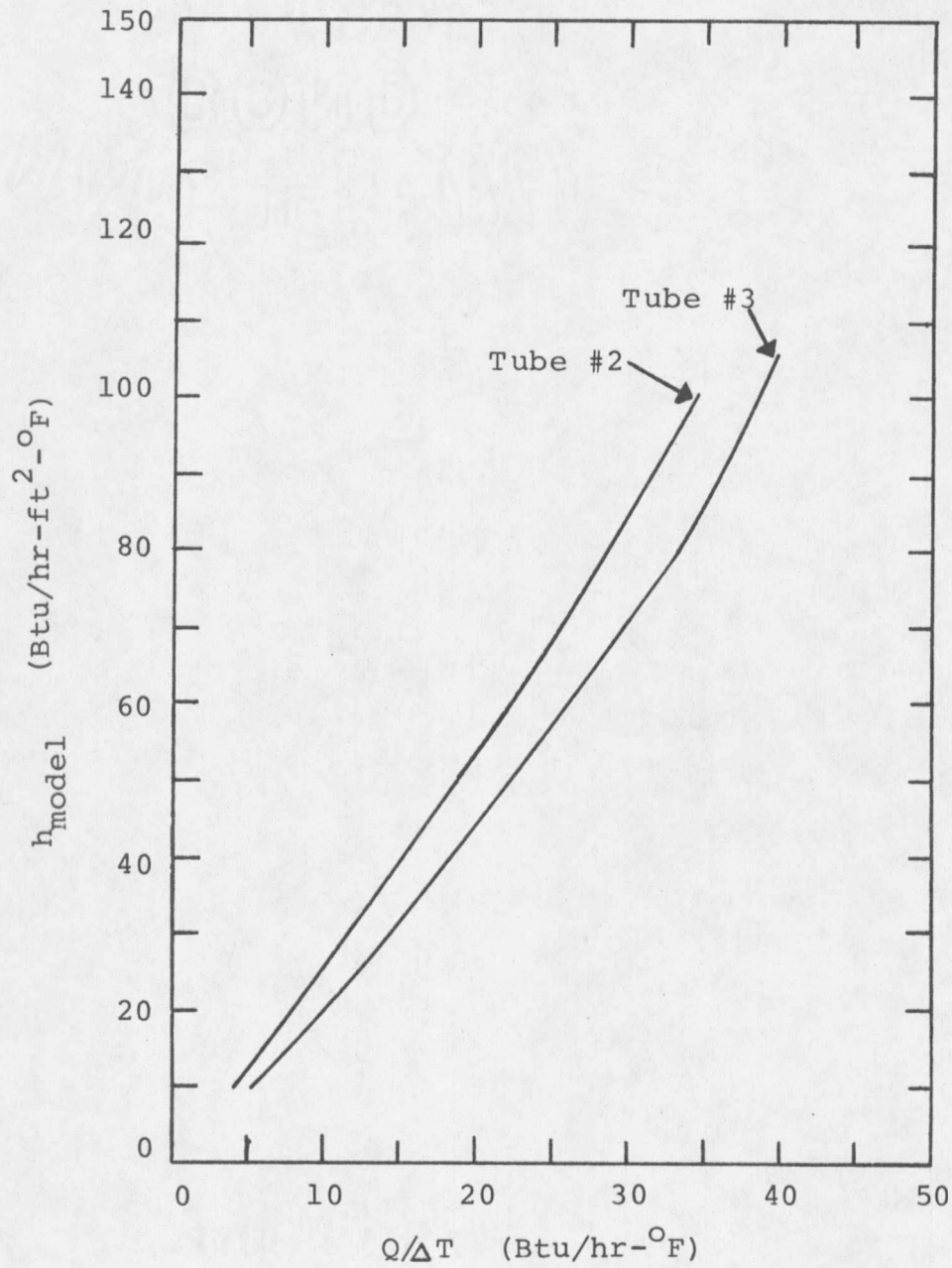


FIGURE 10. ANALYTICAL SOLUTION FOR $Q/\Delta T$, TUBES 2 AND 3.

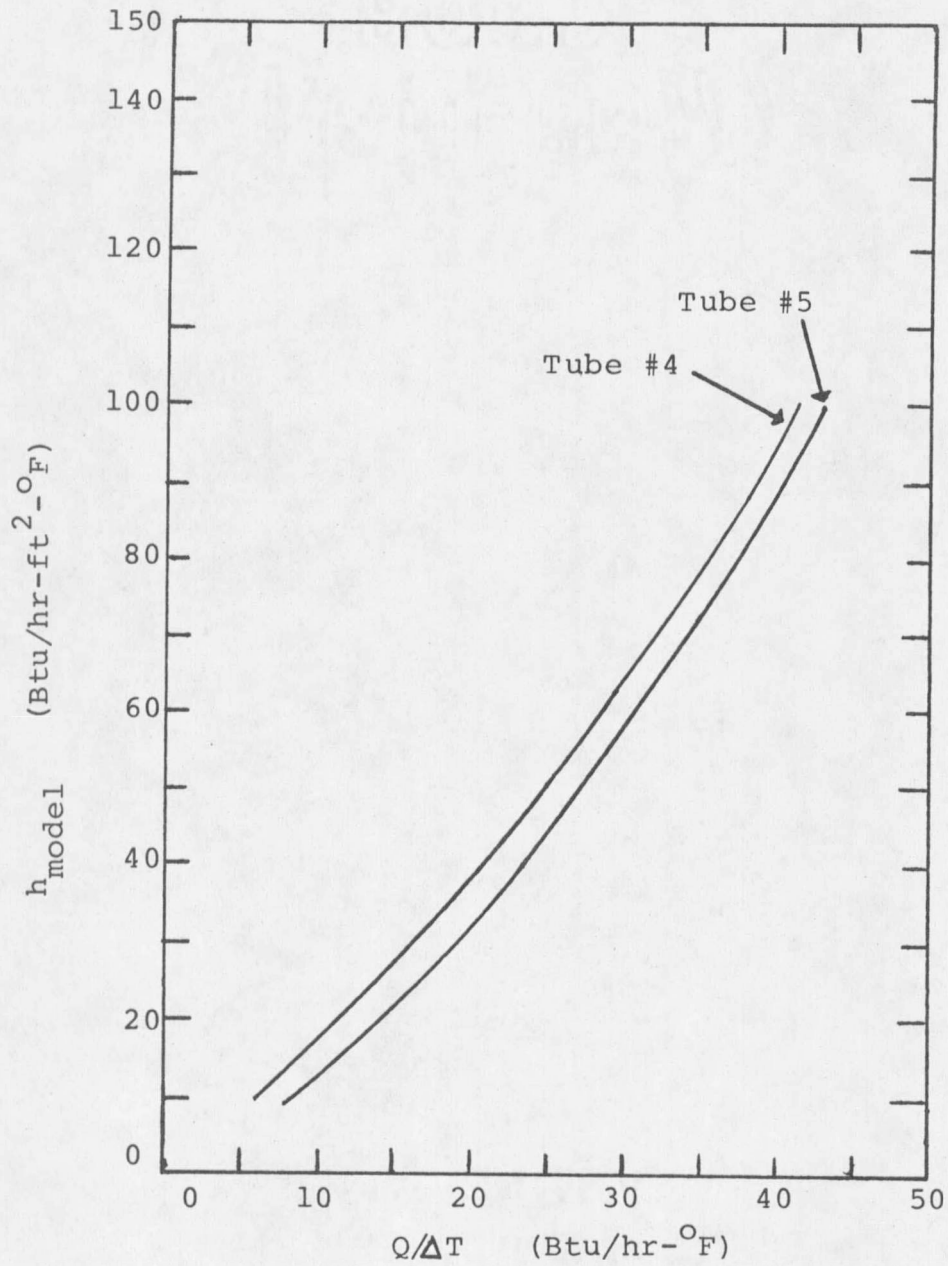


FIGURE 11. ANALYTICAL SOLUTION FOR $Q/\Delta T$ TUBES 4 AND 5.

Modeling of the Multiple Thermocouple Tube

This section has been placed in Appendix A, because the information obtained was of questionable reliability. The information was interesting however, and therefore was included in this thesis.

Development of the Correlating Equation

Correlating equations were developed using dimensionless quantities. These dimensionless quantities relate to certain characteristic variables in the system. The computer program shown in Figures 42 and 43 was written to fit the constants in the equation to the experimental data. The data files used are shown in Figure 44. The program used a simple search subroutine to find the minimum of the difference squared of the experimental and calculated Nusselt numbers.

The dimensionless quantities investigated were:

Re_p	Particle Reynolds Number
G/G_{mf}	Ratio of the gas flow rate to the gas flow rate at minimum fluidization
$(1 - \frac{N_{fin}}{(d_t + L)})$	Ratio of the open space volume to the total volume occupied by the finned tube
Ar	Archimedes number

(1 - Void fraction)

$(1 + \frac{\text{Fin height}}{.0625})$ Fin height ratio

$(.625/\text{Fin height})$ Ratio of the maximum fin
height, .625 (In), to the
actual fin height

These variables were arranged in equations of the following general forms:

$$\text{Nu} = a B^b C^c D^d E^e$$

$$\text{Nu} = a + b C^c D^d E^e$$

where B,C,D, and E represent the dimensionless groups and a,b,c,d, and e represent the quantities which were found using the curve fit program. In this manner equations were constructed, placed into the program, and then evaluated. Lines 46 and 47 of the program shown in Figure 43 contain the objective function that will be evaluated when the program is run. Objective functions are inserted into this program before each new trial. This sequence is continued until all of the objective functions have been evaluated and the best one has been found.

RESULTS AND DISCUSSION

The results and discussion chapter is divided into four sections. The first section shows data obtained for the bare tube, glass bead run compared to values obtained from the literature. The second section discusses the affect of system parameters on the heat transfer coefficient. The third section shows the results of curve fitting the experimental data. The fourth section covers the evaluation of the performance of these finned tubes.

Literature Evaluation of Bare Tube Data

Chen and Withers (27) determined heat transfer coefficients for plain, vertical tubes in a bed of .01 inch diameter glass beads and then compared the results with the correlations of Wender and Cooper, and Genetti, et al. The glass bead, bare tube data of this experiment is compared to their data and the correlations of Wender and Cooper, and Genetti, et al in Figure 12.

Figure 12 shows that the bare tube data of this experiment is between the correlations of Wender and Cooper, and Genetti, et al. It is also near the data of Chen and Withers. Chen and Withers would be expected to have higher heat transfer coefficients because the glass

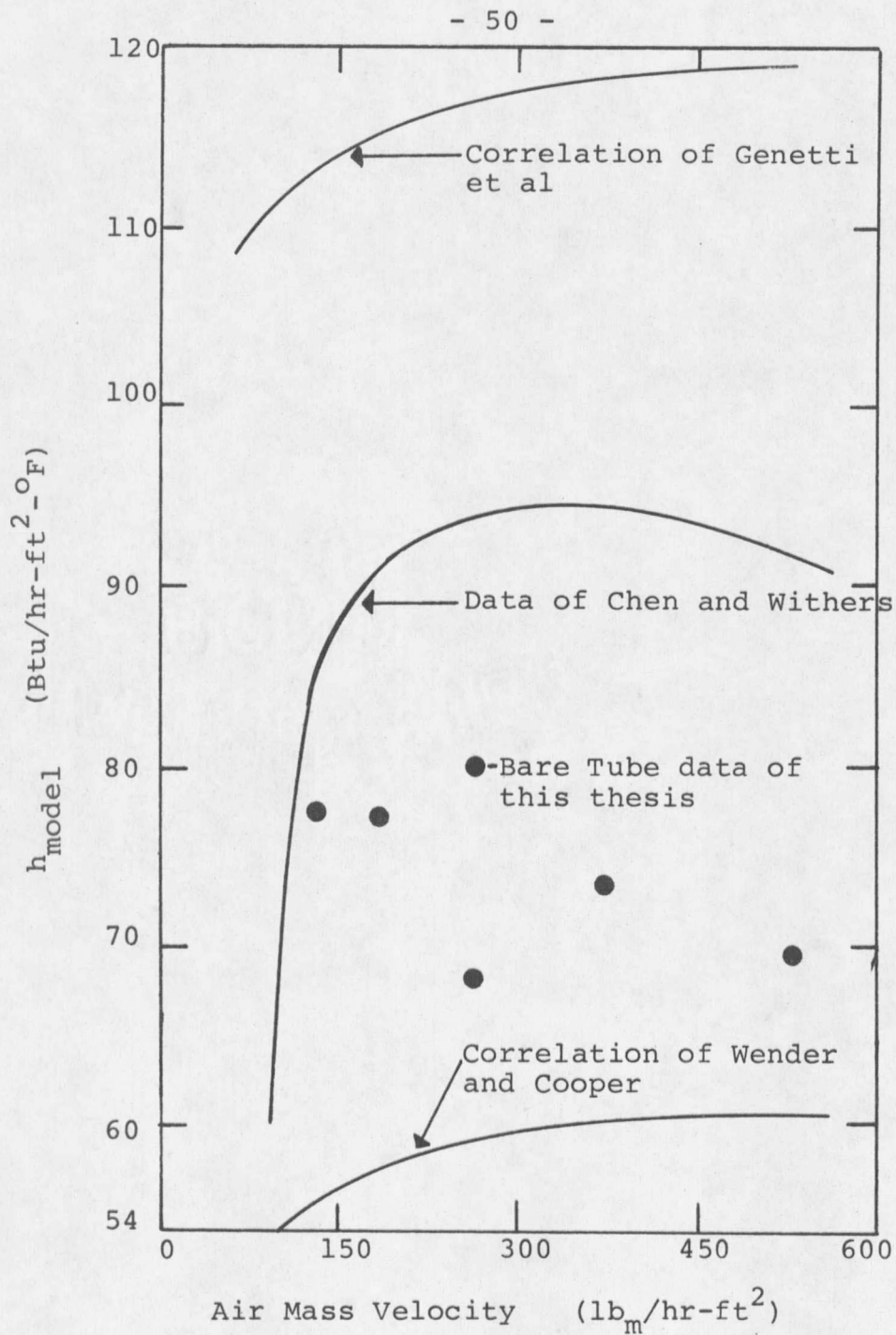


FIGURE 12. BARE TUBE DATA COMPARISON

beads used in their research were smaller than those that were used in this research.

These results lead to some confidence about the validity of heat transfer coefficients calculated in this thesis.

Longitudinal Finned Tube System

Single, longitudinally finned tubes were used to investigate the effects of fin height, particle size, and gas mass velocity.

The effect of increasing fin height was generally a decrease in the heat transfer coefficient. This is shown in Figures 13 and 14, where it can be seen that the glass beads are more predictable in this trend than are the limestone particles. The limestone particles exhibit cross over between the heat transfer coefficients of the 2nd and 3rd tubes and also between the 4th and 5th tubes. In both cases the bare tubes give higher heat transfer coefficients than the finned tubes under similar conditions.

Smaller particles yield better heat transfer coefficients for a given surface and flow rate. Figures 15, 16, 17, 18, and 19 show that the glass beads have higher heat transfer coefficients than the limestone particles

in the region where the gas flow rates are similar. It should be noted that the flow rate through the glass beads is reaching terminal velocity near the minimum fluidization velocity of the larger, limestone particles.

The heat transfer coefficient generally increases with mass flow rate until a maximum is reached. The finned tube, glass particle data given in Figures 16, 17, 18, and 19 shows an increase in the heat transfer coefficient until a leveling off occurs at higher mass velocities of the fluidizing fluid. The bare tube, glass particle data shows a decline in the heat transfer coefficient with an increase in the gas flow rate. This result indicates that the heat transfer coefficient has peaked at a flow rate near minimum fluidization. Figures 16, 17, 18, and 19 show the finned tube, limestone particle data exhibiting the same type results as the glass particles. The heat transfer coefficient increases with increasing gas flow. In this case, however, the heat transfer coefficient does reach a maximum for tubes 2, 3, and 4. This peak is predictable because at high mass flow rates the bed expands, thus decreasing particle scouring action, while not increasing the

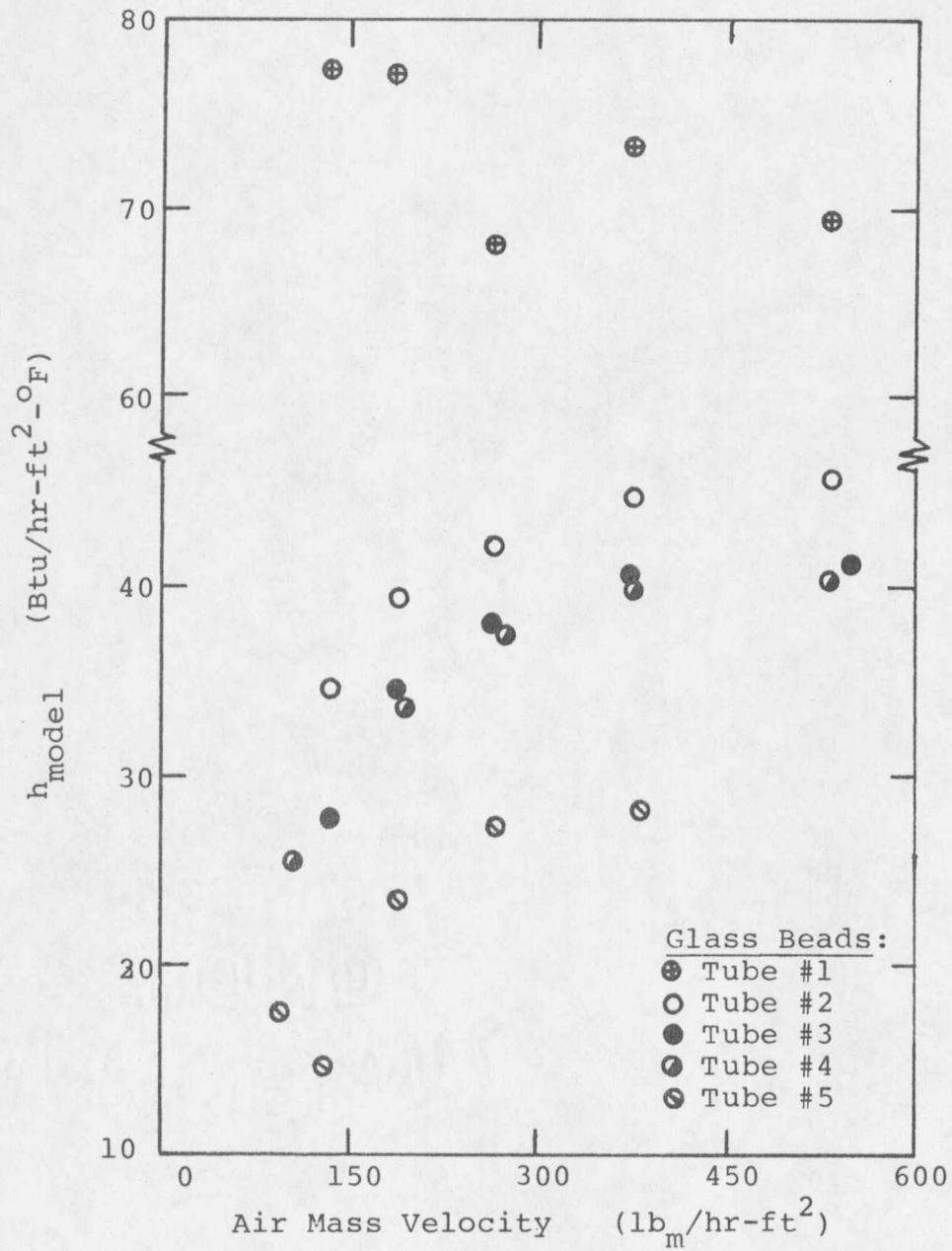


FIGURE 13. h_{MODEL} VERSUS AIR MASS VELOCITY,
GLASS BEADS, ALL TUBES

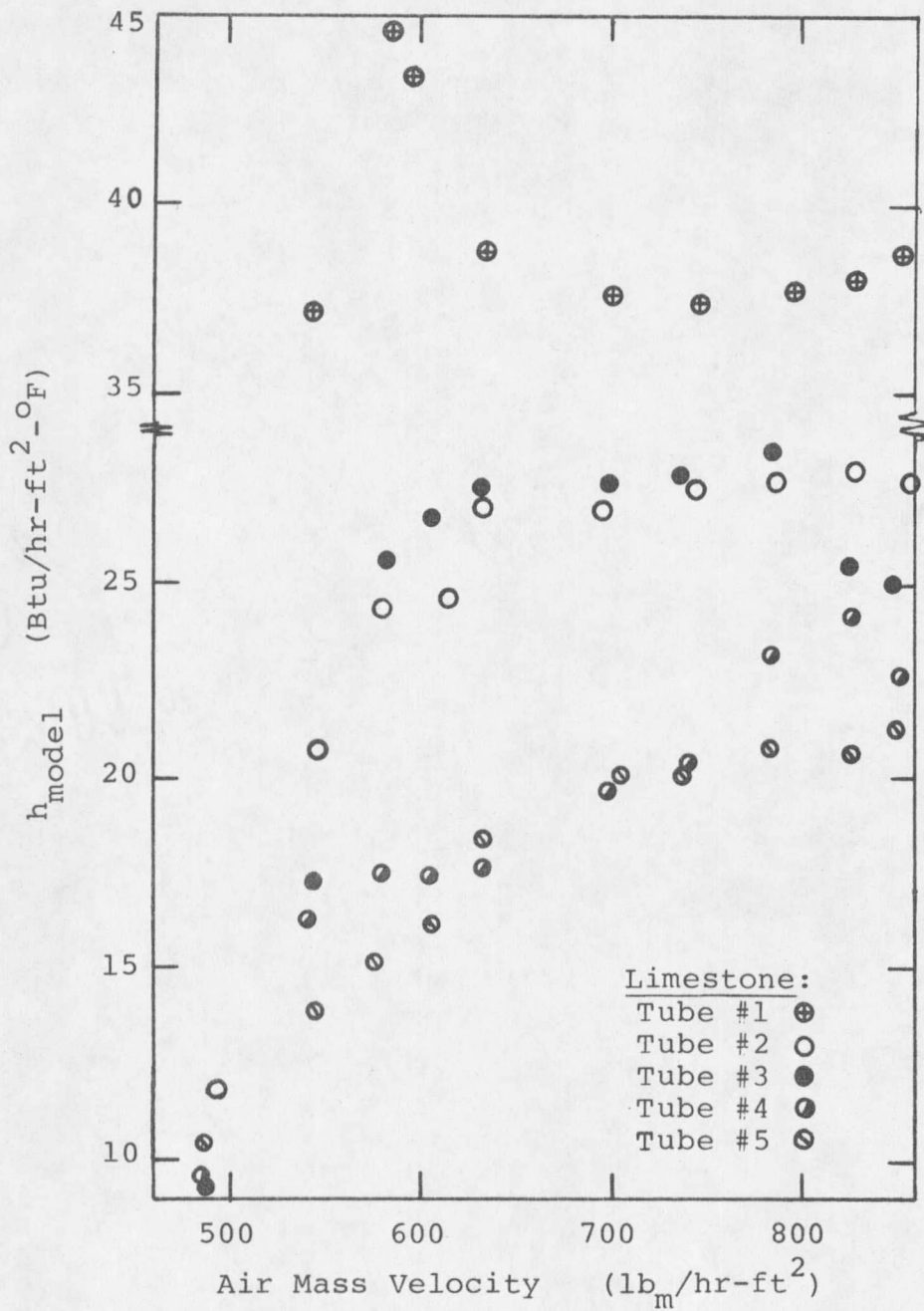


FIGURE 14. h_{MODEL} VERSUS AIR MASS VELOCITY,
LIMESTONE PARTICLES, ALL TUBES

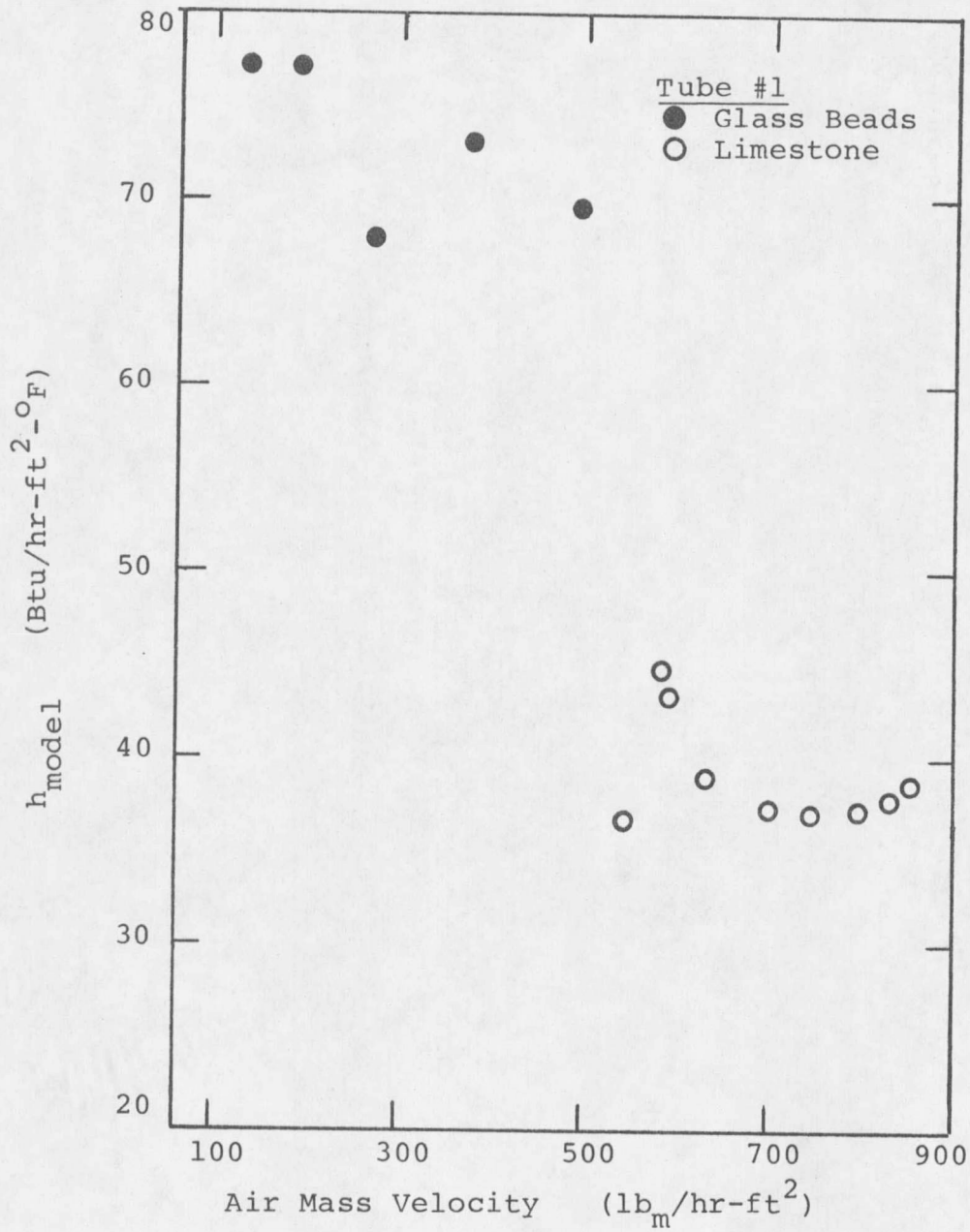


FIGURE 15. h_{MODEL} VERSUS AIR MASS VELOCITY,
TUBE #1, BARE

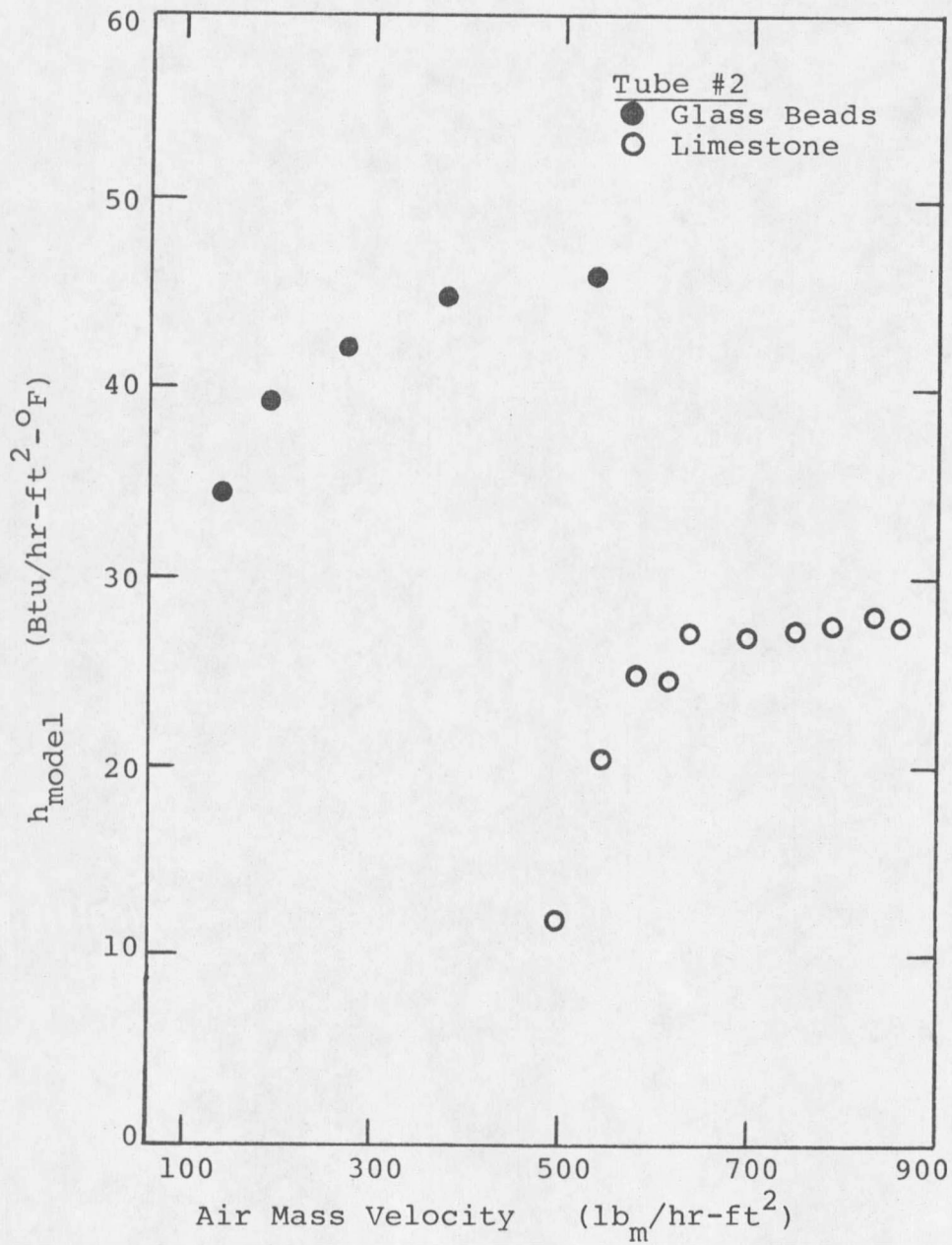


FIGURE 16. h_{MODEL} VERSUS AIR MASS VELOCITY,
TUBE #2, $\frac{1}{4}$ INCH FIN HEIGHT

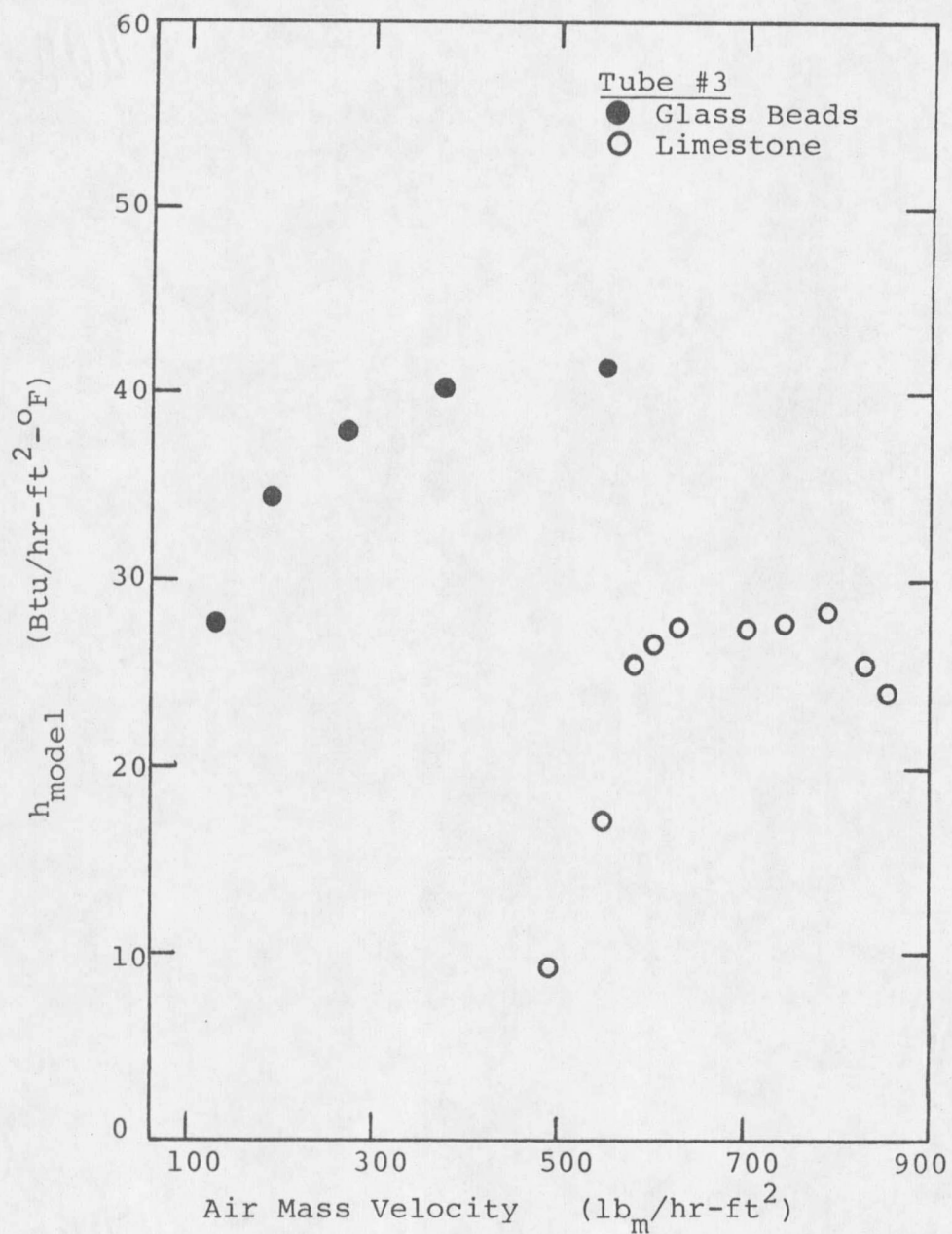


FIGURE 17. h_{MODEL} VERSUS AIR MASS VELOCITY,
TUBE #3, 3/8 INCH FIN HEIGHT

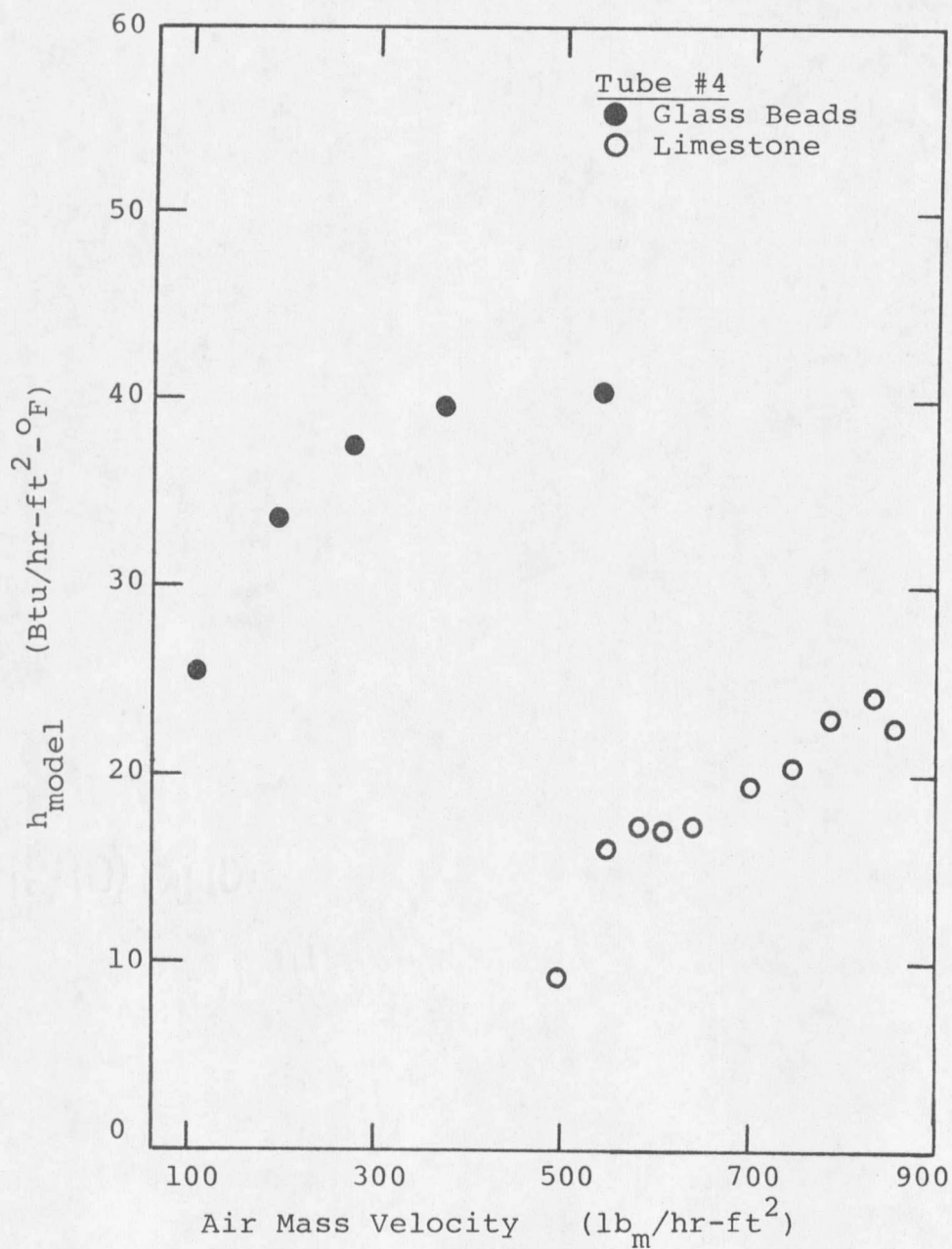


FIGURE 18. h_{MODEL} VERSUS AIR MASS VELOCITY,
TUBE #4, $\frac{1}{2}$ INCH FIN HEIGHT

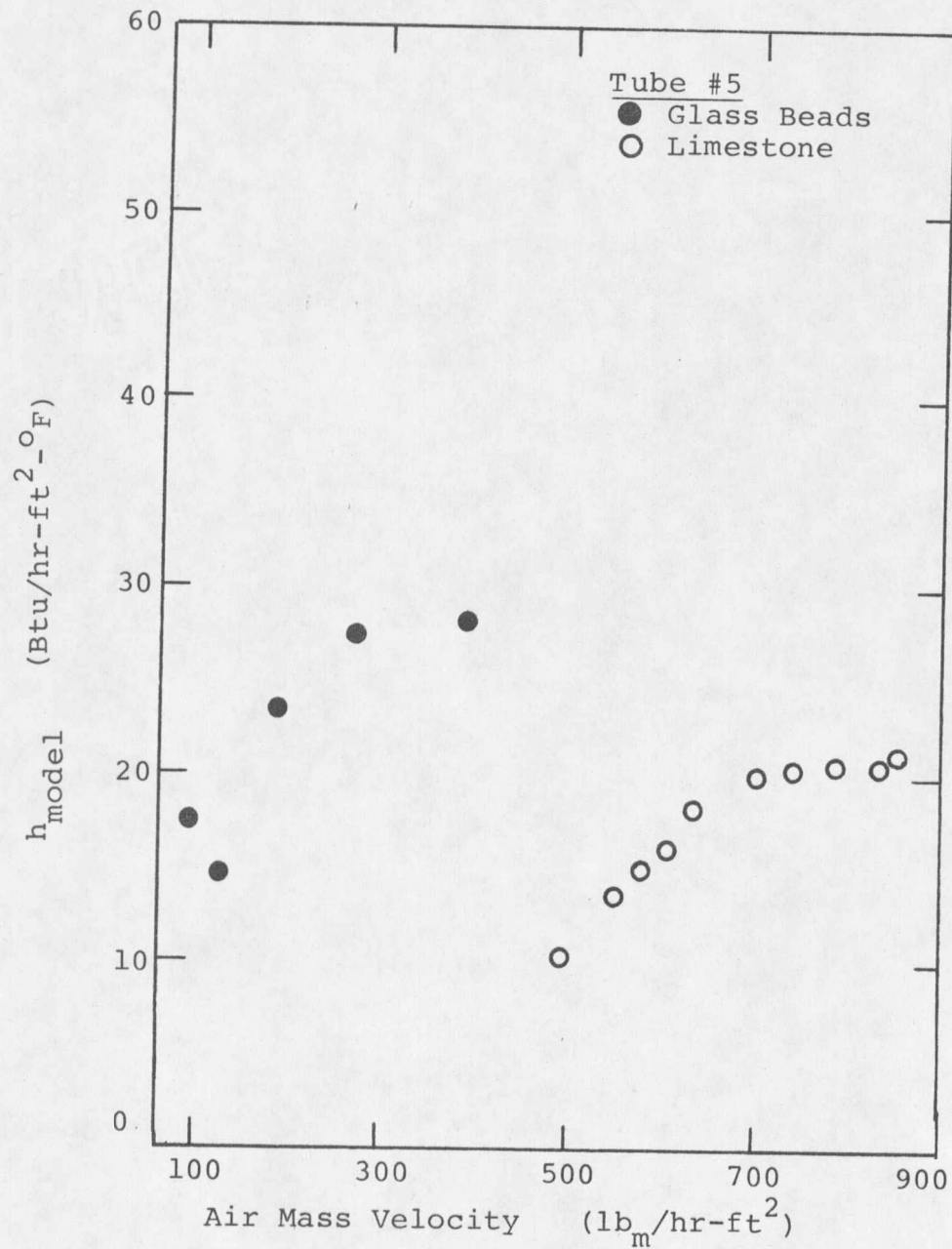


FIGURE 19. h_{MODEL} VERSUS AIR MASS VELOCITY,
TUBE #5, 3/4 INCH FIN HEIGHT

interstitial gas mass velocity. The bare tube, limestone particle data also exhibits an early peak of the heat transfer coefficient near minimum fluidization, but starts to rise again at high flow rates. This is shown in Figure 15. Plots of the Reynolds number vs. the Nusselt number show the same trends. These plots are shown in Figures 45, 46, 47, 48, and 49.

Results of Modeling

A number of models were explored in an attempt to correlate the data of this experiment. The bare tube data was not used in this correlation because its graphic form bears no resemblance to the data for the finned tubes. The equation for the best fit of the limestone particle, finned tube data is:

$$Nu = -0.226 + 0.117 \cdot Re^{0.917} \cdot (1.0 - \text{Void Fraction})^{0.00971} \cdot (0.0625/\text{Fin Height})^{0.322}$$

The best curve fit for the glass beads, finned tube data is:

$$Nu = -0.768 + 0.00542 \cdot Re^{0.132} \cdot Ar^{0.821} \cdot (0.0625/\text{Fin Height})^{0.285}$$

These correlations do not predict a maximum in the

Nusselt number and also should not be used beyond the scope of the experimental data from which they were derived. Figures 20 and 21 show graphically the deviation of the data from the model. Most of the limestone particle data was within 30 percent of the model. The glass beads experimental data was within 15 percent of the model value for all but one point. Tables 4 and 5 show the values of the experimental Nusselt number, the modeled Nusselt number, and the percent difference.

Performance of the Longitudinally Finned Tubes

The ratio of the product of the heat transfer coefficient and the total area of the finned tube to the product of the bare tube heat transfer coefficient and the area of a plain tube occupying the same volume, under identical fluidizing conditions, is a measure of the performance of the finned tube. Performance can be evaluated using the following equation:

$$\text{Performance} = \frac{(Q/\Delta T)_{\text{Finned Tube}}}{(Q/\Delta T)_{\text{Bare Tube}}}$$

where,

Q = heat transferred, Btu/hr

TABLE IV. RESULTS OF CORRELATION

Limestone Particles

Model:

$$\text{Nu} = - 0.226 + 0.117 \cdot \text{Re}^{0.917} \cdot (1.0 - \text{Void Fraction})^{0.00971} \cdot (0.0625/\text{Fin Height})^{0.322}$$

COMPARISON CHART

EXPERIMENTAL NUSSOLT	MODEL NUSSOLT	PERCENT DIFFERENCE
4.10	6.50	-36.91
6.97	7.00	-.47
8.23	7.41	11.09
8.29	7.84	5.68
9.09	8.07	12.69
9.01	8.74	3.09
9.19	9.24	-.55
9.24	9.71	-4.83
9.27	10.19	-8.99
9.18	10.52	-12.70
3.13	5.51	-43.24
5.68	5.91	-3.91
8.41	6.31	33.37
8.78	6.55	34.07
9.00	6.82	31.98
9.02	7.47	20.80
9.08	7.83	15.89
9.24	8.27	11.68
8.24	8.67	-4.91
7.77	8.87	-12.44
3.18	5.02	-36.60
5.28	5.33	-.85
5.70	5.69	.11
5.64	5.93	-4.60
5.71	6.16	-6.52
6.38	6.74	-5.41
6.60	7.09	-6.89
7.52	7.49	.42
7.78	7.87	-1.15
7.32	8.07	-9.28
3.51	4.35	-19.34
4.52	4.66	-3.04
4.91	4.93	-.31
5.21	5.15	1.17
5.98	5.38	11.13
6.45	5.91	9.19
6.54	6.20	5.46
6.71	6.54	2.63
6.63	6.87	-3.54
6.81	7.03	-3.07

TABLE V. RESULTS OF CORRELATION

Glass Beads

Model:

$$\text{Nu} = -0.768 + 0.00542 \cdot \text{Re}^{0.132} \cdot \text{Ar}^{0.821} \cdot (0.0625/\text{Fin Height})^{0.285}$$

COMPARISON CHART

EXPERIMENTAL NUSSELT	MODEL NUSSELT	PERCENT DIFFERENCE
.88	.85	3.09
.77	1.12	-31.26
1.28	1.34	-4.24
1.51	1.53	-1.15
1.56	1.70	-8.02
1.31	1.19	9.64
1.82	1.59	14.22
2.06	1.80	14.43
2.21	1.97	12.30
2.25	2.12	6.18
1.47	1.58	-6.86
1.85	1.81	2.15
2.09	2.01	3.79
2.23	2.20	1.44
2.30	2.38	-3.30
1.82	1.91	-4.80
2.13	2.12	.28
2.31	2.35	-1.58
2.49	2.56	-2.76
2.55	2.70	-5.52

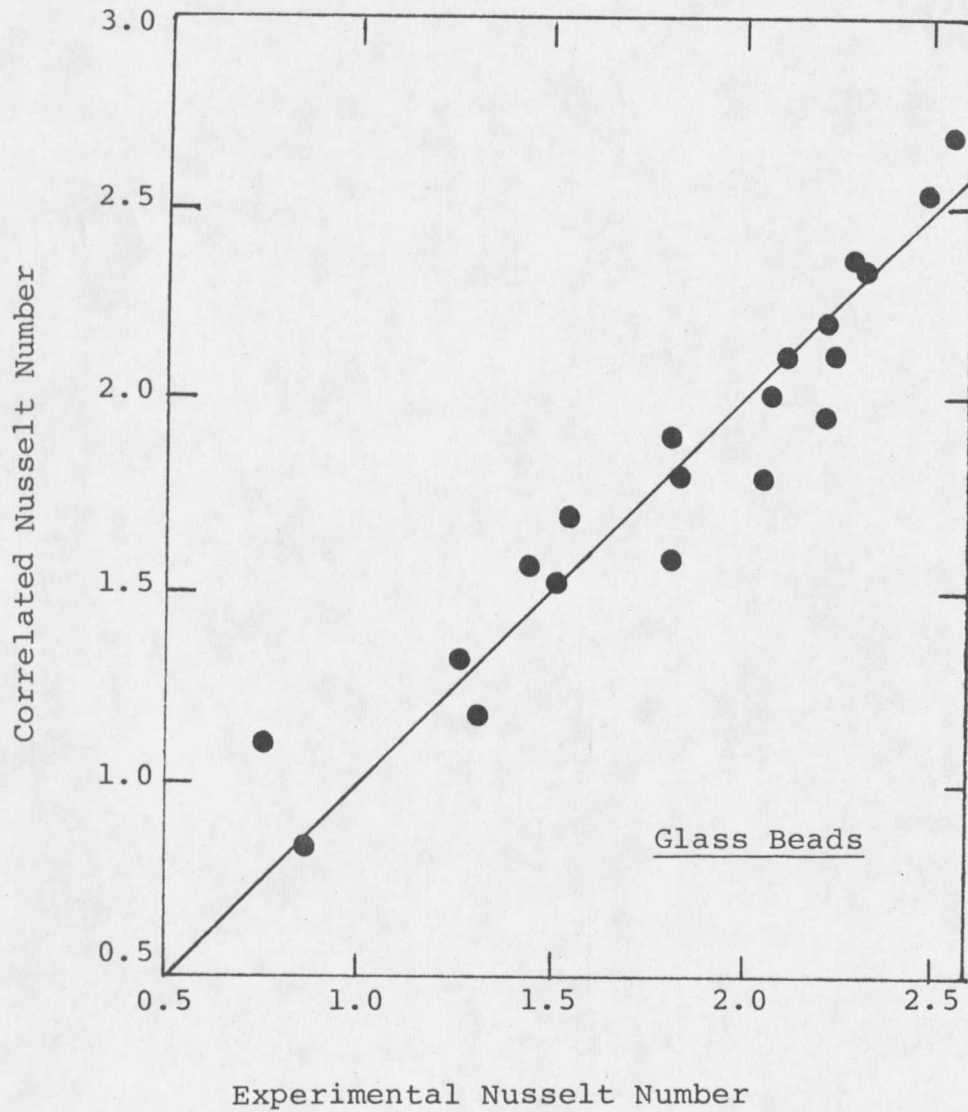


FIGURE 20. CORRELATED NUSSULT NUMBER VERSUS
EXPERIMENTAL NUSSULT NUMBER,
GLASS BEADS, ALL FINNED TUBES

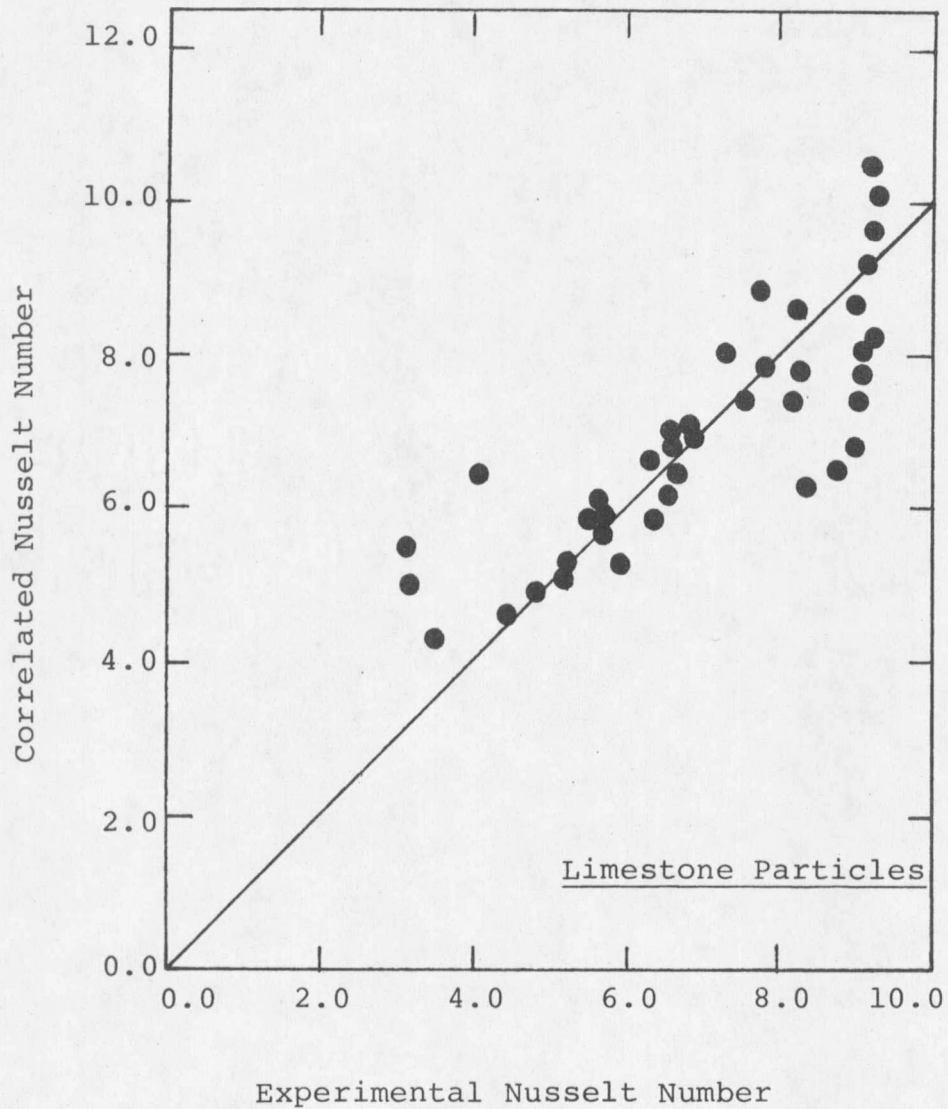


FIGURE 21. CORRELATED NUSSELT NUMBER VERSUS
EXPERIMENTAL NUSSELT NUMBER,
LIMESTONE PARTICLES, ALL FINNED TUBES

ΔT = temperature driving force, $^{\circ}\text{F}$

An overall fin diameter equal to the distance from a fin tip through the radial center to the opposite fin tip was used to calculate the area of the bare tube. This basis means that a finned tube and a bare tube will require the same amount of space in a fluidized bed.

When the performance ratio exceeds one, then for a given volume of the fluidized bed, the finned tube provides higher heat transfer per unit volume than the bare tube.

The best performances were obtained with tube #3 for the limestone particles and tube #4 for the glass beads. Figures 22 and 23 show the performance data for the limestone particles and the glass beads. Tube #3 showed a 70 percent maximum enhancement. Tube #4 for the glass beads showed a 40 percent maximum enhancement.

The glass beads out perform the limestone particles at the same flow rates, but over the ranges which could be considered the normal operating mass flow rate for the particular particle, the limestone particles always yield a higher maximum performance. Figures 24, 25, 26, and 27 show the particle performance for each individual

tube.

The particle size determines the mobility of the fluidized particle in a fluidized bed, for a given air flow rate. Small particles, such as the glass beads used in this experimental program, can move in and out of the space between the fins of a finned tube faster than large particles, thus making particle convection the dominant heat transfer mechanism for small particles. Particle convection is a powerful form of heat transfer in a fluidized bed and therefore when large particles can not utilize this form, their heat transfer coefficients are lower. The narrow space between the fins also limits particle convective heat transfer by increasing the residence time of the larger particles.

The result that several larger fin, finned tubes have higher heat transfer coefficients than the smaller fin, finned tubes, for a given flow rate, is probably due to the small differences between several of the fin heights and an unspecified relationship between the particle size and the fin height. This relationship effects the mobility of the particle in a given situation.

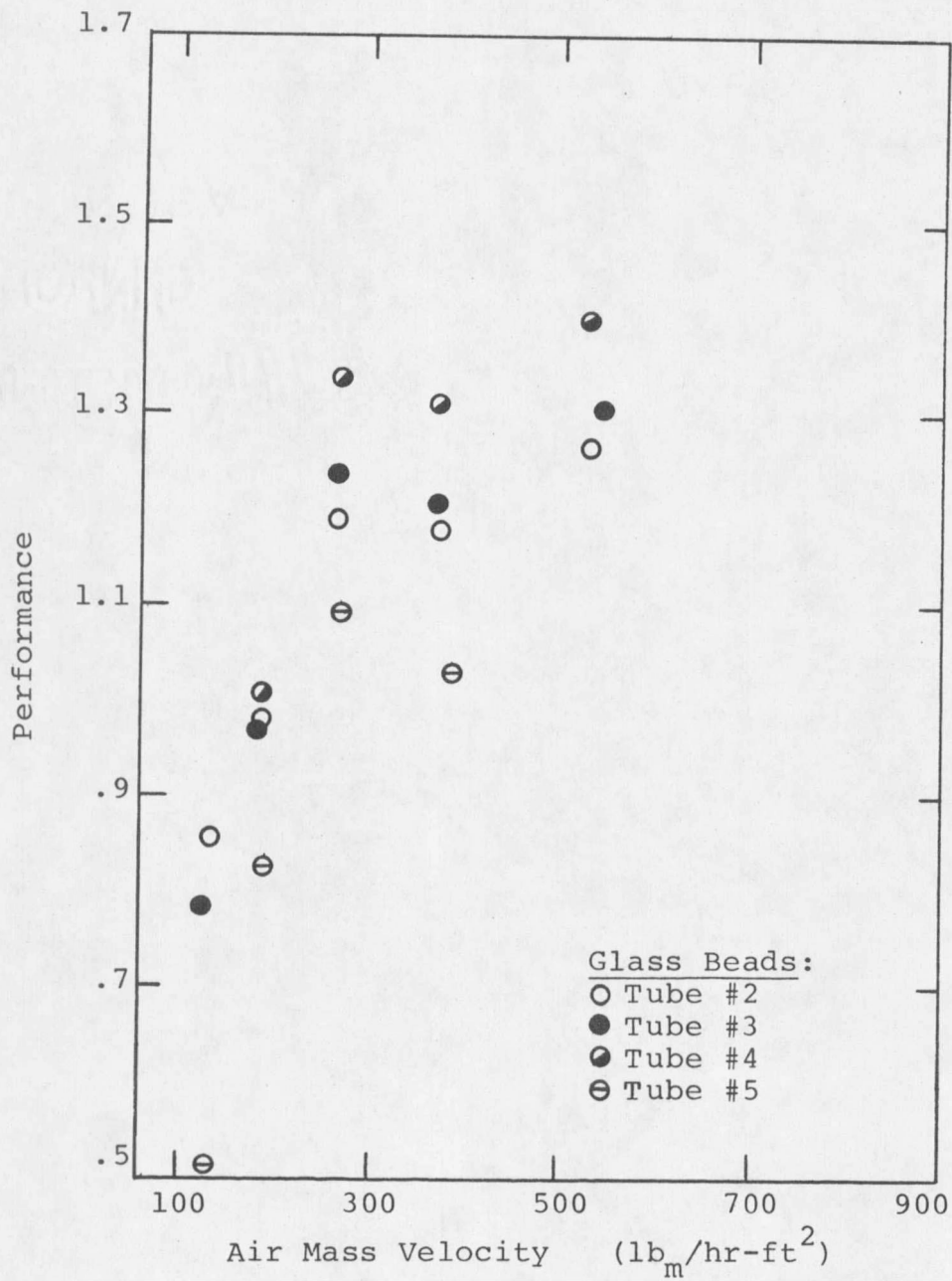


FIGURE 22. PERFORMANCE VERSUS AIR MASS VELOCITY, GLASS BEADS, ALL FINNED TUBES

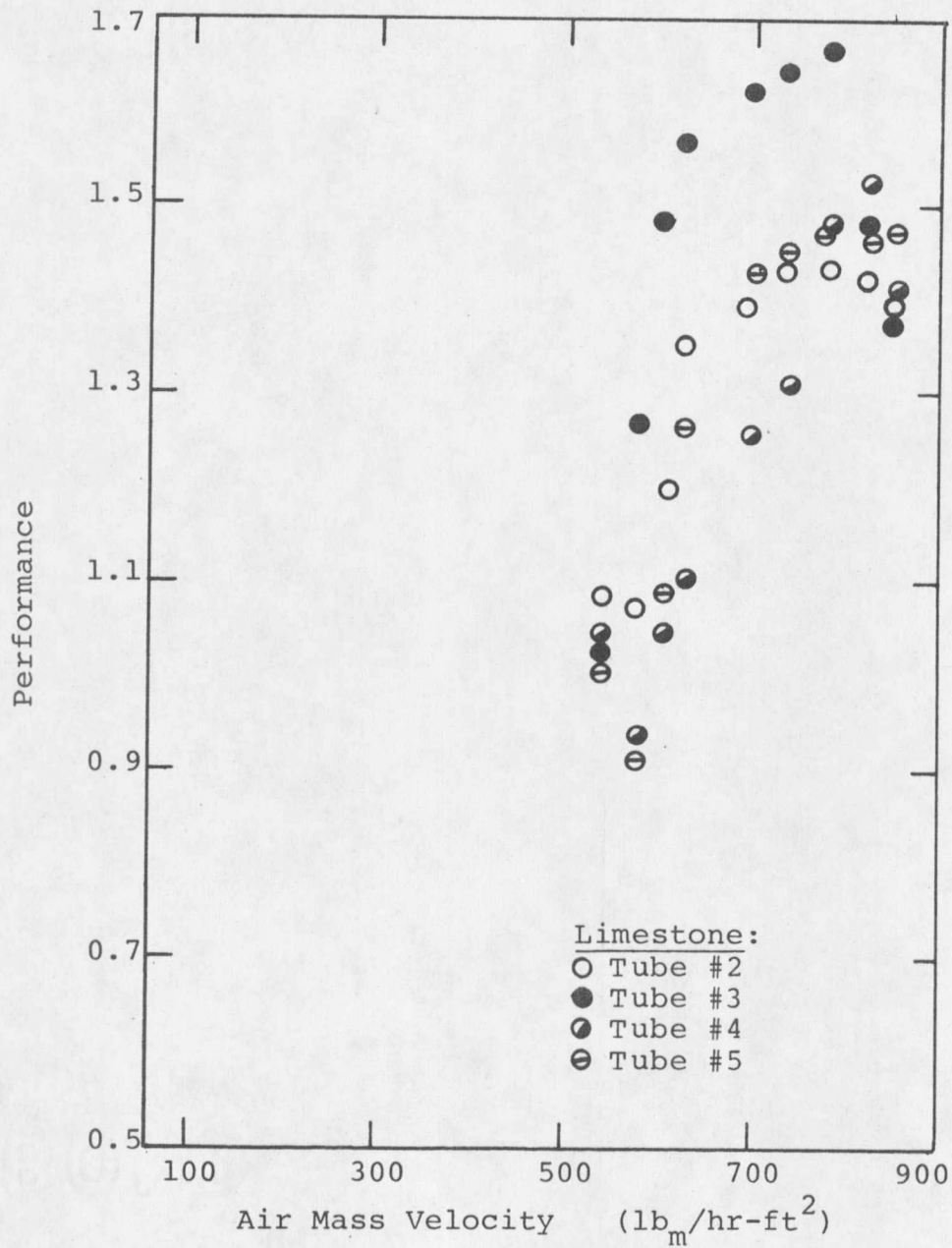


FIGURE 23. PERFORMANCE VERSUS AIR MASS VELOCITY, LIMESTONE PARTICLES, ALL FINNED TUBES

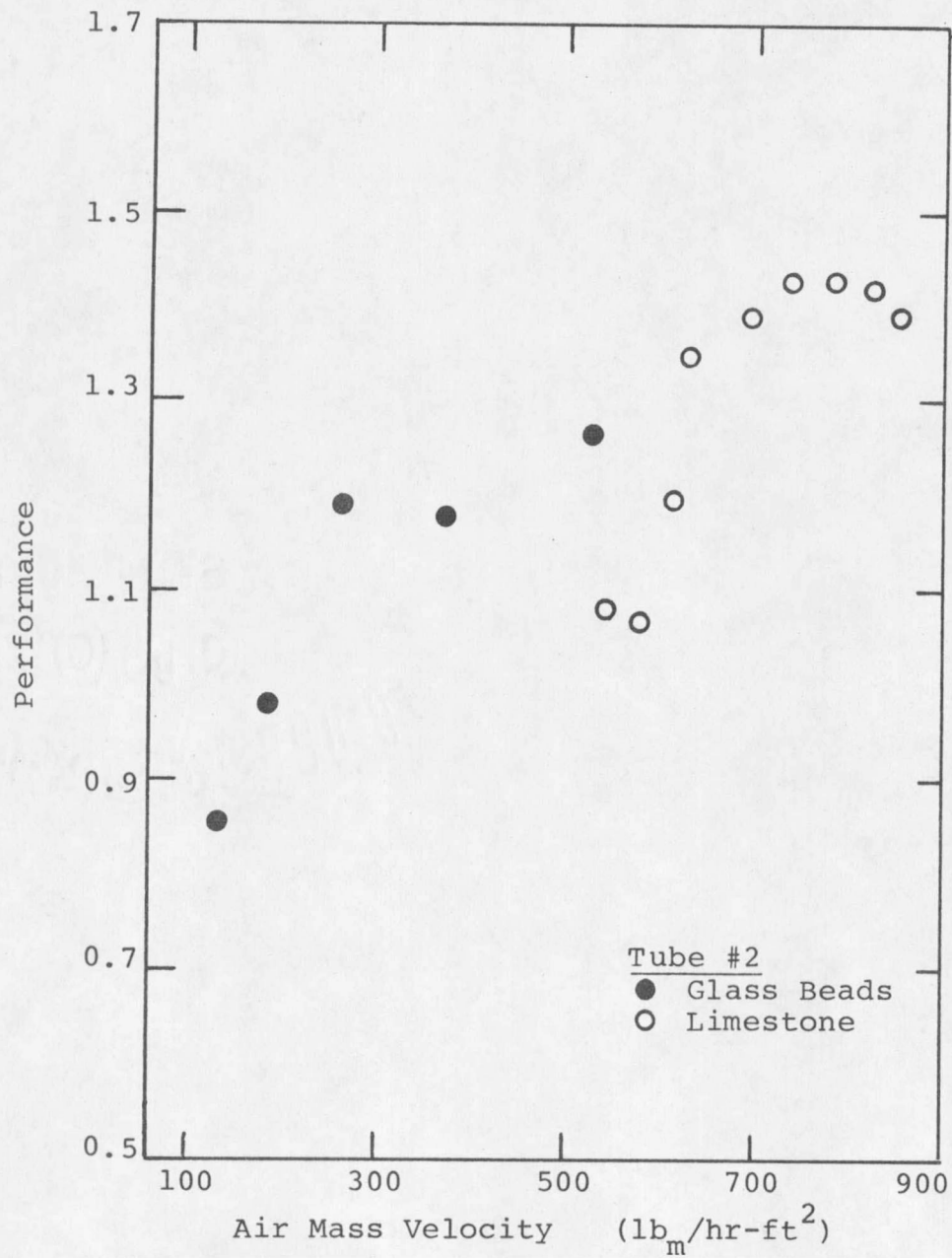


FIGURE 24. PERFORMANCE VERSUS AIR MASS VELOCITY, TUBE #2, $\frac{1}{4}$ INCH FIN HEIGHT

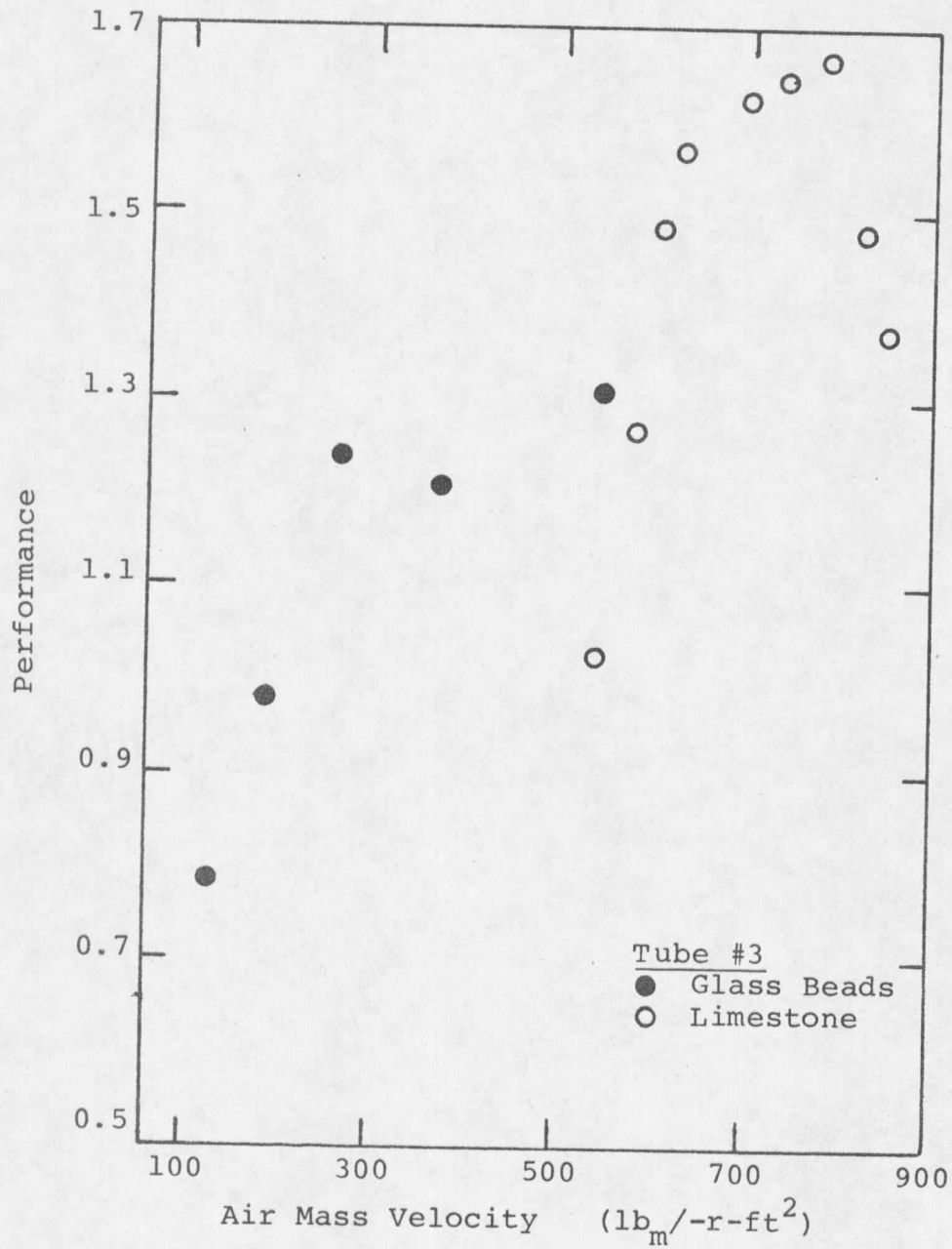


FIGURE 25. PERFORMANCE VERSUS AIR MASS VELOCITY,
TUBE #3, 3/8 INCH FIN HEIGHT

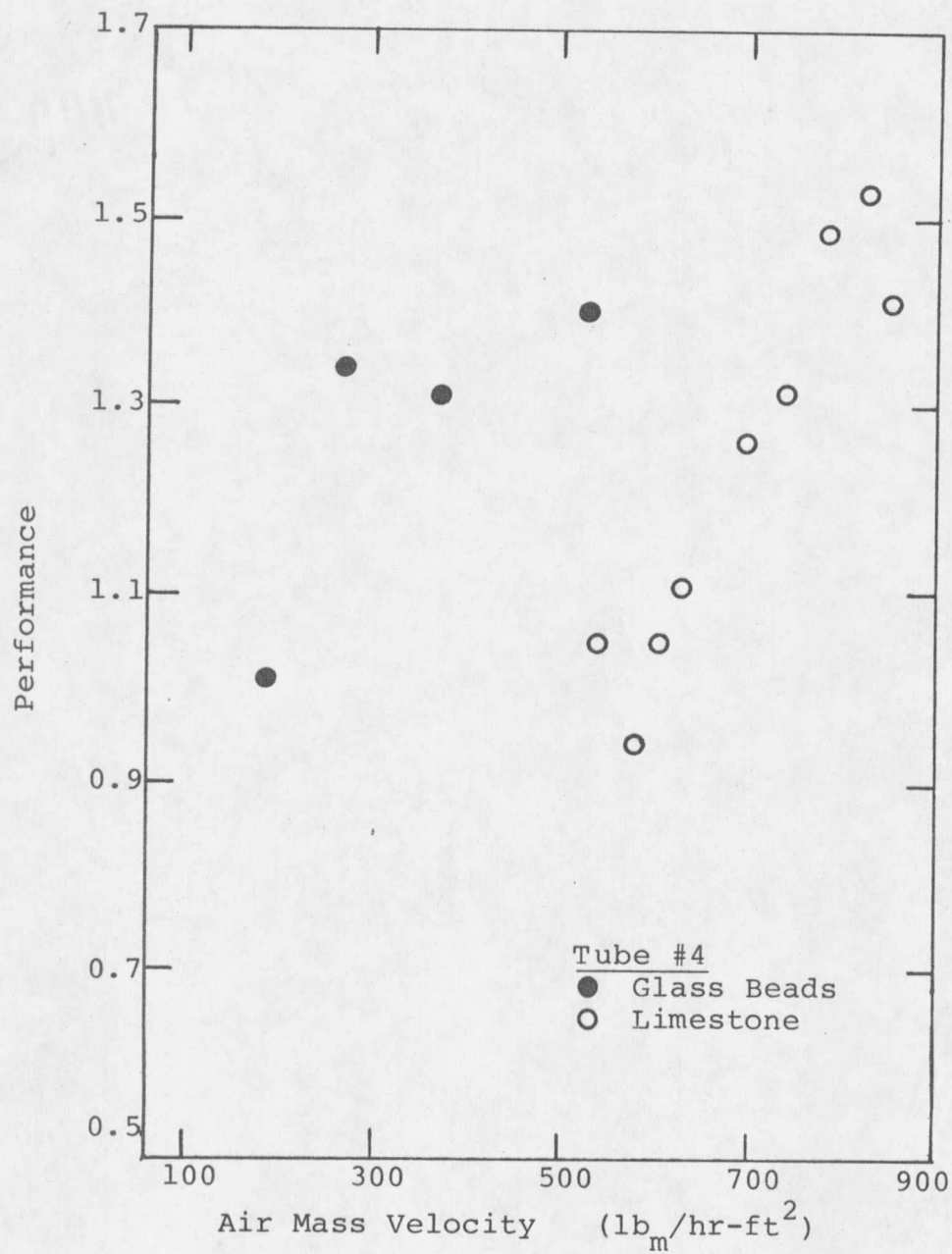


FIGURE 26. PERFORMANCE VERSUS AIR MASS VELOCITY,
TUBE #4, $\frac{1}{2}$ INCH FIN HEIGHT

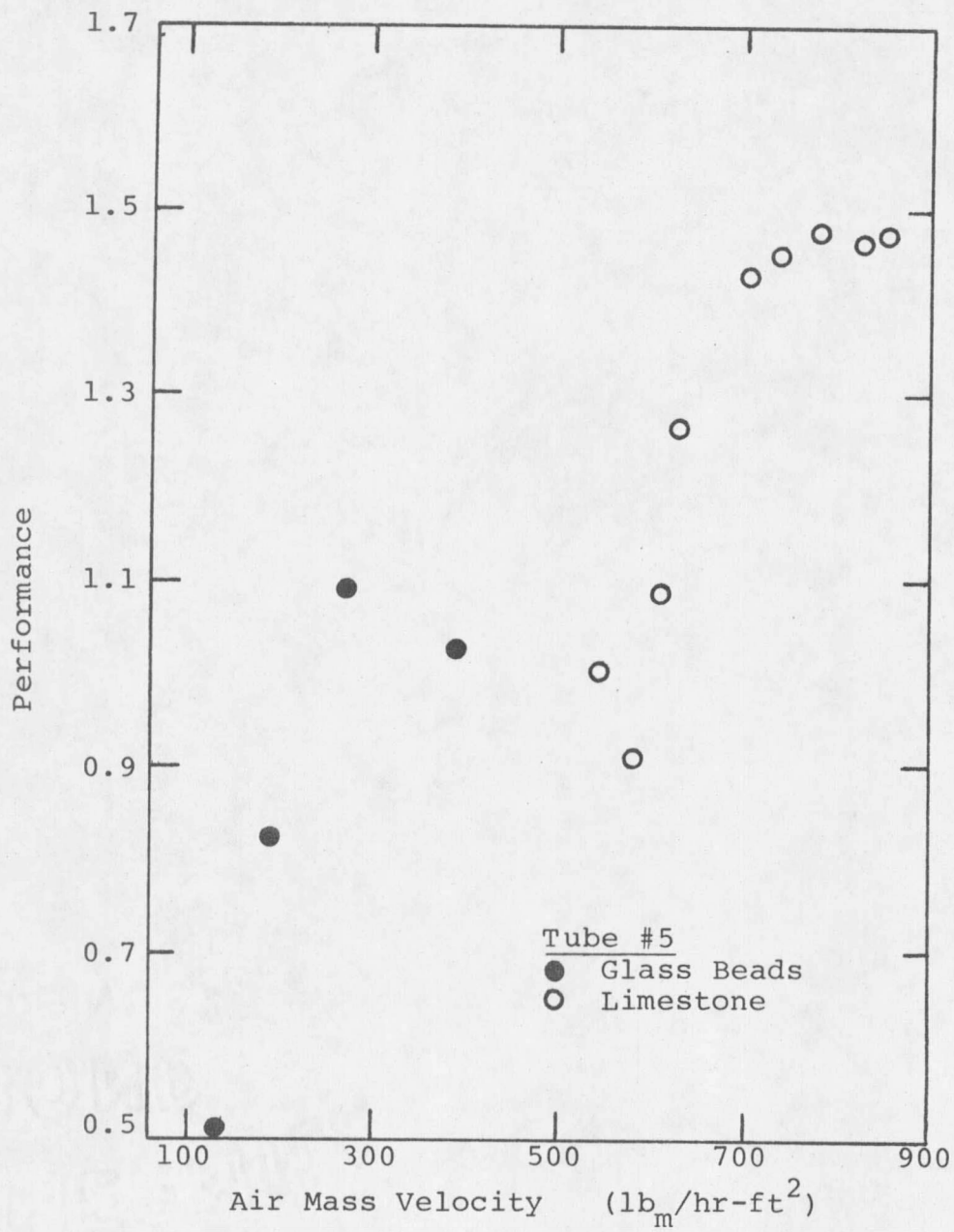


FIGURE 27. PERFORMANCE VERSUS AIR MASS VELOCITY, TUBE #5, 3/4 INCH FIN HEIGHT

ERROR ANALYSIS

The experimental heat transfer coefficient is assumed to be effected by the measurement of q_{exp} and $(T_w - T_b)$ only, therefore the error analysis is to be performed upon the following equation:

$$h_{\text{exp}} = \frac{q_{\text{exp}}}{A_T (T_w - T_b)}$$

The accuracy of the Simpson Model 390 wattmeter is 5% or better of the full scale reading (28). The 3000.0 watt scale was used for wattage measurement, therefore the accuracy of these measurements is $(150.0/1000.0) \times 100$ or $\pm 15\%$. The limits of error for the thermocouples is $\pm 4.0^\circ\text{F}$ (29). The bed temperature is assumed to be measured within this limit and therefore for a worst case $(T_w - T_b)$, the T_b value is 150.0°F and the percentage error is $(4.0/150.0) \times 100$ or $\pm 2.7\%$. The tube wall temperature was assumed to be measured within $\pm 9^\circ\text{F}$, $\pm 4^\circ\text{F}$ for the thermocouple error and $\pm 5^\circ\text{F}$ for the fluctuations observed during a typical experimental run.

The maximum and minimum values of the heat transfer coefficient can now be calculated and the percent maximum error estimated, using the previous assumptions. This analysis is based upon a heat transfer coefficient

of 1.0. The minimum ($T_w - T_b$) value in this investigation was 166.5 °F.

Maximum positive error

$$\frac{h_{\text{exp}}}{h} = \frac{1.15}{1.0 - 13.0/166.5} = 1.247$$

$$\text{Error} = (h_{\text{exp}} - h)/h = (1.247 - 1.0) \times 100 = 24.7 \%$$

Maximum negative error

$$\frac{h_{\text{exp}}}{h} = \frac{0.85}{1.0 + 13.0/166.5} = 0.788$$

$$\text{Error} = (h_{\text{exp}} - h)/h = (0.788 - 1.0) \times 100 = -21.2 \%$$

The proposed experimental error that brackets all of the data is +24.7% and -21.2%. The maximum deviation from the true values of the heat transfer coefficient should be ±24.7%.

CALCULATIONS

Air Mass Velocity

For the main blower a 1.5 inch orifice with vena contracta taps and a water filled manometer were used to measure the air flow rates. The secondary blower also used a 1.5 inch orifice with vena contracta taps and a water filled manometer to measure flow rates. The sum of the flow from these two blowers was the total flow through the bed. The following equation was used for the calculations (8):

$$G = \frac{3600 C_o Y S_c}{A_c} \left[\frac{2 g_c (P_1 - P_2) \rho_1}{1 - \beta^4} \right]^{0.5}$$

G = Air Mass Velocity, lbs/hr-ft²

C_o = Orifice Coefficient, Dimensionless

Y = Expansion Factor, Dimensionless

S_c = Cross Sectional Area of the Orifice, ft²

A_c = Cross Sectional Area of the Column, ft²

g_c = Dimensional Constant, ft-lb_m/sec²-lb_f

P_1 = Pressure at the Upstream Pressure Tap, lb_f/ft²

P_2 = Pressure at the Downstream Pressure Tap, lb_f/ft²

ρ_1 = Density of Air at the upstream Pressure, lb_m/ft³

β = Ratio of the Orifice Diameter to Inside Pipe Diameter, Dimensionless

For a square edged orifice, the expansion factor is given as follows:

$$Y = 1 - \frac{P_1 - P_2}{P_1 K_r} (.41 - .35^4)$$

where,

$$K_r = C_p / C_v$$

The orifice coefficient is a function of the Reynolds number. It was found to be nearly constant at .61 for the range of air flow rates used. This value is used in all air flow calculations.

Tube Temperature

The tube temperatures were measured using thermocouples connected to a Barber Colman Company Potentionmeter for high temperatures (above 600° F) and a Brown Eletronik Potentionmeter for low temperatures. Temperatures were also recorded as a voltage by a Varian G-2000 chart recorder.

Bed Temperature

The bed temperature, T_b , was measured at the mid-height of the bed by a Brown Eletronik Potentionmeter.

Heat Input to Each Tube

The power to each tube was measured with a Simpson wattmeter and then converted to the appropriate units.

$$q = (\text{Watts}) \frac{3.4129 \text{ Btu}}{\text{Watt-hr}}$$

Total Tube Area

The total fin area was found by multiplying the area per fin by 12 (the number of fins). The total tube area was calculated by adding the total fin area and the area of the unfinned portion of the tube.

Air Thermal Conductivity

The air thermal conductivity, k_g , was determined by linear interpolation between the values listed in Kreith (9). The linear equation fitting the thermal conductivity data is given below:

$$k_g = 2.0 \times 10^{-5} T_b + .0134$$

k_g = Thermal Conductivity, Btu/hr-ft-°F

T_b = Bed Temperature, °F

This equation is valid for the temperature range from 32° F to 250° F.

Air Viscosity

The air viscosity was calculated for each temperature from the following equation which was fit to experimental data (9):

$$\mu_g = (2.45 (T_b - 32) + 1538.1) 2.688 \times 10^{-5}$$

μ_g = Air Viscosity, lb/hr-ft

T_b = Bed Temperature, °F

Heat Transfer Coefficient

The heat transfer coefficients were calculated using the following equation and the Wegstein convergence routine.

$$h = \frac{\frac{Q}{(T_w - T_b)} - \frac{N \cdot k \cdot B \cdot \delta \cdot m}{L} \left[\frac{h \cdot \cosh(m) + (km/L) \cdot \sinh(m)}{h \cdot \sinh(m) + (km/L) \cdot \cosh(m)} \right]}{B(\pi D - N\delta)}$$

h = Heat Transfer Coefficient, Btu/hr-ft²-°F

Q = Heat Input per Tube, Btu/hr

T_w = Tube Wall Temperature, $^{\circ}\text{F}$

T_b = Bed Temperature, $^{\circ}\text{F}$

N = The Number of Fins

k = Thermal Conductivity of the Fin Material,
 $\text{Btu/hr-ft-}^{\circ}\text{F}$

δ = Thickness of the Fin, Ft

B = Length of the Fin, Ft

L = Height of the Fin, Ft

π = 3.14159, dimensionless

d_o = Diameter of the Tube, Ft

$m = \sqrt{\frac{2 \cdot h \cdot L^2}{k}}$, dimensionless

Particle Nusselt Number

$$\text{Nu}_p = \frac{h D_p}{k_g}$$

Nu_p = Nusselt Number, Dimensionless

h = Heat Transfer Coefficient, $\text{Btu/hr-ft}^2\text{-}^{\circ}\text{F}$

D_p = Particle Diameter, Ft

k_g = Air Thermal Conductivity, $\text{Btu/hr-ft}^2\text{-}^{\circ}\text{F}$

Particle Reynolds Number

$$\text{Re}_p = \frac{G D_p}{\mu_g}$$

Re_p = Particle Reynolds Number, Dimensionless

G = Air Mass Velocity, Lbs/hr-ft²

D_p = Particle Diameter, Ft

μ_g = Air Viscosity, Lbs/hr-ft

Archimedes Number

$$Ar = \frac{g D_p \rho_g (\rho_p - \rho_g)}{\mu_g^2}$$

Ar = Archimedes Number, Dimensionless

g = Gravitation Constant, Ft/hr²

D_p = Particle Diameter, Ft

ρ_g = Density of Air, Lbs/ft³

ρ_p = Density of the Particles, Lbs/ft³

μ_g = Air Viscosity, Lbs/hr-ft

CONCLUSIONS

The following conclusions can be drawn from this experimental investigation.

1. The heat transfer coefficient for all of the finned tubes increased with increasing fluidizing air mass velocity. It is believed that increasing air mass velocity enhances particle motion, which decreases the resistance to heat transfer at the boundary and increases particle heat convection. In several cases a maximum value was reached. This maximum is observed because of lower particle density at high flow rates.
2. The small particles gave better heat transfer coefficients in the range where the air flow rates were the same for the large and small particles. This occurs because particle mode heat convection by the small particles is greater than by the large particles.
3. For the glass bead runs, the bare tube gave the highest value of the heat transfer coefficient, with Tube #2 ($\frac{1}{4}$ In.) giving the second largest heat transfer coefficients. This is explained by better particle circulation near the surface when there is

minimal flow restriction.

4. For the limestone particle runs, the bare tube gave the highest values of the heat transfer coefficient, with Tube #3 (3/8 In.) giving the second highest heat transfer coefficients and Tube #2 (1/4 In.) only slightly lower values. Tubes #3 and #2 were very similar in fin height and therefore could be expected to yield similar results.
5. Heat transfer coefficients generally decreased for increasing fin height, which can be explained by considering the physical obstruction of the larger fin height on particle motion between the fins and near the tube base.
6. The best performance using the limestone particles was obtained using Tube #3 (3/8 In.), the gain was 70 percent. This result can be attributed to the effects of minimally reduced particle circulation and large surface area.
7. The best performance using the glass beads was obtained using Tube #4 (1/2 In.), the maximum gain was 40%. It would be expected that the smaller particles would circulate better in the narrower confines.

NOMENCLATURE

NOMENCLATURE

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
A,B,C, D,E	Variables in the Nusselt Number Correlation Program	Dimensionless
a,b,c, d,e	Parameters in the Nusselt Number Correlation Program	Dimensionless
A _b	Area of the base section of the tube	ft ²
A _c	Column cross sectional area	ft ²
Ar	Archimedes number	Dimensionless
B	Fin material length	ft
C ₁	Constant	Dimensionless
C ₂	Constant	Dimensionless
C _g	Gas heat capacity	Btu/lb _m -°F
C _o	Orifice coefficient	Dimensionless
C _p	Constant pressure air heat capacity	Btu/lb _m -°F
C _p	Particle heat capacity	Btu/lb _m -°F
C _r	Correlation factor	Dimensionless
C _v	Constant volume air heat capacity	Btu/lb _m -°F
d _p	Diameter of the particles	ft
d _o	Diameter of the finned tube	ft
D	Diameter of the fluidized bed	ft
G	Mass flow rate	lb _m /hr-ft ²
g _c	Dimensional constant	ft-lb _m /sec ² -lb _f

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
h	Heat transfer coefficient	$\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
h_1, h_2, h_3	Sectionally averaged heat transfer coefficients	$\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
h_{mf}	Heat transfer coefficient at minimum fluidization conditions	$\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
h_{model}	Fin model heat transfer coefficient	$\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
k	Thermal conductivity of the fin	$\text{Btu/hr-ft-}^\circ\text{F}$
k_e	Thermal conductivity of the fluid	$\text{Btu/hr-ft-}^\circ\text{F}$
k_f	Thermal conductivity of the fluid	$\text{Btu/hr-ft-}^\circ\text{F}$
k_g	Thermal conductivity of the gas	$\text{Btu/hr-ft-}^\circ\text{F}$
K_r	Ratio of C_p/C_v	Dimensionless
L	Height of the fin	ft
N_{Fin}	Number of fins per tube	Dimensionless
m	Dimensionless number	
Nu_{conv}	Convective Nusselt number	Dimensionless
Nu_{model}	Correlation modeled Nusselt number	Dimensionless
Nu_p	Particle Nusselt number	Dimensionless
ΔP	Manometer fluid height	ft
P_1	Upstream pressure	lb_f/ft^2
P_2	Downstream pressure	lb_f/ft^2

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
Pr	Prandtl number	Dimensionless
$q_1, q_2,$ q_3, q	Rate of heat transfer	Btu/hr
Q	Rate of heat transfer	Btu/hr
Q_{base}	Heat loss from the tube base	Btu/hr
Q_{Fin}	Heat loss from the finned section	Btu/hr
Q_{Total}	Total heat input	Btu/hr
Re_{mf}	Minimum fluidization Reynolds number	Dimensionless
Re_p	Particle Reynolds number	Dimensionless
S_c	Orifice cross sectional area	ft ²
T	Temperature in the fin	°F
T_b	Bed temperature	°F
T_f	Fluid temperature	°F
T_w	Tube wall surface temperature	°F
$T(0)$	Particle temperature	°F
$\underline{x}$	Distance along the fin	ft
x	Dimensionless distance	ft
Y	Expansion factor	Dimensionless
β	Ratio of orifice diameter to inside pipe diameter	Dimensionless

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
δ	Thickness of the fin	ft
δ'	Thermal boundary layer thickness	ft
ϵ	Bed solids void fraction	Dimensionless
λ_g	Thermal conductivity of the fluid	Btu/hr-ft- $^{\circ}$ F
π	Pi , 3.1416	Dimensionless
θ	Dimensionless temperature	
μ	Viscosity of the gas	lb/hr-ft
ρ	Density of the material	lb/ft ³
ρ_g	Density of the gas	lb/ft ³
ρ_p	Density of the particles	lb/ft ³

BIBLIOGRAPHY

BIBLIOGRAPHY

1. Botterill, J.S.M., Fluid-Bed Heat Transfer, 1st ed., Academic Press, New York, 1975.
2. Vanecek, V., M. Markvart, and R. Drbohlav, Fluidized Bed Drying, 1st ed., Leonard Hill Books, London, 1966.
3. Kunii, D., and O. Levenspiel, Fluidization Engineering, 1st ed., Robert E. Krieger Publishing Co., Huntington, New York, 1977.
4. Davidson, J.F., and D. Harrison, ed., Fluidization, Academic Press, New York, 1971.
5. Heylin, M., ed., Chemical and Engineering News, 59, 29, July 20, 1981, pp. 34-37.
6. Montagna, J.C., et al, A.I.Ch.E. Symp. Series, v74, n176, 1978, pp. 203-211.
7. Everly, D.W., M.S. Thesis, Montana State University, (1978).
8. Perry, R.H., and C.H. Chilton, Chemical Engineers' Handbook, 5th ed., McGraw-Hill Book Co., New York, 1973.
9. Kreith, F., Principles of Heat Transfer, 3rd ed., IEP, A Dun Donnelly Publisher, New York, 1976.
10. Botterill, J.S.M., A.I.Ch.E. Symp. Series, v69, n128, 1973, pp. 26-27.
11. Franks, R.G.E., Modeling and Simulation in Chemical Engineering, 1st ed., Wiley-Interscience, New York, 1972.
12. Yurich, S.P., M.S. Thesis, Montana State University, (1979).
13. Davidson, J.F. and D. Harrison, ed., Fluidization, 1st ed., Academic Press, New York, pp.476.

14. Grace, J.R. and J.M. Matsen, ed., Fluidization, 1st ed., Plenum Press, New York, 1980, pp. 225.
15. Catipovic, N.M., T.J. Fitzgerald, and G. Jovanovic, "A Study of Heat Transfer to Immersed Tubes in Fluidized Beds", (Presented at the 71st Annual Meeting of the A.I.Ch.E., November, 1978)
16. Denloye, A.O.O. and J.S.M. Botterill, Powder Technology, 19, 197, 1978.
17. George, A.H., N.M. Catipovic, and J.R. Weltly, "An Analytical Study of Heat Transfer to a Horizontal Cylinder in a Large Particle Fluidized Bed", A.S.M.E. paper No. 79-HT-78, 18th National Heat Transfer Conference, 1979.
18. Adams, R.L. and J.R. Weltly, A.I.Ch.E. Journal, v 25, No. 3, 1979, pp. 395-405.
19. Baskakov, A.P., and V.M. Suprun, International Chemical Engineering, v 12, No. 2, April 1972, pp. 324.
20. Botterill, J.S.M., and A.O.O. Denloye, A.I.Ch.E. Symp. Series, v 74, No. 176, 1978, pp. 194-202.
21. Mickley, H.S., and D.F. Fairbanks, A.I.Ch.E. Journal, v 1, 1955, pp. 374.
22. Wender, L., and G.T. Cooper, A.I.Ch.E. Journal, v 4, No. 1, 1958, pp. 15.
23. Gamson, B.W., Chemical Engineering Progress, v 47, No. 1, 1951, pp. 19.
24. Vreedenberg, H.A., Chemical Engineering Science, v 11, 1960, pp. 274.
25. Gabor, J.D., Chemical Engineering Science, v 25, 1970, pp. 979.

26. Staub, F.W., Heat Transfer to surfaces immersed in fluidized Beds, Corporate Research and Development, General Electric Company. Presented at National Science Foundation Fluidization and Particle System-Research Needs and Priorities, R.P.I., Troy, New York, Oct. 1979.
27. Chen, J.C. and J.G. Withers, A.I.Ch.E. Symp. Series, v74, No. 174, pp. 327-333, 1978.
28. Operator's Manual, Dynamometer Type Model 390, Simpson Electric Company, Chicago, IL, pp. 9, 1969.
29. Temperature Measurement Handbook, Omega Engineering Inc, Stamford, CN, pp. A-4, 1980.

APPENDICES

APPENDIX A

MODELING OF THE MULTIPLE THERMOCOUPLE FINNED TUBE

Modeling work was done using temperature data obtained from tube #5. Temperatures were recorded from a thermocouple attached on the base of the finned tube and from thermocouples attached radially outward on several fins. Figure 7 shows a diagram illustrating the thermocouple placement points. The thermocouples were actually soldered to different fins.

The differential equation,

$$\frac{d^2\theta}{dx^2} = \frac{2hL^2\theta}{k\delta}$$

was solved analytically using boundary conditions which were available from experimental data. The program is shown in Figure 28. The fin was divided into four sections. The sections were defined as the spaces between the thermocouple attachment points shown in Figure 8. The first section had the boundary condition:

a) $\theta = 1$ @ $x=0$

b) $\theta = \theta_1$ @ $x=x_1$

```
1.000 C      FIN MODELING PROGRAM
2.000 C
3.000      INPUT HEXP,BEDT,XT
4.000 30     INPUT XS,XE,XSTART
5.000      XB=XSTART;XA=(XS-BEDT)/(XT-BEDT); XF=(XE-BEDT)/(XT-BEDT)
6.000 10     INPUT XH
7.000      XKM=35.; XLEN=.01563; DEL=.00333
8.000      XM=(2*XH*XLEN**2/(XKM*DEL))**.5
9.000      IF(XT.NE.XS) GO TO 20
10.000     XB=-(XLEN/(XKM*DEL*.6.5))*(HEXP*.9543-XH*.5417*(.2618
11.000     &-12.*DEL))
12.000 20     XTHETA=XB*SINH(XM)/XM+XA*COSH(XM)
13.000      DT=XB*COSH(XM)+XA*XM*SINH(XM)
14.000      DT2=-XH*XLEN*XTHETA/XKM
15.000      XPR=(XTHETA*(XT-BEDT)+BEDT)
16.000      XX=ABS(XF-XTHETA)/XF
17.000      IF(XX.LT..001) GO TO 100
18.000      OUTPUT XPR,XE,XH,DT,DT2,'INPUT ANOTHER VALUE OF H'
19.000      GO TO 10
20.000 100    OUTPUT XPR,XE,XH,DT,DT2
21.000      GO TO 30
22.000      END
```

FIGURE 28. FIN MODELING PROGRAM

The slope at the beginning of the section is needed, because the end boundary condition is satisfied by adjusting the value of the heat transfer coefficient until it is matched. This can be obtained from equation 3 on page 42 of this thesis, with q_{fin} being calculated using equations 5 and 6 on page 43.

$$Q_{Fins} = Q_{Total} - Q_{Base} = -kN_{Fin}\delta B \frac{(T_w - T_b)}{L} \left. \frac{d\theta}{dx} \right|_{x=0}$$

Rearranging this equation and substituting the appropriate terms yields the expression for the slope at the base point:

$$\left. \frac{d\theta}{dx} \right|_{x=0} = - \frac{L}{kN_{Fin}\delta B} \frac{Q_{Total}}{(T_w - T_b)} - h_1 B (\pi d - N_{Fin}\delta)$$

where,

$$Q_{Base} = h_1 B (\pi d - N_{Fin}\delta) (T_w - T_b)$$

A trial and error procedure, varying h , matches the temperature at the first thermocouple on the fin. This h is the "sectionally averaged" heat transfer coefficient for the bare tube surface and the first section of the fin.

The second and third sections are done in a similar manner. The slope of the temperature profile, with respect to x , at the end of the first section is used as the starting slope for the next section. The value of h is varied until the modeled temperature matches the temperature recorded for the end of the section.

The start of the fourth section is the same as the second and third, but the end boundary condition is different. The end boundary condition to be matched in this case is,

$$\frac{d\theta}{dx} = - \frac{hL\theta}{k}$$

The heat transfer coefficient, h , is again varied until the end boundary condition is satisfied.

The analysis of the data using this program was not possible because the temperature profile obtained during the experimental runs was not smooth enough to give accurate values of the first derivative. Very accurate data is required for an analysis of this type. It would also be very helpful to have more thermocouples attached to the fin and also to have all of the thermocouples attached to the same fin. Additional

thermocouples would, however, effect the gas and particle flow past the fin, thus making the heat transfer conditions less realistic. This analysis could have been done using numerical methods, such as Frank's subroutines (11), but the results would still have been unuseable.

A second attempt was made at calculating the sectionally averaged heat transfer coefficients. In this calculation the data was smoothed by calculating the slope of the temperature at a point by adding the temperature of the preceding point to the temperature of the following point and then dividing by the distance between the two points. Using this method dT/dx could be found and then the amount of heat entering and leaving by conduction could be determined. The difference between these two values is the amount convected from the fin surface and with this value a heat transfer coefficient could be calculated.

This method was used to analyze the data and the results are tabulated in Table VI. Some of the low flow rate data is obviously wrong. Negative heat transfer coefficients are not feasible in this system. Higher

TABLE VI. RESULTS OF FIN MODELING

Q_{Total} (Btu/hr)	q_1 (Btu/hr)	q_2 (Btu/hr)	q_3 (Btu/hr)	h_1 (Btu/hr-ft ² -°F)	h_2 (Btu/hr-ft ² -°F)	h_3 (Btu/hr-ft ² -°F)
<u>Limestone data</u>						
3344.	2375.	851.	705.	20.08	2.13	12.29
3378.	2249.	381.	1960.	22.39	-26.27	73.66
3378.	2374.	683.	2027.	27.98	-24.78	86.78
3344.	2438.	997.	1635.	26.79	-14.25	78.54
3378.	2479.	1019.	1568.	29.23	-13.48	80.41
3412.	2557.	1355.	1030.	29.3	11.14	69.49
3447.	2575.	1344.	1019.	29.86	11.18	69.69
3447.	2576.	1372.	940.	30.52	15.96	69.13
3447.	2584.	1388.	957.	29.92	15.75	69.33
3412.	2556.	1378.	913.	30.72	17.95	69.67
<u>Glass Beads</u>						
3378.	2595.	1624.	347.	24.79	52.16	18.58
3412.	2898.	2296.	56.	13.86	98.90	2.42
3412.	2586.	1512.	790.	32.33	37.05	89.36
3412.	2543.	1444.	706.	37.93	46.66	112.06
3447.	2559.	1456.	706.	38.92	49.59	133.6

flow rate data was more consistent. An interesting comparison can be made between the glass bead data and the limestone particle data. The second heat transfer coefficient, h_2 , was lower than h_1 and h_3 for the limestone particle run, while for the glass bead run, h_2 was higher than h_1 , but lower than h_3 . This indicates that there must be a difference in the particle mode heat transfer contribution. This would seem reasonable because the small particles would be less effected by the narrow spacing between fins, while the larger particles would have considerably more problem entering and exiting the narrow space.

The heat transfer coefficients calculated from this analysis are all larger than the average coefficient. It would seem to me that the only explanation for this is that the temperatures measured are erroneous or that the fin modeling program does not calculate enough temperature drop along the fin.

APPENDIX B

CURVE FIT OF VOID FRACTION DATA

The void fraction of the limestone particle runs was calculated from bed expansion data and then a linear model was found to fit the void fraction data. The equation that was found is the following:

$$\text{Void fraction} = \frac{.119 (\text{Mass flow rate})}{(\text{Mass flow rate at minimum fluidization})} + .279$$

The mass flow rates were measured in the units of $\text{lb}_m/\text{hr-ft}^2$.

APPENDIX C

Date: July 1, 1981

Experimental Data
Steven D Hickel
Fluidized Bed

Run # 3

Fin height: medium large

Orientation: vertical, Large particles

Data point #	ΔP main inches	ΔP second inches	ΔP Bed inches	power watts	Bed height inches	Max d bubble inches	Bed Temp. °F	Tube Temp. °F
1	1.65	4.3	22.4	1000.0	18.0	1.0	105.25	675.0
2	2.85	4.25	22.2	1000.0	18.75	2.-4.0	131.00	480.0
3	3.96	4.20	22.2	1000.0	19.00	4.0-6.0	130.25	457.0
4	4.83	4.20	22.4	1000.0	19.50	6.0	132.00	460.0
5	5.90	4.10	22.4	1000.0	20.00	8.0	135.50	457.0
6	8.60	4.00	22.5	1020.0	20.50	8.0	135.00	435.0
7	10.70	3.90	22.6	1010.0	21.00	8.0	138.50	426.0
8	13.10	3.80	22.5	1020.0	21.25	8.0-10.	137.00	397.0
9	15.70	3.75	22.5	1020.0	22.25	12.0	139.00	391.0
10	17.40	3.65	22.5	1020.0	22.50	12.-14.	140.00	405.0

Total heat transfer area: .6835 ft²

Comments: I observed a 1/4 inch material loss during this run.

FIGURE 29. DATA TABLE

```

C    *** MASS FLOW CALCULATION PROGRAM ***
      DIMENSION DZ1(10),DZ2(10),DZ3(10),WATT(10),TS(10),TB(10)
      REAL LENGTH,N,KR
150  READ (10,150) DZ1,DZ2,DZ3,WATT,TS,TB
      FORMAT(10F,3)
      READ(10,160) FT; READ(10,160) DP; READ(10,160) XFINH
      READ(10,160) N; READ(10,160) XGMF; READ(10,160) DENP
160  FORMAT(4F,8)
100  FORMAT(22X,'DP',4X,'DP',3X,'REYNOLDS',3X,'G',5X,
1'Q/(TS-TB)',3X,'HTRUE',4X,'NUSSELT',2X,'G/GMF',4X,'GAS'
2,4X,'ARCHIMEDES',/,21X,'MAIN',2X,
3'SEC',3X,'NUMBER',2X,'(LB/HR-',2X,'(BTU/HR-',3X,
4'(BTU/HR-',2X,'NUMBER',11X,'PRANDTL',3X,'NUMBER',/,21X,
5'(IN)',2X,'(IN)',11X,
6'FT2)',4X,'FT2-F',5X,'FT2-F',21X,'NUMBER',//)
200  FORMAT(20X,F5.2,2X,F4.2,3X,F4.1,4X,F5.1,3X,F6.2,4X
1,F6.2,5X,F5.2,3X,F5.2,3X,F5.2,6X,F8.1,/)
      WRITE (108,100)
      KR=1.4; DW=62.18; XGRAV=4.173E08; THICK=.00333
      XX=1770.94; LENGTH=.54166; XKM=35.
      DO 20, I=1,N
        P1=XX+(DZ1(I)+DZ2(I))*5.182
        P2=XX+(DZ1(I)+DZ3(I))*5.182
        P3=XX+DZ1(I)*5.182
        B1=.608; B2=.741
        Y1=1-(P1-P3)*(.41-.35*B1**4)/(P1*KR)
        Y2=1-(P2-P3)*(.41-.35*B2**4)/(P2*KR)
        DA1=3.2949E-05*P1
        DA2=3.2949E-05*P2
        DA3=3.2949E-05*P3
        BX1=1-B1**4; BX2=1-B2**4
        G=17728.89*Y1*.01227*((DW-DA1)*DZ2(I)*DA1/((12*BX1)**.5
&+17728.89*Y2*.01227*((DW-DA2)*DZ3(I)*DA2/((12*BX2)**.5
        HEXP=WATT(I)*3.41443/((TS(I)-TB(I))*FT)
        XVIS=(2.45*(TB(I)-32.)+1538.1)*2.688E-05
        XRE=G*DP/XVIS
        IF(XFINH.EQ.0.0) HTRUE=HEXP;GO TO 70
        E1=HEXP; E2=HEXP+1.
        EE=E1; NY=0.
50    DIM=(2*EE*XFINH**2/(XKM*THICK))**.5
        EC=((12.*XKM*LENGTH*THICK*DIM*(EE*COSH(DIM)+XKM*DIM*SIN
1H(DIM)/XFINH)/(XFINH*(EE*SINH(DIM)+XKM*DIM*COSH(DIM)/X
2FINH)))+HEXP*FT)/(LENGTH*(3.14159/12.-12.*THICK))
        CALL CONV(EE,EC,1,NY)
        IF(NY-1)55,60,50
55    EE=E2; NY=2.; GO TO 50
60    HTRUE=(EE+EC)/2.
70    XKG=2.0E-05*TB(I)+.0134
        XNUS=HTRUE*DP/XKG; XGMF=G/XGMF; XPRAN=.240*XVIS/XKG
        XARCH=XGRAV*DF**3*DA3*(DENP-DA3)/XVIS**2
        WRITE(108,200)DZ2(I),DZ3(I),XRE,G,HEXP,HTRUE,XNUS,
&XGMF,XPRAN,XARCH
20    CONTINUE
      END

```

FIGURE 30. HEAT TRANSFER PROGRAM


```

1.000 22.4,22.2,22.4,22.4,22.5,22.5,22.5,22.6,22.6,22.6
2.000 1.55,2.9,4.0,4.9,5.95,8.6,10.75,13.15,15.7,17.3
3.000 4.4,4.2,4.2,4.1,4.1,4.0,3.8,3.8,3.7,3.6
4.000 980.0,1000.0,995.0,1000.0,1000.0,1000.0,1000.0,1020.0,1020.0,1020.0
5.000 790.0,513.0,396.0,368.0,383.0,383.0,383.0,386.0,414.0,430.0
6.000 103.5,125.25,127.0,128.25,129.5,130.0,132.0,135.0,136.0,137.0
7.000 .5841
8.000 .005208
9.000 .03125
10.000 10.0
11.000 457.2433
12.000 139.2
13.000      END

```

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FIGURE 31. HEAT TRANSFER PROGRAM DATA FILE

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
1.65	4.40	57.0	494.1	11.67	11.95	4.10	1.08	.71	249453.3
2.83	4.40	61.7	546.8	19.93	20.76	6.97	1.20	.71	238888.9
3.80	4.35	65.5	580.4	23.36	24.49	8.23	1.27	.71	238521.5
4.80	4.50	69.6	618.3	23.58	24.73	8.29	1.35	.71	237465.9
5.85	4.20	71.7	637.6	25.77	27.15	9.09	1.39	.71	237128.3
8.55	4.00	78.1	697.5	25.63	26.99	9.01	1.53	.72	235118.2
10.75	3.95	82.9	743.4	26.21	27.64	9.19	1.63	.72	233623.1
13.15	3.85	87.4	786.0	26.42	27.87	9.24	1.72	.72	231824.3
15.70	3.80	92.0	829.3	26.54	28.01	9.27	1.81	.72	230849.6
17.60	3.75	95.2	858.7	26.33	27.77	9.18	1.88	.72	230462.7

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FIGURE 32. RESULTS FROM RUN #1, LIMESTONE PARTICLES, $\frac{1}{4}$ INCH FIN HEIGHT

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
1.55	4.40	55.3	489.3	8.34	9.28	3.13	1.07	.71	240442.8
2.90	4.20	59.5	542.7	15.08	17.36	5.68	1.19	.72	226031.8
4.00	4.20	63.7	582.4	21.62	25.74	8.41	1.27	.72	225062.1
4.90	4.10	66.3	607.1	22.51	26.90	8.78	1.33	.72	224285.7
5.85	4.10	69.2	634.6	23.06	27.65	9.00	1.39	.72	223574.6
8.60	4.00	76.2	699.6	23.11	27.71	9.02	1.53	.72	223266.7
10.75	3.80	80.2	738.3	23.29	27.95	9.08	1.62	.72	222041.3
13.15	3.80	85.0	785.2	23.76	28.58	9.24	1.72	.72	220282.5
15.70	3.70	89.3	826.6	21.45	25.51	8.24	1.81	.72	219680.9
17.30	3.60	91.6	849.2	20.35	24.07	7.77	1.86	.72	219081.7

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FIGURE 33. RESULTS FROM RUN #2, LIMESTONE PARTICLES, 3/8 INCH FIN HEIGHT

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
1.65	4.30	55.4	490.9	8.77	9.48	3.18	1.07	.71	239243.9
2.85	4.25	59.0	542.6	14.31	16.25	5.28	1.19	.72	222469.5
3.98	4.20	63.3	581.4	15.29	17.50	5.70	1.27	.72	222929.3
4.83	4.20	66.1	608.7	15.23	17.43	5.66	1.33	.72	221980.4
5.90	4.10	68.8	635.9	15.54	17.83	5.76	1.39	.72	219860.7
8.60	4.00	75.7	699.6	16.99	19.74	6.38	1.53	.72	220222.0
10.70	3.90	79.8	741.3	17.55	20.49	6.60	1.62	.72	218187.5
13.10	3.80	84.6	784.1	19.60	23.30	7.52	1.72	.72	219021.6
15.70	3.75	89.2	828.4	20.22	24.17	7.78	1.81	.72	217830.9
17.40	3.65	91.6	852.7	19.23	22.78	7.32	1.87	.72	217239.2

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FIGURE 34. RESULTS FROM RUN #3, LIMESTONE PARTICLES, $\frac{1}{2}$ INCH FIN HEIGHT

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
1.60	4.35	55.1	490.2	8.73	10.47	3.51	1.07	.71	237950.0
2.85	4.35	59.2	546.3	11.07	13.94	4.52	1.20	.72	220644.2
3.90	4.20	62.7	579.1	11.85	15.16	4.91	1.27	.72	220161.6
4.85	4.20	65.7	609.4	12.46	16.15	5.21	1.33	.72	218663.0
5.88	4.10	68.8	635.3	13.84	18.45	5.98	1.39	.72	220826.0
8.75	4.00	75.9	703.0	14.72	19.99	6.45	1.54	.72	219021.6
10.75	3.85	79.9	740.2	14.87	20.25	6.54	1.62	.72	219021.6
13.30	3.75	84.5	785.8	15.22	20.87	6.71	1.72	.72	217239.2
15.80	3.70	89.1	828.1	15.06	20.59	6.63	1.81	.72	217830.9
17.40	3.60	91.2	850.7	15.42	21.24	6.81	1.86	.72	216063.1

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FIGURE 35. RESULTS FROM RUN #4, LIMESTONE PARTICLES, BARE TUBE.

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
2.80	4.30	59.5	542.6	37.16	37.16	12.16	1.19	.72	225686.2
3.90	4.30	63.9	582.8	44.71	44.71	14.64	1.28	.72	225999.2
4.85	3.80	65.1	594.1	43.46	43.46	14.22	1.30	.72	225810.1
5.90	4.10	69.5	636.1	38.90	38.90	12.69	1.39	.72	224563.9
8.60	4.00	75.9	699.7	37.68	37.68	12.22	1.53	.72	221493.2
10.65	4.00	80.7	744.4	37.50	37.50	12.15	1.63	.72	221007.8
13.30	4.00	86.3	796.0	37.86	37.86	12.26	1.74	.72	221007.3
15.60	3.80	89.3	828.8	38.02	38.02	12.25	1.81	.72	218485.0
17.40	3.70	92.0	854.9	38.86	38.86	12.51	1.87	.72	217890.7

FIGURE 36. RESULTS FROM RUN #5, LIMESTONE PARTICLES, 3/4 INCH FIN HEIGHT

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DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
.66	.00	1.8	108.1	21.35	25.76	1.31	1.86	.72	1032.5
2.10	.00	3.4	193.0	26.84	33.94	1.82	3.33	.72	1167.6
4.20	.00	5.0	273.1	29.25	37.75	2.06	4.71	.72	1215.2
8.00	.00	7.0	376.7	30.60	39.95	2.21	6.49	.72	1242.9
16.10	.00	9.9	535.6	30.90	40.44	2.25	9.24	.72	1254.5

FIGURE 37. RESULTS FROM RUN #6, GLASS
BEADS, 3/4 INCH FIN HEIGHT

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER	
.50	.00	1.5	94.1	13.47	17.83	.88	1.62	.73	969.9	
.92	.00	2.2	127.7	11.58	14.74	.77	2.20	.72	1098.3	I
2.00	.00	3.4	188.4	16.80	23.78	1.28	3.25	.72	1168.6	III
4.10	.00	4.9	269.9	18.73	27.55	1.51	4.65	.72	1224.6	I
8.40	.00	7.2	386.5	19.02	28.13	1.56	6.66	.72	1254.3	

FIGURE 38. RESULTS FROM RUN #7, GLASS
BEADS, $\frac{1}{2}$ INCH FIN HEIGHT

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
1.00	.00	2.3	132.8	24.79	27.92	1.47	2.29	.72	1116.3
2.00	.00	3.4	187.8	29.80	34.35	1.85	3.24	.72	1176.1
4.05	.00	4.9	267.3	32.62	38.09	2.09	4.61	.72	1215.6
8.10	.00	7.0	378.4	34.20	40.23	2.23	6.53	.72	1240.8
17.00	.00	10.2	550.2	35.00	41.33	2.30	9.49	.72	1254.2

FIGURE 39. RESULTS FROM RUN #8, GLASS
BEADS, 3/8 INCH FIN HEIGHT

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
1.10	.00	2.5	139.1	32.33	34.51	1.82	2.40	.72	1124.4
2.00	.00	3.4	187.7	36.64	39.44	2.13	3.24	.72	1174.0
4.05	.00	4.9	267.2	38.94	42.12	2.31	4.61	.72	1211.4
8.10	.00	7.0	378.4	41.43	45.02	2.49	6.52	.72	1239.6
16.30	.00	9.9	535.1	42.22	45.96	2.55	9.23	.72	1231.3

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FIGURE 40. RESULTS FROM RUN #9, GLASS
BEADS, $\frac{1}{4}$ INCH FIN HEIGHT

DP MAIN (IN)	DP SEC (IN)	REYNOLDS NUMBER	G (LB/HR- FT ²)	Q/(TS-TB) (BTU/HR- FT ² -F)	HTRUE (BTU/HR- FT ² -F)	NUSSELT NUMBER	G/GMF	GAS PRANDTL NUMBER	ARCHIMEDES NUMBER
1.07	.00	2.4	137.1	77.67	77.67	4.10	2.36	.72	1119.3
2.00	.00	3.4	187.5	77.23	77.23	4.17	3.23	.72	1172.7
4.10	.00	4.9	268.7	67.99	67.99	3.72	4.63	.72	1209.6
8.10	.00	7.0	378.3	73.48	73.48	4.06	6.52	.72	1234.8
16.20	.00	10.0	536.7	69.52	69.52	3.86	9.25	.72	1249.9

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FIGURE 41. RESULTS FROM RUN #10, GLASS BEADS, BARE TUBE.

```

1.000 C      CORRELATION FINDER
2.000 C
3.000      DIMENSION X(5,10),XL(5),XH(5),XC(5),XX(5),FUNC(10)
4.000      1,XP(5,10),XXP(5)
5.000      DIMENSION XNU(74),RE(74),GGMF(74),XLEN(74),AR(74),
6.000      1XER(74),CALCNU(74)
7.000      COMMON/STEVE/ RE,GGMF,AR,XLEN,XNU
8.000      COMMON/JANET/ CALCNU, NN
9.000      SAVE /JANET/
10.000 1      INPUT NN,N,KPRINT,EPSI
11.000      DO 110, I=1,N
12.000 110     INPUT X(I,1)
13.000      DO 120, I=1,N
14.000 120     INPUT XL(I)
15.000      DO 130, I=1,N
16.000 130     INPUT XH(I)
17.000      M=1; KK=10
18.000      DO 10 I=1,NN
19.000      READ(10,150) RE(I),XNU(I),GGMF(I),AR(I),XLEN(I)
20.000 10      CONTINUE
21.000 150     FORMAT(5F,5)
22.000      CALL CMBOX(M,N,KK,KPRINT,X,XL,XH,XC,XX,XP,XXP,FUNC,
23.000      1EPSI)
24.000      DO 200 I=1,NN
25.000      XER(I)=(XNU(I)-CALCNU(I))/CALCNU(I)*100.
26.000 200     CONTINUE
27.000      WRITE(108,300); WRITE(108,400); WRITE(108,500)
28.000 300     FORMAT(20X,'COMPARISON CHART',/)
29.000 400     FORMAT(4X,'EXPERIMENTAL      MODEL',10X,'PERCENT')
30.000 500     FORMAT(7X,'NUSSSELT',7X,'NUSSSELT',8X,'DIFFERENCE',/)
31.000      DO 600 I=1,NN
32.000      WRITE(108,700) XNU(I),CALCNU(I),XER(I)
33.000 700     FORMAT(6X,F6.2,10X,F6.2,10X,F6.2)
34.000 600     CONTINUE
35.000      GO TO 1
36.000      END

```

FIGURE 42. CORRELATION FINDER PROGRAM


```

37.000 SUBROUTINE OBJ(X,N,XP,OB,KOB)
38.000 SAVE /JANET/
39.000 DIMENSION X(N),XP(N)
40.000 COMMON/STEVE/ RE(74),GGMF(74),AR(74),XLEN(74),XNU(74)
41.000 COMMON/JANET/ CALCNU(74), NN
42.000 XDEL=0.; KOB=KOB+1.
43.000 DO 100 I=1,NN
44.000 IF(XLEN(I).EQ.0.) DEL=0.0; GO TO 1000
45.000 DEL=.00333
45.500 XVOID=.119066*GGMF(I)+.27919
46.000 1000 CALCNU(I)=X(1)*RE(I)**X(2)*(1-XVOID)**X(3)
47.000 &*(.0625/XLEN(I))**X(4)
48.000 XDEL=(XNU(I)-CALCNU(I))**2+XDEL
49.000 100 CONTINUE
50.000 OB=-XDEL
51.000 DO 2 I=1,N
52.000 2 XP(I)=X(I)
53.000 OUTPUT OB
54.000 RETURN
55.000 END
56.000 SUBROUTINE CONSTR(X,N,IVI)
57.000 IVI=0.
58.000 DIMENSION X(N)
59.000 G1=X(1)
60.000 IF(G1.LT.0.) IVI=1.; RETURN
61.000 G2=X(2)
62.000 IF(G2.LT.0.0) IVI=1.; RETURN
63.000 G3=X(3)
64.000 IF(G3.LT.0.) IVI=1.; RETURN
65.000 G4=X(4)
66.000 IF(G4.LT.0.) IVI=1.; RETURN
70.000 RETURN
71.000 END

```

FIGURE 43. CORRELATION FINDER PROGRAM, PART B

Glass Bead Data

50.000 1.5, .88, 1.62, 969.9, .0625
 51.000 2.2, .77, 2.20, 1098.3, .0625
 52.000 3.4, 1.28, 3.25, 1168.6, .0625
 53.000 4.9, 1.51, 4.65, 1224.6, .0625
 54.000 7.2, 1.56, 6.66, 1254.3, .0625
 55.000 1.8, 1.31, 1.86, 1032.5, .04167
 56.000 3.4, 1.82, 3.33, 1167.6, .04167
 57.000 5.0, 2.06, 4.71, 1215.2, .04167
 58.000 7.0, 2.21, 6.49, 1242.9, .04167
 59.000 9.9, 2.25, 9.24, 1254.5, .04167
 60.000 2.3, 1.47, 2.29, 1116.3, .03125
 61.000 3.4, 1.85, 3.24, 1176.1, .03125
 62.000 4.9, 2.09, 4.61, 1215.6, .03125
 63.000 7.0, 2.23, 6.53, 1240.8, .03125
 64.000 10.2, 2.3, 9.49, 1254.2, .03125
 65.000 2.5, 1.82, 2.4, 1124.4, .02083
 66.000 3.4, 2.13, 3.24, 1174.0, .02083
 67.000 4.9, 2.31, 4.61, 1211.4, .02083
 68.000 7.0, 2.49, 6.52, 1239.6, .02083
 69.000 9.9, 2.55, 9.23, 1231.3, .02083
 70.000 2.4, 4.1, 2.36, 1119.3, 0.0

Limestone Particle Data

1.000 57.0, 4.1, 1.08, 249453.3, .02083
 2.000 61.7, 6.97, 1.2, 238888.9, .02083
 3.000 65.5, 8.23, 1.27, 238521.5, .02083
 4.000 69.6, 8.29, 1.35, 237465.9, .02083
 5.000 71.7, 9.09, 1.39, 237128.3, .02083
 6.000 78.1, 9.01, 1.53, 235118.2, .02083
 7.000 82.9, 9.19, 1.63, 233623.1, .02083
 8.000 87.4, 9.24, 1.72, 231824.3, .02083
 9.000 92.0, 9.27, 1.81, 230849.6, .02083
 10.000 95.2, 9.18, 1.88, 230462.7, .02083
 11.000 55.3, 3.13, 1.07, 240442.8, .03125
 12.000 59.5, 5.68, 1.19, 226031.8, .03125
 13.000 63.7, 8.41, 1.27, 225062.1, .03125
 14.000 66.3, 8.78, 1.33, 224285.7, .03125
 15.000 69.2, 9.0, 1.39, 223574.6, .03125
 16.000 76.2, 9.02, 1.53, 223266.7, .03125
 17.000 80.2, 9.08, 1.62, 222041.3, .03125
 18.000 85.0, 9.24, 1.72, 220282.5, .03125
 19.000 89.3, 8.24, 1.81, 219680.9, .03125
 20.000 91.6, 7.77, 1.86, 219081.7, .03125
 21.000 55.4, 3.18, 1.07, 239243.9, .04167
 22.000 59.0, 5.28, 1.19, 222469.5, .04167
 23.000 63.3, 5.7, 1.27, 222929.3, .04167
 24.000 66.1, 5.66, 1.33, 221980.4, .04167
 25.000 68.8, 5.76, 1.39, 219860.7, .04167
 26.000 75.7, 6.38, 1.53, 220222.0, .04167
 27.000 79.8, 6.60, 1.62, 218187.5, .04167
 28.000 84.6, 7.52, 1.72, 219021.6, .04167
 29.000 89.2, 7.78, 1.81, 217830.9, .04167
 30.000 91.6, 7.32, 1.87, 217239.2, .04167
 40.000 55.1, 3.51, 1.07, 237950.0, .0625
 41.000 59.2, 4.52, 1.2, 220644.2, .0625
 42.000 62.7, 4.91, 1.27, 220161.6, .0625
 43.000 65.7, 5.21, 1.33, 218663.0, .0625
 44.000 68.8, 5.98, 1.39, 220826.0, .0625
 45.000 75.9, 6.45, 1.54, 219021.6, .0625
 46.000 79.9, 6.54, 1.62, 219021.6, .0625
 47.000 84.5, 6.71, 1.72, 217239.2, .0625
 48.000 89.1, 6.63, 1.81, 217830.9, .0625
 49.000 91.2, 6.81, 1.86, 216063.1, .0625

FIGURE 44. DATA FILES USED IN THE CORRELATION PROGRAMS

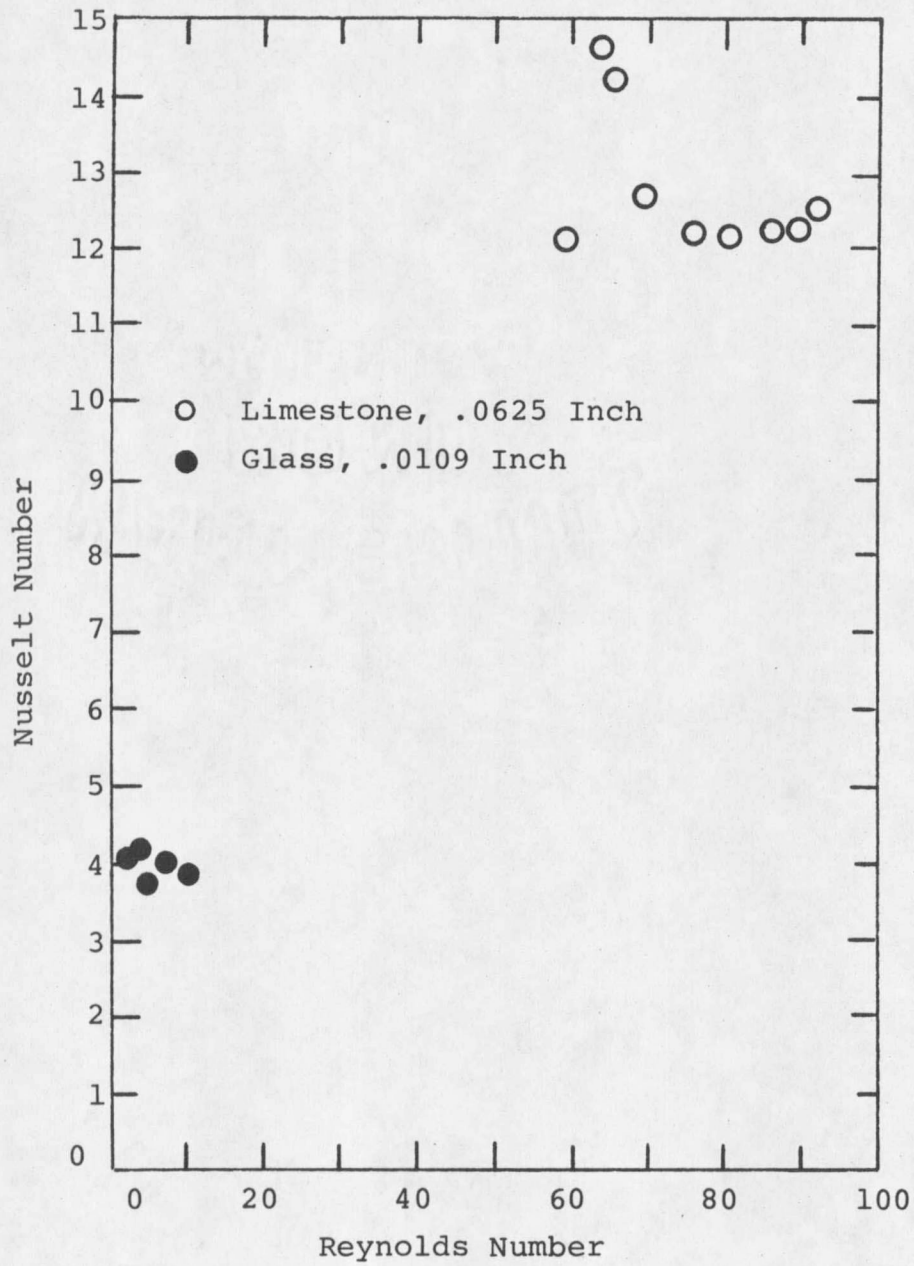


FIGURE 45. Nu VERSUS Re VERSUS PARTICLE DIAMETER (Tube #1)

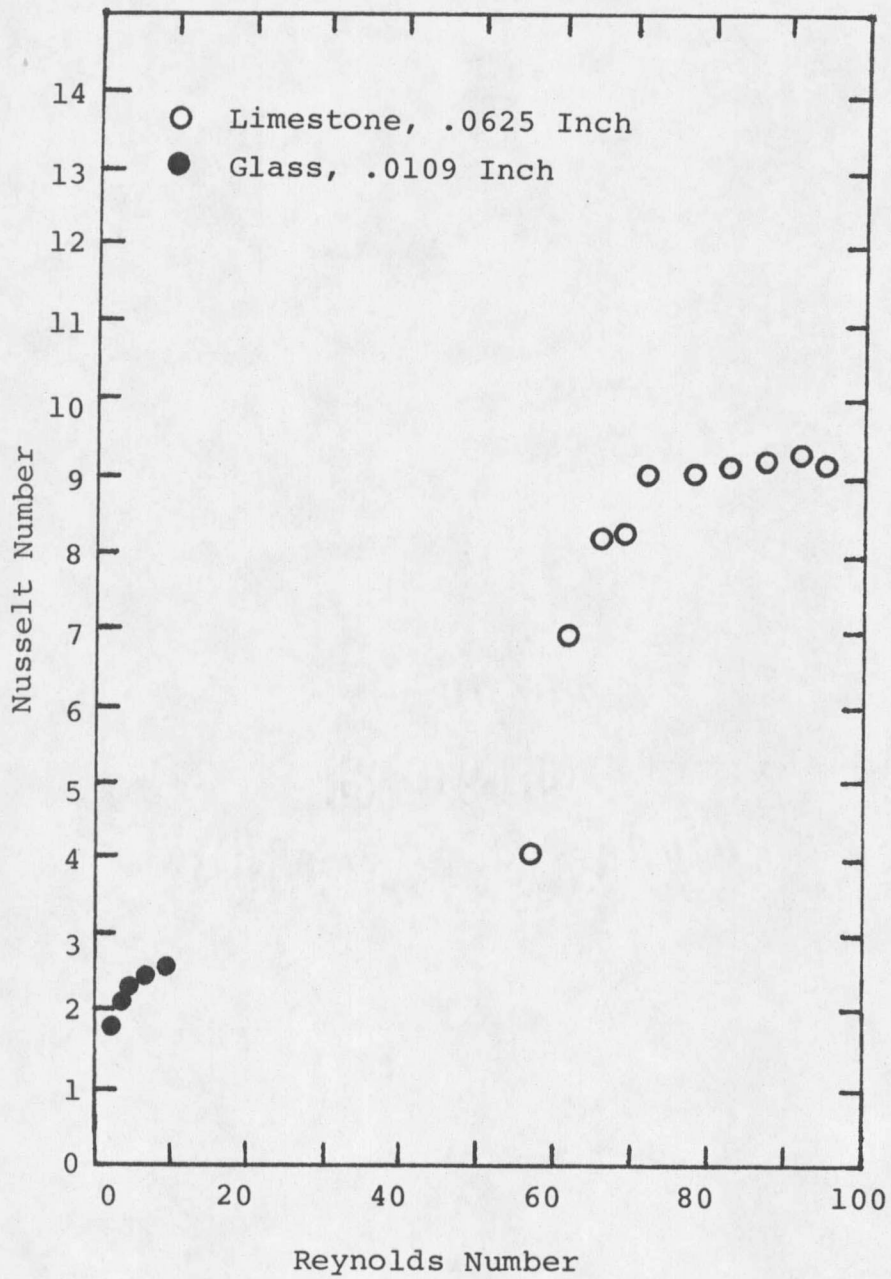


FIGURE 46. Nu VERSUS Re VERSUS PARTICLE DIAMETER (Tube #2)

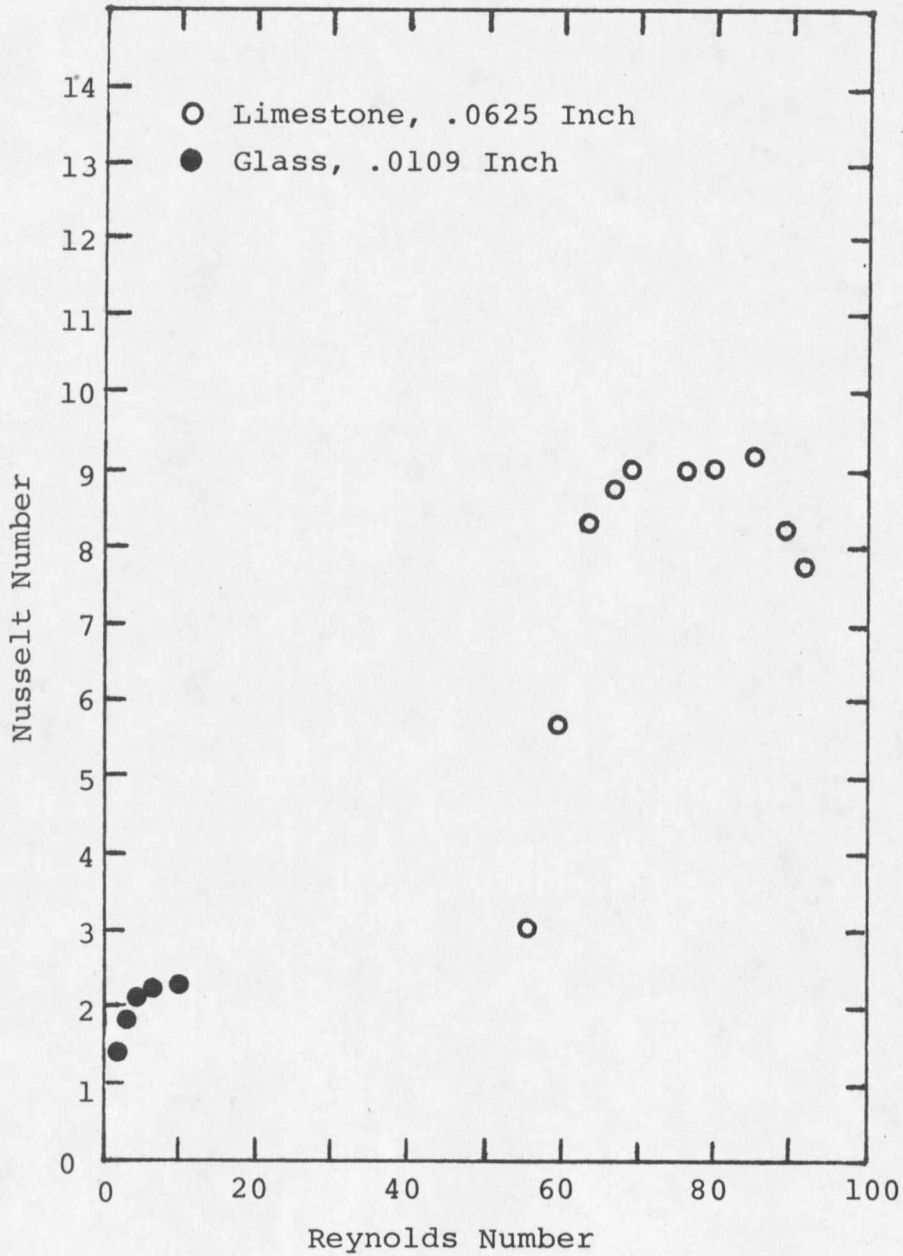


FIGURE 47. Nu VERSUS Re VERSUS PARTICLE DIAMETER (Tube #3)

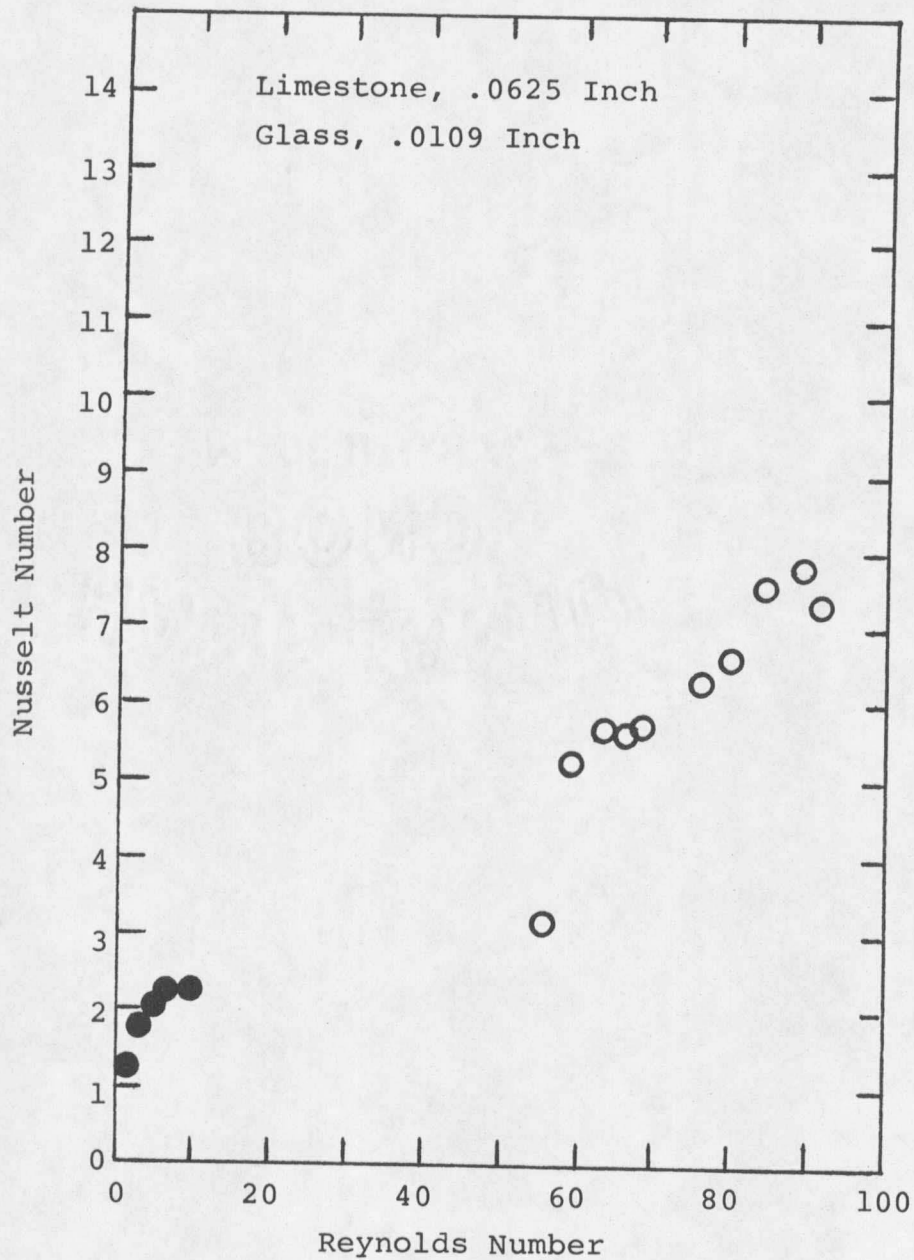


FIGURE 48. Nu VERSUS Re VERSUS PARTICLE DIAMETER (Tube #4)

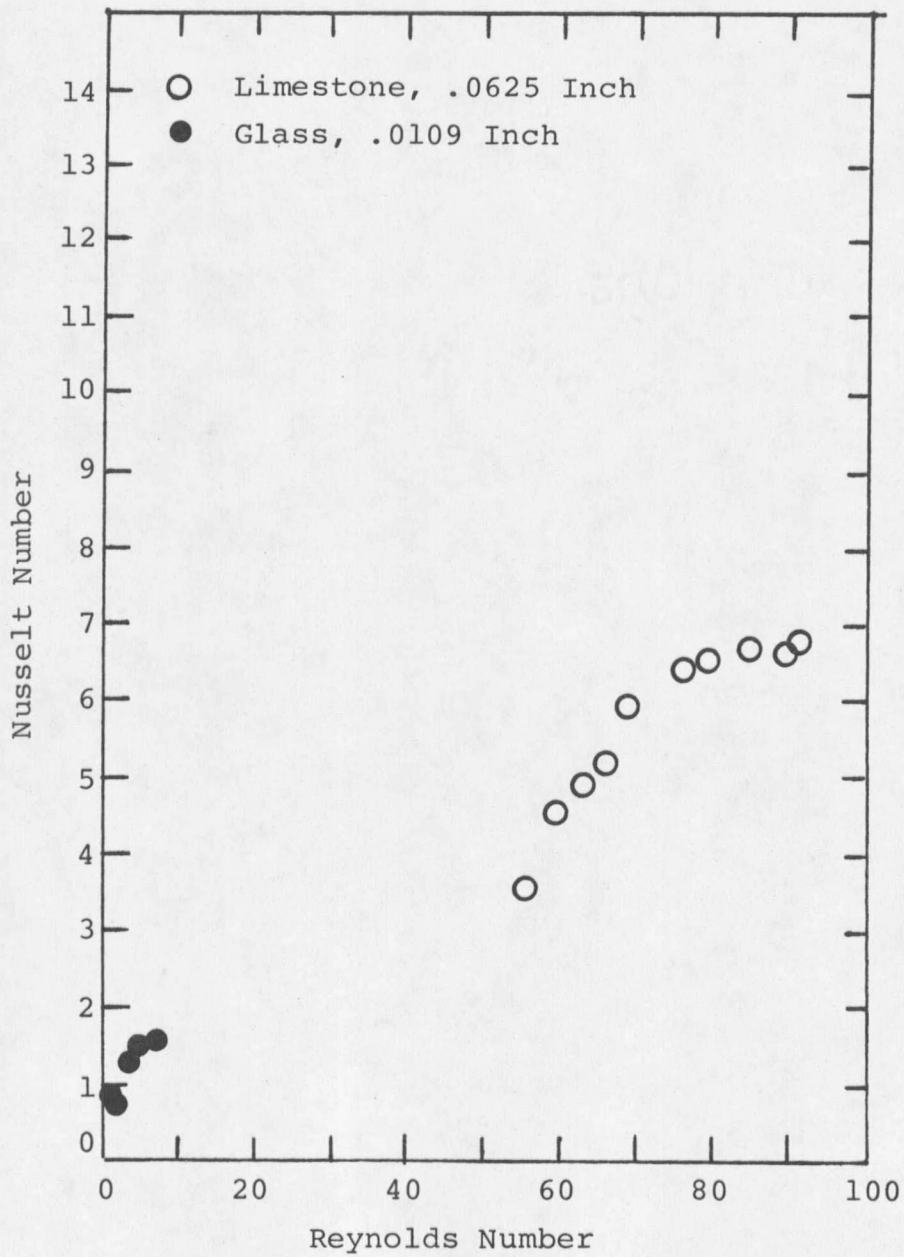


FIGURE 49. Nu VERSUS Re VERSUS PARTICLE DIAMETER (Tube #5)

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H525 Hickel, S. D.
cop.2 Heat transfer from
 single, vertical,
 longitudinally ...

DATE	ISSUED TO
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