



Heat transfer from a vertical bundle of continuous, helical finned tubes and from coiled spiral tubes in an air fluidized bed
by Steven Paul Yurich

A thesis submitted in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE
in Chemical Engineering
Montana State University
© Copyright by Steven Paul Yurich (1979)

Abstract:

Heat transfer coefficients were measured in two systems; one in which heat was transferred from a bundle of continuous helical copper finned tubes to a cylindrical air fluidized glass particle bed and another in which heat was transferred from coiled spiral tubes to the fluidized bed.

Experimental parameters for the two systems included tube geometry, bed particle diameter, and air fluidizing velocity.

Results for the helical finned tubes indicate that the coefficient generally increased with increasing fluid-izing velocity. A maximum coefficient was observed in some cases. The coefficient increased with decreasing particle size. The coefficient increased with increasing fin spacing and decreasing fin height. The coefficient was very sensitive to fin spacings greater than 8 particle diameters and became less sensitive to fin spacings less than 8 particle diameters. The coefficient increased with an increase in the tube bundle center-to-center spacing, until the tubes became located close to the column wall, then the coefficient decreased. Performance increased with increasing air fluidizing velocity, increased fin spacing, and decreased fin height. An average gain in heat transfer of up to 135 percent was obtained with helical finned tubes compared to a bare tube.

Results for the coiled spiral tubes indicated that the coefficient also increased with increasing fluidizing velocity and decreasing particle diameter. The coefficient increased with decreasing groove depth and increasing flute pitch. Most of the data fell within plus or minus 15 percent of a correlation relating experimental parameters to the heat transfer coefficient.

HEAT TRANSFER FROM A VERTICAL BUNDLE OF CONTINUOUS,
HELICAL FINNED TUBES AND FROM COILED SPIRAL TUBES
IN AN AIR FLUIDIZED BED

by

STEVEN PAUL YURICH

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

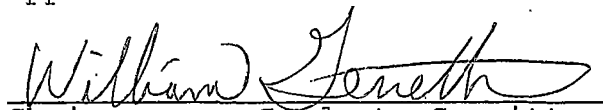
of

MASTER OF SCIENCE

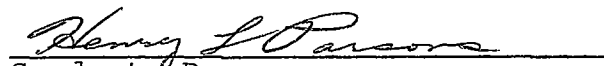
in

Chemical Engineering

Approved:


Chairperson, Graduate Committee


Head, Major Department


Graduate Dean

MONTANA STATE UNIVERSITY
Bozeman, Montana

August, 1979

STATEMENT OF PERMISSION TO COPY

In presenting this thesis in partial fulfillment of the requirements for an advanced degree at Montana State University, I agree that the Library shall make it freely available for inspection. I further agree that permission for extensive copying of this thesis for scholarly purposes may be granted by my major professor, or, in his absence, by the Director of Libraries. It is understood that any copying or publication of this thesis for financial gain shall not be allowed without my written permission.

Signature St. P. Yarnall
Date Aug 33, 1979

ACKNOWLEDGMENT

The author wishes to thank the staff of the Department of Chemical Engineering at Montana State University for the help given in this research.

Special thanks go to Dr. W. E. Genetti, who directed and aided me in all phases of this project.

The author wishes to thank his wife and parents for their encouragement and support.

Finally, the author would like to acknowledge the National Science Foundation, which provided the funding for this project.

TABLE OF CONTENTS

	<u>Page</u>
VITA	ii
ACKNOWLEDGEMENT	iii
LIST OF TABLES	vi
LIST OF FIGURES	vii
ABSTRACT	x
INTRODUCTION	1
THEORY AND PREVIOUS RELATED RESEARCH	7
Mechanism of Fluidization for Heat Transfer	7
Previous Related Research	15
EXPERIMENTAL APPARATUS	19
Helical Finned Tube System	19
Coiled Spiral Tube System	31
EXPERIMENTAL PROCEDURE	42
Minimum Fluidization Velocities	42
Typical Run Procedure for the Helical Finned Tube	44
Typical Run Procedure for the Coiled Spiral Tube	45
DEVELOPMENT OF THEORETICAL MODEL FOR HELICAL FINNED TUBES	47

	<u>Page</u>
RESULTS AND DISCUSSION	57
Helical Finned Tube System	57
Coiled Spiral Tube System	76
ERROR ANALYSIS	89
Helical Finned Tube System	89
Coiled Spiral Tube System	90
CALCULATIONS	92
Common Calculations to Both Systems	92
Helical Finned Tube System	95
Coiled Spiral Tube System	96
CONCLUSIONS	100
APPENDICES	103
NOMENCLATURE	115
BIBLIOGRAPHY	119

LIST OF TABLES

<u>Table</u>	<u>Page</u>
I PREVIOUS RELATED RESEARCH AT MONTANA STATE UNIVERSITY	18
II BLAST-O-LITE BEAD SIZE ANALYSIS	26
III HELICAL FINNED TUBE DIMENSIONS	32
IV COILED SPIRAL TUBE DIMENSIONS	41
V PERFORMANCE OF HELICAL FINNED TUBES AT $G/G_{nf}=4$	77
IV RANGE OF CORRELATION APPLICABILITY	82

LIST OF FIGURES

<u>Figure</u>		<u>Page</u>
1	FILM MODEL FOR HEAT TRANSFER	9
2	PACKET MODEL FOR HEAT TRANSFER	10
3	PARTICLE MODEL FOR HEAT TRANSFER	13
4	SCHEMATIC VIEW OF HELICAL TUBE SYSTEM . . .	20
5	COLUMN OF HELICAL FINNED TUBE SYSTEM	21
6	DETAILED VIEW OF HELICAL FINNED TUBE COLUMN	22
7	DETAILS OF A CARTRIDGE HEATER	27
8	CARTRIDGE HEATER AND FINNED TUBE	29
9	HELICAL FINNED TUBE DETAILS AND NOMENCLATURE	33
10	SCHEMATIC VIEW OF COILED SPIRAL TUBE SYSTEM	34
11	COLUMN OF COILED SPIRAL TUBE SYSTEM	35
12	DETAILED VIEW OF COILED SPIRAL TUBE COLUMN .	37
13	COILED SPIRAL TUBE	39
14	PARTICLE MINIMUM FLUIDIZATION VELOCITIES . .	43
15	PROPOSED MODEL OF HELICAL FIN	49
16	ANALYTICAL SOLUTION FOR Q/ T, No. 1, No. 3, No. 4 TUBES	54
17	ANALYTICAL SOLUTION FOR Q/ T, No. 2, No. 5, No. 7 TUBES	55
18	ANALYTICAL SOLUTION FOR Q/ T, No. 6, No. 8 TUBES	56

<u>Figure</u>		<u>Page</u>
19	HORIZONTAL TUBE CORRELATION WITH VERTICAL TUBE DATA	58
20	h_{model} VERSUS G VERSUS PARTICLE DIAMETER, (TUBE No. 7)	61
21	h_{model} VERSUS G VERSUS FIN SPACING (S)	63
22	h_{model} VERSUS D_p/S (CONSTANT FIN HEIGHT)	64
23	h_{model} VERSUS G VERSUS FIN HEIGHT (L)	67
24	h_{model} VERSUS G VERSUS CENTER-TO-CENTER TUBE SPACING (TUBE No. 8)	69
25	h_{model} VERSUS G VERSUS CENTER-TO-CENTER TUBE SPACING (TUBE No. 8)	70
26	PERFORMANCE VERSUS G VERSUS FIN SPACING (S)	73
27	PERFORMANCE VERSUS G VERSUS FIN HEIGHT (L)	74
28	PERFORMANCE VERSUS G VERSUS CENTER-TO-CENTER TUBE SPACING	75
29	WILSON PLOT ANALYSIS	79
30	CORRELATION FOR SPIRAL TUBES	81
31	h_o VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 5A)	83
32	h_o VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 6A)	84
33	h_o VERSUS G VERSUS GROOVE DEPTH AND PITCH	86

<u>Figure</u>		<u>Page</u>
34	PERFORMANCE RATIO FOR COILED SPIRAL TUBES . .	88
35	h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 1)	104
36	h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 2)	105
37	h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 3)	106
38	h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 4)	107
39	h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 5)	108
40	h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 6)	109
41	h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 8)	110
42	h_{model} VERSUS G VERSUS FIN SPACING (S) . . .	111
43	h_{model} VERSUS G VERSUS FIN SPACING (S) . . .	112
44	h_{model} VERSUS G VERSUS FIN HEIGHT (L)	113
45	h_{model} VERSUS G VERSUS FIN HEIGHT (L)	114

ABSTRACT

Heat transfer coefficients were measured in two systems; one in which heat was transferred from a bundle of continuous helical copper finned tubes to a cylindrical air fluidized glass particle bed and another in which heat was transferred from coiled spiral tubes to the fluidized bed.

Experimental parameters for the two systems included tube geometry, bed particle diameter, and air fluidizing velocity.

Results for the helical finned tubes indicate that the coefficient generally increased with increasing fluidizing velocity. A maximum coefficient was observed in some cases. The coefficient increased with decreasing particle size. The coefficient increased with increasing fin spacing and decreasing fin height. The coefficient was very sensitive to fin spacings greater than 8 particle diameters and became less sensitive to fin spacings less than 8 particle diameters. The coefficient increased with an increase in the tube bundle center-to-center spacing, until the tubes became located close to the column wall, then the coefficient decreased. Performance increased with increasing air fluidizing velocity, increased fin spacing, and decreased fin height. An average gain in heat transfer of up to 135 percent was obtained with helical finned tubes compared to a bare tube.

Results for the coiled spiral tubes indicated that the coefficient also increased with increasing fluidizing velocity and decreasing particle diameter. The coefficient increased with decreasing groove depth and increasing flute pitch. Most of the data fell within plus or minus 15 percent of a correlation relating experimental parameters to the heat transfer coefficient.

INTRODUCTION

In the past 20 years the contributions to the technique of fluidization have been numerous, with the number of theoretical and experimental investigations varied in many directions. The use of fluidized bed equipment has opened wide possibilities for improving various industrial technologies.

The so-called fluidized bed results when a fluid is caused to flow upward through a bed of suitable-sized particles at a velocity sufficiently high to buoy the particles and to impart to them a fluid-like motion. Therefore, a fluidized bed is a relative stable condition of fluid-solid contacting which is intermediate to a packed column on one hand, and pneumatic transport on the other (1).

At low fluidizing velocities, the fluid merely passes through the bed as in a packed bed. The bed of particles offers resistance to the fluid flow through it. As the velocity of the flow increases, the drag exerted on the particles increases. With the fluid flowing upward through the bed, the drag force will tend to cause the particles to rearrange themselves within the bed to offer less resistance to the fluid flow. This causes

the bed of particles to expand. With further increase in the upward fluid velocity, the expansion continues and a stage will be reached where the drag force exerted on the particles will be sufficient to support the weight of the particles. In this state, the fluid-particle system begins to behave like a fluid and will flow under a hydrostatic head. This is the point of incipient fluidization. The velocity corresponding to this condition is termed the minimum fluidization velocity. The pressure drop across the bed at this point will be equal to the weight of the bed (2).

As the fluidizing medium velocity is increased beyond the point of minimum fluidization, the fluid bubbles expand and coalesce as they rise upward through the bed and burst as they reach the upper bed surface. The bubble action tends to agitate the bed, increasing the random motion of the particles. The uniform expansion behavior of the bed is lost. This condition of a freely-bubbling bed is known as aggregative fluidization.

The final regime of fluidization, known as slugging, is evident when the fluid velocity is increased until the size of the bubble diameter approaches the size of the

column. Layers of fluidizing medium are seen rising in the column in piston-like action. Maximum particle motion occurs during slugging. Slugging is usually undesirable since it increases the problems of entrainment and lowers the performance potential of the bed for both physical and chemical operations (3).

All three regimes of fluidization were observed in this investigation.

The fluidized bed has both desirable and undesirable characteristics. Listed below are some of the advantages and disadvantages (4):

Advantages

1. Due to the intense agitation in a well-fluidized bed, local temperatures and solids distribution are much more uniform than in fixed beds. This may be important in many chemical and catalytic processes.

2. Fluidization allows easier implementation of continuous recycling of solids.

3. Increased motion of the particles past internal or external heat transfer surfaces results in heat transfer coefficients much higher in a fluidized bed than in a fixed bed operating under comparable flow conditions. This makes temperature control easier.

4. Use of a fluidized bed lowers maintenance costs because of few movable parts.

Disadvantages

1. Because relative fluid and particle motion is basically co-current, the driving force is not completely favorable, and the fluidized bed acts as a single stage. Multiple beds can be employed, but this runs into more expense.

2. In fluidized operation equipment, erosion may be serious. Special and generally expensive designs may be required to eliminate or minimize wear in reactors and transfer lines.

3. Particle degradation and elutriation may cause severe catalyst losses.

4. Space velocity through the column is limited because the bed fluidizes in a narrow range of velocities. A fluid reactor is in this respect restricted, whereas the fixed bed offers a greater degree of freedom and adjustment of space velocity.

Despite some of its serious drawbacks, the use of fluidized beds in industrial operations has been widespread. The use of a fluidized bed by the U.S. Petroleum industry in the catalytic cracking of oil (4) and in the

catalytic reformer (3) has led to a technical revolution in this field. In recent years there is an intensive effort on research and development to explore the advantages offered by fluidized bed technology to the combustion and conversion of coal (5). Fluidized beds are currently being looked at for the incineration of carbonaceous industrial wastes (6) and the incineration of petroleum refinery wastes (7). Fluidized beds are presently being used in the following industrial operations: the calcining of nuclear wastes; the roasting of sulfide ores; as a heat exchanger to cool hot alumina particles; in the production of alkyl chloride, acrylonitrile, phthalic anhydride, vinyl acetate monomer; and the oxidation of ethylene. These few applications just mentioned are in no way complete, but do show some of the diversity of uses for the fluidized bed in industry today.

As can be seen from these examples, most applications require energy to be transferred to or from the fluidized bed. Heat transfer surfaces immersed in the bed are an efficient method for transferring energy.

The objectives of this investigation are in two parts: The first part was to experimentally study and

analyze heat transfer from vertical, continuous, helical, copper finned tubes arranged in a bundle in an air fluidized bed. The bed material consisted of glass beads of controlled diameters. Experimental variables included fin height, fin spacing, particle diameter, fluidizing gas mass velocity, and tube bundle center-to-center spacing. The second part was to finish the investigation started by David Everly, analyzing heat transfer from coiled spiral copper tubes in an air fluidized bed. Again, the bed material consisted of glass beads of controlled diameters. Experimental variables for this part included groove depth, number of flutes, flute pitch, particle diameter, and gas mass velocity.

THEORY AND PREVIOUS RELATED RESEARCH

The theory and previous research from extended surfaces in fluidized beds is presented in two parts. The first section presents proposed mechanisms for heat transfer from immersed surfaces; the second section describes previous research with immersed heating surfaces.

Mechanism of Fluidization for Heat Transfer

Various researchers (2,3,4,8) have investigated the phenomenon of bubbling and the associated particle motion in fluidized beds. The phenomenon of bubbling is a striking and obvious feature of the gas fluidized beds. Bubbles in gas fluidized beds are very important for they are responsible for most of the features that differentiate a packed bed from a fluidized bed. They modify gas flow through the system that causes particle movement which generally results in rapid and extensive particle mixing. A direct consequence of this is very high heat transfer coefficients that can be obtained between the bed and immersed surfaces.

Several models based on various controlling heat transfer resistances have been presented to explain these high values of heat transfer coefficients.

Levenspiel and Walton (9) presented a "film" model, Figure 1. In the film model a thin laminar film of fluidizing gas is next to the surface. The major resistance to heat flow is considered to be in this film. The scouring action of the fluidized particles against the film decreases its thickness, thereby decreasing the resistance to heat flow. Levenspiel and Walton derived a simple expression in terms of the modified Nusselt and Reynolds numbers for the effective gas film thickness on the assumption that the film is broken whenever a particle touches the transfer surface. They then predicted the overall heat transfer coefficient that would be obtained if the heat transfer was limited by the average thickness of the gas film developing between the points of contact where the film is broken. In order to make their model fit their experimentally determined coefficients, they found it necessary to modify the power to which the Reynolds group was raised and the constant that they had predicted for their expression.

Mickley and Fairbanks (10) proposed a "packet" model, Figure 2. In their model, packets or an emulsion of particles at the bulk bed temperature T_b moves into contact with the transfer surface at a higher

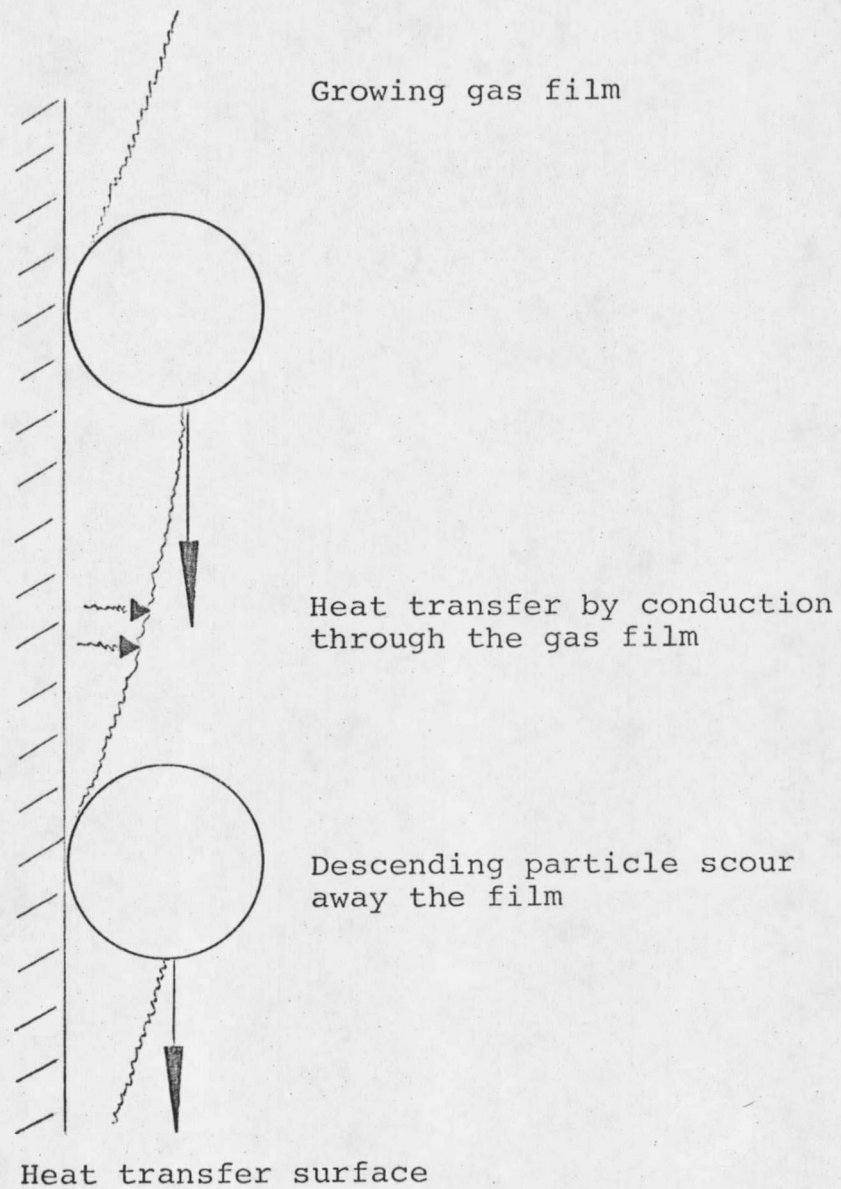


Figure I. FILM MODEL FOR HEAT TRANSFER

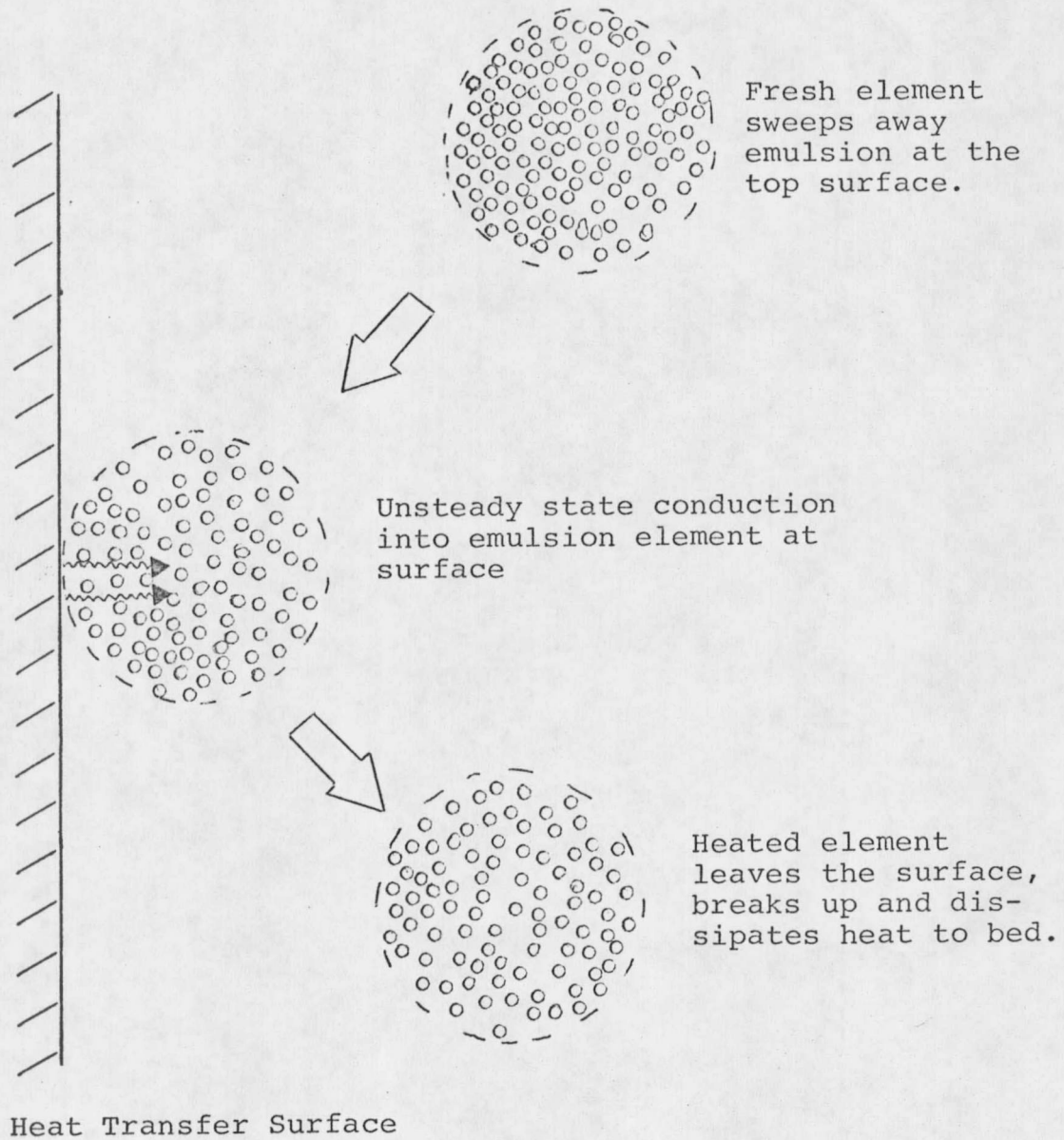


Figure 2. PACKET MODEL FOR HEAT TRANSFER

temperature T_w . Unsteady state conduction from the transfer surface to the packet of particles begins on contact. This conduction of energy into the packet is the controlling resistance. After a short duration of contact with the transfer surface, the packets are visualized as leaving the surface, breaking up, and dissipating heat to the bulk of the bed. The packet properties were assumed to be those of a quiescent bed. In their own experiments, Mickley and Fairbanks found that bed-to-surface transfer coefficients were proportional to the square root of the thermal conductivity of the quiescent bed, as their unsteady state diffusion model predicted. The simplifying assumption that the packet of gas and particles can be treated as a uniform medium, with the thermal properties of the bed at incipient fluidization is obviously unrealistic in the neighborhood of the heat transfer surface, because of the effect of the surface on local particle packing.

Mickley and Fairbanks' proposed packet model was later modified by Ziegler, Koppel, and Brazelton (11). This new model was extended by Genetti and Knudsen (12). Under this new model, the physical properties of the

solids and fluids are constant. The fluidized particles are spheres of uniform diameter. Particles from the bulk of the fluidized bed, having the bulk medium temperature, T_b , move adjacent to the transfer surface, while adjacent to the surface the particle receives energy by convection from the fluid around the particle. This fluid around the particle is assumed to be at the arithmetic mean of the transfer surface temperature and the bulk medium temperature. After some time, the particle leaves the surface and returns to the bulk of the bed. Conduction at the point of contact has been shown to be very small and can be neglected. This mechanism is sketched for a typical particle on figure 3. Ziegler, Koppel, and Brazelton (11) developed the following formula to describe the rate of heat transfer from a surface in a fluidized bed:

$$Nu_p = \frac{h_o D_p}{k_g} = \frac{7.20}{\left[1 + \frac{6k_g \theta}{\rho_s c_{ps} D_p^2} \right]^2}$$

Heat Transfer Surface

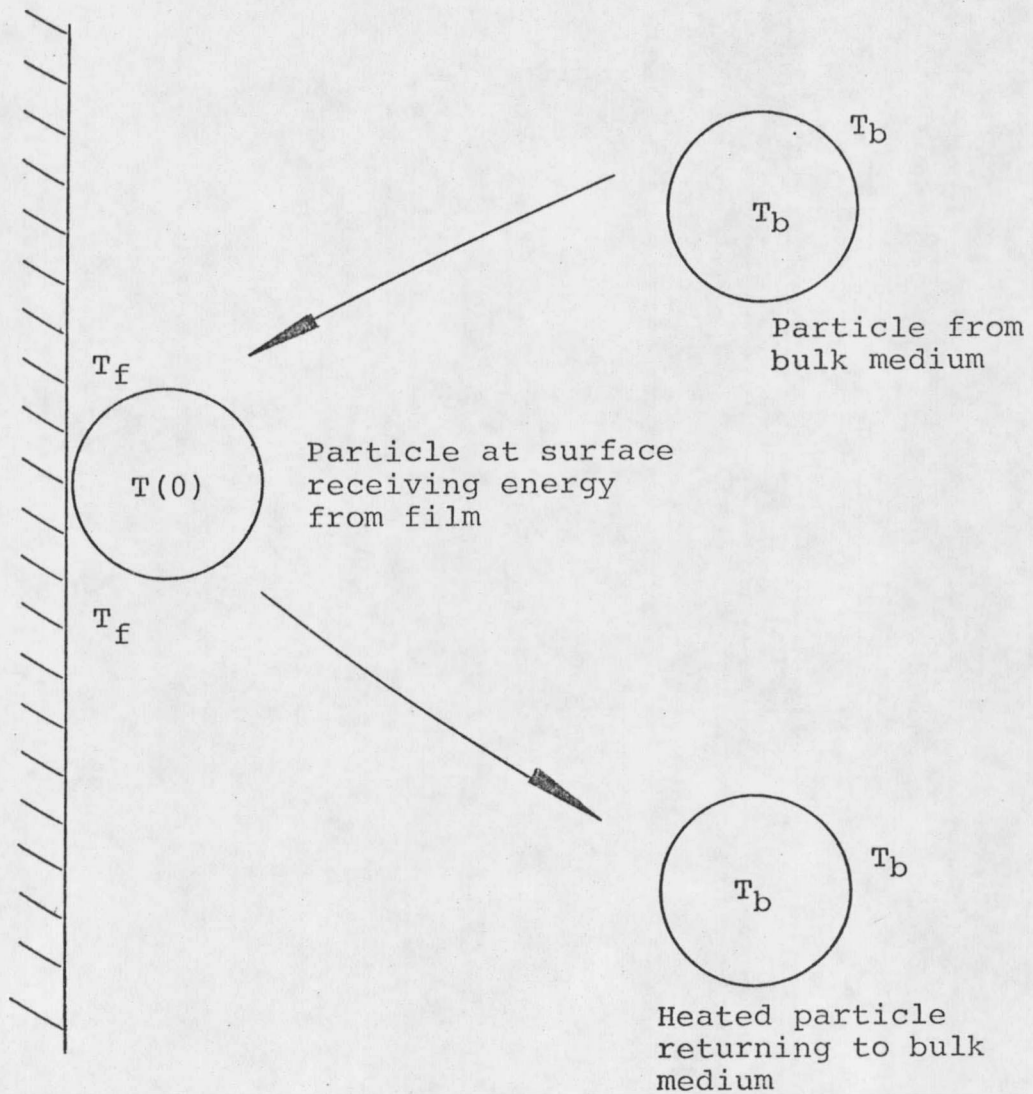


Figure 3. PARTICLE MODEL FOR HEAT TRANSFER

where,

Nu_p = particle Nusselt number, dimensionless

h_o = heat transfer coefficient, $BTU/ft^2-hr-^{\circ}F$

D_p = particle diameter, ft

k_g = thermal conductivity of fluid, $BTU-ft/hr-ft^2-^{\circ}F$

θ = average contact time, hr

ρ_s = solid particle density, lbs/ft^3

C_{ps} = heat capacity of solids, $BTU/lb-^{\circ}F$

When Genetti and Knudsen extended this model, they recommended that $10(1-\epsilon)^{0.5}$ be substituted for the 7.20 in the above equation, where $(1-\epsilon)$ is the particle volume fraction.

Kunii and Levenspiel (3), in an attempt to compare theories and develop criteria to suggest which model applies, suggested the "general" model. The following four heat transfer mechanisms may operate simultaneously in their model:

1. Heat is transferred through a thin gas film with thickness on the order of a particle diameter or less.

2. Heat transfer by conduction in the vicinity of the particle-surface contact points, with frequent replacement of particles at the surface.

3. Unsteady state absorption of heat by fresh emulsion which is swept up to and then away from surface.

4. Steady state conduction through the emulsion layer which is seldom swept away. This represents a film model for the emulsion.

The models that have been presented here are not necessarily in conflict with each other but may correspond to different ranges of operating conditions. These models are not recommended for design purposes, but do give a qualitative understanding of the processes that occur. The mathematical equations developed from these models can be very useful in correlating experimental data.

Previous Related Research

Immersing extended surfaces in a fluidized bed can greatly increase the area for heat transfer. Numerous papers have been published concerning heat transfer from immersed surfaces in fluidized beds. Studies have been done to determine the effect of fluid mass velocity, void fraction, fluid thermal conductivity, particle density, particle heat capacity, particle diameter and shape, and surface geometry.

A study by Genetti, Schmall, and Grimmett (13) showed the effect of bare and serrated fin tube orientation on the heat transfer coefficient. Variables studied included particle size, mass velocity, and orientation angle. A minimum heat transfer coefficient was observed at an orientation angle of 45° for the bare tube and 60° for the serrated fin tube.

Vreedenberg (14) investigated the heat transfer from a horizontal heating tube in a fluidized bed. He correlated the Nusselt number in terms of the Reynolds number, void fraction, and fluid and solid particles. Deviations of experimental values from his correlation were 40%. Variables in his study included bed temperature, fluid mass velocity, particle diameter and shape, and tube diameter.

Petrie, Freeby, and Buckham (15) studied a bundle of 19 horizontal aluminum bare and helical finned tubes. The tubes were heated electrically or with steam condensing on the inside. They found as did Vreedenberg that the heat transfer coefficient increases with air mass velocity, but decreases with particle diameter. They also observed a maximum in heat transfer coefficients with increasing gas mass velocity.

Chen and Withers (16) studied the heat transfer from bare and finned tubes positioned vertically in a fluidized bed. They varied fin height and fin spacing. They reported gains as large as 190% for heat transfer coefficients for helical copper fin tubes compared to plain tubes.

There have been numerous investigations into the heat transfer from tube bundles in an air fluidized bed here at Montana State University under the direction of Dr. William Genetti. Table I is a list of the principle investigators along with some of the results.

Table I. PREVIOUS RELATED RESEARCH AT MONTANA STATE UNIVERSITY

Investigator	Tube Position and Type	Parameters Varied	Agreement With Correlation Developed	Gain in Heat Transfer Compared With Bare Tube
Bartel (17) 1973	horizontal bundle of carbon steel finned and bare tubes	fin height, tube spacing, particle diameter, fluidiz- ing air vel.	$\pm 15\%$	80%
Priebe (18) 1975	horizontal bundle of carbon steel serrated finned tubes, stain- less steel and copper spined tubes	fin and spine height, spine material, spines per turn, particle diameter, fluidizing gas vel.	serrated tubes - $\pm 12.5\%$ spine tubes - $\pm 12.5\%$	spine tubes - 60%
Kratovil (19) 1976	horizontal bundle of copper helical finned tubes	fin height, fin spacing, particle diameter, fluidiz- ing gas vel.	$\pm 20\%$	190%
Vanderhoff (20) 1978	vertical bundle of carbon steel serrated finned tubes	fin height, fin width, fin spacing, particle diameter, fluidizing gas vel.	no correlation	74%
Everly (21) 1978	copper coiled spiral tubes	groove depth, number flutes, flute pitch, particle diameter, fluidizing gas vel.	$\pm 20\%$	40%

EXPERIMENTAL APPARATUS

The discussion of the equipment will be divided into two main sections: the helical finned tube system and the coiled spiral tube system.

Helical Finned Tube System

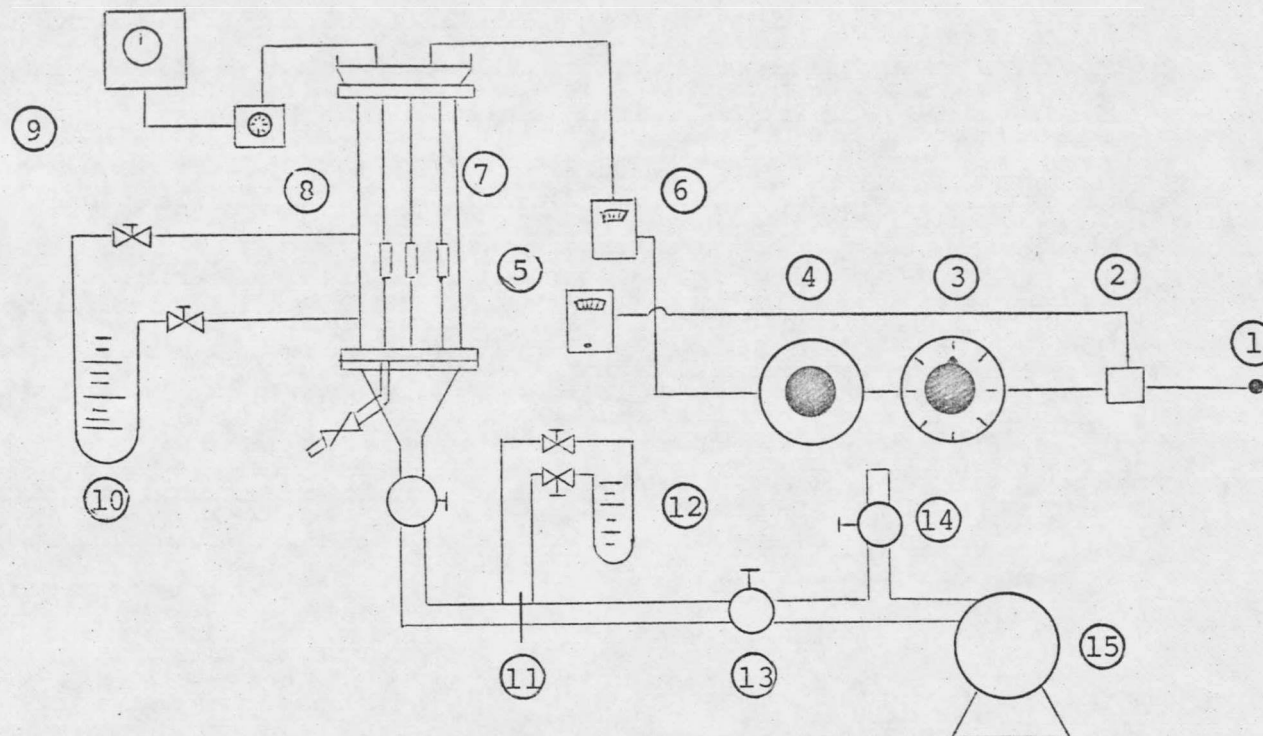
The equipment used in this system was already available, having been once previously used for similar heat transfer investigations of serrated finned tubes by Dan Vanderhoff. Minor modifications were made to improve the efficiency and ease of operation.

The main parts of the helical finned tube system are the fluidizing column, the fluidizing system, and the electrical system.

A schematic drawing of the overall experimental system is shown on Figure 4. It shows the relationship between various parts. A photograph of the column and the surrounding equipment is shown on Figure 5.

Fluidizing Column

Figure 6 shows a detailed view of the column. The column was cylindrical in shape, 59 inches high and $13\frac{1}{2}$ inches inside diameter. It was fabricated of clear, $\frac{1}{4}$ -inch plexiglas. The column was clear to allow visual



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

Figure 4. SCHEMATIC VIEW OF HELICAL TUBE SYSTEM

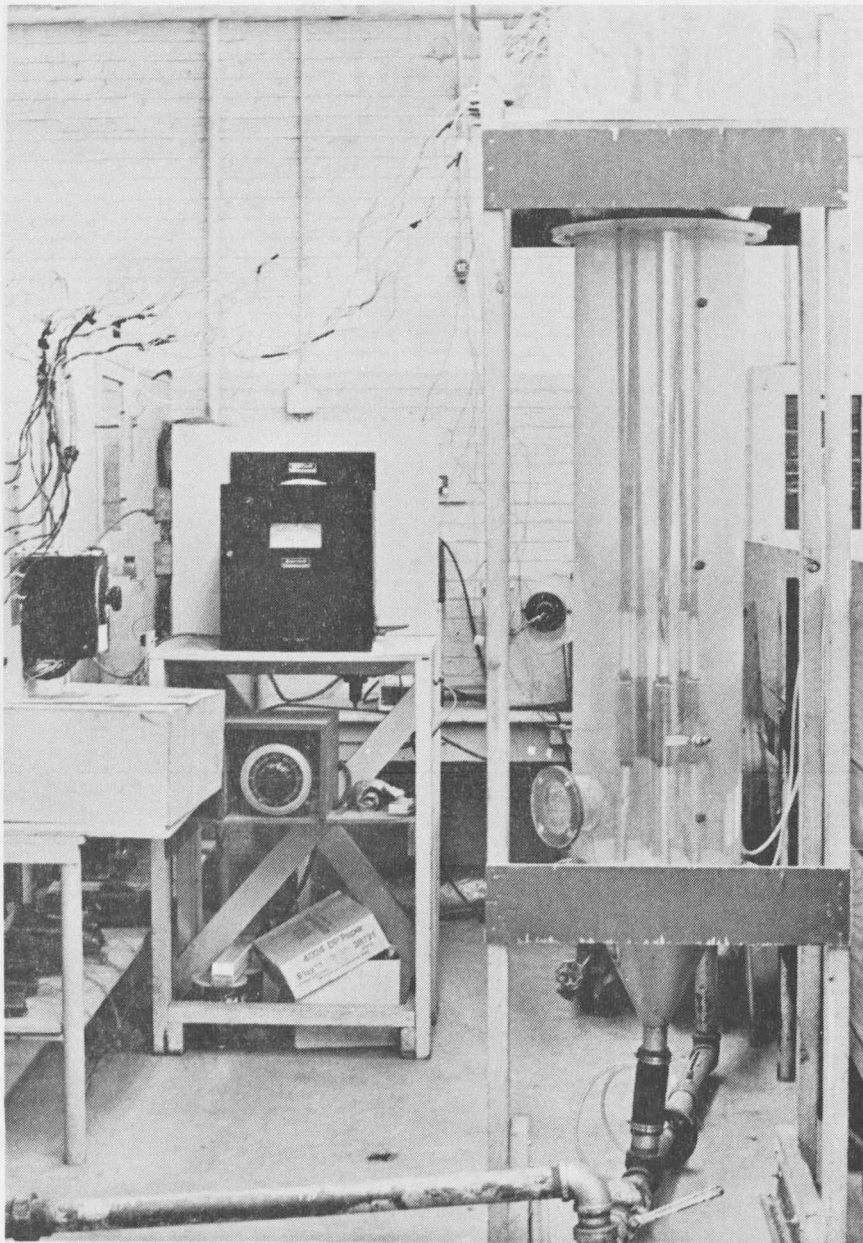


Figure 5. COLUMN OF HELICAL FINNED TUBE SYSTEM

-22-

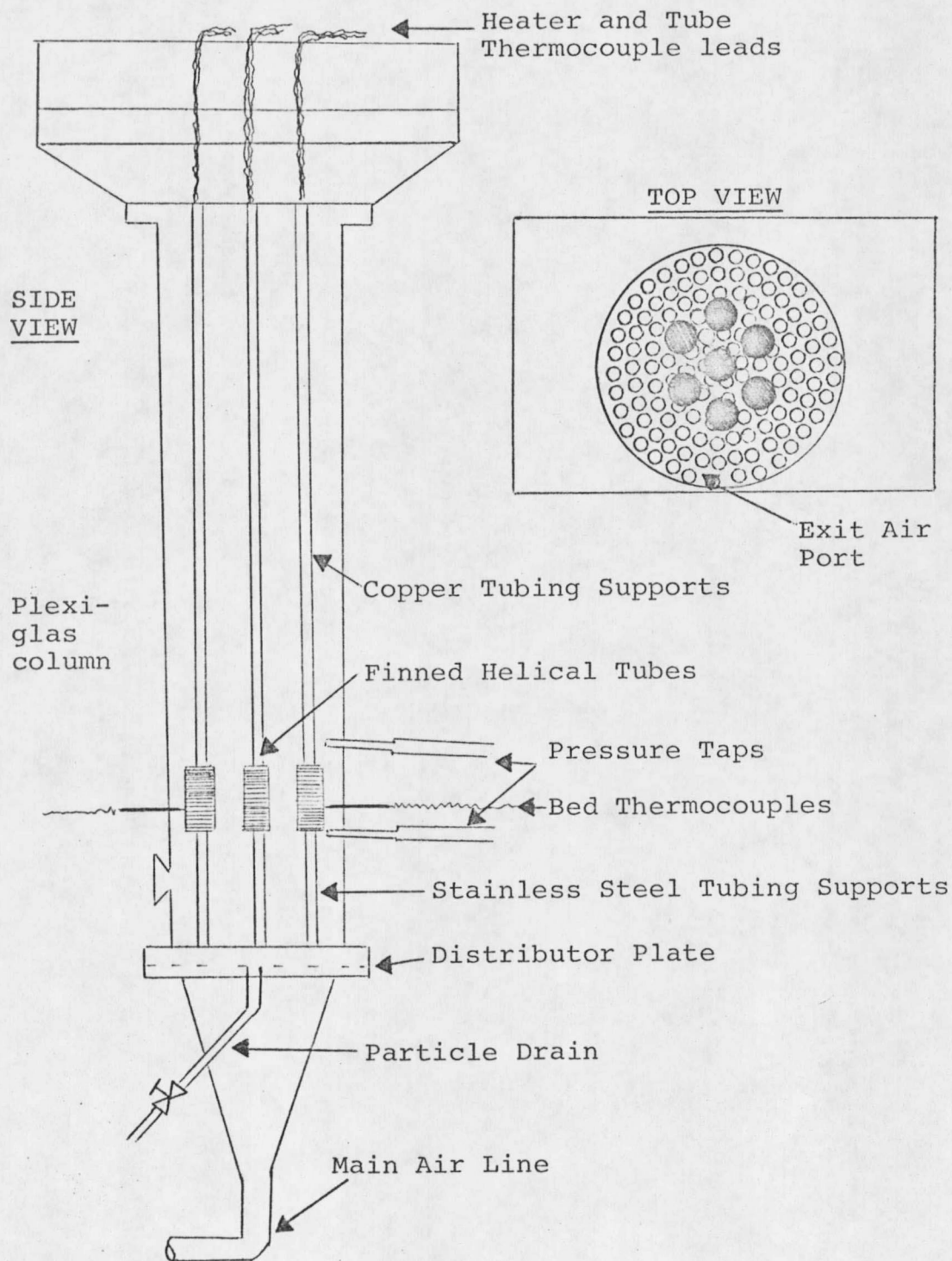


Figure 6. DETAILED VIEW OF HELICAL FINNED TUBE COLUMN

observation of the bed when fluidized, and it was easier to see that the heaters and tubes remained in position. One access port, 4 inches in diameter and located 6 inches from the bottom of the column, was used to clean the bed when changing particle sizes. Flanges, 3/4-inch thick, were placed to the top and bottom of the column.

A small, 6 inches high, galvanized steel funnel, 13½ inches bottom diameter and 19 inches top diameter, fitted with a rubber gasket, was bolted to the top flange of the column. Attached to the top of the funnel was a steel perforated plate sandwiched between two 24 inches by 24 inches plexiglas plates. The plexiglas plates were 3/4-inch thick and had a 19-inch diameter hole cut in the center of them. This steel perforated plate sandwich was the air exit port and also allowed filling of the column. Attached to the top of the steel perforated plate sandwich was a clear plexiglas cylinder 19 inches in diameter and 20 inches high.

A wooden ring, along with a funnel fitted with rubber gaskets, was bolted to the bottom flange of the column. This funnel supported the distributor plate. The funnel was constructed from 16 gauge galvanized steel. The funnel was 12 inches in height, 13½ inches

in diameter at the top, and 2 inches in diameter at the bottom. The spout of the funnel was 2 inches in diameter and 4 inches long. A 1-inch diameter particle drain was fastened to the distributor plate and extended through the side of the funnel. The drain pipe was fitted with a gate valve. 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 $\frac{1}{4}$ inch in diameter with $\frac{1}{2}$ -inch center-to-center distance. This distributor plate proved to provide sufficient pressure drop for uniform fluidization.

The column was supported on a wooden frame anchored to the floor.

Fluidizing System

Air was used as the fluidizing medium. The air was supplied to the column by a Sutorbilt air blower driven by a $7\frac{1}{2}$ hp motor. A $2\frac{1}{2}$ -inch schedule 40 pipe was used both for a main air supply line to the column and for a column by-pass line. The main air supply line was connected to the steel funnel, attached to the bottom of the column, with a flexible rubber hose. A gate valve

was located in the main supply line and the by-pass line. The gate valve in the supply line was left open for all the runs, while the air flow rate to the column was regulated by adjusting the gate valve in the by-pass line.

An orifice in the main air supply line was used to measure the air flow rates to the column. The pressure drop across the orifice was measured with a water manometer. The minimum fluidization flow rates were measured using a micromanometer. The orifice had a $1\frac{1}{2}$ -inch diameter opening, and vena contracta taps.

Back pressure from the column was measured with a Duragauge pressure gauge located downstream from the orifice.

Three sizes of Blast-O-Lite glass beads were used as the bed material. Their sizes and distributions were determined by Everly (21), using a camera-mounted microscope. A stagnant bed height of 18 inches was used in each run. The particle density was $155 \text{ lb}_m/\text{ft}^3$. The glass bead characteristics and distributions are listed in Table II.

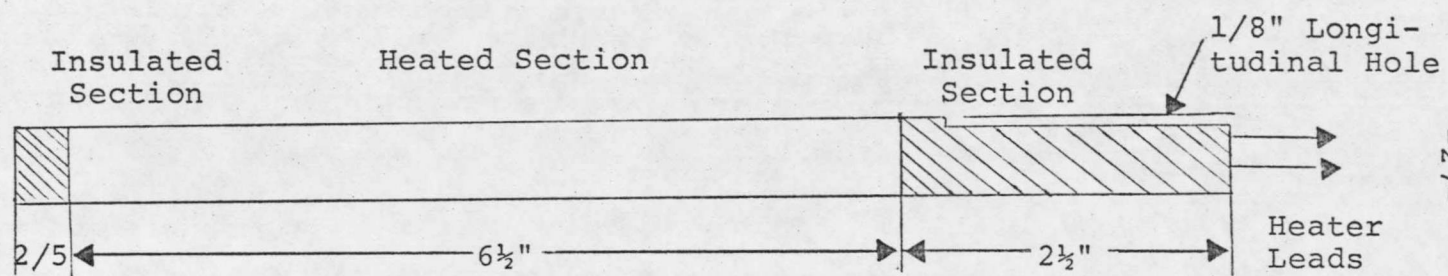
Table II. BLAST-O-LITE BEAD SIZE ANALYSIS

Average Diameter (in)	Distribution (in)	Nominal Name
0.0076	0.005-0.0098	Small
0.0109	0.0098-0.0164	Medium
0.0164	> 0.0164	Large

Electrical System

The electrical system consisted of the heater and tube assemblies, the thermocouple system and the power supply.

Watlow firerod cartridge heaters of appropriate diameters were used as the heat source for the finned tubes. As shown in Figure 7, each cartridge was 10 inches long, comprised of a 6½-inch heated section and two insulated ends. A 1/8-inch diameter longitudinal hole was drilled through the lead end of the heater. A single thermocouple wire was soldered to the surface, at the base of a fin, midway along the length of each tube. The thermocouple wire was passed through the 1/8-inch diameter hole in the heater. The heated section of the cartridge heater was inserted into the finned tube with



-27-

Figure 7. DETAILS OF A CARTRIDGE HEATER

the two insulated ends protruding. A set screw on the finned tube was tightened to secure the tube in place on the heater. Figure 8 is a photograph of the cartridge heater and the finned tube. The smaller insulated end (.4 inch) was inserted into a stainless steel tube 9 inches long and $\frac{1}{2}$ inch inside diameter. The other end of this stainless steel tube was fastened to the distributor plate. The larger insulated end (3 inches) was inserted into a copper tube 52 inches long and $\frac{1}{2}$ inch inside diameter. The other end of this copper tube was inserted through the steel perforated plate on the top of the column.

The leads from the heaters along with the tube thermocouple wire were threaded through the inside of the 52-inch long copper tube and out the top of the column. The leads from the heaters were connected through fuses to seven parallel toggle switches. From these switches the line ran to a rheostat and then through a powerstat where the power could be varied.

A switch box was connected to the powerstat. Also connected to the switch box was a Fenwall model 524 high temperature limit controller. This instrument was a cut-off relay to avoid problems of overheating the bed. A

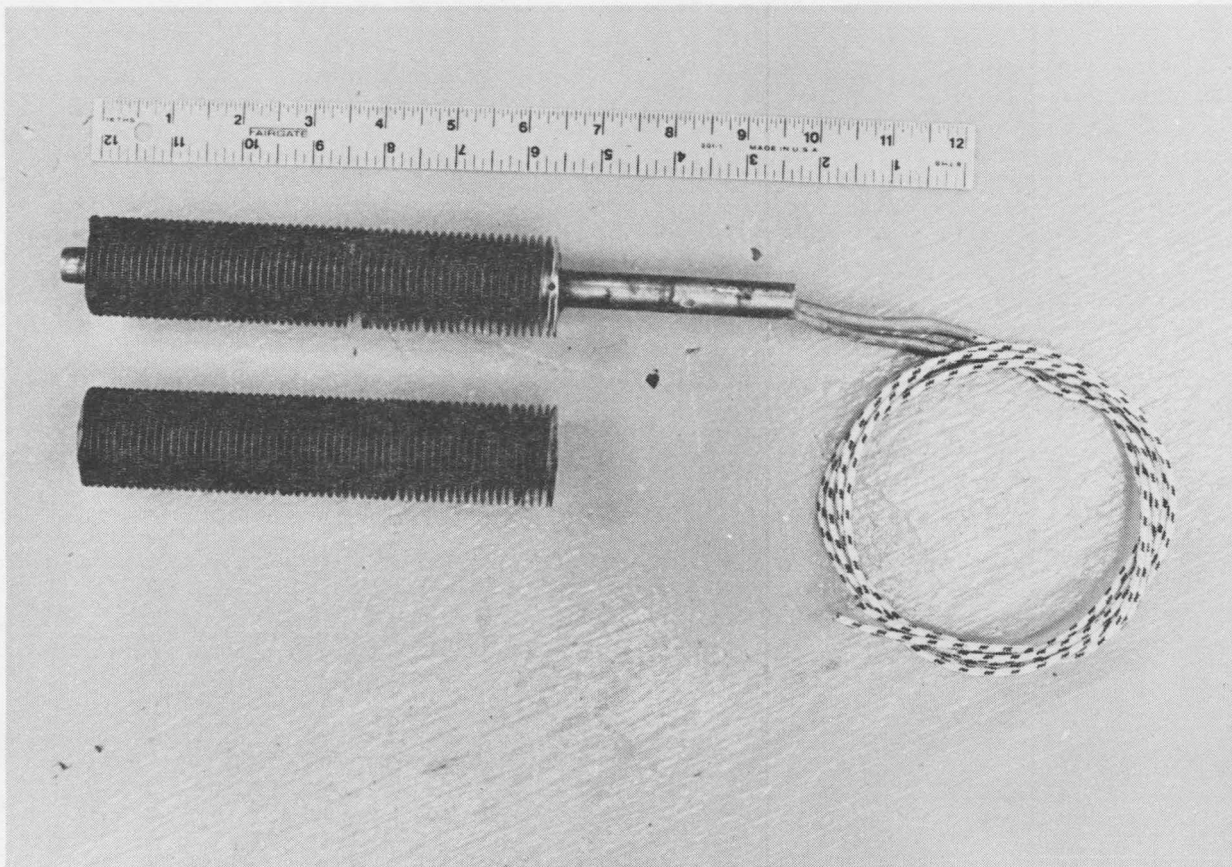


Figure 8. CARTRIDGE HEATER AND FINNED TUBE

thermocouple in the bed acted as the sensing device for the instrument. A 240 volt line source was fed into the switch box.

Along with the single thermocouple attached to each tube, two more thermocouples were used to measure the bed temperature in three different locations. The three thermowells were located $11\frac{1}{2}$ inches above the distributor plate and were equally spaced around the circumference of the bed. The thermowells projected 3 inches into the bed. Another thermocouple was used to measure the incoming air temperature. Its thermowell was located downstream from the orifice in the main air supply line. Leads from the tube thermocouples, the bed thermocouples, and the main air line thermocouple were plugged into a panel board which was wired to a switch box. A model 156x15-P Brown Potentiometer was connected to the switch box. The temperature was measured directly from the potentiometer.

A total of eight different sizes of finned tubes were investigated. A bundle of seven tubes of the same size was used in the column. The tubes were spaced 2 inches center-to-center. After the last tube set was

investigated, the center-to-center spacing was increased to 3 inches and then to 5 inches. The surface areas and various dimensions are given in Table III. An end view and side view of the finned tubes is shown on Figure 9.

Coiled Spiral Tube System

The equipment used in this system was also available, having been once previously used for a study of coiled spiral tubes of a different geometry. Only a brief description of the equipment in this system will be given here; the reader is referred to Everly (21) for a more detailed description.

The main parts of the equipment used in this investigation are the fluidizing column, the fluidizing system, the thermocouple system, and the tube and water heating system.

The fluidizing system was the same as that described for the helical tube system.

A schematic drawing of the overall experimental system is shown on Figure 10. A photograph of the column and the surrounding equipment is shown on Figure 11.

Table III. HELICAL FINNED TUBE DIMENSIONS

Tube No.	Fin O.D. (in)	Tube O.D. (in)	Fin Height (in)	Fin Tip Width (in)	Fin Base Width (in)	Fins Per Inch	Fin Spacing (in)	Avg.* Total Tube Area (ft ²)
1	1.453	.625	.414	.0110	.0164	9	.0951	1.1995
2	1.375	.625	.375	.0134	.0242	9	.0871	1.0604
3	1.328	.625	.352	.0110	.0170	9	.0951	0.97509
4	1.094	.625	.234	.0110	.0120	5	.1840	0.37809
5	1.094	.625	.234	.0090	.0110	14	.0554	0.89712
6	1.094	.625	.234	.0094	.0130	18	.0330	1.0658
7	1.000	.453	.274	.0157	.0330	5	.1750	0.35264
8	1.500	.750	.375	.0140	.0330	7	.1229	0.95433

*The average total tube area was calculated using the average width between the fin tip and the fin base.

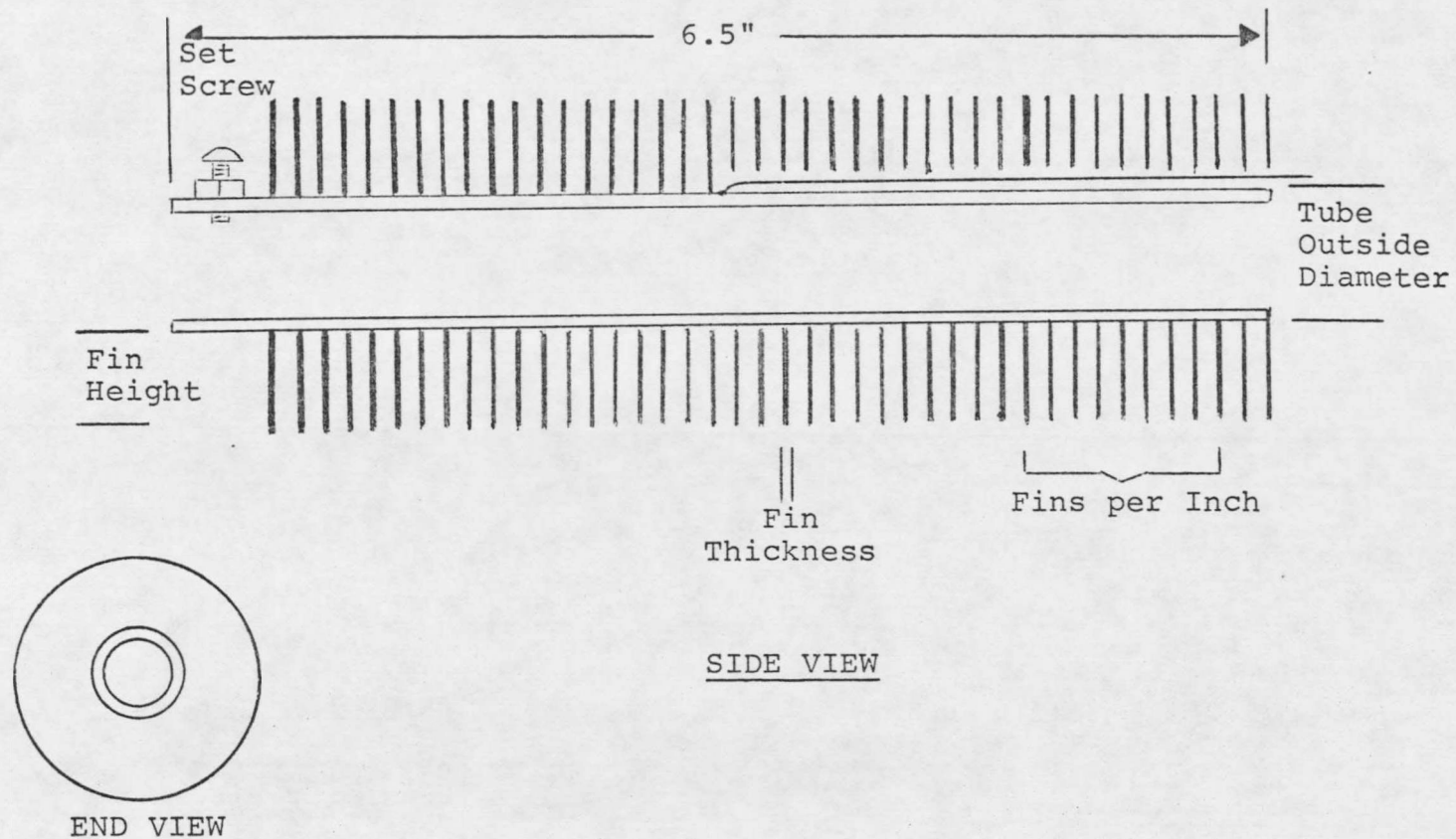
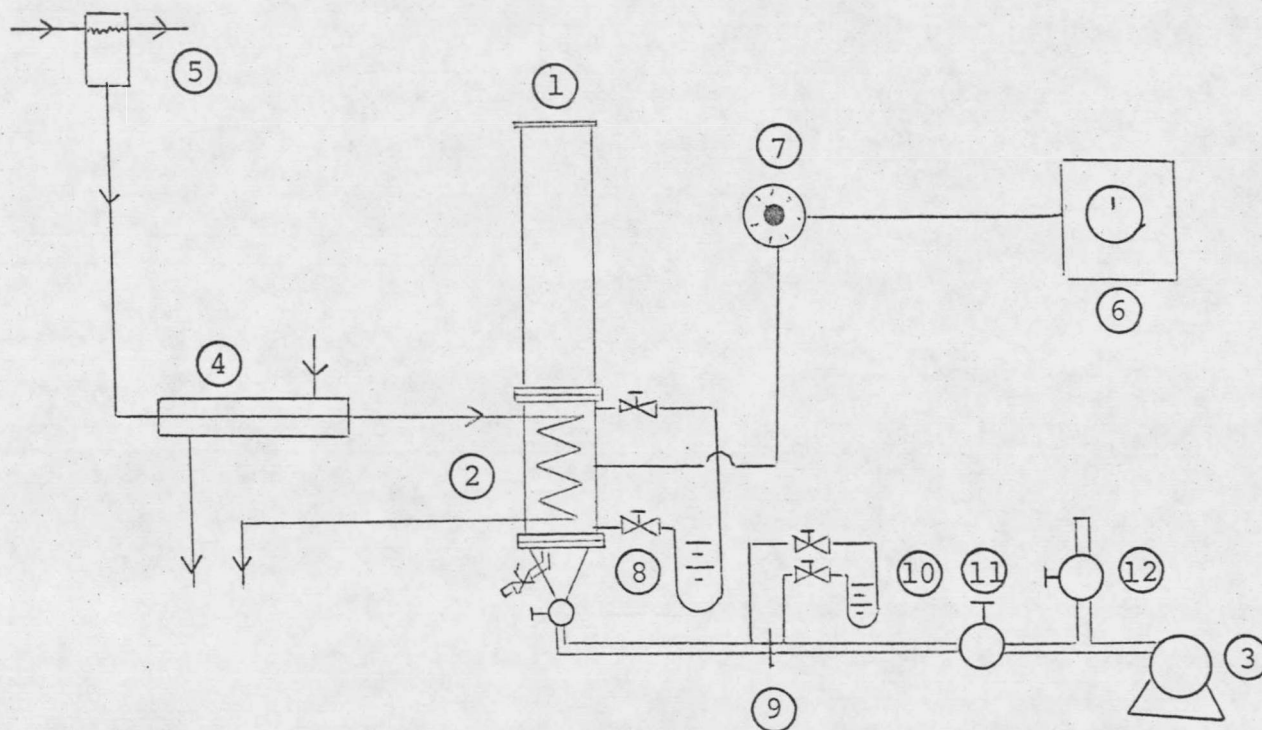


Figure 9. HELICAL FINNED TUBE DETAILS AND NOMENCLATURE



(1) Plexiglas Column; (2) Spiral Tube; (3) Air Blower; (4) Heat Exchanger;
 (5) Overhead Water Tank; (6) Potentiometer; (7) Switch Box; (8) Bed Mano-
 meter; (9) Orifice; (10) Orifice Manometer; (11) Main Air Falve; (12) Bypass
 Valve.

Figure 10. SCHEMATIC VIEW OF COILED SPIRAL TUBE SYSTEM



Figure 11. COLUMN OF COILED SPIRAL TUBE SYSTEM

Fluidizing Column

Figure 12 is a detailed view of the column. The column was constructed from plexiglas $3/8$ inch thick, 9 feet high, and 14 inches in diameter. The column was divided into two sections that could be separated to change the tubes. The lower section was $25-3/4$ inches and the upper section was $90-1/4$ inches. Wire screen was wrapped around the lower section and a third of the upper section. This wire helped dissipate the electrical charge that built up on the outside of the column during fluidization. The top of the column was a removable $1-3/4$ -inch wooden ring covered with a stainless steel screen.

A funnel made from $1/32$ -inch galvanized steel was fastened to the bottom of the bed and supported the distributor plate.

The distributor plate consisted of a piece of 100 mesh stainless steel wire cloth sandwiched between two $1/16$ -inch stainless steel perforated plates. A particle drain pipe was connected to the distributor plate and extended through the side of the funnel section. A quick opening valve was fitted on the end of the particle

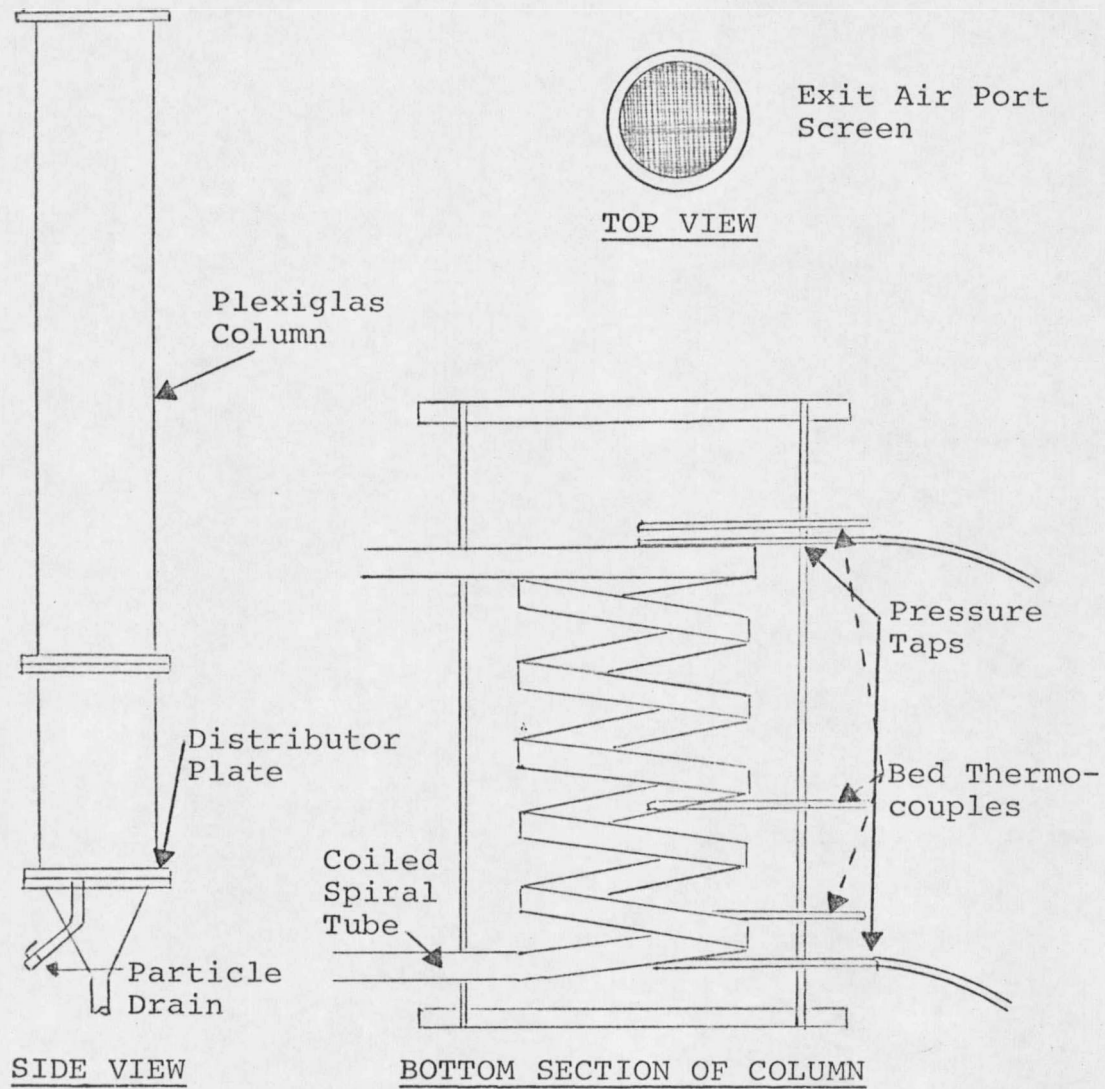


Figure 12. DETAILED VIEW OF COILED SPIRAL TUBE COLUMN

drain pipe. This was used to empty the column of glass beads.

The column was supported by a wooden frame that was anchored to the floor.

Thermocouple System

Thermocouples were used to measure inlet and outlet water temperatures, inlet air temperatures, outside column temperature, and three bed temperatures. The location of the bed thermocouples is shown on Figure 12. The leads of the thermocouples were connected to a switch box which was connected to a 156x15-P Brown Potentiometer.

Tube and Water Heating System

One six-foot length of spiral copper tube was bent into a 7-inch inside diameter coil. A photograph of the coil is shown in Figure 13. Connected to the bottom end of the tube was a one-inch pipe that protruded through the column wall. This one-inch pipe was connected to more piping that lead to a water sink. The top end of the tube was fitted with a one-inch pipe that also protruded through the column wall. Connected to this pipe

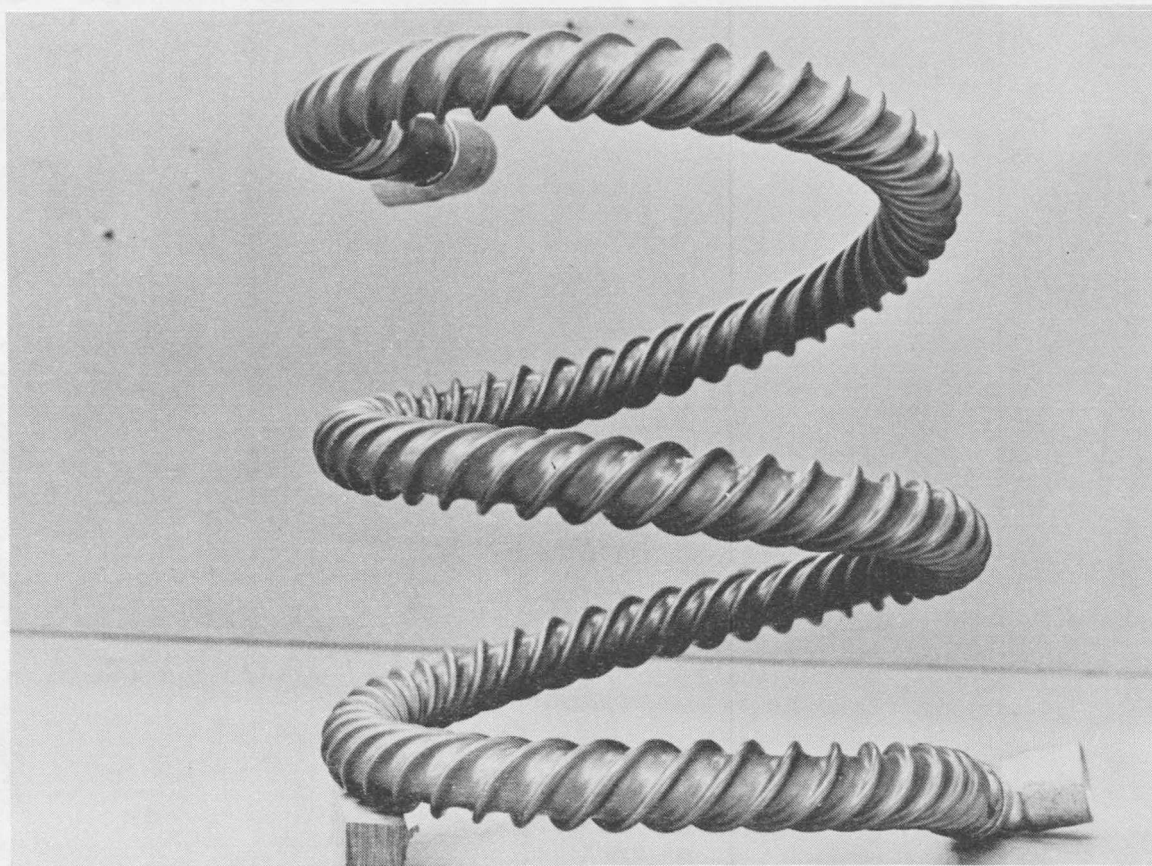


Figure 13. COILED SPIRAL TUBE

was a steam heated countercurrent heat exchanger.

Water was supplied to the heat exchanger and tube by an overhead tank.

Table IV is a list of the physical characteristics of the spiral tubes used in this investigation and previous studies. Everly (21) and Genetti and Everly (22) investigated tubes 1A, 2A, 3A, and 4A. I have investigated tubes 5A and 6A for this investigation.

Table IV. COILED SPIRAL TUBE DIMENSIONS

Tube No.	Wall Thickness	Length (ft.)	Pitch, P (in.)	Groove Depth (in.)	Outside Diameter (in.)	Surface* Area (ft ²)
1A	20 gage	12.0	2.00	0.150	1-1/8	2.57
2A	20 gage	12.0	2.80	0.210	1-1/8	2.02
3A	20 gage	6.0	2.23	0.184	1-1/8	2.51
4A	20 gage	10.4	plain	plain	1-1/8	1.76
5A	20 gage	6.0	2.60	0.166	1-1/8	2.25
6A	20 gage	6.0	3.10	0.101	1-1/8	1.96

* The surface area was based on 6.0 ft. length of tube.

EXPERIMENTAL PROCEDURE

Minimum Fluidization Velocities

Minimum fluidization velocities of the three particle sizes were determined for each of the eight tube bundles in the helical finned tube system. Once the tube bundle was assembled in the column, the column was filled to static height of 18 inches from the distributor plate. The air was turned on and regulated so the bed was bubbling freely. The heaters were then turned on to allow the column to heat up to normal operating temperature and also drive off any moisture in the particles. The column took 3-4 hours to reach steady state.

The minimum fluidization velocity was determined by regulating the air flow rate until initial fluidization or defluidization of the bed was visually verified. A water-filled micromanometer was used to measure the pressure drop across the orifice. This process was repeated several times for each particle size. An average of all the readings was used as the minimum fluidization value. The average minimum fluidization values for the three bead sizes are shown on Figure 14.

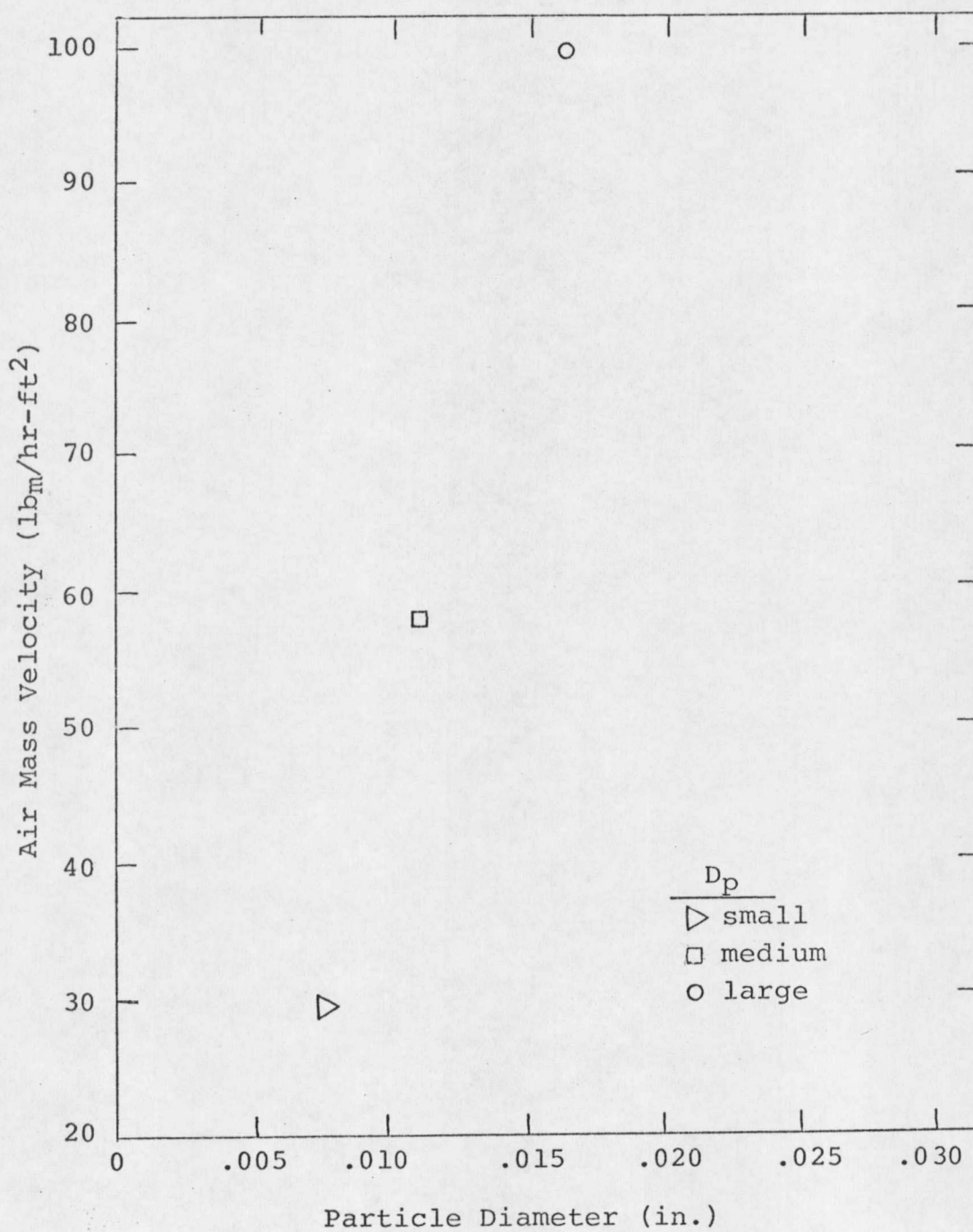


Figure 14. PARTICLE MINIMUM FLUIDICATION VELOCITIES

Typical Run Procedure for the Helical Finned Tube System

The same procedure was used for all runs. A set of finned tubes was selected and the thermocouples were installed. The heaters were layered with aluminum tape and copper anti-sieze compound, to help promote contacting between the heaters and the tubes. The heater was then inserted into the finned tubes. The two protruding insulated ends of the heater were fitted into the stainless steel tubing and the copper tubing supports. The tube thermocouple leads and the heater leads were then threaded through the copper tubing. This assembly of heater, finned tube, and support tubes was positioned and secured into place in the column. The leads from the heaters and thermocouples were connected to their respective panels. Particles of the proper size were poured in the top of the column to a static bed height of 18 inches above the distributor plate.

The power to the heaters was turned on and adjusted to 200 watts per heater. The blower was turned on and the air flow rate adjusted. The column was allowed to reach steady state. Steady state was reached in 4 hours for the first flow rate and 2 hours for each successive flow rate. A reading consisted of recording the seven

temperatures of the finned tubes in the bundle, the seven heater wattages, the three temperatures of the bed, the temperature of the incoming air, the pressure drop across the orifice and the bed, and the ambient conditions. The air flow rate was then increased after the second reading. This procedure was repeated until all of the desired flow rates had been investigated.

All three particle sizes were investigated for each tube. To change particle size, the particles were drained out through the particle drain pipe. The remaining particles were removed by opening the access port and using a vacuum cleaner.

Typical Run Procedures for the Coiled Spiral Tube System

The tubes were properly installed in the column. The column was filled to a static bed height of 20 inches with the desired particle size. The air blowers were turned on and adjusted to the desired flow rate. The water was turned on and adjusted to the desired flow rate. The water flow rate remained constant throughout the run. The inlet water temperature to the tube was adjusted by varying the amount of steam to the exchanger. The inlet water temperature was also maintained constant throughout the run. The column was allowed to reach

steady state, 4 hours for the first air flow rate and 1½ hours for each successive air flow rate. Three readings 15 minutes apart were taken for each air flow rate. A reading consisted of recording the inlet and outlet water temperature, the three bed temperatures, the water flow rate, the incoming air temperature, the pressure drop across the orifice and the bed, and the ambient conditions. The air flow rate was then increased after the third reading. This procedure was repeated until all of the desired flow rates had been investigated.

All three particle sizes were investigated for each tube. The particle sizes were changed in the same procedure as the helical finned tube system.

A run had to be made for each tube using one of the three particle sizes. This run was made using the same procedure described above except the air flow rate was held constant and the water flow rate varied. This run was made to obtain data for the Wilson plot analysis.

DEVELOPMENT OF THEORETICAL MODEL FOR HELICAL FINNED TUBES

As was mentioned previously, a single thermocouple was attached to the surface of each tube in the bundle. This thermocouple measured the temperature of the base of the fin. When using this temperature to calculate the heat transfer coefficient, it would assume that there is a uniform temperature distribution across the tube and fin. Since this is not true, the temperature distribution in the fin must be accounted for.

The helical fins used in this study were slightly tapered. A model was developed that would account for the taper. This model worked well for low air flow rates, but the equation became unstable at high flow rates. A second model was then developed that didn't take into account the taper in the fin. The second model worked well for both low and high air flow rates. The heat transfer coefficients obtained from both models were compared for the low flow rates. There was only a 0.154-2.5% difference between the two models. Kratovil (19) also reported that the taper was so slight as to not significantly affect the final values for the heat transfer coefficients. Based on these findings, the taper was not taken into account in determining the

temperature distribution in the fin. An average fin thickness between the base of the fin and the tip of the fin was used.

To obtain the temperature distribution in the fin, a steady state energy balance was taken around a differential fin element. Assuming angular symmetry, the problem became 1-dimensional in the r-direction, as shown on Figure 15.

Heat flows by conduction into the left face of the element, while heat flows out of the element by conduction through the right face and by convection from the surface. Under steady state conditions, the accumulation of energy within the element is zero.

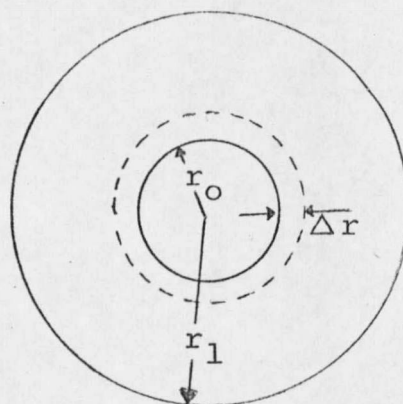
$$-k2\pi rw \left. \frac{dT}{dr} \right|_r + k2\pi rw \left. \frac{dT}{dr} \right|_{r+\Delta r} - 2h2\pi r \Delta r (T - T_b) = 0$$

Rate of heat flow by con- duction into element at r	-	rate of heat flow out of element at (r+Δr)	-	rate of heat flow by con- vection from surfaces be- tween r and (r+Δr)	= 0
--	---	---	---	---	-----

where,

k = thermal conductivity of the fin, BTU-ft/hr-ft²-°F
h = average heat transfer coefficient, BTU/hr-ft²-°F
w = average fin thickness, ft

END VIEW



EDGE VIEW

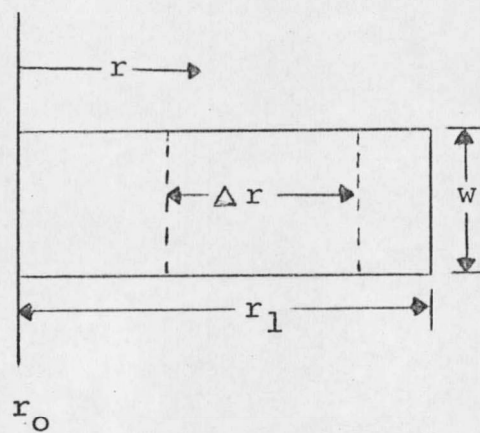


Figure 15. PROPOSED MODEL OF HELICAL FIN

Dividing through by Δr and taking the limit as Δr goes to zero and simplifying, the following differential equation for the temperature distribution in the fin was obtained,

$$\frac{d}{dr} \left[r \frac{d(T-T_b)}{dr} \right] - \frac{2hr}{wk} (T-T_b) = 0$$

Defining $\beta^2 = 2h/wk$, and expanding out the derivative term in the preceding equation gave,

$$\frac{d(T-T_b)}{dr} + r \frac{d^2(T-T_b)}{dr^2} - \beta^2 r (T-T_b) = 0$$

Multiplying through by r and by β^2/β^2 to get the derivative in the form of βr instead of r gave,

$$\beta^2 r^2 \frac{d^2(T-T_b)}{d(\beta^2 r^2)} + \beta r \frac{d(T-T_b)}{d(\beta r)} - \beta^2 r^2 (T-T_b) = 0 \quad (1)$$

Equation (1) is now in the form of the modified (or hyperbolic) Bessel's differential equation,

$$x^2 \frac{dy}{dx^2} + x \frac{dy}{dx} - (x^2 - n^2)y = 0 \quad (2)$$

The complete solution to equation (2) is of the form,

$$y = C_1 I_n(x) + C_2 K_n(x)$$

Therefore, the solution to equation (1), for $n=0$, was of the form,

$$T - T_b = C_1 I_0(\beta r) + C_2 K_0(\beta r) \quad (3)$$

The two boundary conditions to this problem were:

$$a) \quad r=r_0, \quad T=T_w$$

$$b) \quad -k \left. \frac{dT}{dr} \right|_{r=r_1} = h(T \Big|_{r=r_1} - T_b)$$

Applying the two boundary conditions to equation (3) and solving the two equations simultaneously yielded equations for the constants C_1 and C_2 .

$$C_1 = \frac{T_w - T_b (h K_0(\beta r_1) - k \beta K_1(\beta r_1))}{[h [I_0(\beta r_0) K_0(\beta r_1) - I_0(\beta r_1) K_0(\beta r_0)] - k \beta [I_1(\beta r_1) K_0(\beta r_0) + I_0(\beta r_0) K_1(\beta r_1)]}$$

$$C_2 = \frac{T_w - T_b - C_1 I_0(\beta r_0)}{K_0(\beta r_0)}$$

Substituting the equations for C_1 and C_2 back into equation (3) yielded an equation for the temperature distribution in the fin.

The heat transferred to each fin could then be found by applying Fourier's law,

$$q_{fin} = -k2\pi r_o w \left. \frac{dT}{dr} \right|_{r=r_o} \quad (4)$$

Taking the derivative of equation (3) with respect to r and substituting it into equation (4) gave,

$$q_{fin} = -k2\pi r_o w B [C_2 K_1(Br_o) - C_1 I_1(Br_o)]$$

To obtain a relation between the experimental values of $Q/\Delta T$, and the average heat transfer coefficient, h , the above expression was divided by the experimentally measured temperature difference and multiplied by the total number of fins, N_t . A certain portion of the tube was bare, so this was added on to the expression.

The final relation, then, is:

$$\frac{Q}{\Delta T} = \frac{N_t k 2\pi r_o w B}{\Delta T} [C_2 K_1(Br_o) - C_1 I_1(Br_o)] + h A_b$$

There is one major assumption in this model, and that was that the heat transfer coefficient was constant over the entire surface of the tube.

The heat transfer coefficient in the above expression is now based on the model and accounts for the temperature distribution in the fin. A computer program, employing function generator subroutines for the Bessel functions, was used to solve the above equation for $Q/\Delta T$ given a predetermined sequence of values for the heat transfer coefficient. Graphs of the heat transfer coefficient versus the calculated $Q/\Delta T$ were constructed for each of the eight tubes investigated. These graphs are shown on Figures 16, 17, and 18. Using these figures, the heat transfer coefficient accounting for the temperature distribution in the fin could be found for each of the experimental $Q/\Delta T$ values.

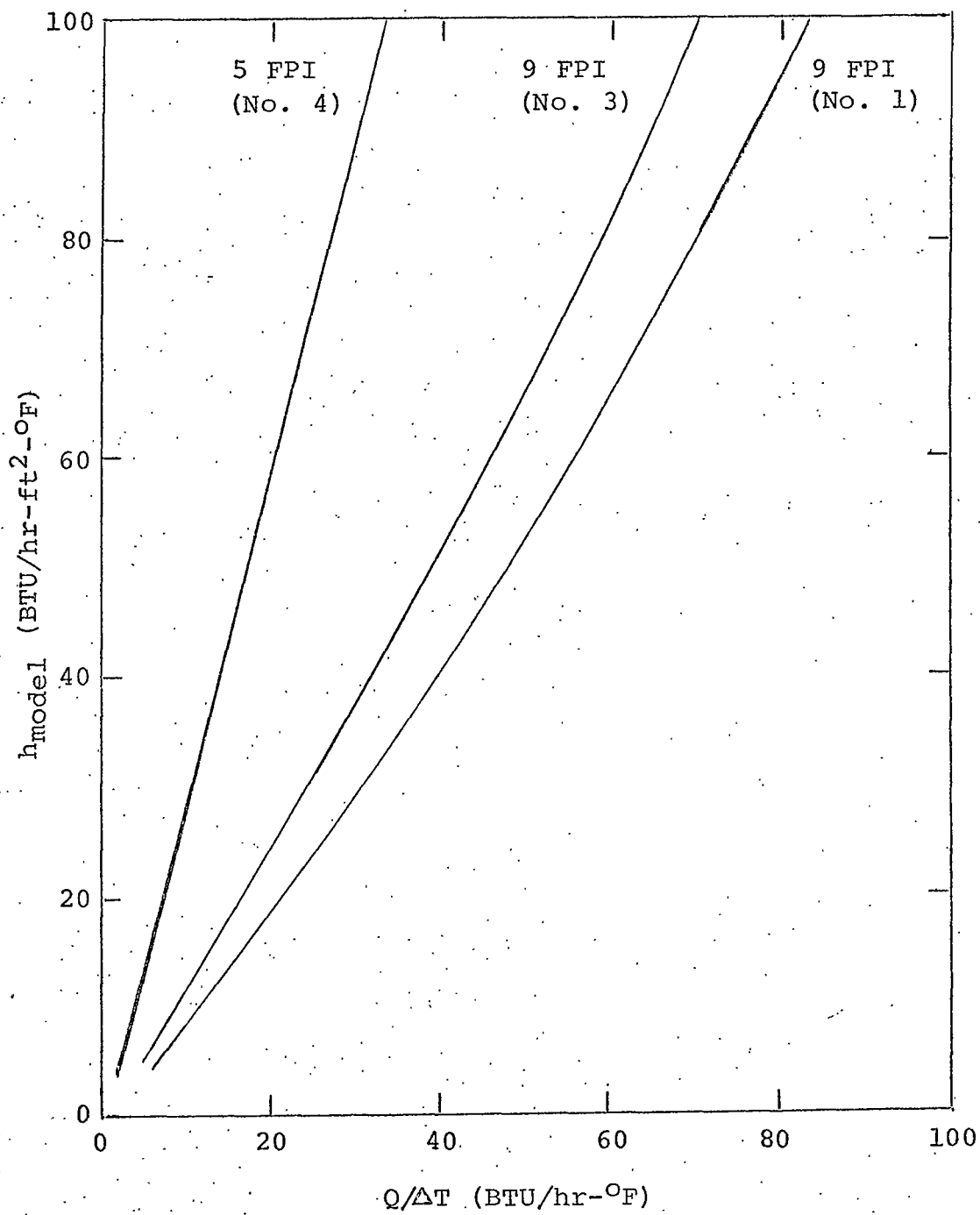


Figure 16. ANALYTICAL SOLUTION FOR $Q/\Delta T$,
No. 1, No. 3, No. 4 TUBES

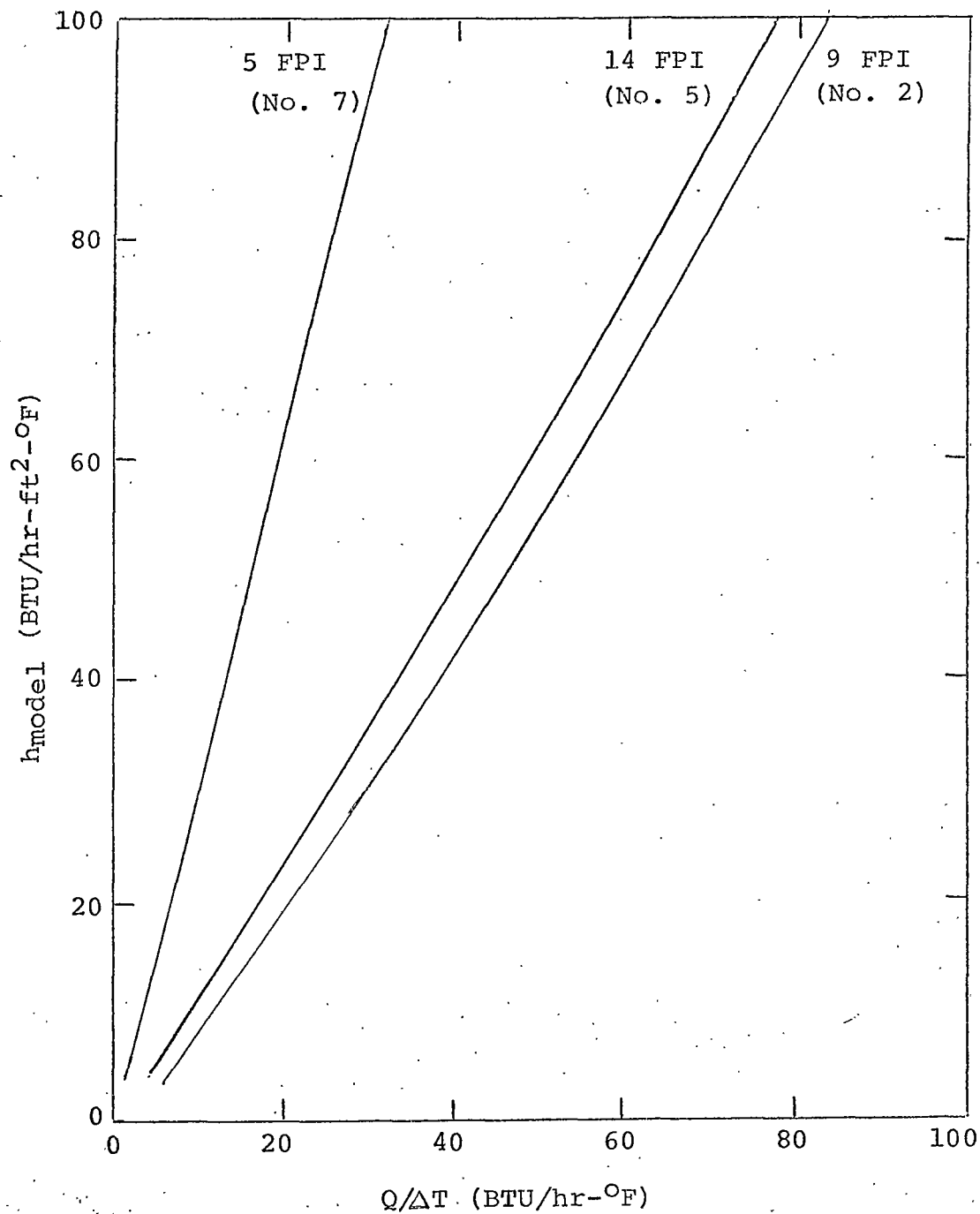


Figure 17. ANALYTICAL SOLUTION FOR $Q/\Delta T$,
No. 2, No. 5, No. 7 TUBES

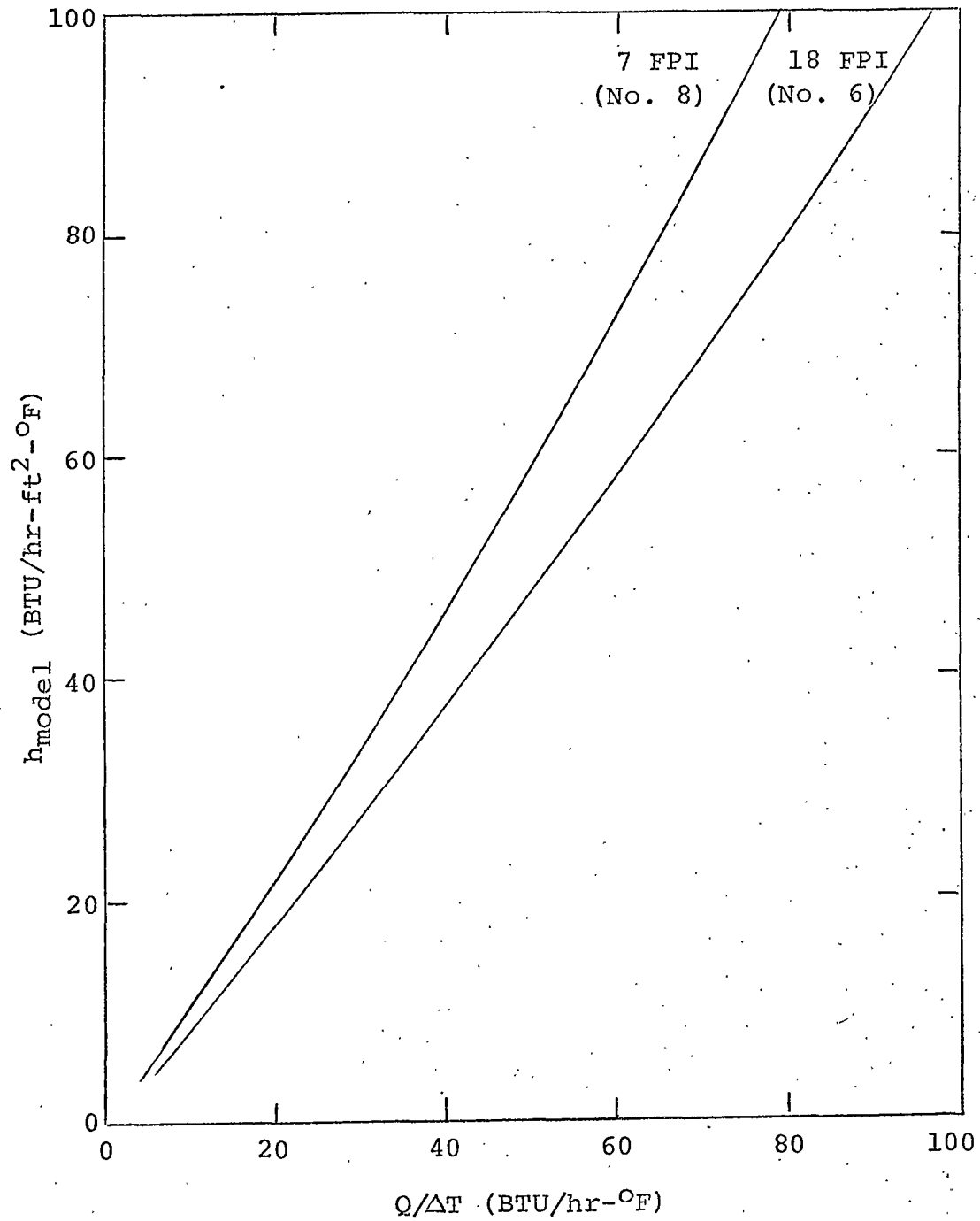


Figure 18. ANALYTICAL SOLUTION FOR $Q/\Delta T$,
No. 6, No. 8 TUBES

RESULTS AND DISCUSSION

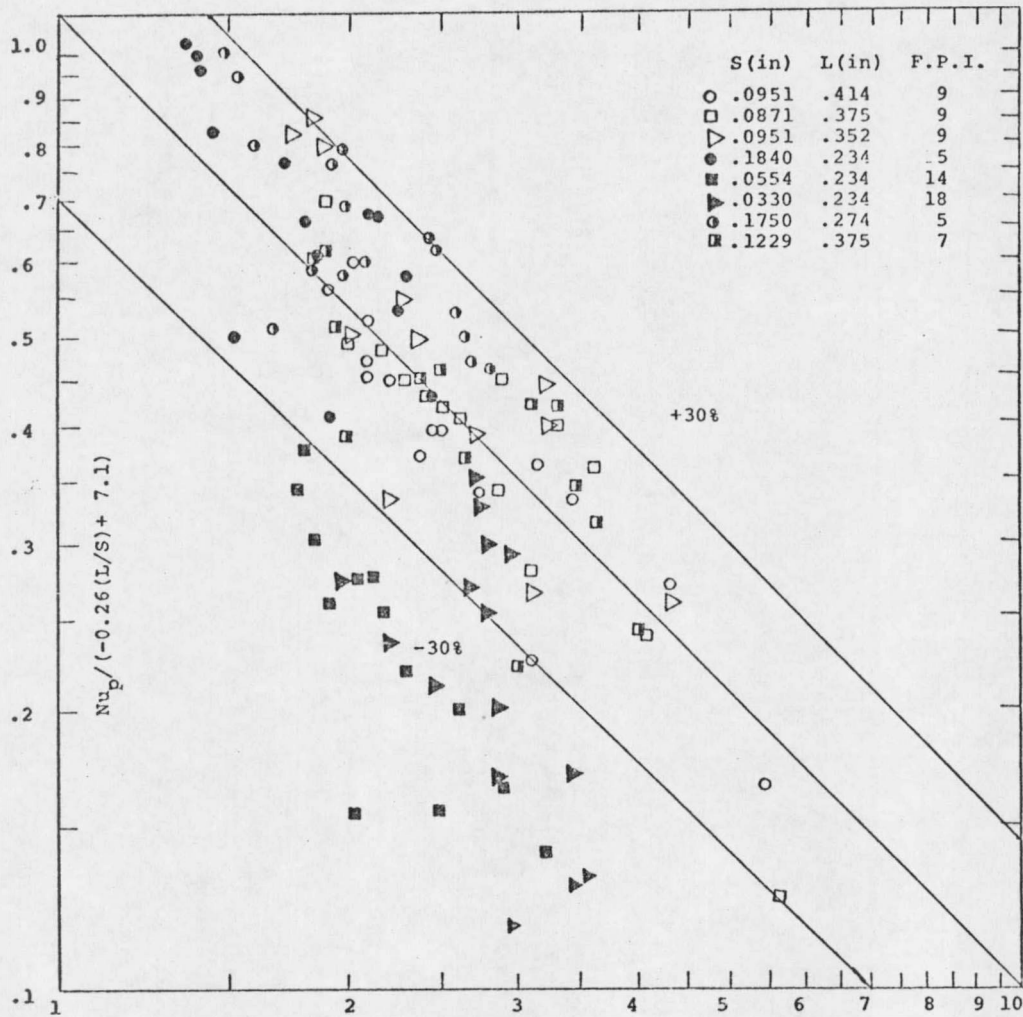
The discussion of the results is presented in two parts. The first section describes the results obtained with the copper helical finned tube system, and the second section presents results obtained with the coiled spiral tube system.

Helical Finned Tube System

A vertical bundle of seven helical finned tubes was used to study the effects of fin height, fin spacing, tube bundle center-to-center spacing, fluidizing gas mass velocity, and particle diameter.

All average heat transfer coefficients, h_{model} , reported in this section allow for the temperature gradient from the base to the tip of the fin.

The same helical finned tubes used in this investigation were used by Kratovil (19), but in the horizontal position. The correlation developed by Kratovil for the tubes in a horizontal position was used with my calculated h_{model} values. With the exception of when the fin spacing becomes small, 14 and 18 fins per inch, most of my data fit his correlation plus or minus 30 percent. His correlation with my data is plotted on Figure 19. The plus or minus 30 percent deviation seems to indicate that Kratovil's



$$\left[1 + .074 \text{Re}_p^{(-8.57(\frac{L}{S})^{-3.4}(\frac{D}{S})^{-.685(L/S)+1.96})(\frac{D}{S})^{-.77}(\frac{L}{S})^{.75}} \right]^{.24(\frac{L}{S})+1}$$

Figure 19. HORIZONTAL TUBE CORRELATION WITH VERTICAL TUBE DATA

correlation for a bundle of helical finned tubes in the horizontal position predicts fairly well the heat transfer from a vertical bundle with large fin spacings of 5, 7, and 9 fins per inch. Using his correlation with the data from the two tubes with small fin spacing gives a deviation as large as 75 percent. This large deviation seems reasonable because, in the vertical position the small fin spacing will hinder particle movement into and out of the fin more than if the tubes were in a horizontal position. In the vertical position the particles can become trapped in the fin spacing, causing an increase in the resistance to heat transfer. This was confirmed by my reported results of lower heat transfer coefficients than Kratovil reported in the horizontal position with these same tubes.

Effect of Mass Velocity and Particle Size on h_{model}

As the fluidizing gas mass velocity was increased, h_{model} generally increased, sometimes reached a maximum and then decreased with further increase in the gas mass velocity. The maximum occurs because of two opposing factors. With an increased mass velocity there is increased particle movement which results in shorter particle-surface residence times and higher heat transfer coefficients.

With the increased mass velocity there is also a higher void fraction which reduces the particle concentration adjacent to the surface and consequently reduces the coefficients.

In this investigation the plots of h_{model} versus air mass velocity for the small particles generally had the steepest positive slopes, followed by the slopes of the plots for medium particles, followed by the slopes of the plots for large particles.

The heat transfer coefficient increased with decreasing particle size. The increase was larger between the medium and small particles than between the large and medium particles. There were increases as large as 35 percent between the large and small particles. The dependence of h_{model} on particle size was lessened with decreasing fin spacing and increasing fin height. A representative plot of h_{model} versus air mass velocity, G , with the three different particle sizes for helical finned tube No. 7 is shown on Figure 20. The plots of h_{model} versus G for the rest of the helical finned tubes are shown on Figures 35-41 in the appendices of this thesis.

Effect of Fin Spacing on h_{model}

Three tubes with fin heights of .234 inch were used to

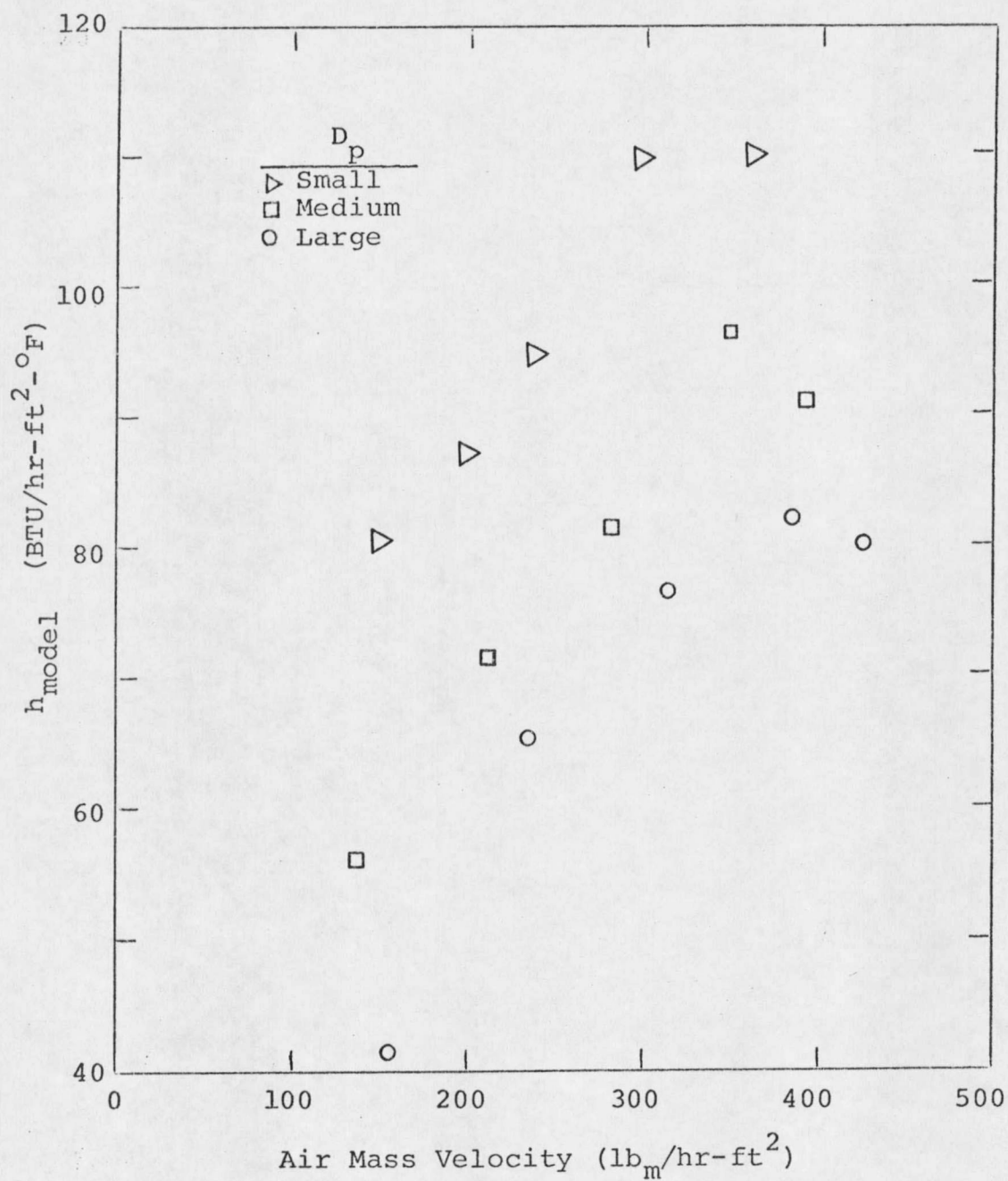


Figure 20. h_{model} VERSUS G VERSUS
PARTICLE DIAMETER (TUBE No. 7)

determine the dependence of h_{model} with fin spacing. The fin spacing varied from .0330 to .1840 inch. Nominal fins per inch were 5, 14, and 18.

The heat transfer coefficient increased with increasing fin spacing. With a smaller fin spacing, particle motion into and out of the fin becomes harder, resulting in an increase in the particle-surface residence time. This increase in particle-surface residence time reduces the rate of heat transfer per unit area, thus decreasing the heat transfer coefficient. Figure 21 is a plot showing the effects of fin spacing on the heat transfer coefficient as the air mass velocity is increased. This plot is for the large size particles. Plots for the small and medium size particles are shown on Figures 42 and 43 in the appendices of this thesis.

Figure 22 is a plot of the heat transfer coefficient versus the ratio of the particle diameter to the spacing between fins (D_p/s) at an air mass velocity ratio (G/G_{mf}) equal to 4.0. The tubes in this plot had constant fin heights of .234 inch.

The curve is steepest when D_p/s is less than .130, increased fin spacing, and flattens out for D_p/s greater

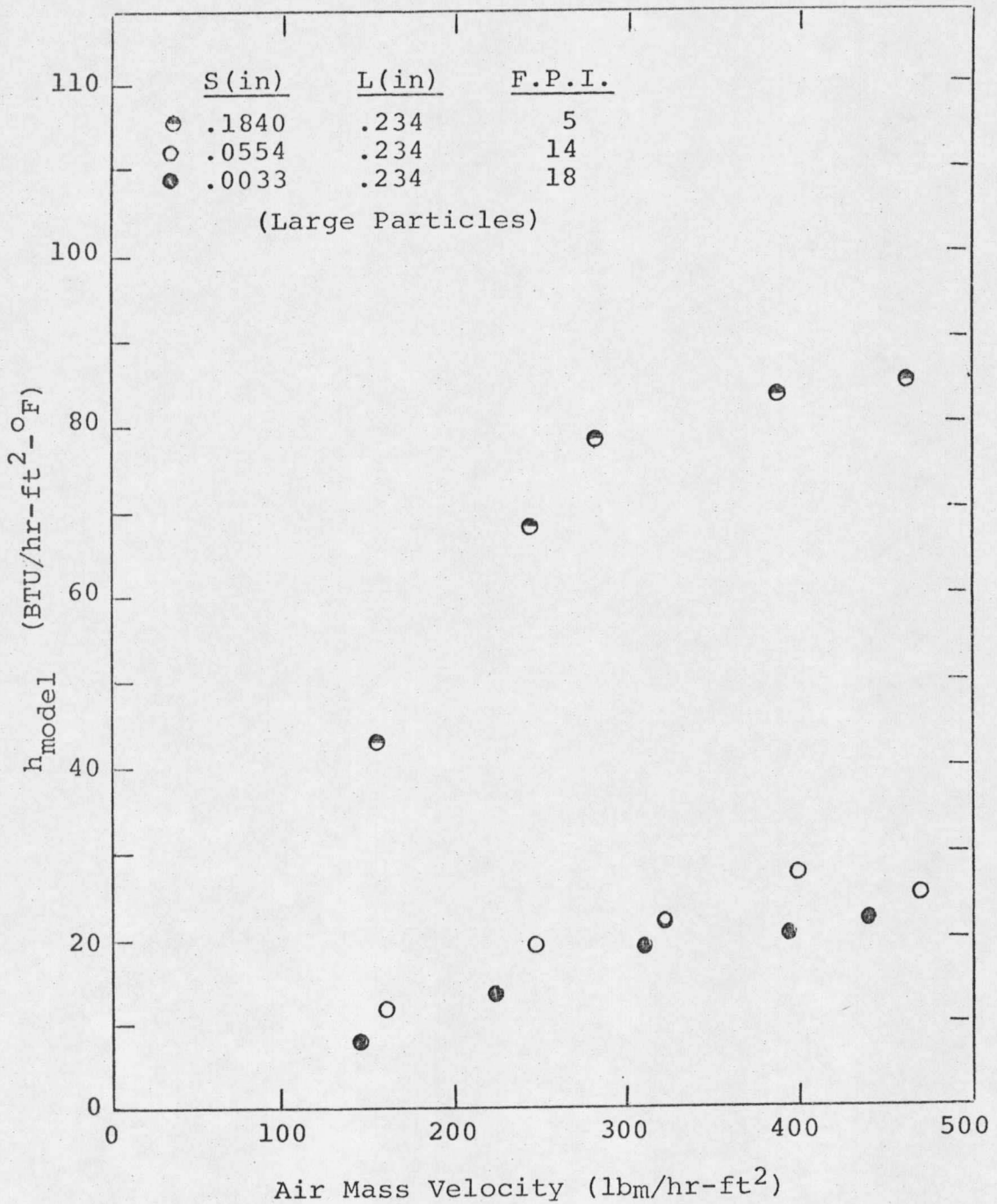


Figure 21. h_{model} VERSUS G VERSUS FIN SPACING (S)

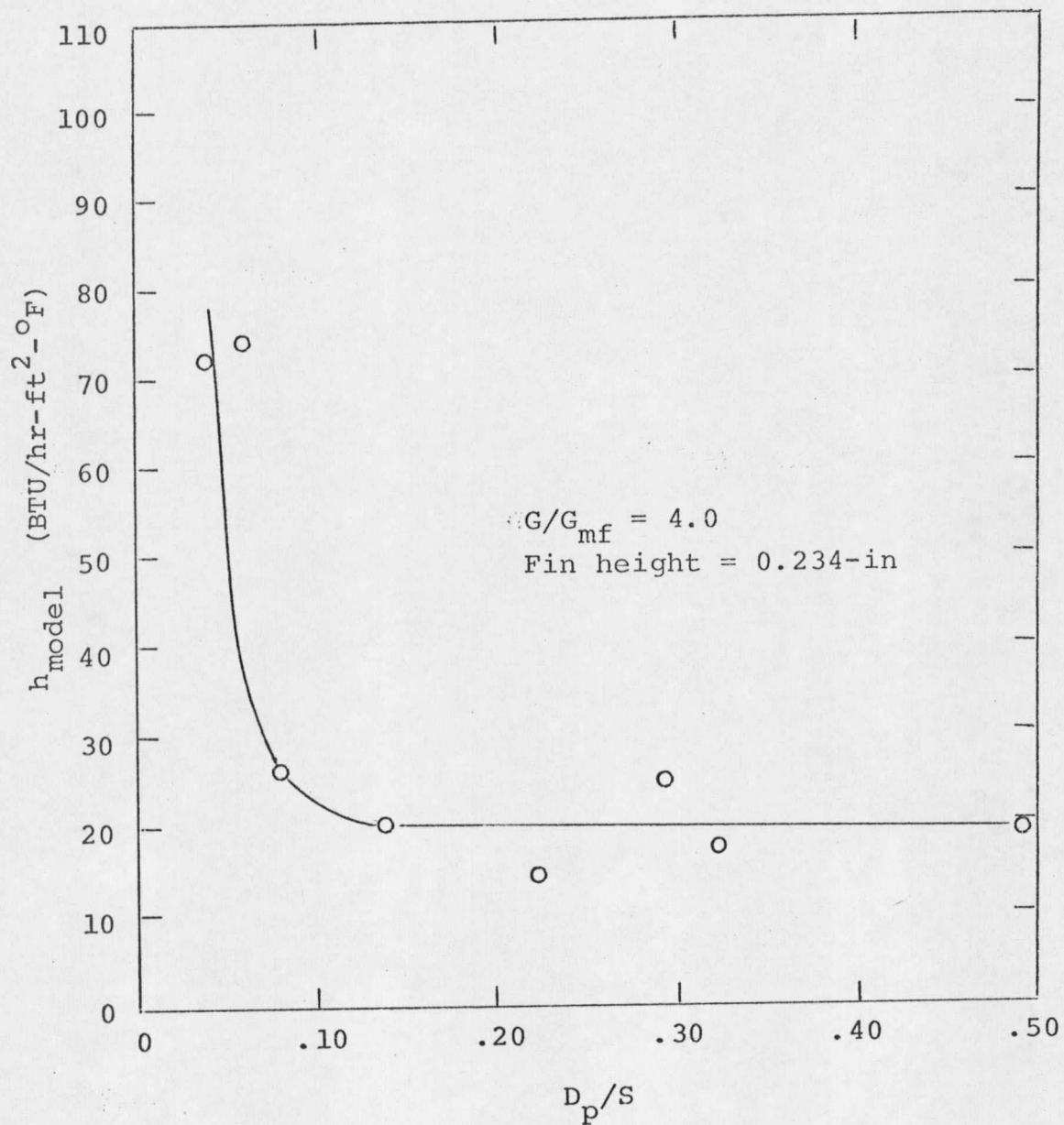


Figure 22. h_{model} VERSUS D_p/S (CONSTANT FIN HEIGHT)

than .130, decreased fin spacing. For D_p/s less than .130 the heat transfer coefficient is quite sensitive to fin spacing (greater than 8 particle diameters). This implies that the particles are free to move into and out of the fins, resulting in higher heat transfer coefficients because of reduced particle-surface residence times. For D_p/s greater than .130 the heat transfer coefficient is less sensitive to fin spacing (less than 8 particle diameters). This implies that the motion of the particles becomes more hindered with smaller fin spacing and the heat transfer coefficient levels out. Fin spacing in this range have a small effect on the heat transfer coefficient.

Kratovil (19) reported that the heat transfer coefficient is sensitive to fin spacings greater than 10 particle diameters and least sensitive to fin spacings less than 10 particle diameters for the helical finned tubes in a horizontal position.

Comparison of my results with Kratovils indicates that the finned tubes in the vertical position are more sensitive to fin spacing than in the horizontal position.

Effect of Fin Height on h_{model}

Three tubes with constant fin spacing of 9 fins per

inch were used to determine the dependence of the heat transfer coefficient with fin height. The fin height varied from .352 to .414 inch.

The heat transfer coefficient decreased with increasing fin height. This is what can be expected since particle movement into and out of the fin is hindered with longer fins. The hindered particle motion increases the particle-surface residence time thus decreasing the heat transfer coefficient.

Figure 23 is a representative plot showing the effect of increasing fin height on the heat transfer coefficient as the air mass velocity increases. This plot is for the medium sized particles. Plots for the small and large size particles are shown on Figures 44 and 45 in the appendices of this thesis.

In the range of fin heights and fin spacings looked at in this investigation, fin spacing seems to have more of an effect on the heat transfer coefficient than fin height.

Effect of Tube Center-to-Center Spacing on h_{model}

After the last helical finned tube No. 8 had been studied. The center-to-center tube spacing in the bundle

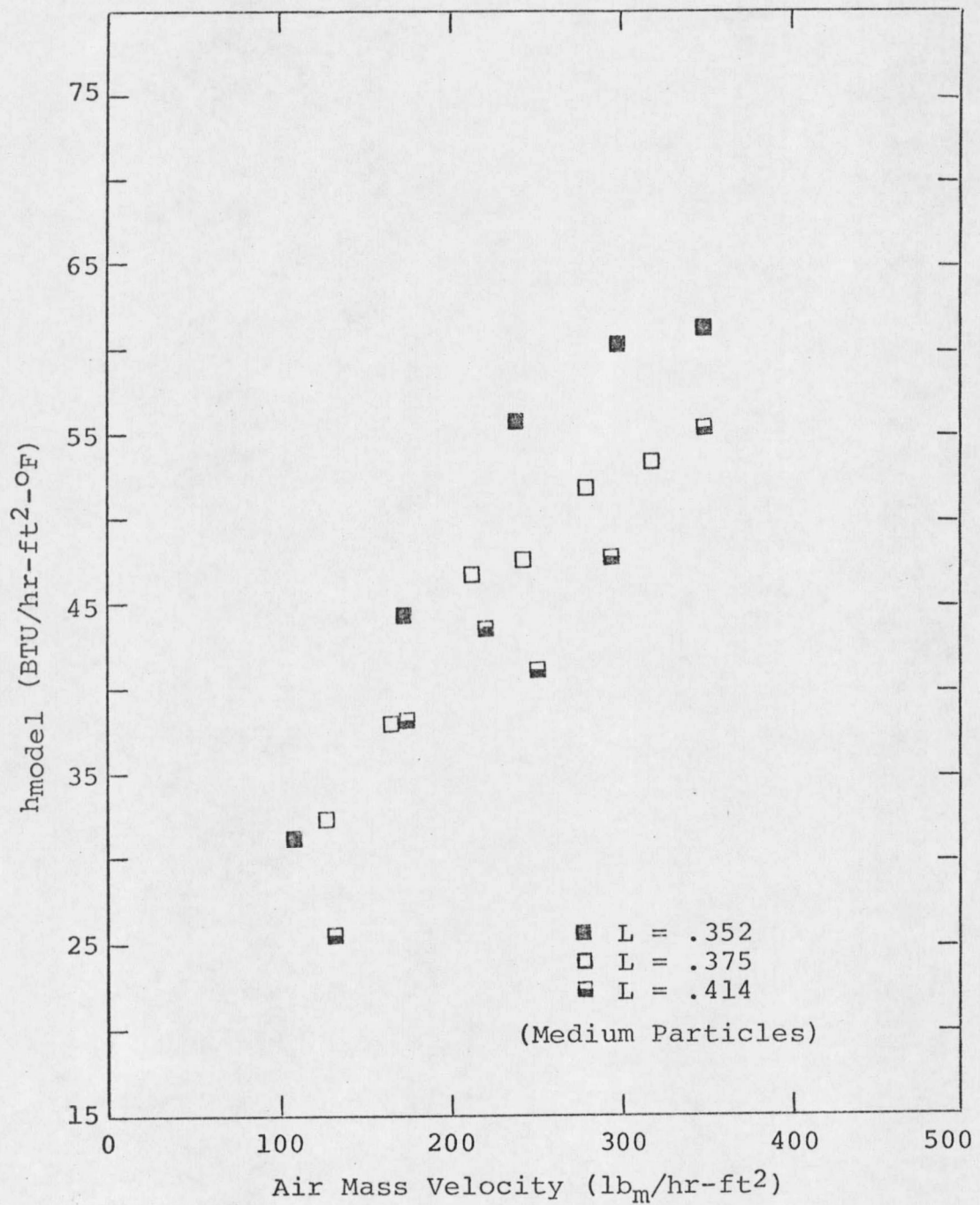


Figure 23. h_{model} VERSUS G VERSUS
FIN HEIGHT (L)

was increased from 2-inches to 3-inches, and then to 5-inches. At each different tube bundle spacing analysis was made using medium and large particle sizes. Figures 24 and 25 show the effect of increasing the center-to-center tube spacing on the heat transfer coefficient.

The heat transfer coefficient increased as the center-to-center spacing was increased from 2 to 3-inches. Increasing the spacing to 3-inches allowed more room for particle movement around the tubes than was available at the 2-inch spacing. This increased spacing enhanced more particle movement into and out of the fins on the tube, which resulted in higher heat transfer coefficients.

When the bundle spacing was increased to 5-inches, the heat transfer coefficient dropped below the values obtained at the 2-inch center-to-center spacing. At the 5-inch spacing the tubes were only 1.75 inches from the wall of the fluidizing column. Visual observation of the column while it was fluidizing revealed that the bubble action of the fluidizing gas was less along the wall than in the center of the column. This defluidization along the wall results in less particle motion which, increases the surface-particle residence time. Therefore, the heat transfer

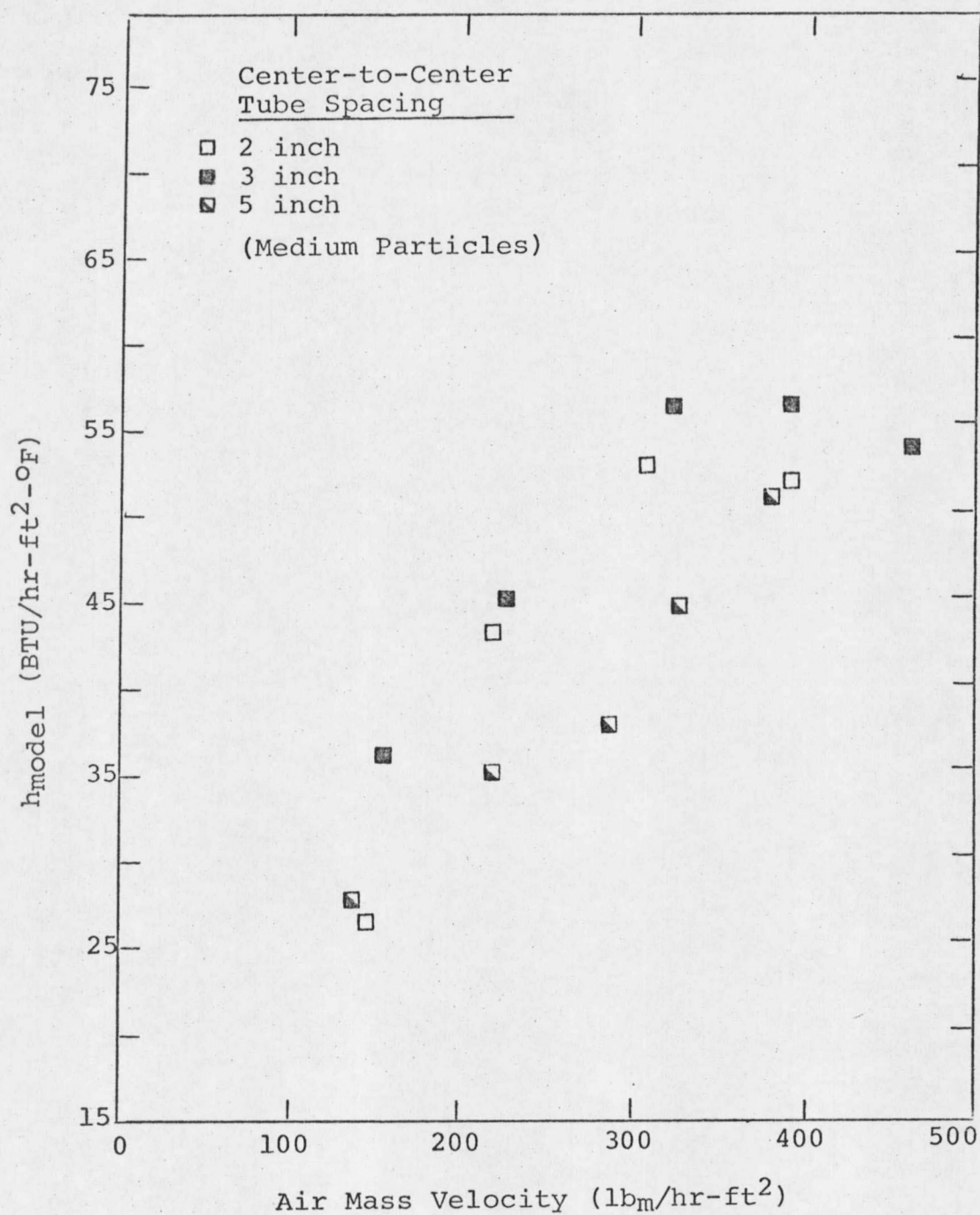


Figure 24 . h_{model} VERSUS G VERSUS CENTER-TO-CENTER TUBE SPACING (TUBE No. 8)

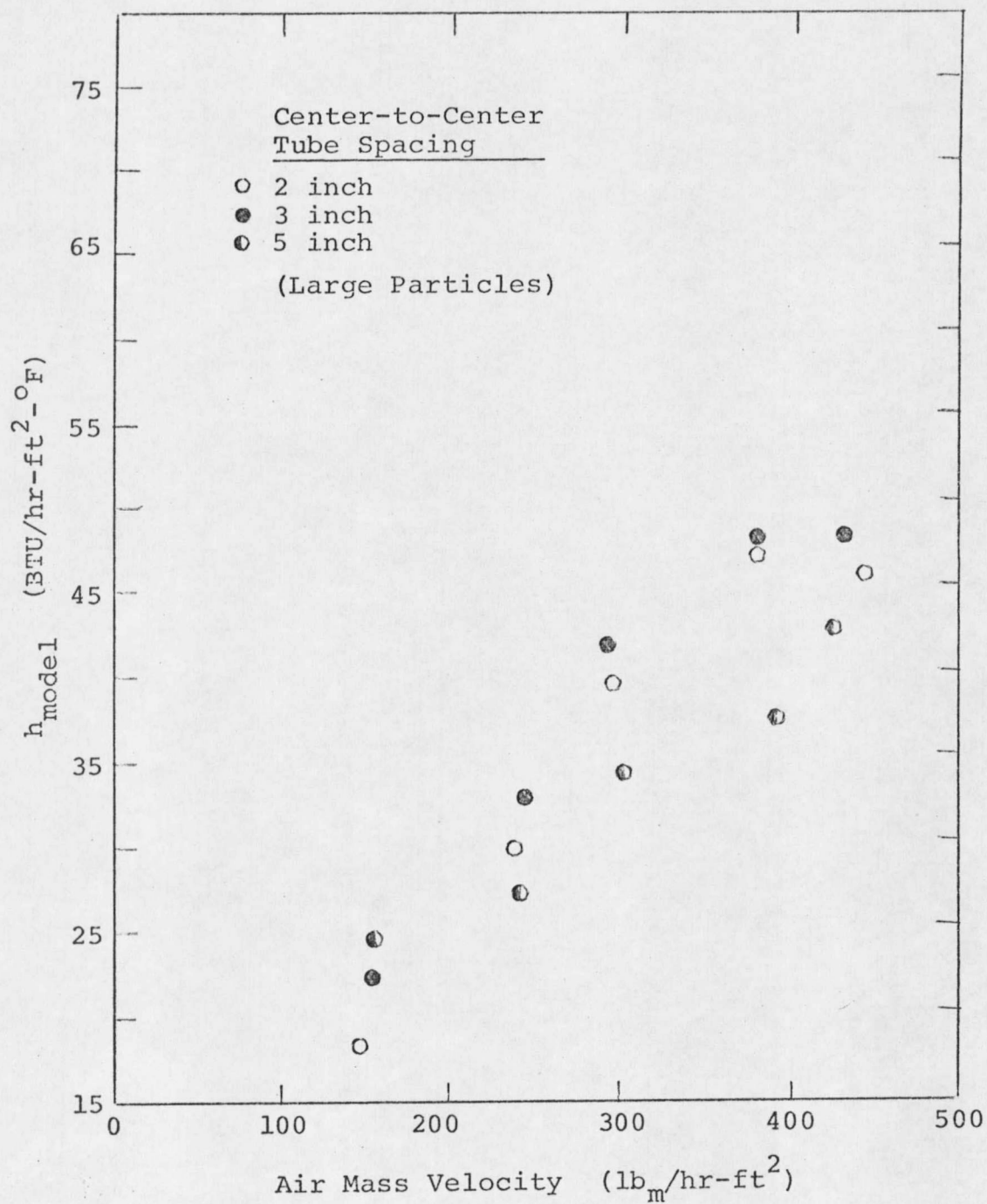


Figure 25. h_{model} VERSUS G VERSUS CENTER-TO-CENTER TUBE SPACING (TUBE No. 8)

coefficient will decrease.

Performance of Helical Finned Tubes

A measure of the helical fin tube performance is the ratio of the product of the heat transfer coefficient and the total area to the ratio of the product of the heat transfer coefficient and area of a plain tube occupying the same volume under identical fluidizing conditions (16).

The performance for this investigation was calculated using the following ratio:

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

where:

Q = heat transfered, BTU/hr

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

The area of the bare tube, used to determine $(Q/\Delta T)$ Bare Tube, was calculated using the 'overall fin diameter'. The 'overall fin diameter' is defined as the tube diameter across the fin tips. The heat transfer coefficients for a bare tube was obtained from work done by Chen and Withers (16).

When the performance ratio or performance exceeds one it can be expected that the finned tube provides higher

heat transfer duty per unit area than a bare tube.

All the helical finned tubes investigated had performances higher than unity. The average increase in the heat transfer coefficient over that of a bare tube was 137 percent. Vanderhoff (20) reported gains of 74 percent using a vertical bundle of carbon steel serrated finned tubes. Kratovil (19) reported gains of 190 percent using a horizontal bundle of helical finned tubes.

Figure 26 is a plot showing the effect of fin spacing on the tube performance as the fluidizing air mass velocity increases. The performance decreases with decreasing fin spacing. By increasing the fin spacing from 18 fins per inch to 5 fins per inch, a maximum increase of 24 percent occurs in the performance of the finned tubes.

Figure 27 is a plot showing the effect of fin height on the tube performance as the fluidizing air mass velocity increases. The performance decreases with increasing fin height. A maximum increase of 10 percent occurs in the performance by decreasing the fin height from .414 inches to .352 inches.

Figure 28 is a plot showing the effect of increasing the tube center-to-center spacing in the bundle on the

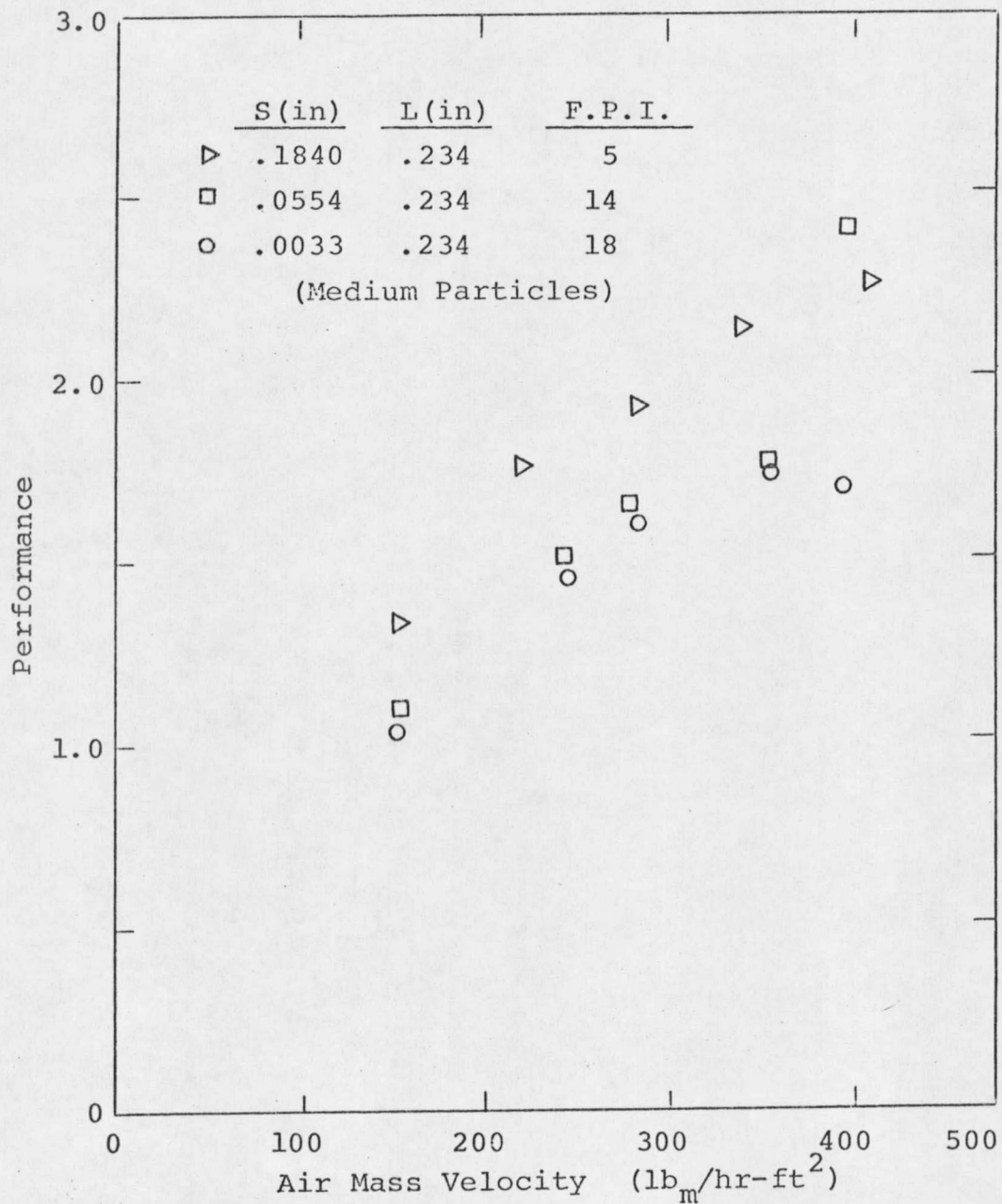


Figure 26. PERFORMANCE VERSUS G VERSUS FIN SPACING (S)

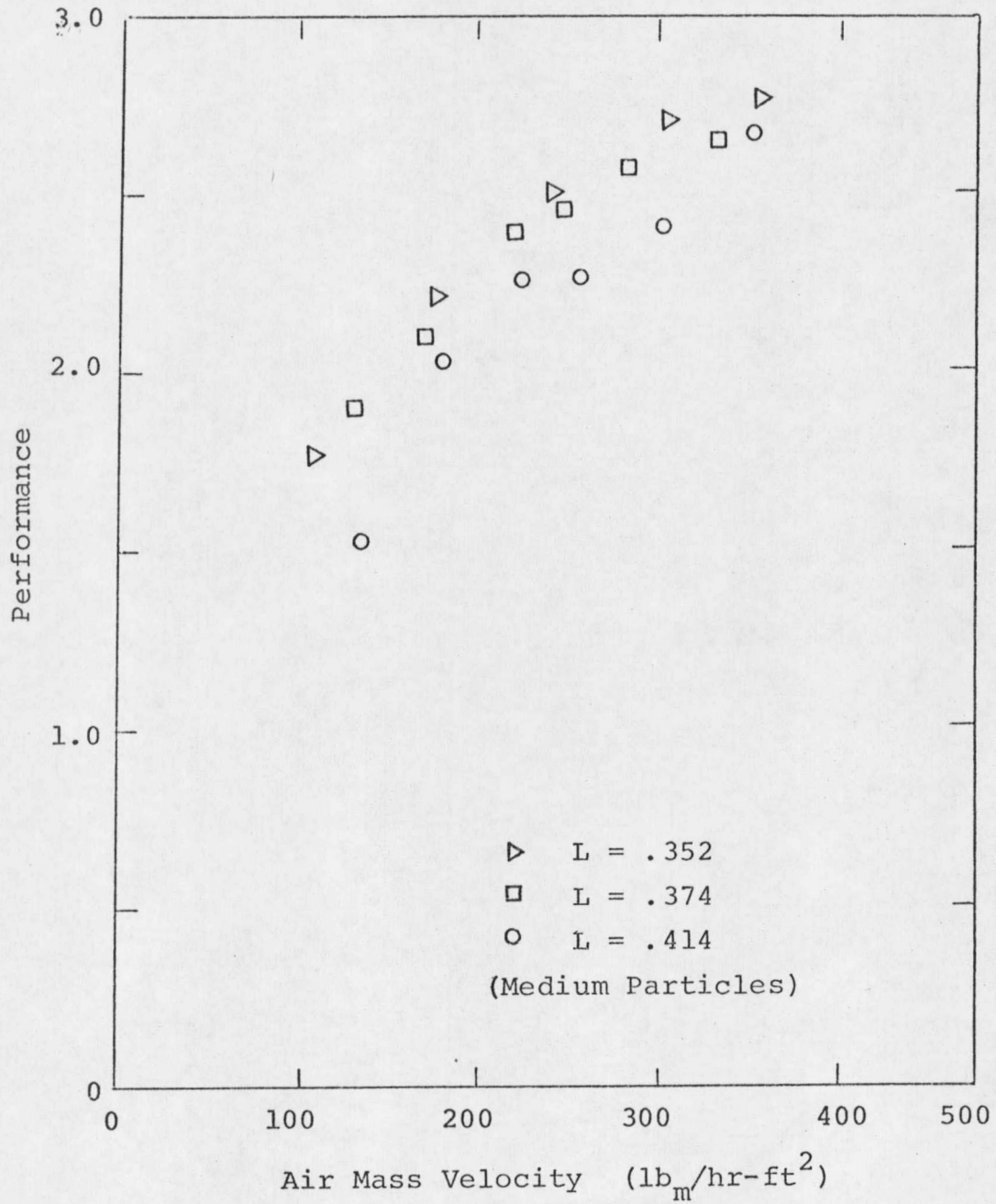


Figure 27. PERFORMANCE VERSUS G VERSUS FIN HEIGHT (L)

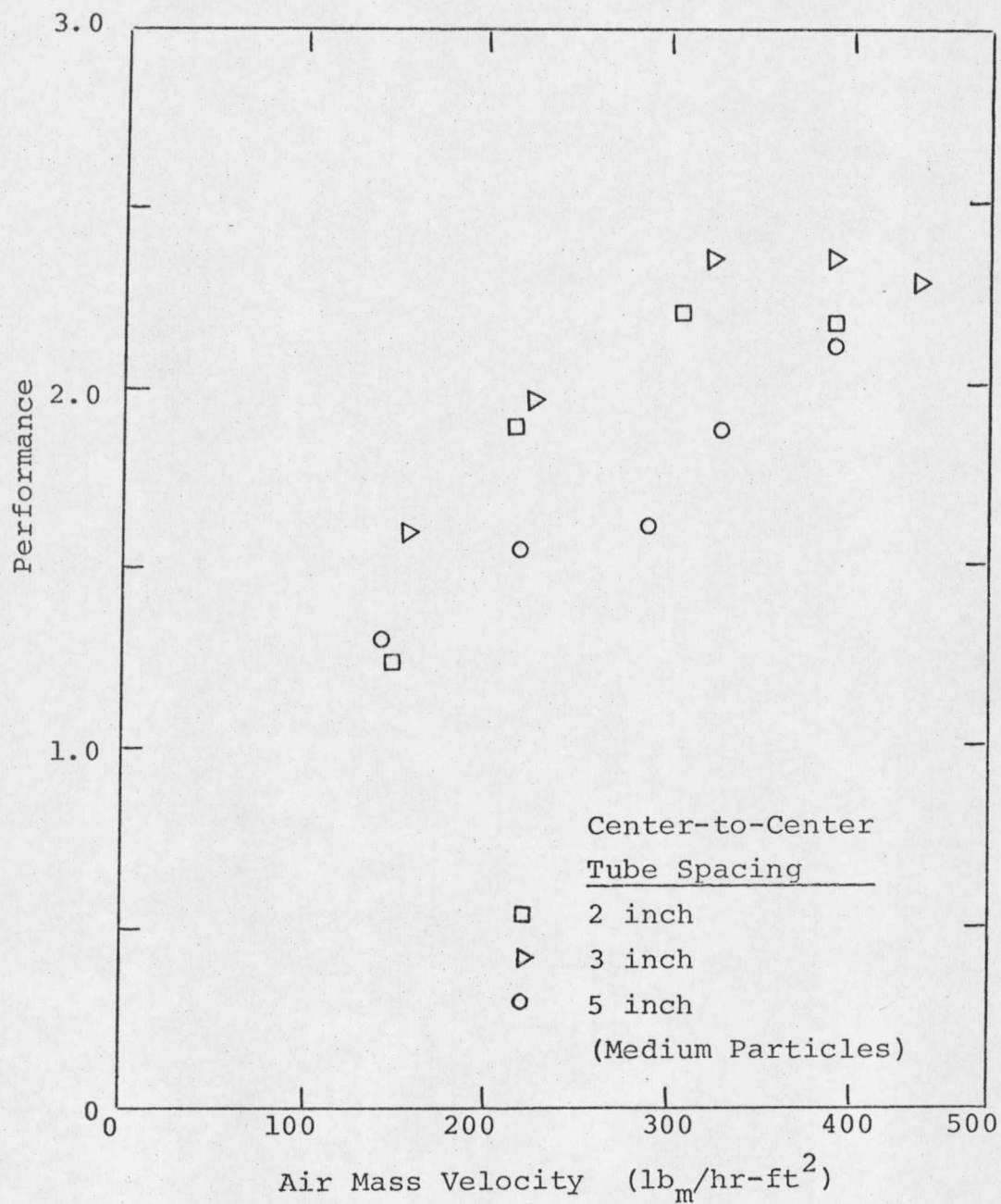


Figure 28. PERFORMANCE VERSUS G VERSUS CENTER-TO-CENTER TUBE SPACING

performance. At high air mass velocities the performance increased 10 percent when the center-to-center spacing was increased from 2-inches to 3-inches. When the center-to-center spacing was increased to 5-inches, a maximum decrease of 21 percent compared to the 3-inch spacing occurred, and a maximum decrease of 25 percent compared to the 2-inch spacing occurred.

In Table V the performance for all eight helical finned tubes are listed at an air mass velocity ratio (G/G_{mf}) equal to 4.0. The best performer based on the average performance of all three particle sizes was a tube of intermediate fin spacing with intermediate fin height.

Coiled Spiral Tube System

As was previously mentioned some of the coiled spiral tubes listed on Table IV were investigated previously by Everly (21) and Genetti and Everly (22). I have studied tubes 5A and 6A in this investigation.

A 6-foot length of spiral copper tube, bent into a 7-inch inside diameter coil was used to study the effects of groove depth, flute pitch, particle diameter, and gas mass velocity.

The following equation is the definition for the

Table V. PERFORMANCE OF HELICAL FINNED TUBES AT $G/G_{mf} = 4.0$

Tube No.	Fin F.P.I.	Fin Spacing	Fin Height	P (Sm Beads)	P (Med Beads)	P (Lg Beads)	Ranking Based on Avg. P
1	9	.0951	.414	1.84	2.21	3.36	3
2	9	.0871	.375	1.69	2.40	3.76	2
3	9	.0951	.352	2.08	2.46	4.17	1
4	5	.1840	.234	1.65	1.80	2.89	5
5	14	.0554	.234	1.07	1.46	2.29	7
6	18	.0330	.234	1.05	1.40	2.09	8
7	5	.1750	.274	1.78	1.84	2.99	4
8	7	.1229	.375	1.51	1.92	2.77	6

relationship between the overall heat transfer coefficient, and the inside and outside heat transfer coefficients.

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{r D_o}{k D_L} + \frac{A_o}{A_i h_i}$$

The overall heat transfer coefficient, U_o , was calculated from experimental measurements using the following equation.

$$U_o = \frac{W_{H_2O} C_{pH_2O} \ln((T_i - T_b)/(T_o - T_b))}{A_o}$$

In order to obtain the inside heat transfer coefficient, h_i , a Wilson plot analysis was used. This was accomplished by varying the water flow rate through the coiled spiral tube holding the other variables of the investigation constant. The Wilson plot analysis using the medium size particles is shown on Figure 29, for tubes No. 5A and 6A. For both tubes, a straight line relationship fit the data for the range of water flow rates investigated. Since the heat transfer properties of the fluidized bed do not depend on the water flow rates in the tube, the following equation was used to obtain the inside heat transfer coefficient:

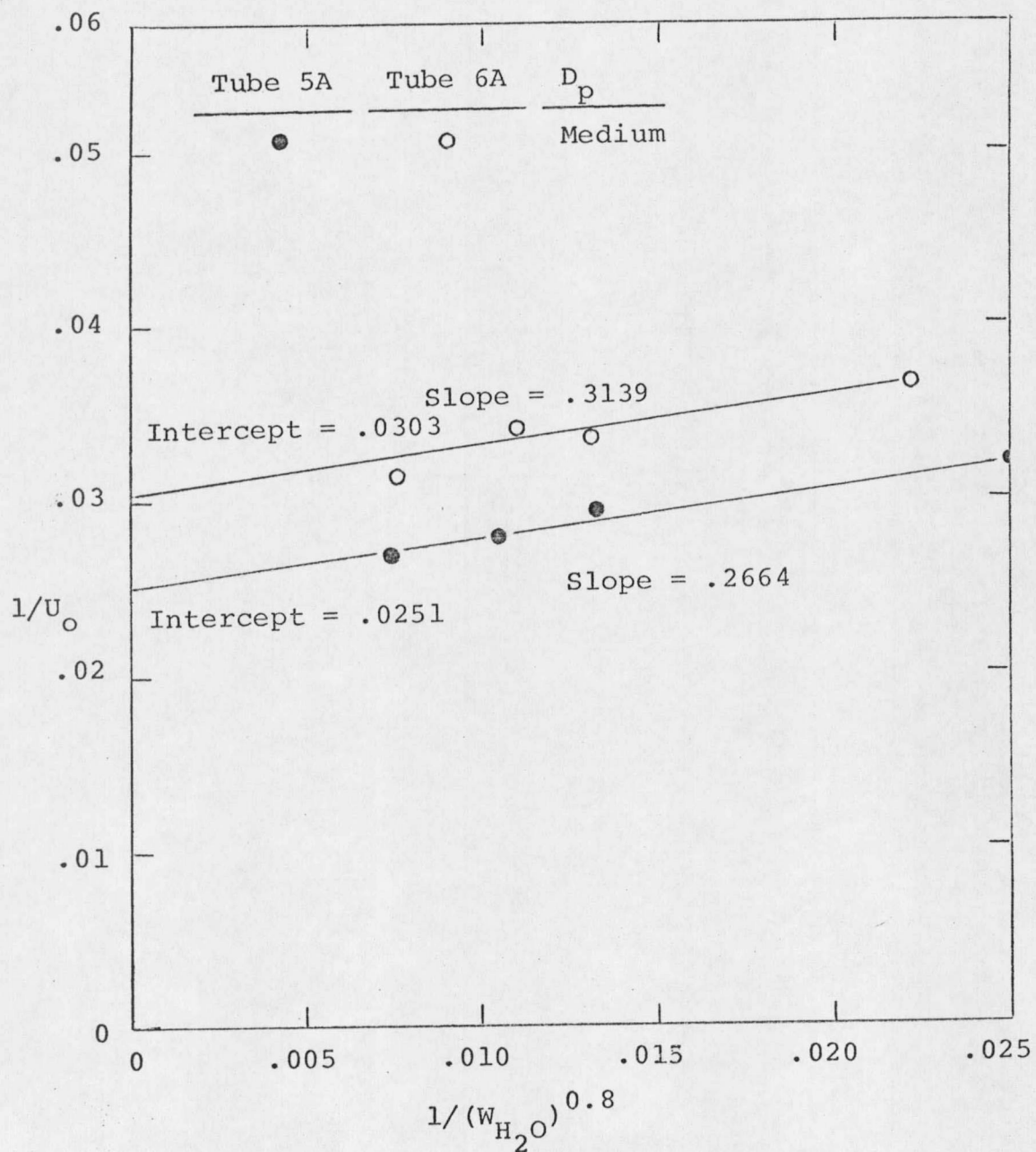


Figure 29. WILSON PLOT ANALYSIS

$$\frac{A_o}{h_i A_i} = \frac{\text{slope}}{(W_{H_2O})^{.8}}$$

The contribution of conduction within the copper tube wall was negligible; therefore, the fluidized bed heat transfer coefficient, h_o , is related to the overall coefficient and the water flow rate by the expression:

$$h_o = \frac{1}{1/U_o - \text{slope}/(W_{H_2O})^{.8}}$$

Everly (21) correlated the heat transfer coefficients he obtained for tubes No. 1A and 2A using the particle model for heat transfer described previously in this thesis. Figure 30 shows his correlation with the experimental points I obtained in this investigation with tubes No. 5A and 6A. Most of the experimental data fit the correlation to within plus or minus 15 percent.

Genetti and Everly (22) were able to obtain a fit of plus or minus 15 percent using Everly's correlation with tube No. 3A. Thus Everly's correlation seems to accurately predict the heat transfer coefficient for coiled spiral tubes for the given range of applicability listed on Table VI.

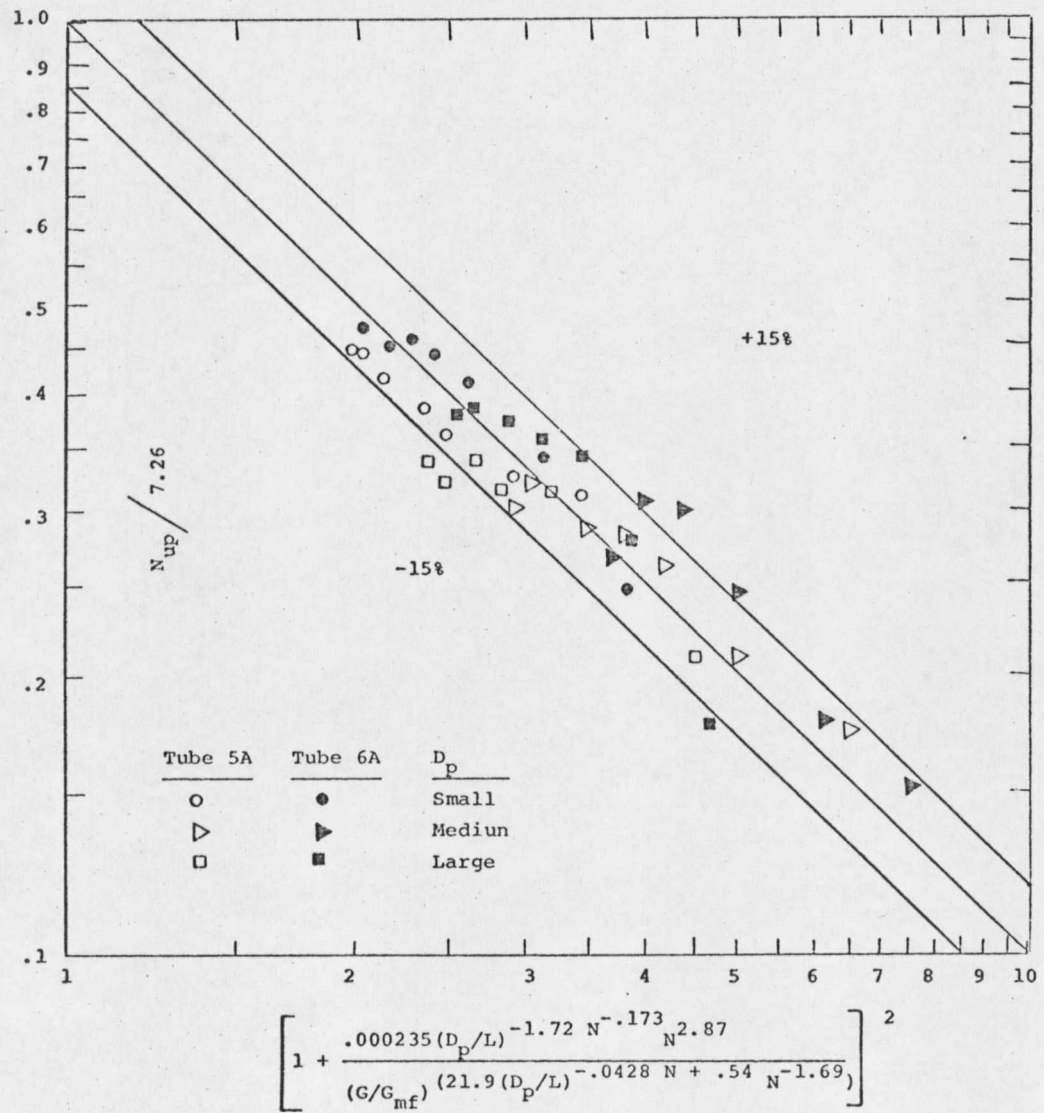


Figure 30. CORRELATION FOR SPIRAL TUBES

Table VI RANGE OF CORRELATION APPLICABILITY

Variables	Range
Particle Diameter	.0076 - .0164, inches
Groove Depth	.101 - .210, inches
Number of Flutes, N	3 to 5
Pitch, P	2 to 3.1, inches
Air Mass Velocity, G	100 to 550, lbm/hr-ft ²
Bed Material	glass spheres

Effect of Mass Velocity and Particle Size on h_o

The fluidized bed heat transfer coefficient, h_o , is plotted versus the air mass velocity, G, on Figures 31 and 32 for the coiled spiral tubes No. 5A and 6A respectively.

It was generally observed that the heat transfer coefficient increased, reaches a maximum value, and then decreased with increasing fluidizing air mass velocity. The maximum is a result of the same two opposing factors mentioned earlier in this section for the helical finned tube system.

Figures 31 and 32 also show that the heat transfer coefficient increased with decreasing particle size.

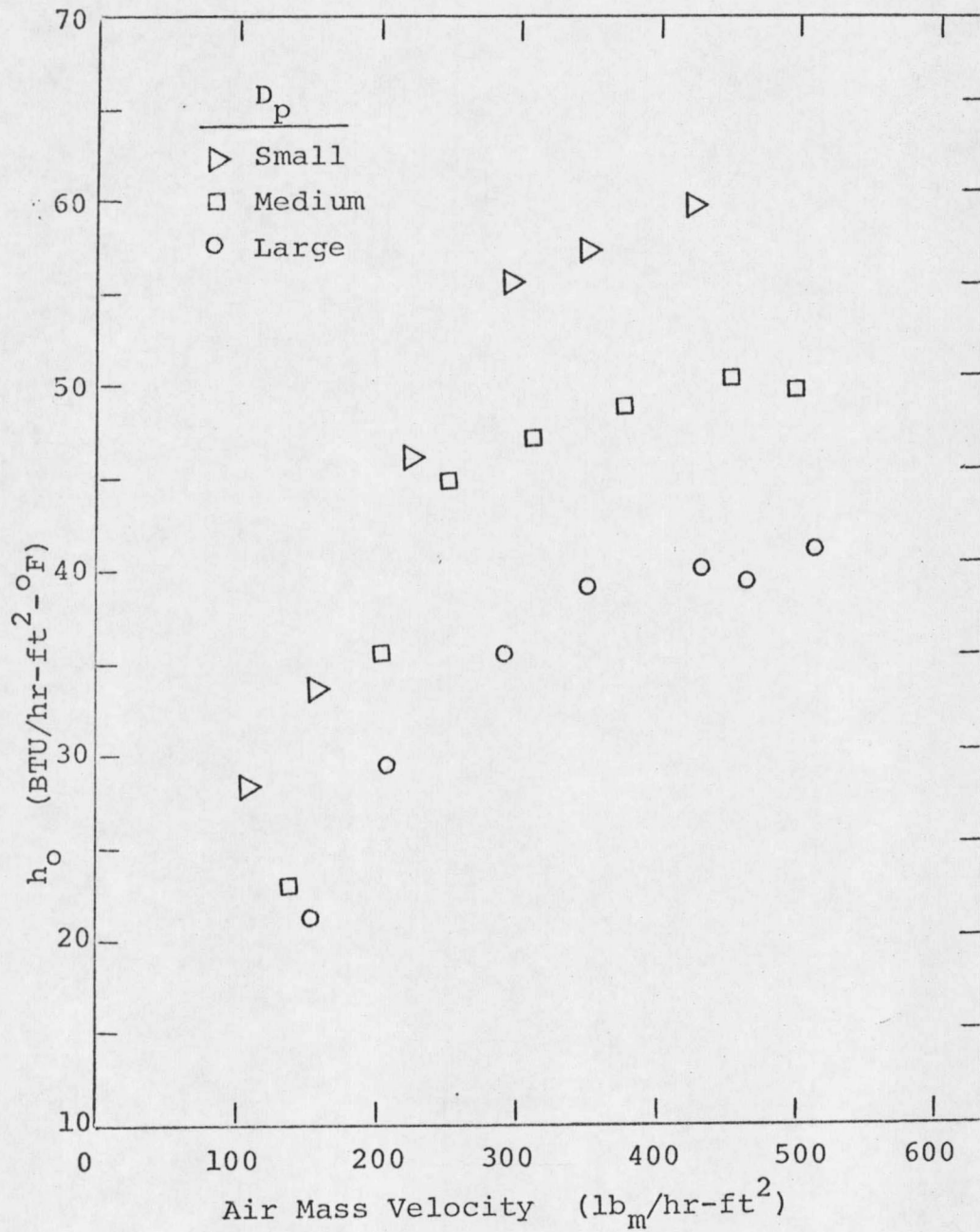


Figure 31. h_o VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 5A)

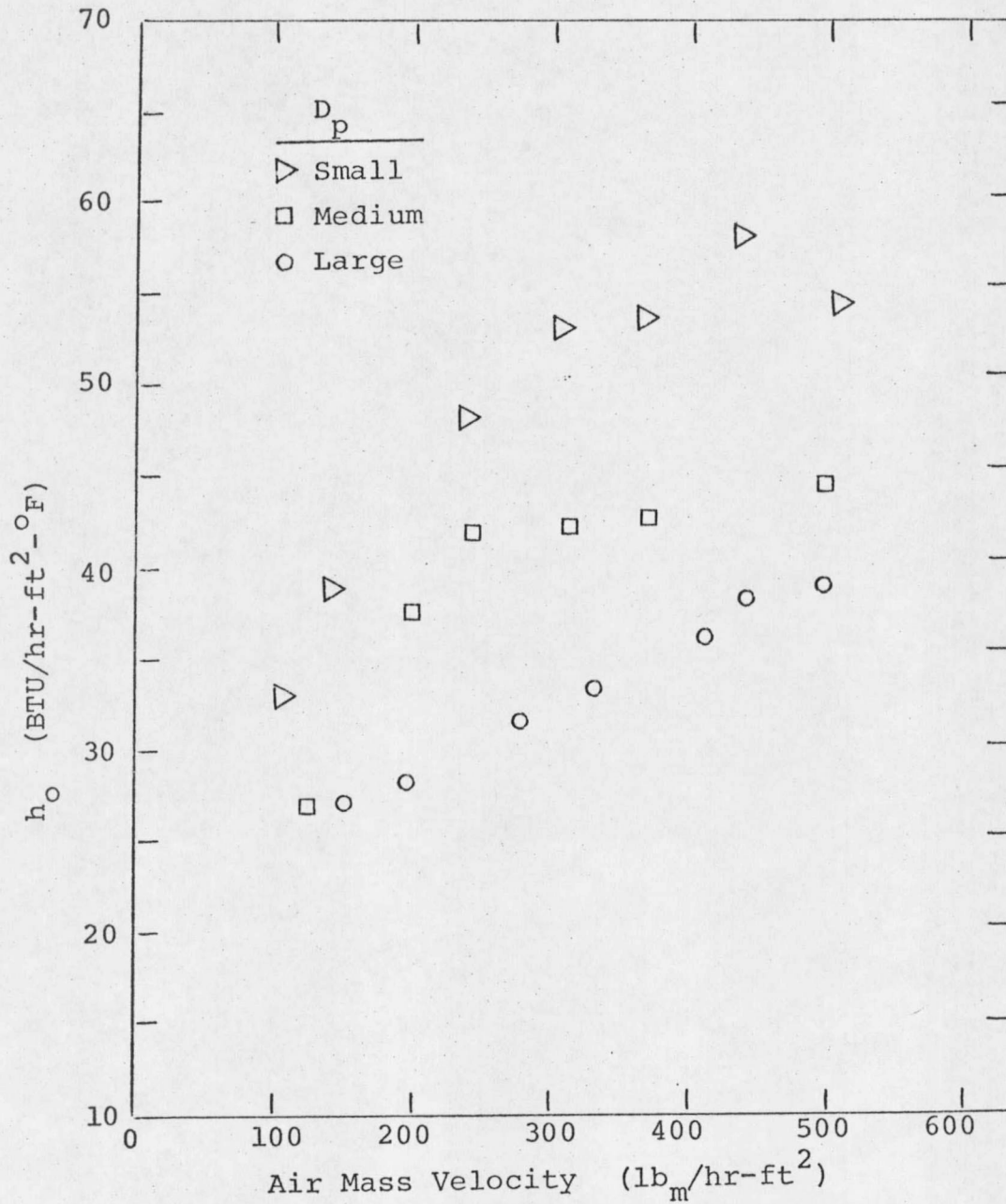


Figure 32. h_o VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 6A)

Increases in the coefficient as large as 45 percent were noted between the small and large particles for tube No. 5A; and 36 percent for tube No. 6A.

Effect of Groove Depth and Pitch on h_o

The two coiled spiral tubes used in this investigation had 4 flutes. Another coiled spiral tube reported by Genetti and Everly (22) also had 4 flutes. Figure 33 shows a plot of the heat transfer coefficient, h_o , versus the air mass velocity, G , for these three coiled spiral tubes.

It was generally observed that the heat transfer coefficient increased with decreasing groove depth, δ , and increasing pitch, P . The maximum increase was 20 percent between tube No. 6A ($\delta = .101$, $P = 3.1$) and tube No. 3A ($\delta = .184$, $P = 2.23$).

As the groove depth decreases and the flute pitch increases, the tube approaches the geometry of a bare tube. It would therefore seem reasonable that the heat transfer coefficient would increase with decreasing groove depth and increasing flute pitch.

Performance of Coiled Spiral Tubes

Genetti and Everly (22) concluded that the performance ratio ($h_o A_o / A_b h_b$) is a function of the groove depth, δ ;

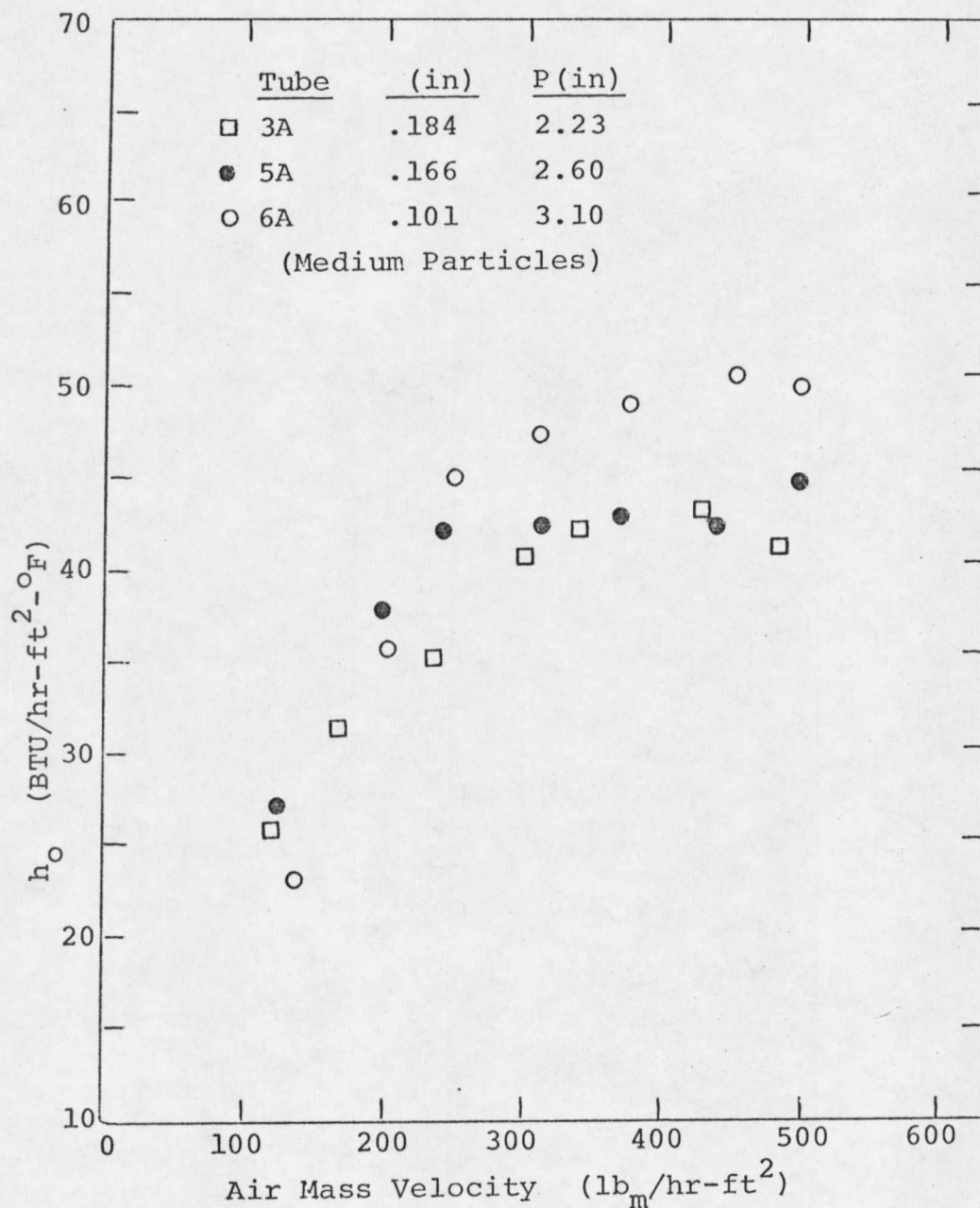


Figure 33. h_o VERSUS G VERSUS GROOVE DEPTH AND PITCH

particle diameter, D_p ; flute pitch, P ; and number of flutes, N . Based on results from this investigation and previous investigations (21,22), the performance ratio appears to go through a maximum as $P\delta/N$ increases. This parameter is proportional to the space between the flutes. As this space increases particle motion is enhanced, thereby increasing the heat transfer coefficient. At the same time the area for heat transfer is decreasing causing the performance ratio to go through a maximum (22). Figure 34 for $G = 200 \text{ lb}_m/\text{hr-ft}^2$ is a plot of the performance ratio versus $P\delta/N$. Tube No. 3A ($\delta = .184$, $P = 2.23$, $N = 4$) of intermediate geometry was the best performer.

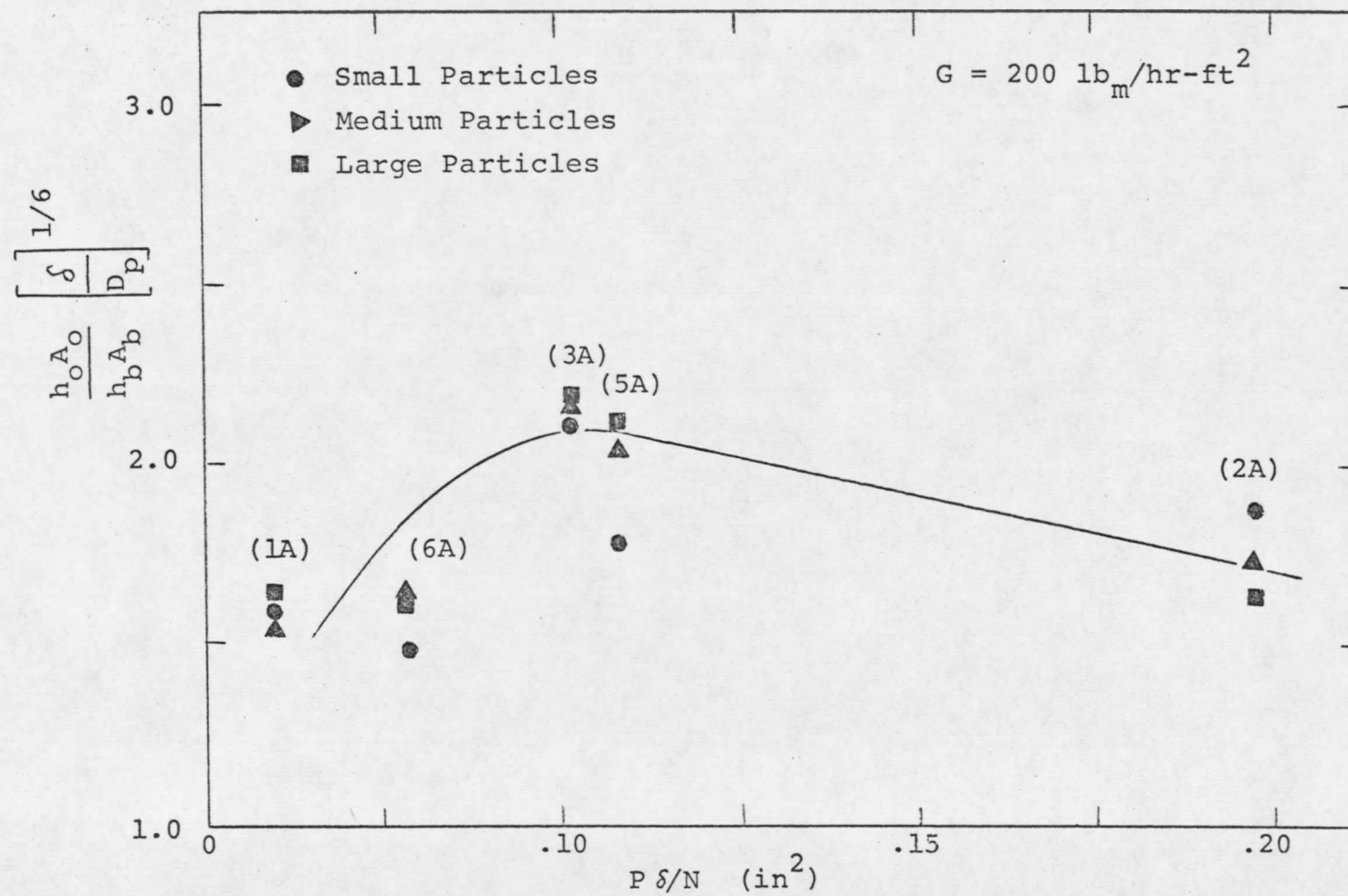


Figure 34. PERFORMANCE RATIO FOR COILED SPIRAL TUBES

ERROR ANALYSIS

The discussion of the error analysis will be divided into two main parts: the helical finned tube system and the coiled spiral tube system.

Helical Finned Tube System

Assuming the experimental heat transfer is only affected by the measurements of q_{exp} and $(T_w - T_b)$, the error analysis is performed on the following equation:

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

The wattmeter was assumed to be accurate within ± 5 percent. The bed temperature is assumed to be measured within $\pm .5^\circ\text{F}$ and the tube wall temperature could be measured within $\pm 1.5^\circ\text{F}$. The minimum $(T_w - T_b)$ value in this investigation was 12.25°F .

The maximum and minimum errors for h_{exp} can be determined using the previous assumptions. This analysis was based on an h_{exp} 'true' value of 1.0.

Maximum h_{exp}

$$\frac{h_{exp}}{h} = \frac{1.05}{1 - 2.00/12.25} = 1.25$$

$$\text{Error} = (h_{exp} - h)/h = (1.254 - 1.0) \times 100 = 25.0 \%$$

Minimum h_{exp}

$$\frac{h_{exp}}{h} = \frac{.95}{1 + 2.00/12.25} = 0.82$$

$$\text{Error} = (h_{exp} - h)/h = (0.82 - 1.0) \times 100 = -18.0 \%$$

Therefore the experimental error range bracketing all results is +25.0 % and -18.0 %. Therefore, maximum deviations from 'true' values should be about $\pm 25 \%$.

Coiled Spiral Tube System

Assuming the overall heat transfer coefficient, U_o , is only affected by the measurements of W_{H_2O} , T_i , T_b , and T_o . The error analysis is performed on the following equation:

$$U_o = \frac{W_{H_2O} C_{pH_2O} \ln((T_i - T_b)/(T_o - T_b))}{A_o}$$

The measurement of W_{H_2O} was assumed to be accurate to 5 lbm/hr. No error was assumed in determining the tube surface area and heat capacity of the water. The temperature measurements are accurate to $\pm .5^\circ\text{F}$. The minimum value for $\ln(T_i - T_b/T_o - T_b)$ in this study was .144. Using the smallest value of U_o , the experimental error is at the maximum effect.

Using the above experimental accuracies, maximum and minimum errors for U_o are determined.

Maximum U_o

$$U_{oMax} = \frac{(W_{H_2O} + 5) (C_{pH_2O}) \ln \frac{T_i + .5 - T_b + .5}{T_o - .5 - T_b + .5}}{A_o}$$

$$\text{Error} = \frac{(U_{oMax} - U_o)}{U_o} \times 100 = 15 \text{ percent}$$

Minimum U_o

$$U_{oMin} = \frac{(W_{H_2O} - 5) (C_{pH_2O}) \ln \frac{T_i - .5 - T_b - .5}{T_o + .5 - T_b - .5}}{A_o}$$

$$\text{Error} = \frac{(U_{oMin} - U_o)}{U_o} \times 100 = -14 \text{ percent}$$

Therefore, since this is the lowest heat transfer coefficient value encountered, the maximum error should be about + 15.0 percent

CALCULATIONS

This section is divided into the following three parts: the equations that were used by both the helical finned tube system and the coiled spiral tube system to analyze the experimental data, the equations that were used to analyze only the experimental data of the helical tube system, and the equations that were used to analyze only the experimental data of the coiled spiral tube system.

Common Calculations to Both Systems

Air Mass Velocity

As mentioned previously, a vena contracta orifice with a water manometer was used to determine the air mass velocity to the columns. A standard equation for the orifice is used.

$$G = \frac{3600 C_o Y_s}{A_c} \sqrt{\frac{2g_c(P_1 - P_2)P_1}{1 - B^4}}$$

where,

- G = air mass velocity, $\text{lb}_m/\text{hr-ft}^2$
- C_o = orifice coefficient, dimensionless
- Y = expansion factor, dimensionless
- S_c = cross-sectional area of orifice, ft^2
- A_c = cross-sectional area of column, ft^2
- g_c = gravitational constant, $\text{ft-lb}_m/\text{hr}^2\text{-lbf}$
- P_1 = pressure at upstream pressure tap, lbf/ft^2
- P_2 = pressure at downstream pressure tap, lbf/ft^2

ρ_1 = density of air at the upstream pressure,
lb_m/ft³

B = ratio of orifice diameter to inside pipe
diameter, dimensionless

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

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

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.

Temperatures

As was mentioned previously, the temperatures were
read directly, using thermocouples connected to a Brown
Potentiometer.

Bed Temperature

The value used for the bed temperature, T_b , was the
average of the three bed thermocouple readings.

Air Thermal Conductivity

Air thermal conductivity, k_g , was determined by linear interpolation between selected table values in Kreith (23). Evaluation temperature was the bed temperature, T_b , and the units are BTU-ft/hr-ft²-°F.

Air Viscosity

Air viscosity was calculated for each bed temperature from the following equation which was fit to experimental data.

$$\mu_g = (2.45(T_b - 32) + 1538.1) \times 10^{-5}, \text{ lb}_m/\text{ft-hr}$$

where,

$$\begin{aligned} \mu_g &= \text{air viscosity, lb/ft-hr} \\ T_b &= \text{bed temperature, } ^\circ\text{F} \end{aligned}$$

Particle Reynolds Number

$$Re_p = \frac{GD_p}{\mu_g}, \text{ dimensionless}$$

where,

$$\begin{aligned} G &= \text{air mass velocity, lb}_m/\text{hr-ft}^2 \\ D_p &= \text{particle diameter, ft} \\ \mu_g &= \text{air viscosity evaluated at bed temperature,} \\ &\quad \text{lb}_m/\text{ft-hr} \end{aligned}$$

Helical Finned Tube System

Heat Input to Each Tube

Electrical power input to each tube was measured with a Simpson Wattmeter. A conversion factor of 3.413 BTU/watt-hr was used to convert the measured watts to BTU/hr.

Surface Area of the Helical Finned Tube

The surface area of each tube was determined by calculating the bare area and adding on the finned area. The finned area was determined by multiplying the area of the fins by the number of fins on the tube.

Experimental Heat Transfer Coefficients

The experimental heat transfer coefficient for each finned tube was calculated from the standard equation for convection from a surface.

$$h_{\text{exp}} = \frac{q}{A_T(T_w - T_b)}, \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$$

where,

h_{exp} = experimental heat transfer coefficient,
BTU/hr-ft²-°F

A_T = total tube area (fin + bare areas), ft²

T_w = heat transfer surface temperature, °F

T_b = bed temperature, °F

The experimental heat transfer coefficient for the bundle of finned tubes was determined by averaging the seven individual finned tube h_{exp} values.

Particle Nusselt Number

$$Nu_p = \frac{h_{model} D_p}{k_g}, \text{ dimensionless}$$

where,

Nu_p = particle Nusselt number, dimensionless

h_{model} = heat transfer coefficient based on analytical temperature profile in a fin, BTU/hr-ft²-°F

k_g = air thermal conductivity, BTU-ft/hr-ft²-°F

D_p = particle diameter, ft

Coiled Spiral Tube System

Water Mass Velocity

The water mass velocity, W_{H_2O} , was measured by weighing the amount of water discharged over a given time period.

Surface Area of the Coiled Spiral Tube

The outside surface area of each coiled spiral tube, A_o , was determined by measuring the reduced length of each spiral tube before it was coiled. This was accomplished by multiplying the area of each ridge by the number of ridges and adding on the bare area between the ridges and the bare area of the ends of the spiral tube.

Overall Heat Transfer Coefficient

$$U_o = \frac{W_{H_2O} C_{PH_2O} \ln \frac{T_i - T_b}{T_o - T_b}}{A_o}, \text{ BTU/hr-ft}^2\text{-}^{\circ}\text{F}$$

where,

U_o = overall heat transfer coefficient, BTU/hr-ft²-°F

C_{PH_2O} = heat capacity of water, BTU/lb_m-°F

W_{H_2O} = water mass velocity, lb_m/hr

A_o = outside surface area of coiled spiral tube, ft²

T_i = inlet water temperature, °F

T_o = outlet water temperature, °F

T_b = bed temperature, °F

Outside Heat Transfer Coefficient

$$h_o = \frac{1}{\frac{1}{U_o} - (\text{slope}/W_{H_2O}) \cdot 0.80}, \text{ BTU/hr-ft}^2\text{-}^{\circ}\text{F}$$

where,

h_o = outside heat transfer coefficient, BTU/hr-ft²-°F

U_o = overall heat transfer coefficient, BTU/hr-ft²-°F

W_{H_2O} = water mass velocity, lb_m/hr

The term $(\text{slope}/W_{H_2O})^{.8}$ was determined for each coiled spiral tube from a Wilson plot analysis which relates the water mass velocity to the inside heat transfer coefficient.

Particle Nusselt Number

$$Nu_p = \frac{h_o D_p}{k_g}, \text{ dimensionless}$$

where,

Nu_p = particle Nusselt number, dimensionless

h_o = outside heat transfer coefficient,
BTU/hr-ft²-°F

D_p = particle diameter, ft

k_g = air thermal conductivity, BTU-ft/hr-ft²-°F

Minimum Fluidization Velocity

The values of minimum fluidization velocity used in the correlation of the data for coiled spiral tubes is calculated from the Leva correlation (4).

$$G_{mf} = \frac{688 D_p^{1.82} \rho_g (\rho_s - \rho_g)^{.94}}{U_g^{.88}}, \text{ lb}_m/\text{hr-ft}^2$$

where,

G_{mf} = minimum fluidization velocity, $lb_m/hr-ft^2$

D_p = particle diameter, in

ρ_g = density of fluidizing gas, lb_m/ft^3

ρ_s = density of bed particles, lb_m/ft^3

U_g = viscosity of fluidizing gas, $lb_m/ft-hr$

CONCLUSIONS

From the investigation of the helical finned tube system the following conclusions were drawn:

1. Heat transfer coefficients increased with increasing fluidizing air mass velocity. For some conditions a maximum was reached.
2. Heat transfer coefficients increased with decreasing particle size. The increase was generally greater between the small and medium sized particles than between the medium and large size particles.
3. Heat transfer coefficients increased with increased fin spacing. The coefficient was very sensitive to fin spacings greater than 8 particle diameters and less sensitive to fin spacings less than 8 particle diameters.
4. Heat transfer coefficients increased with decreased fin height.
5. Heat transfer coefficients increased with an increase in the tube bundles center-to-center spacing, until the tubes became located close to the column wall, then the coefficient decreased.

6. Performance of the tubes increased with increasing fluidizing air mass velocity, increased fin spacing, and decreased fin height.
7. The best performance was with a tube of intermediate fin spacing and intermediate fin height.
8. An average gain of up to 135 percent was obtained with the helical finned tubes compared to a bare tube.

From the investigation of the coiled spiral tube system the following conclusions were drawn:

1. Heat transfer coefficients increased with decreasing particle size for all geometries.
2. It was generally observed that the heat transfer coefficient increased with increasing fluidizing air mass velocity, with a maximum value observed in some cases.
3. The heat transfer coefficient increased with decreasing groove depth and increasing flute pitch.
4. The best performance was from a tube with intermediate number of flutes and intermediate groove depth and flute pitch.

5. Most of the data fell within plus or minus 15 percent of a correlation relating experimental parameters to the heat transfer coefficient.

APPENDICES

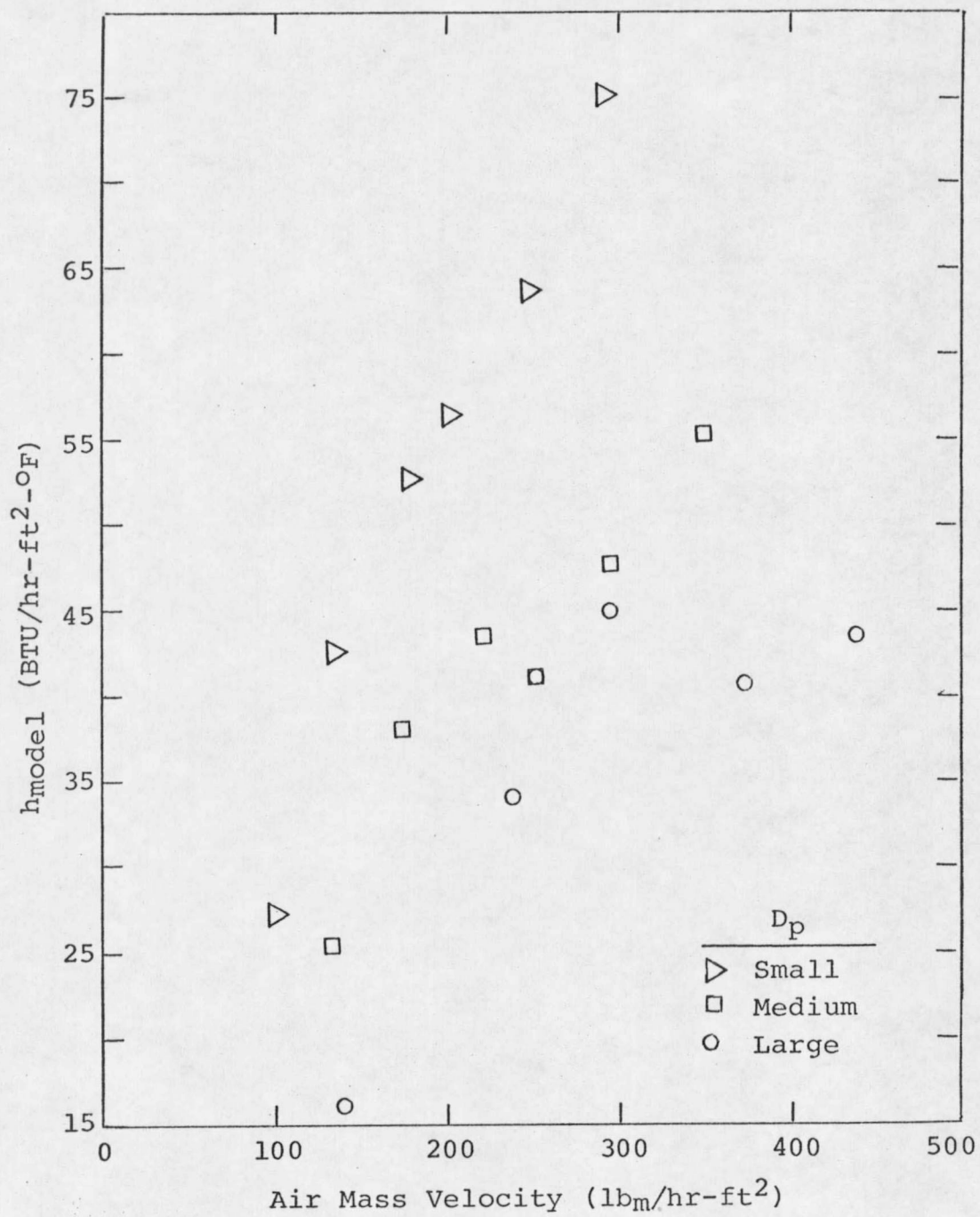


Figure 35. h_{model} VERSUS G VERSUS
PARTICLE DIAMETER (TUBE No. 1)

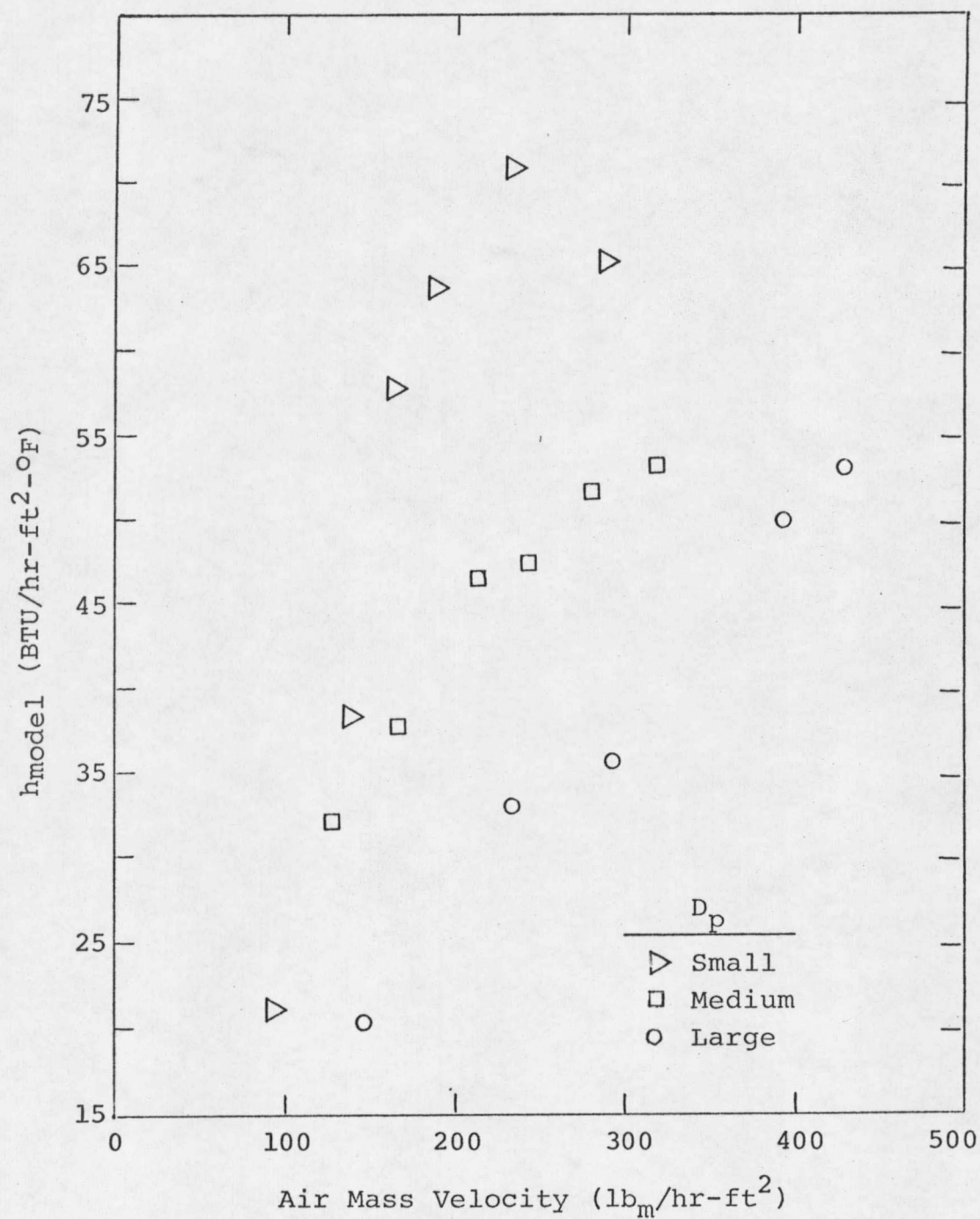


Figure 36. h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 2)

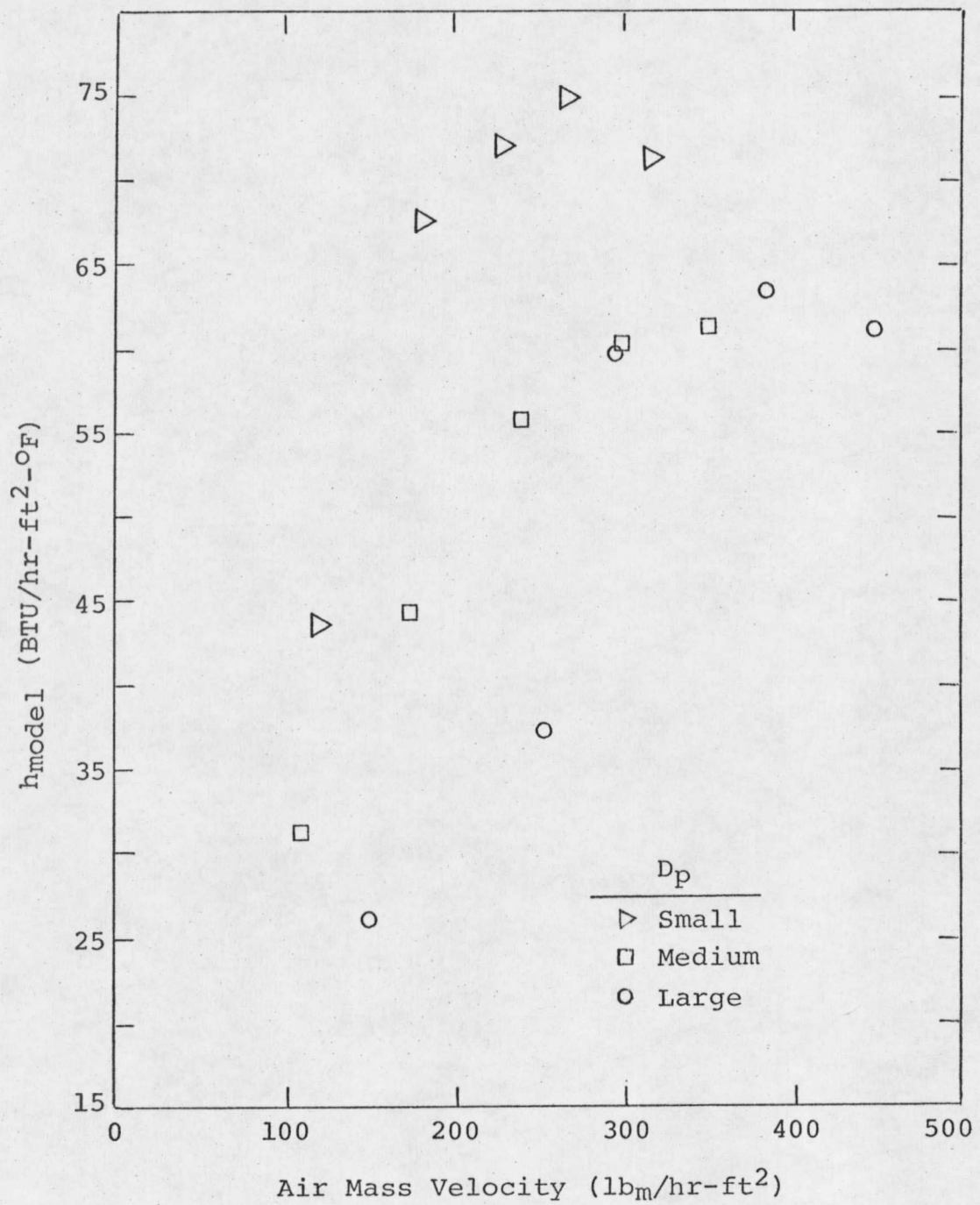


Figure 37. h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 3)

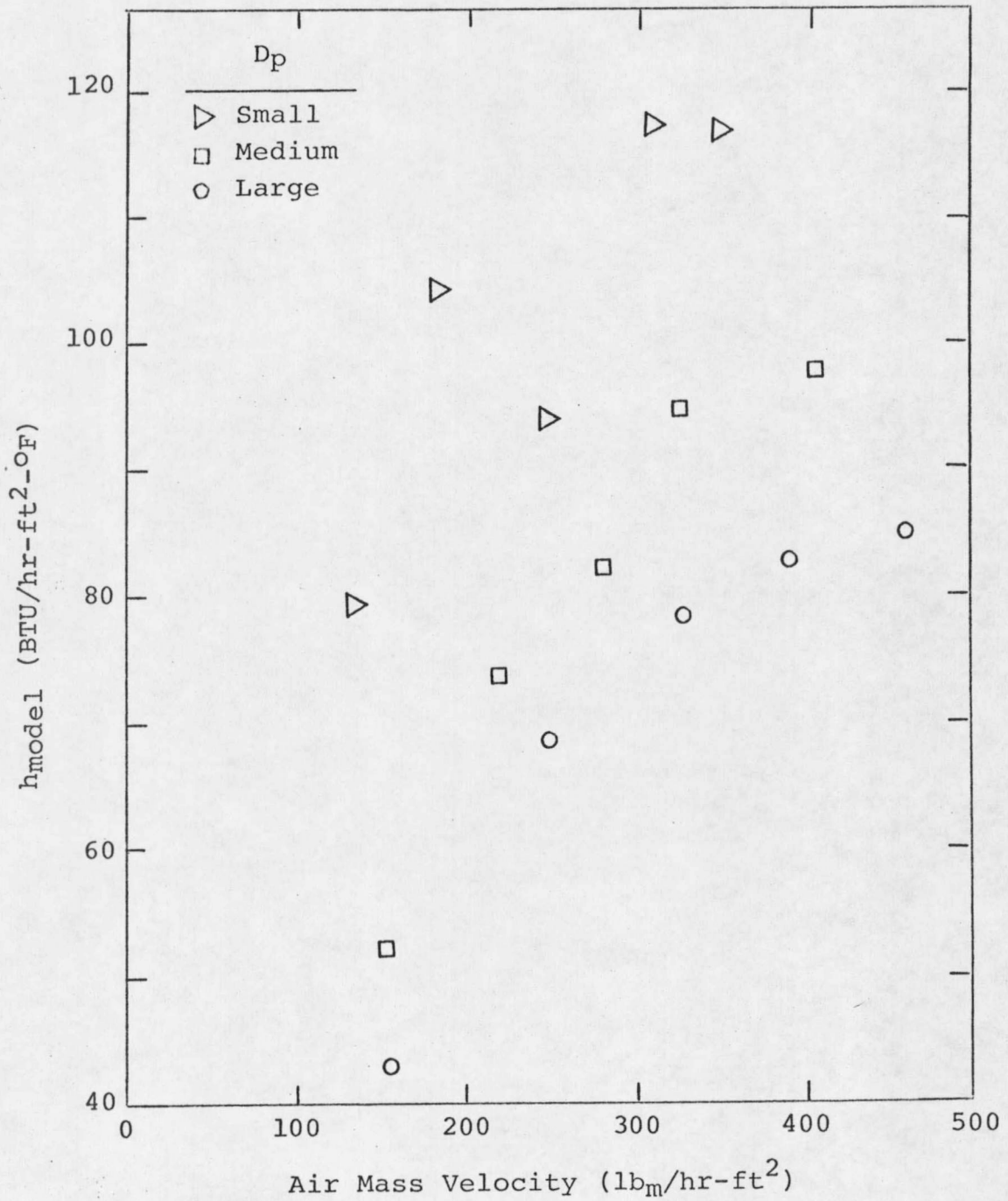


Figure 38. h_{model} VERSUS G VERSUS
PARTICLE DIAMETER (TUBE No. 4)

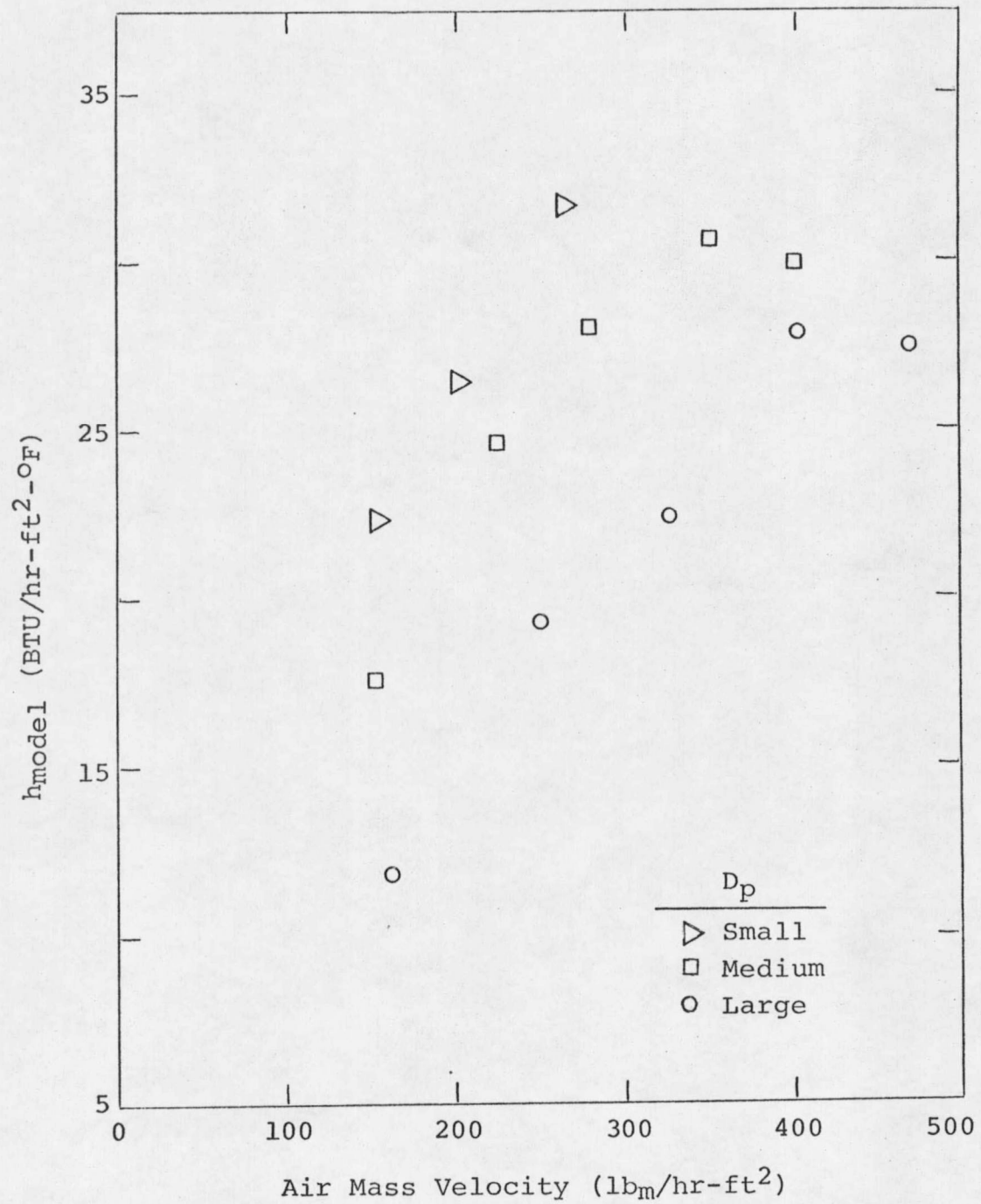


Figure 39. h_{model} VERSUS G VERSUS
PARTICLE DIAMETER (TUBE No. 5)

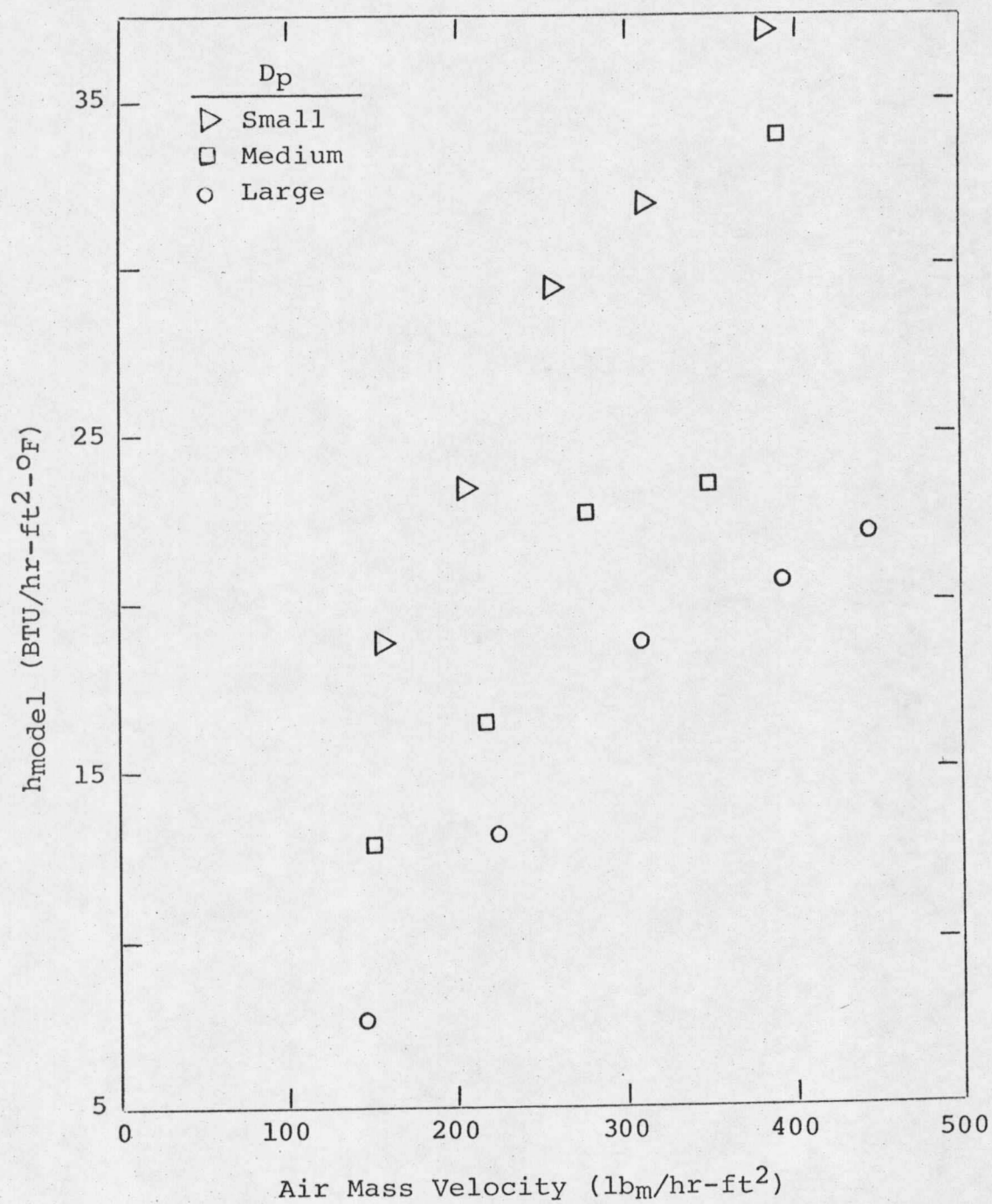


Figure 40. h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 6)

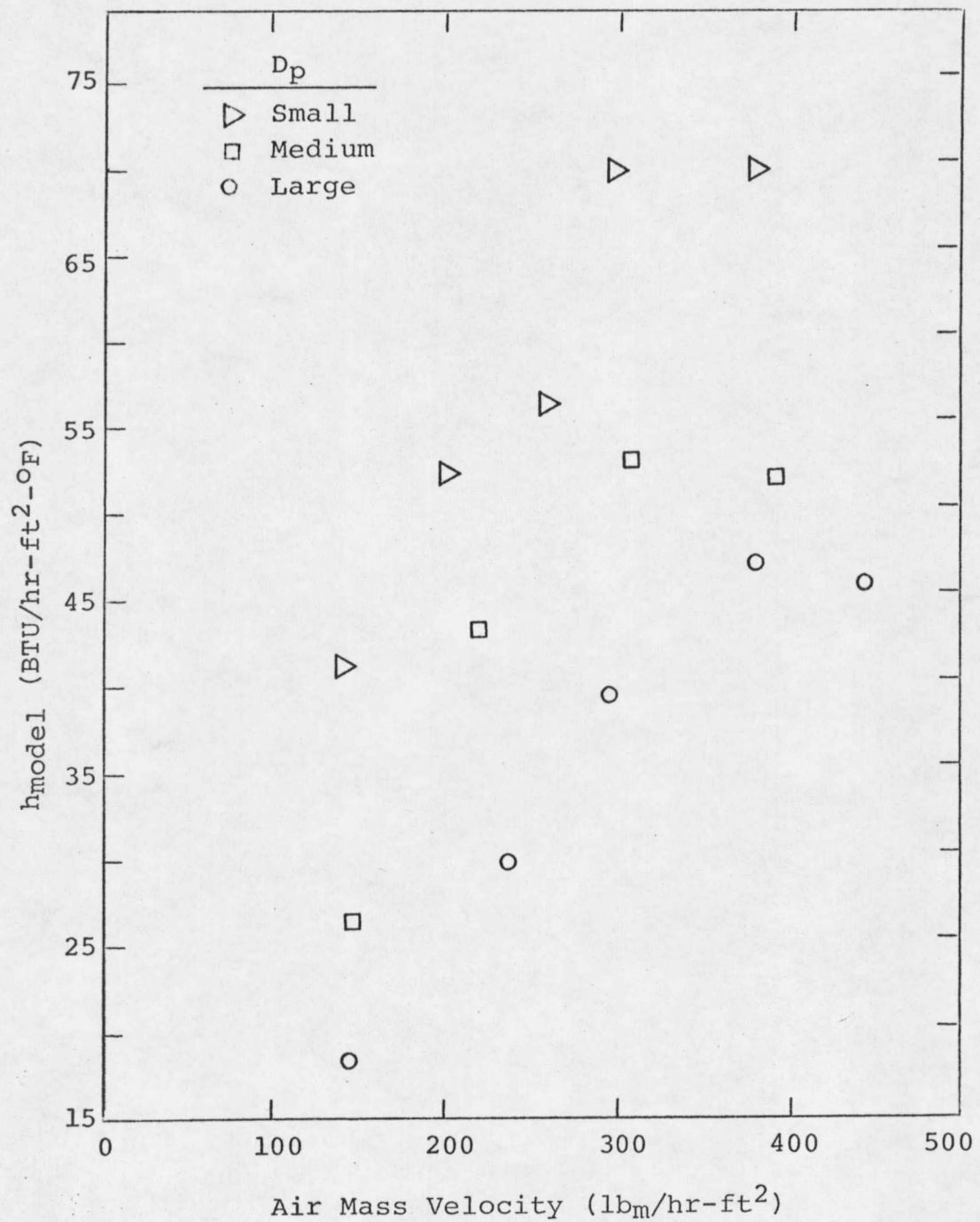


Figure 41. h_{model} VERSUS G VERSUS PARTICLE DIAMETER (TUBE No. 8)

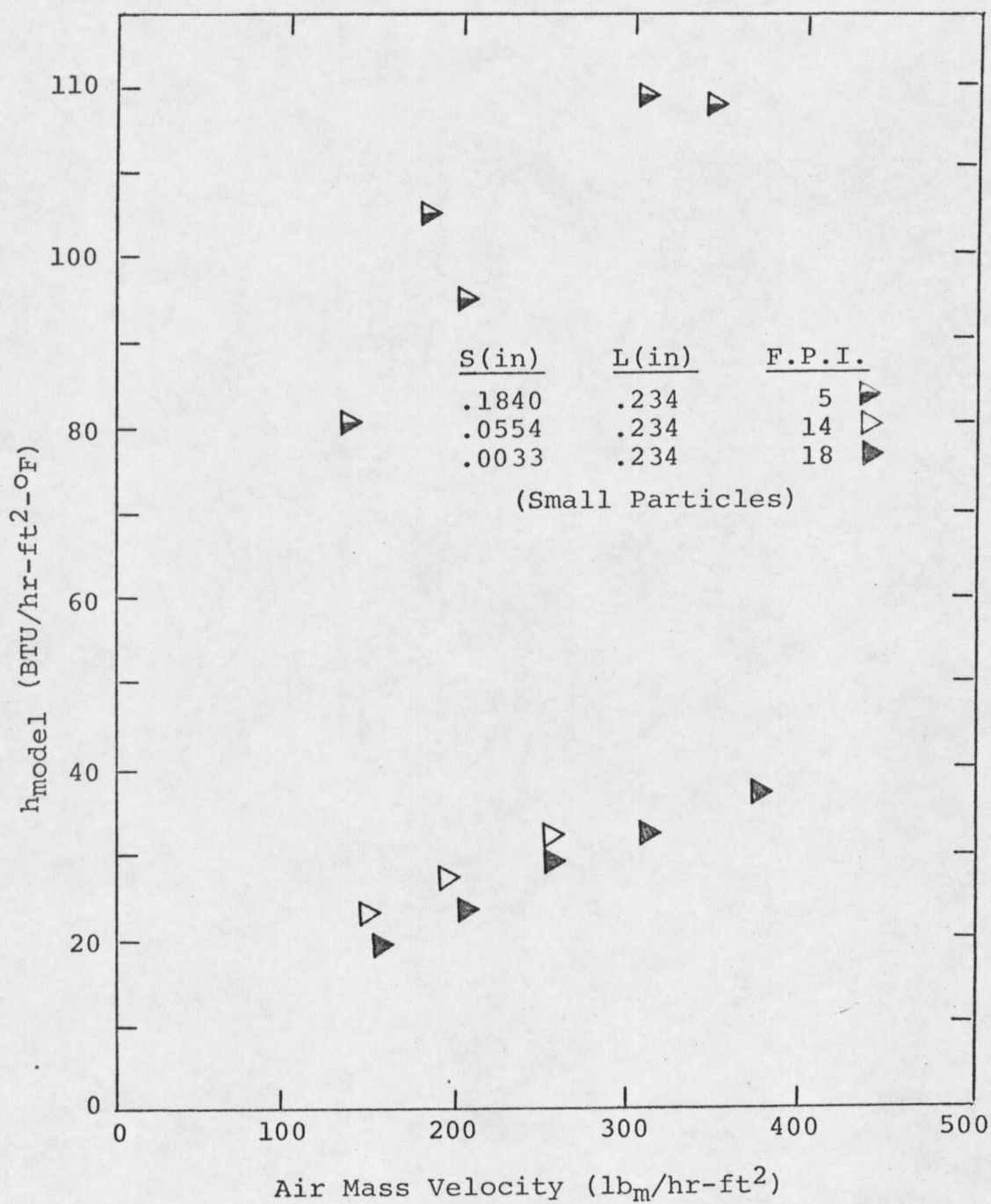


Figure 42. h_{model} VERSUS G VERSUS FIN SPACING (S)

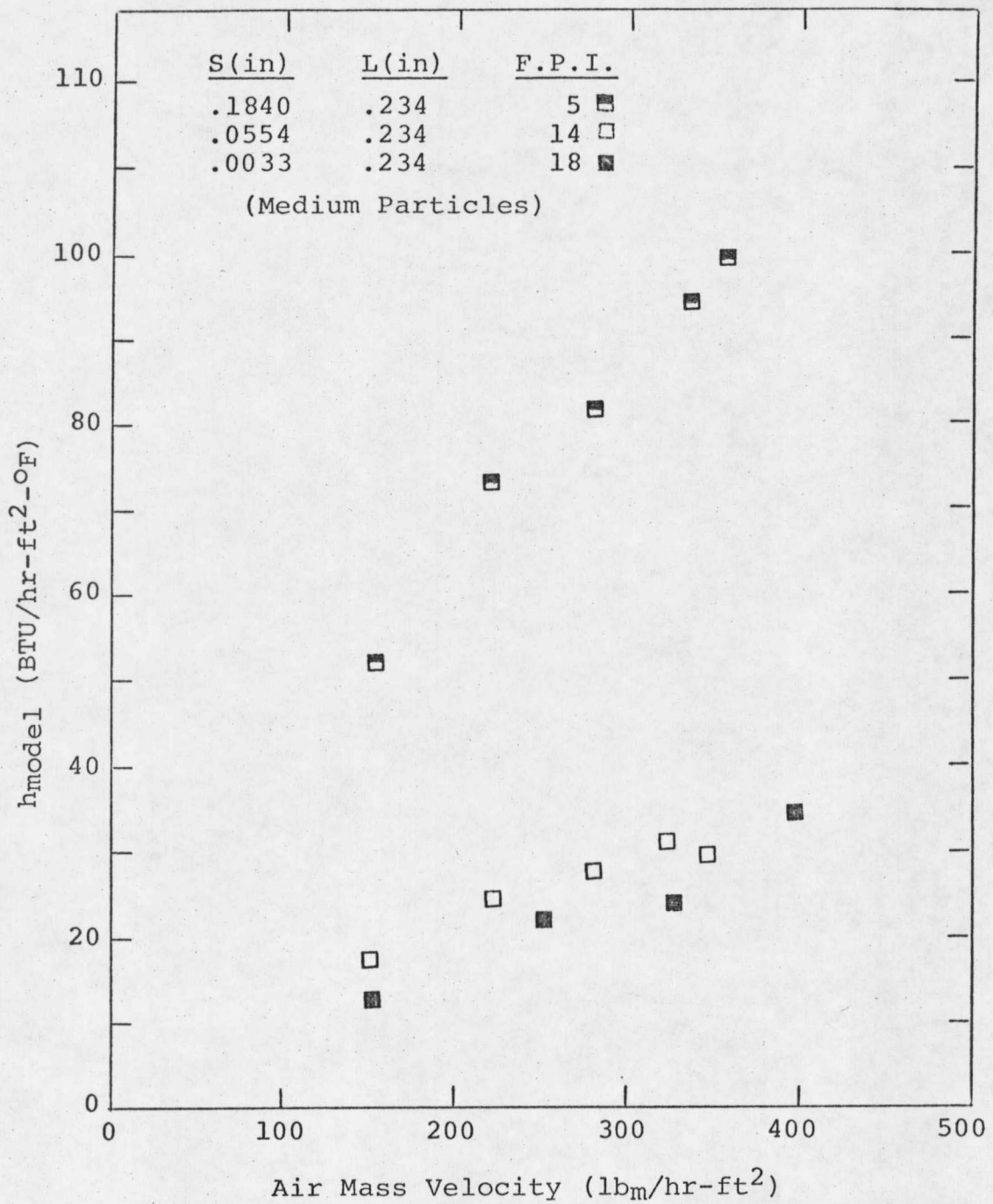


Figure 43. h_{model} VERSUS G VERSUS FIN SPACING (S)

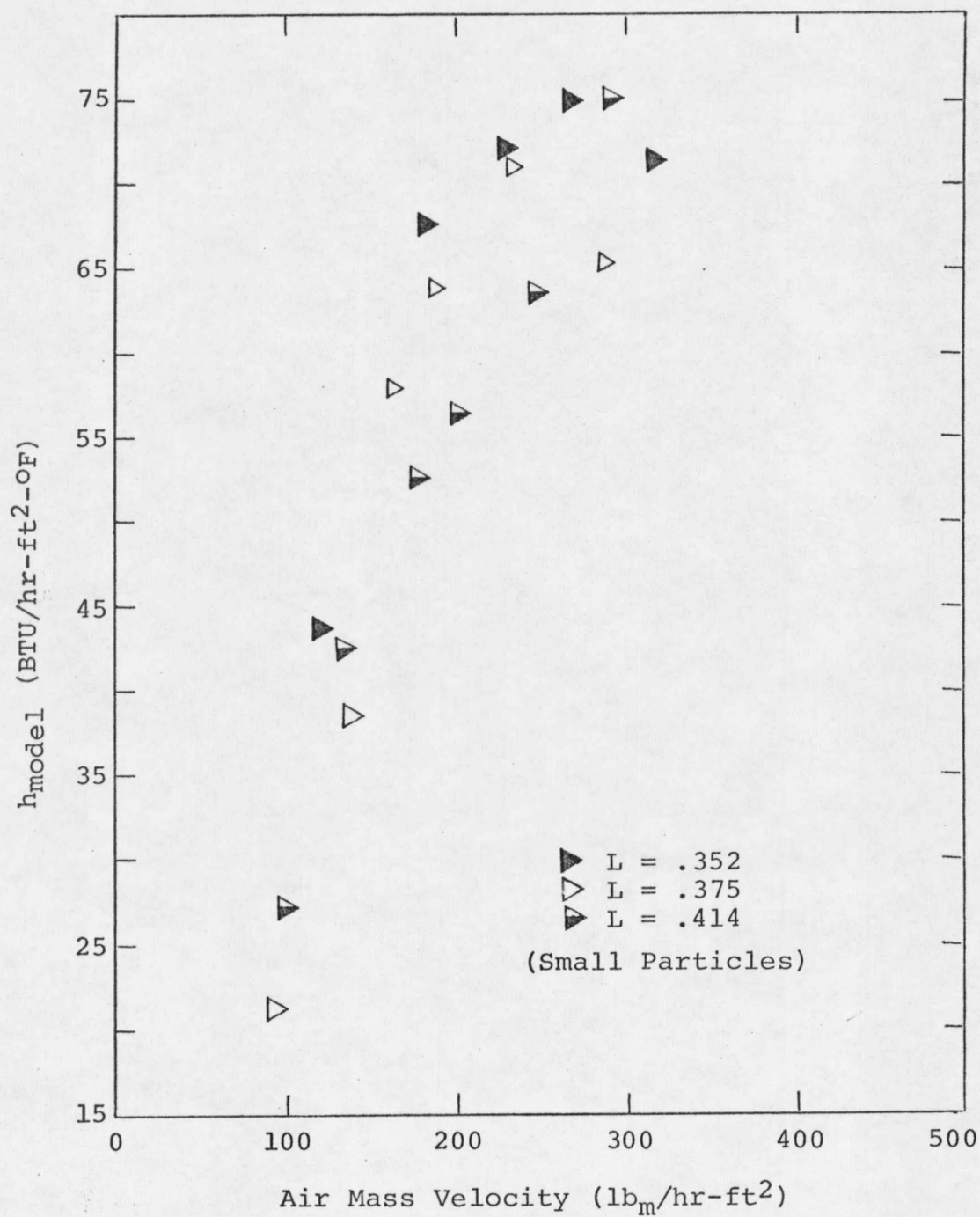


Figure 44. h_{model} VERSUS G VERSUS
FIN HEIGHT (L)

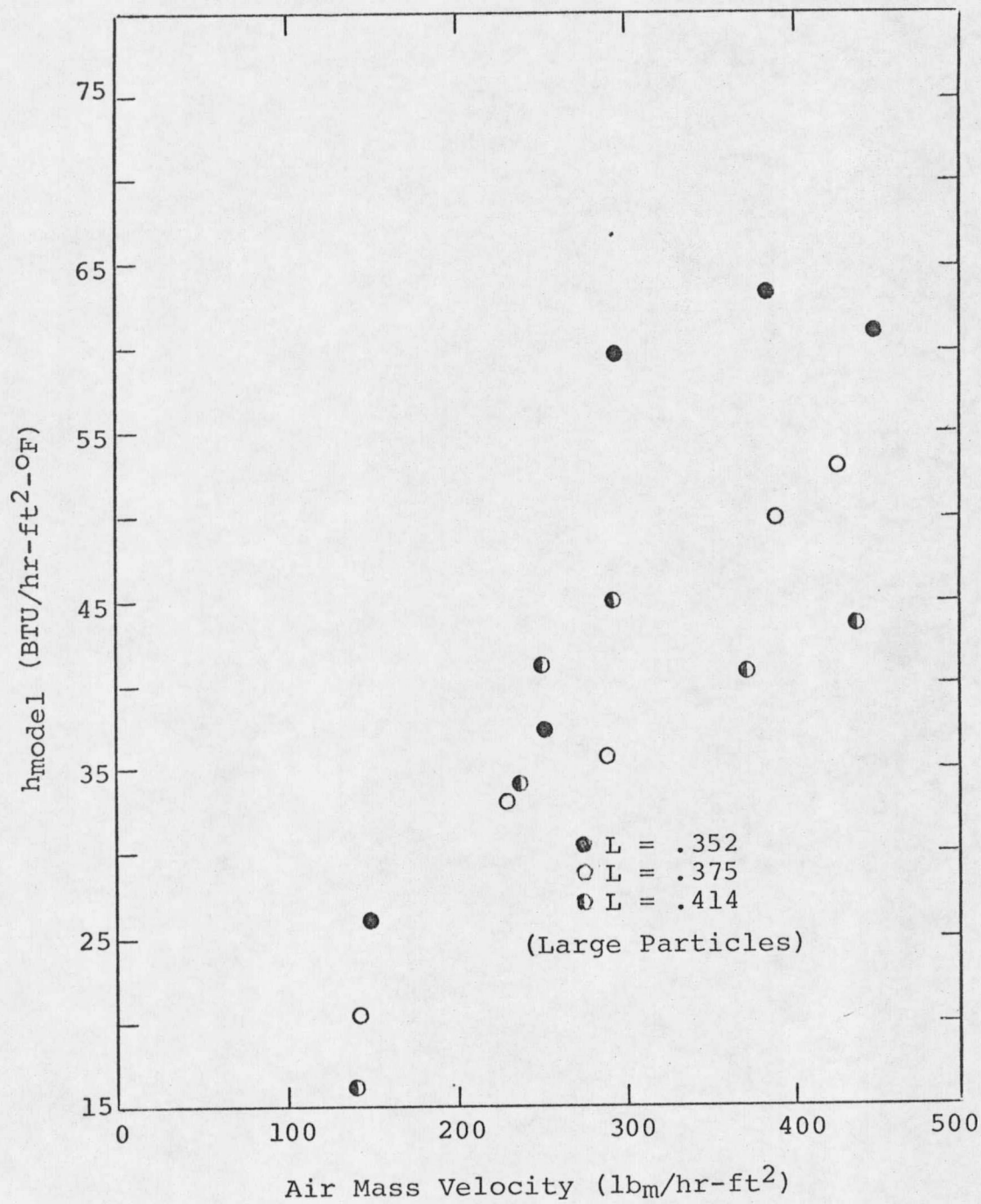


Figure 45. h_{model} VERSUS G VERSUS
FIN HEIGHT (L)

NOMENCLATURE

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
A_b	bare tube surface area	ft^2
A_c	cross sectional area of column	ft^2
A_i	inside surface area, coiled spiral tube	ft^2
A_f	fin surface area, helical finned tube	ft^2
A_T	total tube surface area	ft^2
A_o	outside surface area, coiled spiral tube	ft^2
C_1, C_2	constants in development of temperature distribution in helical fin	dimensionless
C_o	orifice coefficient	dimensionless
C_p	heat capacity at constant pressure	$\text{BTU/lb-}^\circ\text{F}$
$C_{\text{PH}_2\text{O}}$	heat capacity of water coiled spiral tube	$\text{BTU/hr-}^\circ\text{F}$
C_{ps}	heat capacity of solid particles	$\text{BTU/hr-}^\circ\text{F}$
C_v	heat capacity at constant volume	$\text{BTU/hr-}^\circ\text{F}$
D_L	logarithmic mean diameter	inches
D_o	outside diameter, coiled spiral tube	inches
D_p	particle diameter	inches
G	air mass velocity	lbm/hr-ft^2

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
G_{mf}	minimum fluidization velocity	lbm/hr-ft^2
g_c	gravational constant	$\text{ft-lb}_m/\text{hr}^2\text{-lb}_f$
h	average heat transfer coefficient	$\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$
h_{exp}	experimental heat transfer coefficient, helical finned tube	$\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$
h_{model}	heat transfer coefficient accounting for temperature distribution in fin	$\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$
h_i	inside heat transfer coefficient, coiled spiral tube	$\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$
h_o	outside heat transfer coefficient, coiled spiral tube	$\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$
I_o, I_1, K_o, K_1	modified Bessel functions	dimensionless
k	thermal conductivity of copper tube	$\text{BTU-ft/hr-ft}^2\text{-}^\circ\text{F}$
k_g	thermal conductivity of fluidizing medium	$\text{BTU-ft/hr-ft}^2\text{-}^\circ\text{F}$
K_r	ratio of C_p/C_v	dimensionless
L	fin height, helical finned tubes or distance between flutes, coiled spiral tube	inches
N	number of flutes, coiled spiral tube	dimensionless
N_t	number of fins, helical finned tube	dimensionless

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
Nu_p	particle Nusselt No.	dimensionless
P	flute pitch, coiled spiral tube	inches
P_1	upstream pressure	lb_f/ft^2
P_2	downstream pressure	lb_f/ft^2
ΔP	pressure drop across the bed	lb_f/ft^2
q, Q	rate of heat transfer	BTU/hr
Re_p	particle Reynolds No.	dimensionless
r	radius of tube at fin tip	inches
r_o	radius of tube at fin base	inches
S	fin spacing, helical finned tube	inches
S_c	cross sectional area of orifice	ft^2
ΔT	temperature driving force	$^{\circ}F$
T_b	bulk bed temperature	$^{\circ}F$
T_i	inlet water temperature, coiled spiral tube	$^{\circ}F$
T_o	outlet water temperature, coiled spiral tube	$^{\circ}F$
T_w	heat transfer surface temperature	$^{\circ}F$

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
U_o	overall heat transfer coefficient, coiled spiral tube	BTU/hr-ft ² -°F
W_{H_2O}	water mass velocity	lb _m /hr
Y	expansion factor	dimensionless
β	ratio of orifice diameter to inside pipe diameter	dimensionless
$(1-\epsilon)$	particle fraction	dimensionless
θ	average contact time	hr
μ_g	viscosity of fluidizing medium	lbm/ft-hr
ρ_g	density of fluidizing medium	lbm/ft ³
ρ_s	density of particles	lbm/ft ³
ρ_l	density of upstream air	lbm/ft ³
δ	groove depth, coiled spiral tube	inches

BIBLIOGRAPHY

BIBLIOGRAPHY

1. Othmer, D.F., Fluidization, Reinhold Publishing Corp., New York, (1956)
2. Botterill, J.S.M., Fluid-Bed Heat Transfer, Academic Press, New York, (1975)
3. Kunii, D. and O. Levenspiel, Fluidization Engineering, John Wiley and Sons, Inc., New York, (1977)
4. Leva, M., Fluidization, McGraw-Hill Book Co., New York, (1959)
5. Wen, C.Y., A.I.Ch.E. Symposium Series, V74: n 176, (1978)
6. Simons, H.E., Fluid Bed Incineration of Petroleum Refinery Wastes, U.S. Govt. Print. Off., Washington, (1971)
7. Tewksbury, T.L., Fluidized Bed Incineration of Selected Carbonaceous Industrial Wastes, U.S. Govt. Print. Off., Washington, (1972)
8. Davidson, J.F. and D. Harrison, Fluidization, Academic Press, New York, (1971)
9. Levenspiel, O. and J.S. Walton, Chem. Engr. Progr. Symp., V50: n9, (1954)

10. Mickley, H.S. and D.F. Fairbanks, A.I.Ch.E. Journal, V1, 374 (1955)
11. Ziegler, E.N., L.B. Koppel, and W.T. Brazelton, Ind. Eng. Chem. Fundamentals, V3 : n4, 324 (1964)
12. Genetti, W.E. and J.G. Knudsen, I. Chem. Engr. Progr. Symp. Series, V30, 147 (1968)
13. Genetti, W.E., R.A. Schmall, and E.S. Grimmer, A.I. Ch.E. Symposium Series, V67 : n116, (1971)
14. Vreedenberg, H.A., Chem Eng. Science, V9, 52-60 (1958)
15. Petrie, J.C., W.A. Freeby, and J.A. Buckham. Chem. Engr. Prog., V64: n7, July (1968)
16. Chen, J.C. and J.G. Withers, "An Experimental Study of Heat Transfer from Plain and Finned Tubes in Fluidized Beds", A.I.Ch.E paper no. 34, 15th National Heat Transfer Conference, San Francisco (1975)
17. Bartel, W.J. and W.E. Genetti, A.I.Ch.E. Symposium Series, V69 : n128 (1973)
18. Priebe, S.J., Ph.D. Thesis, Montana State University (1975)
19. Kratovil, M.T., M.S. Thesis, Montana State University (1976)
20. Vanderhoff, D.W., M.S. Thesis, Montana State University (1978)

21. Everly, D.W., M.S. Thesis, Montana State University, (1978)
22. Everly, D.W. and W.E. Genetti, "Heat Transfer from Spiral Tubing in an Air Fluidized Bed", paper submitted for presentation at 71st meeting of the AIChE, Miami Beach, (1978)
23. Kreith, F., Principles of Heat Transfer, Intext Press Inc., New York, 3rd edition, 1976

260-1047

AK

MONTANA STATE UNIVERSITY LIBRARIES



3 1762 10022738 6

1598
193
C42