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EVALUATION OF FLUID BED HEAT EXCHANGER
OPTIMIZATION PARAMETERS

Final Technical Report

MASTER

March 1980

Work Performed Under Contract No. AC03-78ET11343

Aerojet Energy Conversion Company
Sacramento, California

U. S. DEPARTMENT OF ENERGY

Division of Industrial Energy Conservation

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FINAL TECHNICAL REPORT

DEPARTMENT OF ENERGY
Contract No. DE-AC22-78ET11343

EVALUATION OF FLUID BED HEAT
EXCHANGER OPTIMIZATION PARAMETERS

REPORT NO. 8100-FB-0001

March 1980

AEROJET ENERGY CONVERSION COMPANY
SACRAMENTO, CALIFORNIA

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C.L.

FOREWARD

An experimental program was undertaken at the Aerojet Energy Conversion Company (AECC) under Contract DE-AC22-78ET11343* with the Department of Energy (DOE), to determine the impact of fluid bed particles on heat exchanger design.

Department of Energy Program Management and Technical Direction was by Michael Perlsweig, Dr. William Thielbahr, and James Hickerson.

Aerojet Energy Conversion Company Program Management was by Mark Rudnicki and Howard Williams, and the Project Engineer was Rosemary Hernandez. Charles Gracey designed and supervised the construction of the test facility and conducted the experimentation with assistance from David Heimlich.

Significant contributions were also made by Frank Miller, Roger Anderson and Clifford Mah.

This report comprises the Final Report under the contract.

* Previous contract numbers assigned were

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EXECUTIVE SUMMARY

The gas-side heat transfer coefficients experienced in fluidized beds are known to be significantly larger than in conventional heat exchangers, often by an order of magnitude. Thus, the fluid bed heat exchanger has the potential for improved performance, as compared to their conventional counterparts, in specific applications.

Uncertainty in the relationship of specific bed material properties to gas-side heat transfer has inhibited the search for optimum bed materials and has led to over-conservative assumptions in the design of fluid bed heat exchangers.

An experimental program was carried out by the Aerojet Energy Conversion Company, to isolate the effects of particle density (ρ), thermal conductivity (k), and heat capacitance (C_p) upon fluid bed heat transfer. Test planning, test system design, fabrication, and test set up were paralleled by a literature search on previous experimental work; the latter indicated a degree of disagreement on the effects of bed material properties. A total of thirty-one tests were then run with eighteen different bed material loads on twelve material types; particle size variations were tested on several material types.

Data reduction and analysis was done by direct comparison of heat transfer data on pairs of materials with nearly equal values of two of the properties under study and with a large difference in the third property. Also, multiple regression analysis was applied to the total data package. These analyses yielded the following relationship:

$$h \propto d_e^{-0.5} \rho^{0.2} C_p^{0.2} k^0$$

where h = gas-side heat transfer coefficient

d_e = equivalent particle diameter

ρ = specific gravity

C_p = specific heat (heat capacitance)

k = thermal conductivity

Similarities and differences with previous findings were discussed.

Conceptual design of a fluidized bed evaporator unit was completed for a diesel exhaust heat recovery system. The evaporator heat transfer surface area was substantially reduced while the physical dimensions of the unit increased. Despite the overall increase in unit size, the overall cost was reduced.

A study of relative economics associated with bed material selection was conducted. For the fluidized bed evaporator, it was found that zircon sand was the best choice among materials tested in this program. More importantly, this study indicated that the selection of bed material substantially influences the overall system costs.

The optimized fluid bed heat exchanger has an estimated cost 19% below a fin augmented tubular heat exchanger; 31% below a commercial design fluid bed heat exchanger; and 50% below a conventional plain tube heat exchanger. The comparisons being made for a 9.6×10^6 BTU/hr waste heat boiler.

The fluidized bed approach potentially has other advantages such as resistance to fouling which could result in the selection of a fluid bed system over a conventional system in some applications.

It is recommended that a study be conducted to develop a systematic selection of bed materials for fluidized bed heat exchanger applications, based upon findings of the study reported herein.

I. INTRODUCTION

In the design of heat exchangers with gas phase heat transfer, the gas-to-surface heat transfer has been the limiting design factor. This low gas-to-surface heat transfer coefficient yields a low overall heat transfer conductance, which typically results in a gas-to-liquid heat exchanger which is approximately four times larger than a liquid-to-liquid heat exchanger for the same heat duty.

Fluidized beds offer very high heat transfer rate potentials.

One of the keys to the optimum utilization of fluidized bed heat exchangers is the selection of a bed material. Both the heat exchanger heat transfer area and the pressure loss are significantly affected by this selection.

The program included a survey of the current literature to establish bed material property effects on fluidized bed heat transfer, selection of materials for testing, fabrication of a fluidized bed facility, performance of fluidized bed heat transfer tests, determination of bed material property effects of particle density, thermal conductivity and heat capacity, and application of the experimental results to the conceptual design of a waste heat recovery heat exchanger.

II. BACKGROUND

A. POTENTIAL FOR FLUIDIZED BED HEAT EXCHANGER

In the design of heat exchangers with gas phase heat transfer, the gas-to-surface heat transfer has been the limiting design factor. Typically, the forced-convection, fluid-to-surface heat transfer coefficients for atmospheric air in conventional tubular heat exchangers are in the range of 5 to 30 Btu/hr-ft²-°F. This low gas-to-surface heat transfer coefficient yields a low overall heat conductance. As a result, the typical gas-to-liquid heat exchanger is approximately four times larger than the typical liquid-to-liquid heat exchanger for the same heat duty. The two conventional approaches for improving the gas-side heat transfer rate are (1) increase the gas velocity through the heat exchanger or, (2) use extended surfaces (fins) on the tubes. But, in the area of waste heat recovery, heat transfer enhancement by these two methods is limited by the available system pressure drop which restricts the extent to which velocity can be increased, and by excessive fouling, where fins cannot be used. An alternative solution is to use fluidized beds for heat transfer enhancement. Fluidized beds have proven, high heat transfer rate potential, with heat transfer coefficients up to 50 times greater than for conventional heat exchangers having been observed with dense phase beds of small (50µm) particles. However, previous attempts to define the phenomenon of heat transfer in fluidized beds have been derived from tests run with many variables changing at the same time. The results have been inconsistent, with heat transfer coefficients that may differ by an order of magnitude for a given system. This forces the fluidized bed heat exchanger designer to fall back to a conservative position which, in turn, prevent the fluid bed system from achieving its full economic/performance potential.

B. HEAT TRANSFER IN FLUIDIZED BEDS

The current fluidizing technique originated from the development work of the Standard Oil Development Company, the M.W. Kellogg Company and Standard Oil of Indiana to fulfill the demand for high

octane aviation fuel during the Second World War. This application of fluidization greatly improved the catalytic cracking process by eliminating the transfer problems associated with catalyst regeneration. Subsequently, fluidized beds have been used in a variety of applications such as the treatment of liquid nuclear wastes, the heat treatment of metals, the desalination of water, and in coal gasification. In most of the applications mentioned, heat must be transferred to or from the bed.

The heat transfer rate in fluidized beds depends on many factors, since the bed represents a complex interaction of gas and solid. Variables to be considered are:

1. Properties of gas: density, viscosity, specific heat, thermal conductivity
2. Properties of solid: particle diameter, true density and bulk density, sphericity, specific heat, thermal conductivity, size distribution
3. Conditions at minimum fluidization: superficial velocity, voidage
4. Flow conditions: superficial velocity, void fraction
5. Geometric properties: bed diameter, diameter of immersed objects, static bed height, length of heat exchange surface, vertical and horizontal spacing in immersed tube bundles

Equally important is the effect of operating temperature and its contribution to the radiative component of the heat transfer rate as well as the distributor plate design. Furthermore, many of these factors are interrelated. Because of this complexity, the data presented by various researchers tend to be widely divergent.

1. General Inadequacy of Correlations

The high rate of heat transfer at heating and cooling surfaces is probably the single most important property of gas-

fluidized beds. Measurements of coefficients of the heat transfer between a fluidized bed and the heat exchange surface have been carried out by many investigators and extensive data are available in the literature. The discrepancy among the results by different workers was illustrated by Wen and Leva (1956) as well as by Van Hurden (1952); they showed that the dense-phase heat transfer derived from correlations by different investigators can vary by sixty-fold. Ainshtein and Gelperin (1966) compared 15 heat transfer correlations for internal heating surfaces and found data scatter of nearly two orders of magnitude. In closely controlled laboratory experiments under similar conditions for heat transfer to vertical tubes, Wender and Cooper showed coefficients of approximately $120 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, whereas Genetti, et. al, showed coefficients of approximately $60 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, and Chen and Withers showed values in between, Figure II-1.

Even though discrepancies exist, progress has been achieved in qualitative estimation and computation of fluidized bed-to-surface heat transfer. Zabrodsky, et. al, 1976, reported that the non-monotonic dependences of the heat transfer coefficient on gas velocity and particle size, as well as the kind of gas and its pressure, are known. They also attributed the enormous gaps between different correlations to the viewers' unjustifiable extrapolation of these equations far beyond the limits within which they were established. Zabrodsky also believed the discrepancies between different correlations may be reduced by modifying some of them through the exclusion of data taken under special conditions. Calculations for these special cases may then be performed by introducing correction factors or terms into a "basic equation" which holds for basic conditions. Again, these are of limited application because they require knowledge of parameters which are not generally available for practical situations. These parameters would include information such as the porosity of the bed, the effective conductivity of a packet, the bed density, the effective stirring factor, and the thermal resistivity at the wall to mention a few. Botterill (1975) also doubted the validity of these empirical correlations because they cannot make adequate allowance for the dynamic behavior of a gas-fluidized system. He contended that, although corre-

Figure 11-7 Variability in Heat Transfer Data

lations give some indication of the importance of factors within the experimental range, it is rarely clear to what extent the results have been influenced by the scale of the equipment, specifically with regard to fluidization behavior.

A sample of calculated heat transfer coefficients based on correlations on fluidized bed heat transfer in the literature is shown below for four different fluid bed materials of varying density, heat capacity, and conductivity:

Correlation	Heat Transfer Coefficient, Btu/hr-ft ² °F			
	Sand/Glass	Graphite	Copper	Tungsten
Levenspiel & Walton 1954	82	82	82	82
Zabrodsky 1976	55	55	72	84
Mickley & Fairbanks 1955	70	100	280	165

As a result of the weak theoretical and empirical base for fluidized bed heat transfer phenomena, many new fluidized bed applications cannot be properly assessed with regard to their economic feasibility, because widely divergent predictive methods project correspondingly divergent values for design parameters.

2. The Effects of Bed Material Variables on Fluidized Bed Heat Transfer

The variation in the effect of bed material variables on fluidized bed heat transfer reported in earlier laboratory tests arise from differences in the detail and scale of the experimental. For example, although it is agreed that heat transfer coefficients increase as the average size of the bed material is reduced, there is a wide variation in the reported data on its effect. The variations in the effect of particle thermal conductivity, particle specific heat and particle density as reported in the literature are reviewed in detail in this section.

a. The Effect of Particle Thermal Conductivity

Mickley and Fairbanks (1955) showed that the effective thermal conductivity of a bed has a significant effect on fluidized bed heat transfer. Baskakov (1966,1968) then proceeded to define an effective bed thermal conductivity as a function of the particle and fluidizing gas thermal conductivity. The function is presented in Figure II-2. The result of the Baskakov correlation would suggest that the particle conductivity would have significant effect on the fluid bed heat transfer rate. Contrary to this, Ziegler (1964) through his experimental work with material of equal size and density and with varying thermal properties, determined that solid thermal conductivity had a negligible effect on the heat transfer coefficient. Later in a book edited by Davidson and Harrison (1971), Gelperin and Ainshtein reported that the thermal conductivity of the solid material has practically no influence on the heat transfer coefficient, although they acknowledge the increase in the heat transfer coefficient with the rise in the effective bed conductivity. In 1972, Chung, Fan and Hwang developed a theoretical model for heat transfer as a modification of Ziegler's 1964 work. Their correlation used particle thermal and physical properties in a complex expression in a thermal diffusivity term. Their model indicated that the heat transfer coefficient can be independent of the particle thermal conductivity. But they also indicated that, under certain conditions, the particle thermal conductivity could affect the heat transfer coefficient. Zabrodsky, et al., in 1976, presented a critical review of recent main works on bed-to-surface heat transfer. They agreed with Botterill (1970) that the thermal conductivity of a solid can be of significance for heat transfer only at the conditions when the thermal resistance of the gas and that of the solid are commensurate. However, later in 1978, Zabrodsky stated that the effect of material thermal conductivity did not show up for fine or large particles during his research on the convective component of the heat transfer coefficient. The one observation that emerges from review of the literature on the effects of particle thermal conductivity on heat transfer is that there is no unanimity or clear position on the issue.

Figure II-2 Approximate Effective Thermal Conductivity of a Fluidized Bed

b. The Effect of Particle Heat Capacity

Again, most workers have found some proportionality between the heat transfer coefficient and particle heat capacity, but the dependence varies widely. Dow and Jakob in 1957 found heat transfer to be proportional to particle heat capacity to the 0.25 power, but Wender and Cooper in 1958 reported the proportionality power factor to be 0.8. This influence of the particle heat capacity on heat transfer was shown especially well by Ziegler, et al., in their tests of particles of equal size and density but with varying values of heat capacity; their results indicate a correlation to the particle specific heat to the 0.40 power. Particle heat capacity was ignored in recent papers by Andeen and Glicksman (1976, 1978) in their work with shallow fluidized beds. In 1977, Selzer and Thomson attempted to correlate fluidized bed heat transfer to horizontal and vertical tubes with modifications made to the packet theory. Their best fit agreed well with Botterill's single particle model, as given by Gutfinger and Abraf (1974), which does not include any particle thermophysical properties. But Thomson did obtain good results with one modification to the packet theory that included the particle heat capacity term to the 0.5 power.

c. The Effect of Particle Density

It has been generally accepted that the primary influences of particle density are upon the fluidizing velocity and the volumetric heat capacity ($\rho_s C_{p_s}$). But Zabrodsky (1978), in his work to determine the convective component of the heat transfer coefficient, stated that the $\rho_s C_{p_s}$ term disappears for particles larger than approximately one millimeter. Gelperin indicated that the density effect on the heat transfer coefficient is in the order of $m \approx 0.2$ in the equation $h \propto \rho^m$. Others have indicated that it might be as high as 1.0. Ainshtein and Gelperin (1966) compared a number of correlations to determine the maximum heat transfer coefficient, $h_{\max}$, using an arbitrary model. They selected four correlations from a field of eleven that determined $h_{\max}$ to within ± 10 -15%. Two correlations use only particle diameter and density through the use of Nusselt and Archimedes numbers. The Baerg correlation shows the heat transfer co-

efficient varying directly with the particle density. The Zabrodsky correlation for the convective component of 'h' uses particle density to the 0.2 power. Lese and Kermode (1972) studied the effect of the number of tubes, their size, configuration and bed material type (glass beads, alumina, and silicon carbide) on fluidized bed heat transfer to horizontal tubes. Their comparison of heat transfer was made against Vreedenberg's correlation with an approximate agreement using glass beads. The Vreedenberg correlation indicates the effect of density on the heat transfer coefficient is to the -0.3 power. In 1977, Yasutomi and Yokota proposed a two layer emulsion phase with unsteady conduction as the heat transfer mechanism. They reported that the dependence of the heat transfer coefficient on the solid particle parameters could not be expressed simply. But they did believe that contact times seemed to be dependent on particle density as well as particle diameter and geometry. In 1976, Andeen (with Glicksman) studied shallow fluidized bed heat transfer to horizontal tubes. Later in 1978, Andeen with Glicksman and Bowman, used flattened horizontal tubes in their studies. In both cases, their best fit correlation was a modification of the Vreedenberg equation which relates particle density to the heat transfer coefficient. In the first study the correlated power was -0.326 and in the second study, -0.37. The discrepancies in the influence of density upon heat transfer coefficient, as indicated in the varied results of the above investigators, point up the need for tests in which bed variables are properly restrained.

d. The Effect of Particle Parameters
in Generalized Functions

Various researchers have presented their findings in generalized functions. But, as reported by Zabrodsky, et al, in 1976, many of these heat transfer "criterial equations" for defining the maximum Nusselt numbers, Nu_{max} , are in fact simple empirical formulas. In such "criterial equations", even the use of the Nusselt number (Nu) and Reynolds (Re) number has no reasonable physical grounds. These equations are not derived from any closed system of differential equations, to say nothing of the fact that many values entering the

Nusselt and Reynolds numbers, such as the thermal conductivity of the gas, the viscosity of the gas and the density of the gas, may not have been varied in the course of the experiments establishing the correlation.

Still many researchers continue to use these generalized functions to present their data. In 1973, Baskakov, et al., presented a modification to the packet theory for heat transfer between a fluidized bed and an immersed object to allow for an effective gap resistance. They present a best fit correlation for the convective component of the heat transfer coefficient with the Nusselt number as a function of both the Archimedes (Ar) number and Prandtl number. More recently, Pillai, in 1976, from his studies of shallow (50 mm) fluidized bed heat transfer to a sphere, correlated 400 data points to within $\pm 10\%$ with a generalized form of the Nusselt number as a function of the Archimedes number. More surprising is the omission of the radiative component of the heat transfer coefficient, even though his work varied from 1000°F to 2000°F. In 1977, Ali and Broughton also used $Nu = f(Ar)$ for the presentation of their data of shallow fluidized bed heat transfer to a movable horizontal tube with bed depths of 50 mm.

Other works showed particle parameters other than particle diameter have no effect on heat transfer. These include Gelperin, et al (1966), Priebe and Genetti (1977) with their work on finned tubes, Korolev and Syromgatnikov (1975) and Grewal and Saxena (1979) in their work on surface roughness. Chen (1976) presented a modified packet model using an effective bed thermal conductivity, density and heat capacity to correlate his studies of base and finned tubes in shallow (200 mm) beds.

All these and many more examples indicate that the physical processes in fluidized bed heat transfer are not well understood. Conductivity experimental tests with many variables changing at the same time will only serve to cloud the controlling parameters and preclude their isolation.

3. Shallow Fluidized Beds

Fluidized bed processes traditionally have involved the use of deep beds of fine solids; commonly, bed depths have exceeded three feet, and particles were smaller than 200 microns. However, heat exchanger and boiler applications dictate that fluidized beds be as shallow as possible to minimize the required fan power. Some work has been done using shallow beds. For example, there were the Courtauld gas cleaning process (1963) which uses a 5-stage unit 7mm deep carbon beds of 100 to 3000 micron particles, the fluidized coal combustor with bed depths of one foot and particles below 1000 microns (1970), and the gas combustors as developed by Elliott (1970) for use in metallurgical furnaces, central-heating systems, etc, with bed depths less than 2 inches. A comparison of shallow beds to deep beds prepared by Ali and Broughton (1977) can be seen in Table II-1.

TABLE II-1
Shallow-Beds Compared With Deep-Beds

	Deep-Beds	Shallow-Beds
Bed pressure drop	>40 in. H ₂ O	<20 in. H ₂ O
Plate pressure drop	>30 in. H ₂ O	< 2 in. H ₂ O
Carryover/Transport disengaging height	>3.3 ft	3.3 ft usually adequate
Bed Expansion(%)	20-40	50-250
Heat transfer coefficient to tubes immersed	30-140 Btu/hr-ft ² °F	30-140 Btu/hr-ft ² °F
Multistage	Seldom practical	Fairly easy
Bubble sizes	Order of 3.3 ft	Less than 4 in.
Amenity to scale-up	Difficult due to slugs in small beds	Usually direct
Slugging	Common	Never
Entry effects	Usually small	Often important
Exit effects	Carryover T.D. Ht.	Several effects
Temperature uniformity	Usually uniform	Gradients often occur

A shallow bed differs in appearance from a deep bed in two important respects:

(1) Bubbles flow in groups of small bubbles and individual bubble studies have little apparent relevance. The large, slugging surface eruptions characteristic of deep bed processes seldom occur, and a lean-phase cloud is formed above the bed. Elliott (1970) reported this lean cloud above the shallow bed has important effects on both radiation and convection processes.

(2) Thermal gradients can readily be established in shallow beds, and entry effects are important.

Little has been reported about heat transfer in shallow beds as compared to deep beds. Zabrodsky and Yuditskii (1966) found that shallow beds exhibit a temperature gradient close to the distributor plate. They reported a significant increase in heat transfer when a heat exchanger tube is very close to the plate. Atkinson (1974) found a definite increase as well. Pillai (1976) mentions a reduction in heat transfer caused by two main factors that are likely to influence behavior at the plate.

(1) Jets, or bubble formation, could cause an alteration in the mechanism of heat transfer. The tube interacts with bubble formation and bubble-blanketing of the surface can occur.

(2) A thermal entry zone exists which is dependent upon the bubbling and/or jetting. When the tube is in this region, the mean temperature driving force differs from that based upon the uniform bed temperature.

These effects have not been studied in detail to date, and the causes can only be inferred from the limited experimental information in the literature. In heat exchangers, heat transfer in the distributor region is apparently higher; in combustors, where cold gas is introduced and then heated, it is lower in the entry region than the mean value used in calculating heat transfer coefficients. The experiments of Zabrodsky and Yuditskii and those of Atkinson used heat exchangers and found a rise in heat transfer coefficients in the entry region. Pillai, using a combustor, found a decrease at the distributor.

Also, when finned horizontal tubes are inserted into shallow beds, the heat flux is greater than that in a deep bed at otherwise similar conditions. This may be explained by the violent coalescence of bubbles that takes place through the majority of the shallow bed depth. This violent coalescence may suggest that solids motion within a shallow bed could be significantly greater than in a deep bed where bubble growth is largely completed within a very small portion of the bed depth. Greater solids motion should result in shorter residence times of particles at the immersed surface, and this, in turn, could explain the high values of heat transfer coefficient which have been measured.

III. TEST PROGRAM

The heat transfer rate in a fluidized bed depends on many parameters, including the properties of the gas, the properties of the bed material, the flow conditions, and the geometric configuration of the unit. The best prospect for system improvement is in the proper selection of bed material. Both the heat exchanger heat transfer area and the pressure loss are significantly affected by material choice. Material properties such as density, heat capacity and thermal conductivity have been frequently cited for their impact on heat transfer and pressure drop. Existing data on the influence of material properties is not adequate for the selection of the optimum bed material for a specified heat recovery service.

A. TEST PROGRAM OBJECTIVES

The objective of the test program was to resolve the question of the influence of material thermophysical properties upon fluidized bed heat transfer. Specifically, the effects of thermal conductivity, density, and heat capacity were to be experimentally determined.

B. EXPERIMENTAL APPROACH

The approach to meeting the above objective was to isolate the influences of the three material properties of interest by holding

all other variables constant, or as nearly so as practicable. Then, the effect of each material property was to be determined by direct-pair testing; e.g., two materials with (ideally) identical densities and thermal conductivities, but with a large difference in heat capacity, would be tested and evaluated in terms of their maximum heat transfer coefficients. The maximum heat transfer coefficients were to be determined by systematic variation of superficial velocity. The two $h_{\max}$ data points would thus be free of influences from other variables, and a reliable measure of the effect of the isolated material property on fluidized bed heat transfer would be determined.

From a practical standpoint, absolute isolation of material variables is not possible. For example, bed materials with a wide range of heat capacities, but with nearly identical density and thermal conductivity, cannot be obtained. The following considerations address the approach to testing with non-ideal materials:

1. Mean particle diameter: Recognizing the impracticability of obtaining bed materials with identical mean particle diameters, a series of tests was planned with one material type (three distinct sizes of spherical grade silica sand) to isolate the effect of particle size such that corrections could be applied to pair comparison test results. Previous findings on size effects would also be used.

2. Particle size distribution: Effects of size distribution were not treated in more than a qualitative way. All materials tested consisted of a range of particle sizes distributed about a mean diameter. Very small particles were removed by the classifying action of the fluidized bed.

3. Particle shape: It was found to be very difficult to control shape to obtain the ideal spherical particle. Care was taken to procure materials which were generally rounded and regular as opposed to being in the form of slivers or flakes.

4. Thermophysical property pairing: It was found diffi-

cult to identify materials giving precise matching of two of the properties while obtaining a significant variation in the third. This was particularly so for thermal conductivity. The approach here was to isolate the effect of thermal conductivity, and then apply corrections, much as in the case of particle diameter compensations noted above.

From the above considerations, it was clear that some uncertainties would be introduced by non-ideal conditions. However, the variations were felt to be second-order in importance, and the planned approach to be a significant step in control of experimental conditions for isolating effects of material properties on fluidized bed heat transfer.

The experimental setup and test conditions were maintained fixed to the greatest extent possible. Superficial velocities were varied to obtain the maximum heat transfer coefficient. The remaining test parameters were controlled to specified values. Thus, all testing was conducted in the same test vessel without any changes to the distributor plate, the in-bed array of heat exchange tubing or the position of the temperature sensing tubes in the array. The bed loading was held constant to 10 inches of static bed height. The bed temperature was maintained at 400°F, except in the case of temperature sensitive materials, such as solder and plastic. The coolant flow was held at 500 lb./hr. The positions of the sensing instrumentation for temperature, pressure and flow were never changed, and the coolant temperature sensors were calibrated prior to each test to within 0.2°F.

C. FLUID BED TEST MATERIALS

To fulfill the basic program objectives of isolating the effect of bed material properties on fluidized bed heat transfer, a careful selection of materials had to be made to ensure that the variable under test had a large enough range in order to isolate the effect of that variable. The ideal situation would be to test several materials with the same physical thermophysical properties except the variable under test. For example, in the selection of materials for the determination of the effect of thermal conductivity on fluidized bed heat

transfer, several materials would be sought with exactly the same size and shape, identical densities and heat capacities, and with several orders of magnitude difference in thermal conductivity. Perfect matching is impractical, if not impossible. Therefore, compromises had to be made in the material selection process, with consideration given to first order effects to increase the probability of isolating material property influences on heat transfer. Materials selected for the test program are listed in Table III-1.

It is noted here, for the benefit of future investigators, that the process of procuring materials for direct comparison tests of this type is difficult, time-consuming, and frustrating. Some of the major hurdles in bed material acquisition are (1) locating vendors of materials prepared in small granule form, (2) interesting vendors in supplying small quantities, (3) actually getting what was specified, even when having received and analyzed a preshipment sample which met the specification, (4) having to arrange for separate screening operations, (5) losses in shipment, and (6) excessive costs. The practical problems of acquiring good bed test materials should not be underestimated.

1. Selection Considerations

a. Particle Size

A uniform particle size was deemed necessary to minimize the dominant effect of particle size on fluidized bed heat transfer. The size of 150 microns (μM) was selected as a reference. As discussed in section B. above, specific bed materials were obtained and tested in various sizes to derive corrections for property pair comparison evaluations in cases where identical mean particle diameters could not be provided. Three sizes of sand and glass ballotini and two sizes of alumina and expanded alumina were procured and tested to isolate the influence of particle size upon heat transfer coefficient. See Table III-1 for details.

b. Particle Thermal Conductivity

For the evaluation of thermal conductivity

TABLE III-1. BED MATERIAL SELECTION

TEST	MATERIAL	NOMINAL PARTICLE SIZE (μM)	SPECIFIC GRAVITY	SPECIFIC HEAT (Btu/lb.-°F)	THERMAL CONDUCTIVITY (Btu/hr-ft-°F)
Thermal Conductivity	Sand	150	2.0 - 2.6	0.195	1
	Graphite	150	2.25	0.202	70-94
	Salt	150	2.16	0.221	2.20
Density	Flint Glass	150	3.89	0.11	0.3-0.5
	316 Stainless Steel	150	7.9	0.117	8
	Glass Ballotini	150	2.45-2.55	0.24	1
	Hollow Glass Beads	150	0.42	0.24	0.5
	Alumina	150	3.89	0.249	11.69
	Expanded Alumina	150	1.0	0.249	12
Specific Heat	Nickel	150	8.9	0.11	53
	Solder	150	8.9	0.045	27
	Flint Glass	150	3.89	0.11	0.3-0.5
Low Density	Alumina	150	3.89	0.249	11.69
	PPS Plastic	150	1.34-1.35	0.3-0.35	24
	Hollow Glass Beads	150	0.42	0.24	0.5
Particle Size	Expanded Alumina	150	1.0	0.249	12
	Sand	150, 300, 500	2.0 - 2.6	0.195	1
	Glass Ballotini	150, 300, 500	2.45- 2.55	0.195	1
	Expanded Alumina	150, 500	1.0	0.249	12
	Alumina	150, 300	3.89	0.249	11.69

Note: Material properties were obtained from available material handbooks or manufacturer's specifications.

effects, materials had to be found with nearly identical density and heat capacity but with a wide variation in thermal conductivity. Silica sand was chosen as a reference; it had been tested by many researchers and the corresponding data was readily available. Although there are no exact limits for the silica content of these sands, they commonly contain more than 95% SiO_2 and some of them more than 99%. This high purity would contribute to the confidence of material property comparisons. It was available in a nominal spherical form in a variety of sizes. Metallics could not be used as high conductivity materials, since they have correspondingly high densities. A search through tables of conductivities for non-metallics revealed only a few materials with high conductivities and densities similar to sand. Graphite was identified as a good thermal conductivity pair for sand; density and specific heat are well-matched, and a large variation in heat conductivity is obtained (see Table III-1). Salt was also selected, but does not provide the large variance in conductivity obtained through comparison of sand with graphite.

c. Particle Density

The effect of particle density on fluidized bed heat transfer required identification of materials with identical thermal conductivity and specific heat and with a wide variation in density. It was found impractical to control thermal conductivity, as mentioned previously; thus, reliance was placed on correcting for conductivity variations on the basis of results of the sand/graphite conductivity pair tests described above.

Density pairing is difficult due to the inherent nature of most materials. Generally, materials adhere to the Kopp-Neumann law, which states that the specific heat of a solid compound at room temperature is equal to the weighted sum of the specific heats of the atoms forming the compound. In addition, the rule of Dulong and Petit that gives the specific heat at constant volume for one atomic weight of any solid element to be 6.4, directly relates specific heat to density; i.e., as density increases, specific heat decreases. Yet, in order to isolate the effect of density, material pairs need to have nearly identical specific heats and differing den-

sities, preferably by a factor of two or more. A material mixture had to be found that did not follow the Kopp-Neumann law. One such mixture is solder-a tin/lead alloy. The tin/lead alloy produces a high density material, and the low specific heat value of the lead keeps the specific heat value of the alloy suppressed below that of many common alloys. But solder could not be used in the density tests since matching specific heat values would result in the selection of precious or rare earth elements such as platinum, lanthanum or tellurium. Solder was then deferred to specific heat test series.

Flint glass and 316 stainless steel were identified as having nearly equal specific heat values and a factor-of-two range in density. Zircon sand was identified as an alternate if the flint glass was unavailable. Two other density pairs were made by matching glass ballotini with hollow glass beads, and by matching alumina with expanded alumina. A potential disadvantage of the expanded alumina is its porosity, an added variable.

d. Particle Specific Heat

The third fluid bed particle thermophysical property to be isolated for testing was specific heat. In this selection, identical densities were sought with a wide variation in the specific heat. Here again, the Kopp-Neumann law relating density and specific heat restricted the number of materials selected. A few materials were identified while researching density test pairs.

Two pairings for specific heat were identified. These were (1) nickel and solder, and (2) flint glass and alumina. Densities are well matched with specific heat varying by roughly a factor of two. The difficulty in fixing thermal conductivity, noted above, is also a factor in selecting materials for specific heat evaluation.

e. Low Density Materials

In the process of studying bed materials, interest was generated in identifying material which might combine good heat transfer characteristics with low pressure drop. Thus, the scope of work was expanded to include selection of low density mater-

ials for limited testing.

In the selection of a low density material, it was first presumed that the material would need to be lighter than sand; i.e., specific gravity less than 2.0. Second, the material should not significantly degrade at temperatures up to 400°F. Three groups of materials were considered, consisting of plastics, hollow spherical materials, and porous materials.

Plastics were considered first, being solid materials that would not introduce the porosity variable to the test program (porosity may or may not significantly affect the heat transfer characteristics of the solid, but its exclusion as a variable is considered preferable).

Hollow spheres, a second choice, come in metallics, plastics, ceramics and glass in a variety of sizes from several inches to less than 100 microns. The hollow spheres would provide the required low density and, since the captured air within the spherical walls would not interchange with the fluidizing gas, the thermophysical properties would not be significantly affected by the enclosed air pocket. Porous materials constitute the third choice for low density testing. Highly porous materials were not considered desirable due to their ability to communicate with the fluidizing gas. This exchange may influence the effect of the thermophysical properties of the material on fluidized bed heat transfer. To reduce the porosity influence, a fused material would be favored over highly porous materials, such as catalysts. This would typically reduce the porosity from over 60% to less than 20%.

Based upon the above considerations, PPS plastic and expanded alumina were chosen for low pressure drop exploration. The expanded alumina also served as one of the materials in a density evaluation pair.

2. Test Material Description

The test materials used in this study are listed in Table III-2 along with the pertinent information in regard to their

TABLE III - 2, TEST MATERIALS

MATERIAL	PARTICLE SIZE MMPD (μM)*	SPECIFIC GRAVITY	SPECIFIC (BTU/lb. $^{\circ}\text{F}$)	THERMAL CONDUCTIVITY (BTU/hr-ft- $^{\circ}\text{F}$)	SOURCE	COMMENTS
1. Silica Sand	176	2.0-2.6	0.195	1	Ottawa Silica Co.	Spherical Grade
2. Silica Sand	279	2.0-2.6	0.195	1	"	"
3. Silica Sand	445	2.0-2.6	0.195	1	"	"
4. Glass Ballotini	174	2.45-2.55	0.24	1	Potters Industries Co.	Spherical, Solid Glass Beads
5. Glass Ballotini	333	2.45-2.55	0.24	1	"	"
6. Glass Ballotini	507	2.45-2.55	0.24	1	"	"
7. Expanded Alumina	202	1.0	0.249	12	Carborundum Co.	Hollow, Porous, Alumina Beads
8. Expanded Alumina	506	1.0	0.249	12	"	"
9. Alumina	168	3.89	0.249	11.69	Nat'l Abrasives Co.	
10. Alumina	283	3.89	0.249	11.69	Norton Co.	
11. Graphite	157	2.25	0.202	70-94	Versar, Inc.	Solid, Rounded Particles
12. Salt	148	2.16	0.221	2.20	Morton Salt	
13. 316 Stainless Steel	105	7.9	0.117	8	Hceganaes Corp	Poor Particle Shape
14. Zircon Sand	149	4.46	0.142	1.11	Industrial Foundry Supply Co.	Spherical to Ovaloid Particles
15. PPS Plastic	205	1.34-1.35	0.3-0.35	24	LNP Corp.	Polyphenylene Sulfide Thermoplastic Plastic
16. Nickel	95	8.9	0.11	53	Belmont Metals, Inc.	99.9% Nickel
17. Solder	98	8.9	0.045	27	Powder Alloy Corp.	60% Sn, 40% Pb
18. Flint Glass	63	3.89	0.11	0.3-0.5	Ferro Corp.	56.2% PbO, 40.5% SiO ₂

* MMPD = Mean mass particle diameter; the particle size at which 50% is either larger or smaller by weight.

NOTE: Material properties were obtained from available material handbooks or manufacturer's specifications.

physical and thermophysical properties. Specific particle size data is given for test materials as received. Also, the sources of the materials are listed. The following descriptions provide additional information on the actual test materials.

a. Silica Sand

All three silica sands used in the testing were obtained from the Ottawa Silica Company. These sands were of a spherical grade. Photomicrographs of these sands are presented in Figures III-1, III-2, III-3. Although there are no exact limits for the silica content of these sands, they commonly contain more than 95% SiO_2 . This high purity contributed to the confidence of the material property comparisons. Three sizes of sand material were obtained and tested, as indicated below. The lot with 176 μm MMPD represented the nominal test size. Samples with 229 and 445 μm MMPD were used in assessing effects of particle size.

<u>Grade</u>	<u>U.S.S. Screen Mesh</u>	<u>Size Range (μm)</u>	<u>Surface Area (cm^2/gm)</u>	<u>MMPD (μm)</u>
F-80	-70 + 140	105-210	176	176
Coarse Special Blend	-45 + 80	177-350	102	279
Sawing	-35 + 50	297-500	56	445

b. Ballotini

Ballotini are solid glass impact beads which are generally used for metal finishing processes such as deburring, cleaning and peening. The glass beads used in the testing were obtained from Potters Industries, Inc. These glass beads were spherical and white or crystal in color, free from surface films. These beads were tested microscopically for broken or angular pieces with a maximum of 3 percent allowable. The soda-lime glass in these beads had a minimum silica content of 67%. Three sizes, listed in Table III-2, were used as a check on particle size effects on heat transfer. The following list provides additional materials data. Photomicrographs of each size are shown in Figures III-4, III-5, III-6.

Figure III-1. Silica Sand
MMPD = 176 microns (33x)

Figure III-2. Silica Sand
MMPD = 279 microns (33x)

Figure III-3. Silica Sand
MMPD = 445 microns (33x)

Figure III-4. Glass Ballotini, MMPD = 174 Microns (33 X)

Figure III-5. Glass Ballotini, MMPD = 333 Microns (33 X)

Figure III-6. Glass Ballotini, MMPD = 507 Microns (33 X)

Ballotini Specifications

<u>Stock Size</u>	<u>U.S.S. Screen Sizes</u>	<u>Size Range (μm)</u>	<u>Minimum % True Spheres</u>	<u>MMPD (μm)</u>
AD	-70 + 140	210-105	80%	174
B	-40 + 60	420-250	70%	333
C	-30 + 40	595-420	60%	507

c. Expanded Alumina

Expanded Alumina (aluminum oxide, Al_2O_3) particles used in the testing were fused, white alumina bubbles. They were an electric arc furnace product using Bayer alumina as the basic raw material. The chemistry of the product is basically the same as white aluminum oxide, with 99% Al_2O_3 . The bubbles were formed from molten alumina to produce a hollow spherical product of low bulk density of 35 lb/cu ft. These alumina bubbles were obtained from the Carborundum Company, Electro Minerals Division. They are commonly used in the production of insulating refractories. This material is available in U.S. Standard Screen Mesh 20 and finer. For this test program, the alumina was classified into two size fractions to produce batches with MMPD=202 μm and MMPD=506 μm . These materials are shown in the photomicrographs of Figures III-7 and III-8.

d. Alumina

Alumina, as a fine powder, is generally white. Aluminum oxide crystals are normally hexagonal, and are minute in size with grain sizes running from 150 microns to less than 50 microns. The larger particles are made up of many crystals. The aluminas used in this study were at least 99.5% pure. Bed loads were obtained in two particle size ranges. The smaller size, with an MMPD=168 μm , was obtained from the National Abrasives Company. It was a first-quality, exolith grade. The larger size (MMPD=283 μm), obtained from Norton, was white Alundum 38 after being crushed from larger crystals. Both aluminas are shown in the photomicrographs of Figures III-9 and III-10.

e. Graphite

The material used in testing was a synthetic

Figure III-7. Expanded Alumina, MMPD = 202 Microns (33 X)

Figure III-8. Expanded Alumina, MMPD = 505 Microns (33 X)

Figure III-9. Alumina, MMPD = 168 Microns (33 X)

Figure III-10. Alumina, MMPD = 283 Microns (33 X)

graphite, obtained from Versar, Inc. These particles were solid and rounded in a size range of 140 to 178 μ m. This material was free-flowing and did not feel 'greasy' like molded graphite. A photomicrograph of this material is given in Figure III-11. MMPD was determined to be 157 μ m.

f. Salt

The salt used in this study was Purex Extra Fine Prepared Salt obtained from Morton Salt. This salt was a food grade sodium chloride with an average NaCl content of 98.24%. The salt was produced by milling vacuum granulated salt to a particle size of less than U.S. Standard Screen Mesh 60(250 microns) and then treated with 1.5% tricalcium phosphate to prevent caking and ensure uniform free-flow when exposed to humidity exceeding 75% rH, the critical humidity of salt. The pour (loose) bulk density range was 57-62 lbs/ft³. See Figure III-12 for a sample photomicrograph. MMPD was 148 μ m.

g. Flint Glass

Flint Glass is a leaded glass; that is, a silicate glass with lead oxide (PbO). The leaded glass used in testing was obtained from the Ferro Corporation. It consisted of 52.6% PbO, 40.5% S_iO₂; 4.8% Al₂O₃, and 2.1% Na₂O. The flint glass was only available as a frit with the 0-50% fraction going through a 63 μ m opening screen. The particles were irregular in shape with sharp, jagged edges. A clear photomicrograph could not be obtained. The size and shape of the flint glass load was considered marginal for useful testing.

h. 316L Stainless Steel

The stainless steel bed material used in testing was obtained from the Hoeganaes Corporation. This stainless steel had 17.5% chromium, 13% nickel, 2% molybdenum and 0.85% silicon. The manufacturer stated that some particles might show slight magnetism. The particle shape was generally poor, as shown in Figure III-13. Particle sizes and shapes were random, with varying thickness. This made the determination of the particle size difficult and uncertain;

Figure III-11. Graphite, MMPD = 157 Microns (33 X)

Figure III-12. Salt, MMPD = 148 Microns (33 X)

Figure III-13. 316 Stainless Steel, MMPD = 105 Microns (33 X)

Figure III-14. Zircon Sand, MMPD = 149 Microns (33 X)

a best estimate of 105 μ m MMPD was determined.

i. Zircon Sand

The Zircon sand used in testing was obtained as a beach sand from Industrial Foundry Supply Company. It was a "Newcastle Grade", with smooth spherical to ovaloid particles, as shown in Figure III-14. The sand was composed of 66.7% ZrO₂ and 33.2% SiO₂. Zircon sand is generally used for alloy steel casting. The MMPD was determined to be 149 μ m.

j. PPS Plastic

The plastic used in this study was polyphenylene sulfide (PPS). Polyphenylene sulfide is a cross-linkable aromatic thermoplastic that can withstand high temperatures. It has a continuous use temperature of 400°F and can withstand short-term exposure to temperatures of 500°F. The plastic was available as a powder in U.S. Standard Screen Mesh of 80 and was obtained from Nitrogen Processing (LNP) Corporation. The particles were angular as can be seen the photomicrograph of Figure III-15. Particle size was determined to be 205 μ m MMPD.

k. Nickel

The nickel powder used in the testing was obtained from Belmont Metals, Inc. It was 99.9% pure and available as a fine powder. The nickel particles were quite uniform and spherical, but the surface of the particles appeared to be dimpled (Figure III-16). Sample size was characterized by a MMPD of 95 μ m.

l. Solder

The solder powder was not available commercially in the required particle size for testing. Consequently, it was specially manufactured by the Powder Alloy Corporation. The solder was produced by atomization with an inert gas. It was composed of 60% tin and 40% lead, and had a low oxygen content. As can be seen in the photomicrograph of Figure III-17, the particles were spherical with smooth surfaces. (MMPD was determined to be 98 μ m).

Figure III-15. PPS Plastic,
MMPD = 205 microns (33x)

Figure III-16. Nickel,
MMPD = 95 microns

Figure III-17. Solder,
MMPD = 98 microns (33x)

D. TEST SET-UP AND INSTRUMENTATION

1. Test Facility

The facility for the fluidized bed material tests was built and activated at the Aerojet Engineering Research Laboratory. The facility was constructed out in the open, with the air and water supply and controls under cover of a corrugated aluminum roof. The entire facility is shown in Figure III-18.

a. Fluidized Bed

The fluidized bed used in the test program was a nominal two-foot diameter, well-insulated vessel. The use of a vessel this size substantially reduced the wall effects inherent in many research investigations. Indeed, Botterill (1972) suggests, "It would seem to be prudent to make full scale unit cell model tests of the fluidization behavior whenever feasible in order to be able to check on details of design". Here "unit cell" means the smallest part of a heat exchanger which contains features representative of the whole. Zabrodsky, et.al. (1976) also states "in future research it is necessary both to explain the decrease in h_{wall} when passing from laboratory scale equipment to commercial and to eliminate this reduction". This reduction in h_{wall} may be explained by the boundary conditions at the heat transfer surface (wall). If the thermal conductivity and the heat capacity of the wall (body) are low, the temperature of the surface changes rapidly, which induces a decline both in the heat flux and in h_{wall} . Obviously, with experimentation conducted in test vessels of less than four inches, the boundary conditions of the walls would substantially contribute to this effect.

The fluid bed test container was constructed from two 25.5-inch I.D., schedule 10, cylindrical 304 stainless steel propellant tanks with hemispherical ends. One hemispherical cap from each tank was removed and a 125-lb. flange was welded in its place. This resulted in a test vessel 6 feet high and two feet in diameter, as shown in Figure III-19. At

Figure III-18. Fluidized Bed Test Facility

Figure III-19. Fluidized Bed Test Facility

the lower hemispherical end, a 4 inch flange was fitted and connected to a 4 inch pipe for delivery of the fluidizing air. At the upper hemispherical end, a 4 inch flange and pipe were fitted for the exhaust system. The top of the vessel was attached to an overhead trolley to facilitate equipment handling during vessel loading. Provisions were also made at the top of the vessel to allow bed particle addition during testing. A valve at the lower end of the vessel was used as the drain. A port was provided for viewing the fluidized bed during testing; the system consisted of a light source and a sight window. The entire test vessel was covered with 2 inches of high temperature Kaowool insulation ($K=0.4 \text{ Btu/hr-ft}^2\text{°F-in}$ at 600°F) and an aluminum skin (Figure III-19).

The distributor plate consisted of 19 bubble caps screwed into a one-inch thick, 316 stainless steel plate. 7/8-inch diameter holes were arranged in two concentric rings about the center of the plate. The caps were fitted into the plate with Voi-shan copper seals to prevent leaking. The first ring from the center is at a radius of 4.5 inches with 6 equally spaced caps, and the second ring is at a radius of 9 inches from the center with 12 equally spaced caps (see Figure III-20). As shown in Figure III-21, each cap was constructed from 316 stainless steel hexagonal bar stock to a length of 2-3/4 inches. The top of caps were positioned two inches above the distributor plate. A half-inch diameter hole was drilled axially along the length of the cap, and six equally-spaced 0.196-inch diameter jets were drilled radially. The jets were inclined downward 15° from the horizontal to minimize bed material leakage into the plenum chamber. The jets were located approximately one inch below the top of the cap. The free area of the distributor plate was 0.76% of the total area. The distributor was mounted on a circular ring 13 inches from the bottom of the vessel, providing a plenum volume of 2.6 ft^3 . The pressure drop characteristics of the distributor plate were established by testing. Results are shown in the Figure III-22.

Figure III-20. Distribution Plate

Figure III-22. Distributor Plate Pressure Drop Characteristics

b. In-Bed Heat Exchanger Array

The water tube system (in-bed heat exchanger array) used in the testing is representative of actual heat transfer in waste heat recovery applications. Although it is true that the time constant of this water tube system is moderately longer than for an electrically heated tube, the data is felt to be more consistent for the former. Electrically heated tubes can give misleading data by exposing the bed particles to radiant energy versus the actual condition of cooling the bed. It has also been noted by Antonishin, et. al.(1972), that if there are internal heat sources in a body, the influence of an essential controlling parameter is less pronounced.

The in-bed heat exchanger array was increased from one as originally proposed to three horizontal rows. Therefore, two additional rows of horizontal tubes were included, with a non-central instrumented tube above and below the central row. Data obtained from this arrangement provided information on heat transfer to a tube in proximity of the distributor plate. Though this increase in the number of horizontal rows increased the depth of the bed, the bed would still be considered shallow, as defined by Ali and Broughton (1977)(a shallow fluidized bed generally has a height of one foot or less with an aspect ratio of 0.5 or less).

The horizontal heat exchanger tube bundle array was a staggered-tube arrangement with a four inch center-to-center horizontal pitch and a two inch center-to-center vertical pitch. Based on data presented by Gelperinand Ainshtein, this arrangement was selected to minimize tube bundle blockage effects and their subsequent influence on fluidized bed heat transfer. There were a total of eleven heat transfer tubes arranged in three rows. The first and third rows had four tubes, and the second row had three. Figure III-23 shows the tube orientation of the first row of tubes in the top cross-sectional view. The vertical section A-A shows all three numbered

Figure III-23

DISTRIBUTOR PLATE AND TUBE ORIENTATION

rows in their respective positions. Row 1 tubes were located two inches above the distributor plate nozzles. There was one 'active' heat exchanger tube per row; i.e., coolant was circulated through one tube in each row. Figure III-23 also shows the direction of water and air flows. Figure III-24 is a photograph of the heat exchanger array mounted in the fluid bed test vessel. The heat exchanger tubes were made from 304 stainless steel, 0.75" O.D. tubing, with a 0.035" wall thickness to minimize axial conduction. Each tube was connected to the vessel by a one-inch Swagelok fitting. One-eighth inch TFE teflon insulators were placed between the Swagelok fittings and the heat exchanger tubes to minimize conduction losses. Detailed connection arrangements can be seen in Figure III-25.

c. Air System

The fluidizing medium was heated air supplied by a positive-displacement air blower with a capability of delivering 350 scfm* at 6 psi. The air was heated by two G.E. air heaters connected in series. The first heater was rated at 12.5 kw and preheated the air to approximately 300°F. The second air heater was rated at 40 kw and completed the air heating. The heater arrangement is shown in Figure III-26. Since testing was based on specified bed temperatures, air temperature regulation was accomplished by a temperature controller sensor located within the bed. This controller sensor automatically controlled the air heaters to maintain the prescribed bed temperature. A cyclone separator was used to recover some of the elutriated bed material and a dust bag capable of retaining particles to 5 microns was used for final filtering (Figure III-27).

*Standard cubic feet per minute

Figure III-24 Unloaded
Fluid Bed Chamber - Plan View

Figure III-25. Coolant Tube Connector Arrangement

Figure III-26. Fluid Bed Heat Exchanger - Test Chamber

Figure III-27. Cyclone Separator

d. Water System

Coolant used during testing was deionized water. The coolant was stored in a 100-gallon reservoir, and delivered by a 50 gpm water pump. The flow then passed through a heat exchanger to eliminate the heat added by the pump. The coolant system was supported by a separate test stand from that of the fluidized bed and heaters, Figure III-28.

2. Instrumentation

The instrumentation used in the heat transfer for tests is shown schematically in the P & ID Fluid Bed Heat Exchanger Experiment drawing of Figure III-29. Each of the item designations on this drawing has been identified in the instrumentation summary shown on Table III-3. General descriptions of the instrumentation used in this program follow below.

a. Temperature Instrumentation

(1) Air Temperature Sensors

Fluidizing air temperatures were taken prior to heating, in each heater element, in the plenum of the bed, within the fluidized bed and in the exhaust air. The upstream air temperature was taken prior to the in-line orifice plate with an API iron-constantan thermocouple. This air temperature sensor was used in the determination of the air mass flow. Air temperature in the first heater was sensed with a Love chromel-alumel thermocouple. The second heater temperature was sensed with a Love iron-constantan thermocouple. The two heater temperature sensors were placed within the heater elements and used in conjunction with an in-bed temperature sensor to control the temperature of the bed to 400°F. The temperature in the plenum of the fluidized bed was sensed with a Chino platinum resistance temperature device (RTD). This sensor was centered in the plenum just in front of the distributor plate. The air temperatures just above

Figure III-28. Coolant Supply

Figure III-29. P & ID Fluid Bed Heat Exchanger Experiment

Table III-3.

INSTRUMENTATION SUMMARY

ITEM DESIGN *	LOCATION	SENSOR TYPE	RANGE	DIAL GAGE	READOUT DEVICE		MFG.	ACCURACY	REMARKS
					RECORDER	DIGITAL METER			
PI-1	Upstr. Air Orifice	Diaphragm	0-150" H ₂ O	X			Dwyer	+ 2" H ₂ O	
PI-2	Orifice Diff.	Diaphragm	0- 25" H ₂ O	X			Dwyer	+ 0.2" H ₂ O	+ 2% of Orifice Range
PI-11	Distr.	Diaphragm	0-150" H ₂ O	X			Dwyer	+ 2" H ₂ O	
PI-12	Distr. Diff.	Diaphragm	0-150" H ₂ O	X			Dwyer	+ 2" H ₂ O	
PI-13	Bed Diff.	Diaphragm	0- 50" H ₂ O	X			Dwyer	+ 0.5" H ₂ O	
PI-14	Bed Disc.	Diaphragm	0-15" H ₂ O	X			Dwyer	+ 0.2" H ₂ O	
PI-16	Coolant	Bourdon	0-100 PSIG	X			Ashcroft	+ 1 PSI	
PI-106	Row 1 - Bot. Diff.	Diaphragm	0-15" H ₂ O	X			Dwyer	+ 0.2" H ₂ O	
PI-206	Row 2 - Bot. Diff.	Diaphragm	0-25" H ₂ O	X			Dwyer	+ 0.2" H ₂ O	
PI-306	Row 3 - Bot. Diff.	Diaphragm	0-50" H ₂ O	X			Dwyer	+0.5" H ₂ O	
TI-1	Upstr. Air Orifice	I/C TC	0-500°F	X			API	+ 5°F	
TI-4	H-1 Element	C/A TC	0-1600°F	X			Love	+ 10°F	
TI-5	H-1 Air Disc.	C/A TC	0-2500°F	X			Love	+ 10°F	+ 5°F Air Control
TI-6	H-2 Element	I/C TC	0-1600°F	X			Love	+ 10°F	
TI-10	Row 1 Bed	I/C TC	0-1600°F	X			Love	+ 10°F	+ 5°F Air Control
TI-17	Coolant	Hg. Bulb	50-400°F	X			Ametek	+ 5°F	+ 3°F Coolant Control
TE-7	Distr. Air	Plat. RTD	200-400°F		X		Chino	+ 2°F	
TE-8	Bot. Bed	Plat. RTD	200-400°F		X		Chino	+ 2°F	
TE-9	Top Bed	Plat. RTD	200-400°F		X		Chino	+ 2°F	
TE-102-1,2	Row 1 Coolant	Plat. RTD	0-200°F		X		Chino	+ 1°F	
TE-103-1,2,3,4	Row 1 Tube	C/A TC	0-2500°F			X	Omega	+ 4°F	
TE-105	Row 1 Bed	Plat. RTD	200-400°F		X		Chino	+ 2°F	
TE-202-1,2	Row 2 Coolant	Plat. RTD	0-200°F		X		Chino	+ 1°F	
TE-203-1,2,3,4	Row 2 Tube	C/A TC	0-2500°F			X	Omega	+ 4°F	
TE-205	Row 2 Bed	Plat. RTD	200-400°F		X		Chino	+ 2°F	
TE-302-1,2	Row 3 Coolant	Plat. RTD	0-200°F		X		Chino	+ 1°F	
TE-303-1,2,3,4	Row 3 Tube	C/A TC	0-2500°F			X	Omega	+ 4°F	
TE-305	Row 3 Bed	Plat. RTD	200-400°F		X		Chino	+ 2°F	

* P = pressure, T = temperature, F = flow rate

Table III-3. INSTRUMENTATION SUMMARY (CONT.)

<u>ITEM DESIGN</u>	<u>LOCATION</u>	<u>SENSOR TYPE</u>	<u>RANGE</u>	<u>DIAL GAGE</u>	<u>READOUT DEVICE</u>		<u>MFG.</u>	<u>ACCURACY</u>	<u>REMARKS</u>
					<u>RECORDER</u>	<u>DIGITAL METER</u>			
FE-101	Row 1 Coolant	Turbine	0-5 GPM			X	Newport	+ 0.05 GPM	
FE-201	Row 2 Coolant	Turbine	0-5 GPM			X	Newport	+ 0.05 GPM	
FE-301	Row 3 Coolant	Turbine	0-5 GPM			X	Newport	+ 0.05 GPM	
TDR-21-1	Row 1 Coolant Diff.	Plat. RTD	0-200°F		X	X	Chino & Nes	+ 0.5°F	Diff. Calibration Pre and Post Test
TDR-21-2	Row 2 Coolant Diff.	Plat. RTD	0-200°F		X	X	Chino & Nes	+ 0.5°F	Diff. Calibration Pre and Post Test
TDR-21-3	Row 3 Coolant Diff.	Plat. RTD	0-200°F		X	X	Chino & Nes	+ 0.5°F	Diff. Calibration Pre and Post Test
PI-18	Fac. Water Reg.	Bourdon	0-60 PSIG	X			AGC	+ 1 PSI	
PI-19	Fac. Water	Bourdon	0-200 PSIG	X			USG	+ 5 PSI	
PI-20	Coolant Pump Seal	Bourdon	0-200 PSIG	X			USG	+ 5 PSI	
PI-21	N ₂ Source	Bourdon	0-200 PSIG	X			USG	+ 5 PSI	
PI-22	N ₂ Reg.	Bourdon	0-60 PSIG	X			Ashcroft	+ PSI	

the distributor plate and at the top of the bed were also sensed with Chino RTDs. Three temperature probes were placed within the fluidized bed to check for bed temperature uniformity (Chino platinum RTD's). These temperature sensors were placed in-line vertically and at the three tube row levels.

The top of the P & ID drawing, Figure III-29, indicates the temperature sensors as applied to the active tubes. Sensors used to measure wall temperatures are designated TE-103, TE-203 and TE-303 for row 1, row 2 and row 3 respectively. There were four wall sensors on each tube (Omega chromel/alumel thermocouples). The details of the sensor placement can be seen in Figure III-30. Thermocouples 1 and 2 were placed on the upstream part of the tube, and thermocouples 3 and 4 were placed downstream. The thermocouples were inserted in 0.020 x 0.020-inch slots on the sides of each tube, then silver soldered to form a smooth outside surface. These thermocouples were used to give an indication of actual versus calculated wall temperatures.

(2) Water Temperature Sensors

Temperature sensors were placed in the water upstream of each active tube and at the inlet and outlet of each active tube. The temperature in the upstream manifold was sensed by an Ametek mercury thermometer placed in a temperature well. This sensor was used to control coolant to within $\pm 3^{\circ}\text{F}$. The inlet and outlet temperatures, TE-102, TE-202, and TE-302, for each row were sensed by Chino platinum RTD's. The detail placement of these RTD probes can be seen in Figure III-25. Each RTD was fitted to its coolant tube by a 3/4-inch Swagelok tee. A 3/4 inch PVC flex hose was then used to deliver coolant water to each tube. Figure III-31 shows the sensor placement on the fluidized bed (the foil and insulation have been removed to expose the RTD sensors and heat exchanger tubes). Temperature differential readouts, TDR-21-1, TDR-21-3, were used to determine the differential temperatures of the coolant from the inlet to the outlet of each active tube. The sensors used to determine the temperature rise of the coolant flow to each active tube were calibrated prior to each

Figure III-30. Wall Temperature Instrumentation - Coolant Tubes

Figure III-31. Coolant Temperature Sensors

Figure III-32. Purge Flow Meters

test run using ice baths and boiling water. Discrepancies were noted and incorporated into the data reduction sheets.

b. Flow Instrumentation

The water flow in each active tube was measured with Newport turbine meters and these turbine meters were first calibrated at the AECC Standards Laboratory, and were subsequently checked prior to each test run.

c. Pressure Instrumentation

Pressure sensors were placed upstream of the air orifice plate, across the air orifice plate, in the fluidized bed plenum chamber, across the distributor plate, within the bed chamber and in the freeboard. All the pressure sensors were diaphragm-type manufactured by Dwyer. The bed differential pressure sensors were placed in-line, with one sensor per tube row. To prevent fluidized solids from entering the pressure lines, a small air flow was delivered to each line and controlled by flow rotameters, as seen in Figure III-32. Dampening pots were placed on-line with these pressure sensors to minimize pressure fluctuations in the dial gages. This pressure instrumentation can be seen in Figure III-33. Pressure sensors were also placed in the coolant flow with Bourdon-type sensors manufactured by Ashcroft and USG. Details are given in Table III-3.

d. Instrumentation Trailer

A trailer (Figure III-34) was used to house most of the temperature recorders (Figure III-35) needed in the fluidized bed program. The trailer also contained the pressure controller and U-meter assembly needed in the test program. The trailer was stationed beside the fluid bed test facility for quick access.

E. TEST PROCEDURES

The operating procedures used in the test program are included as Appendix B.

Figure III-33 Pressure Instrumentation

Figure III-34. Instrumentation and Control Trailer

Figure III-35. Temperature Recorders

F. DATA ACQUISITION AND REDUCTION

The basic experimental data obtained from the heat transfer tests consisted of temperatures, pressures, and frequency signals, the latter being the output from turbine-type liquid flowmeters. The instrumentation for the experimental test setup is shown in Figure III-29 (AECC Drawing No. 1490701). In the course of testing, a set of operating conditions was established using a given bed material. When a steady-state was achieved, as indicated by essentially constant readings of key temperature and flowrate indicators, extensive sets of test data were manually logged onto "Fluidized-Bed Heat Exchanger Data Log" sheets. These log sheets then served as the data input source to two data reduction programs written for the TI-59 programmable calculator. Using these data reduction programs, the raw experimental data were reduced immediately on-the-spot to define bed operating conditions and heat transfer coefficients. Normally, replicate data were logged and reduced approximately one-half hour later. Bed operating conditions (particularly, bed superficial velocity) were then changed and the process was repeated until all the sets of data necessary to characterize the heat transfer behavior of a given fluidized-bed test material were established. Examples of the "Fluidized-Bed Heat Exchanger Data Log" and the "Reduced Data from Fluidized-Bed Heat Exchanger Tests" are shown in Figures III-36, III-37. respectively.

1. Air Circuit Data Reduction

The reduction of the experimental data dealing with the air flow circuit was accomplished using one of the special programs written for the TI-59 programmable calculator as mentioned above. The program utilized logged temperature and pressure data as input to define: (1) the fluidization air mass flow rate, (2) the average air and bed temperature within the fluidized bed region at tube-row No. 2, (3) the average pressure

DATE: _____		BED MATERIAL: _____		NOMINAL PARTICLE SIZE: _____		RUN NO: _____												
TAG NO. _____ SERVICE _____		UNITS	TIME															
AIR SUPPLY SYSTEM																		
PI-21	SOURCE PRESSURE	psig																
PI-22	REGULATED PRESSURE	psig																
COMBUSTOR CONTROL AREA																		
TI-5	H-1 HEATER DISCHG. TEMP.	°F																
TI-4	H-1 HEATER ELEMENT TEMP.	°F																
WATER SUPPLY SYSTEM																		
PI-19	SOURCE PRESSURE	psig																
PI-18	REGULATED PRESSURE	psig																
PI-16	COOLANT PUMP DISCHG. PRESS.	psig																
PI-20	COOLANT PUMP SEAL PRESS.	psig																
TI-17	COOLANT TEMPERATURE	°F																
FLUIDIZED-BED CONTROL AREA																		
TI-1	AIR ORIFICE INLET TEMP.	°F																
PI-2	AIR ORIFICE INLET PRESS.	in.H ₂ O																
PI-3	AIR ORIFICE ΔP	in.H ₂ O																
PI-306	BED ΔP TO ROW 3	in.H ₂ O																
PI-206	BED ΔP TO ROW 2	in.H ₂ O																
PI-106	BED ΔP TO ROW 1	in.H ₂ O																
PI-11	WINDBOX PRESSURE	in.H ₂ O																
PI-13	TOTAL BED ΔP	in.H ₂ O																
PI-14	BED DISCHG. PRESSURE	in.H ₂ O																
TI-10	BED TEMP. CONTROL	°F																
TI-6	H-2 HEATER ELEMENT TEMP.	°F																
PI-12	DISTRIBUTOR PLATE ΔP	in.H ₂ O																
FI-62	PI-12	PURGE AIR	SCFH															
FI-63	PI-106	PURGE AIR	SCFH															
FI-64	PI-206	PURGE AIR	SCFH															
FI-65	PI-306	PURGE AIR	SCFH															
FI-66	PI-13	PURGE AIR	SCFH															
FI-67	PI-14	PURGE AIR	SCFH															

Figure III-36. Fluidized-Bed Heat Exchanger Data Log (Page 1)

DATE: _____ BED MATERIAL: _____ NOMINAL PARTICLE SIZE: _____ RUN NO: _____

TAG NO. _____ SERVICE _____ UNITS _____

TIME

DIGITAL TEMPERATURE INDICATOR, TI-20

TS-20-1	ROW 1	TUBE WALL, POSITION 1
TS-20-2	ROW 1	TUBE WALL, POSITION 2
TS-20-3	ROW 1	TUBE WALL, POSITION 3
TS-20-4	ROW 1	TUBE WALL, POSITION 4
TS-20-5	ROW 2	TUBE WALL, POSITION 1
TS-20-6	ROW 2	TUBE WALL, POSITION 2
TS-20-7	ROW 2	TUBE WALL, POSITION 3
TS-20-8	ROW 2	TUBE WALL, POSITION 4
TS-20-9	ROW 3	TUBE WALL, POSITION 1
TS-20-10	ROW 3	TUBE WALL, POSITION 2
TS-20-11	ROW 3	TUBE WALL, POSITION 3
TS-20-12	ROW 3	TUBE WALL, POSITION 4

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TEMPERATURE INDICATOR/RECORDERS

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TR-21-1 ROW 1 COOLANT TEMP. RISE
TR-21-2 ROW 2 COOLANT TEMP. RISE
TR-21-3 ROW 3 COOLANT TEMP. RISE
TR-22-1 WINDBOX TEMP.
TR-22-2 BED BOTTOM TEMP.
TR-22-3 BED AT ROW 1
TR-22-4 BED AT ROW 2
TR-22-5 BED AT ROW 3
TR-22-6 BED TOP TEMP.
TR-23-1 ROW 1 COOLANT INLET
TR-23-2 ROW 1 COOLANT OUTLET
TR-23-3 ROW 2 COOLANT INLET
TR-23-4 ROW 2 COOLANT OUTLET
TR-23-5 ROW 3 COOLANT INLET
TR-23-6 ROW 3 COOLANT OUTLET

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DIGITAL COOLANT FLOW INDICATORS

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FI-101 ROW 1 COOLANT FLOW
FI-201 ROW 2 COOLANT FLOW
FI-301 ROW 3 COOLANT FLOW

```

COUNTS
COUNTS
COUNTS

DIGITAL COOLANT AT INDICATORS

TI-21-1 ROW 1 COOLANT TEMP. RISE
TI-21-1 ROW 2 COOLANT TEMP. RISE
TI-21-3 ROW 3 COOLANT TEMP. RISE

OFF

Figure III-36. Fluidized-Bed Heat Exchanger Data Log (Page 2)

[illegible]

UNITS: TEMPERATURE IN DEGREE CELSIUS, TEMPERATURE AND PRESSURE									
V_{AIR}	lb/hr								
ρ_{AIR}	lb/ft ³								
T_{AIR}	°F								
P_{AIR}	psia								
V_{AIR}	SUPERFICIAL, ft/sec								

[illegible]

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at tube-row No. 2, (4) the air density at tube-row No. 2, (5) the superficial air velocity through the bed at tube-row No. 2 based on the minimum free cross-sectional area in the heat transfer region (which occurs at tube-rows No. 1 and 3), and (6) the heat input to the fluidized bed by the stream of fluidizing air. Each of the items of reduced data mentioned above were defined as follows:

- 1) Fluidized air mass flow rate, $\dot{w}_{air}$, (lb/hr):

$$\dot{w}_{air} = a_i + b_i x + c_i x^2 \quad (\text{Eqn. 1})$$

$$\text{where } x = \left[\frac{(T_1 + 459.67)(P_3)}{(0.036127 P_2 + 14.696)} \right]^{1/2}$$

and a_i , b_i , and c_i are mathematically derived correlation coefficients for the appropriate air orifices (F0-3) developed from calibration tests against standardized critical orifices.

- 2) Average air and bed temperature within the fluidized bed region at tube-row No. 2, $T_{Avg, air}$, ($^{\circ}\text{F}$):

$$T_{Avg, air} = (T_{22-3} + T_{22-4} + T_{22-5})/3 \quad (\text{Eqn. 2})$$

- 3) Average pressure at tube-row No. 2, $P_{Avg, air}$, (psia):

$$P_{Avg, air} = 14.696 + .036127 \left[\frac{(P_{11} - P_{12}) + (P_{14} + P_{13})}{2} - \frac{(P_{106} + P_{206} + P_{306})}{3} \right] \quad (\text{Eqn. 3})$$

- 4) Air density at tube-row No. 2, ρ_{air} , (lb/ft^3):

$$\rho_{air} = \frac{2.6984(P_{Avg, air})}{(T_{Avg, air} + 459.67)} \quad (\text{Eqn. 4})$$

- 5) Superficial air velocity at tube-row No. 2 based on the minimum free cross-sectional area, $V_{\text{air, superficial}}$, (ft/sec):

$$V_{\text{air, superficial}} = \frac{\dot{w}_{\text{air}}}{3600(\rho_{\text{air}})(A_{\text{min}})} \quad (\text{Eqn. 5})$$

where $A_{\text{min}} = 2.79957 \text{ ft}^2$, the free cross-sectional flow area at tube-row No. 1 and 3.

- 6) Heat input to the fluidized bed by the stream of fluidizing air, q_{air} , (Btu/hr):

$$q_{\text{air}} = \dot{w}_{\text{air}} \{a_0(T_{22-1} - T_{\text{Avg,air}}) + b_0[(T_{22-1})^2 - (T_{\text{Avg,air}})^2]\} \quad (\text{Eqn. 6})$$

where a_0 and $b_0 = 0.23541$ and 1.239×10^{-5} , respectively and are the coefficients to an air enthalpy equation of the form $H = a_0 T + b_0 T^2$

2. Heat Transfer Coefficient Data Reduction

The reduction of the experimental test data related to heat transfer characteristics and coefficients at each instrumented tube in the fluidized-bed heat exchanger test system was accomplished using another special program written for the TI-59 programmable calculator. The program utilized logged temperatures and water flow rate data to define the following parameters:

(1) the average bulk temperature of water flowing through each tube, (2) the specific enthalpy rise of the water flowing through each tube, (3) the mass flow rate of the water through each tube, (4) the heat input to each tube from the fluidized bed, (5) the average overall heat transfer coefficients, (6) the average inside heat transfer coefficients, (7) the resistance to heat transfer across the tube walls, (8) the average outside heat transfer coefficients, (9) the average temperature drop across the inside liquid films, (10) the average temperature drop across the tube walls, and (11) the average temperature drop across the outside films. Each of the items of reduced data mentioned above were defined as follows:

- 1) Average bulk temperature of water flowing through the ith tube where $i = 1, 2, \text{ or } 3$, $\overline{T}_{H_2O,i}$, ($^{\circ}F$):

$$\overline{T}_{H_2O,i} = \frac{T_{23-(2i-1)} + T_{23-(2i)}}{2} \quad (\text{Eqn. 7})$$

- 2) Specific enthalpy rise of the water flowing through the ith tube where $i = 1, 2, \text{ or } 3$, $\Delta H_{H_2O,i}$, (Btu/lb):

$$\Delta H_{H_2O,i} = \int_{T_{in}}^{T_{out}} C_{p,H_2O} dt \quad (\text{Eqn. 8})$$

$$\text{where } C_{p,H_2O} \text{ (Btu/lb } ^{\circ}F) = 1.0109 - 2.5417 \times 10^{-4} (T, ^{\circ}F) + 1.1845 \times 10^{-6} (T, ^{\circ}F)^2$$

$$T_{out} = T_{23-(2i)}$$

$$T_{in} = T_{23-(2i)} - T_{21-(i)}$$

- 3) Mass flow rate of the water flowing through the ith tube where $i = 1, 2, \text{ or } 3$, $\dot{w}_{H_2O,i}$, (lb/hr):

$$\dot{w}_{H_2O,i} = (k_i)(\rho_{H_2O, \text{inlet}})(F_{(i)01}) \quad (\text{Eqn. 9})$$

where k_i = calibration factor for the ith water flow meter

$$\rho_{H_2O, \text{inlet}} = 1.0119 - 1.9 \times 10^{-4} (T_{inlet}, ^{\circ}F)$$

$$\text{and } T_{inlet} = T_{23-(2i-1)}$$

$F_{(i)01}$ = frequency signal from the ith flow meter

- 4) Heat input to the ith tube from the fluidized bed where $i = 1, 2, \text{ or } 3$, $q_{H_2O,i}$, (Btu/hr):

$$q_{H_2O,i} = (w_{H_2O,i})(\Delta H_{H_2O,i}) \quad (\text{Eqn. 10})$$

- 5) The average overall heat transfer coefficient of the ith tube where $i = 1, 2, \text{ or } 3$, $\bar{U}_i$, (Btu/hr·ft²·°F):

$$\bar{U}_i = \frac{q_{H_2O,i}}{(A_{o,i})(T_{\text{Bed},i} - \bar{T}_{H_2O,i})} \quad (\text{Eqn. 11})$$

where $A_{o,i}$ = outside area of the ith tube

$$T_{\text{Bed},i} = T_{22-(i+2)}$$

- 6) The average inside heat transfer coefficient of the ith tube where $i = 1, 2, \text{ or } 3$, $h_{\text{inside},i}$, (Btu/hr·ft²·°F) was iteratively computed using the Colburn equation (Eqn. 12).

$$h_{\text{inside},i} = 0.023 C_p G \left(\frac{\mu_f}{DG} \right)^{0.2} \left(\frac{k}{\mu_f C_p} \right)^{2/3} \quad (\text{Eqn. 12})$$

wherein the viscosity, μ_f , was evaluated at the mean of the average bulk liquid ($\bar{T}_{H_2O,i}$) and average inside surface temperatures (T_{is}) and the heat capacity and thermal conductivity were evaluated at the average bulk liquid temperature ($\bar{T}_{H_2O,i}$). The heat capacity (C_p, H_2O), thermal conductivity (k_{H_2O}), and viscosity (μ_{H_2O}) of the water were defined by Eqn. 12a, 12b, and 12c, respectively.

$$C_{p,H_2O}(\text{Btu/lb } ^\circ\text{F}) = \frac{\Delta H_{H_2O,i}}{T_{21-(i)}} \quad (\text{Eqn. 12a})$$

$$k_{H_2O}(\text{Btu/ft}\cdot\text{hr}\cdot^\circ\text{F}) = .30187 + 7.3504 \times 10^{-4} (T, ^\circ\text{F}) - 1.4216 \times 10^{-6} (T, ^\circ\text{F})^2 \quad (\text{Eqn. 12b})$$

$$\mu_{H_2O}(\text{lb}_m/\text{ft}\cdot\text{hr}) = .058397 \times 10^{446.04/(207.67 + T, ^\circ\text{F})} \quad (\text{Eqn. 12c})$$

The quantity D, the inside diameter of the tubes, was 0.056 ft and the quantity G, the mass flow rate of water per unit area, was $(396.51)(\dot{w}_{H_2O,i}) \text{ lb/hr}\cdot\text{ft}^2$.

- 7) The resistance to heat transfer across the ith tube wall, $R_{\text{wall},i}$, ($\text{hr}\cdot^\circ\text{F/Btu}$) was iteratively computed using Eqn. 13.

$$R_{\text{wall},i} = x_{\text{wall}} / (k_{\text{wall}} \cdot \bar{A}_{\text{wall},i}) \quad (\text{Eqn. 13})$$

where $x_{\text{wall}} = 0.00291667 \text{ ft}$

$\bar{A}_{\text{wall},i}$ = average wall area for tube "i", ft^2

$$k_{\text{wall}} = 8.2625 + 0.00644 (\bar{T}_{\text{wall},i}), \text{ Btu/ft}\cdot\text{hr}\cdot^\circ\text{F}$$

where $\bar{T}_{\text{wall},i}$ is the average wall temperature of tube "i", $^\circ\text{F}$

- 8) The average outside heat transfer coefficient of the ith tube where $i = 1, 2, \text{ or } 3$, $h_{o,i}$, ($\text{Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$) was computed by solving Eqn. 14 for $h_{o,i}$.

$$\frac{1}{\bar{U}_i A_{o,i}} = \frac{1}{h_{o,i} A_{o,i}} + \frac{1}{h_{\text{inside},i} A_{\text{inside},i}} + R_{\text{wall},i} \quad (\text{Eqn. 14})$$

where $\bar{U}_i$ is defined by Eqn. 11

$h_{\text{inside},i}$ is defined by Eqn. 12

$R_{\text{wall},i}$ is defined by Eqn. 13

$A_{o,i}$ = outside area of the ith tube, ft^2

$A_{\text{inside},i}$ = inside area of the ith tube, ft^2

- 9) The average temperature drop across the inside liquid film of the ith tube where $i = 1, 2, \text{ or } 3$, $\Delta T_{\text{inside},i}$, ($^{\circ}\text{F}$):

$$\Delta T_{\text{inside},i} = \frac{q_{\text{H}_2\text{O},i}}{(h_{\text{inside},i})(A_{\text{inside},i})} \quad (\text{Eqn. 15})$$

- 10) The average temperature drop across the wall of the ith tube where $i = 1, 2, \text{ or } 3$, $\Delta T_{\text{wall},i}$, ($^{\circ}\text{F}$):

$$\Delta T_{\text{wall},i} = (q_{\text{H}_2\text{O},i})(R_{\text{wall},i}) \quad (\text{Eqn. 16})$$

- 11) The average temperature drop across the outside film of the ith tube, where $i = 1, 2, \text{ or } 3$, $\Delta T_{o,i}$, ($^{\circ}\text{F}$):

$$\Delta T_{o,i} = \frac{q_{\text{H}_2\text{O},i}}{(h_{o,i})(A_{o,i})} \quad (\text{Eqn. 17})$$

3. Outside Heat Transfer Coefficient Correction

During the course of the experimental work and data reduction using the procedures discussed in Section III, it was first suspected, and subsequently confirmed experimentally, that the outside heat transfer coefficients were somewhat in error. The errors resulted from the fact that the water flowing through the tubes was in the laminar/turbulent transition region, and the Colburn equation used to calculate the inside heat transfer coefficient was inaccurate for the application. That inaccuracy was confirmed in experiments wherein the water mass flow rate was changed significantly ($Re = 4900$ to 12900), while all other test parameters remained essentially constant. In these tests, the outside heat transfer coefficient, as deduced from the data reduction technique described previously, was observed to vary with the water mass flow rate.

Other equations and correlations for evaluating the inside heat transfer coefficient, besides the Colburn equation, and supposedly applicable to the transition region, were also studied; however, the deduced outside heat transfer coefficients still exhibited erroneous dependencies on the water mass flow rate. It was thus concluded that no really good correlation was available for defining the inside heat transfer coefficients in the flow regime generally being employed in the tests. It was found, however, that both the Colburn and Hausen⁽¹⁾ equations consistently predicted for each of the three tube rows, the same values for the inside heat transfer coefficient at certain Reynold's numbers near 9500. Based on those observations, the Colburn equation was assumed to yield a correct value for the inside heat transfer coefficient at those certain mass flow rates corresponding to the Reynold's numbers indicated above and, thus, the corresponding outside heat transfer coefficient would be correct at those specific water mass flow rates. Having established a water mass flow rate at which the outside heat transfer coefficient was deemed to be correct by the data reduction procedures described

⁽¹⁾ Z. Ver. deut. Ing. Beih. Verfahrenstech., No. 4, 91 (1934).

earlier, and recognizing that correct value must be independent of the mass flow rate of the water, correction factors to be applied to the "data-reduced" outside heat transfer coefficients (which reflected errors in the calculated inside coefficients) were empirically defined as functions of the mass flow rate of water through the tubes. These correction factors are shown in Figure III-38 and were applied to the "data-reduced" outside heat transfer coefficients to yield the corrected values shown in the various graphical data presentations in this report. Generally, the correction amounted to about an 8% increase in the "data-reduced" outside heat transfer coefficient.

G. ERROR ANALYSIS OF TEST DATA

The error analysis procedures of the AECC test data are included as Appendix C.

Figure III-38. Correction Factors Vs Water Flow

IV. TEST RESULTS

A. DESCRIPTION OF BASIC DATA

A summary of the test run on this program is given in Table IV-1. A total of 31 tests were run using a variety of bed materials and particle sizes.

The mean particle sizes were established by a screen analysis taken before and after each test. A summary of the screen analysis results is given in Table IV-2. Particle sizes were observed to change in testing, either measuring slightly larger, due to elutriation of fines, or slightly smaller, due to attrition. This variation averaged less than $\pm 7\%$, and the sizes listed in Table IV-2 are mean values. Notable exceptions to this were found for the silica sand, size F-80, which increased in size by 26%, the 316 stainless steel powder which decreased by 15%, the expanded alumina, size -40 mesh, which decreased by 22%, and the expanded alumina, size +40 mesh, which decreased by 24%.

The bed temperatures were nominally 400°F, with the exception of three tests which were run at 250°F in order to accommodate the temperature-sensitive materials, which were plastic, Test Run No. 22, and solder, Test Run No. 26. Nickel, Test Run No. 21, was also run at 250°F, since it was part of a comparison test involving the above materials.

The superficial velocities shown in Table IV-1 are defined as the velocity through the tube bundle at the minimum cross section. This is about 15% greater than the approach velocity to the bundle. In most cases, the bed was fluidized sufficiently for the heat transfer coefficient on the gas-side of the tube bundle to reach a maximum, or nearly so.

The nominal water flowrate, for all tests but one, was 500 lb/hr. The one exception was Test 20, in which the water flowrate was varied from 363

TABLE IV-1

TEST SUMMARY

Run No.	Bed Description		Purpose of Test	Bed Temp (°F)	Test Conditions			Comments
	Material	MMPD Dia. (μM)			Superficial Velocity Range (fps)	Water Flowrate Range (per Row) (lb/hr)	Water Enthalpy Change (Btu/lb)	
1	Silica Sand	176	Checkout	400	2.2-2.7	482-505	11.1-14.7	
2,3	Silica Sand	176	Particle Size/ Thermal Conductivity	400	.7-1.8	483-496	11.0-15.5	Calibration questionable
4	Silica Sand	279	Particle Size	400	1.1-2.8	491-515	11.0-13.0	H.T. not peaked
5	N/A	N/A	H.T. with no bed	400	.5-3.0	462-510	0.6- 2.4	
6	Silica Sand	445	Particle Size	400	.7-2.9	481-509	6.3-11.2	
7	Zircon Sand	149	Density	400	.7-2.9	482-498	11.0-16.4	H.T. not peaked
8	Alumina	168	Density/Particle Size	400	.6-2.6	459-505	9.5-16.5	
9	Glass Beads	174	Particle Size	400	.7-2.7	457-519	12.2-18.3	
10	Glass Beads	507	Particle Size	400	1.1-2.7	478-510	7.9-10.6	
11	Glass Beads	333	Particle Size	400	1.0-2.9	461-532	11.5-13.5	
12	NaCl	148	Thermal Conductivity	400	.4-1.0	467-505	8.4-13.6	H.T. not peaked - Low superficial velocity - Large % fines in bed load
13	Void Test	---	---	---	---	---	---	
14	316 SS	105	Density	400	.7-2.6	467-503	6.0-14.0	Invalid data..flow meter calibration shift
15	Glass Beads	174	Rerun	400	.7-2.5	375-531	13.5-19.6	Poor particle shape. MMPD highly uncertain
16	Glass Beads	174	Bed Depth Effects	400	1.6-1.7	494-504	1.1-11.0	
17	316 SS	105	Density	400	1.0-1.9	497-501	9.6-12.8	
18	Ni	95	Specific Heat	400	.6-1.6	476-530	11.8-17.5	H.T. not peaked
19	Alumina	283	Particle Size	400	1.1-2.6	488-514	10.6-13.6	
20	Ni	95	Determine Inside H	400	1.1-1.1	363-1000	9.6-21.5	One data point taken
21	Ni	95	Specific Heat	250	.6-1.3	353-511	7.2-10.6	
22	Plastic	205	Density	250	.6-.9	481-508	2.7-4.7	Extreme "clumping" was observed due to static elec.
23	Graphite	131	Thermal Conductivity	250	.3-.7	500-519	2.8-6.2	H.T. not peaked. Graphite tended to blow away.(Poor particle shape)
24,25	Silica Sand	176	Rerun	400	.9-2.7	485-498	12.9-15.8	
26	Solder	98	Specific Heat	250	.5-2.3	472-506	4.8-8.6	
27	Graphite	157	Thermal Conductivity	400	.6-2.4	468-515	10.5-17.4	
28	Void Test	174						Bed agglomerated due to high moisture
29	Expanded Alumina	202	Density	400	.8-2.7	483-500	8.4-12.4	
30	Expanded Alumina	505	Particle Size	400	.7-2.6	485-509	2.6-9.3	
31	Silica Sand	176	Check-out	400	.9-1.8	492-502	12.5-15.1	

TABLE IV-2
BED MATERIAL SIZE

<u>Material Description</u>	<u>Condition</u>	<u>MMPD, Microns</u>
Glass Beads		
Size B	As Received	---
	As Received	507
	After Run No. 10	<u>507</u>
	Average	507
Size C	As Received	---
	As Received	328
	After Run No. 11	<u>337</u>
	Average	332.5
Size AD	As Received	---
	As Received	172.8
	After Run No. 9	171.4
	During Run No. 13 (1330 hrs)	178
	During Run No. 13 (1420 hrs)	174
	During Run No. 16 (1045 hrs)	167.3
	During Run No. 16 (1230 hrs)	174
	During Run No. 16 (1335 hrs)	<u>177.6</u>
	Average	173.6
Nickel Powder	As Received	---
	As Received	95.7
	After Run No. 18	98.4
	After Run No. 20	96.2
	After Run No. 21	<u>91</u>
	Average	95.3

TABLE IV-2 (cont.)

<u>Material Description</u>	<u>Condition</u>	<u>MMPD, Microns</u>
Alumina (Al_2O_3) Large Size	As Received	288.8
	As Received	303
	After Run No. 19	265.5
	After Run No. 19	<u>275.1</u>
	Average	283.1
Salt (NaCl)	As Received	144
	After Run No. 12	<u>151</u>
	Average	147.5
Polyphenylene Sulfide	As Received	205
	After Run No. 22	<u>---</u>
	Average	205
Graphite	As Received	133
	After Run No. 23	<u>129.4</u>
	Average	131.2
316 Stainless Steel	As Received	---
	As Received	108.4
	After Run No. 14	114.6
	After Run No. 17	<u>91.6</u>
	Average	104.9

TABLE IV-2(cont.)

<u>Material Description</u>	<u>Condition</u>	<u>MMPD, Microns</u>
Silica Sand ~100 Mesh	As Received	169.4
	After Run No. 3	<u>182</u>
	Average	175.7
F-80	As Received	204
	After Run No. 4	257.3
Coarse Blend	As Received	447
	As Received	447
	As Received	441.5
	After Run No. 6	447
	After Run No. 6	<u>443</u>
	Average	445.1
Zircon Sand	As Received	150.3
	As Received	---
	After Run No. 7	<u>148</u>
	Average	149.1
Alumina (Al ₂ O ₃) Exolith	As Received	170
	As Received	---
	After Run No. 8	<u>166</u>
	Average	168

TABLE IV-2 (cont.)

<u>Material Description</u>	<u>Condition</u>	<u>MMPD, Microns</u>
Rounded Graphite	As Received	157.5
	After Run No. 27	<u>156.7</u>
	Average	157.1
Solder Powder	As Received	98.4
	Before Run No. 26	95.5
	After Run No. 26	<u>100.0</u>
	Average	98
Expanded Alumina Size -40 Mesh	Before Run No. 30	227
	After Run No. 30	<u>177.8</u>
	Average	202.4
Expanded Alumina Size +40 Mesh	Before Run No. 31	575.4
	After Run No. 31	<u>435.5</u>
	Average	505.4

to 1000 lb/hr. This was done to determine the effect of Reynolds Number on the liquid-side heat transfer coefficient in the transition regime.

The water enthalpy change for each test is given in column 8 of Table IV-1. Since the specific heat of water is approximately 1.0, this column represents the temperature change of the water as it flows through the test section. The accuracy of the temperature measurements is estimated to be $\pm 0.5^{\circ}\text{F}$ yielding an uncertainty in the heat transfer coefficient of approximately $\pm 10\%$.

Figures IV-1 through IV-25 present the gas-side heat transfer coefficient versus superficial velocity for the bed materials and test conditions noted in Table IV-1. The intent of varying superficial velocity was to determine the point at which the heat transfer coefficient reached a maximum value. Results are given for all 3 tube rows.

Data was taken on one tube in each of the three tube rows. Tube Row 1 (nearest the distributor plate) generally yielded the lowest heat transfer coefficient. The row effect variation was generally less than the estimated uncertainty in the heat transfer coefficient ($\pm 10\%$), and so it was difficult to determine whether this apparent row effect was real, or due to a small bias in the instrumentation of the system.

As a baseline for comparison, Run 5 was conducted with no bed material. The results of this test may be seen in Figure IV-3. Row 1, the leading tube, gave the highest value of heat transfer coefficient, while Row 3, the trailing tube, gave the lowest value. The published literature indicates that, for staggered tube bundles, the leading tube should experience the lowest heat transfer coefficient. It is hypothesized that the row closest to the distributor received an unusual degree of free stream turbulence, which enhanced heat transfer. If the enhanced heat transfer to Row 1 is due to free stream turbulence induced by the distributor plate, then the free stream turbulence must be strongly inhibited by the addition of bed material.

Figure IV-1. Fluidized Bed Heat Transfer Coefficients - Runs 2 & 3

Figure IV-2. Fluidized Bed Heat Transfer Coefficients - Run 4

Figure IV-3. Fluidized Bed Heat Transfer Coefficients - Run 5

Outside Film Coefficient, h_0 , Btu/ft² hr. °F

Figure IV-4. Fluidized Bed Heat Transfer Coefficients - Run 6

Figure IV-5. Fluidized Bed Heat Transfer Coefficients - Run 7

Figure IV-6. Fluidized Bed Heat Transfer Coefficients - Run 8

Figure IV-7. Fluidized Bed Heat Transfer Coefficients - Run 9

Outside Film Coefficient, h_o , Btu/ft² hr, °F

Figure IV-8. Fluidized Bed Heat Transfer Coefficients - Run 10

Outside Film Coefficient, h_o , Btu/ft² hr. °F

Figure IV-9. Fluidized Bed Heat Transfer Coefficients - Run 11

Outside Film Coefficient, h_o , Btu/ft² hr., °F

Figure IV-10. Fluidized Bed Heat Transfer Coefficients - Run 12

Figure IV-11. Fluidized Bed Heat Transfer Coefficients - Run 14

Outside Film Coefficient, h_o , Btu/ft².hr.°F

Figure IV-12. Fluidized Bed Heat Transfer Coefficients - Runs 14 & 17

Figure IV-13. Fluidized Bed Heat Transfer Coefficients - Runs 9 & 15

Figure IV-14. Bed Depth Effects - Run 16

Figure IV-15. Relative Bed Depth Effects - Run 16

Outside Film Coefficient, h_0 , Btu/ft². hr. °F

Figure IV-16. Fluidized Bed Heat Transfer Coefficients - Run 18

Figure IV-17. Fluidized Bed Heat Transfer Coefficients - Run 19

Figure IV-18. Fluidized Bed Heat Transfer Coefficients - Runs 18, 20 & 21

Figure IV-19. Fluidized Bed Heat Transfer Coefficients - Run 22

Figure IV-20. Fluidized Bed Heat Transfer Coefficients - Run 23

Figure IV-21. Fluidized Bed Heat Transfer Coefficients - Run 24 & 25

Outside Film Coefficient, h_o , Btu/ft² °F

Figure IV-22. Fluidized Bed Heat Transfer Coefficients - Run 26

Outside Film Coefficient, h_o , Btu/hr ft² °F

Figure IV-23. Fluidized Bed Heat Transfer Coefficients - Run 27

Figure IV-24. Fluidized Bed Heat Transfer Coefficients - Run 29

Figure IV-25. Fluidized Bed Heat Transfer Coefficients - Run 30

A careful look at bed test data in Figures IV-1 through IV-25 reveals that, on occasion, Row 1 yielded the lowest heat transfer coefficient, and, on other tests, Row 1 gave the highest coefficient or crossed the other curves. This inclined one to think that, although there may be a maldistribution or free stream turbulence effect with no bed material, the presence of a bed greatly reduces this effect and that the resultant row to row variations are within the uncertainty of the data.

In an effort to assess the effect of bed depth on the gas-side heat transfer coefficient, test 16 was run, Figure IV-14. The bed depth was varied from the low point, where Row 1 was partially covered, to where all rows were covered. The unusual "humpback" curves are attributed to the increased turbulence on the top of the tubes when they were just barely covered by the bed. As the bed depth increased, there tended to be an accumulation of bed material that stagnated on the top of the tubes, thus insulating them from heat conduction.

When the bed depth is normalized with respect to the bottom of the tube (i.e., bed depth equals zero at the bottom of the tube), we get the curve shown in Figure IV-15. Each curve peaks at the same relative bed depth, indicating that similar mechanisms are apparent on each tube.

Most of the tests yielded valid results. However, there were notable exceptions, as summarized in Table IV-3. These runs were either terminated or eliminated in the data analysis due to either poor resultant data or incomplete testing. First, Runs No. 2 and 3, silica sand with an MMPD = 176 microns, were eliminated from the data analysis due to suspicion that calibrations were improper.

Test Run No. 12 was a test with salt bed material to determine the effects of thermal conductivity on the heat transfer coefficient. As can be seen in Figure IV-10, the maximum heat transfer coefficient was not obtained. Elutriation was severe throughout the test due to the high percentage (21%) of fines in the material. Consequently, the run was terminated at a low superficial velocity (one foot per second) prior to reaching the maximum heat transfer coefficient.

TABLE IV-3
INVALID TESTS

Run No.	Material	Basis for Data Rejection
2,3	Sand	Questionable calibration
12	Salt	Material had high (21%) percentage of fines. Elutriation resulted in shutdown before maximum heat transfer was attained.
13	Glass Ballotini	Shift in flow meter calibration.
14,17	316 Stainless Steel	Particle shape poor. Not possible to obtain a valid measure of effective particle diameter.
22	Polyphenylene Sulfide	Severe agglomeration of bed material due to static electricity.
23	Graphite	High rate of elutriation resulted in shutdown before maximum heat transfer was obtained. Poor particle shape.
28	Glass Ballotini	High moisture resulted in bed agglomeration.
--	Flint Glass	Hygroscopic particles agglomerated. Unable to conduct test.

Test Run No. 13 was the first attempt to assess the effect of bed depth on the gas-side heat transfer coefficient. The material used in this test was the glass ballotini (beads), size AD, with a MMPD = 174 microns. This test was run without any apparent difficulty. But upon the completion of the data reduction, it was noted that results were not consistent with those of previous test run (Test Run No. 9) which used the same material. The facility and instrumentation were examined, and a shift in the operating point of the flow instrumentation was discovered, probably as a result of a prolonged down-time during the December holidays. The test run was voided, and the facility equipment was re-calibrated.

The test results on Runs 14 and 17, 316 stainless steel, are considered suspect because of the irregular particle shape (see Section III photomicrograph). This makes determination of a meaningful particle diameter highly questionable, and screen analysis to be misleading. Since particle diameter is the major influence on heat transfer, it was decided to eliminate these tests in the data analysis process.

Run No. 22 was a test intended to be used to determine the effects of a low density material such as Polyphenylene Sulfide (PPS), on the heat transfer coefficient. As can be seen in Figure IV-19, the reduced data was markedly different from previous test runs. Once the upper half of the fluidized bed vessel was removed for inspection, the cause of the erratic test data was made clear. The material developed static electricity, causing it to cling to the walls of the vessel, despite the fact that the entire internal surface of the fluidized bed was sprayed by an anti-static agent. This can be seen in Figures IV-26 and IV-27. Because of this adherence of the plastic material to the internal surfaces of the fluidized bed, it was felt that the determination of valid heat transfer coefficients would not be possible; therefore, the test was terminated.

Run No. 23 was a graphite test to determine the effects of particle thermal conductivity on the gas-side heat transfer coefficient. The resultant superficial velocity vs. outside film coefficient curve can be seen in Figure IV-20. During the test, it was found that the graphite elutriated rapidly at relatively low superficial velocities. For this reason, Test No. 23 was terminated before peak heat transfer coefficients could be established. The cause of this excessive elutriation was poor particle shape. The particles had a somewhat flattened appearance as shown in the photomicrograph of Section III. This resulted in a high drag-to-weight ratio, and the bed material readily elutriated. Subsequently, a rounded graphite material was obtained and tested (Test Run No. 27).

Test Run No. 28 was a check-out. The glass ballotini beads, size AD, were used. This material is hygroscopic. During the course of this test, the material did not dry out and fluidize properly, so valid data could not be obtained.

FLUIDIZED BED INTERNALS AFTER TESTING WITH PPS PLASTIC

FIGURE IV-26. FLUIDIZED BED INTERNALS AFTER RUN WITH POLYPHENYLENE SULFIDE BED SHOWING THE BED MATERIAL CLINGING TO ALL SURFACES BECAUSE OF ELECTROSTATIC ATTRACTION

FIGURE IV-27. FLUIDIZED BED INTERNALS SHOWING DISTURBED COATING OF PPS PLASTIC

Flint glass was obtained from the Ferro Corporation as a glass frit. This material was approximately 200 U. S. Standard Screen Mesh size, which was found to be too small to be effectively tested. This material is also hygroscopic. As with the glass beads, the material formed into an agglomerate. Once the heated air flow was applied to this mass, it fractured into random size pieces, but the material did not return to particulate form. The test run was terminated.

B. DATA ANALYSIS

The data plotted and discussed in the previous section was manipulated to arrive at heat transfer coefficient values from which particle size and thermophysical property effects could be determined. Applied to the basic data were adjustments for, (1) selection of the maximum heat transfer coefficients, (2) data averaging, (3) temperature correction, and (4) normalization by equivalent particle diameter. These adjustments are described in detail below. Results are presented in Table IV-4.

1. Maximum Heat Transfer Coefficient Determination

In work by Ainshtein and Gelperin (1966), and Botterill (1975) it was pointed out that the missing feature in existing correlations is the hydrodynamic condition in the bed. Though the governing hydrodynamic conditions of the bed are not entirely understood, they are dependent upon superficial velocity. To ensure reasonable similarity of the hydrodynamic conditions in the bed, the maximum heat transfer coefficients were selected on the basis of comparable multiples of the minimum fluidization velocity, U_{mf} ; i.e., at equal fluidization numbers (U/U_{mf}).

U_{mf} was determined through the use of the correlation developed by Leva, et al. (1956). This correlation avoids the use of the voidage at the minimum fluidizing velocity, ϵ_{mf} , and the particle shape factor, ϕ , factors which cannot be estimated reliably. The minimum fluidizing velocity, as defined by Leva, is:

$$U_{mf} = \frac{4.6 \times 10^{-10} d_p^{1.8} \rho_p (\rho_p - \rho_g)^{0.94}}{0.88 \mu_g}$$

where, U_{mf} is the minimum fluidizing velocity
 d_p is the mean particle diameter
 ρ_g is the fluid density
 ρ_p is the particle density
 μ_g is the gas viscosity

Table IV-4
ADJUSTED MAXIMUM HEAT TRANSFER COEFFICIENTS

Data Point No.	Material	Test Run Reference	Equivalent Particle Diameter (μM)	Maximum Heat Transfer Coefficient ($\text{Btu}/\text{ft}^2\text{hr}\text{-}^\circ\text{F}$)					
				Row 1	Row 2	Row 3	Average	Temperature Corrected	Diameter Corrected
1	Silica Sand	24, 25	172	90	85	90	88	-	94
2	Silica Sand	4	279	72	71	76	73	-	100
3	Silica Sand	6	439	55	58	60	58	-	99
4	Graphite	17	137	100	90	88	96	-	89
5	Alumina	8	170	80	94	92	89	-	94
6	Alumina	19	266	69	68	75	71	-	95
7	Expanded Alumina	29	163	68	62	61	64	-	67
8	Expanded Alumina	30	444	48	45	42	45	-	77
9	Zircon	7	147	88	94	97	93	-	92
10	Nickel	21	91	112	102	-	107	114	89
11	Solder	26	88	100	85	93	93	99	76
12	Glass Ballotini	9	164	97	95	100	97	-	101
13	Glass Ballotini	11	333	68	66	68	67	-	100
14	Glass Ballotini	10	494	51	53	58	54	-	98

This method was used to determine the U_{mf} and fluidization number for each material in the data analysis. The fluidization number was used as an index to test the comparability of the data. For all materials with particle diameter near 150 microns, the fluidization number used for data normalizing was 25. There were two exceptions. The heat transfer rate for the graphite powder and the expanded alumina did not attain a maximum until the fluidization number was over 80. The larger diameter materials were normalized at fluidization numbers of 15 and 5 for diameters near 300 and 500, respectively. The larger expanded alumina material (equivalent particle diameter of 444 microns) was evaluated at a larger fluidization number of 15. The reason for these exceptions for graphite and expanded aluminas is not clear.

2. Data Averaging

From the overview of all the data presented in Section IV A, it was observed that the resulting heat transfer coefficients varied from row to row. This row variation, after several inspections of the instrumentation, appeared to be actual. But the magnitude of the variations were not beyond the limits of the possible overall error of the test results. Consequently, it could not be positively assumed that Row 3 data was generally higher nor that Row 1 data was generally lower for most of the tests. Also, this pattern was not consistent throughout the entire test program. As an example, in Run 26 (solder) and Run 27 (graphite) higher heat transfer coefficients were found for Row 1. To minimize any possible bias due to either a row effect or instrumentation, the data used in the analysis was averaged from the three rows. This equalized the generally higher coefficients of Row 3 with generally lower coefficients of Row 1.

3. Temperature Corrections

In Section IV. A., it was reported that the bed temperature of the solder test run reduced to 250°F to maintain the integrity of this material. This solder (60% tin, 40% lead) has a low melting temperature because the two main constituents, lead and tin, form a eutectic with a melting point of 361°F.

To preserve the pair testing concept, solder and its pair nickel were tested at the reduced bed temperature of 250°F. The data from these test runs, Run No. 21 - Nickel and Run No. 26 - Solder, were then standardized to the bulk of the data tested at 400°F. This was done according to G. A. Henwood, United Kingdom Atomic Energy Report No. AERE-R 6027, 1969. His curve on the variation of the maximum value of the heat transfer coefficient with gas conductivity indicated a 6.8% increase in the heat transfer coefficients when the bed temperature was increased from 250°F to 400°F. The effect of this increase can be seen in Table IV-4.

4. Correcting for Equivalent Diameters

The mean particle diameter, d_p , used in the Leva correlation for minimum fluidization velocity is the "equivalent particle diameter", d_e , defined by:

$$d_e = \frac{1}{\sum \left(\frac{x}{d_{avg}} \right)}$$

where d_e is the equivalent particle size in microns, x is the weight fraction of the sieved particles retained between two consecutive screens, and d_{avg} is the average of the size openings of these two consecutive screens. Equivalent particle diameters were computed and are shown in Table IV-4. These were used in all correlations and data corrections.

The particle size identification reported previously was "mean mass particle diameter" (MMPD). This method designates the particle diameter at the mid-point of the sample as the representative particle size. At that point, 50% of the weight of the material would have a particle diameter either larger or smaller than the MMPD. The MMPD method provided a quick identification of material throughout the test program. The use of equivalent particle diameter in the data analysis is more in keeping with analytical techniques used in the current literature (Botterill, 1975). Generally, the equivalent particle diameters are less than those obtained with the MMPD method, but the deviation is generally less than 10%.

For purposes of making direct material pair comparisons of heat transfer rate data, a correction was applied to normalize this data to a common particle diameter of 150 μ m. The results, shown in the last column of Table IV-4, were obtained by multiplying the average heat transfer coefficient by $(d_e/150)^{0.5}$, where d_e is the equivalent particle diameter in μ m. The exponent 0.5 was taken from previous work (Yasutomi and Yakota) and subsequently found to fit the results of the current investigation, as discussed in the following discussion of results.

C. DISCUSSION OF RESULTS

Covered in this section are the derivation of material property influences on heat transfer coefficient by two methods. The first of these involves evaluation of results from tests on materials which have nearly equal values of two of the thermophysical properties, with the third property being widely different; this is referred to as "direct pair comparison". The second method applies all of the data in a regression analysis. Discussed also are the results of tests to isolate the effect of particle diameter upon heat transfer rate.

The results of both techniques are quite comparable. The analyses yield the following correlation for heat transfer rate as a function of the bed material parameters:

$$h \propto d_e^{-0.5} \rho^{0.2} C_p^{0.2} k^0$$

where h = maximum gas-side heat transfer coefficient Btu/hr-ft²-°F

d_e = equivalent particle diameter μm

ρ = specific gravity of particle

C_p = specific heat of particle, Btu/lb-°F

k = particle thermal conductivity, Btu/hr-ft-°F

1. Particle Diameter Effects

The averaged (three tube rows) maximum heat transfer coefficients for various equivalent particle diameters for sand, glass beads, alumina, and expanded alumina are plotted in Figure IV-28. Also shown are curve fits which relate heat transfer coefficient to particle diameter to the -0.5 power. Most of the data points fall within $\pm 5\%$ of the curve fit.

Figure IV-28. Particle Diameter Effects - Four Bed Materials

The exception is expanded alumina, where the error is roughly $\pm 10\%$. Overall, the results show good correlation of maximum heat transfer coefficient to the -0.5 power of particle diameter.

The finding noted above is in complete agreement with the work of Yasutomi and Yakota (1977). For each material....copper, sand and glass....their analysis indicated the maximum heat transfer coefficients were proportional to the -0.5 power of particle diameter. Their results can be seen in Figure IV-29. They reported differences of $h_{\max}$ between sand and glass particles were small and remained within 10%, whereas copper particles showed 25% higher values than sand. The AECC results also showed small differences between the sand and glass particles (within 5%). The work by Yasutomi and Yakota was conducted in small beds ($4''\phi$, $5.5''\phi$ and $12''\phi$) using a vertical heater. The magnitudes of their coefficients were higher, but generally as would be expected for vertical surfaces as opposed to horizontal surfaces.

The AECC results vary somewhat from the widely accepted Zabrodsky correlation, where the reported dependence of the maximum heat transfer coefficient was found to be proportional to the 0.36 power of particle diameter. His empirical relationship for the maximum particle convective bed-to-surface transfer coefficient correlated the experimental data for spherical particles using the products of combustion. Attempting to fit the AECC data to Zabrodsky's result roughly doubles the error band of data versus the curve fit; clearly, the correlation of the findings of the current study according to the -0.5 exponent yields the better result.

2. Direct Material Pair Comparisons

Addressed below are results of direct pair comparison tests in terms of material property effects on gas-side heat transfer coefficient. Also discussed are the findings of this investigation in relationship to results reported by other researchers.

Figure IV-29. Dependence of Maximum Heat Transfer Coefficient On Particle Diameter

a. Thermal Conductivity Effects

The results of the direct pair comparison of particle thermal conductivity effects upon the maximum heat transfer coefficient can be seen in Figure IV-30. The materials compared were silica sand, size 100 mesh, and the rounded graphite powder. This should represent a valid comparison of thermal conductivity effects since for sand and graphite the specific heats are 0.195 Btu/lb-°F and 0.202 Btu/lb-°F, the specific gravities are 2.3 and 2.25, while the thermal conductivity varies from 1 Btu/hr-ft-°F and 82 Btu/hr-ft-°F, respectively. Once the maximum heat transfer coefficients had been adjusted to eliminate the effect of particle diameter, the results showed a slightly higher coefficient for the sand material (94 Btu/hr-ft²-°F) than for the graphite powder (89 Btu/hr-ft²-°F). This would imply a slight dependence on particle thermal conductivity (straight line on log-log plot gives exponent of -.03). But the data scatter is well within the overall error band of the test system (Figure IV-30). It is concluded that the effect of particle thermal conductivity on heat transfer is negligible. This is consistent with the findings of the regression analysis, in which a very small exponent was found for the thermal conductivity effect.

The AECC findings are in agreement with the majority of test results reported in the current literature by Zabrodsky (1978), Davidson & Harrison (1971), Gelperin & Einstein (1965) and Ziegler (1964). Ziegler's experimental works closely resembled the AECC pair-testing concept. He tested copper and nickel of equal size, and of nearly equal density and specific heat, but with a four-fold variation in thermal conductivity. His test results indicated that particle thermal conductivity had a negligible effect on the heat transfer coefficient. However, the AECC results differed from the results of Botterill, et. al. (1970), which showed a 30-fold increase in the heat transfer coefficient following a shift from glass to copper beads. Zabrodsky (1976) explains this by suggesting that the heat wave penetrates only particles next to the wall and a change in the conductivity of the solid can be of significance for heat transfer only at the conditions when the thermal resistance of the gas and that of the solid particle are commensurate. Under uniform bed expansion

Figure IV-30. Thermal Conductivity Effects - Direct Pair Comparison

conditions, high gas flowrates lead to an increase in the average distance between the wall and the adjacent row of solid particles; i.e., the thermal resistance of the gas becomes much greater than that of the particles, and the influence of k would be of no significance. The AECC test program was conducted with high gas flowrates, so the results are not necessarily in conflict with the results of Botterill.

The use of effective bed thermal conductivity as proposed by Baskakov (1964) does suggest that the particle conductivity would have a significant effect on the fluid bed heat transfer rate. Gelperin and Ainshtein reported (Fluidization by Davidson & Harrison, 1972) that the increase of the heat transfer coefficient with the effective bed thermal conductivity was in complete agreement with the packet theory. The reason for disagreement between the AECC data and the theory based on Baskakov's effective bed thermal conductivity is not entirely understood.

b. Heat Capacitance Effects

The results of direct pair comparison tests for heat capacitance are shown in Figure IV-31. The materials used in testing were nickel and solder, with specific gravities of 8.9 and 8.9, thermal conductivities of 53 Btu/hr-ft-°F and 27 Btu/hr-ft-°F, and with heat capacitances of 0.11 Btu/lb-°F and 0.045 Btu/lb-°F, respectively. Note that thermal conductivity was found to have a negligible effect on heat transfer. The two data points yield an exponent of 0.20 in the relationship $h \propto C_p^x$.

It is noted that the surface roughness of the nickel particles (see photomicrograph of Section III) may have influenced the results of testing on that material. The particle surfaces are dimpled rather than smooth. Baerg, Klaussen and Gishler carried out experiments with smooth glass beads and with the same glass beads with etched surfaces. They noted a decrease in the maximum heat transfer coefficient for the etched particles. Zabrodsky later explained that the smooth particles yielded higher transfer coefficients

Figure IV-31. Heat Capacitance Effects - Direct Pair Comparison

due to the decreased friction between the particles. The testing of smooth nickel particles might have yielded a larger coefficient.

There are a number of researchers who have found dependency of the heat transfer coefficient upon particle heat capacity, but the dependence varies widely, viz:

Andeen & Glicksman (1976, 1978) and Zabrodsky (1966, 1978)	C_p ignored
Wender & Cooper (1958)	$h \propto C_p^{0.8}$

The most comparable work to the AECC test program was that of Ziegler (1964). He tested nickel and solder powders of equal size and nearly equal densities, but with a two-fold variation in specific heat. Ziegler found the maximum Nusselt number increased 60% with a 100% increase of heat capacity in comparing data on nickel and solder. He concluded that heat transfer was proportional to particle specific heat taken to the 0.4 power. This is twice the exponent obtained during the AECC test program. This is not a significant discrepancy considering the potential error band of the AECC test data and the widely differing experimental conditions under which the heat transfer characteristics were determined.

The AECC particle specific heat results do appear to conflict with Zabrodsky's maximum particle convective component, inasmuch as this parameter is not included in his correlation. A review of his book, Hydrodynamics and Heat Transfer in Fluidized Beds, 1966, gives an explanation for this omission. Zabrodsky states that the specific heat parameter can be approximately replaced by the particle density parameter in the formulas for maximum wall heat transfer coefficients because of the nearly single-valued inverse dependence of specific heat on the specific weight (or bulk density) of the material (the rule of Dulong and Petit, which states that the specific heat at constant volume for one atomic weight of any solid elements is 6.4). This directly relates the specific heat to density of any solid element. Zabrodsky has recognized that particle specific heat indirectly affects the maximum wall heat transfer coefficient by its relationship to particle density. Through pair testing, AECC has tested and shown that heat capacitance does have a direct effect on the heat transfer coefficient.

c. Density Effects

The results of the particle density effects on the maximum heat transfer coefficients are presented in Figure IV-32. The materials used in testing for this effect were solid alumina and expanded (hollow) alumina. The solid alumina produced a 40 percent higher maximum heat transfer coefficient than the expanded alumina using data that had been normalized to eliminate the effect of differing particle diameters. The differences in their densities was approximately four-fold with identical specific heats. Curve fitting of Figure IV-32 gives an exponent of 0.26 in the proportionality relationship.

As is the case with other bed particle parameters, there is controversy regarding the impact of particle density on heat transfer. First, the heat transfer coefficient was reported to be an inverse function of particle density by Vreendenberg (1976), and Andeen and Glicksman (1978). Zabrodsky (1966) explained their results by conjecturing that there may be an appreciable decrease in the maximum heat transfer in a fluidized bed of low density material due to low turbulence. In this case, the limit of bed stability is reached at low superficial fluid velocities. The bed has a high voidage even at low superficial fluid velocities, so that the particle velocities are also low. As a result, the particles get overheated at the heat transfer surface. Under otherwise equal conditions, the high bed voidage is accompanied by a reduction in the heat transfer coefficient.

For materials above a minimum weight, Zabrodsky (1966) and Gelperin (1969) both report that the heat transfer coefficients are proportional to the 0.2 power of particle density. This differs with the coefficient found in the AECC direct pair comparison tests. The problem may be in the AECC data. In the AECC tests, a true density pair was not available, and the tests had to be based on the use of a porous material in which the captured air within the spherical walls can interchange with the fluidizing gas. In addition, the

Figure IV-32. Particle Density Effects - Direct Pair Comparison

expanded alumina material had thin walls that were susceptible to fracture under vigorous fluidizing condition and evenly distributed weight fractions throughout the particle size range. This even distribution made particle size characterization difficult. The ease of fracture of the particles was shown by the 16% reduction in particle size after the test run.

3. Regression Analysis

A multiple covariance (regression) analysis of the AECC data was completed to correlate the heat transfer coefficients in terms of the thermodynamic properties of the particle. The Multiple Covariance Analysis Computer Program (FD0088) was used for this purpose. This program is based on the multiple regression analysis procedure as developed by Aerojet. This method utilizes the analysis of covariance to estimate the regression coefficients and simultaneously obtain estimates of the between-unit and within-unit variance components. These variance estimates are then used in a generalized least square fit to obtain regression estimates.

The regression analysis was made with all of the data from the last column of Table IV-4, with the heat transfer coefficients adjusted for a particle size of 150 microns according to the -0.5 power of diameter. The results were:

$$h = 115 \rho^{0.19} C_p^{0.23} k^{-0.04}$$

where h is in BTU/hr-ft²-°F
 ρ is the specific gravity
 C_p is in BTU/lb-°F
 k is in BTU/hr-ft-°F

This function is shown in Figure IV-33 with respect to the data points for the eight material data points given in Table IV-4. The standard deviation was 0.077. The density exponent of 0.19 found in the regression analysis was the factor, when combined with uncertainties in testing expanded alumina, that led to the

Figure IV-33. Results of Regression Analysis

adaptation of a density exponent of 0.2 despite a larger value having been indicated by direct pair testing.

4. Selected Curve Fit

The results of direct pair comparison and multiple regression analyses are given below according to the relationship

$$h \propto d_e^{-0.5} \rho^x c_p^y k^z.$$

	$\frac{x}{0.26}$	$\frac{y}{0.20}$	$\frac{z}{-0.03}$
pairs comparison	0.26	0.20	-0.03
regression analysis	0.19	0.23	-0.04

Considering experimental uncertainty, the final selection of a curve fit is made by combining the pairs comparison and regression results and then rounding off to one significant figure. This yields

$$h \propto d_e^{-0.5} \rho^{0.2} c_p^{0.2} k^0$$

where h = maximum gas-side heat transfer coefficient, Btu/hr-ft²-°F
 d_e = equivalent particle diameter, μm
 ρ = particle specific gravity
 c_p = particle specific heat, Btu/lb-°F

V. APPLICATION OF RESULTS

In the design of heat exchangers with gas phase heat transfer, the gas-to-surface heat transfer has been the limiting design factor. Typically, the forced-convection, fluid-to-surface heat transfer coefficients for atmospheric air in conventional tubular heat exchangers are in the range of 5 to 30 Btu/hr-ft²-°F. This low gas-to-surface heat-transfer coefficient yields a low overall heat transfer conductance, the result being that the typical gas-to-liquid heat exchanger will be approximately four times larger than the typical liquid-to-liquid unit for the same heat duty. The two conventional approaches for improving the gas-side heat transfer rate are (1) increase the gas velocity through the heat exchanger or (2) use extended surfaces (fins) on the tubes. Both methods can enhance heat transfer considerably. But, in the area of waste heat recovery, heat transfer enhancement by these two methods is limited by the available system pressure drop and, in some cases, by excessive fouling, where fins cannot be used. An alternative solution is the fluidized bed heat exchanger.

Discussed in this section are the conceptual design of a fluidized bed heat exchanger based upon previous findings as well as a comparison of relative heat exchanger costs using various bed materials.

A. WASTE HEAT BOILER CONCEPTUAL DESIGN

The primary area in which fluidized bed heat exchangers offer potential advantages over conventional heat exchangers is in waste heat recovery from gas streams. For this application, the fluidized bed heat exchanger offers heat transfer area reductions by factors of three or more even when compared with finned tubes. Furthermore, with the use of shallow beds and light weight bed materials, the air side pressure drops can be kept at the same level as in conventional heat exchangers.

A conceptual design of a fluidized bed heat exchanger for waste heat recovery has been made to assess its potential for energy recovery in comparison with a conventional steam generator. The application investigated was the recovery of the waste heat from the exhaust gases of a large diesel-electric generator. The Binary Rankine Cycle Waste Heat Recovery System designed by Mechanical Technology, Inc., (MTI) was chosen as the basis for comparison. This selection was based on compatibility of fluidized bed characteristics with the heat exchanger requirements of the conventional system and on the availability of operating specifications and fabrication costs.

The MTI binary cycle uses both water and Freon as working fluids to recover 500 kw from the exhaust of a turbocharged diesel engine. Figure V-1 shows a schematic of the cycle. The steam portion of the binary cycle is implemented by a conventional waste-heat steam generating unit with economizer, evaporator and superheater sections using 9.6×10^6 Btu/hr of heat recovered from the exhaust gases. The steam generator is a conventional finned-tube construction with provisions for soot blowers. Exhaust gas flow of 205,000 lb/hr at 520°F is ducted into this low pressure steam generator to raise 9000 lb/hr of 430°F, 60 psia steam from 215°F feedwater. Detailed fabrication costs are given below in Table V-1 and the system specifications are given in Table V-2.

Figure V-1. Schematic of MTI Binary Rankine Cycle System for Waste Heat Recovery/Electric Power Generation

TABLE V-1

MTI DESIGN SPECIFICATIONS				
	<u>Preheater</u>	<u>Vaporizer</u>	<u>Superheater</u>	<u>Overall</u>
Heat Transfer Rate, Btu/hr	713,700	8,237,800	648,900	9,600,000
Exhaust Gas Flow Rate, lbm/hr	205,000	205,000	205,000	205,000
Water/Steam Flow Rate, lbm/hr	9,000	45,000	9,000	9,000
Entering Gas Temperature, F	345.6	506.1	520	520
Exiting Gas Temperature, F	333	345.6	506.1	333
Entering Water Temperature, F	215	293	293	215
Exiting Water Temperature, F	293	293	430	430
Gas Heat Transfer Coefficient, Btu/ft ² -hr-F	12.6	12.6	12.6	12.6
Water Heat Transfer Coefficient, Btu/ft ² -hr-F	1090	493	65.6	-
Overall Heat Transfer Coefficient, Btu/ft ² -hr-F	9.27	8.54	4.41	-
Total Surface Area, sq ft	816	8160	1224	10200
Tube Diameter, in.	1.875	1.875	1.875	1.875
Fin Diameter, in.	3.375	3.375	3.375	3.375
Fin Spacing, per in.	3	3	3	3
Tube Length, ft	221	2213	332	2766
Number of Tube Rows	2	20	3	25
Gas Pressure Loss, in. H ₂ O	0.34	3.36	0.50	4.2
Water Pressure Loss, psia (2.29)		6.27	2.06	10.52

TABLE V-2

MTI Steam Generator Fabrication Costs

Unit	Cost
Superheater	\$ 3,000
Preheater	2,500
Evaporator	26,200
Steam Separating Drum/ Circulating Pump System	13,300
Installation Costs	<u>5,000</u>
	\$ 50,000

1. Design Configuration

The element of the steam generator chosen to be designed as a fluidized bed heat exchanger was the evaporator section. This section alone carries 86 percent of the total thermal load, with the remaining thermal load divided equally between the preheater and superheater sections. Also, the design of this section as a fluidized bed heat exchanger does not incur a significant system penalty due to a reduction in the log mean temperature difference (LMTD); i.e., both the fluidized bed temperature and the temperatures of the liquid-vapor mixture inside the boiler tubes are relatively uniform. The resultant conceptual design is then a fluidized bed for the evaporator section with the conventional finned tube preheater and superheater as proposed by MTI. This conceptual design can be seen in Figure V-2.

As can be seen by Figure V-2, a parallel fluidized bed approach was taken in the design of the evaporator. The parallel bed concept would allow for an optimum volume to package the requisite heat transfer surface area without incurring the increase in gas pressure drop as for series fluidized beds. The evaporator is formed by flat shallow fluidized beds stacked on top of one another, with a single row of finned heat exchanger tubes immersed in each bed. The bottom of each bed serves as both a separator to compartmentize each unit and as the distributor plate of each unit. The distributor plates are designed to have one-third of the pressure drop of the bed itself in order to maintain fluidizing stability as recommended by Zenz (1960).

In the steam generator, the diesel exhaust gases enter the superheater section at 520°F and are cooled to 506.1°F, identical to the MTI design. The gases would then enter the plenum areas of the fluidized bed evaporator section. A small temperature loss in the transition between the superheater section and the fluidized bed would result in a reduced evaporator inlet gas temperature of 503.6°F. The gases would then flow up through each distributor plate to the fluidized beds. The gases would leave the fluidized bed at the bed temperature of 342.8°F. The gas streams are then collected and passed through the preheater section at 340.3°F. Other than the small temperature losses before and after entering the fluidization bed evaporator section, the changes to the MTI design are limited to the evaporator section alone. These changes are compared in Table V-3.

Figure V-2. AECC Fluidized Bed Waste Heat Steam Generator.

TABLE V-3
EVAPORATOR COMPARISONS

SPECIFICATION	MTI DESIGN	AECC DESIGN
Heat Transfer Rate, Btu/hr	8,237,800	8,237,800
Exhaust Gas Flow Rate, lbm/hr	205,000	205,000
Water/Steam Flow Rate, lbm/hr	45,000	45,000
Entering Gas Temperature, °F	506.1	503.6
Exiting Gas Temperature, °F	345.6	342.8
Entering Water Temperature, °F	293	293
Exiting Water Temperature, °F	293	293
Gas Heat Transfer Coefficient, Btu/ft ² -hr-°F	12.6	90
Water Heat Transfer Coefficient, Btu/ft ² -hr-°F	493	2,700
Overall Heat Transfer Coefficient, Btu/ft ² -hr-°F	8.54	75
Total Surface Area, sq ft	8,160	2,223
Tube Diameter, in.	1.875	0.709
Fin Diameter, in.	3.375	1.5
Fin Spacing, per in.	3	7
Tube Length, ft	2,213	1,280
Number of Tube Rows	20	1/tray
Gas Pressure Loss, in. H ₂ O	3.36	3.0
Water Pressure Loss, psia	6.27	2.6

The fluidized beds incorporate a single row of Wolverine Trufin copper-finned heat exchanger tubes. The tube specification is shown in Table V-4.

Table V-4
Tube Specification

Diameter of fins	1.50 inch
Root diameter of finned section	0.709 inch
Inside diameter	0.625 inch
Fin height	0.520 inch
Mean fin thickness	0.021 inch
Number of fins per inch	7
Outside area per linear foot	1.738 ft ²

The fin spacing was selected to maximize heat transfer surface in the evaporator without interfering with the particle circulation. This provided for a more compact design of the unit. Korolev, et al. (1975) stated that, if the height of the projections and the distance between them are greater than the particle size, then higher heat transfer coefficients are obtained. Higher coefficients were not assumed, but fin spacing was checked to minimize heat transfer degradation.

Silica sand was selected as a reference bed material because of its universal use. From AECC test data, an averaged maximum heat transfer coefficient was selected at the highest superficial velocity without serious elutriation. This gave a heat transfer coefficient of 90 Btu/hr-ft²-°F at 2.5 feet per second. The overall heat transfer coefficient was 75 Btu/hr-ft²-°F. The total required heat transfer area was 2,223 ft² (1280 feet of tubing). This represented a 70% reduction in the heat transfer area over the conventional design. Since the beds were made as shallow as possible to reduce the gas pressure losses, the tubes were placed as close to the distributor plate as possible without obstructing particle circulation. Consequently, the flow area had to be

enlarged to account for the cross-sectional area of the tubes in the path of the gas flow so that the desired superficial velocity would be maintained. This increased the total flow area from 456 ft² to 531 ft², or a flow area per tray of 177 ft². The optimum volume/tube spacing configuration was then a tray size of 13.5 ft x 13.5 ft with 32 tubes per tray on 5-inch centers.

To keep the gas pressure loss to a minimum, the center line of the heat exchanger tubing was placed one inch above the plate. A 60% expansion of the bed at the operating superficial velocity was assumed, with a one-third bed pressure drop through the distributor plate. These two pressure losses, plus the losses due to the changing gas velocities through the unit, result in an overall gas pressure loss of three inches of water.

The transport disengagement height was calculated at approximately one foot, and the back panel of each flattened pan was extended by this amount to limit elutriation. An additional two feet of freeboard was added above each bed to minimize the velocity change from the bed to the preheater section. A two-foot-high entrance to the evaporator section was also incorporated to the unit. This resulted in an overall unit height of 13 feet.

2. ROM Cost Estimates

A rough order of magnitude cost estimate for the parallel fluidized bed heat exchanger was made to assess the economics of this conceptual design. Several items were taken into account for the economic analysis, including all of the construction costs and the cost of the superheater, preheater and steam separating drum/circulating pump system for the MTI design. The cost of the evaporator/steam generator unit included material costs of the heat exchange tubing, feed lines and manifolds, distributor and frame costs, bed material, and assembly costs. The construction costs were based on a characteristic volume and included site preparation and erection costs. Site preparation included costs for excavation, drainage channels and drainage lines. Erection costs included costs of materials and structural members as well as labor. The costs of the superheater, preheater

and steam separating drum/circulating pump system were taken from the MTI report since these units would remain unchanged.

The results showed a total unit cost of \$46,700. This is \$3,300 less than the MTI conventional heat exchanger design.

It has been demonstrated that the selection of heat exchanger bed material can have a significant effect on heat exchanger size and cost.

This effect will be considered in the next section where significant size and cost reductions will be shown.

B. EFFECTS OF BED PARTICLE PARAMETERS ON HEAT EXCHANGER DESIGN

1. Material Optimization

The recovery of waste heat is a matter of economics. If the cost of recovery is lower than the value of an equivalent amount of heat recovered, the process is feasible. To realize the full potential of fluidized beds in reducing heat exchanger costs, the effects of bed materials properties on heat exchanger costs must be assessed to allow selection of the optimum material. To assess the impact of bed materials on heat exchanger costs, a comparative analysis has been made on the evaporator costs, as this is the unit which will be immediately affected by material selection. There will be secondary influences which will affect other elements of the total heat exchanger cost, but they are neglected in this study.

This comparison of the impact of bed material on the evaporator cost was made at a reference condition set by particle size (150 microns), identical fluidization indices and the design of specification of the MTI unit. The major items of interest were the costs of heat transfer, flow cross-sectional area, parasitic losses (gas pressure loss) and bed material. The costs were grouped into heat transfer area and flow area costs to produce unit costs. The heat transfer area costs included tubing, manifolds, feed lines and costs of assembly. The reference cost was \$4.10 per length of tube. The flow area costs included the distributor and bed structures. The reference cost was \$15.5 per square foot of flow area. The cost of materials were assessed on an equal volume basis, with unit costs taken from actual costs paid for the AECC test materials. The cost of the parasitic losses were primarily assessed from the gas pressure loss through the bed and distributor plate based on equal bed material volume. An uncertainty factor of 2 was used to account for other losses in the system, and were assessed as a fraction of the recoverable energy. The combined costs of the heat transfer area and flow area were then increased by this amount.

The results of this comparison can be seen in Table V-5. The results were graded from 1 to 8 according to lowest costs. The first column shows glass ballotini as having the lowest cost for the heat transfer area. This is because the glass ballotini had the highest heat transfer coefficient. The highest costs were for the expanded alumina, the material with the lowest heat transfer coefficients. The spread for these costs was 150% between the lowest to the highest. In the second column, the nickel material had the lowest costs with expanded alumina as the highest. This is reasonable since the flow area is dependent primarily on particle density when particle size and the fluidization index are kept constant. Heat transfer coefficients also influence the flow area since the flow area includes an accommodation of the cross-sectional area of the heat transfer tubing, even though this is only a secondary effect. This explains why nickel has a smaller flow area and consequently lower cost than solder with the same density. The total variation in cost between the various materials was by a factor of six. Generally, flow area costs were larger than heat transfer area costs. Expanded alumina had a flow area cost three times as high as the heat exchange area cost due to the low superficial velocity at the specified fluidization index. Nickel and solder both had lower flow area costs than heat transfer area costs because of their high densities and high superficial velocities.

The third column grades the costs of the parasitic losses. This could be interpreted as the loss of recoverable energy. Here, the expanded alumina had the lowest costs because it required the lowest fan power for fluidization. The total cost variation for the various materials was less than 25%. The fourth column rates the bed material costs. These costs proved to be the decisive element in the overall heat exchanger costs. Up to this point, the heat exchanger costs were governed predominantly by the material density, with nickel showing the lowest costs and expanded alumina the highest. But in fluidized bed heat exchanger designs, bed material can greatly influence the overall cost of the unit, not only in initial costs but also in maintenance costs due to material elutriation and attrition. The material costs were assessed on an equal volume basis with the rates based on the actual AECC costs per pound. This may unjustly prejudice some of the materials because of their special nature. For example, the graphite powder used in the test program cost \$40 per pound. The solder powder was also very expensive. On the other hand, silica sand had the lowest material costs, followed closely by the zircon sand.

TABLE V-5

HEAT EXCHANGER COSTS COMPARISONS - AECC TEST DATA

MATERIALS	HEAT TRANSFER AREA COSTS	FLOW AREA COSTS	COSTS OF PARASITIC LOSSES	MATERIAL COSTS	TOTAL COSTS
Glass Ballotini	1	5	3	3	3
Alumina	2	4	4	4	2
Silica Sand	2	6	2	1	4
Zircon Sand	3	3	5	2	1
Nickel	4	1	6	5	5
Graphite	4	7	3	8	8
Solder	5	2	7	7	7
Expanded Alumina	6	8	1	6	6

The results showed that the lowest heat exchanger costs would be obtained with a bed material of zircon sand. The second lowest was a heat exchanger with an alumina bed. This design had a cost 20% higher than that for zircon sand. Designs for glass ballotini and silica sand had costs approximately 30% higher. When material costs were not excessive, particle density was the major influence on heat exchanger costs.

More importantly, this cost comparison showed a \$6,000 reduction in the evaporator costs. This would roughly translate into a 20% reduction in the overall unit cost. This bed material comparison illustrates the value of the material selection in the overall design. The use of sand as a bed material resulted in a marginal cost advantage whereas the use of zircon sand, resulted in a worthwhile advantage for the conversion to a fluidized bed system from a conventional system.

A second cost comparison study was made based on an empirically derived correlation for the fluidized bed heat transfer. In this correlation, particle thermal conductivity was neglected and the pc_p product was taken to the 0.2 power. The design particle diameter was assumed to be 150 microns. The same procedure described earlier in this section for costing was used. The results showed that overall costs agreed with those generated using the actual AECC heat transfer data to $\pm 6\%$.

2. Optimized Fluid Bed Heat Exchanger Conceptual Design

Once the optimum material, zircon sand, had been identified, then the effect of the bed material on the conceptual design could be evaluated. Again, as in Section V.A., the designed specifications for the MTI Binary Rankine Cycle Waste Heat Recovery System were used as the basis of comparison. The evaporator section was designed as a fluidized bed unit with the conventional finned tube designed preheater and superheater as proposed by MTI. The evaporator section was designed as a parallel fluidized bed with three shallow fluidized beds, (see Figure V-3 .) The pressure drop through this unit has increased to 5.7 inches of H_2O . This is due to the increase in density of the zircon sand. The increase in pressure loss would incur a system penalty that would eventually result in an increase in cost, but since this penalty was not included in baseline MTI system, it was not included in the AECC design.

The bed depth was unchanged from that used in the evaporator design with silica sand as a bed material. Placement and selection of finned copper tubing was also unchanged. The difference in heat transfer coefficient was negligible, 2%. This resulted in nearly identical heat exchange surface area requirements. The significant change in the design of the fluidized bed evaporator with zircon sand as the bed material as opposed to silica sand was in the required flow area. The flow area was reduced by 30% to 304 ft². Inclusion of the cross-sectional area of the heat exchanger tubes increased this flow area to 369 ft². This reduced the overall size to 2600 ft³, or a 24% reduction over the conceptual design with a silicon sand fluidized bed.

As discussed in Section V.B, the unit cost for the flow area was \$15.5/ft². With a reduction in flow area of 30%, a substantial savings in overall system cost was realized. This resulted in an overall savings of \$6,300 from the conceptual design with silica sand used as the bed material in the fluidized bed evaporator. Cost estimate details, Table V-6, are identical to those used in Section V.A2. This significant reduction in system cost illustrates the importance of bed material selection in reducing both heat exchanger size and cost.

DESIGN HEAT LOAD = 9.6×10^6 BTU/HR

Figure V-3. AECC Optimized Fluid Bed Waste Heat Steam Generator

Table V-6. ROM COSTING FOR OPTIMIZED FLUIDIZED BED HEAT EXCHANGER

<u>Item</u>	<u>Quantity</u>	<u>Unit Cost</u>	<u>Cost</u>
I. Construction			
A. Site Preparation			490
B. Building			<u>7,900</u>
Subtotal - Construction			8,390
II. Superheater		3,800	3,800
III. Preheater		2,500	2,500
IV. Steam Separating Drum/ Circulating Pump System		13,300	13,300
V. Evaporator			
A. Finned Copper Tubing	1908 ft.	2.65	2,909
B. Manifolds	253 lb.	1.00	253
C. Water Feed Lines	127 lb.	1.00	127
D. Tube Assembly			
1. Drill Manifolds	81	2.00	162
2. Weld Tubes	81	12.00	972
3. Misc. Fittings	-	450.00	450
4. Frame Handling	75 hr.	20.00	1,500
5. Steel Plate	1500 lb.	0.30	450
6. Distributor	16 sht.	240.00	3,840
7. Weld Distributor	54 hr.	25.00	1,350
8. Zircon Sand	4195 lb.	0.105	<u>440</u>
Subtotal Evaporator			<u>12,453</u>
TOTAL			<u><u>40,443</u></u>

3. Comparisons of Heat Exchanger Types

A comparison was made of four heat exchanger systems with respect to shape, overall volume, size, gas-side pressure drop, and ROM cost. The comparisons were baselined on the MTI 9.6×10^6 BTU/hr waste heat boiler of conventional finned-tube superheater, preheater and evaporator design. This system was compared to three similar systems using different designs for the evaporator unit. These designs included an AECC optimized fluidized bed evaporator with zircon sand as the bed material, a bare-tube evaporator and a commercially available fluidized bed used as the evaporator as reported in DOE Contract No. EC-77-C-03-1433, Report No. R78-912942. All systems compared were of a rectangular shape. The total volume of the systems compared varied from 440 ft³ for the nonfluidized, finned tube design of MTI to 6,570 ft³ for the scaled-up commercially available fluidized bed system. The heat transfer area requirements varied by nearly a factor of 6 with the optimized AECC fluid bed heat exchanger showing the lowest requirements. Gas-side pressure loss varied by 160% with the highest losses incurred by the scaled-up, commercially available unit and the lowest losses associated with the nonfluidized bed designs. Results in Table V-7.

The comparisons also indicated that the AECC optimized system with fluid bed evaporator with a bed material of zircon sand had a 19% reduction in cost when compared the same system with a conventional finned-tube evaporator, a 50% reduction in cost when compared to system with a bare-tube evaporator, and a 31% reduction in cost when compared to a system with a scaled-up version of a commercially available fluidized bed evaporator.

Table V-7. HEAT EXCHANGER COMPARISONS

Item	1.Optimized Fluid Bed Heat Exchanger (AECC)	2.Conventional Bare Tube Heat Exchanger	3.Augmented Surface Heat Exchanger	4.Commercially Available Fluidized Bed Heat Exchanger
Shape	Rectangular	Rectangular	Rectangular	Rectangular
Volume (ft ³)	2,600	1,620	440	6,570
Heat Exchange Area (ft ²)	1,980	10,860	10,200	7,100
Gas-Side Pressure Drop (in.H ₂ O)	5.7	4.2	4.2	6.6
Estimated Cost	\$40,400	\$80,000	\$50,000	\$58,400

NOTE: Comparisons were baselined for a 9.6×10^6 BTU/hr. Waste Heat Boiler.

1. Optimized Fluid Bed Design.
2. Conventional Non-fluidized bed design with a bare tube evaporator (extrapolated from Design 3).
3. Augmented non-fluidized design with a finned-tube evaporator (Rhinehart)..
4. Design scaled-up version of a commercially available fluidized bed evaporator
DOE Contract No. EC-77-C-03-1433, Report No. R78-912942.

VI. CONCLUSIONS AND RECOMMENDATIONS

On the basis of the experimental and analytical work conducted under this contract, the following conclusions can be made for fluid bed heat exchangers:

1. Maximum heat transfer coefficients are proportional to the -0.5 power of the particle diameter of the bed material.
2. Maximum heat transfer coefficients are proportional to the 0.2 power of the particle specific heat of the bed material.
3. Maximum heat transfer coefficients are proportional to the 0.2 power of the particle density of the bed material.
4. Maximum heat transfer coefficients have a negligible dependence on the particle thermal conductivity of the bed material.
5. Bed material selection significantly affects heat exchanger costs. Based on optimization analysis considering the bed materials tested, the use of zircon sand resulted in the lowest cost heat exchanger.
6. The optimized fluid bed heat exchanger has an estimated cost 19% below a fin augmented tubular heat exchanger; 31% below a commercial design fluid bed heat exchanger; and 50% below a conventional plain tube heat exchanger. The comparisons being made for a 9.6×10^6 BTU/hr waste heat boiler.

Results of this contract established that bed material properties can significantly affect fluidized bed heat exchanger cost and performance. The following suggestions are made for future investigations.

1. Prospective bed materials suitable for heat exchangers should be surveyed and rated according to their projected heat exchanger

performance in light of the relationships developed in this work.

2. Heat exchanger requirements should be classified into general categories with potential bed materials identified for each category.

3. Fluidized bed heat exchangers should be evaluated for advantages in addition to heat transfer duty such as filtration and fouling prevention.

4. Methods should be developed to increase gas throughput without loss of heat transfer enhancement or elutriation.

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V. G. Ainshtein and N. I. Gel'perin, "Heat Transfer Between a Fluidized Bed and a Surface", International Chemical Engineering, January 1966.

The authors are attempting to compare a number of correlation equations for the determination of ' $h_{\max}$ ' using an arbitrary model system. Their model used glass spheres with $\rho = 2.66$, $\rho_{\text{BULK}} = 1.66$, $C_p = 0.19 \frac{\text{BTU}}{\text{lb}^\circ\text{F}}$ and air at 68°F . The bed height of $H_0 = 1.31$ ft. and a bed diameter $D = 0.66$ ft. were used. The maximum heat transfer coefficient at the optimum velocity, heat exchange with an external wall and an in-bed surface on the ascending and descending branch of the h vs. V curve were sought in this comparison study.

Determination of ' $h_{\max}$ ' was best correlated by the following works to within ± 10 -15% from a field of eleven correlations:

AUTHORS	CORRELATION	REMARKS
Martyushin and Varygin	$Nu_{\max} = 0.86 Ar^{0.2}$	s is the distance between the axes of tubes with a diameter of d_t
Baerg et al.	$\alpha_{\max} = \frac{2.395 \log 7.05 \cdot 10^{-3} \rho p}{d}$	
Zabrodskii	$\alpha_{\max} = 33.7 \rho p^{0.2} Kg \cdot d^{-0.36}$	
Gel'perin et al.	$Nu_{\max} = 0.64 Ar^{0.22} \frac{s}{d_t}^{0.09}$	

The determination of the optimum velocity could not be determined from the 8 correlations studied. Results differed by almost an order of magnitude. This may be due to the use of average velocities for the system whereas ' $h_{\max}$ ' may be taken at a local point.

The authors also state the "the hydrodynamic conditions may ensure ' $h_{\max}$ ' at a given point in the bed but not necessarily at others where the value of ' h ' has yet to achieve its maximum". Also, since the average velocity is not the same as the local value due the dependence of the locations of the heat transfer surface, construction and distributing grid, the average velocity can in no way accurately characterize the heat transfer process.

These deviations in the correlations of V_{opt} appear in comparison of heat transfer rate for an external wall and an internal heating element, or for various tubes of a vertical bundle (Gelperin showed significant variation in the ascending curve of 'h' vs V for different tube arrangements even though ' h_{max} ' for different tubes were very close). Graphical comparisons were made for these cases with results differing by nearly two orders of magnitude. Probable cause may be:

1. Experimental conditions differ considerably including operating conditions of pressure and temperature, physical properties of the solids and fluidizing media, the geometric characteristics of the bed, experimental technique and calculation procedures.
2. Extending specific experimentation to generalized correlations beyond the limits of the original work.
3. Using generalized values of the void fraction which may distort the general picture of the data.

The authors suggest better results could be obtained if some uniformity in the presentation of the data could be adopted. Also, new dimensionless thermal quantities in the 'packet theory' would be of interest rather than use of traditional quantities from classical problems of heat transfer and hydrodynamics in tubes, channels, etc.

Neither the Reynolds number nor the fluidization number can be used because the fluidization number is in the order of 80 for systems with small values of Ar and about 8-10 for ones with a large Ar.

B.M.A. Al Ali and J. Broughton, "Shallow Fluidized Bed Heat Transfer", Applied Energy (3), 1977, pp. 101-113.

Testing was done in a rectangular bed 600mm x 150mm with a pierced plate distributor. Heated air was used to get bed temperatures from 75-150°C (167-302°F).

A movable horizontal bare copper tube and a 14.3mm dia copper sphere were used to measure heat flux. Materials used were:

BED MATERIAL	SIZE μm	DENSITY	SHAPE FACTOR
sand	151	2.709	1.19
	253	2.684	1.13
	345	2.662	1.15
blown alumina	1100	1.777	1.00

Results in a 50mm bed using the copper tube devise were (h in $\frac{\text{Btu}}{\text{hr-ft}^2-\text{°F}}$)

134 μm sand h=92.5 @ 57mm above the plate
 253 μm sand h=70 @ 65ma above the plate
 345 μm sand h=66 @ 27mm above the plate
 1100 μm alumina h=41 @ 15mm above the plate

Results of 'h' with increasing fluidizing velocity were at 25mm above the plate

134 μm sand h=84 @ 1.23 ft/sec
 253 μm sand h=70 @ 1.64 ft/sec
 345 μm sand h=66 @ 1.97 ft/sec
 1100 μm alumina h=43 @ 3.6 ft/sec

Best data correlation is

$$\text{Nu}_m = 0.562 \text{ Ar}^{0.255}$$

with 20% data scatter. U_{op} was in the range $0.28 U_{op} - U_{mf}$ 0.44 M/S
 (1.44 ft/sec) or $U_{op} - U_{mf} = 0.35$ M/S (1.15 ft/sec).

Inside film coefficient of heat transfer was used, calibrated against water flow rate, since the water flow was in the transitional regime, in order to obtain a reasonable temperature rise.

Ali compares shallow and deep beds. Defines shallow beds as less than 1 ft. with an aspect ratio less than 0.5, whereas deep beds are generally more than 3 feet.

	<u>Deep-beds</u>	<u>Shallow-beds</u>
Bed pressure drop	+ 10kN/m ²	- 5kN/m ²
Plate pressure drop	+ 7-5kN/m ²	- 0.5kN/m ²
Carryover/Transport disengaging height	+ 1 m	1 m usually adequate
Bed expansion (%)	20-40	50-250
Heat transfer coefficient to tubes immersed	26-141 Btu/ft MF (150-800(W/m ² K))	150-800(W/m ² K)
Multistage	Seldom practical	Fairly easy
Bubble sizes	Order 1 m	Less than 0.1m
Amenity to scale-up	Difficult due to slugs in small beds	Usually direct
Slugging	Common	Never
Entry effects	Usually small	Often important
Exit effects	Carryover/T.D. Ht.	Several effects
Temperature uniformity	Usually uniform	Gradients often occur

Ali states a lean phase cloud forms above the bed, effects both the radiation and reaction processes, i.e., increases 'h'. Also 'h' reduces in the dense phase as the depth reduces. Form of U vs h curve remains the same with U_{op} roughly constant independent of depth and tube position but h_{max} does vary with depth and tube position.

The large alumina particles did not show any effect to depth or tube location. Heat transfer surfaces near the distributor result in an increased 'h' for systems using heated fluids and in a system introducing cold fluids 'h' is reduced. This confirmed by Zabrodsky and Yuditskii (1966), Atkinson (1974), and Pillar (1976).

Gas-blanketing, results from using fine particles, can reduce heat transfer and jets can improve it. Effect is less than 20%.

B. R. Andeen, L. R. Glicksman and R. Bowman, "Heat Transfer from Flattened Horizontal Tubes", Fluidization, Cambridge University Press, 1978, pp. 345-350.

Horizontal tubes in fluidized beds have been observed to have stagnant particles residing on the top surface and a region of low voidage directly below the tube. The overall heat transfer to the tubes should be increased if the effects of these two regions are minimized. The authors will evaluate this postulate by experimentally measuring the film coefficients with flatten-sided tubes that have a smaller percentage of their surface area at the top and bottom.

TEST SET-UP. The bed is 2-foot square with a distributor plate of 60 mesh brass wire cloth backed with one to two layers of cotton cloth. The pressure drop ranged from 30 to 70% of the total bed pressure drop. A heated tube measured film coefficients. The flattened tubes were 3.18cm x 1.27cm in cross-section (equal to a 2.54cm round tube). The tubes were arranged in a single row on 5.08cm centers and 1.1cm above the distributor plate. Operating temperature and static bed height not given. Three sizes of Ottawa sand used with mean diameters of 360, 510 and 710 microns. Round tubes (2.5cm dia.) were tested for comparison.

Results.

Dp	h(Btu/hr-ft ² -°F)
360 μm	53
510 μm	44
710 μm	36

The flattened tubes have a 1/3 improvement in average heat transfer coefficient over round tubes. Round tubes exhibit a very narrow peak in the heat transfer coefficient while flattened tubes decreases slightly to a relatively constant value. Peak heat transfer for the round tube occurs at about the same superficial velocity as the flattened tube. The least square fit of the data results in

$$\frac{hd_p}{K_g(1-\epsilon)} = 1240 \left[\frac{Vd_p\mu}{p_p g d_p^3} \right]^{0.37} \left[\frac{dp}{dt} \right]^{.63} Pr^{0.3}$$

B. R. Andeen and L. R. Glicksman "Heat Transfer to Horizontal Tubes in Shallow Fluidized Beds", ASME 76-HT-67.

Authors aimed to understand heat transfer to bare tubes in shallow fluidized beds to evaluate their usefulness in dry cooling towers.

The bed was two feet square of clear plastic. Distribution plate consisted of a 60 mesh brass wire cloth backed with one to two layers of cotton cloth. Measurements were made with a hot wire probe. An instrumented, heated 3/4" aluminum tube was used to measure 'h'. Material used was:

	size range	$\bar{d}_p$
Sand	840 - 590 μm	710
	590 - 420 μm	510
	420 - 300 μm	360

Initial data was taken with two tubes in a triangular stagger. Tubes had a pitch of two for all tests. Superficial velocities ranged from 5 to 13 ft/sec.

Results were about 20-30 $\frac{\text{Btu}}{\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}}$. A 50-50 mix of coarse and light sands resulted in same 'h' as light sand alone. When fluidized a mixture of light and coarse particles, the particles segregated with fines above coarse particles.

Best fit correlation used was:

$$\frac{hd_t/Kg}{Pr^{1/3}} = 900 (1-\epsilon) \left[\frac{6d_t\rho_p}{\rho g \mu} \cdot \frac{\mu^2}{d_p^3 \rho_p^2 g} \right]^{.326}$$

They suggest tubes rather than the gross bed on distribution geometry must have the predominant influence on the bed fluid dynamics near the tubes. They do not recommend their correlation for vertical tubes/surfaces or with fluidized gases other than air at standard conditions.

A. P. Baskakov, B. V. Berg, O. K. Vitt, N. F. Filippovsky, V. A. Kirakosyan, J. M. Goldobin and V. K. Maskaev, "Heat Transfer to Objects Immersed in Fluidized Beds", Powder Technology, 8 (1973), pp. 273-282.

The authors present a modified packet model for heat transfer between a gas-fluidized bed and an immersed object to allow for the effective gap resistance between the packet of particles and surface (contact thermal resistance R_K), and for the heat transferred directly by convection through the gas filtering through the particles and between the particles and the immersed object.

The authors also experimentally determined packet contact time, τ , and the fraction of time that the bubbles shrouded the transfer surface, f_0 . Empirical correlations were presented for each. They found particle density and gas dynamic viscosity had little apparent effect, except for their effect on U_{mf} , and the variation in mean probe and bubble contact time with gas flow rate was independent of particle diameter and shape. The contact resistance R_K reduced almost linearly with reduction in particle diameter, but did not tend to zero in the limit of small particles. Thus, providing further evidence for the existence of a gap between the bed and the surface.

The interparticle gas convective heat transfer component given by Baskakov and Suprun was studied. The relationship given suggests that the gas convective component of heat transfer, h_{CONV} , is proportional to $d^{.038}$ whereas the conductive component reduces with increasing particle diameter because of the increases in R_K and τ . Thus, the convective contribution toward the maximum heat transfer coefficient, h_{MAX} , is found to increase with increasing particle size, being approximately 10, 15, 30, 60 and 90% respectively for corundum and chamotte particles having average diameters of 160, 320, 500, 2500 and 4000 microns.

The work done with the effect of shape of the immersed body indicated that vertical objects produce higher heat transfer rates than do horizontal objects. The effect of particle moisture increases the effective gas thermal conductivity 3%, but water sprayed into the bed affects the bed behavior directly by aiding the discharge of electrostatic charge. Also, the radiant

heat transfer effects increases with the increase in the temperature of the radiative surface component reached 20-30% for a surface temperature of 1340°F and a bed temperature of 1560°F. The larger the particle diameter, the larger is the fractional quantity of heat transferred by radiation, although the absolute quantity of heat transferred by radiation is independent of particle diameter and of the fluidizing velocity.

A.P. Baskakov and V.M. Suprun, "Determining the Convective Component of the Heat Transfer Coefficient to a Gas in a Fluidized Bed", translated from Khimicheskoe i Neftyanoe Mashinostroenie, No. 3, pp 20-21, March 1971.

This work was to determine the convective component of the heat transfer coefficient to a gas in a bed and to refine the empirical functions on the basis of experimental values of mass transfer coefficients from a vertical naphthalene surface to a fluidized bed.

This study did show that the convection component of the heat transfer coefficient h_{conv} , was less than 8 Btu/hr-ft²-°F for 160 μm corundum particles, 9 Btu/hr-ft²-°F for 320 μm corundum particles and 10 Btu/hr-ft²-°F for 500 μm corundum particles.

They conclude that an increase in average particle diameter from 160 μm to 2.5 mm results in a lower 'h' since the reduction in the conductive heat transfer coefficient exceeds the increase in h_{conv} . When diameters are increased from 2.5 mm to 4 mm, 'h' increase due to a reversal of the trend noted at smaller particle sizes.

Based on their data, they derived a formula for the convective component of the heat transfer coefficient

$$\text{Nu}_{\text{conv}} = 0.0175 \text{ Ar}^{0.46} \text{ Pr}^{0.33} \left(\frac{w}{w_o} \right)^{0.3}$$

J. C. Chen, "Heat Transfer to Tubes in Fluidized Beds", ASME paper 76-HT-75, August 1976.

The heat transfer of plain and finned tubes in fluidized beds are examined. Compares experimentally obtained data with correlations for vertical and horizontal tubes. For the horizontal tube experiment he uses 240 μm glass spheres fluidized with room temperature air in a rectangular bed, 15 cm x 30 cm, with a packed bed height of 20 cm. The tube is 2.5 cm in dia. Very little agreement was obtained when compared with the correlation of Genetti, et.al, (1971) or with Vreedenberg (1958). Chen presents a modified "packet model" that successfully predicts 'h' if average residence times are known:

$$h = 2 \left[\frac{k_E \rho_E C_{PE}}{\pi \bar{\theta}_B} \right]^{1/2} (1-f_0)$$

Where θ_B is residence times for the bubble phase

Detailed heat transfer measurements around the tube indicate the renewal mechanism only occur on the sides, at the bottom of the tube a convective transport mechanism should be used, and at the top heat conduction through the packed particles should be used. The main point to remember is that the heat transfer mechanism at the surface can be of different character at different positions around the tube and for different operating conditions.

Study of finned tubes indicate an increase in heat transfer of up to ~300%.

John C. Chen, James G. Withers, "An Experimental Study of Heat Transfer from Plain and Finned Tubes in Fluidized Beds", 15th National Heat Transfer Conference, AIChE Paper No. 34, 12 August 1975.

Heat transfer coefficients on the surface of two plain and four finned copper tubes located vertically in fluidized beds were measured experimentally. It was expected that the results would give indications of optimum fin characteristics to provide maximum heat transfer capabilities.

TEST SET-UP. The cylindrical bed vessel was 5-1/2 inch I.D. and four foot high. The distributor was a 1/8" thick sintered bronze plate with 10 micron porosity. Electric cartridge heaters were the heat sources. Wall temperatures measured by sheathed T/C embedded in wall. Packed bed height was 12 inches giving fluidizing depth of 12 to 30 inches. Three sizes of glass beads were used. Operating bed temperatures were not given.

Size	Avg dp	G(#/hr.ft ²)
small	127 μ m	42
medium	254 μ m	74
large	609 μ m	216

RESULTS: Part of the data, maximum 'h', is shown in the table below.

Dp (m)	Tube	h	Tube	h	Tube	2
127	.5 OD plain	110				
127	.865 OD plain	105	9FPI, 1/16" FH	110	19FPI, 1/16" FH	78
254	plain	95	9FPI, 1/16" FH	104	19FPI, 1/16" FH	78
609	plain	65	9FPI, 1/16" FH	68	19FPI, 1/16" FH	56

*h is given in Btu/hr-ft²-°F based on total heat transfer area.

Results show that the ability of the fluidized particles to penetrate into the root region of fin tubes is affected by both fin height and fin count. When the fin tube performance, ratio of the effective heat transfer coefficient compared to a plain tube, is near at or above unity, one may expect the finned tube to

provide higher heat transfer duty per unit length than a plain tube. The authors used a capacity function (hA/h_pA_p) to determine optimum fin geometry (W/d_p) at a particular air mass flux where W is the fin gap. Their results indicate that for a given fin geometry (W/d_p), the capacity function increases to a maximum then decreases. Increases in heat transfer capability were from 20% to 190%.

Benjamin T. Chung, L. T. Fan and C. L. Hwang, "A Model of Heat Transfer in Fluidized Beds", Journal of Heat Transfer, February 1972.

An expression for estimating the heat transfer coefficient in a fluidized bed has been developed theoretically on the surface renewal and penetration concept, a modification of Ziegler's work. The proposed expression is

$$h = \frac{13.5 k_g \left[b^2 + 0.4 \frac{k_g}{k_p} \left(b^2 + 5b - \frac{d_p^2}{2\alpha\tau} \right) \right]}{d_p \left(b + 1.6 \frac{k_g}{k_p} \right)}$$

$$\text{where } b = \frac{d_p}{2\alpha\tau} \coth \frac{d_p}{2\alpha\tau} - 1$$

α = thermal diffusivity of fluidized particles $\left(\frac{K}{\rho C_p} \right)$

τ = mean residence time

This expression between walls and beds agrees with the experimental results of Botterill (1964). The thermal conductivity of solid particles has little effect on the transfer rate if the value of $d_p/\sqrt{2\alpha\tau}$ is less than unity. The only condition under which the conductivity of particles may have influence on the heat transfer coefficient is when both $d_p/\sqrt{2\alpha\tau}$ and k_p/k_g are large. The model predicts a maximum Nusselt number of 13.5 for any gas fluidized system. The Nusselt number is defined as $Nu = \frac{hd_p}{k_g}$.

N. I. Gel'perin, V. G. Ainshtein, L. A. Korotyanskaya, "Heat Transfer Between a Fluidized Bed and Staggered Bundles of Horizontal Tubes", International Chemical Engineering, Vol. 9, No. 1, Jan. 1969, pp. 137-142.

The authors propose to use the Baskakov modified packet theory to compare to experimental data on heat transfer between a fluidized bed and staggered bundles of tubes.

TEST SET-UP. The bed is 15"x15" with a perforated grid. Three sizes of quartz sand were used with $d_e = 160, 263$ and 350 microns with $U/U_{mf} = 1.1$ (0.11, 0.26 and 0.43 ft/sec respectively) to $U/U_{mf} = 8$. Fluidizing agent was air. Static bed height was 1.25 to 1.44 feet. The tubes (heaters and thermocouples) were 0.79 in. dia. The bundles were 5.9" above the grid and the bundles had 2, 5 and 9 rows.

RESULTS. With straight-line bundles of horizontal pipes resulted in a constaining effect on the bed. The heat transfer coefficient depended noticeably on the horizontal spacing, s_h , whereas the influence of the vertical spacing s_v , up to $s_v = 0$, was not great. In staggered bundles, small values of s_v will also have an influence on the heat transfer. The decrease in the heat transfer coefficient on transition to tighter bundles is explained by the decrease in the intensity of movement of particles in the bed and the decrease of its effective thermal conductivity. The experimental data on the maximum heat transfer coefficients for 5 and 9 row bundles were presented in the form of an emperical formula which reflects the influence of the vertical and horizontal spacings on heat transfer

$$Nu_{max} = 0.74 Ar^{0.22} \left[1 - \frac{d_s}{s_h} \left(1 + \frac{d_s}{s_y + d_s} \right) \right]^{0.25}$$

$$\text{where } Nu_{max} = \frac{h_{max} d_e \phi}{K_g}$$

$$Ar = \frac{g d_e^3 \phi^3}{2} \cdot \frac{\rho_s - \rho_g}{\rho_g}$$

V. N. Korolev and N. I. Syromyatnikov, "Heat Transfer from a Surface with Artificial Roughness to a Fluidized Bed", Journal of Engineering Physics, Vol.28, 6 June 1975, pp. 698-700

Experiments used four 55x55mm heat emitting plates with artificial roughness:

Plate	Height	Spacing
1	0.5 μ m	
2	2 μ m	60 μ m
3	10 μ m	100 μ m
4	400 μ m	500 μ m

Particles used were:

corundum	120 μ m
chamotte	320 μ m
	490 μ m
spherical polystyrene	720 μ m

The bed is a column 150mm in dia. The fluidizing agent is air. The calorimeter was fastened at a distance of 30mm from the perforated gas-distributing grill. The height of the material was 120mm. Measurements were taken with copper-Constantan thermocouples. The heat transfer coefficient is given per unit of smooth surface.

If the height of the roughness projections and the distance between them are greater than the particle size then higher heat-exchange coefficients are obtained than for a smooth surface. This increase occurs because of the increase in actual surface. If actual surface is taken as the calculated surface, the coefficient would be less due to the increase in the effective gas layer between the surface and the first layer of particles, and the retarding effect of the roughness on the particles on the mobility of the particles. But the increase in surface area exceeds the reduction in the heat-exchange intensity and the overall heat flux increases considerably. For large particles (720 μ m) the effect of roughness was not detected.

Some results were

		Btu/hr-ft ² -°F		
MATERIAL	SIZE	'h' (Mft ² °F)	U _f (ft/sec)	plate
Corundum	120 m	141	.66	1
		132	1.07	2
		122	1.07	3
		202	>1.31	4
Chamotte	320	92	.98	1
		86	1.3	2
		72	1.3	3
		110-132	1.3	4

H. K. Lese and R. I. Kermode, "Heat Transfer from a Horizontal Tube to a Fluidized Bed in the Presence of Unheated Tubes", The Canadian Journal of Chemical Engineering, Vol. 50, February 1972.

This experiment studied the effect of the number of tubes, their size, configuration and type of fluidized materials. Certain configurations caused as much as a 56% reduction in heat transfer over the most favorable which was 35 Btu/hr-ft²-°F. Found the top of the tube was covered by stagnant solids, the bottom partially by a gas film, and the sides by a small region of vigorously agitated solids. Most of the heat transfer occurred in the side region.

The bed was a rectangular box 6-3/8 in.x6-3/8 in. of 1/16 in. mild steel sheet and insulated with one inch of Kaylo block. A porous ceramic plate one inch thick was used as the distributor and base plate. The bed was heated by a single horizontally mounted tube. It was a copper tube cut axially in 4 full length pieces at 90° intervals. A Chromalox CDE cartridge heater rated at 450 watts fit inside the tube. Copper-Constantan T/C were placed at each cut and a 0.003 inch brass shim stock was wrapped around the entire assembly.

Materials were fluidized with air. Weighted harmonic mean particle diameters were used.

MATERIAL	dp	SHAPE
glass beads	295 μ m	spherical
tabular alumina	76 μ m	angular, almost square or rectangular; no fines
silicon carbide	99 μ m	uneven surfaces, large amounts of fines

They experienced poor heat transfer when unheated tubes were placed close to the heated tube. This disappeared at a separation distance of 1/2 inch. Their closely packed array of unheated tubes (separations of 1/8 inch) for both square and triangular pitch experienced good heat transfer using silicon carbide. Results with glass beads were slightly lower.

Only a brief series of runs were made to study the effect of bed material. Maximum 'h' for glass beads was $45 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, at $U/U_{mf}=5$; silicon carbide had greater data scatter and a maximum was not evident.

Correlation with Vreedenberg's correlation was made. Suggested data of silicon carbide and alumina was below the predicted values due to their non-spherical shape which lowered mobility.

Krishna K. Pillai, "Heat Transfer to a Sphere Immersed in a Shallow Fluidized Bed," Letters in Heat and Mass Transfer, Vol. 3, pp 131-146, 1976.

Pillai proposes a correlation for heat transfer in a shallow fluidized bed at temperatures of 1100°C. Four hundred data points are used to within $\pm 10\%$ using

$$Nu_{max} = 0.365 \theta^{.82} Ar^{0.22} \quad \text{where } \theta = \frac{\text{abs. bed temp } ^\circ K}{273}$$

Bed depth is less than 100 mm. Bed heating was accomplished by premixed gas combustion of propane and air. But combustion was stopped during testing. Two beds were used:

1 mild steel, 100 mm dia up to 700°C and

1 stainless steel, 140 mm dia up to 1100°C.

Both used a 25 mm ceramic distributor with 20 mm of asbestos insulation.

Probes to measure heat transfer were spherical with chromel/alumel thermocouples placed at the center of a copper probe 6 mm in dia. (most frequently used). Thermocouples were sheathed in 1 mm stainless steel.

Air flow varied from $U_{mf} >$ to $10 U_{mf}$ (~ 1.5 m/s). All tests were done at 50 mm static bed height with probe at 25 mm above the distributor plate. Since bed heating occurred above the bed, bed temperature decreased with time. But author states testing time was short (1 min), therefore time was not a dominate factor.

Material	dp. (μm)	ρ	$C_p (\frac{J}{kgK})$	$K (w/mk)$	Sphericity	ϵ_{mf}
Silica sand	243-755	2.63	845	1.87	0.81	0.43
Silicon carbide	255-787	4.615	900	1.95	0.79	0.43
Zircon sand	212-513	3.583	550	16.2	0.73	0.43

One data curve showed $h_{max} = 110 \frac{Btu}{hr/ft^2 ^\circ F} @ 3 \frac{U_f - U_{mf}}{U_{mf}}$ for

243 μm silica sand with a static bed height of 50 mm @ $T_B = 550^\circ C$ (1022°F).

He suggests heat transfer coefficients are higher in shallow beds due to greater solids motion and shorter residence times. Pillai states a 10% reduction in "h" when the bed increases from 20 to 80 mm.

Pillai does not mention radiative heat transfer which could contribute to the increased 'h' from both the bed and the metal containers. He used air at 20°C (68°F) which results in low 'h's' near the distributor plate.

S. J. Priebe and W. E. Genetti, "Heat Transfer from a Horizontal Bundle of Extended Surface Tubes to an Air Fluidized Bed", AIChE Symposium Series, Vol 73, 1977, pp 38-43.

Heat transfer coefficients were measured from a horizontal bundle of finned tubes to an air fluidized bed. Two types of tubes were used: (1) discontinuous finned tubes to study heat flux and fin spacing, and (2) spined tubes to study spine height, spines per turn, and spine material.

TEST SET-UP. A clear plexiglass rectangular column 8 ft. tall with inside cross-section of 14"x6.5" was used. The distributor plate consisted of 2 layers of 200 mesh stainless steel wire cloth sandwiched between 2 perforated steel plates. Seven tubes were used in each bundle and heated electrically with Calrod cartridge heaters. An I/C T/C was attached at midpoint of each tube and bed temperatures were measured at 3 locations with T/C.

Results of discontinuous finned tubes. Results indicate that the heat transfer coefficient decreases when the particle diameter approaches the fin spacing ($\frac{D_p}{S} > 0.1$). Final correlation is:

$$Nu_p = \frac{(1013 D_p - 3.8) (Q/A_t)^{0.24} \frac{D_p}{S}^{-(0.55+5.9D_p)}}{\left[1 + \frac{8.19 + 5.88 S}{(G/G_{mf})^{0.164}} \right]^2}$$

where $\frac{D_p}{S}$ is the ratio of particle diameter to fin spacing; D_p & S are in inches

Q/A_t is in Btu/hr.-sq. ft.

All data falls within $\pm 12.5\%$. The correlation accounts for the thermal conductivity of the tube.

Results of spinned tubes: 16 different tube bundles were used. Coefficients obtained for copper spines are greater than for stainless steel. There is little difference in the coefficient with a large number of spines per turn, but the increased area will yield higher heat transfer rates. Silica sand and glass beads were the bed material used, but the particle size is not given. Air

temperature and height of bed are not given. A noticeable feature of all the plots is the lack of a maximum coefficient. No clear trends with respect to spine height. Final correlation is:

$$Nu_p = \frac{7.34 \left(\frac{A_t'}{L} \right)^{-0.311}}{\left[1 + \frac{2.1 - 0.0081 D_f^{7.24}}{Re_p^{1.72}} \right]^2}$$

where A_t' is the total area unit length

L is the spine length

All the data fall within $\pm 12.5\%$. The correlation accounts for tube thermal conductivity.

V. W. Selzer and W. J. Thomson, "Fluidized-Bed Heat Transfer - The Packet Theory Revisited", AIChE Symposium Series, Vol. 73, 1977, pg. 29-37.

The major portion of this paper deals with the simultaneous measurements of bubble activity and the instantaneous temperature fluctuations for fluidized bed heat transfer from immersed tubes.

Test Set-Up. A 2-D rectangular bed made from 1/4 in. thick plexiglass was used. It is 3.5 ft. tall x 1 ft. long x 4 cm thick. 6 positions are available at the bed axis for a vertical tube. The distributor plate has 2 layers of perforated stainless steel plates with slits 31 mils wide x 9 mils high. The test chamber had a calming section. Static bed height was 20 inches. The heated tube was a 1.5 Kw electric Vulcan cartridge heater surrounded by a 38.6 mil thick brass sheath. Four I/C T/C's were placed flush on this tube.

Results:

BED MATERIAL	Dp	U _{mf}	VELOCITY RANGE	H _{max} (BTU/hr ft ² -°F)	U _{opt} (cm/sec)
Glass	230 μ m	5.7 cm/sec	1-6 U _{mf}	113	U-U _{mf} =22.8
Beads	470 μ m	33 cm/sec	1-2 U _{mf}	60	U-U _{mf} =24.8

For the small beads, the heat transfer coefficient were 12% higher for the horizontal tube at 30 cm above the plate than at 15 cm. Vertical tube data compared with horizontal tube at 30 cm above the plate up to $U/U_{mf}=3$; after which they were 10-15% lower.

Attempts were made to predict the experimentally determined values of the localized heat transfer coefficients by utilizing various modifications of the packet theory. One variation gave reasonable results for the small beads. But there was a lack of agreement with the larger particles. Comparison of results were made with correlations by others. The best results for the horizontal tube for both particle sizes were obtained by Botterill's single particle model as given by Gutfinger and Abuaf -

$$\bar{h}_1 = \frac{170dp f_{BC}}{\bar{\tau}_{BC}} \left[1 - \exp \frac{-476 \bar{\tau}_{BC}}{dp^2} \right]$$

where $\bar{h}_1$ = local time - averaged heat transfer coefficient

Toshio Yasutomi and Teiji Yokota, "Heat Transfer from the Surface of a Body Fixed in a Fluidized Bed", Heat Transfer-Japanese Research, Jul-Sep 1977, Vol 6, No. 3, p. 26-35.

A model is proposed for the mechanism of heat transfer from the surface of a body fixed in a fluidized bed. The heat transfer mechanism is regarded as a transient heat conduction through composite solids consisting of two layers. The first layer contacts the surface and has a thickness of half the diameter of a solid particle with a large void fraction. The second layer with a void fraction at minimum fluidization follows the first layer and its thickness is infinity. Total times of bubbles rising along the body were measured to estimate contact times.

TEST SET-UP. The fluidized bed consists of cylindrical towers which are 104mm, 140mm and 300mm in dia. The heights are 500mm. The distributor is a perforated plate with a square pitch of 10mm. The heat transfer tube is a heater held vertically in the center of the bed. The bed temperature varied from 68°F to 122°F. Materials used were:

MATERIAL	dp	ρ	Cp	ϵ	Umf (ft/sec)
sand	246	2.7	0.191	.435	.26
sand	351	2.7	0.191	.450	.32
sand	700	2.7	0.191	.440	1.08
sand	495	2.7	0.191	.480	.58
copper	174	8.96	0.092	.410	.31
copper	256	8.96	0.092	.425	.60
copper	487	8.96	0.092	.425	1.49
glass	443	2.5	0.184	.405	.51
glass	852	2.5	0.184	.400	1.44

RESULTS. Maximum heat transfer coefficients are proportional to -0.5 power of particle diameters. Differences in h_{MAX} between sand and glass are within 10%, whereas copper particles are 25% higher than sand. No affect due to bed diameter. The effect of fluid heat capacity is negligible. It is not necessary to take account of contact resistance for $d_p < .3mm$.

MATERIAL	Dp	h_{MAX}
sand	246	82
sand	351	51
copper	174	123
glass	443	59

S. S. Zabrodsky, Yu. G. Epanov and D. M. Galershtein, "On the Dependence of Fluidized Bed-Wall Heat Transfer Coefficients on the Thermal Conductivity and Volumetric Heat Capacity of the Particles", Fluidization, Cambridge University Press, 1978, pp. 362-370.

Zabrodsky is attempting to explain the contradictions of material thermophysical properties on fluidized bed heat transfer.

For fine particles or larger particles, effect of k_s does not show.

He used the expression for the conductive wall-fluidized bed heat transfer component to analyze the effect of $\rho_s C_s$

$$h_{cd} = 0.807 d_p C_s \rho_s \frac{(1-\epsilon)^{2/3}}{t_{ex}} [1-e^{-n}]$$

$$\text{where } n = \frac{1.5 k_f t_{ex}}{\rho_f C_s \rho_s d_p}$$

He states for larger particles, +1mm, $\rho_s C_s$ effect disappears. These results were for velocities at $\sim U_{mf}$ ($\frac{H}{H_0} = 1.1$) with variation made with increasing contact times and increasing $C_s \rho_s$. He states larger particles result in slightly larger 'h' but this is due to the increase in the convective 'h' component. Zabrodsky states the part of $C_s \rho_s$ will decrease with decreasing 'hcd' component.

Victor Zakkay, Sergio Panunzio and Andrea Brentan, "Fluid Dynamics of a Fluidized Bed Packed with Heat Exchangers", Interim Report for Feb.-May 1976, E(49-18)-2256.

This report reviews contract work from Feb.- Apr. 1976. The initial task was a review of the recent literature. They report a wide scatter for the various configurations with no single correlation valid. They report that the average density of the particles is a major parameter in all heat exchanger processes. High performance of fluidized beds have been associated with their high density when compared with the density of the gas alone.

Temperature effects the value of 'h', other than due to radiant heat transfer, by decreasing the density of the gas and lowering the dynamic pressure and drag on the particles, and by increasing the kinematic viscosity and gas conductivity.

Pressure effects the value of 'h' by increasing air density and by contributions of the gas convective component of heat transfer. Also, minimum fluidizing velocities occur sooner due to the higher density.

Particle properties of heat capacity (mass x solid specific heat), size, size distribution, and surface state contribute to heat transfer values. Higher heat capacities cause a small temperature gradient among particles adjacent to the heat exchange surface which results in a higher average temperature gradient between the particles and surfaces. Therefore, moving particles have less of a temperature drop. A low solid density results in a higher bed voidage and therefore results in a lower ' h_{max} '.

The authors state that the configuration of the in-bed cooling tubes is critical in the design of an industrial fluidized bed. They suggest that a suitable radial distribution of the gas velocity would improve the overall heat exchange performance. From recent literature, they found that objects placed between the distributor plate and the exchange surface can reduce the value of 'h' by as much as 80%. This is because the upstream body shades the upper zone of the bed and reduces the temperature of the rising particles. They also report that vertical spacing in an array effects the 'h' slightly (3.5%) and that the horizontal spacing can affect the 'h' by 20% due to poor particle circulation.

E. N. Ziegler, L. B. Koppel, and W. T. Brazelton, "Effects of Solid Thermal Properties on Heat Transfer to Gas Fluidized Beds", I & EC Fundamentals, Vol. 3, No. 4, November 1964, pp. 324-328.

The authors measured heat transfer coefficients from a sphere and a cylinder to an air fluidized bed to determine the effects of solid thermal properties.

TEST SET-UP. A 4 inch diameter lucite column with a porous plate distributor was used. Either a 1/2 inch Celite sphere or a 1/2 inch diameter, 2.3 inches long cylindrical cartridge heater were used to measure heat transfer coefficients. Bed temperature, static bed height were not given. Materials used were:

MATERIAL	dp(μm)	ρ(#/ft ³)	Cp (Btu/#°F)	K (Btu/hr-ft-°F)
Copper	136.4	559	0.092	223
Nickel	136.4	555	0.108	53
Solder (50% Pb-50% Sn)	136.4	554	0.053	27

RESULTS. Average Nusselt numbers of 6.1, 5.7 and 4.6 were obtained for the nickel, copper and solder at $u/umf = 3$. Copper and nickel produced a relatively small difference in Nusselt numbers. Their conductivities differ by a factor of over 4 but have similar heat capacities. On the other hand, the maximum Nusselt number increased 60% with a 100% increase of heat capacity in the nickel-solder comparison.

The proposed correlation is

$$N_u = \frac{h_o D_p}{K_g} = \frac{4\pi/\sqrt{3}}{\left[1 + \frac{3 K_g \bar{\theta}}{\rho_s C_s R p^2 (\alpha+1)} \right]^{\alpha+1}}$$

The maximum observable value of the Nusselt number is 7.2. The effect of gas conductivity, k_g , on the film coefficient is proportional to 0.5. Solid thermal conductivity, k_s , has a negligible effect on the surface-to-fluidized bed heat transfer coefficient, h_o . Coefficient h_o increases with increasing solid heat capacity, C_s . The rate of increase is less than linear.

APPENDIX A
FLUIDIZED BEDS AND THEIR
MATHEMATICAL MODELS

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FLUIDIZED BEDS AND THEIR MATHEMATICAL MODELS

I. DESCRIPTION OF FLUIDIZED BEDS

The term fluidization is used to designate the gas-solid process in which a bed of solid particles is lifted and agitated by a rising stream of gas or liquid.

Basically, a fluidized bed consists of a vessel containing solid particles, which are supported by a distributing grid. A supply of gas or liquid acts as the fluidizing medium. The vessel is generally a column with a rectangular or circular cross-section. The bed may consist of metallic or nonmetallic particles with particle diameters ranging from 30 microns to several hundred microns. Bed depth may vary from less than one inch to several feet. The fluidizing medium may be chosen from a variety of liquids or gases. In some of the earlier concepts, heat transfer was accomplished through the walls of the column. But this severely restricted the available surface area for heat transfer. As a solution to this problem, tubes were immersed in the bed either horizontally or vertically, in bundles or singly.

Various regimes of fluidization are shown schematically in Figure A-1. At a low superficial velocity the fluid percolates through the void spaces between the particles. This is essentially a fixed bed. At a higher velocity there is a shifting of the solids with a slight increase in bed volume, and the frictional drag on each particle becomes high enough to support it without contact with other particles. At this point, the bed of solids is in a state of minimum fluidization, characterized by a pressure drop which is essentially equal to the weight of bed per unit cross section. Since the particles forming the bed are separated by fluid, the bed is unable to resist a shearing force and the bed behaves as a liquid. In liquid-solid systems, an increase in the flow of the fluidizing medium results in a smooth, progressive expansion of the bed with only a slight increase in particle motion. This regime is called particulate fluidization. But in gas-solid systems, higher fluidizing velocities almost

Figure A-1. Fluidization Regimes

Appendix A

I, Description of Fluidized Beds (cont.)

immediately result in formation of gas bubbles at the distributor plate. These bubbles coalesce to form larger bubbles as they rise through the bed which finally burst at the surface. This bed is called an aggregative fluidized bed, and is characterized by a violent agitation and vigorous movement of the solid particles. The degree of agitation is determined by the size and number of bubbles. The bubble behavior is, in turn, dependent upon the nature of the distributor plate and bed height.

At still higher fluidizing velocities, the bubbles become large enough to occupy the entire cross-section of the bed and force up masses of particles above them. This condition is called slugging. Finally, as the velocity is increased beyond

II. HEAT TRANSFER MODELS

When studies showed that the heat transfer coefficients were higher in fluidized systems than under similar conditions without the bed of particles, interest was generated in postulating the mechanism for the increase.

Basically, four different types of surface-to-bed heat transfer mechanisms in gas fluidized beds of fine particles have been proposed by researchers:

- (1) Steady state conduction of heat across the fluid film.
- (2) Unsteady state heat conduction by single particles in direct contact or close proximity to the heat transfer surface.
- (3) Unsteady state heat transfer to "packets" of particles which directly contact the heat transfer surface and then removal by violent disturbances in the core of the fluidized bed.
- (4) Steady state conduction through an emulsion layer.

Appendix A

II, Heat Transfer Models (cont.)

These explanation fall into two classes, those viewing the resistance to transfer within a relatively thin region ($< d_p$, the particle diameter) at the heat exchange surface, and those viewing the resistance in a relatively thick emulsion layer ($> d_p$) adjacent to the heat exchange surface.

A. STEADY STATE CONDUCTION ACROSS THE FLUID FILM

Steady state conduction across the fluid film is one of the earliest models proposed by several researchers to account for the high heat transfer rates in gas-fluidized beds of fine particles. In 1949, Leva et al. proposed a heat transfer model from earlier studies (1947, 1948) that indicated heat flow through fixed beds is primarily opposed by a film resistance inside the vessel and a core resistance in the bed interior. In applying these findings to fluidized beds, the following assumptions were made to derive a heat transfer model:

- (1) Film resistance is decreased due to a reduction in film thickness caused by the vertical motion of particles along the tube wall.
- (2) Core resistance is reduced greatly due to transverse agitation and may be considered negligible compared with the more important film resistance.
- (3) The film thickness is chiefly dependent upon the fluid kinematic viscosity and the vertical component of the average absolute particle velocity within the bed.

Appendix A

II, A, Steady State Conduction Across the Fluid Film (cont.)

The bed to wall heat transfer coefficient of gas fluidized beds of fine particles is given as:

$$h = \left(\frac{\beta k_f}{\delta_f} \right)$$

Where β = constant, k_f = fluid thermal conductivity, δ_f = film thickness.

Levenspiel and Walton (1954) also proposed a steady state conduction mechanism. They attributed the scouring action of descending solids on the gas film to be responsible for decreasing the effective film thickness, thereby increasing the amount of heat transferred. The main features of their thin-film model can be seen in Figure A-2. They derived a simple expression for the effective film thickness for both streamline and turbulent flow conditions which accounted for the frequency of scouring of the film and its growth rate as given by flat plate film growth.

Basically, the thin-film model does not take into account the effect of the physical and thermophysical properties of the solid material. The concept assumes a steady state heat transfer process and results in the conclusion that heat transfer can be increased by increasing the fluidizing gas flow. This conclusion is at odds with most test results.

B. UNSTEADY STATE HEAT CONDUCTION BY SINGLE PARTICLES

Botterill and Williams (1963) presented another thin-film mechanism showing that conductive heat transfer between a single particle immersed in a fluid and a heat exchange surface can be regarded as the basic element in heat transfer to a gas-fluidized bed of spherical particles under conditions where radiative heat transfer is negligible. Botterill based this model on the generally accepted fluidized-bed heat transfer processes listed below:

Figure A-2. Main Features Of The Thin-Film Model Of Levenspiel And Walton

Appendix A

II, B, Unsteady State Heat Conduction by Single Particles (cont.)

- (1) Thermal equilibrium is very quickly established between the gas and the particles in the bulk of the bed.
- (2) The entire thermal gradient is restricted in the immediate region near the heat transfer surface.
- (3) Groups of randomly packed particles convey heat from the surface to the bulk of the bed.

Botterill and Williams viewed the heat transfer process as follows: When fresh, cold emulsion contacts the hot surface, heat flows by conduction alone into the gas and into the uniformly distributed spherical particles touching the surface. For the short emulsion contact times, heat cannot travel far into the emulsion, therefore, attention is focused on a layer of emulsion a bit more than one particle diameter thick. By neglecting conduction through the point of contact, radiation effects and thermal convection through the surrounding fluid, the unsteady-state conduction equations were solved numerically between a plane and an isolated spherical particle surrounded by static gas, initially at the temperature of the bed. Computations were made of 200 micron particles for various fluid and solid properties to give isotherms for different contact times; two are shown in Figure A-3. The results of these calculations show that the solids with their large heat capacity provide an effective heat sink, so heat transfer is located primarily in the region around the contact points of particles with the surface. Because of this, the heat flux can be taken to be proportional to the number of contact points per unit surface.

Ziegler et al (1964) proposed a particle heat absorption model similar to the one proposed by Botterill. Particles from the bulk of the fluidized medium arrive at the heating surface at the fluid bulk temperature. During the time that a particle remains at the surface it receives heat by

Appendix A

Figure A3 Isotherms For Conduction From A Wall To A Glass Sphere In Still Air (From Botterill and Williams)

Appendix A

II, B, Unsteady State Heat Conduction by Single Particles (cont.)

convection from the fluid adjacent to the surface. This fluid is assumed to be at the surface temperature. After some time, the particle leaves the wall and returns to the bulk fluidized medium. The mechanism is pictured in Figure A-4. They concluded that the major portion of the heat transfer occurs by this mechanism, and that the heat transfer coefficient was independent of the particle thermal conductivity but increased with the particle heat capacity.

The single moving particle model assumes that the particles are bathed by fluid at the wall temperature, absorb heat by transient conduction, then give up the heat to the gas. This model results in a conclusion that suggests particle size has very little effect on fluidized bed heat transfer. This conclusion is also at odds with general test results.

C. UNSTEADY STATE TRANSFER ON "PACKETS" OF PARTICLES

This thick emulsion layer mechanism was first introduced by Mickley and Fairbanks (1955). They viewed the unsteady heating of elements of packets of emulsion phase as the vehicle for heat transfer. These packets are viewed to rest on the surface for a short time only to be swept away and replaced by fresh emulsion from the main region of the bed. This is shown in Figure A-5.

Such a "packet" of particles may then be considered as an entity, for the particles will move comparatively little relative to each other. A knowledge of the thermal properties of the quiescent bed is necessary in order to set up the model. Then, if a packet of particles from the bed at temperature T_{bed} is swept into contact with the heat transfer surface at temperature T_s and gives up heat by unsteady state diffusion, then the rate of heat transfer, q , is given by

Appendix A

Figure A4. PROPOSED MECHANISM OF HEAT TRANSFER

Figure A5. Main Features Of The Emulsion-Contact Model Of Mickley And Fairbanks

Appendix A

II, C, Unsteady State Transfer on "Packets" of Particles (cont.)

$$q = \frac{A_m}{\pi} \frac{K_m \rho_m C_{pm}}{\tau} (T_{bed} - T_s)$$

Where A_m = contact surface area

K_m, ρ_m, C_{pm} = thermal conductivity, density and specific heat of the packet respectively

τ = contact time of the packet

The local heat transfer coefficient can then be written as:

$$h = \sqrt{K_m \rho_m C_{pm}} S$$

Where S is a stirring factor introduced by the authors to account for the frequency of occurrence of packets and contact times.

This transient conduction model suggests that thermal conductivity, heat capacity and density are equally important fluid bed particle parameters. However, this relationship has not always been borne out of test results.

The work of Mickley and Fairbanks was refined by Baskakov and Gelperin. Baskakov (1964) felt that this model was deficient because the instantaneous heat transfer coefficient is inversely proportional to the square root of the contact time of the packet leading to an infinite instantaneous heat transfer rate at zero contact time. In contrast to the "packet" model, allowance was made for the effective gap heat resistance between the packet of particles and surface (the contact thermal resistance) and for the heat transferred directly by convection through the particles and between the particles and an immersed heat transfer surface. The fact that the heat transfer coefficient stays finite for very small contact times is further evidence for the existence of a gap between the bed and the heat transfer surface.

Appendix A

II, C, Unsteady State Transfer on "Packets" of Particles (cont.)

Gelperin et al (1966) and Gelperin and Ainshtein (1971) have subsequently developed this model further by considering other details of the heat transfer process. Thus, as well as transfer between the surface and packets of particles, they also made allowance for periods where the heat exchange surface would become shrouded by gas bubbles and for the quantity of heat transferred by gas convection. Expressions were obtained for these different boundary conditions and the mean values of the heat transfer coefficients were found by integration of the expression within the limits corresponding to the heating time of the packet.

In all these surface-renewal-and-penetration type models, particle residence time or contact time at the heat transfer surface is very important. Mickley, et.al., (1961) estimated the contact time of packets by measuring the temperature fluctuations and their frequency on the heater surface. Other techniques include the gamma statistical distribution function of Ziegler, et al., (1964), the so-called multiple capacitance contact time distribution proposed by Perlmutter (1961) and used by Chung, et al., (1972), and the tracer particle counts of Koppel, et al., (1970). The best representation of particle or packet residence times from the above estimates would be difficult to determine at the present, so the methods' usefulness is limited.

D. STEADY STATE HEAT CONDUCTION THROUGH AN EMULSION LAYER

Van Heerden et al., (1951) proposed the first model of this type where they accounted for the thermal resistance of the thin gas film as well as the layer of emulsion with its high heat capacity solids flowing parallel to the wall. They observed that the heat capacity of particles per unit volume is about a thousand times greater than that of the fluidizing medium. They suggested that most of the heat is transferred by particles while the fluidizing gas simply provides for the suspension of particles. They also observed that

Appendix A

II, D, Steady State Heat Conduction Through an Emulsion Layer (cont.)

there is no radial movement of particles as they move along the heat transfer surface. Therefore, they concluded that radial heat transfer will be governed by the thermal conductivity of the suspension (emulsion layer).

A similar model was presented by Wicke and Fetting (1954). They reported that the heat from an immersed surface is first transferred by conduction through a gas layer with thickness δ_f , by two components:

- (1) Heat taken by particles flowing parallel to the surface in the emulsion zone
- (2) Heat transferred into the core of the bed by the interchange of particles

Heat transferred to the core of the bed was found to be negligible as compared to heat taken by particles flowing parallel to the surface. The authors proposed the following relationship:

$$2h L_H/K = 1 - \exp(-2 L_H k_f / \delta_f k_s)$$

$$K = \rho_s (1 - \epsilon_{mf}) C_{p,s} v_s \delta_e$$

Where

- h = the heat transfer coefficient
- L_H = the length of the heat transfer surface
- k_f = the thermal conductivity of the fluid
- δ_f = thickness of gas layer
- k_s = the thermal conductivity of the solid
- ρ_s = the density of the solid
- ϵ_{mf} = the voidage fraction at minimum fluidization velocity
- $C_{p,s}$ = the specific heat of the solid
- v_s = average particle velocity along the surface
- δ_e = thickness of emulsion layer

Appendix A

II, D, Steady State Heat Conduction Through an Emulsion Layer (cont.)

By properly selecting numerical values for δ_f and δ_e , Wicke and Fetting found that they could account for the effect of gas velocity on the heat transfer coefficient observed by themselves, by Baerg, et al., (1950) and by Van Heerden, et al., (1953). This steady state heat conduction model suggests that the most important particle parameters are density and heat capacity but not particle size. But dependency of heat transfer on particle diameter has been observed in tests, pointing out a deficiency in this model.

E. COMBINED MODELS

All of the models discussed earlier are limited in their applicability. Available correlations to evaluate heat transfer coefficients are mostly empirical and confined to specific sets of experimental conditions.

In recent years investigators started suspecting that there may be several heat transfer mechanisms operating simultaneously. Botterill et al., (1967) suggested that the discrepancies between measured and predicted values (of unsteady-state heat transfer by the single particle model discussed earlier), can be accounted for if a gas film of 10 μm thickness is assumed to separate the particles and the heat transfer surface. They concluded that even with short residence times, thermal properties of particles are of secondary importance and the controlling factor is the thermal conductivity of gas.

Yoshida, et al., (1969) presented a unification of the unsteady state heat transfer to "packets" of particles and steady-state conduction through the emulsion layer postulated a mechanism, and gave a criterion to suggest when one or the other may be expected to apply. The bubbling bed model is that mechanism. They give a simple representation of the bubble flow and the interaction of these streams in a fluidized bed. It was observed that, in the core portion of a bubbling fluidized bed, solids are mixed violently by gas bubbles and heat is transported by the solids at a high rate. On the other hand, solids mostly move

Appendix A

II, E, Combined Models (cont.)

downward at the vessel wall and form a solid sublayer which is replaced partially by fresh emulsion elements from the core portion. When a small heater such as a sphere or cylinder is immersed in a fluidized bed, rising bubbles hit the heater frequently; consequently, the contact time of emulsion elements with the heat exchange surface is short. Therefore, solid elements may be anticipated to absorb heat by an unsteady-state conduction. On the other hand, when a long wall surface is used such as in a heat exchanger, a steady temperature gradient will be set-up across the emulsion layer because the frequency of passing bubbles is small near the wall, and the contact time of solids is much longer. In the case of intermediate contact times, both mechanisms should be taken into account. The above model is analogous to and can be explained by the film-penetration-theory for gas absorption into liquids proposed by Toor and Marchello (1958). Residence times for emulsion elements are computed from both random surface renewal, probably most representative of a surface in the main body of the bed continually contacted by rising bubbles, and uniform surface renewal, probably most representative of an emulsion flowing smoothly past the heating surface. Theoretical estimation of heat transfer coefficients roughly coincide with experimental results and were found to be consistent with the bubbling bed model proposed by Kunii and Levenspiel (1968).

In an attempt to develop a generalized model, Kunii and Levenspiel (1969) suggested that all four different mechanisms may operate together as follows: A thin film of gas of thickness l_g coats the surface, some solids are in direct contact with the surface, and the emulsion of equivalent thickness l_e flows past the surface and is replaced occasionally by fresh emulsion, Figure A-6. The theoretical treatments previously have considered one mechanism to be dominant. However, the controlling mechanism may change with the fluidizing conditions as well as the location and geometry of the heat exchange surface. A small immersed body is likely to be frequently contacted by rising bubbles, and each time this happens, the emulsion layer will be swept away. Thus, short contact time may be

Figure A-6 General Model for Bed-Wall or Bed-Surface Heat Transfer

Appendix A

II, E, Combined Models (cont.)

expected with the surface. On the other hand, the frequency of bubbles passing the wall surface will be much lower and solids will more likely descend steadily along the wall giving a much longer contact time. Consequently, it would not be surprising to find the mechanism of unsteady-state conduction through the emulsion layer controls for the long wall surface. Kunii and Levenspiel suggests that the model presented to date are not necessarily contradictory but may correspond to different ranges of emulsion contact times which result from different operating conditions.

APPENDIX B

FLUID BED HEAT EXCHANGER - OPERATING PROCEDURES

APPENDIX B

FLUID BED HEAT EXCHANGER - OPERATING PROCEDURES

Pretest:

1. Conduct screen analysis on test bed material.
2. Determine bed material minimum fluidization velocity, $u_{min. fl.}$, and bed porosity @ $u_{min. fl.}$.
3. Fill 5 gal. containers with bed material. Four (4) containers will provide a bed height of approximately 10 inches, covering all three (3) heat exchanger tube rows. Two (2) containers will provide a bed height of approximately 6 inches, covering 2 inches above the first heat exchanger tube row.
4. Weigh all filled containers and determine total bed weight by subtracting container empty weight.
5. Remove middle flange bolts on fluid bed tank T-1 and disconnect all air and N₂ flex line on upper section. N₂ supply valve V-47 should be closed prior to disconnecting flex line.
6. Raise upper section of tank T-1 by tightening top support bolts on monorail carriers. Push tank toward coolant tank direction as far as possible.
7. Fill tank T-1 with bed material contained in 5 gal. containers. Valve V-34 should be closed prior to filling.
8. Seal tank T-1 by reversing procedures 6 and 5, respectively. Close valves V-33 and V-46.
9. Install a clean bag on bed discharge filter F-1.
10. Install required air orifice F0-3. 1", 1-1/2", 2", and 2-1/2" dia. orifices are available for specified flow ranges. See calibration curves for ranges.
11. Check water level in coolant tank T-3. Fill is necessary. Drain valves V-37 and V-39 should be closed.
12. Clean coolant pump facility water filter F-3. Close valve V-41 before removing filter. After reinstalling filter, open valves V-41 and V-42.
13. Install a clean bag on air discharge filter F-2. Weigh and record bag weight prior to installation on filter F-2.

Pretest:

14. Check instrumentation recorders for chart content and prepare test data sheet.
15. Calibrate coolant RTD probes for differential output. Place all coolant inlet probes in a 32°F ice bath and all coolant outlet probes in 212°F boiling water bath.

Start Up:

16. Place all 110 VAC and 440 VAC electrical switches in OFF position.
17. Place 440 VAC main switch in ON position.
18. Place gas combustor main switch in ON position.
19. Turn trailer instrument key switch to ON position.
20. Place all temperature recorders to ON position. Three Chino and one Omega recorders are involved.
21. Open valve V-44 and adjust valve V-43 to a pressure setting of 60 psig, indicated by the PI-18.
22. Adjust valve V-42 to a flow setting of 50 to 100 cc/min. PI-20 should indicate 50 to 60 psig.
23. Open nitrogen supply valve V-46 and adjust valve V-47 to a pressure setting of 20 psig, indicated by PI-21.
24. Open valves V-48, V-49, V-50, and V-51.
25. Set coolant water temperature controller TIC-17, at 100°F. Controller pneumatically adjusts valve V-40.
26. Place coolant pump P-2 main power switch to ON position.
27. Turn fluid bed heat exchanger key switch HS-28 and coolant pump switch HS-23 to ON position.
28. Set coolant water pressure at 40 psig, indicated by PI-16, by pneumatic adjustment of valve V-39.
29. Set coolant water flows thru Rows 1, 2, and 3 heat exchanger tubes. Valves V-100, V-200, and V-300 should be open. Adjust valves V-104, V-204, and V-304 to require flow rate.
30. Turn gas combustor key switch HS-27 to ON position.
31. Open air valves V-30 and V-31.

Start Up:

32. Place air blower main switch to ON position.
33. Turn on air blower P-1 by pressing START button HS-23.
34. Set air flow rate thru adjustment of valve V-30. Set for highest rate.
35. Set air heater H-1 temperature controller TIC-5 at an air temperature of 400°F and heater element temperature of 800°F.
36. Set air heater H-2 temperature controller TIC-10 at an air temperature of 400°F and heater element temperature of 800°F.
37. Place air heater H-2 main switch to ON position.
38. Turn air heater H-1 and H-2 switches HS-25 and HS-26 to ON position.
39. Record all temperatures, pressures, and flow rates every 30 minutes until steady-state condition is observed. Make adjustments to flows and thermal conditions as required.

Steady State:

40. Record data per test plan. Make three reading sets at 10 min. intervals.
41. Set new air flow rate by adjusting air valve V-30. Set new bed temperature by adjusting TIC-10.
42. Wait approximately 30 minutes and repeat procedures 40 and 41.
43. If air orifice changes are required, turn off air blower P-1 by pushing STOP button HS-23. Install new orifice as soon as possible and restart air blower by pushing START button HS-23. Wait approximately 30 minutes and repeat procedures 40 and 41.
44. Visually observed bed height between flow rate changes to detect serious attrition losses. Bed material may be added thru the bed loader T-2. To operate the bed loader, open valve V-33 and slowly open valve V-45 to assist in blowing the entire bed into the fluid bed tank T-1. After unloading bed in T-2, close valve V-45 and V-33 in respective order. If more bed is to be added, open top cover on T-2 and fill. Close cover and repeat above procedure.

***Shutdown:**

45. Turn air heater H-1 and H-2 switches, HS-25 and HS-26, to OFF position.
46. Place air heater H-2 main switch in OFF position.
47. When bed temperature TI-10 reaches 160°F turn coolant pump P-2 switch HS-24 to OFF position.
48. Place coolant pump P-2 main power switch in OFF position.
49. If bed unloading is desired, open valve V-34 and fill bed discharge filter F-1. Close valve V-34 after 5 minutes and remove bag and bed contents from filter F-1. Continue above procedure until nothing appears in filter F-1.
50. Turn off air blower P-1 by pressing STOP button HS-23.
51. Turn fluid bed heat exchanger and gas combustor key switch HS-28 to OFF position.
52. If trailer instrumentation, air blower, air heater H-1 source power is desired off, place gas combustor main power switch in OFF position.
53. Place 440 VAC main switch in OFF position.

Post-Test:

54. Repeat procedure 15.
55. Disconnect bag on air discharge filter F-2. Weigh and record bag weight. Determine bag content weight by subtracting bag empty weight.
56. Determine total weight of bed material from procedure 49.
57. Collect recorder and all tabulated data. Identify by test number, bed material, and date.
58. Clean area and set up for next test.

*For shutdown in emergency situation, place 440 VAC main switch in OFF position. Contact lead test engineer before starting system again.

APPENDIX C

ERROR ANALYSIS OF TEST DATA

APPENDIX C

Error Analysis of Test Data

An error analysis of the test data was made to evaluate the most probable overall instrumentation error based on the method of root-sum square (rss). This method of estimating uncertainty in experimental results is based on a careful specification of the uncertainties in the various primary experimental measurements.

To estimate the uncertainty in the calculated result on the basis of the uncertainties in the primary measurements, the following method was used:

- (1) The quantity to be computed is N , where N is a known function of the n independent variables, $u_1, u_2, u_3, \dots, u_n$. That is,

$$N = f(u_1, u_2, u_3, \dots, u_n)$$

The u 's are the measured quantities (instruments or component outputs) and are in error by $\pm \Delta u_n$, respectively. These errors will cause an error ΔN in the computed result N .

- (2) Compute the partial derivatives $\partial f / \partial u_1, \partial f / \partial u_2, \dots, \partial f / \partial u_n$ and evaluate each by substituting in the basis data $u_1, u_2, u_3, \dots, u_n$.

- (3) Let u_R be the uncertainty in the result and compute u_R by:

$$u_R = \left[\left(\frac{\partial f}{\partial u_1} \Delta u_1 \right)^2 + \left(\frac{\partial f}{\partial u_2} \Delta u_2 \right)^2 + \dots + \left(\frac{\partial f}{\partial u_n} \Delta u_n \right)^2 \right]^{1/2}$$

The computed quantity N will then have an uncertainty of u_R .

The heat transfer relations governing the AECC test program were

$$q_{H_2O} = W_{H_2O} \Delta H_{H_2O} \quad (\text{Eqn. 1})$$

$$\text{where } \Delta H_{H_2O} = \int_{T_{i, H_2O}}^{T_{o, H_2O}} c_{p_{H_2O}} dT$$

q_{H_2O} is the heat transferred to the water

w_{H_2O} is the water flow in each tube

ΔH_{H_2O} is the change in water enthalpy

T_{i,H_2O} is the inlet water temperature

T_{o,H_2O} is the outlet water temperature

Cp_{H_2O} is the water specific heat

$$UA_o = \frac{q_{H_2O}}{T_{bed} - \bar{T}_{H_2O}} \quad (\text{Eqn. 2})$$

$$\text{where } \bar{T}_{H_2O} = \frac{1}{2} (T_i + T_o)$$

U is the overall coefficient of heat transfer

T_{bed} is the fluidized bed temperature

$\bar{T}_{H_2O}$ is the average water temperature between inlet and outlet

A_o is the outside surface area of the tube

$$\frac{1}{UA_o} = \frac{1}{h_o A_o} + \frac{t_{wall}}{k_{wall} \bar{A}_{wall}} + \frac{1}{h_i A_i}$$

h_o = gas-side heat transfer coefficient

h_i = water-side heat transfer coefficient

t_{wall} = thickness of the tube wall

k_{wall} = thermal conductivity of the tube material

A_i = inside surface area of the tube

$\bar{A}_{wall}$ = average wall surface area between the inside and outside

The instrumentation used and accuracies are shown in Table III-3. The major sources of error used in the analysis are shown in Table C-1. The error of $\bar{T}_{H_2O}$ was determined as one-half the sum of T_{o,H_2O} and T_{i,H_2O} as presented by J. Topping.

The relations were consolidated and the error in the outside film coefficient was determined using the above method. Results were:

$$\text{error in } h_o = \left[(3.25)^2 + (2.95)^2 + (-0.42)^2 + (-0.15)^2 + (-1.94)^2 \right]^{1/2}$$

$$E_{\text{rss in } h_o} = 5.65 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$\% \text{ error} = 10.46$$

Table C-1
Significant Error Sources

<u>Item</u>	<u>Accuracy</u>
Temperature	
Inlet Water	± 0.5°F
Outlet Water	± 0.5°F
Fluidized Bed	± 2°F
$T_{\text{H}_2\text{O}}$	± 0.7°F
Flow	
Water	± 0.05 gal/min