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Report R78-912942

Investigation of Waste Heat Recovery
Utilizing Fluidized Bed Heat Exchangers

Contract No. EC-77-C-03-1433

Final Report

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FOREWORD

An experimental and analytical investigation of fluidized bed heat exchangers was conducted by the United Technologies Research Center (UTRC) under Contract EC-77-C-03-1433 with the Department of Energy (DOE). The Program Manager for DOE was Michael Perlsweig, Fossil Fuel Utilization Division, and the Technical Monitor of the Contract for DOE was Dr. William Thielbahr. The Technical Program Manager for UTRC during the Contract period was Dr. Mikio Suo. This report comprises the required Final Technical Report under the Contract.

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Report R78-912942

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Report R78-912942

Investigation of Waste Heat Recovery
Utilizing Fluidized Bed Heat Exchangers

SUMMARY AND CONCLUSIONS

An experimental and analytical investigation was conducted to evaluate the economic potential of shallow fluidized bed heat exchangers to recover the waste heat in the exhaust gases from large diesel engines. The program involved:

- (1) experiments to measure the heat transfer, pressure loss, and fouling characteristics of a commercially available shallow fluidized bed heat exchanger, and
- (2) conceptual design of a fluidized bed heat exchanger for a particular diesel exhaust waste heat recovery application. Two alternative designs were developed. The first design was based upon the measured characteristics of the tested unit while the second design used extrapolations of these results.

The shallow fluidized bed heat exchanger used in the experimental program is a commercial design manufactured by Stone-Platt Fluidfire Ltd. of England. This unit was designed to transfer 140,000 BTU/hr from 436 F flue gases to 160 F water. The heat transfer surface is comprised of 1.5-in.-OD finned tubes and the bed material is 0.026-in.-dia alumina. Measured heat transfer coefficients were approximately 40 BTU/ft²-hr-F, based upon fin side total surface area after correction for water film resistance and fin conductivity. This coefficient is a factor of four larger than the typical values of 10 BTU/ft²-hr-F achieved in conventional nonfluidized bed heat exchangers. The heat transfer coefficient was insensitive to air flow rate showing only a 5 percent decrease with a 40 percent reduction in superficial velocity from approximately 1.8 to 1.0 ft/sec.

One problem encountered in present day waste-heat heat exchangers is fouling of the heat transfer surfaces. Fouling considerations dominate the heat exchanger design through performance penalties, limiting material and configuration options, as well as requiring surface cleaning equipment (such as soot blowers) to be included in the design. However, it is believed that fluidized bed waste-heat heat exchangers should have improved fouling characteristics when compared to conventional, nonfluidized bed heat exchangers. Exploratory tests were conducted to investigate the fouling characteristics of the Fluidfire heat exchanger. Small carbon particles were added to the airstream to produce a particulate burden of approximately 0.04 gr/SCF over a total of six hours. During this period the system pressure losses did not vary. However, the heat transfer coefficient declined a small amount (approximately 5 percent), but recovered to its original value upon removal of the carbon burden.

During this test the particle color turned to dark black but there was only a slight discoloration of the finned tubes. Thus the addition of the carbon particles had a negligible fouling effect over the duration of these tests.

The physical characteristics and performance related parameters of the Fluidfire unit were used to formulate a baseline conceptual design for a waste-heat heat exchanger to recover approximately 10^7 BTU/hr from diesel engine exhaust gases. Design conditions correspond to those for the waste-heat heat exchanger designed by Mechanical Technology Incorporated (MTI) and currently under installation at Rockville Center, N.Y. In the conceptual design, preheater and superheater stages utilize conventional finned tubes and a fluidized bed serves as the vaporizer (85 percent of the total thermal load). The total heat transfer surface area is reduced 30 percent from the 10,200 sq ft in the MTI unit to 7100 sq ft for the fluidized bed baseline conceptual design. The estimated cost of this unit is \$58,000, \$8000 more than the estimated installed cost of the MTI unit. This cost increase may be compensated for to some degree by the reduced operation and maintenance costs associated with the improved fouling characteristics of a major portion of the total heat exchanger system. However, since at least a portion of the heat exchanger system is not immersed in fluidized beds, fouling resistance has probably not been improved enough to justify the initial cost increase.

A number of technical improvements to this baseline fluidized bed heat exchanger design were identified and used to formulate an alternative design. The alternative design has two fluidized beds staged in counterflow, allowing preheater and superheater stages, as well as the vaporizer stage, to be fluidized beds. The total heat transfer surface area required is reduced 50 percent from 10,200 sq ft in the MTI unit to 5600 sq ft for the alternative design. The initial cost estimate for the alternative design is \$42,000, \$8000 less than the estimated installed cost of the MTI unit. Analysis of performance and construction options for the alternative design show the potential for further reducing the unit cost. In addition to an initial cost advantage, the alternative design should have significantly reduced operation and maintenance costs as compared to a conventional nonfluidized bed waste-heat heat exchanger. This cost reduction is due to the improved fouling resistance of an all fluidized bed design.

Fluidized beds are a viable technical option for waste-heat heat exchangers. They offer the potential advantages of improved surface fouling characteristics at a reduced cost as compared to conventional finned tube heat exchangers. Although this study was specifically directed at the application of waste heat recovery from large diesel engines, the results are applicable to many alternative applications involving heat recovery from low temperature gases. In particular, this includes gas turbine and combined cycle energy recovery and conversion systems.

INTRODUCTION

There are many power generating and industrial processes where thermal energy is currently discarded to the environment. Examples of these are chemical processing plants as well as gas turbine and diesel power plants. This thermal energy is often at a temperature where power could be extracted from it or, at the very least, residential or commercial building heat could be provided. Since the annual use and cost of energy in the United States is projected to increase, efficient utilization of waste heat will be a future requirement. This requirement is even further accentuated by the foreign oil importation problem.

The waste heat from a prime mover, either a gas turbine or diesel engine, could be utilized in a bottoming cycle to increase the power output with no increase in fuel consumption. Without the bottoming cycle, current high technology engines can achieve mid-thirties range thermal efficiency. With the same engine in a combined cycle (utilizing the waste heat), a thermal efficiency in the mid-forties can be achieved. Thus a significant improvement in energy resource conservation can be made through use of combined cycles.

Existing gas turbine installations for electric power generation in the United States for 1973 are tabulated by Sawyer (Ref. 1). The total capacity was 28,900 MW and the total power generated was 25.4 million MW-hr. The average load factor, defined as power generated divided by total generating capacity, for these units is 10 percent, characteristic of utilization during peak demand periods only. Though total utilization would be greater than 10 percent, due to part load operation, this utilization does not lend itself to advantageous use of bottoming cycles. However, with the development of advanced bottoming cycles, these plants could economically be used for intermediate load operations. Thus there appear to be significant benefits from more economical waste-heat heat exchangers in this applications.

Similarly, the American Public Power Association estimates that approximately 3000 MW of capacity presently exists in diesel electric generating plants. This equipment is used to generate approximately 10.5 million MW-hr per year. These figures indicate that the average load factor for the diesel engine electric generating plants is nearly 40 percent. However, operational hours would be greater due to part load operation of the units. These utilization figures indicate that the addition of bottoming cycles to the diesel engine electric generator installations, as utilized today, would produce substantial energy savings.

One reason that waste heat is not more extensively utilized is the cost of the heat exchangers which is a major fraction of the combined cycle power plant cost. Thus even a small decrease in the cost of waste-heat heat exchangers will improve the economics of combined cycles. The dominant thermal resistance is usually on the

waste heat side of the heat exchanger. Thus any attempt to decrease the heat exchanger cost will focus upon enhancing the waste heat side heat transfer coefficient. At least two methods are currently used to increase this heat transfer coefficient. The first method is to increase the velocity in the heat exchanger. This, however, leads to increased pressure losses which usually limits the extent to which velocity can be increased in a given heat exchanger.

The second method is to use extended surfaces for heat transfer coefficient enhancement. One example of this is a pin-fin heat exchanger. The pins provide additional heat transfer surface area and also enhance the heat transfer coefficient. Heat transfer coefficient enhancement is achieved in this device as well as most other commonly used enhancement devices by generation of turbulence at the surface. Increased pressure loss usually results. Many extended surface configurations have been investigated and the results of these investigations reported in the literature. In practice, the problems associated with augmenting surface area and enhancing the heat transfer coefficient include increased complexity, with associated cost increases, and a tendency to promote fouling.

A concept which has great potential for lowering heat exchanger cost and improving waste heat recovery efficiency is the fluidized bed. It has long been recognized that high heat transfer coefficients may be obtained within fluidized beds. This feature has been utilized in many applications, but most widely in the chemical processing industry. Currently the emphasis in fluidized bed development is directed at improved combustion efficiency and pollution control when burning coal.

Fluidized beds have not been extensively utilized as heat exchangers per se, probably because of the large pressure losses normally associated with them. Recently, several authors (Refs. 2, 3, and 4) have discussed the use of shallow fluidized beds as heat exchangers. Additionally, such heat exchangers are being sold for waste heat recovery applications by Stone-Platt Fluidfire Ltd. of England. Shallow fluidized beds have lower pressure losses, higher heat transfer coefficients, and improved fouling resistance as compared to conventional nonfluidized bed heat exchangers.

Prediction of heat transfer coefficients in fluidized beds requires a considerable amount of subjective judgment and involves very large uncertainties. It is common for existing correlations to predict heat transfer coefficients that differ by an order of magnitude for a given system. This discrepancy is due to the complex mechanisms controlling fluidized bed heat transfer rates and to researchers reporting their results in a generalized manner not justified by the range of experimental conditions investigated. As a result, an educated guess is frequently more accurate than an existing correlation to predict the heat transfer coefficient and in all cases experimental verification is required.

In this program, a fluidized bed heat exchanger was purchased from Stone-Platt Fluidfire Ltd. and tested. The results of these tests were utilized to formulate the conceptual design and cost estimate for a heat exchanger to recover approximately 10^7 BTU/hr from the exhaust gases of a diesel engine. Additionally, an alternative design was developed incorporating a number of technical improvements to the baseline design. Presented herein are the results of the experimental program and the heat exchanger conceptual designs.

EXPERIMENTAL PROGRAM

The objective of the experimental program was to provide the information required to estimate the economic potential of shallow fluidized bed heat exchangers to recover waste heat from large diesel engine exhaust gases. The test program encompassed measuring the pressure losses and heat transfer rates of a fluidized bed heat exchanger as well as exploratory tests of the fouling characteristics of the heat exchanger. Described herein is the experimental equipment and data analysis procedures used as well as the experimental results and their analysis. A data uncertainty analysis is presented in APPENDIX I.

Experimental Equipment and Procedures

The shallow fluidized bed used in the experimental program was manufactured by Stone-Platt Fluidfire Ltd., and is a design commercially marketed by them. Figure 1 is a diagram of the unit and Fig. 2 is a photograph of the unit as installed at UTRC. This unit is the result of over six years of research and development by Fluidfire into the performance of shallow fluidized beds for both waste heat recovery and combustion systems. The unit was designed to transfer 140,000 BTU/hr into hot water for space heating using 436 F flue gases. As tested at UTRC, heated air was substituted for the flue gases. The specifications for the unit are given in TABLE I.

Air System

The tests were performed using heated air as the fluidizing medium. To minimize the heating requirements the air was recirculated, passing through a heater, then through the bed, and back to the heater. Circulation was provided by a fan provided with the heat exchanger rated at 800 ACFM at 6 in. H₂O pressure rise at an air temperature of 250 F. The fan was mounted on top of the unit as shown in Fig. 2. As installed at UTRC, the fan discharge is routed to an electric heater mounted within a duct adjacent to the heat exchanger (see Fig. 2). The heater is a 60 kW, 480 V unit connected into four manually controlled circuits of 6, 12, 18, and 24 kW capacity. A temperature controller monitors heater output temperature and is connected as an over-temperature protection for the heater. Electric power to the heaters was measured by two Weston Model 310 Watt meters, one on each of two phases. The Watt meters required both current and potential transformers to measure these power levels. Transformer calibration factor was determined by measuring the voltage ratio across the potential transformer and from the specifications of the current transformer. Phase shift errors were found to be negligible by comparison with a laboratory standard transformer. Estimated uncertainty in power measurement is ± 1.9 percent (see APPENDIX I). After passing through the heater, the air passes through a butterfly control valve into the plenum beneath the fluidized bed. The air then passes through the distributor plate and the fluidized bed into the freeboard region.

DETAILS OF FLUIDIZED BED HEAT EXCHANGER
MANUFACTURED BY STONE-PLATT FLUIDFIRE LTD

(ALL DIMENSIONS IN FEET)

FIG. 1

FLUIDIZED BED HEAT EXCHANGER TEST INSTALLATION

TABLE I

FLUIDFIRE HEAT EXCHANGER SPECIFICATIONS

Performance (Ref. 5):

Thermal Rating: 140,000 BTU/hr
 Heat Source: Propane products of combustion
 Gas Inlet Temperature: 436 F
 Gas Exhaust Temperature: 247 F
 Water Inlet Temperature: 160 F
 Water Outlet Temperature: 180 F

Heat Exchanger Surface (Measured):

Tube and Fin Material: Copper, assumed thermal conductivity of 220 BTU/ft-hr-F
 Tube to Fin Attachment: Alloy bonded
 Number of Tubes: 12 (in series/parallel combination)
 Tube Outer Diameter: 0.715 in.
 Tube Inner Diameter: 0.50 in.
 Fin Outer Diameter: 1.375 in.
 Fin Spacing: 73/10 in.
 Fin Thickness: 0.025 in.
 Outside Surface Area: 1.54 ft²/ft
 Ratio Outside to Inside Area: 11.7
 Total Surface Area in Bed: 50 ft²

Gas Flow Area (Measured):

Total Grate Area: 7.36 ft²
 Active Distributor Area: 5.94 ft²
 Tube Blockage: 1.78 ft²

Particle Properties (Measured):

Material: Alumina
 Diameter: 0.026 ±0.005 in. (by microscopic examination)
 Density: 245 lbm/cu ft
 Voidage of Poured Bed: 0.43

Air flow rate is determined by measuring the differential pressure in the connection between the freeboard above the bed and the fan entrance. This connection includes a 180 deg turn. Pressures were measured with 2 in. inclined manometers having 0.01 in. H₂O resolution. One manometer was connected to measure the pressure drop across the elbow, while a second manometer was used to measure the transverse pressure gradient at the 100 deg point of the turn (see Fig. 2). Calibration of this flowmeter was provided by removing the connection between the fan and the heater duct and inserting a flow nozzle constructed to meet ASME standards (Ref. 6). This nozzle discharged into the room with fresh air entering the system at the top of the heater duct. The pressure drop across the elbow was found to correlate to the flow nozzle with a 2 σ standard deviation of 7.5 percent (40 points) and the transverse pressure gradient was found to correlate to the flow nozzle with a 2 σ standard deviation of 5 percent (45 points). Both measurements were corrected for Reynolds number effects (Ref. 6). The final air flow rate was determined by averaging the values obtained by these two techniques. Total estimated uncertainty at the nominal operating point is 6.5 percent (see APPENDIX I) with 1.8 percent of this being interdata variances and the balance an unknown, but constant, bias.

Pressure losses across all components of the system were measured and compared to the pressure rise of the fan. Instrumentation locations are shown in Fig. 3. All measurements were with oil filled U-tube manometers and the summation of pressures around the air circuit was always less than 0.1 in. H₂O. The pressure loss of interest was that of the distributor and the fluidized bed. The distributor pressure loss was determined during operation without the fluidized bed. These data were correlated using the procedures in Bean (Ref. 6). Pressure loss of the fluidized bed was determined by subtracting the calculated distributor pressure loss from the measured sum of the distributor and bed pressure loss.

The air temperature was measured in the elbow above the freeboard of the bed, at the inlet to the heater duct, and in the plenum beneath the distributor, as shown in Fig. 3. The former two measurements were made with bare thermocouple beads, and the latter with a shielded, aspirated thermocouple. The temperature was found to be relatively constant at each location and the thermocouple was located in the center of the constant temperature zone. Observed temperature variation across each duct was used in APPENDIX I to estimate the measurement uncertainty. Thermocouple wire used was special limit copper-constantan wire manufactured to a $\pm 3/8$ percent tolerance to NBS calibration standards. Each thermocouple was installed with the adjacent length of wire on the spool used as its reference junction. This decreased the uncertainty in temperature measurement considerably below that quoted by the manufacturer. Thermocouple outputs were measured with a digital voltmeter with an accuracy of $\pm 2 \mu V$ (approximately ± 0.1 F). The heater inlet temperature varied less than ± 0.1 F over a one minute interval. The heater outlet and plenum temperature both varied ± 0.5 F over the same time. The aspirated probe showed a 6 F temperature rise with suction as compared to without suction at a gas temperature of 440 F.

FLUIDIZED BED HEAT EXCHANGER FLOW PATHS

FIG. 3

In addition to the thermocouples mentioned above, an additional thermocouple bead was mounted in a probe to characterize the thermal properties of the fluidized bed. Translating the probe vertically through the bed showed the temperature to be uniform to within 0.25 in. (approximately ten particle diameters) of the distributor plate. Below this plane there was a significant rise in bed temperature, increasing as the distributor plate was approached. This measured bed temperature distribution is in agreement with data of other researchers (Ref. 7). The internal bed probe was also translated horizontally through the bed to check for temperature uniformity. The bed temperature was found to vary by about ± 2.5 F at the nominal operating point. The lowest bed temperature occurred at the water entrance plane and the highest bed temperature occurred at the water exit plane, indicative of incomplete horizontal particle mixing. With air flow rates and temperatures measured, it is possible to calculate the amount of heat transferred to the water by the fluidized bed. The estimated uncertainty in this parameter is 7.8 percent (see APPENDIX I).

The amount of heat transferred to the air by the heater can also be calculated. This value can be compared to the measured electrical input to provide a heat balance. The outline of the calculations is summarized in APPENDIX I. Experimentally the heat balance across the heater showed a difference between air heat addition and electrical power measured of between 5 and 15 percent. The greater value was always the measured electrical power and the difference increased with decreasing air flow rate. This difference is consistent with expected thermal losses. These measurements provided an independent means of verifying the air flow rate.

Water Flow Systems

Thermal load to the heat exchanger was provided by passing water through the finned tubes located in the fluidized bed. The water was recirculated in a closed loop with a pump withdrawing the water from a tank, pumping it through the heat exchanger, and then back to the tank as shown in Fig. 3. A cold water addition control in the return line provided temperature control of the water system with an overflow spout on the tank maintaining the water level. Water flow rate was measured with three rotameters. Two rotameters with a full-scale reading of about 9 gal/min were manifolded in parallel. A third rotameter was connected in series with the first two. Consequently, two water flow rate measurements were always available increasing the accuracy of the measurement. The three rotameters were calibrated in place by measuring the weight of water passed through the rotameters in a known time period. Accuracy of the weight measurement was verified by using standard weights. Correlation coefficient of the calibration curves in all three cases was greater than 0.998 and for two of the rotameters greater than 0.9998. All water flow rate measurements were corrected for water density variations. The two water flow rate measurements were averaged to produce the best estimate for water flow rate. This average has an estimated uncertainty of 1.9 percent at the nominal operating point (see APPENDIX I). Temperature of the water was also measured with

the special limit copper-constantan thermocouple wire (see previous description of the air temperature measurement). Variability of this measurement was found to be 0.05 F over a one minute interval.

The temperature rise of the water and water flow rate were used to calculate the heat transferred by the fluidized bed. Estimated uncertainty in this measurement is 3.1 percent. The heat balance across the bed (i.e., water heat gain less air heat loss) differed by an average value of 1.6 percent, increasing with decreased air and water flow rates. However, only a small (0.15 percent) bias is present in the data. This compares favorably to a calculated uncertainty limit of 8.4 percent and a bias limit of 2.5 percent (see APPENDIX I).

Calculation of Heat Transfer Coefficients

Prior to calculating the heat transfer coefficient it is necessary to characterize the fluidized bed and select the proper analytical model. Two distinct models have been developed by Suo (Ref. 8): that of the mixed bed (complete horizontal particle mixing) and the distributed bed (no horizontal particle mixing). The real case is, of course, between these extremes. However, for the conditions tested in this program ($NTU \leq 4$, $C_{min}/C_{max} \leq 0.2$) heat transfer coefficients predicted using these models differed by less than 1 percent. For simplicity, the mixed bed model was used.

One critical parameter in calculating the heat transfer coefficient is the temperature of the fluidized bed. This parameter was determined by measuring the air temperature in the elbow above the bed and correcting this value for thermal losses in the freeboard region. This correction ranged from 2.5 F to 3.5 F depending upon air flow rate. This corrected bed temperature was always found to lie within the temperature range measured by the thermocouple within the fluidized bed.

The overall heat transfer coefficient can be calculated using either the air temperature loss or the water temperature loss as a measure of the total heat transferred. The best estimate of the heat transfer coefficient is obtained by averaging these values, producing a total uncertainty of 6.3 percent. Bias of this parameter is 4 percent, with interdata variations significant at the 2.3 percent level (for a large number of data points).

The overall heat transfer coefficient is the result of three distinct heat transfer mechanisms: forced convection from the water to the tube wall, fin conduction, and heat transfer from the fluidized bed to the fins. The former two, calculated using the procedure in APPENDIX I, were used to calculate the fluidized bed heat transfer coefficient.

Soot Tests

In addition to the heat transfer tests, an exploratory test series was conducted under simulated sooting conditions. These conditions were produced by injecting small carbon particles into the air stream at the plenum entrance. Monarch 1300 particles manufactured by the Cabot Corp. were used. Particle size was nominally 15 millimicrons. A Plasmadyne Model 1000 A powder feeder was used to inject this material. Measurement of the size and concentration of injection particles was obtained by extraction of filter samples. Filter paper used was Nuclepore membranes with 0.2 μ holes.

Discussion of Results

The main emphasis in the experimental program was characterizing the heat transfer properties of the Fluidfire shallow fluidized bed heat exchanger. Additional efforts focused on the effects of soot addition upon the heat exchanger performance. In all of the tests fluidization was observed to be uniform over the cross section of the unit with fluidization more intense between the tubes than vertically above them.

Heat Transfer Tests

The heat transfer data can be classified into four categories: heat transfer in the region of the nominal operating point, effect of water temperature, effect of reduced water flow rate, and effect of reduced air flow rate. Fifteen data points were obtained and are summarized in TABLE II. Four data points were obtained in the region of the nominal operating point, though none of them exactly at that point. This is due to an inability to set the test facility controls at a specific point (i.e., the heater is usable in fixed increments only, flow rates drift slightly from the set values with temperature and time). The average bed Nusselt number was 4.79 with a standard deviation of 0.15. The Nusselt number was calculated using the bed-to-tube heat transfer coefficient, the particle diameter, and the thermal conductivity of air evaluated at the mean film temperature. At the nominal operating point the bed-to-tube heat transfer coefficient was 40.1 BTU/ft²-hr-F (based on total bed side surface area and the average Nusselt number). Twenty-six percent of the overall thermal resistance resulted from the water film resistance and the fin conductivity. Combining these effects with the bed-to-tube thermal resistance results in an overall heat transfer coefficient equal to 29.4 BTU/ft²-hr-F. Overall heat transferred was 119,000 BTU/hr as compared to the original specification of 140,000 BTU/hr. The manufacturer's attributed this difference to the fact that air rather than products of combustion was used for the fluidizing medium. No assessment of the effect of this substitution was made in this study.

TABLE II DATA SUMMARY

DATA POINT	AIR			PRESSURE LOSS		WATER			HEATER POWER			HEAT TRANSFER AT BED			HEAT TRANSFER COEFFICIENT			
	Flow Rate lbm/sec	Inlet Temp F	Outlet Temp F	Distributor in. H ₂ O	Bed in. H ₂ O	Flow Rate lbm/min	Inlet Temp F	Outlet Temp F	Electric BTU/hr	Air BTU/hr	Deviation %	Water BTU/hr	Air BTU/hr	Deviation %	BTU/ft ² -hr-F			Bed Nusselt Number
															Overall	Water	Bed	
NOMINAL OPERATING POINT:																		
1.	0.73	429	246	3.5	2.3	120	160.1	176.1	129,000	123,000	4.4	115,000	117,000	1.2	29.9	1720	40.6	4.83
2.	0.74	423	246	3.5	2.3	115	162.2	178.8	126,000	121,000	4.6	114,000	114,000	0.54	30.4	1670	41.8	4.98
3.	0.73	435	253	3.5	2.3	109	162.5	180.2	132,000	128,000	2.6	115,000	116,000	0.8	28.6	1610	39.0	4.62
4.	0.71	423	245	3.3	2.4	117	161.6	177.5	125,000	117,000	6.4	111,000	110,000	0.73	29.4	1700	39.8	4.74
CALC. MEAN*	0.73	436	250	3.5	2.3	114	160.0	177.4	-----	-----	----	119,000	-----	----	29.4	1660	40.1	4.79 ± 0.15
VARIABLE WATER TEMPERATURE:																		
5.	0.72	502	238	3.7	2.3	118	103.1	127.0	185,000	171,000	7.6	170,000	166,000	2.0	27.6	1370	39.0	4.80
6.	0.71	520	250	3.7	2.2	116	119.1	143.0	191,000	173,000	9.3	167,000	168,000	0.8	28.2	1460	39.4	4.79
CALC. MEAN*	0.73	436	250	3.5	2.3	114	160.0	177.4	-----	-----	----	119,000	-----	----	29.4	1660	40.1	4.79
7.	0.76	342	242	3.4	2.3	116	195.3	204.6	74,000	71,000	4.8	65,000	66,000	2.1	30.9	1860	41.5	4.85
																		4.81 ± 0.03
ONE HALF NOMINAL WATER FLOW:																		
8.	0.71	445	262	3.4	2.2	59	163.5	195.6	130,000	122,000	6.5	114,000	114,000	0.2	27.8	1030	44.0	5.13
9.	0.75	362	234	3.5	2.3	61	159.2	182.6	92,000	89,000	3.0	86,000	84,000	2.2	27.4	1010	43.5	5.20
10	0.74	403	249	3.5	2.2	57	161.0	189.3	112,000	106,000	5.4	97,000	99,000	2.4	27.1	980	44.6	5.22
11.	0.73	438	260	3.5	2.2	57	161.7	193.7	131,000	118,000	9.7	110,000	112,000	2.2	27.6	990	44.4	5.19
CALC. MEAN**	0.73	436	260	3.5	2.2	57	160.0	192.7	-----	-----	----	113,000	-----	----	27.4	970	44.3	5.19 ± 0.04
REDUCED AIR FLOW:																		
12.	0.63	417	238	2.6	2.2	110	161.0	175.5	111,000	104,000	6.5	96,000	99,000	2.8	28.3	1600	38.5	4.62
13.	0.61	459	248	2.5	2.3	112	161.8	178.4	129,000	117,000	9.0	112,000	113,000	0.5	29.0	1640	39.5	4.68
14.	0.47	417	224	1.4	2.4	114	162.0	173.6	92,000	82,000	10.7	80,000	79,000	1.8	28.4	1620	38.6	4.62
15.	0.43	426	222	1.2	2.4	115	159.6	171.1	92,000	80,000	13.0	79,000	76,000	3.6	27.6	1640	37.1	4.49

* Calculated mean conditions from data points 1 through 4 using measured bed Nusselt Number.

** Calculated mean conditions from data points 8 through 11 using measured bed Nusselt Number.

The second data set in TABLE II is for points with variable water temperature. Examination of these data show that both the overall and bed-to-tube heat transfer coefficient increased with water temperature. This trend is consistent with the increase in gas thermal conductivity with temperature. Calculated Nusselt numbers for this data set are constant within the reproducibility of the data and differ by less than 0.5 percent from the previous data set.

The third data set is for points at reduced water flow rate. Reducing the water flow rate reduced the overall heat transfer coefficient from 29.4 BTU/ft²-hr-F to 27.4 BTU/ft²-hr-F. After correcting this parameter for the water-to-tube heat transfer resistance and fin effectiveness, the fluidized bed Nusselt number was found to increase from 4.8 to 5.2. Insufficient data were available to determine if this increase in the bed Nusselt number is real or results from inaccuracies of the method used to calculate the water film resistance.

The fourth data set concerns the dependence of the heat transfer coefficient upon air flow rate. A slight dependence is shown here; however, only a 6 percent reduction is observed for a 40 percent reduction in air flow. This insensitivity of the heat transfer coefficient to air flow rate is as predicted by published correlations. Plotted in Fig. 4 are several correlations for heat transfer between a fluidized bed and an immersed surface as well as the data obtained in this program. Only a slight dependence of the heat transfer coefficient upon fluid velocity is predicted by these correlations. Most of these correlations predict higher heat transfer coefficients than measured in this program. One explanation for this is the stagnant zones observed in the bed. These zones were visualized during the soot tests and are discussed with those test results.

Pressure Losses

A second area of interest was the pressure loss across the fluidized bed and the distributor. As shown in TABLE II, the pressure loss was constant (within the experimental accuracy) at 2.3 in. H₂O over the range of test conditions. A force balance indicates that the pressure loss across a fluidized bed should equal the bed weight per unit flow area. For a measured static bed thickness of 1.5 in., this results in a pressure loss estimate of 3.35 in. H₂O. The measured pressure loss was only 70% of this theoretical pressure loss. This value is consistent with the measurements of Frantz (9) where pressure losses for shallow fluidized beds (2 in. thick) were found to be 65% of the theoretical pressure loss. This discrepancy is best explained by the existence of stagnant zones in the bed. In these zones the bed weight would not be supported by the air flow, reducing the pressure loss.

MEASURED HEAT TRANSFER COEFFICIENTS COMPARED TO EXISTING CORRELATIONS

Soot Tests

One problem encountered in present day waste-heat heat exchangers is fouling of the heat transfer surfaces. Fouling considerations dominate the heat exchanger design through performance penalties, limiting material and configuration options, as well as requiring surface cleaning equipment (such as soot blowers) to be included in the design. However, it is believed that fluidized bed waste-heat heat exchangers should have improved fouling characteristics when compared to conventional, non-fluidized bed heat exchangers. In addition to the heat transfer tests previously discussed, an exploratory test series was conducted to determine if the addition of fine carbon particles to the air flow would create significant changes in the performance of the heat exchanger.

The particulate burden in the exhaust gases of large diesel engines was determined from a review of published data. Average particulate burden at the Catalina diesel-electric generator station is 0.675 g/kWH (Ref. 14). Assuming exhaust products have the same density as air (0.07 lb/SCF) and a specific air consumption of 20 lb/kWH, results in a particle burden of 0.035 gr/SCF. Data from other large diesel plants (Ref. 15) indicate a particulate burden range of 0.075 to 0.31 lb/MMBTU. Assuming exhaust products have the same density as air (0.07 lb/SCF), a fuel oil heating value of 0.02 MMBTU/lb, and an air to fuel ratio of 35 (Ref. 14), results in a particulate burden range of 0.021 to 0.087 gr/SCF. These numbers are consistent with those in the literature for mobile diesel engines showing a range of 0.013 to 0.35 gr/SCF (Refs. 16, 17, and 18).

In the first soot test the particle feeder was operated for thirty-eight minutes producing an average particulate burden in the air of 0.066 gr/SCF with an average particle diameter of 0.6 mils (16 microns). Pressure drop across the bed and distributor combination remained constant at 5.7 in. H₂O throughout the test. However, the heat transfer coefficient declined 5 percent over the duration of the test from an initial value of 29.5 BTU/ft²-hr-F as shown in Fig. 5. When the unit was started three days later, the heat transfer coefficient was again 29.5 BTU/ft²-hr-F. During the second soot test the particle burden averaged 0.004 gr/SCF. Total test duration was 105 minutes with both heat transfer rate and pressure drop remaining constant throughout the duration of the test.

A third soot test of 240 minutes duration was conducted at an average particulate burden of 0.04 gr/SCF. During this test the pressure drop remained constant while the heat transfer coefficient declined 5 percent. However, particulate inventory in the system was built up sufficiently to maintain a carbon burden of 0.017 gr/SCF without further particle addition. Thus post-soot test heat transfer measurements without carbon present were not possible.

VARIATION OF OVERALL HEAT TRANSFER COEFFICIENT AND PRESSURE LOSS WITH TIME DURING SOOT TESTS

R78-912942

FIG. 5

78-07-95-3

Results of the tests employing carbon particles indicated no significant change in heat exchanger performance from that obtained using clean air. The approximately 5 percent reduction in overall heat transfer coefficient is within the estimated experimental uncertainty of 6.3 percent. However, the measured change is greater than the interdata variations obtained in the tests with clean air. Additional effort to ascertain the cause of the small change in heat transfer coefficient which occurred during the tests with carbon is believed to be unwarranted.

Post-test inspection showed the inside surface of the unit to have a uniform black coating. The distributor plate was black but did not appear to have any significant blockage. Similarly the heat exchanger tubes were slightly discolored. The bed material was uniformly black except for pockets of white material between the fins on top of each tube as shown in Fig. 6. Adjacent to these pockets the copper fins also maintained their original copper color. The presence of the white material and bright copper on the tubes indicates there are stationary unfluidized areas in the bed (with or without soot). As a result, not all of the heat transfer surface area is effective and measured heat transfer coefficients would be lower than predicted by heat transfer correlations in the literature.

BED APPEARANCE AFTER SOOT TEST

R78-912942

FIG. 5

HEAT EXCHANGER CONCEPTUAL DESIGN STUDIES

To evaluate the potential of fluidized bed waste-heat heat exchangers for energy recovery and conversion, a conceptual design study was performed for a specific application. The heat exchanger application investigated was the recovery of waste heat from the exhaust gases of a large diesel-electric generator. This heat is envisioned to power a bottoming cycle, thus increasing gross electrical output. Three bottoming system concepts are currently being investigated by three contractors under DOE sponsorship: one by Thermoelectron, Inc.(Ref.19); a second by Sundstrand Corporation (Ref.20); and a third by Mechanical Technology, Inc. (MTI) (Ref.21). For this study it was decided to choose one of these systems and incorporate a fluidized bed waste heat boiler. Choice of one of these bottoming system concepts determined the process conditions and provided a direct basis for comparing cost estimates. The MTI system was chosen on the basis of compatibility of fluidized bed characteristics with the heat exchanger requirements and availability of information.

The MTI designed Binary Rankine Cycle Waste Heat Recovery System uses both steam and Freon as working fluids. The system is designed to generate 500 kWe using 9,600,000 BTU/hr of heat recovered from the exhaust of a diesel-electric generator. A system description is presented by Rhinehart (Ref.21). The primary waste heat boiler for this system is a conventional finned tube design incorporating both superheater and preheater stages in addition to the steam vaporizer. The vaporizer employs forced recirculation raising 20 percent quality steam per pass with vapor separation in a separating drum. Detailed specifications of the unit are given in TABLE III (Ref. 22). Fabrication cost of this unit is \$45,000 (Ref.23) and installation cost is estimated to be \$5,000 for a total cost of approximately \$50,000.

The performance parameters for the MTI heat exchanger were used as design values for the fluidized bed heat exchanger (i.e., raise 9000 lb/hr of 430 F, 60 psia steam from 215 F feedwater using 205,000 lb/hr of 520 F exhaust gases). Two heat exchanger designs were developed during the studies: a baseline design which duplicated as many of the performance related parameters of the Fluidfire unit as possible; and an alternative design incorporating a number of technical innovations on the Fluidfire unit design.

Baseline Design Configuration

A single fluidized bed cannot serve as preheater, vaporizer, and superheater due to varying temperature level requirements and the isothermal properties of fluidized beds. Consequently, it was decided to use the fluidized bed for the vaporizer only (85 percent of the total thermal load) with the conventional finned tubes of the MTI design serving as the preheater and superheater stages. The resulting conceptual design is shown in Fig. 7. The exhaust gases first pass through

TABLE III: HEAT EXCHANGER SPECIFICATIONS

	MTI Design (Ref. 21)				Fluidized Bed Design			
	Preheater	Vaporizer	Superheater	Overall	Preheater	Vaporizer	Superheater	Overall
Heat Transfer Rate, Btu/hr	713,700	8,237,800	648,900	9,600,000	713,700	8,237,800	648,900	9,600,000
Exhaust Gas Flow Rate, lbm/hr	205,000	205,000	205,000	205,000	205,000	205,000	205,000	205,000
Water/Steam Flow Rate, lbm/hr	9,000	45,000	9,000	9,000	9,000	45,000	9,000	9,000
Entering Gas Temperature, F	345.6	506.1	520	520	340	503	520	520
Exiting Gas Temperature, F	333	345.6	506.1	333	327	342	506	327
Entering Water Temperature, F	215	293	293	215	215	293	293	215
Exiting Water Temperature, F	293	293	430	430	293	293	430	430
Gas Heat Transfer Coefficient, Btu/ft ² -hr-F	12.6	12.6	12.6	12.6	12.6	47	12.6	-
Water Heat Transfer Coefficient, Btu/ft ² -hr-F	1090	493	65.6	-	1090	2700	65.6	-
Overall Heat Transfer Coefficient, Btu/ft ² -hr-F	9.27	8.54	4.41	-	9.27	33.3	4.41	-
Total Surface Area, sq ft	816	8160	1224	10200	816	5000	1224	7100
Tube Diameter, in.	1.875	1.875	1.875	1.875	1.875	0.75	1.875	-
Fin Diameter, in.	3.375	3.375	3.375	3.375	3.375	1.5	3.375	-
Fin Spacing, per in.	3	3	3	3	3	7	3	
Tube Length, ft	221	2213	332	2766	221	2870	332	3420
Number of Tube Rows	2	20	3	25	2	1	3	6
Gas Pressure Loss, in. H ₂ O	0.34	3.36	0.50	4.2	0.34	5.8	0.50	6.6
Water Pressure Loss, psia	2.29	6.27	2.06	10.52	2.3	3.5	2.1	7.9

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SKETCH OF BASELINE WASTE-HEAT EXCHANGER

DESIGN HEAT LOAD = 9.6×10^6 BTU/HR

ALL TEMPERATURES IN F

ALL DIMENSIONS IN FT.

FIG. 7

the superheater and into a plenum beneath the fluidized bed. The fluidized bed vaporizer has been separated into two different sections in parallel to minimize the total enclosed volume of the heat exchanger. The thermal performance of this split bed configuration is identical to that of a single contiguous bed. After passing through the fluidized bed the exhaust gases are collected and exit through the preheater on top of the heat exchanger. Overall specifications are presented in TABLE III for comparison with those of the original MTI design. Use of a fluidized bed heat exchanger reduces the total surface area requirements by 30 percent to 7100 sq ft.

The exhaust gases enter the heat exchanger through the superheater which cools them from 520 F to 506 F. The superheater is identical to the original MTI design. After exiting the superheater the exhaust gases enter the vaporizer section of the heat exchanger.

Criteria for the vaporizer design was to duplicate the Fluidfire design as close as possible. Distributor design and superficial velocity are identical to that used in the Fluidfire unit. The tubes are the closest standard size to those in the Fluidfire unit and the same bed material and particle size is used. The overall heat transfer coefficient was calculated using the measured value of the bed-to-fin heat transfer coefficient (after correction for the change in gas temperature from the experimental conditions to the design condition), a calculated fin effectiveness of 84 percent, and an inside boiling heat transfer coefficient of 2700 BTU/ft²-hr-F calculated with the method of Chen (Ref.24). The number of tubes (ninety-five) and their length (30 ft) was selected based upon a calculated water side pressure loss using the method of Martinelli (Ref.25). The resulting overall heat transfer coefficient is 33 BTU/ft²-hr-F.

The very low gas velocity (1.8 ft/sec) used in this design results in a large distributor and bed area, producing a large overall unit volume. Several construction techniques for the heat exchanger enclosure were considered: poured concrete, poured concrete with a metal roof, all metal with prefabricated buildings. Configurations with both a contiguous bed and split beds were considered. The lowest cost configuration was found to be poured concrete with a metal roof and the fluidized bed split into two sections as shown in Fig. 7. For this configuration thermal losses in the plenum are estimated to be 3 F, lowering the exhaust gas temperature entering the bed. Total vaporizer heat transfer surface area, based on this number is 5000 sq ft, a 39 percent reduction from the original 8160 sq ft in the MTI design. Detailed design calculations are presented in APPENDIX II.

After exiting the fluidized bed the exhaust gases rise through the freeboard for particle de-entrainment. The correlation of Soroko (Ref.26) was used to confirm the transport disengagement height used in the Fluidfire unit. The gases are then collected and exit through the preheater. Temperature loss in the freeboard is approximately 2 F. Gas temperature entering the freeboard is 340 F, 6 F below

the value in the original MTI design. The preheater is identical to the original MTI design except for an 8 percent area increase to account for this temperature loss.

Total cost for this unit is estimated to be \$58,000 as shown in APPENDIX II. The cost is 20 percent higher than the estimated cost for the MTI unit. This increase in cost may be compensated for to some degree by the improved fouling characteristics of a major portion of the total heat exchanger system. However, since at least a portion of the heat exchanger system is not immersed in fluidized beds, fouling resistance has probably not been improved enough to justify the initial cost increase.

Alternative Design Configuration

In the baseline conceptual design study it became apparent that a fluidized bed heat exchanger system would be much more attractive if all the heat exchanger surfaces could be immersed in fluidized beds for reduced fouling and if the overall initial costs could be reduced. The alternative design study was undertaken to identify a design concept which would have both these features.

The most important modification is to divide the fluidized bed into two separate stages in series. Thus two separate temperature zones would be created, allowing all heat transfer surfaces to be immersed in fluidized beds and reducing heat transfer surface area requirements. Figure 8 schematically illustrates one possible implementation of this concept showing gas flow paths and temperature levels. Actual hardware design would look similar to the baseline design illustrated in Fig. 7. The exhaust gas flow is split upon entering the heat exchanger with approximately 30 percent of the flow passing through a fluidized bed superheater and the balance through a first stage fluidized bed vaporizer. The superheater bed temperature is 490 F and the vaporizer bed temperature 366 F. The exhaust gases then pass through a second fluidized bed with additional vaporizer and preheater stages. The gases from the first bed were assumed to mix before entering the second bed. Second stage vaporizer bed temperature is 342 F and the preheater bed temperature is 303 F. The final gas exit temperature is 333 F. Total surface area requirement is 5600 sq ft, a reduction of 50 percent from the MTI design and a reduction of 20 percent from the baseline design. At this time only a preliminary analysis has been performed on the two-stage heat exchanger concept. Further analysis concerning flow distribution and stage temperature levels may significantly improve its estimated performance.

The primary technical problem with this design concerns the overall pressure loss. The pressure loss of the Fluidfire heat exchanger is 5.8 in. H₂O. If two Fluidfire type heat exchangers were used in series, the pressure loss would double

THE TWO STAGE FLUIDIZED BED HEAT EXCHANGER

ALL TEMPERATURES IN F

to 11.6 in. H_2O . This is probably in excess of the acceptable pressure loss for diesel exhaust waste heat recovery. Fortunately, this pressure loss is also judged to be considerably greater than that required to maintain acceptable fluidization quality.

The pressure loss of the distributor in the Fluidfire unit is 1.5 times that of the bed alone and approximately 19 times that of the dynamic head in the inlet plenum. This is in excess of the requirements identified by Glicksman (Ref. 4) where good fluidization was obtained with the distributor pressure loss as low as 50 percent of the bed pressure loss.

A further reduction in total pressure loss can be realized by using a thinner fluidized bed. In the baseline design, sufficient bed material was used to completely immerse the finned tubes in the packed bed. When fluidized, the bed expands as much as 60 percent in thickness. From a heat transfer effectiveness viewpoint, it is only necessary to immerse the heat transfer surfaces in the fluidized bed and any bed material above the heat transfer surface is of little value. Consequently, a thinner bed might be used for a further reduction in pressure loss while maintaining the same heat transfer rate. With these two changes, it is estimated that the pressure loss through the two-stage bed may be reduced to approximately 4 in. of H_2O , the same as the MTI design.

Additional technical improvements to the baseline design have been identified which will significantly reduce the heat exchanger cost. These improvements include increasing the superficial velocity, reducing the cost of the gas distributor, and increasing the heat transfer coefficient. The most important of these improvements is to increase the superficial velocity. The gas velocity in the Fluidfire unit is 1.8 ft/sec, over an order of magnitude less than that used in conventional heat exchangers. This translates directly to increased unit size and cost. In the alternative design configuration it is assumed that this velocity can be increased by a factor of two. Gas flow cross-sectional area is thereby reduced from 576 sq ft to 288 sq ft.

The technical risk in this assumption concerns the freeboard requirement above the bed to separate particles from the gas (termed the transport disengagement height (TDH)). This requirement was calculated to be one foot using the correlation of Soroko (Ref. 26). However, this correlation is based on limited data and its utilization presents a risk. If further investigation shows the TDH to be greater than this correlation predicts, then larger or denser particles must be employed to reduce the TDH.

In the alternative design it was assumed that the overall heat transfer coefficient could be increased 30 percent over the value obtained in the Fluidfire unit. This assumption was based on the comparison of measured values with values calculated from correlations in the literature (Fig. 4), together with the assumption that an alternative heat transfer surface could be found which would not have the large unfluidized zones above the tube. Both of these assumptions have the effect of increasing the heat transfer coefficient. Additionally, the alternative design assumes that a lower cost distributor can be used.

The cost of this alternative design is estimated to be \$42,000, 28 percent less than the estimated cost of the baseline design. The assumptions used in arriving at this design and cost estimate are believed to be conservative. However, experimental verification of these assumptions is required. This alternative design is not an optimized configuration and further parametric analyses and tradeoff studies could result in additional cost reductions.

The results of the alternative conceptual design indicate that a fluidized bed heat exchanger with all heat transfer surface immersed in fluidized beds is possible. This design has a reduced initial cost when compared to a conventional nonfluidized bed design. Additionally, the improved fouling properties of fluidized beds should further reduce the life cycle costs of the heat exchanger. Further efforts should therefore pursue evaluating the features incorporated into the alternative conceptual design.

CONCLUDING REMARKS

Fluidized beds are a viable technical option for waste-heat heat exchangers. They offer the potential advantages of improved surface fouling characteristics at a reduced initial cost as compared to conventional finned tube heat exchangers. These features could result in significantly reduced life-cycle costs. However, the technology of fluidized bed heat exchangers requires further development. In particular, the multistage fluidized bed concept must be validated. Multiple stages, when configured in counterflow, can be utilized to either provide a higher thermal effectiveness than a single-stage unit or to reduce the heat transfer surface area requirements for a given thermal effectiveness. Specific problems in developing this concept concern actual configurations for its implementation and reducing pressure losses to an acceptable level.

Detailed parametric performance analyses are also required to develop multi-stage fluidized bed waste-heat heat exchangers. In particular, flow division between sections of a particular stage and temperatures for each section must be optimized to improve heat exchanger performance and cost. Additionally, trade-off studies should be conducted to investigate the many physical characteristic options available. Areas of interest include the heat transfer surface configurations, bed material and its size distribution, as well as the superficial velocity. Preliminary indications are that these studies should produce fluidized bed waste-heat heat exchanger designs with significantly improved performance and reduced initial and life cycle costs when compared to conventional finned tube designs.

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APPENDIX I: DATA UNCERTAINTY ANALYSIS

Presented herein is an uncertainty analysis for a single data point obtained at approximately the nominal operating condition (Point No. 2 in TABLE II). The standard method of uncertainty analysis (Ref. 26) is used with the bias and precision error calculated separately. The total uncertainty is presented at the 95 percent confidence level (bias plus twice the precision). Measured values, with their estimated bias and uncertainty, are presented in TABLE AI-1.

Uncertainty in Water Flow Rate

The water flow rate for each rotameter is calculated from a linear regression analysis of the calibration data. This analysis is summarized below.

<u>Rotameter</u>	<u>Weight Flow (lbm/min)</u>	<u>r</u>	<u>n</u>	<u>S.R.</u>	<u>Weight Flow</u>	<u>Precision</u>
No. 1	0.567 + 0.633 (S.R.) C.F.	0.99996	4	79	51.4	0.27
No. 2	-0.27 + 0.795 (S.R.) C.F.	0.99830	7	78	62.8	1.11
No. 3	0.74 + 1.466 (S.R.) C.F.	0.99981	14	77	115.5	0.62

Where the precision was calculated by using:

$$\frac{t_n}{t_\infty} = \frac{(n-1)}{(n-2)} \frac{S_y^2 (1-r^2)}{\left[1 + \frac{1}{n} + \frac{(X - \bar{x})^2}{(n-1)S_x^2} \right]^{1/2}}$$

This precision in the calibration of the rotameters represents a bias in each rotameter reading. Using these numbers, the individual uncertainties in the weight flow from each rotameter in pounds per minute are:

TABLE AI-1

UNCERTAINTY IN MEASURED VALUES

<u>PARAMETER</u>	<u>VALUE</u>	<u>BIAS</u>	<u>PRECISION</u>	<u>TOTAL UNCERTAINTY</u>
Pressure:				
Barometer, in Hg	29.84	0.01	0.02	0.05 (0.16%)
Freeboard, in H ₂ O	7.3	0.1	0.2	0.5 (6.8 %)
Distributor/Bed				
Combination, in H ₂ O	5.8	0.1	0.2	0.5 (8.6 %)
Transverse to Elbow, in H ₂ O	0.29	0.01	0.01	0.03 (10.3 %)
Loss of Elbow, in H ₂ O	0.76	0.01	0.01	0.03 (3.9 %)
Temperature: F				
Above Bed	243.3	1	1	3
Heater Inlet	235.3	0.5	0.1	0.7
Bed Inlet	422.7	2	2	6
Water Inlet	162.2	0.3	0.1	0.5
Water Outlet	178.8	0.3	0.1	0.5
Water Flow Rate: Percent of Full Scale Reading				
Rotameter No. 1	79	0	0.5	1.5 (1.9 %)
Rotameter No. 2	78	0	0.5	1.5 (1.9 %)
Rotameter No. 3	77	0	0.5	1.5 (1.9 %)
Wattmeter:	487	1.25	2	5.3 (1.1 %)
Absolute Humidity: gr/lb	34	0	2	4 (12 %)

Rotameter	Weight Flow	Scale Bias	Calibration Bias (2x Precision)	Total Bias	Precision	Total Uncertainty
No. 1	51.4	0.26	0.54	0.60	0.26	1.11 (2.2%)
No. 2	62.8	0.31	2.22	2.24	0.31	2.87 (4.6%)
No. 3	115.5	0.58	1.24	1.37	0.58	2.53 (2.2%)

The best estimate of the water flow rate was obtained by taking the sum of the readings divided by two (the two small rotameters are in parallel). This results in a flow rate of 114.9 lb/min. Using the common scale bias, the total uncertainty becomes 2.1 lb/min or 1.9 percent.

Uncertainty in Air Flow Rate Measurement

The air flow rate was determined by measuring the pressure drop in a 180° elbow and the transverse pressure within the elbow. Both of these procedures require use of the specific volume. Specific volume is calculated using:

$$v = \frac{0.754 (T + 460)}{P} \left[1 + \frac{W}{4360} \right]$$

and the uncertainty is:

Parameter	Value	Bias	Precision	Total Uncertainty	$\partial v / \partial ()$
P, in Hg	29.39	0.02 (0.07%)	0.04 (0.14%)	0.10 (0.34%)	v/P
T, F	243.3	1	1 (0.14%)	3 (0.4 %)	v/T
W, gr/lbm	34	0 (0%)	2 (5.9 %)	4 (11.8 %)	$0.754(T+460)/4360P$
Calculation	-----	(0.1%)	-----	(0.1 %)	1
v	18.2	0.034 (0.19%)	0.037 (0.20%)	0.1 (0.55%)	---

For Air Flow Rate:

$$\dot{M} = C(\Delta P/v)^{1/2}$$

Parameter	Value	Bias	Precision	Total Uncertainty	$\partial M / \partial ()$
v, ft ³ /lbm	18.2	0.034 (0.19%)	0.037 (0.20%)	0.1 (0.55%)	$M/2v$
Calculation	----	0.1%	----	0.1%	M
Flow Nozzle	----	1%	----	1%	M

For Pressure Loss Measurement:

<u>Parameter</u>	<u>Value</u>	<u>Bias</u>	<u>Precision</u>	<u>Total Uncertainty</u>	<u>$\partial M / \partial ()$</u>
P, in H ₂ O	0.76	0.01 (1.4%)	0.01 (1.4%)	0.03 (3.9%)	M/2 p
C	3.56	0.26 (7.42%)	----	0.26 (7.4%)	M/k
M, lbm/sec	0.743	0.055 (7.5%)	0.0005 (0.7%)	0.065 (8.9%)	---

For Transverse Pressure Measurement:

<u>Parameter</u>	<u>Value</u>	<u>Bias</u>	<u>Precision</u>	<u>Total Uncertainty</u>	<u>$\partial M / \partial ()$</u>
P, in H ₂ O	0.29	0.01 (3.4%)	0.01 (3.4%)	0.03 (10.2%)	M/2 p
C	5.67	0.29 (5.2%)	----	0.29 (5.2%)	M/C
M, lbm/sec	0.728	0.040 (5.6%)	0.012 (1.7%)	0.065 (9.0%)	---
Mavg, lbm/sec	0.735	0.035 (4.7%)	0.007 (0.9%)	0.049 (6.5%)	---

where Mavg is the average flow rate from the two independent measurements.

Uncertainty in Heat Addition by the Heater

The total amount of heat added by the heater was calculated by two different methods: through measurement of the electrical power and the heat received by the air. The uncertainty in the measurement by each method is presented below:

By Air:

$$Q = \dot{M} C_p (T_{HO} - T_{HI})$$

<u>Parameter</u>	<u>Value</u>	<u>Bias</u>	<u>Precision</u>	<u>Total Uncertainty</u>	<u>$\partial Q / \partial ()$</u>
M, lbm/sec	0.735	0.035 (4.6%)	0.007 (0.9%)	0.049 (6.5%)	Q/M
Cp, Btu/lbm-F	0.243	----- (0%)	0.0019 (0.8%)	0.0038 (1.6%)	Q/Cp
T _{HO} - T _{HI} , F	234.9	1.5 (0.64%)	2 (0.85%)	5.5 (2.3%)	Q/(T _{HO} - T _{HI})
Q, Btu/hr	120,500	5715 (4.7%)	1776 (1.5%)	9200 (7%)	----

By Electrical Power:

<u>Parameter</u>	<u>Value</u>	<u>Bias</u>	<u>Precision</u>	<u>Total Uncertainty</u>	<u>$\partial Q / \partial ()$</u>
Meter	487	1.25 (0.25%)	2 (0.41%)	5.3 (1.2%)	1
Current					
Transformer	---	(1%)		(1%)	1
Voltage					
Transformer	---	(0.3%)		(0.3%)	1
Q, Btu/hr	126,400	1640 (1.07%)	520 (0.41%)	2400 (1.9%)	

The difference between these two methods of calculating heat addition could be expected to be as great as 8600 Btu/hr (7.9%) in the absence of thermal losses. Actual deviation is 5900 BTU/hr (4.7%).

Uncertainty in Heat Transferred by the Fluidized Bed

Again, the total amount of heat transferred was calculated by using two separate methods: by the heat lost from the air and by the heat gain of the water.

By Air:

$$Q = \dot{M}C_p(T_{HO} - T_B)$$

<u>Parameter</u>	<u>Value</u>	<u>Bias</u>	<u>Precision</u>	<u>Total Uncertainty</u>	<u>$\partial Q/\partial ()$</u>
M, lbm/sec	0.735	0.035 (4.7%)	0.007 (0.9)	0.049 (6.5%)	Q/M
Cp, Btu/lbm-F	0.242	---	0.0019 (0.8%)	0.0038 (1.6%)	Q/Cp
T _{HO} -T _B , F	<u>176.9</u>	<u>3.2 (1.8%)</u>	<u>2.3 (1.2%)</u>	<u>7.8 (4.2%)</u>	Q/(T _B -T _{HO})
Q, Btu/hr	113,000	5690 (5.0%)	1920 (1.7%)	9500 (8.4%)	

By Water:

$$Q = \dot{M}C_p(T_{WO} - T_{WI})$$

<u>Parameter</u>	<u>Value</u>	<u>Bias</u>	<u>Precision</u>	<u>Total Uncertainty</u>	<u>$\partial Q/\partial ()$</u>
M, lbm/min	114.9	1.42 (1.2%)	0.35 (0.3%)	2.1 (1.8%)	Q/M
Cp, Btu/lbm-F	1	0	0.002 (0.2%)	0.004 (0.4%)	Q/Cp
T _{WO} -T _{WI} , F	<u>16.6</u>	<u>0.1 (0.6%)</u>	<u>0.14 (0.85%)</u>	<u>0.38 (2.3%)</u>	Q/(T _{NO} -T _{WI})
Q, Btu/hr	114,400	1530 (1.3%)	1060 (0.92%)	3600 (3.1%)	

and the difference could be expected to be as great as 10,000 Btu/hr (9.0%) in the absence of thermal losses. Actual difference was 1400 BTU/hr (1.2%).

Uncertainty in the Heat Transfer Coefficient

The heat transfer coefficient was calculated by two separate methods, one using the water temperature rise and the second using the air temperature loss. Calculating the former first:

$$U = - \frac{\dot{M}_w C_p}{A} \ln \left[1 - \frac{T_{WO} - T_{WI}}{T_B - T_{WI}} \right]$$

Parameter	Value	Bias	Precision	Total Uncertainty	$\partial U / \partial ()$
M, lbm/min	114.9	1.42 (1.2%)	0.35 (0.3%)	2.1 (1.8%)	U/M
Cp, Btu/lbm-F	1	---	0.002 (0.2%)	0.004 (0.4%)	U/Cp
A, ft ²	50	1.2 (2.4%)	---	1.2 (2.4%)	U/A
T _{WO} -T _{WI} , F	16.6	0.1 (0.6%)	0.14 (0.85%)	0.38 (2.3%)	1.12U/(T _W -T _{WI})
T _B -T _{WI} , F	83.8	2 (2.5%)	1 (1.2%)	4 (5%)	1.12U/(T _B -T _{WI})
U	30.4	1.2 (3.9%)	0.51 (1.7%)	3600 (7.3%)	

and utilizing the temperature loss of the air:

$$U = - \frac{\dot{M}_w C_{pw}}{A} \ln \left[1 - \frac{\dot{M}_A C_{pa} / \dot{M}_w C_{pw}}{\frac{1}{\epsilon} - 1} \right]$$

where:

$$\epsilon = \frac{T_{HO} - T_B}{T_{HO} - T_{WI}}$$

Parameter	Value	Bias	Precision	Total Uncertainty	$\partial U / \partial ()$
T _{HO} -T _{WI} , F	260.5	2.0 (0.77%)	2.0 (0.77%)	6.0 (2.3%)	$\epsilon / (T_{HO} - T_{WI})$
T _{HO} -T _B , F	176.9	3.2 (1.8%)	2.3 (1.2%)	7.8 (4.2%)	$\epsilon / (T_{HO} - T_{WI})$
	0.68	0.013 (2.0%)	0.0097 (1.4%)	0.032 (6.8%)	0.77 U/ ϵ
Ma, lbm/sec	0.735	0.035 (4.7%)	0.007 (0.9%)	0.049 (6.5%)	1.12 U/M
C _{pa} , Btu/lbm-F	0.243	---	0.0019 (0.8%)	0.0038 (1.6%)	1.12 U/C _{pa}
M _w , lbm/min	114.29	1.42 (1.2%)	0.35 (0.3%)	2.1 (1.8%)	2.12 U/M
C _{pw} , Btu/lbm-F	1	0	0.002 (0.2%)	0.004 (0.4%)	2.12 U/C _{pw}
A, ft ²	50	1.2 (2.4%)	---	1.2 (2.4%)	U/A
U, Btu/ft ² -hr-F	30.4	2.0 (6.5%)	0.58 (1.9%)	3.1 (10.3%)	

The final reported heat transfer coefficient is the average of these two values with a total uncertainty of 1.9 BTU/ft²-hr-F (6.3%).

The heat transfer coefficient of final interest is that for the fluidized bed to the immersed surfaces. The correlation of Nusselt:

$$Nu = 0.036 Re^{0.8} Pr^{1/3} (\mu/\mu_s)^{0.14} (d/l)^{0.0556}$$

was used to calculate the water film resistance, compensating for both the temperature difference between the tube and the bulk fluid and the tube length. The fin effectiveness was calculated by the method of Gardner (Ref. 28). Bed to surface heat transfer coefficient was then calculated from:

$$h_B = \frac{1}{n \left[\frac{1}{U} - \frac{A_F}{A_i h_w} - \frac{A_F r_o}{A_i} \ln \frac{r_o}{r_i} \right]}$$

The principal factor in the uncertainty in this parameter is the unknown bias of the water film correlation. No estimate was made for this bias.

The uncertainties presented above depend upon actual measured values and will differ for each data point. The greatest dependence is upon air flow rate, increasing dramatically, with a reduction in air flow rate.

Nomenclature

A	Total heat transfer surface area, sq ft
A_F	Total tube outside area, sq ft
A_i	Total tube inside area, sq ft
C	Calibration constant
C_p	Specific heat, BTU/lbm-F
C.F.	Rotameter density correction factor, unitless
d	Tube diameter, ft
h_B	Bed-to-tube heat transfer coefficient, BTU/ft ² -hr-F
h_w	Water-to-tube heat transfer coefficient, BTU/ft ² -hr-F
l	Tube length, ft
$\dot{M}$	Mass flow, lbm/sec or lbm/min
n	Number of data points
Nu	Nusselt number
P	Pressure, in H ₂ O or in Hg
Pr	Prandtl number

Q	Total heat transfer rate, BTU/hr
r_i	Inside tube radius, ft
r_o	Outside tube radius, ft
r	Linear regression correlation coefficient
Re	Reynolds number
$S.R.$	Rotameter scale reading
S_x^2	Variance of independent variable
S_y^2	Variance of dependent variable
T	Temperature, F
t_n	Student's t parameter based on n degrees of freedom
t_∞	Student's t parameter based on infinite degrees of freedom
U	Overall heat transfer coefficient, BTU/ft ² -hr-F
v	Specific volume, ft ³ /lbm
W	Specific humidity, gr/lbm
X	Value of independent variable
$\bar{x}$	Average of independent variable

Greek Symbols

ΔP	Pressure loss, in H ₂ O
ϵ	Thermal effectiveness, $(T_{HO}-T_B)/(T_{HO}-T_{WI})$
η	Fin effectiveness
μ	Viscosity of water evaluated at bulk water temperature, lbm/ft-hr
μ_s	Viscosity of water evaluated at tube surface temperature, lbm/ft-hr

Subscripts

a	Air
avg	Average
B	Bed
HI	Heater inlet
HO	Heater outlet
w	Water

APPENDIX II: DESIGN CALCULATIONS

Supporting calculations for the design of a 9.6×10^6 BTU/hr fluidized bed heat exchanger are presented herein. Highlights of these calculations have been presented in the body of this report.

Heat Losses

Due to the changing gas velocities, the heat transfer coefficient varies throughout the plenum and freeboard region. The approach taken here was to calculate the heat transfer coefficient at the entrance condition (using forced convection) and at the exit condition (using free convection) and then average them to obtain an effective heat transfer coefficient. Eight inches of concrete was assumed for all cases with 50 F ground water for below grade surfaces and zero F no wind conditions outside above grade walls (free convection for outside heat transfer coefficient). The overall heat loss was calculated separately for each plenum and used to calculate the equivalent temperature loss of the air. Temperature loss for both plenums was 2.5 F for a vaporizer inlet temperature of 502.8 F.

Heat Transfer Surface Area Requirement:

$$\epsilon = \frac{T_{B1} - T_B}{T_{B1} - T_{M1}} = 0.765$$

for $C_{MIN}/C_{MAX} = 0$

$$NTU = \frac{\epsilon}{1-\epsilon} = 3.25 \quad (\text{Ref. 8})$$

$$C_{MIN} = \frac{Q}{\Delta T} = 51,300 \frac{\text{BTU}}{\text{hr-F}}$$

$$UA = NTU * C_{MIN} = 167,000 \text{ Btu/hr-F}$$

Calculating Distributor Area

In the Fluidfire unit the superficial gas velocity, based on distributor open area, is 2.76 ft/sec. However, the total cross section of the bed considerably exceeds the distributor open area, resulting in a superficial velocity of 1.8 ft/sec. For this design, an intermediate value of 2 ft/sec will be used, resulting in a total distributor area of 576 ft².

Calculation of the Overall Heat Transfer Coefficient

Tube specifications: Outside Diameter - 0.75 in.
 Fin Depth - 0.375 in.
 Fin Thickness - 0.012 in.
 Fin Spacing - 7 in.

The outside heat transfer coefficient is:

$$h = Nu = \frac{k}{d} = 46.8 \text{ BTU/ft}^2\text{-hr-F}$$

where the experimentally obtained Nusselt number of 4.7 was used. The fin effectiveness was calculated to be 84 percent (Ref.28). The inside heat transfer coefficient can be calculated using the method of Chen (Ref.24). Assuming 96 tubes:

$$G = \frac{\dot{m}}{A} = 58.3 \text{ lbm/ft}^2\text{-sec}$$

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g}{\rho_f} \right)^{0.5} \left(\frac{\mu_f}{\mu_g} \right) = 0.47$$

$$F = 4.5$$

$$h_c = 0.023 \left[\frac{G(1-x)d_T}{\mu_f} \right]^{0.8} Pr^{0.4} \frac{k_f}{d} F = 2440 \text{ Btu/ft}^2\text{-hr-F}$$

$$Re_{TP} = \frac{G(1-x)d_T F^{1.25}}{\mu} = 143,500$$

$$S = 0.3$$

$$h_{NcB} = 0.00122 \frac{k_F^{0.79} C_{pf}^{0.45} \rho_f^{0.49}}{\sigma^{0.5} \mu_f^{0.29} \rho_g^{0.24} \rho_g^{0.24}} \Delta T_{sat}^{0.24} \Delta P_{sat}^{0.75} S = 234 \text{ Btu/ft}^2\text{-hr-F}$$

where $T_{sat} = 7.3 \text{ F}$ and $P_{sat} = 1009$ from iteration

$$h_{tp} = h_c + h_{NcB} = 2680 \text{ Btu/ft}^2\text{-hr-F}$$

$$U = 33.3 \text{ Btu/ft}^2\text{-hr-F}$$

tube length is therefore:

$$L_T = \frac{UA}{U(A/L)_t} = 30 \text{ ft.}$$

The pressure drop can be calculated using the method of Martinelli and Nelson (Ref. 24).

$$X^2 = \left(\frac{Re_g}{Re_f} \right)^{0.2} \left(\frac{\mu_f}{\mu_g} \right)^2 \frac{C_{fvf}}{C_{vg}} = 0.3$$

$$\phi^2 = 39$$

$$\Delta P = \phi^2 f \frac{L_T}{d_T} \frac{G_{vf}^2}{2g_o} = 3.5 \text{ psia}$$

Freeboard Heat Loss

The total heat loss through the freeboard region was estimated by the same method used to estimate the heat losses in the plenum region. Overall temperature loss of the air was 3.5 F, resulting in a 339.5 F preheater inlet temperature. Thus the log mean temperature difference in the preheater is 8 percent less than in the MTI design.

Nomenclature

A	Area, sq ft
C_{MIN}	Minimum thermal capability, BTU/hr-F
C_p	Specific heat, BTU/lbm-F

d	Diameter, ft
F	Factor in Chen correlation
f	Friction factor
G	Mass velocity, $\text{lbm/ft}^2\text{-sec}$
g_0	Gravitational constant, $32.2 \text{ lbm-ft/lbf-sec}^2$
h	Heat transfer coefficient, $\text{BTU/ft}^2\text{-hr-F}$
i	Latent heat of vaporization, BTU/lbm
k	Thermal conductivity, BTU/ft-hr-F
L_T	Tube length, ft
NTU	Number of transfer units, UA/C_{MIN}
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
S	Suppression factor of Chen
U	Overall heat transfer coefficient, $\text{BTU/ft}^2\text{-hr-F}$
v	Specific volume, ft^3/lbm
x	Mass vapor quality
X	Martinelli Parameter
X_{tt}	Parameter in correlation of Chen

Greek Symbols

ΔP_{sat}	Pressure rise due to ΔT_{sat} , lbf/ft^2
ΔT_{sat}	Wall superheat at nucleation, F
ϵ	Thermal effectiveness $(T_{B1}-T_B)/(T_{B1}-T_{M1})$

ρ	Density, lbm/ft ³
σ	Surface tension, lbf/ft
μ	Viscosity, lbm/ft-hr
ϕ	Martinelli pressure drop multiplier

Subscripts

B	Bed
Bl	Bed inlet
c	Convective
f	Fluid
g	Gas
Ml	Water inlet
NcB	Nucleate boiling
T	Tube
TP	Two-phase
P	Particle

APPENDIX III: COST ESTIMATE FOR BASELINE DESIGN CONFIGURATION

Presented in the following TABLE is the detailed cost estimate for the heat exchanger. The total cost is \$58,400 and is presented in five categories: Construction; Superheater; Preheater; Steam Separating Drum/Circulating Pump; and, the Vaporizer. Nearly \$20,000 of the total cost results in the use of MTI designed components (Superheater, Vaporizer, Steam Separating Drum/Circulating Pump).

COST ESTIMATE FOR BASELINE DESIGN CONFIGURATION

<u>Item</u>	<u>Quantity</u>	<u>Unit Cost</u>	<u>Cost</u>	<u>Ref.</u>
I. Construction				
A. Site Preparation				
1. Excavation	200 cu yd	\$ 0.90	\$ 180.	29+20%
2. Curtain Drains	100 linear ft	2.50	250.	29+20%
3. Drain Line	50 linear ft	6.00	300.	29+20%
4. Backfill	80 cu yd	1.20	100.	29+20%
SUBTOTAL - SITE PREPARATION			\$ 830.	
B. Building				
1. Cast Walls	40 cu yd	225..	9,000.	29+20%
2. Bottom Slab	8 cu yd	55.	440.	29+20%
3. 17 4" I-Beams 10'4" long	1800 lb	0.30	540.	30
4. 1/8" Steel Plate	3000 lb	0.30	900.	31
5. Roof/Divider Installation	100 Mhr	20.	2,000.	32
6. Preheater Installation	25 Mhr	20.	500.	32
SUBTOTAL - BUILDING			\$13,300.	
SUBTOTAL - CONSTRUCTION			\$14,230	
II. Superheater	LS	3,000.	\$ 3,800.	26
III. Preheater	LS	2,700.	\$ 2,700.	26+8%
IV. Steam Separating Drum/ Circulating Pump System	LS	13,300.	\$13,300.	26
V. Vaporizer				
A. Finned Cooper Tubing				
3/4" OD, 1.5" OD fins x 30'	2870 ft	2.65	7,600.	30
B. Manifolds	600 lb	1.00	600.	32
C. Water Feed Lines	300 lb	1.00	300.	32
D. Tube Assembly				
1. Drill Manifolds	192	2.	380.	32
2. Weld Tubes	192	12.	2,300.	32
3. Miscellaneous Fittings	LS	600.	600.	32
4. Frame Handling	100 Mhr	20.	2,000.	32
5. Steel Plate	3000 lb	0.30	900.	31
6. Distributor	24 sht	240.	5,760.	30
7. Welding of Distributor	80 Mhr	25.	2,000.	32
8. Alumina Powder	7000 lb	0.278	1,950.	30
SUBTOTAL VAPORIZER			\$24,390.	
TOTAL			\$58,400.	