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ENHANCEMENT OF HEAT TRANSFER IN WASTE-HEAT
HEAT EXCHANGERS

Final Report

By
Richard C. Stoeffler

July 1980

Work Performed Under Contract No. AC03-77ET11348

United Technologies Research Center
East Hartford, Connecticut

U. S. DEPARTMENT OF ENERGY

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Enhancement of Heat Transfer
in Waste-Heat Heat Exchangers

Contract DEAC03-77-ET11348

Final Report

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DATE July 1980

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SUMMARY

The Fluidfire shallow fluidized bed heat transfer facility was modified during this program to give increased air flow capacity and to allow testing with different distributor plates and with two-stage heat exchangers. Tests were conducted using this heat transfer facility to investigate the effect of reduced distributor plate pressure loss and amount and type of bed material on the heat transfer performance of a single-stage fluidized bed heat exchanger. Elutriation from the bed was measured for different bed materials and distributor plates; alternate heat exchanger surfaces having different fin spacings were also tested. Two types of two-stage fluidized bed heat exchangers were tested: one having a baffle (having almost no pressure loss) located between the stages and which allowed bed material to recirculate between upper and lower beds; the second having two distributor plates in series with no recirculation of the bed material.

The results obtained in the experimental program were used in concept design studies of multi-stage fluidized bed heat exchangers for waste heat recovery from diesel engine exhaust gases. Information was obtained from the literature and from diesel engine manufacturers to determine allowable diesel engine operating back pressures. The costs were estimated for two- and three-stage designs and were compared with costs obtained previously for single-stage fluidized bed and conventional heat exchanger designs.

CONCLUSIONS

Fluidized bed heat exchangers continue to appear viable. Staging of fluidized bed heat exchangers is one feature which improves their viability. In this program several valuable technical accomplishments were achieved which are important for implementing staged fluidized bed heat exchangers; they are:

1. A two-stage heat exchanger with two distributor plates in series was operated successfully. For the test duration (six hours), plugging of the upper plate by material from the lower did not occur and the performance of the heat exchangers in the two-stage configuration was equivalent to or slightly better than their individual performance.
2. Conceptual design studies were performed which indicate that two- and three-stage fluidized bed heat exchangers can be fabricated for less cost, approximately 20 and 37 percent, respectively, than for a single-stage fluidized bed heat exchanger. The studies also indicate that the costs for the two- and three-stage fluidized bed heat exchangers are 15 and 30 percent less than the cost of a conventional heat exchanger.

These major conclusions are supported by tests performed in this program which showed that:

1. Distributor pressure loss can be decreased to 50 to 85 percent of the bed pressure loss without decreasing heat transfer performance.
2. Within a limited range of variables Al_2O_3 is a suitable bed material. Heat transfer performance obtained using SiC was equivalent to that obtained using Al_2O_3 and bed pressure losses were also approximately the same, thus SiC appears no better than Al_2O_3 . Performance with steel shot was up to 70 percent better; however, the steel shot bed pressure loss was larger than that of Al_2O_3 by a factor of 2 to 2.5.
3. The transport disengagement heights (TDH) determined from elutriation measurements were approximately the same for Al_2O_3 and SiC bed materials. TDH was less for the heavier steel shot. The TDH is a function of the distributor plate, but is low enough to allow reasonable heat exchanger packaging.
4. A baffle was successfully used to separate the fluidized beds in a two-stage heat exchanger; however, in these tests the rate of recirculation of bed material between the beds was high, reducing the effectiveness of staging with the baffle. Thus, until further testing is performed, staged heat exchangers must be designed with separate distributor plates.

INTRODUCTION

Many power generating and industrial processes currently discharge thermal energy into the environment. This energy is often at temperatures which are compatible with power generation or residential or commercial building heating requirements. Considering the energy situation in the United States at present, efficient utilization of waste heat will be a future requirement.

One reason that waste heat is not used more extensively is the high cost of the waste heat exchangers. Even a small decrease in the cost of these heat exchangers will improve the economics of waste heat recovery. 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: (1) increasing the gas velocity in the heat exchanger; and (2) use of extended surfaces which tend to enhance heat transfer coefficients. The former leads to increased pressure losses which usually limits the extent to which velocity can be increased in a given heat exchanger. An example of the latter is a pin-fin exchanger. Heat transfer coefficient enhancement is achieved in this device as well as most other commonly used enhancement devices by generation of turbulence; increased pressure loss usually results. Many configurations have been investigated and the results of these investigations reported in the literature. It is expected that these two methods of enhancing heat transfer coefficients will continue to be used. However, alternative concepts for enhancing heat transfer coefficients would be desirable, if they can be shown to have cost or operational advantages.

A concept for enhancing heat transfer coefficients which has not yet been greatly exploited in heat exchangers 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 used extensively as waste heat exchangers because of the large pressure losses normally associated with the deep beds used in the chemical industry. However, shallow fluidized beds such as those sold for waste heat recovery applications by Stone-Platt Fluidfire Ltd. of England are being investigated for use in heat exchangers with reduced pressure loss. Shallow fluidized beds have 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 for predicting the heat transfer coefficient, and, in all cases, experimental verification is required.

In the previous program (Ref. 1), a fluidized bed heat exchanger purchased from Stone-Platt Fluidfire Ltd. was tested. The results of these tests were used to formulate the conceptual design and cost estimate of a fluidized bed heat exchanger to recover approximately 10^7 Btu/hr from the exhaust gases of a diesel engine. The cost of a single fluidized bed heat exchanger was found to be higher than that of a conventional heat exchanger; however, it was also found that a two-stage fluidized bed heat exchanger was competitive with a conventional heat exchanger. Accordingly, in the program described herein, experimental investigations were conducted that would further characterize both single- and multi-stage fluidized bed heat exchangers. These studies included tests with different distributor plates and bed materials to determine the effect of reduced distributor plate pressure loss and amount and type of bed material on heat transfer performance. Transport disengagement height (TDH) from measurements of elutriation from the bed for different distributor plates, and bed materials are discussed. Tests of alternate heat exchangers having different fin spacings, and tests of two types of two-stage fluidized bed heat exchangers are also presented. One of the two-stage units had a baffle (having almost no pressure loss) as the second stage which allowed bed material to recirculate between upper and lower beds. The second two-stage unit had two distributor plates in series with no recirculation of the bed material. The results obtained from these experiments are then used as input for the conceptual design of a multi-stage fluidized bed waste heat recovery heat exchanger.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

The shallow fluidized bed heat exchanger used in the previous program and described in Ref. 1 was modified to increase the air velocity through the heat exchanger and to adapt the heat exchanger housing for testing two-stage heat exchangers. (The fluidized bed heat exchanger, as manufactured by Stone-Platt Fluidfire, Ltd., designed to transfer 140,000 Btu/hr into hot water using 436 F flue gases.) A sketch of the unit used for single-stage tests is shown in Fig. 1 and a schematic drawing showing air and water flow circuits is given in Fig. 2. Sketches of the unit as modified for two-stage baffle tests is shown in Fig. 3, while the unit as used for two fluidized beds in series is shown in Fig. 4.

Air System

Heated air used in these tests to fluidize the bed was circulated by a fan rated at 2600 ACFM at 25 in. H₂O pressure rise and at an air temperature of 250 F. The air was recirculated to minimize heating requirements. This fan was floor mounted and replaced the original 800 ACFM at 6 in. H₂O pressure rise fan which was mounted on top of the heat exchanger. The fan discharge air passes over electric heaters 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. 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 fluidized bed into the freeboard region. Air flow rate, hence, superficial velocity were controlled using the butterfly valve. Air flow rate is determined using a 6-in.-dia orifice plate installed according to ASME standards between the bed exit and the fan inlet. The orifice plate differential pressure was measured using an inclined manometer having a resolution of 0.01 in H₂O.

The pressure loss of the air for the entire fluidized bed heat exchanger was measured for all tests. The bed pressure loss was either measured directly using a pressure probe, or was determined by subtracting the distributor plate pressure loss from the heat exchanger loss. The distributor plate pressure loss was either measured directly or determined from a calibration of the distributor plate pressure loss made without the bed material present. Pressures were measured using inclined or vertical manometers.

The air temperature was measured before it entered the fluidized bed, after leaving the fluidized bed, at the fan inlet and at the heater inlet. The locations for temperature measurement are shown in Fig. 2. All bed inlet flow

temperatures were measured using shielded aspirated thermocouples, whereas the other temperatures were measured using bare thermocouple beads. The thermocouple wire was special limit copper-constantan wire manufactured to a $\pm 3/8$ percent tolerance to NBS calibration standards. Thermocouple outputs were measured with a digital voltmeter. An ice point reference junction was used in all tests. The heat transferred from the air was calculated using measured airflow rates and the measured temperatures.

Water Flow Systems

The heat loss from the air was transferred to water passing through finned tubes located within the fluidized bed. Water was circulated through the tubes using a pump with cold water added to and hot water drained from the flow to maintain a constant water inlet temperature (see Fig. 2). A temperature controlled valve in the cold water make-up line was used to control the rate of cold water addition. The water flow rate was measured with rotameters calibrated by measuring the weight of the water flowing through the rotameters in a known period of time. The temperature of the water was measured using thermocouples and the water flow rate measurements were corrected for water density variations. The temperature rise of the water and water flow rate were used to calculate the heat transferred to the water.

Calculation of Heat Transfer Coefficient

Two analytical models were developed by Suo (Ref. 2) for characterizing the fluidized bed. In one model, the bed is assumed to be completely mixed horizontally, in the other the bed is assumed to be distributed with no horizontal mixing. Measurements made of the horizontal temperature distribution (Ref. 1) in a fluidized bed indicated that mixing in a real fluidized bed is between that of the two models. However, for the test condition of this program ($NTU < 6$ and $C_{min}/C_{max} > 0.2$) heat transfer coefficients predicted using either of the two models differ by less than 1.25 percent. For simplicity, the mixed bed model was assumed in the calculation of the heat exchanger's overall heat transfer coefficient from the measured temperatures and flow rates. The equations used in this calculation are given in APPENDIX I of this report.

The overall heat transfer coefficient, U , was calculated using an average of the heat transferred from the air and that transferred to the water (see APPENDIX I), the heat exchanger water inlet and exit temperatures, and the fluidized bed temperature. The temperature of the fluidized bed was determined by measuring the exit air temperature and correcting this value for the thermal losses in the freeboard region. This correction ranged from 1.5 to 3.5 F depending upon the air flow rate.

The overall heat transfer coefficient, U , results from forced convection between the water and the tube wall (film coefficient, h_w), conduction through the tube wall and fins (thermal conductivity, k), and heat transfer from the fluidized bed to the finned tubes (bed heat transfer coefficient, h_b). The former two were calculated using equations described in APPENDIX I (also see Appendix I of Ref. 1), and were used with the equations for the overall heat transfer coefficient (Eq. (5) in APPENDIX I) to calculate the bed heat transfer coefficient, h_b . The uncertainty (method of calculation given in Ref. 1) in the overall heat transfer coefficient generally varied from 13 percent at V_s of 4 ft/sec to 20 percent at V_s of 1 ft/sec.

DESCRIPTION OF TESTS

Tests were conducted using the single-stage shallow fluidized bed with the twelve-tube Fluidfire heat exchanger surfaces to investigate (1) the effect of reduced distributor plate pressure loss on the performance of the heat exchanger, (2) the effect of amount and type of bed material on the performance of the heat exchanger, and (3) elutriation from the bed for different bed materials and distributor plates. Also, alternate heat exchanger surfaces having different fin spacings and two-stage fluidized bed heat exchangers were tested.

Single-Stage Tests

Distributor Plate Tests

Tests were conducted with the original (Ref. 1) Fluidfire twelve-tube heat exchanger surface (see TABLE I) and original Fluidfire distributor plate (0.05-in.-thick) and distributors having 3.4, 6, 12, and 24 percent open area (0.049-in.-thick). The Fluidfire distributor plate consisted of slots as shown in Sketch A. The exact area of the slots was difficult to ascertain but the slot width was small enough that a fluidized bed with a mean particle diameter of 0.0143 in. did not pass through it. Distributor plate pressure loss tests indicated that the open area was roughly 2 to 3 percent. The 3.4 percent open area plate consisted of 0.050-in.-dia holes located in rows and columns separated by 11/64 in. and 21/64 in., respectively. The 24 percent open area plate consisted of 0.050-in.-dia holes located in staggered rows and columns separated by 1/16 in. as shown in sketch B. The plate having 12 percent open area was made from the plate having 24 percent open area by masking half of the holes with tape (columns 1-3, 7-9, etc. were covered with tape). The plate having 6 percent open area was made from the plate having 12 percent open area by masking half of its open holes with tape (rows 1-3, 7-9, etc. were covered with tape). The distributors with 0.050-in.-dia holes were tested using a 100 mesh screen on top of the plate to prevent the bed material from falling through the 0.050-in.-dia holes. The tests were made using nominally 110 lbs of Al_2O_3 as the bed material (see TABLE II) and for the superficial velocities, V_s , of approximately 1 to 4 ft/sec.

Bed Load Tests

Tests were made to determine the effect of the amount of material on heat transfer coefficient for the Fluidfire and 3.4 percent open area distributors. In one test (Fluidfire distributor) V_s was constant at 2.25 ft/sec and bed load was varied. In other tests (3.4 percent open area distributor) V_s was varied from approximately 1 to 4 ft/sec and the bed load was fixed at 100 and 200 lb. The bed material was Al_2O_3 for these tests.

SKETCH A

SKETCH B

Bed Material Tests

Tests were conducted with fluidized beds of Al_2O_3 , SiC, and steel shot using the Fluidfire distributor. The bed materials (having a nominal size of 0.015 in.) were used as received from vendors. The distribution of particle sizes (determined by sieving bed material samples) are compared in TABLE II. The mass average particle sizes for Al_2O_3 , SiC, and steel shot were 0.0143, 0.0150, and 0.0180 in., respectively. Other properties of these materials are listed in TABLE III. The SiC is 20 percent less dense and 43 percent more thermally conductive than Al_2O_3 . Steel shot is twice as dense, 54 percent more conductive, and has one-half of the heat capacity of Al_2O_3 .

Elutriation Tests

Elutriation measurements were obtained for configurations having the original Fluidfire distributor and the 3.4 percent open area distributor and using Al_2O_3 , SiC, and steel shot bed materials. In these tests the elutriation (i.e., the flow rate of bed materials which is transported upward by the entraining gas flow) was measured at different heights above the bed. Measurement of this flow rate with distance above the bed was used to determine the TDH, which is the height above which the elutriation is a relatively small constant value. The rate was determined from samples of flow collected by an aspirated sampling probe (probe area of $2.23 \times 10^{-3} \text{ ft}^2$) mounted in the flow. The particles were collected on millipore membrane type filters (these filters collected all particles larger than 0.2 to 0.8 microns) for measured times and then weighed to determine the flow rate. The gas flow into the sampling probe was adjusted so that the velocity into the sampling probe was equal to the superficial velocity, V_s . Elutriation data were obtained for nominal superficial velocities of 2.25 and 3.75 ft/sec. The height above the bed was defined as distance above the bed in its quiescent state.

Tests of Alternate Heat Exchanger Surfaces

Tests were made to determine the heat transfer performance of a baffled two-stage heat exchanger configuration. This configuration is sketched in Fig. 3 and consists of a baffle plate having large open area and positioned close to the bed separating the water-cooled Fluidfire heat exchanger in the upper bed region from a simulated heat exchanger (containing dummy heat exchanger tubes with no coolant flow) in the lower bed region. The Fluidfire distributor supports the simulated heat exchanger and the lower bed. The baffle provides support for the upper bed with negligible pressure loss; however, it allows material to recirculate between upper and lower beds. The baffled configuration was tested with baffle plates having 22.5 and 10.5 percent open area, both baffles having 0.25-in.-dia holes. The baffle plate was positioned approximately 1.6 in. above the distributor and the total bed load consisted of 200 lbs of Al_2O_3 .

During these tests, the variation of heat transfer performance, recirculation rate of bed material through the baffle, and amount of bed material in upper and lower beds with superficial air velocity was determined. The performance and the amount of bed materials in each bed were inferred from the individual bed pressure loss.

Tests of Two-Stage Heat Exchangers With Two Distribution Plates

Tests were made to determine the performance of a two-stage heat exchanger with two distributors. This configuration is sketched in Fig. 4 and consists of the original twelve-tube Fluidfire (lower) heat exchanger and a new twelve-tube (upper) heat exchanger. Distributor plates having 3.4 percent open area were used with 100 mesh screen to support the beds for the upper and lower heat exchangers. In this configuration, there was no recirculation of bed material between upper and lower beds. The freeboard between heat exchangers was 12 in. and was selected, based on results of the elutriation tests described previously, to permit operation without the occurrence of plugging of the upper distributor by particles from the lower bed. The two-stage exchanger was tested with 110 lbs of Al_2O_3 in each bed for V_g of 0.95 to 3.75 ft/sec. The specifications for the two twelve-tube heat exchangers are given in Fig. 4.

DISCUSSION

Single-Stage Tests

Tests of Fluidfire Heat Exchanger

Data obtained in tests of the unmodified Fluidfire heat exchanger with the Fluidfire distributor plate and with a fluidized bed of Al_2O_3 are presented in Fig. 6. Data are presented for Al_2O_3 both as received from the vendor, having a 28x65 mesh size (0.025 in. to 0.009 in. range of particles sizes with average particle size of 0.014 in. (see TABLE II)), and sieved Al_2O_3 , having a 40x45 mesh size (0.016 in. to 0.014 in. range of particle sizes with average particle size of 0.014 in.). For the range of superficial velocity, V_S , tested, the data indicate that the dependence of Nusselt number (Nu) ($Nu=hgd/K$) on V_S was similar for the two size distributions, but with the Nu for the 28x65 mesh being consistently (up to 8 percent) less than that for the 40x45 mesh. Thus, the heat exchanger performance is not strongly dependent on particle size distribution. Accordingly, the 28x65 mesh Al_2O_3 as received from the vendor was used in all other tests with Al_2O_3 beds in this program except for the tests with variable bed loading.

Tests With Different Distributor Plates

Heat transfer performance data obtained in tests of the Fluidfire distributor and distributors having 3.4 and 6 percent open area are presented in Fig. 7. The data indicate that the heat exchanger can be operated successfully at larger V_S with more open area (less pressure loss) than that of the Fluidfire plate; however, at V_S less than ~ 3.5 ft/sec the Nu and, hence, heat transfer coefficient decreases with increasing open area.

The degradation in heat transfer probably results from nonuniformities in the fluidization of the bed which occurs when the distributor pressure loss becomes a fraction of the bed pressure loss. In these experiments, the bed pressure loss was approximately 2 to 2.5 in. H_2O . The variation of distributor plate pressure loss with V_S is given for the different open areas in Fig. 8 and the variation with inlet dynamic head is given in Fig. 9. The distributor plate pressure loss decreased with decreasing V_S and the data (Figs. 7 and 8) indicated that for distributor pressure losses less than approximately 65 to 85 percent of the bed pressure loss, the Nu decreases (with decreasing V_S) from its maximum value. The data also indicate for V_S of 3.5 to 4 ft/sec Nu for the three distributor plates was approximately the same, while the distributor plate pressure loss decreased by up to a factor of 5.

Observation of the bed during tests of the plate having 6 percent open area revealed that fluidization of the bed was quite uniformly active for superficial velocities greater than about 2 ft/sec and less uniformly active for V_S less than 2 ft/sec. Similar observations of the bed during tests of the plates having 24 and 12 percent open area revealed that the fluidization was extremely nonuniform throughout the range of test velocities (1 to 4 ft/sec), and performance data were not obtained for these tests. It appears that decreasing the resistance of the distributor (increasing open area) decreases the amount of suppression of any nonuniformities that may exist in the inlet flow. Apparently when the distributor loss becomes a fraction of the bed loss (generally 0.50 to 0.85 based on data obtained in these experiments) suppression of the inlet nonuniformities is inadequate resulting in a nonuniformly activated bed and reduced heat transfer (note decreased Nu at low V_S , Fig. 7). It is possible that uniformly activated beds can be obtained for smaller ratios of distributor to bed loss with improved inlet flows (apparently uniform beds have been achieved previously in shallow bed experiments conducted by Andeen and Glicksman (Ref. 3) for ratios as small as 0.3). No attempts were made in these tests to streamline or make a more uniform inlet flow.

Tests With Different Bed Loads

Data obtained in tests of the Fluidfire distributor plate with Al_2O_3 bed material and V_S of 2.3 ft/sec for different bed loads from 0 to 110 lbs are shown in Fig. 10. The data indicate that Nu increases with increasing bed load for bed loads up to 40 lbs and then remains constant for bed loads up to 110 lbs. A possible explanation for this phenomenon follows. Heat transfer from the bed to the finned tubes increases with increasing bed load (for a given V_S) until the fluidized bed covers the top of the fins and any further addition of material results in no further increase in heat transfer. In fact, if bed material is added until the ratio of distributor loss to bed loss becomes less than 0.5 to 0.85 (see previous section), the heat transfer will decrease because of the occurrence of nonuniformities in the bed activation. This is indicated by the data presented in Fig. 11. For instance, for V_S of 1.75 ft/sec and a bed load of 110 lbs, the Nu is 3.0 and the ratio of distributor to bed pressure loss is 0.71, whereas for an increased bed load of 200 lbs the ratio of distributor to bed loss is reduced to 0.39 and Nu reduced to 2.4.

For best performance at a given superficial velocity, the bed load (in a fluidized state) should be large enough to cover the heat exchanger fins but not so large as to result in a distributor pressure loss which is less than 0.5 to 0.85 of the bed pressure loss. For consistency in comparing different configurations a bed load of 110 lbs was used for most tests. This insured complete fin coverage but resulted in a slightly larger overall pressure loss.

Tests With Different Bed Materials

Heat transfer data obtained with the Fluidfire distributor and Al_2O_3 , SiC, and steel shot bed materials are presented in Fig. 12. The bed materials (having a nominal size of 0.015 in.) were used as received from vendors. The distribution of particle sizes (determined by sieving bed material samples) for SiC and steel shot are compared with that for Al_2O_3 in TABLE II. The mass average particle sizes for Al_2O_3 , steel shot, and SiC were 0.0143, 0.0180, and 0.0150, respectively. Other properties of these bed materials are presented in TABLE III. Enlarged photographs of the bed materials are shown in Fig. 13. The photographs indicate that the shape of the Al_2O_3 particles is highly irregular, the shape of the SiC particles is somewhat more regular, and the shape of many of the steel shot particles is spherical. The tests were conducted for 110 lbs of Al_2O_3 , 100 lbs of SiC, and 250 lbs of steel shot. These bed loadings were selected so that the heat exchanger fins were just covered with bed material with the bed in the unfluidized state.

The heat transfer performance data obtained using SiC and steel shot as bed materials are compared with performance obtained using Al_2O_3 in Fig. 12. The data indicate that performance obtained using SiC was equivalent to that obtained using Al_2O_3 for the range of test velocities. The performance obtained using steel shot was better (up to 63 percent higher Nu) for V_s greater than 1.5 ft/sec than that obtained using Al_2O_3 or SiC and worse (up to 26 percent lower Nu) for V_s less than 1.5 ft/sec. It should be noted that since the average diameter of the steel shot particles was 20 percent larger than that of the Al_2O_3 and SiC particles, the $\text{Nu} = h_p d / k$ (k is the conductivity of air) would be 20 percent higher for the steel shot than for Al_2O_3 even if the heat transfer coefficient, h_p , were the same. Thus a 63 percent increase in Nu only represents a 43 percent increase in heat transfer coefficient. The reason for this increase is probably related to the density and possibly the particle shape.

A limited number of tests were conducted with 200 lbs of steel shot in the bed. The lighter load resulted in Nu approximately 8 percent larger than Nu obtained by a 250 lb bed load for V_s near 2 ft/sec. For V_s greater than 3 ft/sec, Nu was up to 8 percent less than Nu obtained for a 250 lb bed load. This behavior is unexplained at present and indicates more testing with different bed loads is required.

The data showing the variation of bed pressure loss with V_s for the three bed materials are given in Fig. 14. The pressure was determined using a probe located at the center of the flow duct between the distributor plate and the bed. If the bed material and air flow were uniformly distributed for the

range of test V_S , the pressure would be constant for velocities greater than the minimum velocity for fluidization, approximately 0.4 ft/sec. For the tests with 110 lbs of Al_2O_3 and 100 lbs of SiC the data may indicate that the bed material depth increased in the central region (where probe was located) for low V_S (1.0 ft/sec) and high V_S (4 ft/sec). For the tests with 200 lbs of Al_2O_3 the bed load appeared to decrease in the central region with decreasing V_S . For tests with steel shot the bed pressure loss was constant for V_S (as low as 2.5 ft/sec) and then increased to a maximum and decreased with further decreases in V_S . Observation of the bed during tests with steel shot indicated that the steel shot bed was nonuniformly activated for V_S less than approximately 2 ft/sec with the central region quiescent. For these conditions the flow is probably nonuniform and the pressure measured by the probe an inaccurate representation of the bed pressure (the decrease in Nu (Fig. 12) at these velocities (less than 2 ft/sec) also indicates incomplete fluidization of steel shot in this velocity range).

In summary, approximately 43 to 50 percent higher heat transfer coefficient, h_B , was obtained (for V_S greater than 1.5 ft/sec) when using the bed of steel shot than when Al_2O_3 or SiC beds were used. However, the bed pressure loss for steel shot was about 125 percent greater than that for beds of the less dense Al_2O_3 and SiC, thus making steel shot unattractive for use in low pressure drop fluidized beds.

Elutriation Tests

Test data obtained during elutriation tests are presented for Al_2O_3 , SiC, and steel shot in Figs. 15, 16, and 17, respectively. In general, the entrainment rate (weight of particles collected by probe per unit time) increased with increasing height above the bed to a fixed low minimum value for heights above the bed greater than the TDH. The value of TDH was subjectively chosen to be the height (in the knee of the curve of elutriation rate versus height) below which elutriation increases very quickly with decreasing height. The magnitude of the minimum value which was attained generally increased with increasing superficial velocity, V_S . Examination of the filters indicated that the material collected consisted of particles having the nominal 0.015-in.-dia (i.e., the normal bed material) and a very fine dust. For heights greater than the TDH, the minimum rates which were measured resulted, almost exclusively, from these fine dusts. For an actual fluidized bed waste-heat heat exchanger, the minimum value of elutriation would depend on the rate of production of the fine dust through self-abrasion of the bed material. If there were no self-abrasion, then ideally the minimum value of elutriation could be zero. The entrainment rates measured using Al_2O_3 and SiC (for the same V_S) were similar for similar superficial velocities and were greater than those measured using steel shot. This is not unexpected, since the density of the steel is much greater than that of the Al_2O_3 and SiC which are quite similar in density. The data also

generally indicate that the entrainment is less for the 3.4 percent open area distributor plate than for the original Fluidfire distributor plate. This difference could result from the occurrence of smaller local velocities (near the distributor plate) for the distributor plate having more open area.

In some tests with SiC made at V_S equal to 2.25 ft/sec, the probe inlet velocity, V_C , was 0.25 ft/sec greater than V_S . Data from these tests (2.25 ft/sec) are compared in Fig. 16 with the data with both V_S and V_C equal to 2.25 ft/sec (compare singly- and doubly-spaced flagged circular symbols in Fig. 16). A comparison of the data indicates that the slight difference in V_C does not appreciably affect the measured entrainment rate and has no effect on the TDH.

The data in Figs. 15 and 16 show that the TDH for Al_2O_3 and SiC increased from approximately 6 in. for V_S equal to 2.25 ft/sec to approximately 12 in. for V_S equal to 3.75 ft/sec. The TDH for steel shot (Fig. 17) increased from 1 in. for V_S equal to 2.25 ft/sec to approximately 6 in. for V_S equal to 3.75 ft/sec. The variation in TDH with V_S for these tests is given in Fig. 18.

Alternate Heat Exchanger Surface Tests

Data obtained for alternate heat exchanger surfaces in two different arrangements (see Fig. 5) are presented in Figs. 19 and 20. The specifications for the alternate heat exchanger surfaces are compared with those of the Fluidfire finned tubes in TABLE I.

Test results obtained for configuration 1 are shown in Fig. 19. The variations in $Nu = h_B d/k$, with superficial velocity, V_S , are shown in Fig. 19a. These data indicate that the Nusselt number varies directly with fin spacing and for small fin spacing 7.33 and 10 fins/in.) Nu increases with increasing V_S , while for large spacings (0 and 4.35 fins/in.) Nu decreases with increasing V_S . It appears that the effect of the fins is to reduce the heat transfer from the bed to the heat exchanger and that increasing V_S makes this effect less pronounced (i.e., Nu becomes more similar for the different heat exchanger surfaces). The heat transfer rate (bed heat transfer coefficient, h_B , multiplied by the heat exchanger surface area, A_T) for the four alternate heat exchanger surfaces are presented in Fig. 19b. The data indicate that the heat transfer rate of the heat exchanger surfaces having 7.33 and 10 fins/in. is quite similar and generally larger than those having 4.35 and 0 fins/in. The data are also compared with that obtained for the Fluidfire heat exchanger (having twelve-tubes with 7.33 fins/in.) in Fig. 19a. The comparison indicates that the alternate heat exchanger surfaces having 7.33 fins/in. performed very differently from the Fluidfire heat exchanger surface with performance being most similar at large V_S . The discrepancy in performance may result from differences in the fluidization of the alternate and Fluidfire heat exchanger

beds. These differences probably resulted from (1) the proximity of the 7.33 fins/in. alternate heat exchanger surface to the bed sidewall (the performance of the Fluidfire exchanger surface was based on tests of twelve tubes uniformly spaced in the bed, whereas the performance of the 7.33 fins/in. alternate exchanger surface was based on tests of two tubes located near one sidewall (see Fig. 5a), or (2) differences in geometry of the two heat exchanger surfaces (the alternate consisted of eight tubes with four sets of different tubes, whereas the Fluidfire consisted of twelve identical tubes). This caused concern because, if the presence of different types of tubes in the bed influenced the results, the current test results could only be viewed in a qualitative sense.

Accordingly, to determine the validity of the tests in Configuration 1, a second configuration was tested in which the alternate heat exchanger tubes were interchanged so that the 7.33 and 10 fins/in. tubes were not located near the sidewalls (see Fig. 5b). The performance data obtained in these tests (Configuration 2) are presented and compared with Configuration 1 performance in Fig. 20. In general, the data indicate that both the 7.33 and 10 fins/in. tubes performed better when located toward the middle of the bed than when located near the sidewalls with the most improvement occurring for the 10 fins/in. tube. Similarly, the 0 fins/in. and 4.35 fins/in. tubes performed better when located toward the center of the bed (Configuration 1) for small V_g . However, the discrepancy in data between that obtained with the Fluidfire heat exchanger surface and the similar geometry (i.e., 7.33 fins/in.) alternate heat exchanger surfaces cannot be explained by location in the bed.

The results of this testing unfortunately showed that the method of test was not appropriate to obtain quantitative data. There appears to be a strong interaction between adjacent heat exchanger tubes in a shallow fluidized bed heat exchanger. There does appear to be a qualitative decrease in heat transfer coefficient with decreasing fin spacing, but this cannot be quantified here. Future tests must be run with only one surface geometry at a time.

Two-Stage Tests

Tests of Two-Stage Heat Exchanger With a Baffle

Heat transfer performance data obtained for the baffled two-stage heat exchanger configuration shown in Fig. 3 is compared with that obtained for the unbaffled single-stage configuration in Fig. 21. The data were obtained with the Fluidfire heat exchanger surface installed in the upper bed and with an uncooled heat exchanger surface in the lower bed to provide flow resistance. For the 22.5 percent open area baffle plate, the data showed that the variation of Nu with superficial velocity, V_g , was approximately the same as that

obtained without the baffle. It was assumed that the amount of material in each bed (bed loading) is proportional to the measured pressure loss for each bed and that the heat transfer depends to some degree on the bed loading. The variation of pressure loss with V_S for the upper (solid circular symbols) and lower (open circular symbols) beds are compared in Fig. 22a and indicate for V_S of 1.0 ft/sec the pressure loss and, therefore, probably the bed loading and heat transfer capability are equal for the upper and lower beds. At larger values of V_S such as 3 to 4 ft/sec (larger values of V_S are desirable because they result in smaller and, therefore, less costly beds) the pressure loss for the lower bed is substantially reduced, indicating that most of the bed material is in the upper bed, with the probable result that heat transfer capability was reduced for the lower bed.

It was believed that decreasing the open area of the baffle would increase the value of V_S where equal bed loadings occur (equal upper and lower bed pressure losses). Accordingly, a baffle plate having a 10.5 percent open area was tested. The data (open and solid square symbols) presented in Fig. 22a indicate that the V_S for equal bed loads is approximately the same as that obtained for the baffles with 22.5 percent open area. Also, the upper bed heat transfer data (Fig. 21) for the baffle with 10.5 percent open area indicates that decreasing the baffle open area has a significant effect on the heat transfer characteristics of the upper bed. The Nu is equal to that obtained without a baffle for V_S of 3.7 ft/sec, is less than that obtained without a baffle for V_S less than 3.7 ft/sec, and does not vary appreciably with V_S between 1.25 and 3.7 ft/sec.

The amount of bed material recirculated between the upper and lower beds through the baffle (small recirculation flows are required if baffles are to be effective) was deduced from measured temperatures and heat balances for the two baffled configurations and is shown in Fig. 22b. The amount of recirculation is characterized by the ratio, f (see APPENDIX II for derivation of f), which is equal to the product of the bed material recirculation flow rate and its specific heat ($C_p = 0.22$ Btu/lb F for Al_2O_3) divided by the product of the air mass flow rate and its specific heat (C_p for air = 0.24 Btu/lb F). The data indicate that the variation of f with V_S is quite irregular and unexplained at present. In general (except for V_S between 1.4 and 2.0 ft/sec), f was less for the baffle with 10.5 percent open area than for the baffle with 22.4 percent open area. At $V_S \sim 1$ ft/sec, the recirculation was small and the bed loadings about equal. Thus, it appears that for these baffle configurations the optimum operating point is about 1 ft/sec. The baffle configuration did not provide equal bed loadings and small values of recirculation rate at higher velocities.

In summary, results from tests of the baffled two-stage heat exchanger configuration (Fig. 3) indicate that for operation with upper and lower bed loads approximately equal, V_S must be about 1.0 ft/sec resulting in f in the range of 4 to 10 and Nu of 2.0 to 4.0. Further experiments with different configurations (i.e., using baffles with different hole sizes and different spacings above the distributor) should be done to determine if the V_S for equal bed loads can be increased and the recirculation rate decreased.

Calculations were made to determine the effects of using a two-stage fluidized bed with and without baffling on the NTU (number of transfer units) requirements for a heat exchanger having the same requirements as given for the MTI heat exchanger discussed in Ref. 1. In the calculations of Ref. 1 the fluidized beds were used to provide heat transfer to the vaporizers only and not a preheater or a superheater. The results of the calculations for a range of f ratios are summarized in the following TABLE.

Configuration	f Ratio	NTU
Single-Stage	-	3.40
Baffled Two-Stage	10	3.20
" "	4	2.97
" "	1	2.59
Two-Stage (no baffle)	0	2.20

The results in the TABLE indicate that the use of a nonbaffled two-stage heat exchanger instead of a single-stage configuration decreases the NTU (and, therefore, the heat transfer surface area) requirements by 35 percent (from 3.4 to 2.2). However, for a baffled two-stage configuration the decrease in NTU requirements is less and depends on the f ratio. For an f ratio of 4 the reduction in NTU requirements is 13 percent. The analysis indicates that the effectiveness of a baffled two-stage configuration is reduced by recirculation of the bed material through the baffle; however, for f ratios measured in these experiments ($f=4-10$), baffled two-stage NTU requirements are reduced (13 to 6 percent, respectively) from NTU requirements for a single-stage configuration.

The results of the studies of the baffled two-stage heat exchanger configuration demonstrated that baffles could be used successfully to stage fluidized bed heat exchangers. However, the velocity at which the unit was most effective was low (approximately 1 ft/sec) and the effectiveness was much less than that achievable with a two-stage unit having two distributor plates. It appears that additional studies would be required to obtain increased effectiveness at higher velocity for baffled two-stage heat exchangers. Therefore, in additional tests on staged heat exchangers conducted in this program separate distributor plates were used.

Tests of Two-Stage Heat Exchangers With 3.4 Percent Open Area Distributors

Results obtained in previous tests were used in the design of a two-stage fluidized bed heat exchanger having separate distributor plates for each stage. The heat exchanger was designed for stage heat transfer performance comparable to that provided by the original Fluidfire heat exchanger at superficial velocities of 3 to 4 ft/sec and at a nominal overall heat exchanger pressure loss of 12 in. of H_2O .

Distributor plates having 3.4 percent open area which provided approximately 45 percent less pressure loss than the Fluidfire distributor plate were selected to support the beds. Although distributors having 6 percent open area would have provided even less pressure loss, the heat transfer performance obtained with the distributor having 3.4 percent open area was better for a wider range of V_s than that obtained with the distributors having 6 percent open area.

Of the three bed materials tested, steel shot provided the best heat transfer performance; however, bed pressure losses were approximately twice as large as those obtained with either SiC or Al_2O_3 . Since heat transfer performance obtained with SiC was no better than that obtained with Al_2O_3 , and pressure losses were similar, Al_2O_3 was selected as the bed material for the two-stage unit to keep the bed pressure losses within acceptable limits.

A freeboard of 12 in. between heat exchangers was allowed to prevent plugging of the upper distributor plate by material from the lower bed. This distance was based on TDH estimated from elutriation tests discussed previously.

Heat transfer coefficient data obtained for the two-stage heat exchanger configuration schematically shown in Fig. 4 are presented in Fig. 23. Data are presented for both the Fluidfire (lower) and new (upper) heat exchanger surfaces in both single- and two-stage configurations. For the new (upper) heat exchanger surfaces data were the same when operated in either single- or two-stage configurations (compare square symbols in Fig. 23). However, the Nu obtained for the new unit was between 34 and 52 percent less than that obtained for the Fluidfire unit (compare square and circular symbols in Fig. 23). The differences in heat transfer of the two heat exchanger surfaces probably results from the small differences in geometry (see TABLE in Fig. 4). The largest dissimilarity between the two units was the spacing between the distributor plates and the bottom edge of the fins, being 0 in. for the Fluidfire unit and 0.5 in. for the new unit and occurred as a result of the method of construction of the two heat exchangers. Apparently, these geometric differences caused the beds to be fluidized differently resulting in dissimilar heat transfer coefficients.

The data for the Fluidfire heat exchanger surface indicate that the heat transfer of this unit was different for the two-stage than for the single-stage configuration (in Fig. 23 compare circular symbols with solid curve from Fig. 6). For V_S greater than 1.75 ft/sec, Nu was up to 13 percent greater; however, for V_S less than 1.75 ft/sec, Nu was up to 30 percent smaller. This change in heat transfer may result from effects on the flow caused by the additional back pressure imposed by the presence of the upper heat exchanger. However, it is curious that the performance of the Fluidfire unit should be affected by the presence of the upper unit and yet the performance of the upper unit is unaffected by the presence of the Fluidfire unit.

Bed, distributor plate, and heat exchanger pressure loss data for the two-stage heat exchanger are presented in Fig. 24. The data indicate that the bed pressure loss decreased slightly with increasing V_S from approximately 2.3 in. of H_2O at V_S of 0.95 ft/sec to 1.9 in. of H_2O at V_S of 3.73 ft/sec. It was expected that the bed pressure loss would not vary with V_S and would equal that pressure required to support the bed load. A pressure loss of 2.1 in. of H_2O would support 61 lbs of bed material over the 5.55 sq ft of active bed area indicating that 49 lbs of the 110 lb bed load was being supported by inactive areas such as heat exchanger tubes, manifolds, and distributor plate support hardware.

The data also indicate that, as expected, the losses for both distributor plates increases with increasing V_S . In addition, the pressure loss was less for the upper plate than for the lower plate. Since the distributor plates are identical, the smaller pressure loss obtained for the upper plate results from the decrease in inlet dynamic head to the upper plate due to the cooling of the flow by the lower heat exchanger. Predictably, the exchanger loss (i.e., the sum of the lower and upper distributor plate plus the bed losses) increases with increasing superficial velocity, V_S . As shown in Fig. 24, the heat exchanger loss increased from 5.3 in. of H_2O at V_S equal to 0.95 ft/sec to 14.7 in. of H_2O at the maximum operating V_S of 3.73 ft/sec.

The tests of the two-stage heat exchanger were conducted over a six hour time period. During this period, the pressure loss performance did not change, indicating that plugging of the upper distributor plate by lower bed material did not occur and, therefore, the 12 in. of freeboard between heat exchangers was adequate.

In summary, the results of the two-stage tests indicated that for the test duration (about six hours) plugging of the upper distributor plate by bed material from the lower bed did not occur and the performance of the heat exchangers in the two-stage configuration was equivalent to or slightly better than their individual performance. However, some as yet unexplained differences in the heat transfer coefficients of the beds (as noted above) do exist.

CONCEPTUAL DESIGN OF STAGED HEAT EXCHANGERS FOR DIESEL EXHAUST WASTE HEAT RECOVERY

Investigations of Allowable Diesel Exhaust Back Pressure

Information was obtained from manufacturers and the literature regarding the operation of large diesel engines at increased back pressure. Reference 4 describes tests of a fluidized bed heat exchanger used to recover waste heat from the exhaust of a 12000 bhp slow-speed marine diesel. Operation with the bed resulted in an increase in engine back pressure from 4 in. of H_2O to approximately 12 to 14 in. of H_2O . The increased back pressure resulted in a 59 F temperature rise of the exhaust gas. A back pressure increase from 12 to 14 in. H_2O was due to plugging (by soot) of the distributor plate and when the back pressure reached 14 in. of H_2O wire brushes were used to clean soot from the distributor plate reducing the back pressure to 12 in. of H_2O . Three manufacturers (Refs. 5, 6, and 7) indicated that operation of diesel engines with back pressures up to 10 to 12 in. of H_2O was acceptable. They stated that operation at higher back pressures would result in increased fuel consumption, loss of power, and higher exhaust gas temperatures; the last item resulting in shorter lifetimes for valves and turbochargers. Reference 7 indicated that the power loss would be 2 to 3 percent for each additional 4 in. of H_2O back pressure. One manufacturer (Ref. 8) indicated that a back pressure up to 5 in. of H_2O was acceptable and that the power loss would be 0.5 percent for each additional 4 in. H_2O of back pressure. The manufacturers generally indicated that fouling may or may not be a problem depending on the type of fuel used. On the basis of the above information, an allowable back pressure of 12 in. H_2O was chosen for the Design Study.

Conceptual Design Study of Multi-Staged Shallow Fluidized Bed Heat Exchangers

Conceptual two- and three-stage fluidized bed heat exchangers were analyzed using the data obtained in the previously completed performance and surface development tests. The previously obtained data were used in the selection of the heat exchanger geometry, the distributor plate configuration, the bed material, the freeboard between distributor plates, and the superficial velocity, V_s . These designs are compared with a single-stage fluidized bed heat exchanger previously analyzed during this program and described in Ref. 1. The heat exchangers were designed to operate with superficial velocities consistent with a heat exchanger pressure loss of 12 in. of H_2O or less and with sufficient surface area to transfer the required heat (the heat load of 9.5×10^6 Btu/hr is the same as that used in Ref. 1).

A 0.75-in.-dia tube (0.684-in.-ID) having seven 1.5-in.-dia by 0.025-in.-thick fins per inch was the geometry selected for the two- and three-stage heat exchanger surfaces, since most experience in this program has been obtained using this geometry.

Distributor plates having 3.4 percent and 6.0 percent open area were selected for two- and three-stage heat exchangers, respectively. (The selection of 3.4 percent open area for the two-stage distributor plates is somewhat arbitrary since a 6 percent open area would result in less pressure loss. However, heat transfer coefficient is less sensitive to superficial velocity for a 3.4 percent than a 6 percent open area (see Fig. 7).) Test data shown indicated that for V_s of approximately 3.5 to 4.5 ft/sec the heat transfer performance obtained from the 3.4 and 6.0 percent open area distributor plates was approximately the same as that obtained from the original Fluidfire distributor plate (Nu was approximately 2.7). For the design of the two and three-stage heat exchangers, it was assumed that Nu equal to that obtained in the single-stage tests, namely 2.7, would be achieved for all stages. Nu of 2.7 is also approximately equal to the average of Nu obtained for the upper and lower stages of the two-stage heat exchanger tested in this program. Nu equal to 2.7 corresponded to a bed heat transfer coefficient of 37 Btu/hr-ft²-F in these tests and up to approximately 47 Btu/hr-ft²-F for the water and gas temperatures anticipated in the diesel exhaust waste heat recovery heat exchanger. The two-stage fluidized bed heat exchanger was designed using a conservative h_b of 37 Btu/hr-ft²-F, a realistic h_b of 47 Btu/hr-ft²-F and h_b of 61 Btu/hr-ft²-F attainable (Ref. 1) with improvements in the state-of-the-art of fluidized beds. The three-stage unit was designed only using the more conservative h_b of 37 Btu/hr-ft²-F.

Al_2O_3 was the bed material selected for the two- and three-stage heat exchangers. Test data shown in Fig. 12 indicated that no improvements in heat transfer was obtained using less dense silicon carbide (SiC) and the increased pressure loss obtained using steel shot (Fig. 13) more than offset the increase in heat transfer.

The two- and three-stage fluidized bed heat exchangers are shown schematically in Figs. 25 and 26, respectively. The specifications of these heat exchangers are compared in TABLE IV with those for the conventional MTI heat exchanger and the shallow fluidized bed single-stage heat exchanger described previously in Ref. 1. Superheating and vaporization of steam are accomplished in the first stage of both the two- and three-stage units. The superheater and vaporizer beds are separated by a vertical baffle of sufficient height to keep the material from the two adjacent beds from mixing. The required baffle height is equal to the TDH and was determined from extrapolation of data presented in Figs. 14 through 18. A vertical baffle was provided to allow the mixture (a freeboard between stages greater than the baffle height

is required for air mixing) of superheater and vaporizer air to exit at a temperature less than the required steam exit temperature of 430 F and at the same time to maintain the superheater bed at a temperature greater than the steam exit temperature. Such exit temperatures are required to obtain minimum heat exchanger areas and still maintain heat exchangers having equal surface areas. Steam is vaporized in the second stage of the three-stage unit. Water is preheated and steam vaporized in the second and third stages of the two- and three-stage units, respectively. The bed temperature and amount of steam vaporized in each stage were selected to equalize the per stage heat exchanger area and to minimize the total heat exchanger area required. Also, the distributor plate area was chosen to provide an overall gas pressure loss less than 12 in. of H₂O (it was assumed that a pressure of 1 in. H₂O per stage was required to fluidize the bed). The distributor plate length is equal to the stage heat exchanger tube length and the width is equal to the number of tubes multiplied by the tube pitch. Pitch for the two-stage heat exchanger is 1.75 in. consistent with the heat exchangers tested in this program. Pitch for the three-stage unit is 2.2 in. and resulted in stage superficial velocities greater than 4.0 ft/sec. The extrapolation of previously obtained data was required to obtain bed heat transfer coefficients for V_s greater than approximately 4 ft/sec. Because an extrapolation of the data is required and because the data were obtained for a pitch of 1.75 rather than 2.2 in., the three-stage design results should be viewed with caution.

The specifications for the three fluidized bed and conventional MTI heat exchanger configurations are given in TABLE IV. The heat exchangers were designed to provide 9000 lb/hr of 430 F steam from water at 215 F using 205,000 lb/hr of air entering at 520 F and exhausting at 327 F consistent with the conditions of Ref. 1. The superheater and preheater of the single-stage heat exchanger (Ref. 1) consisted of 1.875-in.-dia tubes having three 3.375-in.-dia fins per inch while the vaporizer consisted of 0.75-in.-dia tubes having seven 1.5-in.-dia fins per inch. Since only the vaporizer was fluidized, an h of 47 Btu/hr-ft²-F was used for the vaporizer and h of 12.6 Btu/hr-ft²-F for the superheater and preheater. The preheater, superheater, and vaporizers of the two- and three-stage units all consisted of 0.75-in.-dia tubes having seven 1.5-in.-dia fins per inch and all stages were fluidized.

The required heat exchanger area for the single-stage unit described in Ref. 1 is 7100 ft². The required heat exchanger area for the two- and three-stage units is 6616 and 5368 ft², respectively, and was calculated using a conservative value of h_g of 37 Btu/hr-ft²-F. Thus, the two-stage unit resulted in a 7 percent reduction in the required heat exchanger area. The gas pressure loss is 6.6, 8.1, and 12.0 in. of H₂O for the single-stage (original Fluidfire distributor plate), two-stage (3.4 percent open area distributor plate), and three-stage (6 percent open area distributor plate) unit, respectively. It was assumed that no advantage was realized in reduced pressure loss so long as the overall pressure loss was less than 12 in. H₂O.

The estimated costs for the three heat exchangers is given in TABLE V. The costs were estimated using the costs of the single-stage unit (Ref. 1) as a base. Since the size of the two- and three-stage heat exchangers was not greatly different than that of the single-stage unit, the cost of the steam separator, water pumps, and manifolding for the single-, two-, and three-stage units was estimated to be the same as that for the single-stage unit. The building costs for the two- and three-stage units were computed using the ratio of their cross-sectional areas to that of the single-stage unit and the building cost of the single-stage unit. The heat exchanger (superheater, preheater, and vaporizer) costs for the two- and three-stage units was computed using the ratio of the heat exchanger area to the single-stage vaporizer area and the cost of the single-stage vaporizer (\$7.5K). The distributor plate costs were estimated using manufacturers prices for perforated plate and wire mesh screen and included \$2K for welding costs (the same welding costs were included in the estimate of Ref. 1). The amount of bed material was determined from the required distributor plate area and by assuming that the bed depth in each stage would be the same as that for the single-stage unit. Bed material cost was estimated using the unit cost given in Ref. 1.

The cost of the two-stage unit was estimated using a conservative value of h_B of 37 Btu/hr-ft²-F, a more realistic value of 47 Btu/hr-ft²-F, and a value of 61 Btu/hr-ft²-F which may be attained through advances in the state-of-the-art and of fluidized bed heat exchangers. Costs are compared with those for the conventional MTI unit and the single-stage unit described in Ref. 1 in TABLE V.

The cost of the conventional MTI heat exchanger was estimated by MTI to be \$50K (Ref. 1). The cost of the single-stage fluidized bed was estimated to be \$52.5K using MTI costs for steam drums and pumps, and using costs estimated in Ref. 1 for the superheater, preheater, vaporizer, distributor plate, bed material, manifolding, and building. The cost of the conventional MTI unit was approximately 5 percent less than that of the single-stage fluidized bed heat exchanger. The estimated cost of the two-stage unit with $h_B = 37$ Btu/hr-ft²-F was approximately 20 percent less than the cost of the single-stage unit of Ref. 1. With $h_B = 47$ Btu/hr-ft²-F the cost of the two-stage unit was 32 percent less than that of the single-stage unit and if an h_B of 61 Btu/hr-ft²-F were attainable (as assumed in Ref. 1), the cost of the two-stage unit would be approximately 39 percent less than the cost of the single-stage unit. The cost of a three-stage unit ($h_B = 37$ Btu/hr-ft²-F) is approximately 13 percent less than that of the two-stage unit. Thus, appreciable savings can be achieved through staging of fluidized bed heat exchangers.

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LIST OF SYMBOLS

A	Heat exchanger surface area, ft^2
C_{pA}	Specific heat of air, 0.24 Btu/lb-F
C_{pB}	Specific heat of Al_2O_3 , 0.22 Btu/lb-F
C_{pw}	Specific heat of H_2O , 1 Btu/lb-F
d	Diameter of particle, ft
f	Recirculation ratio = $M_B C_{pB} / M_A C_{pA}$, dimensionless
h_B	Bed to heat exchanger coefficient, $\text{Btu/hr-ft}^2\text{-F}$
H	Height above bed, in.
k	Thermal conductivity of air, $\text{Btu/hr-ft}^2\text{-F/ft}$
M_A	Air flow rate, lb/sec
M_B	Bed recirculation rate, lb/sec
M_w	Heat exchanger water flow rate, lb/hr
M_{wc}	Heat exchanger make up water flow rate, lb/hr
Nu	$h_B d / k$
P	Pressure, lb-f/ft ²
P_w	Pump work, Btu/hr
Q_A	Heat transferred from air, Btu/hr
Q_M	Heat transferred to water, Btu/hr
Q_{Mc}	Heat transferred to make up water, Btu/hr
Q_L	Heat loss, Btu/hr
Q_1	Heat transferred from bed 1, Btu/hr
Q_2	Heat transferred from bed 2, Btu/hr

LIST OF SYMBOLS (Concluded)

T_1	Temperature of inlet air, F
T_{B1}	Temperature of bed 1, F
T_{B2}	Temperature of bed 2, F
T_5	Water inlet temperature, F
T_6	Water exit temperature, F
T_7	Make up water temperature, F
TDH	Transport Disengagement Height

APPENDIX I: CALCULATION OF HEAT TRANSFER COEFFICIENT

The heat transferred from the air to the heat exchanger of the fluidized bed is given by Eq. (1) in which T_1 and T_B

$$Q_A = M_A C_{PA} (T_1 - T_B) \quad (1)$$

are the air inlet and bed temperatures, respectively, and M_A is the air flow rate. The heat transferred to the water was calculated from the temperature rise of the water through the heat exchanger, Eq. (2), and from the temperature rise of the make up water, Eq. (3).

$$Q_M = M_w C_{Pw} (T_6 - T_5) \quad (2)$$

$$Q_{MC} = M_{we} C_{Pw} (T_7 - T_6) - P_w + Q_L \quad (3)$$

In Eq. (2), M_w is the water flow rate through the heat exchanger and T_6 and T_5 are the water exit and inlet temperatures, respectively. In Eq. (3), M_{we} is the make up water flow rate, T_7 is the make up water inlet temperature, T_6 is the make up water exit temperature, P_w is the pumpwork, and Q_L are heat losses from the make up water circuit.

The overall heat transfer coefficient for the fluidized bed heat exchanger is given by Eq. (4) and is that obtained using the mixed model fluidized bed discussed in Ref. 2.

$$U = Q / A (T_6 - T_5) \ln (T_B - T_5) / (T_B - T_6) \quad (4)$$

The temperatures in Eq. (4) have been identified above and A is the heat exchanger surface area. The heat Q is an average of the heats given by Eqs. (1), (2), and (3). In this analysis uncertainties were calculated for each parameter using the method described in Ref. 1, and averages were then calculated using the uncertainties as weighting factors to minimize the uncertainty of the average.

The bed heat transfer coefficient, h_B , was calculated using the overall coefficient given by Eq. (4) and using Eq. (5)

$$1/U = 1/\eta h_B + R_F + R_C \quad (5)$$

In Eq. (5) the terms R_F and R_C are the resistance of the internal water film and the resistance of the tube wall to heat transfer, respectively, and η is the fin effectiveness of the fin tube heat exchanger. R_C is given by Eq. (6).

$$R_C = (A/A_i)(r_i/k) \ln r_o/r_i \quad (6)$$

where A is the heat exchanger surface area, A_i is the area of the internal tube surface, k is the thermal conductivity of the tube wall, and r_i and r_o are the inner and outer tube radii, respectively. R_F is given by Eq. (7)

$$R_F = A/A_i h_w \quad (7)$$

where the internal film coefficient, h_w , is given by Eq. (8)

$$h_w d/K = 0.023 Re^{0.8} Pr^{0.4} \quad (8)$$

In Eq. (8), d is the tube diameter, Re is the Reynolds number, and Pr is the Prandtl number. The bed heat transfer coefficient is calculated, then, using Eq. (5) with the U determined from measurements and calculated values of R_F and R_C .

APPENDIX II: RECIRCULATION OF BED MATERIAL IN BAFFLED TWO-STAGE HEAT EXCHANGER

The amount of recirculation of bed material between upper and lower beds in the baffled two-stage heat exchanger configuration shown in Fig. 3 and in the sketch below was determined from energy conservation equations for each bed (see Sketch C).

$$T_1 = T_{\text{AIR IN}} \downarrow$$

$$T_2 = T_{\text{AIR OUT}} \downarrow = T_{\text{BED 2}}$$

$$T_{B1} = T_{\text{BED 1}}$$

$$M_A = \text{AIR FLOW RATE}$$

$$M_B = \text{BED MATERIAL RECIRCULATION}$$

Sketch C

The heat balance equations for beds 1 and 2 are given by Eqs. (6) and (7), respectively.

$$M_A C_{PA} (T_1 - T_{B1}) = Q_1 + M_B C_{PB} (T_{B1} - T_{B2}) \quad (6)$$

$$M_A C_{PA} (T_{B1} - T_{B2}) = Q_2 - M_B C_{PB} (T_{B1} - T_{B2}) \quad (7)$$

The specific heats of air and bed material are C_{pA} and C_{pB} , respectively, and for these tests Q_1 was zero since simulated heat exchanger tubes were used in bed 1. Equations (6) and (7) are rewritten after dividing both equations by $M_A C_{pA}$ and letting $M_B C_{pB} / M_A C_{pA}$ equal f , the recirculation ratio.

$$f = (T_1 - T_{B1}) / (T_{B1} - T_{B2}) \quad (8)$$

$$f = Q_2 / M_A C_{pA} (T_{B1} - T_{B2})^{-1} \quad (9)$$

In these experiments an average value of f was obtained from Eqs. (8) and (9) using measured temperatures and flow rates.

TABLE I: HEAT EXCHANGER SURFACE SPECIFICATIONS

Heat Exchanger	Fins (in.)	Tube OD (in.)	Tube ID (in.)	Fin OD (in.)	Fin Thick. (in.)	Out-side Area (Ft ²)	Ratio of Outside to Inside Area
Fluidfire	7.33	0.715	0.50	1.375	0.025	8.4	11.7
Alternate	10	0.75	0.69	1.5	0.025	13.2	10.39
" "	7.33	0.75	0.69	1.5	0.025	10.1	7.94
" "	4.35	0.75	0.69	1.5	0.025	6.7	5.24
" "	0	0.875	0.765	-	-	1.64	1.15

TABLE II: PARTICLE SIZES

Size Range (Mesh)	Avg. Size (in.)	Al ₂ O ₃ (Percent)	Steel Shot (Percent)	SiC (Percent)
16-20	0.0362	-	0.1	-
20-25	0.0304	-	0.8	-
25-30	0.0256	-	3.0	-
30-35	0.0216	7	25.5	0
35-40	0.0181	19	36.1	16.8
40-45	0.0152	24	21.3	71.0
45-60	0.0118	38	12.6	11.6
60	0.0098	12	0.6	0.5
Average Diameter, in.		0.0143	0.0184	0.0150

TABLE III: PARTICLE PROPERTIES

Material	Density (lb/ft ³)	Specific Heat (BTU/lb-F°)	Thermal Conductivity (BTU/hr-ft ² -°F/in.)
Al ₂ O ₃	245	0.22	202
Si	200	0.22	290
Steel Shot	491	0.11	312

TABLE IV: HEAT EXCHANGER SPECIFICATIONS

	Conventional MTI (Ref. 1)	Single-Stage (Ref. 1)	Two-Stage	Three-Stage
Heat Transfer Rate, Btu/hr-ft ² -F	9,600,000	9,600,000	9,500,000	9,500,000
Gas Flow Rate, lb/hr	205,000	205,000	205,000	205,000
Water/Steam Flow Rate, lb/hr	9,000	9,000	9,000	9,000
Gas Inlet Temperature, F	520	520	520	520
Gas Exit Temperature, F	333	327	327.3	327.3
Water Inlet Temperature, F	215	215	215	215
Steam Exit Temperature, F	430	430	430	430
Gas Heat Transfer Coefficient, Btu/hr-ft ² -F	12.6	47(12.6)*	37	37
Total Heat Exchanger Area, ft ²	10,200	7100	6616	5368
Tube Diameter, in.	1.9	0.75(1.9)**	0.75	0.75
Fin Diameter, in.	3.4	1.5(3.4)***	1.5	1.5
Fin Spacing, Per Inch	3	7(3)****	7	7
Tube Pitch, in.	-	1.5	1.75	2.2
Number of Tubes	400	96	100	100
Tube Length, ft	2766	3420	4640	3765
Superficial Velocity, ft/sec (1st/2nd/3rd)	24	1.8	4.2/3.6	6.2/5.3/5.1
Transport Disengagement Height, in.	-	5	14	19;16
Distributor Plate (Fluidfire) Open Area	-	2-3%	3.4%	6%
Gas Pressure Loss, in. H ₂ O	4.2	6.6	8.1	12.0
Water Pressure Loss, psi	10.5	7.9	14.2	12.8

*h=47 BTU/hr-ft²-F for vaporizer and 12.6 for superheater and preheater.

**Tube Diameter=0.75 for vaporizer and 1.875 for superheater and preheater.

***Fin Diameter=1.5 for vaporizer and 3.375 for superheater and preheater.

****Fin Spacing=7 fins/in. for vaporizer and 3 fins/in. for superheater and preheater.

TABLE V: HEAT EXCHANGER COSTS

Heat Exchanger	h-BTU/ hr-ft ² -F	COSTS - DOLLARS					Building and Installation	TOTAL
		Steam Sep., Water Pumps & Manifolds	Superheater & Preheater	Vaporizer	Distri- butor Plate	Al ₂ O ₃		
MTI								
Conventional	12.6	13,400	-	31,600*	-	-	5,000	50,000
Single-Stage (Ref. 1)	47	14,441	6,500	7,600	7,760	1,950	14,230	52,481
Two-Stage	37	14,441	4,081	8,215	6,103	2,287	8,350	43,477
	47	14,441	3,896	6,625	5,542	1,974	7,164	39,642
	61	14,441	3,739	5,300	5,433	1,916	6,967	37,800
Three-Stage	37	14,441	3,347	6,633	6,098	2,285	5,558	38,362

*Includes cost of superheater and preheater.

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

(ALL DIMENSIONS IN FEET)

FIG. 1

SCHEMATIC OF FLUIDIZED BED HEAT EXCHANGER

FIG. 2

SKETCH OF BAFFLED TWO-STAGE FLUIDIZED BED CONFIGURATION

SKETCH OF TWO — STAGE FLUIDIZED BED HEAT EXCHANGER

HEAT EXCHANGER SURFACE DIMENSIONS		
	LOWER (FLUIDFIRE)	UPPER (NEW)
TUBE OUTER DIA., in.	0.715	0.750
TUBE INNER DIA., in.	0.500	0.684
FIN OUTER DIA., in.	1.375	1.480
FIN THICKNESS, in.	0.025	0.025
FIN SPACING, FINS/in.	7.3	7.3
HEIGHT OF FIN ABOVE DISTRIBUTOR PLATE, in.	0	0.5

SCHEMATIC OF ALTERNATE HEAT EXCHANGERS

TOP ELEVATION

a) CONFIGURATION 1

b) CONFIGURATION 2

FIG. 5

VARIATION OF NUSSELT NUMBER WITH SUPERFICIAL VELOCITY FOR BEDS WITH NARROW AND WIDE RANGES OF PARTICLE SIZES

FLUIDFIRE DISTRIBUTOR PLATE
BED LOAD = 110 lb OF Al_2O_3

SYMBOL	RANGE OF PARTICLE SIZES — MESH — dia. in.	AVERAGE DIA. — in
○	40 × 45—0.016—0.014	0.014
□	28 × 65—0.025—0.009	0.014

VARIATION OF NUSSELT NUMBER WITH SUPERFICIAL VELOCITY FOR FLUIDFIRE AND 3.4 AND 6.0 PERCENT OPEN AREA DISTRIBUTOR PLATES

BED = 110 lb OF Al_2O_3

SYMBOL	DISTRIBUTOR PLATE
—	FLUIDFIRE (FIG. 6)
□	3.4 PERCENT OPEN AREA
△	6.0 PERCENT OPEN AREA

VARIATION OF PRESSURE LOSS WITH SUPERFICIAL VELOCITY FOR FLUIDFIRE AND 3.4 AND 6.0 PERCENT OPEN AREA DISTRIBUTOR PLATES

SYMBOL	DISTRIBUTOR PLATE
○	FLUIDFIRE
□	3.4 PERCENT OPEN AREA
△	6.0 PERCENT OPEN AREA

VARIATION OF PRESSURE LOSS WITH DYNAMIC HEAD FOR FLUIDFIRE AND 3.4 AND 6.0 PERCENT OPEN AREA DISTRIBUTOR PLATES

DYNAMIC HEAD BASED ON INLET CONDITIONS

SYMBOL	DISTRIBUTOR PLATE
○	FLUIDFIRE
□	3.4 PERCENT OPEN AREA
△	6.0 PERCENT OPEN AREA

EFFECT OF BED LOAD ON HEAT TRANSFER

$V_S = 2.3$ ft/sec
BED MATERIAL IS Al_2O_3

SYMBOL	RANGE OF PARTICLE SIZES — MESH	AVERAGE DIA. — IN.
○	40 × 45	0.014
□	28 × 65	0.014

VARIATION OF NUSSELT NUMBER WITH SUPERFICIAL VELOCITY FOR DIFFERENT BED LOADS

DISTRIBUTOR PLATE OPEN AREA = 3.4 PERCENT

SYMBOL	BED LOAD — lb
□	110
○	200

VARIATION OF NUSSELT NUMBER WITH SUPERFICIAL VELOCITY FOR THREE DIFFERENT BED MATERIALS

FLUIDFIRE DISTRIBUTOR PLATE

SYMBOL	BED MATERIAL	BED LOAD — lb	NOMINAL PARTICLE SIZE — in.
—	Al ₂ O ₃ (FIG. 5)	110	0.015
○	SiC	100	0.015
□	STEEL SHOT	250	0.018
◻	STEEL SHOT	200	0.018

PHOTOGRAPHS OF BED MATERIALS ENLARGED BY A FACTOR OF 25

a) ALUMINA

b) SILICON CARBIDE

c) STEEL SHOT

VARIATION OF BED PRESSURE LOSS WITH SUPERFICIAL VELOCITY FOR THREE BED MATERIALS

FLUIDFIRE DISTRIBUTOR PLATE

SYMBOL	BED MATERIAL	BED LOAD — lb	BED HEIGHT — in.
□	STEEL SHOT	250	1.5
◻	STEEL SHOT	200	1.25
○	SiC	100	1.5
△	Al ₂ O ₃	110	1.5
⋈	Al ₂ O ₃	200	2.7

VARIATION OF ENTRAINMENT WITH HEIGHT ABOVE ALUMINA BED FOR FLUIDFIRE AND 3.4 PERCENT OPEN AREA DISTRIBUTOR PLATES

INLET AREA OF PROBE = $2.23 \times 10^{-3} \text{ ft}^2$

VARIATION OF ENTRAINMENT WITH HEIGHT ABOVE SILICON CARBIDE BED FOR FLUIDFIRE AND 3.4 PERCENT OPEN AREA DISTRIBUTOR PLATES

INLET AREA OF PROBE $\equiv 2.23 \times 10^{-3} \text{ ft}^2$

VARIATION OF ENTRAINMENT WITH HEIGHT ABOVE A STEEL SHOT BED FOR DISTRIBUTOR PLATE HAVING 3.4 PERCENT OPEN AREA

INLET AREA OF PROBE = $2.23 \times 10^{-3} \text{ ft}^2$

VARIATION OF TRANSPORT DISENGAGEMENT HEIGHT WITH SUPERFICIAL VELOCITY FOR DIFFERENT BED MATERIALS AND DISTRIBUTOR PLATES

SYMBOL	BED MATERIAL	DISTRIBUTOR PLATE
○	Al ₂ O ₃	FLUIDFIRE
□	Al ₂ O ₃	3.4 PERCENT OPEN AREA
△	SiC	FLUIDFIRE
▽	SiC	3.4 PERCENT OPEN AREA
◇	STEEL SHOT	3.4 PERCENT OPEN AREA

PERFORMANCE OF ALTERNATE HEAT EXCHANGER SURFACES

CONFIGURATION 1

a) NUSSELT NUMBER

b) HEAT TRANSFER RATE

FIG. 19

PERFORMANCE OF ALTERNATE HEAT EXCHANGER SURFACES

a) NUSSELT NUMBER

CONFIGURATION 2

b) HEAT TRANSFER RATE

VARIATION OF NUSSELT NUMBER FOR BEDS WITH AND WITHOUT BAFFLES

SYMBOL	OPEN AREA OF BAFFLE — PERCENT
○	22.4
□	10.5
—	NO BAFFLE (DATA FROM FIG. 6)

VARIATION OF BED PRESSURE LOSS AND BED MIXING RATIO WITH SUPERFICIAL VELOCITY

SYMBOL	OPEN AREA OF BAFFLE — PERCENT
○	22.4
□	10.5

a) BED PRESSURE LOSS

b) MIXING RATIO

VARIATION OF NUSSELT NUMBER WITH SUPERFICIAL VELOCITY FOR ONE AND TWO STAGE HEAT EXCHANGERS

BED MATERIAL IS Al_2O_3 DISTRIBUTOR PLATE OPEN AREA = 3.4 PERCENT

SYMBOL	HEAT EXCHANGER	CONFIGURATION
—	FLUIDFIRE	SINGLE STAGE (FIG. 6)
○	FLUIDFIRE	TWO STAGE
◻	NEW	SINGLE STAGE
□	NEW	TWO STAGE

VARIATION OF PRESSURE LOSS WITH SUPERFICIAL VELOCITY FOR TWO STAGE HEAT EXCHANGER

SCHEMATIC OF TWO-STAGE FLUIDIZED BED HEAT EXCHANGER

SCHEMATIC OF THREE-STAGE FLUIDIZED BED HEAT EXCHANGER