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ICP-1151

LIQUID-FLUIDIZED-BED HEAT EXCHANGER FLOW DISTRIBUTION MODELS

January 1979

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DEPARTMENT OF ENERGY

IDAHO OPERATIONS OFFICE UNDER CONTRACT EY-76-C-07-1540

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Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, Virginia 22161
Price: Printed Copy \$4.00; Microfiche \$3.00

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ICP-1151

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LIQUID-FLUIDIZED-BED HEAT EXCHANGER
FLOW DISTRIBUTION MODELS

by

L. T. Cole
C. A. Allen

Date Published - January 1979

ALLIED CHEMICAL CORPORATION
IDAHO CHEMICAL PROGRAMS - OPERATIONS OFFICE

PREPARED FOR THE
DEPARTMENT OF ENERGY
IDAHO OPERATIONS OFFICE
UNDER CONTRACT 41-1570-I-598

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ABSTRACT

Allied Chemical Corporation at the Idaho National Engineering Laboratory is developing liquid-fluidized-bed shell-and-tube heat exchangers for geothermal applications. Sand fluidized by geothermal water on the shell side prevents scaling and increases heat transfer coefficients over conventional heat exchangers. Tests were conducted on two instrumented fluidized-bed heat exchanger models, constructed primarily of plexiglass, which differ in tube bundle orientation. One contains a horizontal bundle and the other a vertical tube bundle.

Plexiglass construction allowed visual observation of flow patterns. The vertical model proved to have more uniform flow distribution and higher heat transfer coefficients than the horizontal model. The horizontal heat exchanger experienced piling on top of the tubes and areas of poor fluidization existed in the bed. Geometric considerations show that a horizontal design is more conducive to large flow rates than a vertical design. New design concepts for both vertical and horizontal assemblies and recommendations for further developmental work are presented.

SUMMARY

Developmental work on liquid-fluidized-bed heat exchangers was performed at the Raft River geothermal site, part of the Idaho National Engineering Laboratory. This report discusses information gained as a result of tests conducted on two different liquid fluidized-bed heat exchanger models.

The concept of a fluidized bed is not new, since they are used in many industrial applications. The concept of using a liquid fluidized bed to eliminate shell-side scaling in tube-and-shell heat exchangers is new. Scaling fluids such as geothermal water can be made to suspend or fluidize a bed materials, i.e., sand on the shell side of the heat exchanger. The action of the fluidized bed "scrubs" the exchanger tubes and eliminates scale and reduces corrosion. The added benefit of increased heat transfer coefficients is inherent with the fluidized bed.

Work done to date has not allowed detailed observation of a fluidized bed in a heat exchanger tube bundle configuration. For this reason, tests were conducted on two large-scale fluidized bed models constructed mainly of plexiglass. The models differ mainly in tube bundle orientation. One utilizes a horizontal tube bundle, while the other has a vertical tube bundle. The simulated tube bundles consist of 1.9 cm plexiglass rods in the place of tubing. In both units some of the dummy tubes were replaced with instrumented heater rods to determine shellside heat transfer coefficients. Results show that the vertical unit has higher heat transfer coefficients than the horizontal unit.

Visual observations showed the vertical unit flow distribution to be more uniform than that of the horizontal unit. The vertical unit operation is comparatively trouble free, while the horizontal unit results in piling of particles on top of the tubes and areas where the bed does not fluidize well due to constantly changing flow cross-sectional areas.

Even though the horizontal unit has more design problems, development of both vertical and horizontal type exchangers is desirable. Geometric considerations show that a horizontal unit is more conducive to large shell-side flow rates than a vertical unit. This is due to the window of fluid velocities (3-9 cm/sec), which must be maintained to properly fluidize the bed and maximize heat transfer. Horizontal vessels offer more cross-sectional area than vertical vessels for the same tube length.

Based on these tests, new design concepts for both horizontal and vertical assemblies were developed. Recommendations for further developmental work are made.

CONTENTS

ABSTRACT	ii
SUMMARY	iii
I. INTRODUCTION	1
II. EXPERIMENTAL OBJECTIVE	1
III. EXPERIMENTAL APPARATUS	2
1. Horizontal Model	2
2. Vertical Model	2
3. Instrumentation	5
IV. RESULTS AND DISCUSSION	5
1. Porosity	5
2. Flow Distribution	6
3. Heat Transfer	8
4. Design Recommendations	11
V. CONCLUSIONS	12
VI. NOMENCLATURE	13
VII. REFERENCES	15

FIGURES

1. Horizontal Liquid Fluidized-Bed Model	3
2. Vertical Liquid Fluidized-Bed Model	4
3. Measured Porosity as a Function of Velocity for 16 Mesh Silica Sand	7
4. Liquid Fluidized Bed Heat Exchanger Designed for Improved Flow Distribution	8
5. Typical Angular Dependence of the Heat Transfer Coefficient for a Horizontal Tube in a Fluidized-Bed ($W/m^2 - ^\circ K$)	9
6. Shell-Side Heat Transfer Coefficients Measured Using Fluidized-Bed Models	10

I. INTRODUCTION

Liquid fluidized-bed technology offers the potential for scale control and increased heat transfer coefficients in heat exchangers. Extensive work has been done with gas fluidized beds, but liquid fluidized beds are still in the developmental stage. Grimmett (et. al.)¹ reported the progress made to date.

Fluidized beds consist of a bed of solid particles (e.g. sand) with fluid passing upward through them. When the fluid reaches a velocity which causes the drag force on the individual particles to equal the particle weight, the particles are suspended or fluidized. The bed of particles will continue to expand as the velocity is increased and it will behave as a fluid until the terminal velocity² is reached. At terminal velocity, the particles will be entrained in the fluid.

This process of fluidization has been applied to liquid heat exchangers to eliminate the common problem of heat transfer surface scaling. The primary fluid (e.g. geothermal water) may be used to fluidize a bed material such as sand. The fluidizing action of the bed creates two distinct advantages over conventional shell-and-tube flow arrangements: (1) the scouring action of the bed prevents scaling and limits corrosion on the tubes and (2) the heat transfer coefficient for the fluidized bed is almost double the coefficient for a conventional exchanger. These two advantages make development of this type of heat exchanger desirable since preliminary cost estimates have shown fluidized bed heat exchangers are cost competitive with conventional units⁽¹⁾.

The successful development of a working, economical, liquid fluidized bed promises relief to the common industrial problem of shell-side heat exchanger fouling and costly downtime for cleaning. The technology is being developed for the Department of Energy to be used to transfer heat from geothermal water which causes severe scaling and corrosion problems in industrial domestic use.

A number of studies on heat transfer in liquid fluidized beds dealt exclusively with heat transfer, either from a tube wall to liquid or from a heated object to liquid in vertical cylindrical glass vessels of small scale (3,4,5,6,7,8). This paper reports results from full scale instrumented plexiglass "unit cell" shell-and-tube heat exchanger models. These models allowed visual observation of the fluidized bed in a tube bundle configuration. Flow distribution characteristics and problems of different fluidized bed designs were noted. Experimental comparisons of outside tube heat transfer coefficients were made in both vertical and horizontal assemblies with and without a bed present.

II. EXPERIMENTAL OBJECTIVES

The primary objectives were to: (1) determine local shell side heat transfer coefficients, (2) visually observe flow distribution behavior and problems, and (3) give design information for full-scale commercial units.

The models were designed to measure outside heat transfer coefficients as a function of fluid velocity. Since heat transfer coefficients vary with circumferential position on fixed exchanger tubes, the instrumented tubes were designed to rotate a full 360 degrees for measurement of angular dependence.

III. EXPERIMENTAL APPARATUS

1. Horizontal Model

The horizontal fluidized bed model consists of a 0.91 m diameter shell which is 0.30 m wide. The shell houses 688 dummy tubes made from 1.9 cm diameter plexiglass rods placed in an equilateral triangular pitch arrangement with a pitch-to-tube-diameter ratio of 1.5.

Water used to fluidize the bed of 16 mesh (1 mm) sand enters the unit through the bottom as shown in Figure 1. Water passes through a distributor plate which serves two functions: (1) to give uniform flow distribution to the bed by providing a flow resistance, and (2) to support the bed when flow is stopped. The distributor plate consists of a 0.63 cm carbon steel plate drilled with 0.63 cm diameter holes to give a 23% void area. A screen was placed on the back to prevent sand from falling through when flow is stopped. A second plate with 0.63 cm holes drilled to 23% voidage is placed above the top of the tubes to help prevent bed carryover as the flow exits the top of the unit.

The tube bundle assembly is sandwiched between two 5.1 cm thick plexiglass plates to allow visual observation. The model can be visualized as a 30 cm long slice out of a 91 cm diameter heat exchanger. The back plexiglass plate is drilled with five 2.5 cm holes to accommodate the instrumented heater rods used to determine heat transfer coefficients.

2. Vertical Model

The vertical fluidized-bed model shown in Figure 2 was constructed from a piece of 122-cm-high, 25.4-cm-O.D., 0.95-cm-wall-thickness plexiglass tubing. The tubing has a plexiglass flange made to bolt to a 25.4 cm weld-neck flange. The diameter is reduced to an 8 cm supply line by 25 x 15 cm reducer and then to a 15 x 8 cm reducer and then to an 8 cm weld-neck flange. This assembly serves as a plenum to supply the vertical unit with uniform flow. A distributor plate with 0.32 cm diameter holes drilled to give a 30% void area is bolted between the two flanges.

The simulated tube bundle, constructed of 1.9 cm diameter plexiglass rods, is inserted and supported from the top of the unit. The tubes are arranged on an equilateral triangular pitch having a pitch/tube diameter ratio of 1:5. Water passes through the distributor plate to fluidize the bed around the tubes and exits the exchanger via a weir located in the top of the unit. The weir consists of eight 1.3-cm-wide slots and a 5.1-cm-long cut around the periphery of the shell. A weir is used to maximize the amount of tube covered by the fluidized bed and still provide an adequate disengagement space for the bed. Vertical

Figure 1. Horizontal Liquid Fluidized Bed Model

Figure 2. Vertical Liquid Fluidized Bed Model

cylindrical configurations are simpler to build if the tubes are introduced from the top of the vessel.

3. Instrumentation

The models accept instrumented heater rods in the place of dummy tubes at several locations in the tube bundle. The test section consists of a 0.95 -cm-diameter, 17.8-cm-long 500-W heater press-fitted into a 1.9 cm O.D. sheath. The heater sheath has a 0.1 cm deep groove machined from one end to the middle of the section. A 0.16 cm diameter type J stainless steel clad thermocouple is soldered in place so that the thermocouple measures the skin temperature of the sheath. The heater tube is designed to allow 360 degree rotation of the thermocouple to determine angular dependence of the heat transfer coefficient. The bulk temperature of the fluid is measured in both the vertical and horizontal vessels, by three thermocouples at various locations in the vessel.

Water flow rate through each exchanger is measured by pressure drop across an orifice plate. The pressure differential is measured with a calibrated differential pressure transducer. Heater output is determined by direct measurement of the heater voltage and current.

IV. RESULTS AND DISCUSSIONS

1. Porosity

A test was conducted using the vertical model to determine the relationship between fluid cross-sectional velocity and porosity of 16 mesh (1 mm mean partical diameter) sand.

The data shown in Figure 3 were fitted using the least squares method. Equation 1 relates porosity (ϵ) to cross sectional velocity, V , in cm/sec.

$$\epsilon = .222 V^{.662} \quad (1)$$

This formula predicts a terminal velocity of 9.7 cm/sec. The standard formula for terminal velocity⁽⁹⁾ is:

$$v_t^2 = \frac{4gd_p (\rho_s - \rho_f)}{3 D_c \rho_f} \quad (2)$$

Using a drag coefficient (D_c) of 1.6, equation 2 predicts a terminal velocity of 11.6 cm/sec. This gives a difference of 16.5% between extrapolated results and calculated predictions. This may be due to sand size distribution around the mean particle diameter and could be accentuated by sand particle attrition.

2. Flow Distribution

Visual observation of the two operating models showed the vertical model to have more uniform flow distribution than the horizontal model. Sand piling occurs on top of the horizontal model tubes due to flow separation from the tubes in the cross-flow design. The horizontal model also shows dead areas at the sides of the bed. This is due to the constantly changing cross-sectional area of the cylindrical shell.

Visual observation of dead areas in the horizontal model prompted modification of the distributor plate. Fifty percent of the 0.63 cm holes located in the outer four in. of either side of the distributor plate were enlarged to 1.26 cm holes in an attempt to increase flow to the edges. Efforts to force flow to the edges were only partially successful, since the flow seeks the least resistant path through the bed. Dead areas in commercial heat exchangers would be a serious problem. Therefore, it seems necessary to design the tube bundle with a constant cross-sectional area. This can be done by using a rectangular tube bundle and side baffles as shown in Figure 4. The lost shell volume could be used as a header system as shown in the diagram. This will simplify flow distribution and ensure complete fluidization in horizontal exchanger.

Insufficient pressure drop across the distributor plate allows the bed to collapse in an area and not be refluidized, since the plate offers relatively little resistance in non-collapsed areas. The pressure drop across a fluidized bed is constant and is described by equation 3.

$$P_b = (\rho_s - \rho_f) (1 - \epsilon) H \frac{g}{g_c} \quad (3)$$

$$\text{where: } (1 - \epsilon) (H) = \text{constant.} \quad (4)$$

As with a gas fluidized bed⁽⁹⁾, it is not easy to postulate universal rules for the proper ratio of pressure drop across the distributor (P_d) to pressure drop across the bed (P_b). As flow increases, P_d changes while P_b remains constant. Limited experience with distributor plate pressure drops across over models indicates that smaller P_d/P_b ratios are needed for a liquid fluidized bed than in a gas fluidized bed. P_d/P_b ratios of 10% at moderate cross-sectional flow velocity (6 cm/sec) appears to be adequate.

Figure 3. Measured Porosity as a Function of
Velocity for 16 Mesh Silica Sand

Figure 4. LIQUID FLUIDIZED BED HEAT EXCHANGER DESIGNED FOR IMPROVED FLOW DISTRIBUTION

A corrosion study revealed that the areas of piling on top of horizontal tubes are susceptible to corrosion and scaling. This piling may be reduced by improved tube bundle design. A horizontal velocity component across the top of the tubes, which will reduce piling, may be introduced by fabricating the tube sheet in configurations other than the traditional equilateral triangular or square pitch. A nonequilateral triangular pitch can increase both the tube-side tangential velocity as separation from each tube to reduce piling. This development work could increase the shell side heat transfer coefficient as discussed in the following section.

3. Heat Transfer

The heat transfer analysis assumes a uniform heat flux out of the heater rods, and shell-side heat transfer coefficients were calculated using "Newton's Law of Cooling":

$$q = hA (T_s - T_f) \quad (5)$$

The horizontal tubes showed a definite angular dependence of the heat transfer coefficient. As discussed in the Flow Distribution section of this paper, the horizontal unit experiences flow separation and piling. Non-uniform flow patterns around the horizontal tube cause a circumferential difference in heat transfer coefficients. The coefficients were measured at 45° increments from the vertical and the representative coefficient reported is an average of these readings. A typical representation of the radial dependence of these coefficients is shown in Figure 5. Reduction of the piling could increase the average of the coefficients by as much as 15%. The vertical unit did not show any angular dependence. The results of the tests are shown in Figure 6.

Figure 5. Typical Angular Dependence of the Heat Transfer Coefficient for a Horizontal Tube in a Fluidized Bed ($W/m^2 \cdot ^\circ K$)

Heat transfer coefficients for the vertical fluidized bed are higher than for the horizontal bed at the same cross-sectional velocity. With no bed, the converse is generally true⁽¹⁰⁾ as shown in Figure 6. When the two models are run with a bed, the vertical coefficients are higher, due to more uniform fluidization. Most experimenters agree that the increased heat transfer of a fluidized bed is due to destruction of the boundary layer around the tubes⁽⁵⁾. The boundary layer in a crossflow situation will probably be more difficult to uniformly minimize than in a parallel flow situation in the presence of a fluidized bed. Therefore, the vertical unit provides an upper limit for the coefficient achievable in the horizontal unit.

Interpretation of the data shown in Figure 6 is more complex in the case of the horizontal unit. The cross-section velocity reported is based on the total open cross-sectional area and neglects dead areas, since some flow does go through this non-moving bed. As the flow rate increases, the dead areas at the edge of the horizontal unit become smaller creating a more uniformly fluidized bed. This means that the actual cross-sectional velocity does not change linearly with increased flow rate. Bed flow patterns in both units constantly shift which accounts for scatter in the data.

Figure 6. Shell-Side Convection Coefficients Experimentally Determined in Fluidized Bed Models

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Several authors report relationships for fluidized-bed heat transfer coefficients which agree with the basic form of equation 6(1,5,6) but not upon the value of a, b, c, d, e, and f.

$$Nu = a Re_t^b Pr^c \epsilon^d (1-\epsilon)^e \left(\frac{d_p}{d_t}\right)^f \quad (6)$$

This equation can be simplified for a constant fluid temperature to:

$$Nu = K O(\epsilon) \quad (7)$$

where K is a constant and,

$$O(\epsilon) = \epsilon^d (1-\epsilon)^e \quad (8)$$

Using the correlation given by Grimmett, et al.

$$Nu = 1.83 \left(\frac{d_p}{d_t}\right)^{0.2} Re_t^{0.52} Pr^{0.33} \epsilon^{1.53} (1-\epsilon)^{0.48} \quad (9)$$

the heat transfer coefficient for 130°C water is 1.7 times greater than the coefficient for water at 20°C due to changes in fluid properties. This temperature dependence explains why the heat transfer coefficients measured at 20°C are relatively low. It should be stressed that the heat transfer coefficient is strongly dependent on both porosity and temperature.

4. Design Recommendations

All evidence given so far indicates that the vertical model is preferable to the horizontal. However, the horizontal model is superior in two aspects:

- (1) A vertical exchanger requires tube bundle introduction from the top of the unit which favors an even number of tube passes and makes it difficult to put several stages in a single vessel. A disengagement section of approximately 15 cm must be left above the bed to prevent particle carryover. This means that the top 15 cm of tubes will not be covered with the fluidized bed. This short section of tubes will be susceptible to increased corrosion rates and scaling.
- (2) Geometric considerations show that the horizontal design is more conducive to large flow rates. This is due to a larger cross-sectional area presented by horizontal vessels for the same length and number of tubes. The velocity used to fluidize the bed has a narrow range (3-9 cm/sec for 1 mm silica

sand) which must be maintained for proper bed operation. Applications such as geothermal power plants demand large flow rates. Thus, the horizontal unit still merits development and consideration.

One possible way of increasing the flow capacity of the vertical unit is to supply a larger shell than is needed for the tube bundle. Since the bed is isothermal and well mixed, having portions of a vertical assembly with no tubes will not impair the heat transfer efficiency.

A second possibility to increase flow rates through heat exchangers is to increase the density of the bed material. Examination of Equation 2 shows that this would allow higher fluidization velocities. For example, use of 1 mm steel shot as a bed material would give a terminal velocity of 23.6 cm/sec or more than twice the velocity for 1 mm sand, which has an experimentally measured terminal velocity of 9.7 cm/sec. Experimental work needs to be done with heavier particles since aggregate fluidization may take place and cause design problems. The pressure drop associated with fluidization of a silica sand bed material is approximately equal to the head loss of the water through the exchanger as calculated by Equation 3. If steel shot, having a density of three times that of silica sand, is used for a bed material, the pressure drop due to the fluidization of the material is four times the pressure drop with a silica sand bed.

V. CONCLUSIONS

Vertical liquid-fluidized-beds are characterized by:

- (1) Uniform fluidization.
- (2) Slightly higher heat transfer coefficients than a horizontal unit.
- (3) Approximately 15 cm of each tube at the top of the unit must be left uncovered by the bed.

Horizontal liquid-fluidized-beds are characterized by:

- (1) Capability of handling larger shell-side fluid flows than a vertical unit for the same tube lengths and pitch-to-tube-diameter ratio.
- (2) Complete covering of all tubes by the fluidized-bed.
- (3) Non-uniform fluidization.

The vertical fluidized bed heat exchanger offers more trouble free operation than a horizontal unit. The horizontal fluidized bed offers more design problems and will take more developmental effort, but it is capable of handling larger flow volumes necessary for many industrial projects due to the greater cross-sectional area presented to the flow. The fluidizing velocity is limited to a narrow range (3-9 cm/sec for 16 mesh silica sand) to give proper fluidization.

Two options are open to increase the flow capability of a vertical unit:

- (1) Put the vertical tube bundle in a larger shell than the tube bundle requires. The isothermal and mixing characteristics of a fluidized bed should cause the heat exchanger to operate normally at increased flow rates.
- (2) Increase the density of the bed particles to give higher fluidization velocity. This option may give problems with aggregate fluidization.

The most valuable interpretation of the heat transfer data comes from direct comparison between the two models. The horizontal unit with a bed has lower coefficients than the vertical with a bed due to poorer fluidization properties. Design improvement can bring these closer together, but the vertical unit provides the probable upper limit coefficient achievable in the horizontal unit.

Regardless of the design configuration used, liquid fluidized beds provide a solution of tube surface scaling and will reduce corrosion caused by fluids, such as geothermal water, in a heat exchanger. Estimates have shown the liquid fluidized bed to be cost competitive to a conventional tube-and-shell type heat exchanger. These estimates did not include the significant cost factor associated with down-time for cleaning conventional units. Down-time for cleaning can be minimized with a properly designed liquid fluidized bed.

VI. NOMENCLATURE

a,b,c,d,e = dimensionless variables

A = heat transfer area (m^2)

d_p = particle diameter (m)

d_t = tube diameter (m)

D_c = drag coefficient (dimensionless)

g = acceleration of gravity

g_c = gravitation constant

h = convection heat transfer coefficient (W/m^2-K)

H	= bed height (m)
K	= dimensionless constant
Nu	= Nusselt number (dimensionless)
P _b	= pressure drop across the bed (Pa)
P _d	= pressure drop across the distributor plate (Pa)
Pr	= fluid Prandtl number (dimensionless)
q	= rate of heat transfer (W)
Re _t	= Reynolds number based on terminal velocity (dimensionless)

$$Re_t = \frac{V_t d_p}{\nu}$$

T	= temperature (°C)
T _f	= bulk fluid temperature (°C)
T _s	= heater surface temperature (°C)
V	= cross-sectional velocity (m/sec)
V _t	= terminal velocity, = 1 (m/sec)
ε	= porosity, void fraction (dimensionless)
ρ _f	= fluid density (g/cm ³)
ρ _s	= solid density (g/cm ³)
ν	= kinematic viscosity (m ² /sec)

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