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Heat Exchanger

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FROM: J. L. Wantland

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THERMAL CHARACTERISTICS OF A DELTA ARRAY
HEAT EXCHANGER

ABSTRACT

The heat transfer and fluid friction characteristics have been determined for a liquid flowing parallel to the tube bundle of a heat exchanger consisting of one hundred and two 1/16-inch O.D. tubes arranged in a delta or triangular array. These results may be expressed by the following empirical equations:

From $N_{Re} = 900$ to $N_{Re} = 2,300$, $N_{Nu}/N_{Pr}^{0.4} = 0.47 N_{Re}^{0.36}$ and,

from $N_{Re} = 2,300$ to $N_{Re} = 13,000$, $N_{Nu}/N_{Pr}^{0.4} = 0.0066 N_{Re}^{0.91}$

From $N_{Re} = 900$ to $N_{Re} = 13,000$, $f = 90/N_{Re} + 0.033$,

where f is defined by $f = \frac{\Delta P}{\frac{L}{D_w} \frac{\rho u^2}{2g_c}}$.

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NOMENCLATURE

- A heat transfer area, outside tube surface, ft^2
- c specific heat of water, $\text{Btu}/\text{lb}_m\text{-}^\circ\text{F}$
- d_i tube inside diameter, ft
- d_o tube outside diameter, ft
- D_h equivalent heated diameter used in N_{Nu} calculated for a single central flow channel or the so-called infinite array, ft
- D_w equivalent wetted diameter = 4 cross-sectional flow area/wetted perimeter, used in N_{Re} , ft
- f friction factor defined by $f = \frac{\Delta P}{\frac{L}{D_w} \frac{\rho u^2}{2g_c}}$
- g_c gravitational conversion factor = $4.170 \times 10^8 \text{ lb}_m\text{-ft}/\text{lb}_f\text{-hr}^2$
- h tube outside surface coefficient of heat transfer based on A, $\text{Btu}/\text{hr-ft}^2\text{-}^\circ\text{F}$
- k thermal conductivity of water, $\text{Btu}/\text{hr-ft-}^\circ\text{F}$
- L total length of tube bundle parallel to axis of shell, ft
- ΔP static pressure drop through the tube bundle, lb_f/ft^2
- Q heat transferred from tube to water, Btu/hr
- s tube spacing, outside wall to outside wall, ft
- s' spacing from peripheral tubes to heat exchanger shell wall, ft
- S tube spacing ratio, s/d_o
- S' wall spacing ratio, s'/d_o
- t_{wi} average inside tube wall temperature, $^\circ\text{F}$
- t_{wo} average outside tube wall temperature, $^\circ\text{F}$
- t_a average mixed-mean water temperature, $^\circ\text{F}$

u average velocity of water flowing around the tube bundle, ft/hr

μ dynamic viscosity of water, $\text{lb}_m/\text{hr-ft}$

ρ density of water, lb_m/ft^3

N_{Pr} Prandtl modulus, $c\mu/k$

N_{Nu} Nusselt modulus, $h D_h/k$

N_{Re} Reynolds modulus, $D_w u \rho / \mu$

THERMAL CHARACTERISTICS OF A DELTA ARRAY

HEAT EXCHANGER

INTRODUCTION

The need for the transition region heat transfer and fluid friction characteristics of systems with flow parallel to closely spaced bundles of cylindrical tubes resulted in the initiation of an experimental program at the Oak Ridge National Laboratory on compact tubular heat exchangers. The first study^{1,2} related to the characteristics of a square array tube bundle. This report presents the results for fluid friction and heat transfer in a delta array tube bundle which was designed with physical characteristics similar to those of the square array system.

EXPERIMENTAL APPARATUS

The heat exchanger studied consisted of a triangular array of 102 approximately 3/16-inch O.D. tubes enclosed in a hexagonal cross-section plexiglas shell. This is illustrated in Figure 1, and the pertinent physical characteristics of the heat exchanger are listed in Table 1. The tubes were spaced 0.036-inch apart by spacers made from 0.006-inch thick stainless steel strips. Six complete spacer assemblies, located at 11-inch intervals along the tube bundle were used. The heat exchanger used for the experimental study is shown partially installed in Figure 2. The copper header

1. Superscript numbers refer to Bibliography.

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Fig. 1 Cross-Section of a Full-Scale Model of the Delta Array Heat Exchanger

TABLE I

PHYSICAL CHARACTERISTICS OF THE DELTA ARRAY HEAT EXCHANGER

Number of tubes	102
Rectified length of tubes between water side header faces, ft	5.94
Total length of tube bundle parallel to axis of shell, L, ft	6.05
Tube inside diameter, d_i , ft	0.0116
Tube outside diameter, d_o , ft	0.0158
Tube spacing, outside wall to outside wall, s, ft	0.00300
Tube spacing ratio, S	0.190
Average tube to shell wall spacing, s' , ft	0.00242
Wall spacing ratio, S'	0.153
Minimum flow cross-sectional area around tubes (excluding spacers), ft^2	0.0126
Tube outside heat transfer surface area, A, ft^2	30.0
D_h , ft	0.00880
D_w , ft	0.00878
L/D_w	688

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Fig. 2 The Delta Array Heat Exchanger Being Installed for Experimentation

plates were electrically connected to the secondary of a heavy duty welding transformer and the tube bundle was used as an electric resistance heater. The electrical input of the tube bundle was controllable over a wide power range.

The water flow through the heat exchanger was determined by weighing the water discharged from the system using a calibrated platform balance and an electrical interval timer. The inlet and outlet mixed-mean water temperatures were measured by copper-constantan thermocouples placed in four-section modified disc and doughnut mixing chambers. At each of the temperature measuring locations, two thermocouple wells were located in a plane perpendicular to the axis of the mixing chamber such that the junction of one thermocouple coincided with the axis, while the junction of the other was approximately midway between the axis and the mixing chamber wall. The output emf of each thermocouple was fed into a calibrated Brown recording potentiometer. The inside tube wall temperatures were measured at four longitudinal locations: 11-inches upstream of the tube bundle center line, right at the tube bundle center line, 11-inches downstream of the tube bundle center line, and 22-inches downstream of the tube bundle center line. At each of these longitudinal temperature measuring locations, a 102 element copper-constantan thermopile was situated with a hot junction in each of the 102 tubes and the cold junctions in a common ice bath. These thermocouples were not attached to the tubes but approached thermal equilibrium with the tube wall at each location by radiant heat transfer. The output emf of each thermopile was fed through voltage suppression circuits into calibrated Brown

recording potentiometers. Two sets of piezometric rings, each consisting of six static pressure taps, were used in conjunction with two-fluid manometers to measure the pressure drop through the heat exchanger. The pressure drop was obtained across both the entire tube bundle and a four-foot section of straight tubes midway between the headers.

METHOD OF CALCULATION

The average inside tube wall temperature at each location was obtained by dividing the output emf of each thermopile by 102. The relationship between emf and temperature for these thermocouples was sufficiently linear in the operating range utilized to permit this averaging technique to be used with a high degree of accuracy. Thus, the average inside tube wall temperature gradient along the length of the tube bundle was established.

An analytical investigation showed that for the electrical frequency and current range of this study, skin effect in the tubes would be negligible. Also, since all tubes have the same length and constant cross-sectional area and since the change of electrical conductivity of the tube metal in this temperature range was negligible, the heat generation per unit volume could be considered constant and the equation for uniform heat generation in a cylindrical tube with heat transfer from only the outer wall would apply. Using the physical properties and parameters that apply to this heat exchanger, the equation $t_{wi} - t_{wo} = 4.16 \times 10^{-6} Q$ was derived to determine the outside tube wall temperature from the inside tube wall temperature. This term, the temperature drop across the tube wall, was found to average

about 2°F.

For uniform heat transfer from the tubes and small axial Δt in the fluid, it may be assumed that the tube wall temperature varies linearly along the straight section of the tube bundle. This was verified by inspection of the data obtained. From the inlet and outlet mixed-mean water temperatures, the average mixed-mean water temperature was calculated and the average temperature difference for heat transfer, $(t_{wo} - t_a)$, was determined. The coefficient of heat transfer was then calculated from

$$h = \frac{Q}{A(t_{wo} - t_a)}.$$

In analyzing the heat transfer data, the fluid properties used in evaluating the Reynolds, Nusselt, and Prandtl moduli were taken at the arithmetic average of the inlet and outlet mixed-mean water temperatures. In all cases, the Reynolds moduli were based on the equivalent wetted diameter, D_w , and the Nusselt moduli were based on the equivalent heated diameter, D_h .

The friction factor was calculated using the Fanning equation, $f = \frac{\Delta P}{\frac{L}{D_w} \frac{\rho u^2}{2g_c}}$, where the hydraulic diameter, D_w , is based on the total

wetted perimeter of the flow channel. The friction characteristics presented were determined from the pressure drop across the entire tube bundle. However, these data agree quite closely with the friction characteristics determined from the four-foot length of straight tubes in the center of the tube bundle. While the friction characteristics reported were based on isothermal data, the isothermal pressure drop data agree quite closely

with those determined with heat transfer at the tube wall. The fluid properties were based on the arithmetic average of the inlet and outlet mixed-mean water temperatures.

CORRELATION OF RESULTS

The heat transfer parameter, $N_{Nu}/N_{Pr}^{0.4}$, and friction factor, f , are plotted versus the Reynolds modulus in Figures 3 and 4. These data can be expressed to $\pm 10\%$ by the following empirical equations:

From $N_{Re} = 900$ to $N_{Re} = 2,300$, $N_{Nu}/N_{Pr}^{0.4} = 0.47 N_{Re}^{0.36}$ and,

from $N_{Re} = 2,300$ to $N_{Re} = 13,000$, $N_{Nu}/N_{Pr}^{0.4} = 0.0066 N_{Re}^{0.91}$.

From $N_{Re} = 900$ to $N_{Re} = 13,000$, $f = 90/N_{Re} + 0.033$,

where f is defined by $f = \frac{\Delta P}{\frac{L}{D_w} \frac{\rho u^2}{2g_c}}$.

A discussion of errors is given in the Appendix. In Figures 3 and 4, the delta array data are also compared with the data obtained from the square array heat exchanger previously reported^{1,2}. In addition the conventional correlations for ordinary ducts in the turbulent regime and the correlations for parallel plate systems in the laminar regime are shown in these figures.

Fig. 3. Heat Transfer Characteristics of the Delta Array Heat Exchanger.

Fig.4 . Isothermal Friction Characteristics of the Delta Array Heat Exchanger.

DISCUSSION AND CONCLUSIONS

The heat transfer characteristics for the delta array configuration were on the average 25% lower than would be predicted from the conventional correlation for turbulent flow in ordinary ducts, while the friction characteristics were 40% higher. At lower Reynolds moduli, the heat transfer data indicate a leveling off which is typical of laminar flow, and at higher Reynolds moduli the data suggest an approach to the conventional heat transfer correlation. One reference³ indicates that in the Reynolds modulus range of from 70,000 to 700,000, the heat transfer coefficients of a delta array tube bundle were 40% higher than would be predicted by

$$N_{Nu}/N_{Pr}^{1/3} = 0.023 N_{Re}^{0.8} .$$

The data indicate a definite break in the heat transfer characteristics at a Reynolds modulus of 2,300, and a faint break in the friction characteristics in the same region. However, it is indicated that there is not a definite change from laminar flow to turbulent flow, but that in this type of configuration there is a wide transitional region with mixed flow. Since the heat transfer is less than would be predicted from the conventional correlation, it is believed that fully established turbulent flow is not developed below Reynolds moduli of 20,000 or more. It is difficult to interpret transitional friction characteristics from the data presented since it is a composite of spacer drag as well as tube skin friction.

Throughout the entire experimental range, the heat transfer characteristics of the delta array heat exchanger are equal to or superior to those of the square array heat exchanger while the friction characteristics of the delta array configuration are consistently lower than those of the square array configuration. Consequently, for a given flow rate and pressure drop (i.e., heat removal capacity and required pumping power), the delta array configuration is significantly superior to the square array configuration since the heat transfer surface can be more effectively swept by the heat transfer fluid. It should be noted that at both extremes of the experimental range where the friction characteristics of both heat exchangers are close together, the heat transfer characteristics are further apart indicating a correlation between heat transfer and fluid friction characteristics. However, in this region the heat transfer is less and the pressure drop is greater for either configuration that would be obtained by flow through cylindrical ducts.

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APPENDIX - DISCUSSION OF ERRORS

The water temperature measurements and flow measurements were believed to be accurate to $\pm 1\%$. All the instruments were calibrated before experimental operation and reproducible readings were obtainable. It is believed, therefore, that the quantity of heat transferred is accurate to $\pm 2\%$.

The heat exchanger was accurately constructed and during final assembly its physical measurements were checked to an accuracy of better than $\pm 1/2\%$. The calculated Reynolds moduli are believed to be accurate to $\pm 3\%$.

The voltage suppression circuits and associated recording potentiometers were calibrated and checked periodically. The inside tube wall temperature measuring thermopiles were checked against the water temperature measuring thermocouples by allowing preheated water to flow through the heat exchanger with no heat transfer. Under these conditions, all thermocouples and thermopiles should indicate the same temperature. The tube wall temperatures were believed to be accurate to $\pm 2\%$, the temperature difference for heat transfer to $\pm 5\%$ and the resulting heat transfer characteristics to $\pm 10\%$.

As can be seen from Figure 4, the precision obtainable in determining the fluid friction characteristics was greater than that obtained in determining the heat transfer characteristics. The major uncertainty in the calculated friction factor was probably in the correct determination of the pressure drop across the tube bundle. It is possible that disturbances in the flow pattern, particularly where the fluid emerged from the tube bundle, produced erroneous pressure drop readings. However, the friction characteristics obtained across the entire tube bundle agreed quite closely with those obtained

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across the four-foot length of straight tubes in the center of the bundle and the friction characteristics presented are believed to be accurate to $\pm 10\%$.

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