# Heat Transfer to Water Flowing Parallel to a Rod Bundle

**PHILIP MILLER, JAMES J. BYRNES, and DAVID M. BENFORADO**

Walter Kidde Nuclear Laboratories, Inc., Garden City, New York

**Heat transfer was measured for water flowing parallel to 5/8-in. rods in a triangular lattice (0.914-in. spacing). Velocity was varied from 5 to 20 ft./sec., temperature from 150° to 325°F., and heat flux from 50,000 to 200,000 B.t.u./(hr.)(sq. ft.). Reynolds numbers ranged from 70,000 to 700,000. Coefficients were 40% higher than predicted from the Colburn equation for flow in pipes by use of the equivalent diameter. The effect of heated length was studied. No variation in local coefficient around the rod periphery was found. Pressure drops were 65% higher than predicted by the Fanning equation (by use of $D_e$).**

No published data have been found for the case of heat transfer to a fluid in turbulent flow outside rods or tubes in a bundle, with flow parallel to the rod axes. In the most recent general texts on heat transfer the following information is offered. McAdams (*8*) and Jakob (*7*) do not mention this case. Eckert (*5*) recommends using the equation for turbulent flow inside tubes, with an equivalent diameter $D_e = 4 \times$ flow area $\div$ wetted perimeter. His recommendation is apparently made on theoretical grounds, as no data are cited. In the section on heat transfer in "The Chemical Engineers' Handbook" (*10*) McAdams states that test data for gases are in rough agreement with equations for gases inside pipes when the equivalent diameter is used. Experimental results have been published on shell-side heat transfer in unbaffled shell-and-tube heat exchangers, and these were correlated by Donohue (*4*). However, in these studies the local velocity within the tube bundle was not known, and the average velocity was used in the correlation. Because of the considerable space between the tube bundle and the shell, the leakage was undoubtedly sufficient to make the local velocity within the tube bundle much less than the average velocity. Presumably on this account Donohue's correlation predicts values of shell-side coefficients that are one-half to one-third the values obtained by Eckert's procedure. Consequently, it is not safe to apply Donohue's equation to cases other than shell-and-tube heat exchangers of the type for which the original data were obtained.

The lack of information on parallel-flow heat transfer in rod bundles no doubt reflects the absence of examples of its occurrence in industrial equipment. In baffled shell-and-tube heat exchangers true parallel flow probably does not occur; and in unbaffled heat exchangers it has evidently proved expedient to use empirical correlations involving average velocities and various correction factors rather than to measure the leakage and make more basic correlations. However, a need for reliable data on parallel-flow heat transfer has developed with the advent of nuclear reactors. Among the large variety of reactor designs under serious consideration for various applications, a number involve parallel-flow heat transfer from cylindrical rods at high heat fluxes and large Reynolds numbers. In particular, some of these contemplate the use of pressurized water at high temperature as the coolant for power-producing reactors. Both technical and economic reasons make it important to know the heat transfer coefficients for reactor design with considerable accuracy. For this reason an experimental investigation was undertaken of parallel-flow heat transfer to water flowing through a lattice of cylindrical rods.

The experimental equipment was of a rather large scale, as evident from the photograph in Figure 1; the maximum water-circulation rate was 1,100 gal./min., provided by a 25-hp. pump. Several factors contributed to this large size. The first was the decision to study a section of a full-scale lattice of ⅝-in. rods, rather than a small-scale model. Then, in the absence of advance knowledge of the wall effect, a lattice of reasonably large cross section was needed; the one used contained thirty-seven rods enclosed in a 6¼-in. cylinder (plus eighteen partial rods attached to the boundary wall). Finally, the interest in relatively high velocities, up to 20 ft./sec. maximum, contributed to the high circulation rate.

The lattice was made up of unheated dummy aluminum rods except for one test rod, which had a 4- or 8-in.-long middle section internally heated by an electrical resistance coil. Measurements were made at relatively high heat fluxes [50,000 to 200,000 B.t.u./(sq. ft.)(hr.)], corresponding to a maximum power density as high as 0.8 kw./in. of heated length. Development of a satisfactory heated test rod, and of a method of measuring the temperature of the heated surface, proved to be the major problems of this investigation. All the rods had tapered ends and were supported vertically between perforated plates, so that no change of flow direction was required in the entrance and exit sections. Frictional pressure drops were measured along the rod length.

This work was concerned primarily with obtaining heat transfer data for parallel flow of water in a triangular lattice of cylindrical rods at high temperature, high flow rate, and high heat flux. The maximum water temperature used was 325°F. at an imposed pressure of 140 lb./sq. in. gauge to ensure the absence of boiling at the heated surface, which reached a maximum temperature of 360°F. This temperature limit, which is several hundred degrees below the temperatures that would be encountered in water-cooled power reactors, is a compromise with the rapidly increasing cost and complexity accompanying the higher temperatures. In addition to the normal lattice studies, measurements were made in four different lattice positions to study the wall effect, the effect of a neighboring control-rod thimble (⅞-in. diam.), and the effect of severe lattice distortion. Information was also obtained on the variation of the local heat transfer coefficient around the rod periphery and on the effect of heated length.

## DESCRIPTION OF EQUIPMENT

### Circulation System

A flow diagram of the water loop is shown in Figure 2. The loop was constructed mainly of welded standard carbon steel pipe with bolted flange connections. Starting at the lower left-hand corner of the diagram and proceeding clockwise, water flowed upward through the test lattice, which was enclosed in a flanged section of 10-in. pipe. The water then flowed through an 11-ft. horizontal run (22 pipe diameters) of 6-in. pipe before passing through an orifice for flow measurement. After a 2½-ft. horizontal run (5 pipe diameters) it entered a vertical downcomer (8-in. pipe) leading to the suction side of the circulating pump. A surge tank consisting of an 18-in. section of 12-in. pipe with a total volume of 9 gal. was mounted above this downcomer. The tank was separated from the circulating system by a bottom plate with a 1-in. opening. System pressure was adjusted by introducing compressed nitrogen into the gas space in the upper part of the surge tank, which was equipped with a liquid-level gauge glass. The pump was a Weinman bronze-fitted, horizontal split-case, cast-iron centrifugal with outboard bearings, direct-driven by a 25-hp. motor. Flow rate was controlled by means of both a gate valve in a 4-in. by-pass line around the pump and a gate valve in the main 6-in. discharge line from the pump. The entire loop was insulated with 2 in. of 85% magnesia.

The loop was operated at water temperatures of 150°, 200°, 250°, 300°, and 325°F.

---

For complete tabular data, order document 4831 from A.D.I. Auxiliary Publications Photoduplication Service, Library of Congress, Washington 25, D. C., remitting \$1.75 for microfilm or \$2.50 for photoprints readable without optical aid.

For the lower temperatures the water was heated by introducing live low-pressure steam. To reach higher temperatures five electrical Calrod heaters with a combined capacity of 14 kw. were used. One 3-kw. unit, on automatic control, was used mainly in keeping the system hot overnight between tests (two-shift operation was used for most of the data taking). To maintain steady conditions at the lower temperatures, it was necessary to remove heat because the pump added more heat than was lost from the insulated system. The heat was removed by circulating a small by-pass stream through a Heliflow water-to-water heat exchanger. The heat exchanger had 2.76 sq. ft. of heat transfer area in the form of a helical coil of ½-in. tubing and used a maximum flow of 10 gal./min. of cooling water. Both the temperature and pressure in the loop were controlled manually, with satisfactory results.

Water was injected into the system intermittently to make up for losses through the pump packing glands. Leakage rates became quite high at the higher temperatures and pressures; the 7-gal. effective liquid capacity of the surge tank permitted operation without make-up for 2 hr. at the highest pressure used (140 lb./sq. in. gauge). Make-up water, partially preheated, was added by means of a pressurized storage tank made of a 4-ft. section of 10-in. pipe.

#### Instrumentation

The general instrumentation of the circulation system is indicated in Figure 2. Water flow was measured with an orifice meter equipped with standard horizontal flange taps, a 4.175-in. orifice in a 6-in. line (6.065 in. I.D.) being used. In order to cover the entire flow range with a single orifice, a mercury-filled manometer was used to measure the pressure differentials for the higher flow rates, and an inverted gas-filled manometer (using the water in the system as the manometer fluid) measured the lower rates. In this way the smallest differential measured was 8 in.

Static pressures in the system were measured with Bourdon gauges with a range of 0 to 160 lb./sq. in. gauge, graduated to 4 lb./sq. in. gauge, at the locations shown in Figure 2. The surge tank and make-up water tank were equipped with safety valves set at 150 lb./sq. in. gauge. The top of the surge tank was also equipped with a ball-float water-drain valve.

The following instrumentation was provided for the test lattice. Pressure drops were measured across the entire lattice (over a 4½-ft. length of the test section) and over the 3-ft. straight length of the dummy rods. For the lattice measurements pressure connections were made just below the bottom support plate and above the top one. In each case four taps in the 10-in. pipe spaced at 90° were connected to a piezometer ring. The internal measurements were made by means of suitably modified dummy rods, in which the pressure was likewise averaged around the circumference by means of four openings 90° apart. In addition to one of the standard ⅝-in. rods (*A* in Figure 3), the ⅞-in. rod used to simulate a control-rod thimble (*B* in Figure 3) was used for pressure-drop measurements).

The bulk water temperature was measured upstream of the lattice with both a thermometer and a thermocouple. The thermometer, which was used as an operating control and a check on the thermocouple, was located in an oil-filled steel well in the horizontal line approaching the lattice. It was a Fisher total-immersion thermometer with a scale covering 0° to 200°C. over a length of 18 in. and graduated to 0.2°C. The thermocouple was chromel-alumel and was inserted in an oil-filled copper well below the bottom support plate. The reference junction was placed in an ice bath. Both the thermometer and the thermocouple were calibrated in an oil bath against a set of four Anshutz precision thermometers covering the range from room temperature to 200°C. and in addition were calibrated at 0°C. in an ice bath and at 100°C. with condensing steam. When a standard immersion correction was applied to the thermometer reading, it generally checked the thermocouple reading within 0.5°F.

The temperature difference between the heater surface and the water was measured directly by placing a reference junction for each surface thermocouple in the same well that contained the chromel-alumel thermocouple for measuring water temperature. Since the test heater raised the temperature of the adjacent water by a small but appreciable amount, a correction was calculated on the basis that all the heat released upstream of the given thermocouple location was used to raise the average temperature of one rod's "share" of the total water flow. The correction reduced the measured temperature difference by 1 to 2%. The thermocouple voltages were read with a Leeds and Northrup type K-2 potentiometer and a Leeds and Northrup D.C. (No. 2430) galvanometer. A 12-point rotary switch was used to permit rapid readings. The surface thermocouples were calibrated in the water loop by comparison with the calibrated chromel-alumel thermocouple used to measure the bulk water temperature. By means of an ice-bath reference junction for all thermocouples, comparative steady state readings were made of the temperature of the circulating water, with no heat input to the test heater.

The test element was heated with d.c. current from a Westinghouse type R.A. arc welder, rated to supply 200 amp. at 40

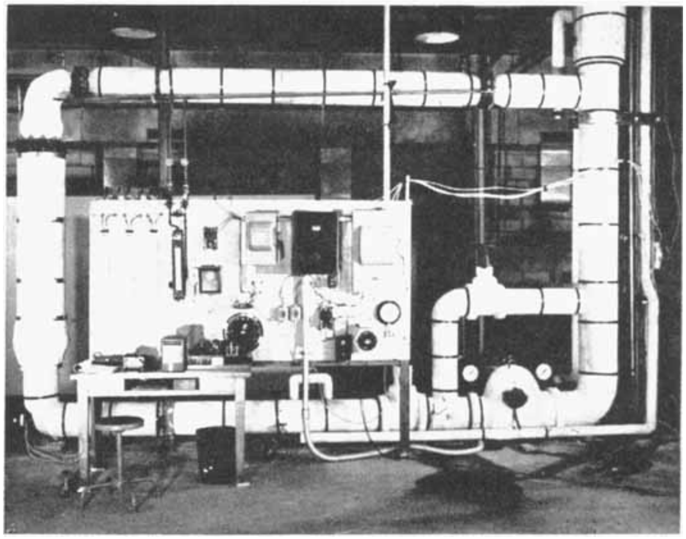

Fig. 1. Water-circulation loop for heat transfer measurements.

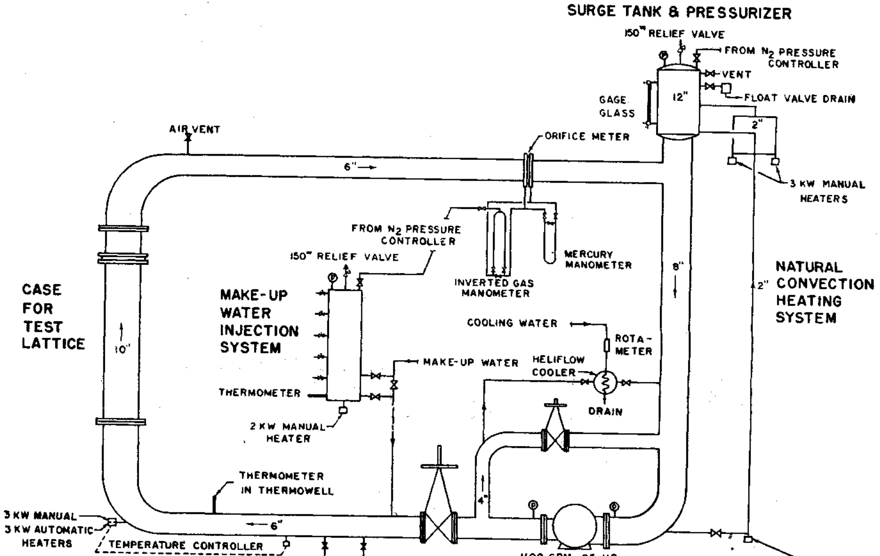


Fig. 2. Flow diagram including general instrumentation for circulation system.

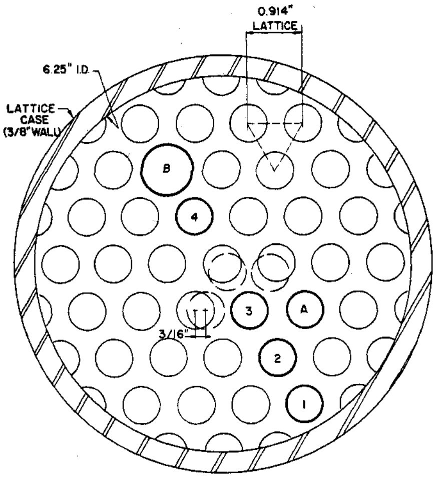


**Fig. 3. Cross section through lattice showing arrangement of rods.**

volts. The maximum power input obtainable with the heaters used (close to 1 ohm of resistance) was approximately 3.2 kw. corresponding to a current of 56.5 amp. and a voltage drop of 56.5 volts across the heater. The power input was measured with a Weston model 310 wattmeter, which had scales of 1,000, 2,000, and 4,000 watts. Independent calibrations of this meter by the vendor and by another laboratory showed the maximum error on any scale to be less than ¼ of 1% of full-scale value.

#### Test Lattice

The test lattice consisted of a vertical bundle of solid 61ST-6 aluminum rods supported between two perforated plates. The lattice was enclosed in a cylindrical steel case of 6¼-in. I.D., which was contained in a 10-in. pipe. The case was supported at the top by a flange which was sandwiched between two flanges connecting two sections of the 10-in. pipe. The annular space between the pipe and inner case was filled with water at the system pressure, but the top supporting flange prevented any flow through this space, forcing all the water through the lattice.

The rods were 4.0 ft. long (clear distance between support plates). They had an O.D. of 0.625 in. and were arranged in a triangular lattice with 0.914 in. between centers. Figure 3 shows a cross section through the lattice; it is seen that the 6¼-in. circular boundary encloses thirty-seven full-sized rods and intersects eighteen additional rods. Partial rods of the indicated cross section were attached to the cylinder wall in these eighteen positions. A single rod of 0.875-in.-diam., designated *B*, was installed to simulate a control-rod thimble. The rod positions numbered from 1 to 4 indicate the locations where heater rods were installed in various tests. The three displaced circles (dashed line) adjacent to position 3 indicate the locations to which these three dummy rods were moved in a special test of lattice distortion.

The dummy rods were tapered for 6 in. at each end to avoid blocking the openings in the support plates. The rods were fastened rigidly to the top plate by means of an extension pin, threaded for a standard nut. A lower extension pin made a slip fit in an opening in the lower plate, to allow for longitudinal thermal expansion or contraction. [It was found that a coating of liquid moly (molybdenum disulfide) prevented freezing of this pin.] The support plates were made of 61ST-6 aluminum; the top one was ¼ in. thick and the bottom one ⅛ in. The openings for water flow had a diameter of 0.485 in.; they were arranged in a hexagonal pattern which resulted in an average of two openings for each rod and a free-flow area of 44.3%, or 13.6 sq. in.

**Table 1. Dimensional Tolerances for Test-lattice Components**

| Component | Dimension | Magnitude, in. | Specified tolerance, in. | Actual tolerance, in. |
|---|---|---|---|---|
| Support plates | Rod support spacing | 0.914 | ±0.015 | ±0.003 |
| | Diameter | 7.998 | +0.000 −0.003 | +0.000 −0.003 |
| | Concentricity of center rod with 7.998 O.D. | | 0.005 | 0.005 |
| Dummy rods | Diameter of support | 0.202 | +0.000 −0.005 | +0.000 −0.002 |
| | Concentricity of support pins and rod | | 0.010 | 0.005 |
| | Rod straightness | | 0.010 | 0.005 |
| | Rod diameter | 0.625 | ±0.005 | ±0.002 |
| Steel cylinder | Inside diameter | 6¼ | ±1/64 | ±0.005 |

Dimensional tolerances for the test lattice were established on the basis that they should contribute to the probable error of the measurements equally with other sources, such as flow measurement, temperature measurement, etc. (See discussion on accuracy of results, page 231.) Table 1 lists the resulting specified tolerances. It is of interest that the error analysis resulted in a considerable relaxation from more rigid tolerances originally felt to be necessary, which led to a considerable reduction in time and cost of machining. The tolerances actually achieved are also listed in Table 1; they were considerably better than the specified values.

#### Description of Heater Rod

Three heated test rods, all basically similar in construction, were used in obtaining the data reported in this paper. The construction of rod 15, an aluminum-jacketed unit that was used in most of the work, was described completely in a recent paper (9). Figure 4 shows some of the construction details of this heater rod, which had a 4-in. heated section located midway in its length.

Heater rod 18, which was used in some cases when rod 15 was under repair and in some cases in the same tests as rod 15, differed from it only in the following respects. The heater jacket was nickel instead of aluminum. The thermocouples were chromel-alumel instead of copper-nickel; the thermocouple wires were 5 mils in diameter and were insulated with resin-impregnated Fiberglas. Because of the larger O.D. of this insulation, the grooves for the two external thermocouples with which the heater was provided had to be enlarged to 12 mils wide by 12 mils deep. The junctions for these thermocouples were soldered, and the two wires for each thermocouple were only 10 degrees apart on the circumference. In addition, rod 18 was provided with two "internal" thermocouples, which were used to test a method of thermocouple installation that involved minimum disturbance of the heater surface. A 6-in. groove 25 mils wide by 20 mils deep was made along the inside of the nickel jacket. Two holes, each 31 mils in diameter, were drilled out to the surface, each being ½ in. from the center of the groove on either side. Two chromel-alumel thermocouples were brought in along this groove, one from either end, and the junctions were bent into the holes and brought out to within about 5 mils of the surface. The hole was then filled in with hot solder and smoothed over. The leads for these two thermocouples were brought out through the hollow end pieces of the rod, along with the power and voltage leads.

The two internal thermocouples were located respectively ½ in. upstream (No. 1) and ½ in. downstream of the middle. (See Figure 5.) One external thermocouple (No. 3) was located ½ in. downstream, 180 degrees away from No. 2, and the other (No. 4) was located ½ in. upstream, 90 degrees from No. 1.

Heater rod 20, which was constructed late in the investigation, had an 8-in. heated length and a copper jacket. It was provided with eight thermocouples spaced axially along the rod from ⅝ to 7½ in. past the upstream end and 45 degrees apart circumferentially. Each thermocouple required only a single groove, for a nickel wire, as the copper jacket completed the thermocouple circuit. (It was found by test that the thermoelectric effect between the copper tube and copper wire was negligible.)

#### Surface Temperature Measurement

Measurement of the surface temperature presented serious difficulties. The thermocouple had to be mounted in a manner that would avoid local flow disturbance and at the same time would place the junction at the surface. The use of a plated thermocouple, as described by Bonilla (*6*), was considered but had to be discarded because it would require either an insulating layer between the thermocouple and the rod jacket or the use of a nonmetallic rod, and neither was practical. Both calculation and experiment showed that a thermocouple plated directly on the metal jacket would read the temperature only at the point where the insulated lead wire made a junction with the plate, and so the plate served no purpose.

The method used, that of embedding the thermocouple wires in shallow downstream grooves which were then filled and smoothed, avoided interference with water flow. However, it presented several possibilities for error that had to be investigated. First, because the thermocouple junction was below the surface and there was a temperature gradient in the jacket, the thermocouple might read high. Second, because the thermal conductivities of the thermocouple itself, or the cold solder used to wedge it in, and of the intermetallic boundaries were different from those of the jacket material, a disturbance might result in the thermal flux pattern which could make the thermocouple read either high or low. Both of these effects would vary with the thermal conductivity of the jacket material, and for this reason heated elements were initially constructed with both aluminum and nickel jackets [$k$ = 119 and 34 B.t.u./(hr.)(ft.)(°F.), respectively], and later with a copper jacket ($k$ = 218).

rods were incorporated in an annulus heat transfer experiment.

#### Heat Transfer Measurements in a Tube

The tubular tests were made in a vertical copper tube 0.548 in. I.D. Water flowed upward. An 8-in. length was heated by an external winding of asbestos-insulated chromel-A wire, covered with 1 in. of 85% magnesia. The inner wall temperature was measured by two thermocouples located 90 degrees apart, each consisting of a 3-mil nickel wire inserted in a groove 8 mils wide by 8 mils deep, with the copper tube completing the circuit. The soldered junctions were located 6 in. from the upstream end of the heater. Two external thermocouples were similarly installed in grooves on the outside of the tube, 90 degrees apart from each other and each 90 degrees apart from one of the interval thermocouples. A layer of aluminum foil was wrapped around the tube, covering the external thermocouples, before the heating coil was applied, to minimize the possibility that these thermocouples could reach a temperature higher than that of the outer copper surface.

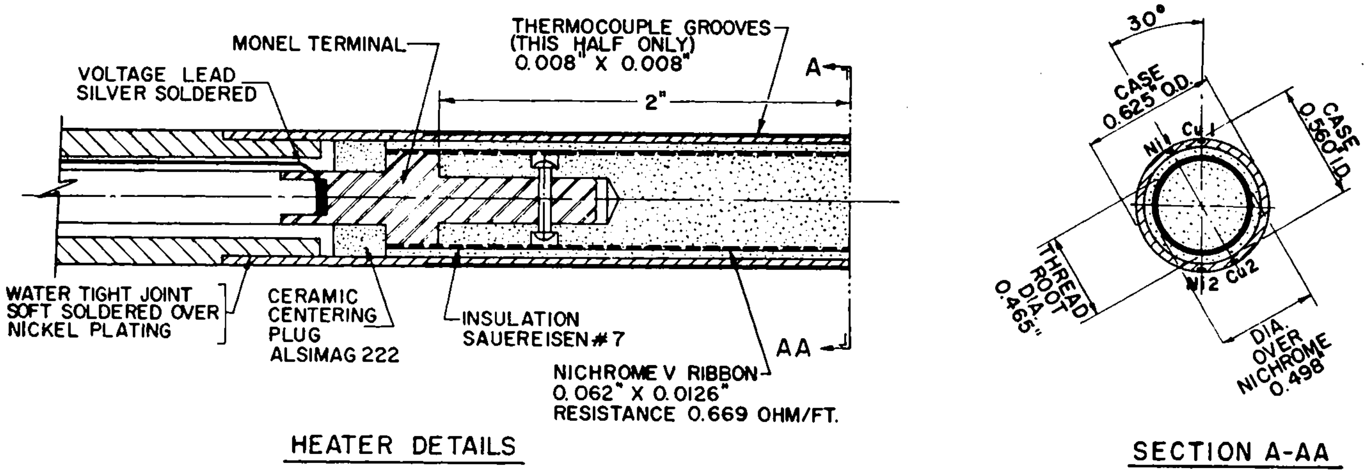


Fig. 4. Details of construction of heater in rod 15.

adequate for full development of the thermal boundary layer. The total straight upstream length was 29 in. (53 diameters), sufficient for a fully developed hydraulic boundary layer.

The two external thermocouples differed by 8% (based on $\Delta T$), and their mean was consistently higher than the mean of the internal thermocouples. The difference varied from 0.1° to 1.9°F. with a mean of 0.7°F. This compares with a calculated temperature drop of 0.8°F. across the copper tube wall.

The reliability of the measurements was confirmed by a heat balance, which indicated that the water picked up 95 ± 5% of the total electrical heat input. This was corroborated by an external loss of 3 to 5% from the heating coil, as indicated by both measurement and calculation. A value of 95% of the measured power input was used in calculating experimental heat transfer coefficients.

Calculations were made for nickel and aluminum to evaluate these two sources of error. Rather drastic assumptions had to be made with regard to thermal resistances at boundaries, etc., and these limit the applicability of the calculated results. They supported the conclusion, however, that higher temperatures (lower values of $h$) should be measured for the nickel rod than for the aluminum rod under comparable conditions. This agrees with the experimental results.

In view of the difficulty of establishing directly the accuracy of the surface-temperature measurement, it was desirable to check the over-all accuracy of the heat transfer measurements by comparison with previous work. There are no pertinent data available for an open lattice. Two other methods of checking the measurements were resorted to. In one, heat transfer coefficients were determined for water flowing inside a tube, the same technique of thermocouple installation being used here as for the heater rods. In the second, the actual

Measurements were made at water temperatures of 73° and 140°F., velocities of 3½ and 7 ft./sec., and a single flux of 34,000 B.t.u./(hr.)(sq. ft.). The water-film temperature drops measured by the two internal thermocouples differed consistently by about 7%. Values of the heat transfer coefficient based on an average of the two were about 20% higher than values predicted by the Colburn equation (*10*):

$$\frac{hD}{k} = 0.023\left(\frac{DG}{\mu_f}\right)^{0.8}\left(\frac{C_p\mu_f}{k}\right)^{1/3} \qquad (1)$$

When the heated upstream length was decreased from 6 to 2 in., the values of $h$ increased by 4%; this small change indicated that a 6-in. heated length was

The agreement of the external and internal thermocouples was direct evidence that the method of installation did not introduce any systematic error. It demonstrated even more strongly that the internal thermocouples were not reading erroneously low (leading to high values of $h$), as it is difficult to conceive of factors that would make the external thermocouples read too low. The accuracy of the temperature measurements was corroborated by the agreement of the measured heat transfer coefficients, to 20%, with the Colburn equation.

#### Annulus Heat Transfer Measurements

The annulus measurements have been reported separately (*9*). The results indicated that coefficients measured with rod 15 (aluminum jacketed) and rod 20 (copper jacketed) were about 20% higher than predicted by the Colburn equation, by use of the equivalent diameter $D_e = D_2 - D_1$. Coefficients measured with

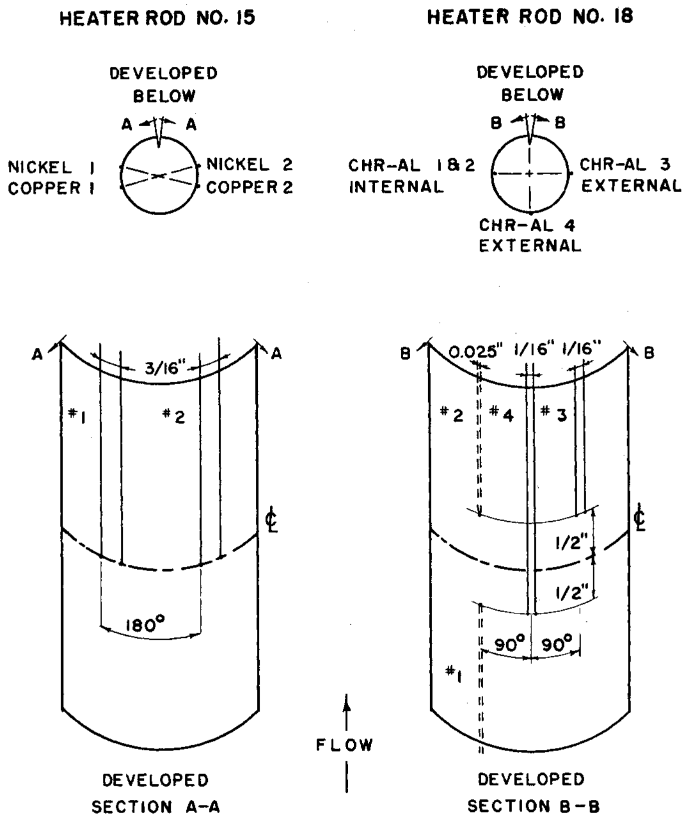


Fig. 5. Developed views of heater surfaces showing thermocouple locations.

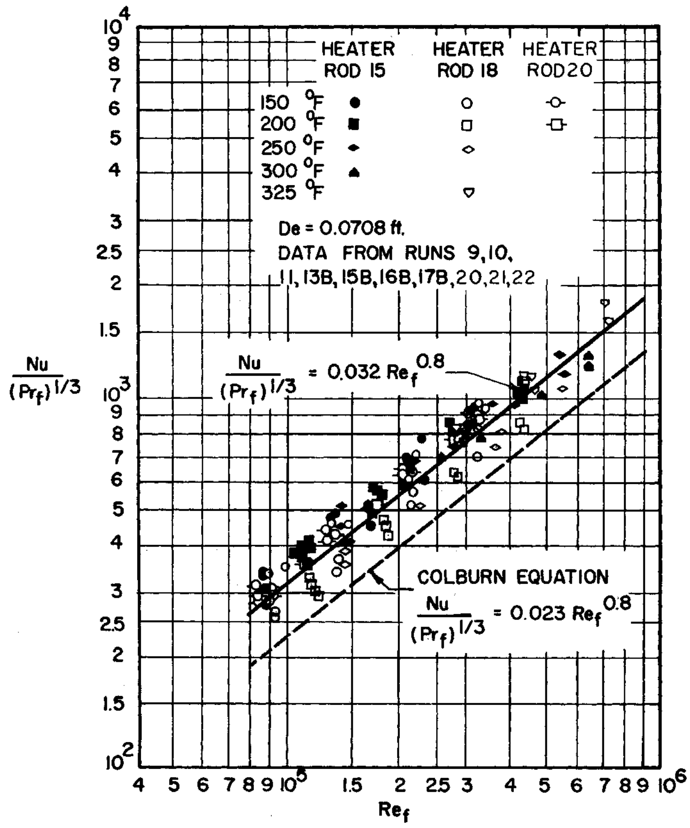


Fig. 6. Heat transfer data for water flowing parallel to cylindrical rods.

nickel-jacketed rod 18 (not reported elsewhere), based on thermocouples 3 and 4, were about 10% higher than the Colburn values, which is in accord with the prediction that thermocouples embedded in nickel should indicate values higher than the true surface temperature. From these results, and the considerations mentioned above, it is concluded that a true measure of the surface temperature was obtained with rod 15, and that the temperatures measured with rod 18 indicated water-to-jacket temperature differences that were 10% too high.

However, rather than to make the indicated adjustment of 10% in the rod 18 measurements, it has been deemed preferable to report the uncorrected results. While this increases the scatter of the data taken as a whole, the scatter s still not excessive for measurements of this type. (*See* Figure 6.)

#### Behavior in Lattice Tests

Heating element 15 (aluminum) was equipped with two copper-nickel thermocouples designated No. 1 and No. 2, located 180 degrees apart at the mid-length of the heater. (*See* Figures 4 and 5.) The film temperature drop measured with thermocouple 1 was consistently higher than that measured with No. 2 by about 10 to 20%. The difference was virtually unaffected by rotating the heating element 30 or 180 degrees (runs 9, 10, and 11), which indicated that it was not caused by local differences in flow conditions in a triangular lattice. It was likewise unchanged by at least three successive repairs to the thermocouples, in which the wire tips were removed from their grooves, the grooves cleaned, and the junctions reconstituted. This behavior indicated that the thermocouple readings were not sensitive to minute differences in the method of installation.

After considering and eliminating bowing of the heated element as the cause of the difference in reading between No. 1 and No. 2, the authors concluded that it was probably due to nonsymmetrical heat flux around the periphery. The layer of Sauereisen insulation separating the heating coil from the aluminum jacket was 31 mils thick, and because of its relatively poor conductivity there was an estimated temperature drop of as much as 1,500°F. across it. For this reason, even small differences in the thickness of the layer or in its thermal conductivity around the circumference could result in a nonuniform flux distribution. Measurement of the thermal conductivity of several samples of Sauereisen in a Cenco-Fitch apparatus (*2*) gave values varying from 0.27 to 0.48 B.t.u./(hr.)(ft.)(°F.); although the absolute accuracy of these measurements is questionable (owing to boundary effects), this degree of variation would readily account for the observed differences in thermocouple readings.

The procedure was followed of averaging the readings to No. 1 and No. 2 whenever an average coefficient was being measured with rod 15. In some cases rod 15 was used in locations where the coefficient was expected to vary around the circumference, i.e., next to the wall (run 12) or next to a simulated control rod (run 13*A*). In such cases the values measured with the individual thermocouples were compared directly with the readings of the same thermocouple when the rod was tested in a central, or "normal," location (run 10, lattice position 3).

For rod 18 (nickel jacketed), thermocouples 3 and 4 agreed quite closely (within 3%), and their average value was 10 to 15% above the values given by No. 2. Number 1 failed at an early stage and was not repaired. Number 2 failed after it had been used in several runs; however, it was the only thermocouple functioning on rod 18 in run 16*B*, because of failure of the external leads of No. 3 and No. 4.

#### ACCURACY OF RESULTS

From the degree of scatter in Figure 6, the reproducibility of measurements in different runs, and the tubular- and annular-flow heat transfer measurements, it is estimated that the over-all accuracy of the results was $\pm 8\%$. This uncertainty was greater than the maximum probable error, which was estimated as $\pm 4.1\%$ from procedures described by Sherwood and Reed (*11*). $Q$ was measured to $\pm 0.5\%$ or better, and $A$ was known with at least this accuracy; calculations confirmed that axial heat flow was negligible in the vicinity of the thermocouples. The orifice measurement was accurate to $\pm 2\%$, and this combined with the uncertainty in the free-flow area gave $V$ a probable error of $\pm 2.8\%$. However, a further uncertainty is introduced by the possible difference between the average velocity, $V$, and the local velocity at the heated rod. The added surface (or reduced equivalent diameter) at the wall tends to give a lower velocity near the wall, making the velocity in the inner zone higher than the average. A calculation of this effect, based on the Fanning equation and the assumption of two zones of constant velocity, indicated that the velocity in the inner zone might be 4.5% higher than the average velocity.

#### HEAT TRANSFER MEASUREMENTS IN NORMAL LATTICE

Table 2 lists the variables studied and their range of variation; the rod diameter and normal lattice spacing were not varied.

TABLE 2. RANGE OF TEST VARIABLES

| | |
|---|---|
| Water temperature, °F. | 150–325 |
| Water velocity, ft./sec. | 5–20 |
| Lattice positions | 4 |
| Reynolds number | 70,000–700,000 |
| Power density, kw./ft. | 2.5–10 |
| Heat flux, B.t.u./(sq. ft.)(hr.) | 52,000–208,000 |

TABLE 3. TYPICAL HEAT TRANSFER DATA

Run 9†† Purpose-normal lattice — Test element 15, Lattice position 3

| Bulk water temp.*, °F. | Average water velocity, ft./sec. | Heat flux, B.t.u./(hr.)(sq. ft.) | $Re_f$ $\times 10^{-3}$ | Corrected film temp. drop, °F. T.C.1 | T.C.2 | $h$,† B.t.u./(hr.)(sq. ft.)(°F.) |
|---|---|---|---|---|---|---|
| 200.3 | 4.95 | 104,300 | 118.0 | 46.6 | 43.3 | 2,320 |
| 200.2 | 9.94 | 105,200 | 211.8 | 27.8 | 25.0 | 3,990 |
| 200.4 | 14.91 | 105,200 | 299.8 | 20.7 | 18.8 | 5,340 |
| 199.4 | 19.90 | 104,500 | 434.3 | 16.8 | 15.0 | 6,600 |
| 199.2 | 4.97 | 104,600 | 117.2 | 44.5 | 40.7 | 2,460 |
| 199.5 | 4.97 | 52,100 | 110.7 | 21.3 | 19.4 | 2,570 |
| 199.4 | 9.97 | 52,100 | 215.3 | 12.8 | 11.3 | 4,350 |
| 199.9 | 14.93 | 52,000 | 319.2 | 9.5 | 8.5 | 5,790 |
| 201.0 | 19.91 | 52,000 | 426.6 | 7.9 | 7.0 | 7,020 |
| 202.2 | 4.80 | 52,000 | 107.8 | 21.2 | 19.3 | 2,570 |

*Upstream of heater, uncorrected for heat input.
†Based on calculation using average corrected film temperature drop.
††Data for 150°, 250°, and 295°F. are omitted to conserve space.

TABLE 4. EFFECT OF HEAT FLUX

| Run | Heater rod | Bulk water temp., °F. | Ratio of power densities | Ratio of heat transfer coefficients (avg.) | Values No. | Range |
|---|---|---|---|---|---|---|
| 9 | 15 | 200 | 1650/825 | 0.939 | 5 | 0.910–.980 |
| 10 | 15 | 200 | 1650/825 | 0.964 | 5 | 0.930–.981 |
| 17B | 18 | 325 | 1650/825 | 0.929 | 3 | 0.918–.937 |
| 16B | 18 | 150 | 3200/1650 | 1.085 | 5 | 0.937–1.170 |
| 16A | 15 | 150 | 3200/1650 | 1.063 | 3 | 1.035–1.078 |

A total of twenty-two runs was made, of which runs 1 through 8 were preliminary, being devoted to testing and improving the heated test elements and the method of surface-temperature measurement. Heater rod 15 was tested in positions 1, 3, and 4 (*see* Figure 3), heater rod 18 was tested in positions 2 and 3, and heater rod 20 was tested in position 3. Table 3 gives detailed results for a typical run.

Figure 6 presents a plot of all the heat transfer data that were obtained for normal lattice conditions.* These comprise the measurements for lattice positions 2 and 3, with rods 15, 18, and 20. For rod 15 the results for thermocouples 1 and 2 were averaged, and for rod 18 the values are either an average for "external" thermocouples 3 and 4 or the measurement with "internal" thermocouple 2, whichever was functioning. For rod 20, thermocouples 2 to 7 were averaged.

The slope of the best straight line through the data is very close to 0.8, in agreement with heat transfer behavior in pipes. With the slope fixed at 0.8, least squares analysis (using vertical deviations) gave the following equation:

$$\frac{hD_e}{k} \Big/ \left(\frac{C_p\mu_f}{k}\right)^{1/3} = 0.032\left(\frac{D_eG}{\mu_f}\right)^{0.8} \quad (2)$$

The data for copper rod 20 (tagged open symbols) fall slightly higher than those for aluminum rod 15 (solid symbols), which are higher than those for nickel rod 18 (open symbols). The equations for the three sets of data taken separately would have constants of 0.034, 0.033, and 0.031, respectively (with the slopes kept arbitrarily at 0.8).

##### Effect of Heat Flux

Most of the measurements were made at a power density of 5 kw./ft. [heat flux of 104,000 B.t.u./(hr.)(sq. ft.)], but comparative measurements were made also at power densities of 2.5 and 10 kw./ft. The available results are compared in Table 4. Heat transfer coefficients at 5 kw./ft. were 6 or 7% lower, on the average, than values measured either at the higher or lower power density. No explanation of this behavior has been suggested. Allowance for increased film temperature would indicate a small increase in coefficient with power density, less than that observed.

##### Heater Entrance Effects

The effect of heated length on apparent heat transfer coefficient was studied in some detail with rod 20, which had an 8-in. heated section. Representative results are shown in Figure 7. To facilitate comparison a relative value of the coefficient $h_r$ has been plotted; $h_r$ is the ratio

*The physical property data were taken from Eckert (*5*).

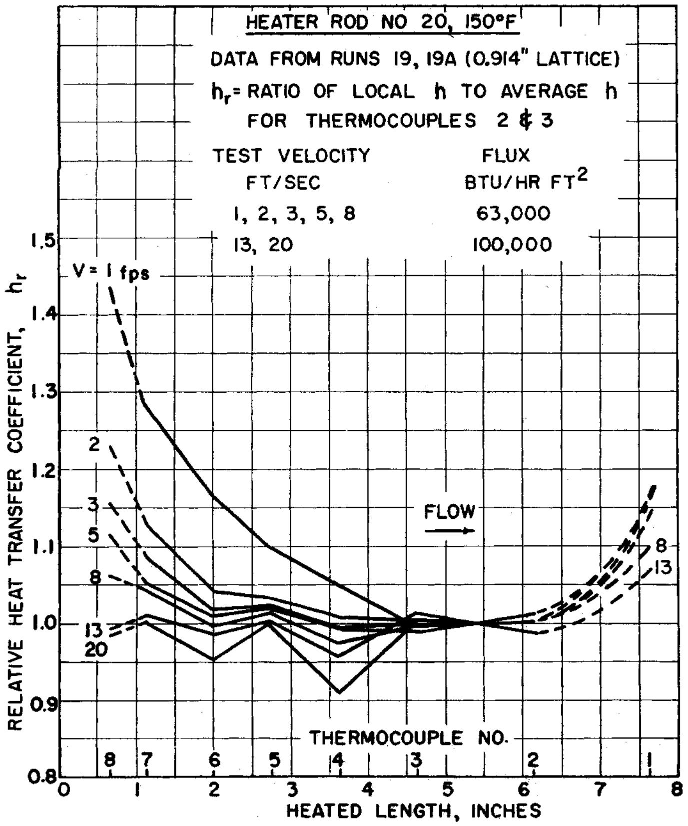


Fig. 7. Effect of heated length on heat transfer coefficient.

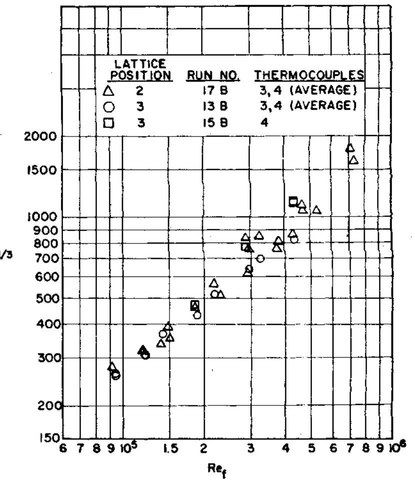


Fig. 8. Effect of lattice position on heat transfer (rod 18).

of $h$ as measured by a particular thermocouple to the $h$ calculated from the average readings of thermocouples 2 and 3 at the given velocity. Thermocouples 2 and 3 were chosen as the reference because they were the furthest downstream and agreed with each other at all velocities. (Number 1 was actually the furthest downstream but was evidently close enough, ⅜ in., to the end to have its reading lowered by axial conduction.) For these tests the velocity range was extended down to 1 ft./sec.

An effect of heated length is clearly evident at the lower velocities, which increases with decreasing velocity. However, for thermocouple 7 (1⅛-in. station) it was less than 5% maximum for the velocities of 5 ft./sec. and higher at which the general heat transfer study was made. At the 2-in. station, corresponding to the thermocouple location in the 4-in. heater rods, no significant effect of heated length was evident at velocities above 3 ft./sec. It will be observed that the readings of thermocouple 8 (⅝-in. station) may or may not have been influenced by axial cooling; calculations placed it on the border line of such effects.

Measurements at two different heat fluxes, 63,000 and 100,000 B.t.u./(sq. ft.)(hr.) showed close agreement.

The results indicate that a 2-in. heated length ($L/D_e = 2.4$) was sufficient for full development of the thermal boundary layer at velocities of 5 to 20 ft./sec. The systematic indication of an increasing effect of heated length with decreasing velocity below 5 ft./sec. is evidence of the adequacy of the method to detect any effects that do exist.

The results also confirm that the hydraulic boundary layer ($L/D_e = 21$) was fully established.

Additional evidence on the question of heated length was obtained with heater rod 18, on which thermocouple 3 was located 1 in. further downstream than thermocouple 4. (*See* Figure 5.) Data for a comparison are available from runs 13*B* and 17*B*. In run 13*B* the average value of $h_4/h_3$ was 1.045 ± 0.039 (based on twelve values), and in run 17*B* it was 1.028 ± 0.031 (based on twenty-one values). The difference between $h_3$ and $h_4$ is within the experimental accuracy.

#### Simultaneous Heating of Two Rods

In run 17 advantage was taken of the opportunity to test two heated rods simultaneously, with rod 15 in lattice position 3 and with its No. 2 thermocouple directly adjacent to rod 18 in lattice position 2. A comparison of heat transfer coefficients measured with this thermocouple, with and without simultaneous heating of the adjacent rod, showed no effect.

### EFFECT OF LATTICE VARIABLES ON HEAT TRANSFER

Tests were made to obtain information on the effect upon the heat transfer coefficient of lattice position, lattice distortion, and thermocouple orientation in the unit lattice cell. Rod 15 was used in all these tests. As any of these variables might result in a peripheral variation in heat transfer coefficient, the average value for thermocouples 1 and 2 was not of much use in evaluating the results. Instead, the value measured with each thermocouple was compared directly with the readings of the sample thermocouple under the same conditions in run 10 (lattice position 3). The deviations of the bulk water temperatures, power densities, and flow rates for the various runs from the nominal values were small enough to make such a comparison valid. As the values of $h$ obtained in run 10 were somewhat higher than the average results, this basis of comparison exaggerates slightly the effect of lattice variables.

#### Effect of ⅞-in. Rod

Run 13*A* was made with rod 15 in lattice position 4, adjacent to the ⅞-in.-diam. dummy rod which simulated a typical control thimble. Heat transfer was measured at 150° and 200°F. at a heat flux of 104,000 B.t.u./(hr.)(sq. ft.). Thermocouple 2 was located directly adjacent to the ⅞-in. rod.* The average value of the ratio of $h_2$ (run 13*A*) to $h_2$ (run 10) was 0.916 ± 0.028†, and for $h_1$ the average ratio was 0.956 ± 0.030. The comparison indicates a reduction of about 8% in the coefficient directly adjacent to the ⅞-in. rod and a reduction of about 4% on the opposite side.

#### Wall Effect

Run 12 was made with rod 15 in lattice position 1, next to the wall, with thermo-

*Actually the nickel wire of thermocouple 2 had this location, but the Ni–Al junction contributes about 80% of the total Ni–Cu voltage.

†The value ±0.028 is the probable error = 0.6745 × standard deviation.

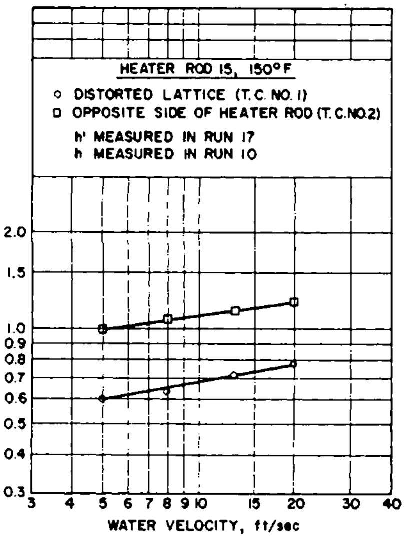


Fig. 9. Effect of lattice distortion on heat transfer.

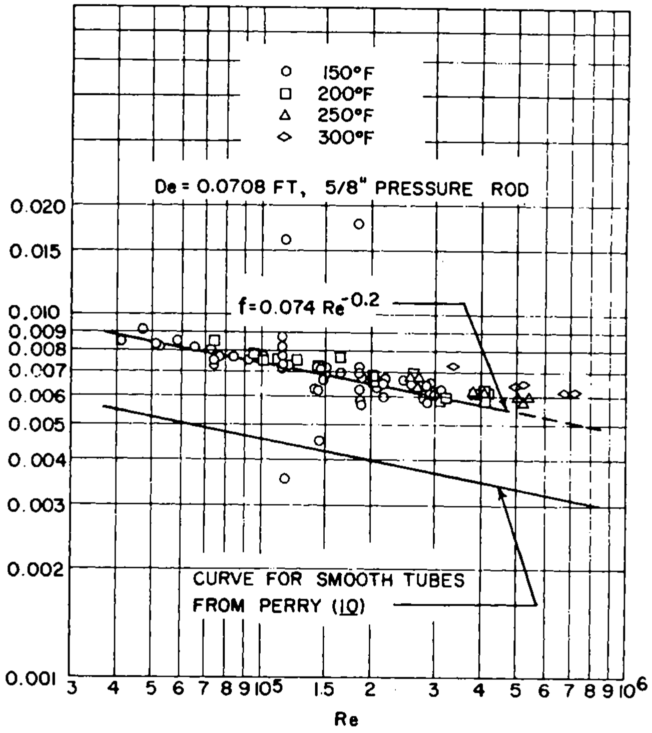


Fig. 10. Friction-factor data for open lattices.

couple 1 adjacent to the wall. Measurements were made at 150°, 200°, and 250°F. and a heat flux of 104,000 B.t.u./(hr.)(sq. ft.). The average value of the ratio of $h_1$ (run 12) to $h_1$ (run 10) was 0.616 $\pm$ 0.022, and for $h_2$ the average value of the ratio was 0.876 $\pm$ 0.020. Thus the coefficient was reduced about 40% on the side nearest the wall and about 12% on the opposite side.

These figures indicate that the wall effect does not extend very far into the lattice. This conclusion is confirmed by a comparison of the results obtained in lattice positions 2 and 3 (obtained with rod 18; rod 15 was not tested in position 2). Tests were made in position 3 in runs 13*B* and 15*B* (with thermocouple 3 facing the wall) and in position 2 in run 17*B* (with thermocouple 2 facing the wall). As a point-to-point comparison of these tests would leave a majority of the data out of consideration, it is more convenient to make the comparison in a plot, and this is done in Figure 8. The data showed considerable scatter, with the points for lattice position 2 bracketed by values obtained for position 3 in runs 13*B* and 15*B* respectively. The evidence, within its limits of accuracy, indicates that the wall effect did not extend to lattice position 2 (and on this basis the data for position 2 were included in Figure 6).

#### Lattice Distortion

The effect of severe fuel-rod distortion was simulated by reducing the minimum clearance between heated rod 15 and three of its six neighbors by 3/16 in., from 0.289 to 0.101 in. (*See* Figure 3.) Three sets of measurements were made at 150°F. and a heat flux of 104,000 B.t.u./(hr.)(sq. ft.), with thermocouple 1 facing the crowded lattice (run 17*A*). The results are compared with those of run 10 in Figure 9. The ratio of the heat transfer coefficient in the crowded lattice to that in the normal lattice varied from 0.596 to 0.773. When this ratio is plotted against the velocity on log-log paper, the points are fitted quite well by a straight line.

Also plotted in Figure 9 are values of the ratio of heat transfer coefficients measured with thermocouple 2 (180 degrees from No. 1) in runs 17*A* and 10 respectively. The values ranged from 0.992 to 1.213 and also fell on a straight line.

#### Local Coefficient

Between runs 10 and 11 heater rod 15 was rotated 30 degrees, with no other change in the lattice. This change moved thermocouple 1 from a position on a line of centers to a point of maximum removal from the neighboring rods and did the reverse for thermocouple 2, thus providing a test of the maximum potential variation in local coefficient around the periphery. The ratio of heat transfer coefficients measured with the two thermocouples $h_2/h_1$ was compared for the conditions common to both runs: 150°, 200°, and 250°F. at a flux of 104,000 B.t.u./(hr.)(sq. ft.).

The average value of the ratio was 1.113 $\pm$ 0.018 in run 10 and 1.141 $\pm$ 0.045 in run 11. The difference is less than the experimental uncertainty and is considered to indicate the absence of peripheral variation.

Average values of $h$ were 97% as high in run 11 as in run 10.

### PRESSURE-DROP MEASUREMENTS

The frictional pressure drops within the lattice were measured over a 3-ft. length of the ⅝-in.-diam. pressure rod. Pressure drops were not measured in all runs, but values were obtained covering the entire range of operating conditions. Measurements made with the ⅞-in. pressure rod agreed with those made with the ⅝-in. rod, apart from some random variation.

The observed pressure drops were used to calculate values of the friction factor $f$ from the relation

$$f = \frac{FgD_e}{2LV^2} \tag{3}$$

obtained from the Fanning equation. The results are plotted against the Reynolds number in Figure 10. On the whole, the data are fitted quite well on a log-log plot by a straight line with a slope of $-0.2$, having the equation

$$f = 0.074\, Re^{-0.2} \tag{4}$$

This slope agrees with predicted behavior for turbulent flow in pipes.

The experimental values of $f$ fall considerably above the curve for turbulent flow in smooth tubes, taken from Perry (*10*). On the average, the measured values are about 65% higher than the curve. It is believed that the rods were initially comparable to smooth tubes in surface finish and remained so as evidenced by physical appearance and lack of a trend in heat transfer and pressure-drop data with time.

The measured frictional pressure drops varied from about 0.076 lb./(sq. in.)/(ft.) at 5 ft./sec. to about 0.91 lb./(sq. in.) (ft.) at 20 ft./sec. at 150°F. The pressure drop over the entire lattice, including both support plates, ranged from 0.55 to 8.04 lb./sq. in.

### RELATION OF HEAT TRANSFER TO PRESSURE DROP

Colburn (3) showed that the heat transfer factor

$$j = \frac{h}{C_pG}\left(\frac{C_p\mu_f}{k}\right)^{2/3}$$

was equal to $f/2$ for turbulent flow in pipes and parallel to plane surfaces but was less than $f/2$ for flow across tube banks. A recent report (1) shows that for heating and cooling of water in rectangular ducts, the ratio of $j$ to $f/2$ varied from about 0.8 for square ducts to 0.64 for a duct with an aspect ratio of 7.9. For the present case, by combining Equation (2)

$$Nu = 0.032\, Re_f^{0.8}\, Pr_f^{1/3} \qquad (2)$$

and Equation (4)

$$f = 0.074\, Re^{-0.2} \qquad (4)$$

one obtains

$$\frac{j}{f/2} = 0.86$$

This procedure is not strictly correct, as the pressure-drop measurements were made for *unheated* rods; however, it is estimated that the correction for this discrepancy would be less than 2% for most of the range studied. It would be less than 6% for a heat flux of 208,000 B.t.u./(hr.)(sq. ft.), a temperature of 150°F., and a velocity of 5 ft./sec., which represents the worst case.

The fact that the ratio is less than 1.0 indicates that the pressure-drop data support the correctness of the relatively high values found for the heat transfer coefficients.

### SUMMARY

Heat transfer coefficients were determined for parallel flow of water through a bundle of cylindrical rods at water temperatures of 150° to 325°F., velocities of 5 to 20 ft./sec., and heat fluxes of 50,000 to 200,000 B.t.u./(hr.)(sq. ft.). No previous data for this case were found in the literature. For this range of conditions the apparent coefficient remained unchanged for heated lengths greater than 1½ in. ($L/D_e$ = 1.6). The experimental values were 40% higher than would be predicted by use of Colburn's equation for flow inside tubes, with the equivalent diameter $D_e$ replacing the tube diameter, as proposed by Eckert. The frictional pressure losses were likewise higher, by 65%, than would be predicted by substituting $D_e$ in the Fanning equation. These two results are mutually consistent in the light of the modified Reynolds analogy.

The heat transfer data are fitted by the equation

$$\frac{hD_e}{k} = 0.032\left(\frac{D_eG}{\mu_f}\right)^{0.8}\left(\frac{C_p\mu_f}{k}\right)^{1/3} \qquad (2)$$

which is analogous to the Colburn equation. However, there is no assurance that this relation will prove correct for lattices of dimensions different from those of the single lattice studied in the present work. Heat transfer measurements are needed in various lattices before a general relation can be established.

No variation in local coefficient around the rod periphery was found. The coefficient was reduced by 40% directly adjacent to the wall of the lattice case, but there was practically no reduction at the second rod from the wall. The effect of local distortion of the lattice was measured.

### ACKNOWLEDGMENT

Most of the results reported in this paper were obtained during work performed by the Walter Kidde Nuclear Laboratories, Inc., under Contract AT(30-1)-1374 for the United States Atomic Energy Commission. The permission of the A.E.C. to publish this paper is hereby gratefully acknowledged.

The authors are indebted to K. P. Cohen and W. I. Thompson for advice and encouragement; to J. J. Barker, R. F. Benenati, A. Weiss, and H. Yanowitz for assistance in the design and operation of the equipment; and to G. G. Bartolomei for construction of the heated test rods.

### NOTATION

$A$ = heat transfer area, sq. ft.
$C_p$ = Specific heat of water at bulk temperature, B.t.u./(lb.)(°F.)
$D$ or $D_1$ = outside diameter of rod, ft.
$D_2$ = for annulus, jacket diameter, ft.
$D_e$ = Equivalent diameter, ft. =

$$\frac{4 \times \text{flow area}}{\text{wetted perimeter}}$$

$f$ = friction factor, defined by Equation (4)
$F$ = pressure loss, ft. of fluid flowing
$g$ = gravitational constant, ft./(sec.)$^2$
$G$ = mass velocity of water, avg., lb./(hr.)(sq. ft.)
$h$ = heat transfer coefficient, B.t.u./(hr.)(sq. ft.) ($h$ is used with various subscripts through report, as defined in text in each case)
$j$ = heat transfer factor =

$$\frac{h}{C_pG}(Pr_f)^{2/3}$$

$k$ = thermal conductivity of water at bulk temperature, B.t.u./(hr.)(ft.)(°F.)
$L$ = length of flow passage, ft.
$Nu$ = Nusselt number,

$$\frac{hD}{k} \text{ or } \frac{hD_e}{k}$$

$P$ = frictional pressure drop, lb./(sq. in.) (ft.)
$Pr$ = Prandtl number,

$$\frac{C_p\mu}{k}$$

$Pr_f$ = Prandtl number,

$$\frac{C_p\mu_f}{k}$$

$Q$ = heat-flow rate, B.t.u./hr.
$Re$ = Reynolds number,

$$\frac{DV\rho}{\mu} \text{ or } \frac{D_eV\rho}{\mu}$$

$Re_f$ = Reynolds number,

$$\frac{DV\rho}{\mu_f} \text{ or } \frac{D_eV\rho}{\mu_f}$$

$T$ = temperature, °F.
$\Delta T$ = temperature difference from rod surface to bulk water, °F.
$V$ = water velocity, avg., ft./sec. or ft./hr.
$\mu$ = viscosity of water at bulk temperature, lb./(ft.)(hr.)
$\mu_f$ = viscosity of water at average film temperature, lb./(ft.)(hr.)
$\rho$ = density of water at bulk temperature, lb./cu. ft.

### LITERATURE CITED

1. Armour Research Foundation, *Rept. NEPA*-1559, unclassified (Aug. 3, 1950).
2. Central Scientific Co., "Selective Experiments in Physics: Thermal Conductivity," No. 71990-H52b (1940).
3. Colburn, A. P., *Trans. Am. Inst. Chem. Engrs.*, 29, 174 (1933).
4. Donohue, D. A., *Ind. Eng. Chem.*, **41**, 2499 (1949).
5. Eckert, E. R. G., "Introduction to the Transfer of Heat and Mass," McGraw-Hill Book Company, Inc., New York (1950).
6. Hyman, S. C., and C. F. Bonilla, *A.E.C. Rept. NYO*-560, unclassified (June 30, 1950).
7. Jakob, M., "Heat Transfer," VI, John Wiley and Sons, Inc., New York (1949).
8. McAdams, W. H., "Heat Transmission," 3d ed., McGraw-Hill Book Company, Inc., New York (1954).
9. Miller, P., J. J. Byrnes, and D. M. Benforado, *A.I.Ch.E. Journal*, 1, 501 (1955).
10. Perry, J. H., ed., "Chemical Engineers' Handbook," 3d ed., McGraw-Hill Book Company, Inc., New York (1950).
11. Sherwood, T. K., and C. E. Reed, "Applied Mathematics in Chemical Engineering," McGraw-Hill Book Company, Inc., New York (1939).

*Presented at the Nuclear Science and Engineering Congress, Cleveland.*