GENERAL ATOMIC

DIVISION OF GENERAL DYNAMICS

GA-1787

MEASUREMENTS OF HEAT-TRANSFER COEFFICIENTS, FRICTION FACTORS, AND VELOCITY PROFILES FOR AIR FLOWING PARALLEL TO CLOSELY SPACED RODS

November 28, 1960

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

GENERAL ATOMIC

DIVISION OF

GENERAL DYNAMICS

JOHN JAY HOPKINS LABORATORY FOR PURE AND APPLIED SCIENCE

P.O. BOX 608. SAN DIEGO 12. CALIFORNIA

GA-1787

MEASUREMENTS OF HEAT-TRANSFER COEFFICIENTS, FRICTION FACTORS, AND VELOCITY PROFILES FOR AIR FLOWING PARALLEL TO CLOSELY SPACED RODS

by

Luther D. Palmer and Leonard L. Swanson

Submitted for presentation at the Second International Heat Transfer Conference, to be held at Boulder, Colorado, August 28-September 1, 1961, and for publication in the Proceedings of the Conference.

Project 32.230
Contract AT(04-3)-314

November 28, 1960

ABSTRACT

The design of gas-cooled reactors with solid-rod fuel elements is predicated on knowledge of the pressure-drop and heat-transfer characteristics within the reactor core. There have been very few experimental investigations of systems in which the gas flows parallel to closely spaced rods. Therefore, heat-transfer and fluid-dynamic studies were made with air flowing parallel to the rods of a seven-rod cluster. The parallel rods were equally spaced in a triangular array with a 1.015 ratio of center-to-center distance to rod diameter, the rod surfaces forming flow passages of tricuspid-shaped cross section. Average and local heat-transfer film coefficients, pressure drops, and velocity profiles were measured under established hydrodynamic conditions.

The measured average heat-transfer film coefficients were approximately equal to those for smooth round ducts; however, the peripheral variation of the local coefficients at a Reynolds number of 20,000 was 0.5 to 1.3 of the average values. The data were obtained under the following forced-convection conditions:

Reynolds number	10,000 to 60,000
Rod heat flux	400 to 1,000 Btu/(hr)(ft^2)
Rod surface temperature	100° to 160°F

Fanning friction factors calculated from static-pressure-drop measurements made along the length of a flow passage were approximately 5% higher than those for smooth round ducts over a Reynolds number range of 3,000 to 30,000.

The velocity structure in the flowing gas was determined at a Reynolds number of 20,000. The peripheral variation of the ratio of local to bulk mean velocity was found to be very nearly equal to the ratio of the local to average heat-transfer coefficient. This agreement of the local and average heat-transfer data is to be expected from the analogy between fluid friction and heat transfer.

MEASUREMENTS OF HEAT-TRANSFER COEFFICIENTS, FRICTION FACTORS, AND VELOCITY PROFILES FOR AIR FLOWING PARALLEL TO CLOSELY SPACED RODS*

by

Luther D. Palmer† and Leonard L. Swanson†

INTRODUCTION

Recent designs of high-temperature heat exchangers and nuclear reactors incorporate coolant-flow lengthwise passages formed by the spaces between parallel rods or tubes in closely spaced equilateral arrays. The resulting coolant shear stress distribution is quite different from that found in ordinary ducts and may cause unusual heat-transfer rates within the flow passages, as well as pressure drops through the passages. Previous mathematical analyses[(1)(2)**] predict large circumferential variations of the friction and average heat-transfer film coefficients from the accepted empirical correlations. However, certain experimental studies,[(3-7)] in which the rods are widely spaced, indicate that the circumferential variation in the local heat-transfer film coefficients is much less than predicted. These experimental studies indicate, also, that conventional correlations may be used to calculate the friction and average heat-transfer film coefficients. A question then arises as to the heat-transfer rates within and pressure drops through the flow passages under conditions of closely spaced rods, that is, when the gaps between the rods are of the order of 0.01 of the rod diameter. Because of the difficulty of representing forced convection and hydrodynamic phenomena for turbulent flow in mathematical form, experimental measurements must be made. Two air-flow systems (see Fig. 1)

*These studies were sponsored by the U. S. Atomic Energy Commission, under Contract No. AT(04-3)-314.

†John Jay Hopkins Laboratory for Pure and Applied Science, General Atomic Division of General Dynamics Corporation, P. O. Box 608, San Diego 12, California.

**References are listed at the conclusion of this paper.

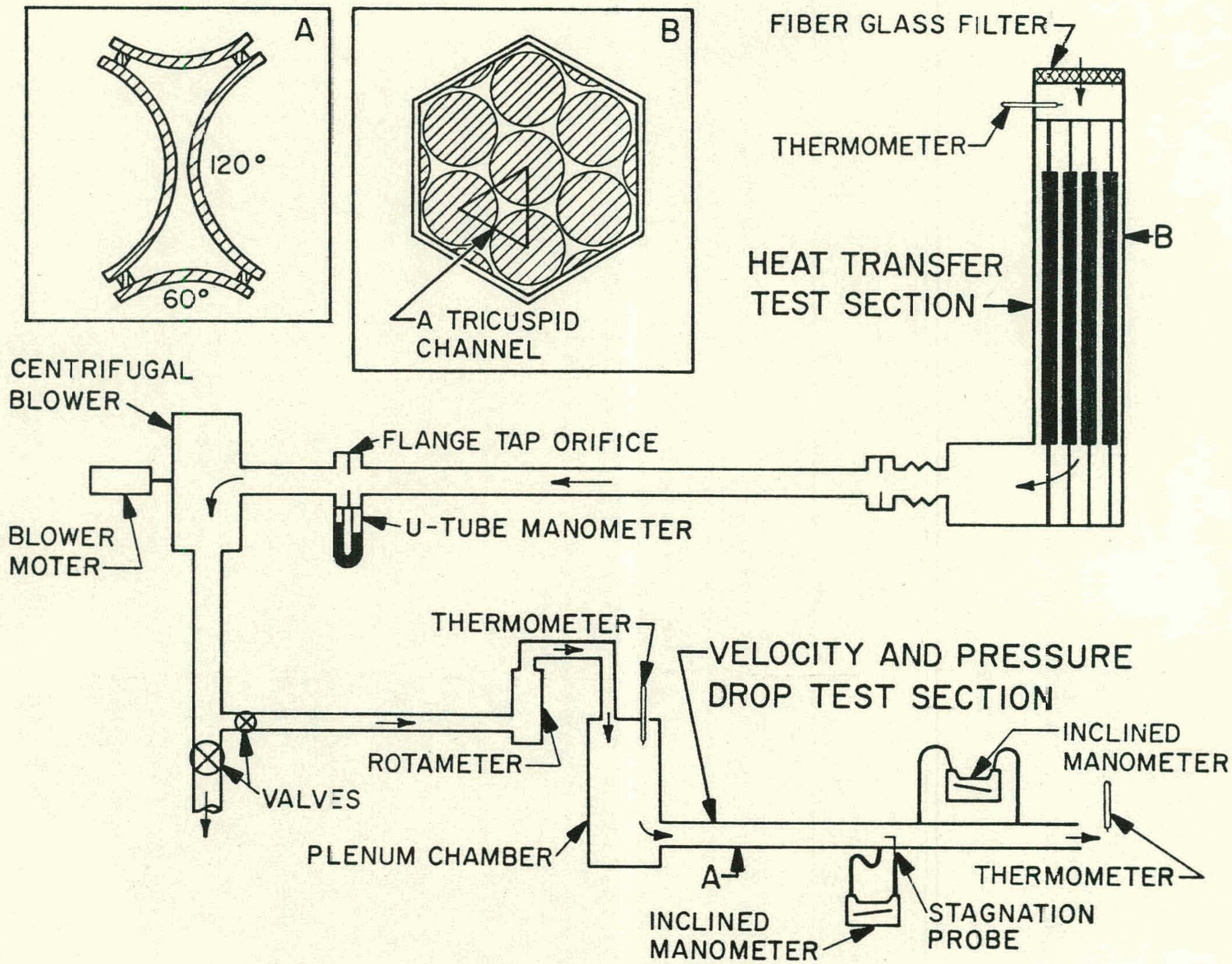


Fig. 1--Schematic drawing of the heat transfer and velocity and pressure drop test systems, showing cross-section views of each test section

were used to make these measurements, the pressure drops and velocities being measured under isothermal conditions in a dual tricuspid flow channel and the constant-wall-temperature heat-transfer coefficients being measured in a seven-rod cluster.

DESCRIPTION OF SYSTEMS*

Velocity and Pressure-drop Test System

For this system, air was delivered by a centrifugal blower through a rotameter and a plenum chamber to a dual-channel test passage from which the air was vented to the atmosphere. This test passage (see Insert A in Fig. 1) was formed by joining two 120° and two 60° circumferential segments of aluminum tubing. The leading edges of the tubing were rounded and were positioned within the plenum chamber so as to reduce entrance disturbances. The inside surfaces of the flow passage were machined and hand-finished to correspond to the surfaces of the rods in the heat-transfer lattice.

Static pressure taps were located at points 110 and 134 equivalent diameters downstream from the channel entrance. The stagnation probe penetrated the channel wall upstream from the static pressure taps. A thermometer in the plenum chamber was used to measure the temperature of the flowing air.

Heat-transfer Test System

The heat-transfer system utilized the same blower as the velocity and pressure-drop system, being connected to the inlet side of the blower. Air from the room (which served as an infinitely large plenum) was drawn through the heated test lattice, was passed through a calibrated orifice, and was then vented to the atmosphere by means of valve arrangements.

The test lattice consisted of seven rods spaced in an equilateral cluster within a hexagonal chamber, with circumferential segments of

---

*Further descriptive details of the experimental systems are presented in an Addendum to this paper.

tubing attached to the chamber walls to simulate the adjacent rods of a large array (see Insert B of Fig. 1). A thermometer for measuring the inlet-air mean temperature was located at the entrance of the chamber below a fiberglass filter, which served to remove particles from the air and to suppress flow disturbances caused by air currents within the room.

Each rod of the test lattice was composed of three contiguous hollow aluminum cylinders: (1) a thin-walled, upstream flow development portion; (2) a thick-walled, heated portion; and (3) a thin-walled, downstream exit portion. The heated portion of each rod contained a helical electrical resistance heater tightly wound on a stainless steel pipe which passed through this thick-walled aluminum cylinder. The asbestos-insulated, stainless-steel-sheathed heater wire was in firm contact with the inner surface of the cylinder, and calibrated copper-constantan thermocouples were embedded in the rod surface along this heater portion.

The central rod of the seven-rod cluster contained additional calibrated surface thermocouples, as well as a small temperature-controlled surface heater installed within the rod wall in such a manner that its surface was continuous with that of the rod itself. The electrical leads were exited through a hole located between the resistance heater and the surface of the rod in order to avoid subjecting the wire to a temperature gradient which would result in heat transfer along the wires.

EXPERIMENTAL TECHNIQUE*

Velocity and Pressure Drop

The static pressure losses in the dual-channel flow passage were measured at entrance lengths of 110 and 134 equivalent diameters over a Reynolds number range of from 3,000 to 30,000 for air at $\sim 90^{\circ}F$. The stagnation pressures were measured at an entrance length of 105 equivalent diameters for a Reynolds number of 20,000 as functions of the perpendicular

*Definitions of notation used are listed at the end of this paper.

distances from the channel wall at peripheral positions of 0°, 10°, 15°, 20°, and 30°. The Reynolds numbers for both the static-pressure and the stagnation-pressure measurements were based on the total flow and the equivalent diameter of the dual-channel passage.

Heat Transfer

The constant wall-temperature heat-transfer film coefficients, both local, as a function of peripheral position, and average, were measured and correlated as a function of a Reynolds number based on the total flow and equivalent diameter of the test lattice.

The average heat-transfer film coefficients were calculated by using the equation

$$\bar{h} = \frac{\bar{q}}{A\left(t_{s,x} - \frac{x}{X}\frac{\bar{q}}{\dot{m}c_p} - t_{m,in}\right)} . \tag{1}$$

The rod surface temperatures, $t_{s,x}$, were measured at three hydrodynamic entrance lengths of 38.4, 57.3, and 74.0 equivalent diameters; the upstream heated length of each was 22.2 equivalent diameters. The total flow rate through the lattice, $\dot{m}$, which necessitated the use of the total heating rate, was used because of the difficulty involved in measuring the individual flow rate through each passage. Velocity measurements with pitot-static probes located at the center of each of the tricuspid passages indicated less than 2% difference in flow rate. Comparison of these velocity measurements with the velocity profiles measured in the velocity and pressure-drop test passage, showed that the indicated flow rates were within 4% of those calculated using the total flow and cross-sectional area of the lattice.

The local circumferential variations of the heat flux were determined from measurements of the electrical power necessary for equalizing the temperature of the small heater surface to that of the adjacent tube wall while the central rod was rotated about its longitudinal axis, the small heater passing from the smallest to the largest interstice between the rods. This technique is valid if (1) the rod containing the small heater has a circumferentially constant wall temperature and (2) the small heater is either thermally insulated or guard-heated and calibrated. The first condition was satisfied: An electrical analog plot indicated that in the worse case measured (where the simulated circumferential variation of the heat flux was 0.5 to 1.3 and the small heater was positioned at the largest gradient of the surface heat flux) the maximum circumferential temperature difference was 1.8^{o}F and the temperature drop across the surface heater was 0.7^{o}F. The second requirement was fulfilled by having a guard heater below the surface heater to prevent heat losses through the lead wires and by calibrating the central rod containing the small surface heater in an annulus system under proper temperature and heating conditions.* Integration of the local heat flux at entrance lengths of 38.4, 57.3, and 74.0 equivalent diameters yielded average values which were within ±10% of those determined from the overall lattice measurements. Errors in measurement which could be attributed to the technique were minimized, in dimensionless comparisons, by using the average heat flux obtained by integrating the peripheral values. Consequently, the ratio of local to average heat flux can be considered equivalent to the local-to-average film coefficients.

RESULTS OF MEASUREMENT

Velocity and Pressure Drop

The static pressures were correlated in dimensionless form (Fanning friction coefficient) as a function of channel Reynolds number (see Fig. 2). The values, which have a maximum possible range of error of 6%, are between those for smooth ducts and those for commercial pipe.[8] Published correlations of pressure losses with air flowing through smooth tubes[9] are also shown in Fig. 2 for comparison. It appears that the pressure losses

---

* See Addendum.

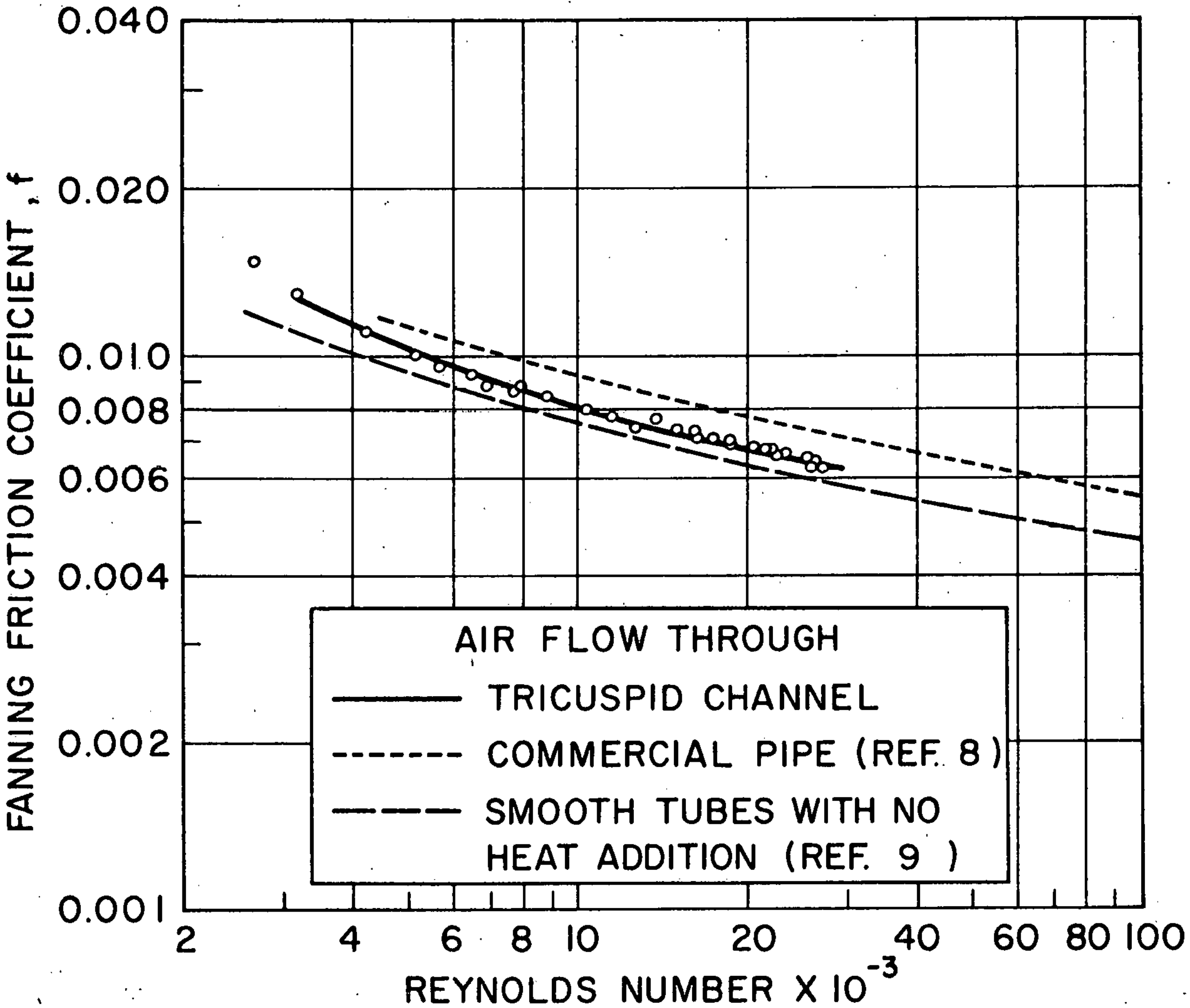


Fig. 2--Measured Fanning friction as a function of Reynolds number

reported here are consistent with other air-flow measurements.

The stagnation pressures for air flowing at a Reynolds number of 20,000 were converted to velocities, which are shown in Fig. 3 as a function of dimensionless distance from the duct wall with peripheral position as a parameter. An error analysis of these measurements indicates a maximum possible error of 3% in the pressure measurements and a possible 5% in the distance measurements.

Heat Transfer

The average heat-transfer data in dimensionless form are shown in Fig. 4 as a function of Reynolds number. The values tend to be below the Dittus-Boelter relation which applies to fluid forced convection in smooth pipes; however, upon comparison with others,[9, 10] the data appear to be consistent with previous measurements.

The measured peripheral variations of the local heat-transfer film coefficients are shown in Fig. 5 as a function of peripheral position with the Reynolds number as a parameter. The values obtained by graphically integrating these curves were within ±10% of the averages calculated from the over-all measurements.

It is interesting to note the insensitivity of the data to guard heating, which indicates that the conduction heat losses from the surface heater, in the absence of guard heating, are nearly uniform with respect to peripheral position.

An important requirement of this study was that the measurements be made under stabilized hydrodynamic conditions; therefore, three sets of upstream flow development sections of different lengths were used to create varying entrance lengths. The three lengths of the flow development sections resulted in the small surface heater being located at 38.4, 57.3, and 74.0 equivalent diameters from the entrance. The measured heat-transfer rates shown in Fig. 4 demonstrate that at Reynolds numbers greater than 20,000 there is no appreciable difference in the values for

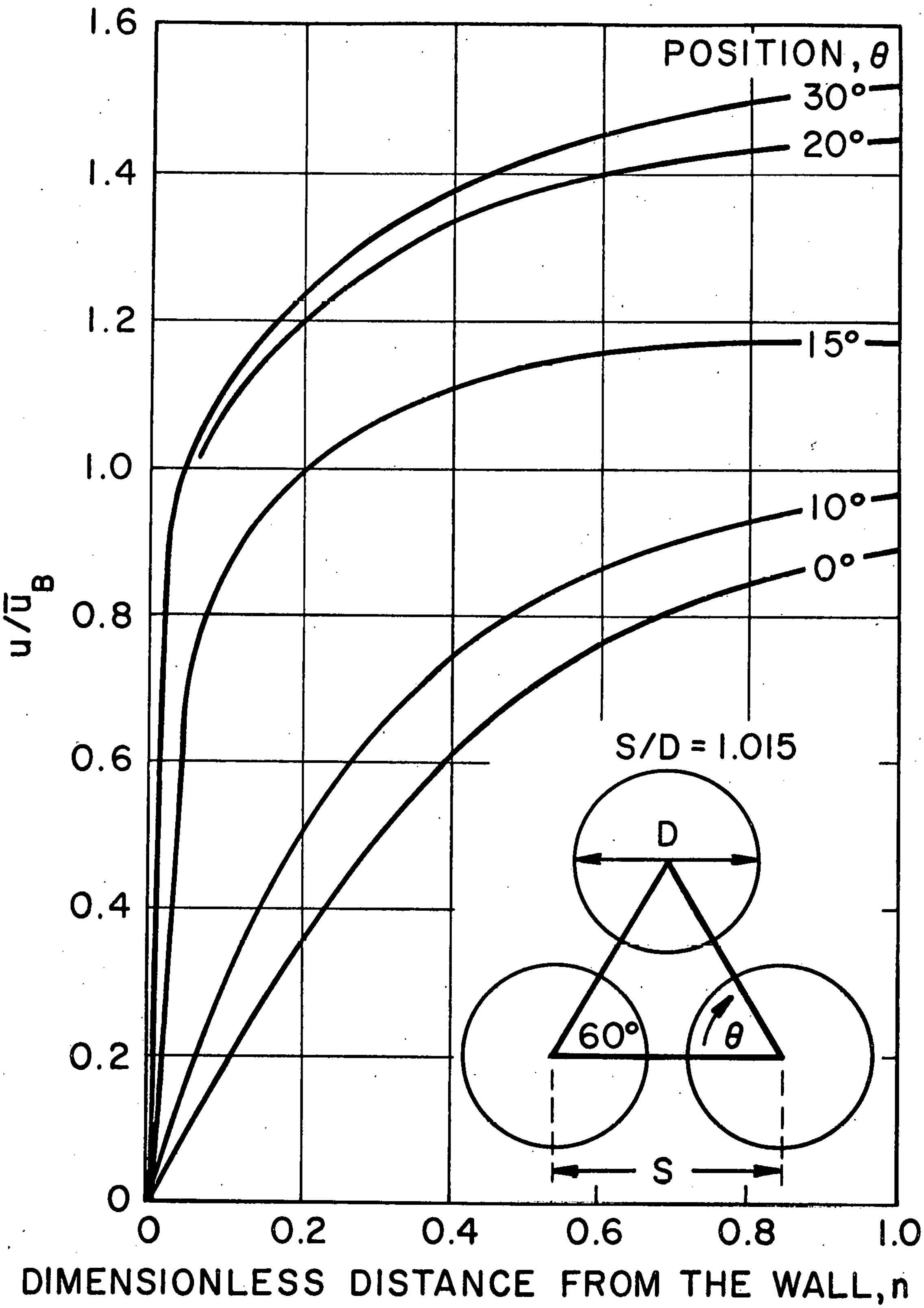


Fig. 3--Velocity profile at a Reynolds number of 20, 000

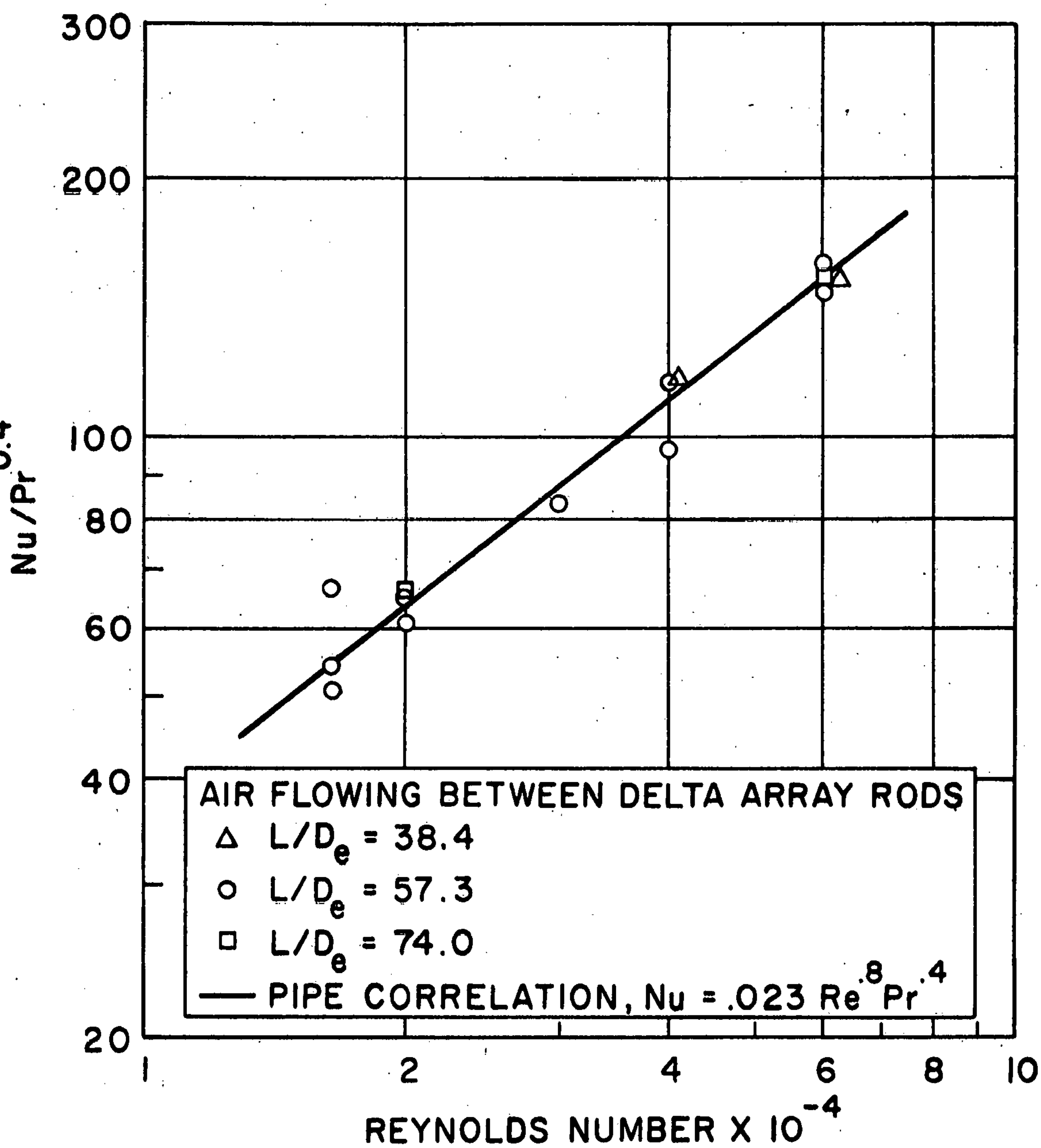


Fig. 4--Measured over-all heat-transfer rates from the rod surface to air, as a function of Reynolds number

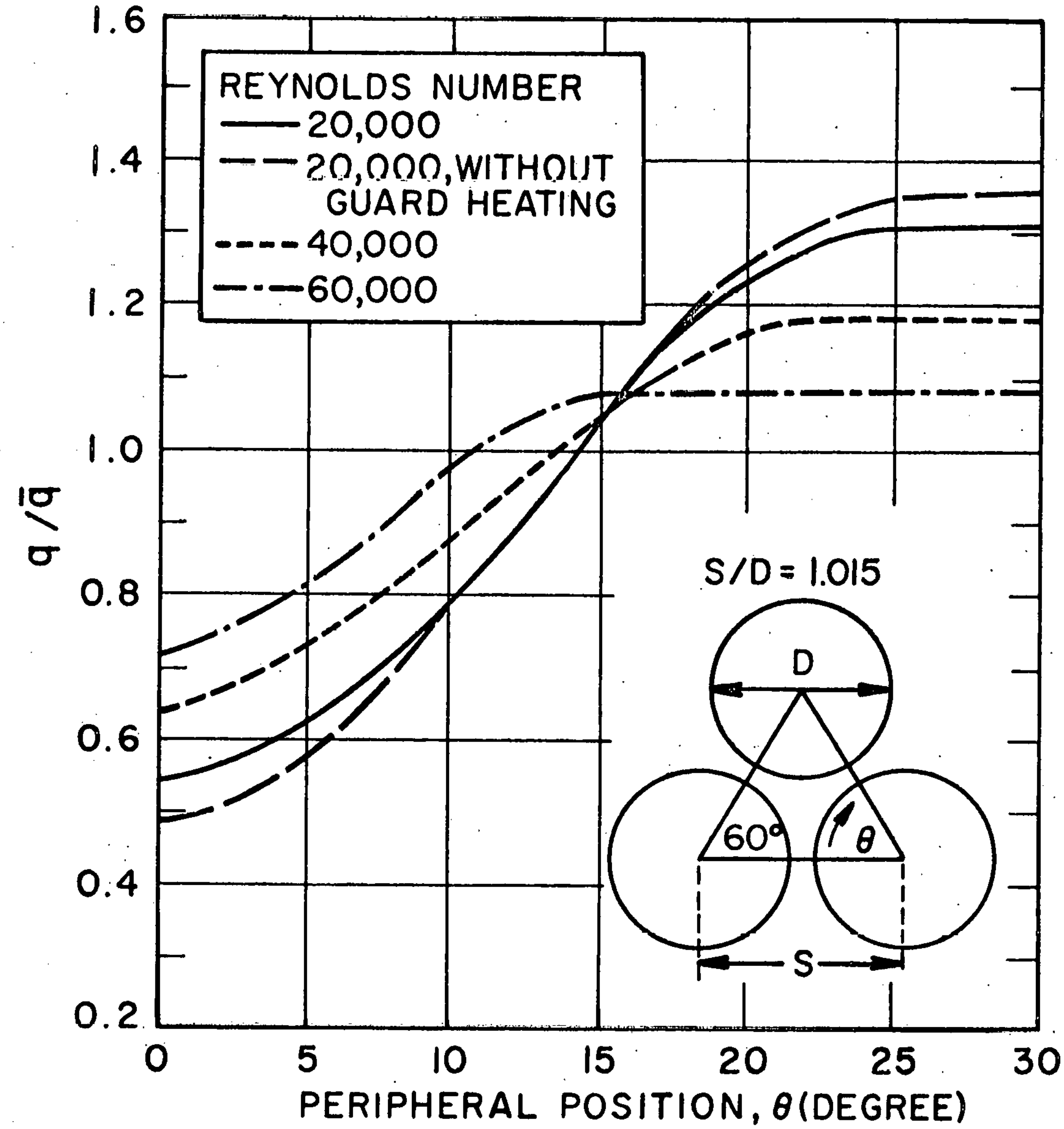


Fig. 5--Peripheral variation of the measured heat-transfer rates

the three entrance lengths. It appears that the correlation presented here can be used in heat-transfer calculations for cases far downstream where fully developed flow is known to exist.

DISCUSSION OF RESULTS

The results of this investigation lend credence to the practice of using the equivalent-diameter concept in conventional heat-transfer and fluid-friction correlations for the purpose of determining the over-all characteristics of flow along tube bundles with a ratio of pitch to rod-diameter of 1.015. However, no such empirical relations exist which describe the peripheral variation of the local heat-transfer coefficient under these conditions. This nonuniformity of heat conductance is, therefore, experimentally indicated as a serious problem in the use of this type of solid-to-gas heat-transfer process.

A comparison of the average heat-transfer and friction film coefficients can be made by utilizing the Colburn function, which is usually given as

$$f/2 = \mathrm{St}\,\mathrm{Pr}^{2/3} . \tag{2}$$

However, previous experiments with gases have yielded values of f/2 which are ~10% higher than the St $\mathrm{Pr}^{2/3}$ term. This difference is also found in the measurements presented here, as shown in Fig. 6, and is in agreement with an analysis published in Ref. 11.

A comparison of the ratio of the local mean velocity to the duct mean velocity and the ratio of the local to mean heat-transfer coefficients is shown in Fig. 7 as a function of peripheral position. The local mean velocities ($\bar{u}_{r,\theta}$) were obtained by integrating the velocity profiles normal to the rod surface with respect to the distance from the wall to the line of symmetry within the duct. The dependence of the local heat-transfer rates on the local mean velocities is quite evident.

The data presented here for air flowing under the described conditions can be summarized as follows:

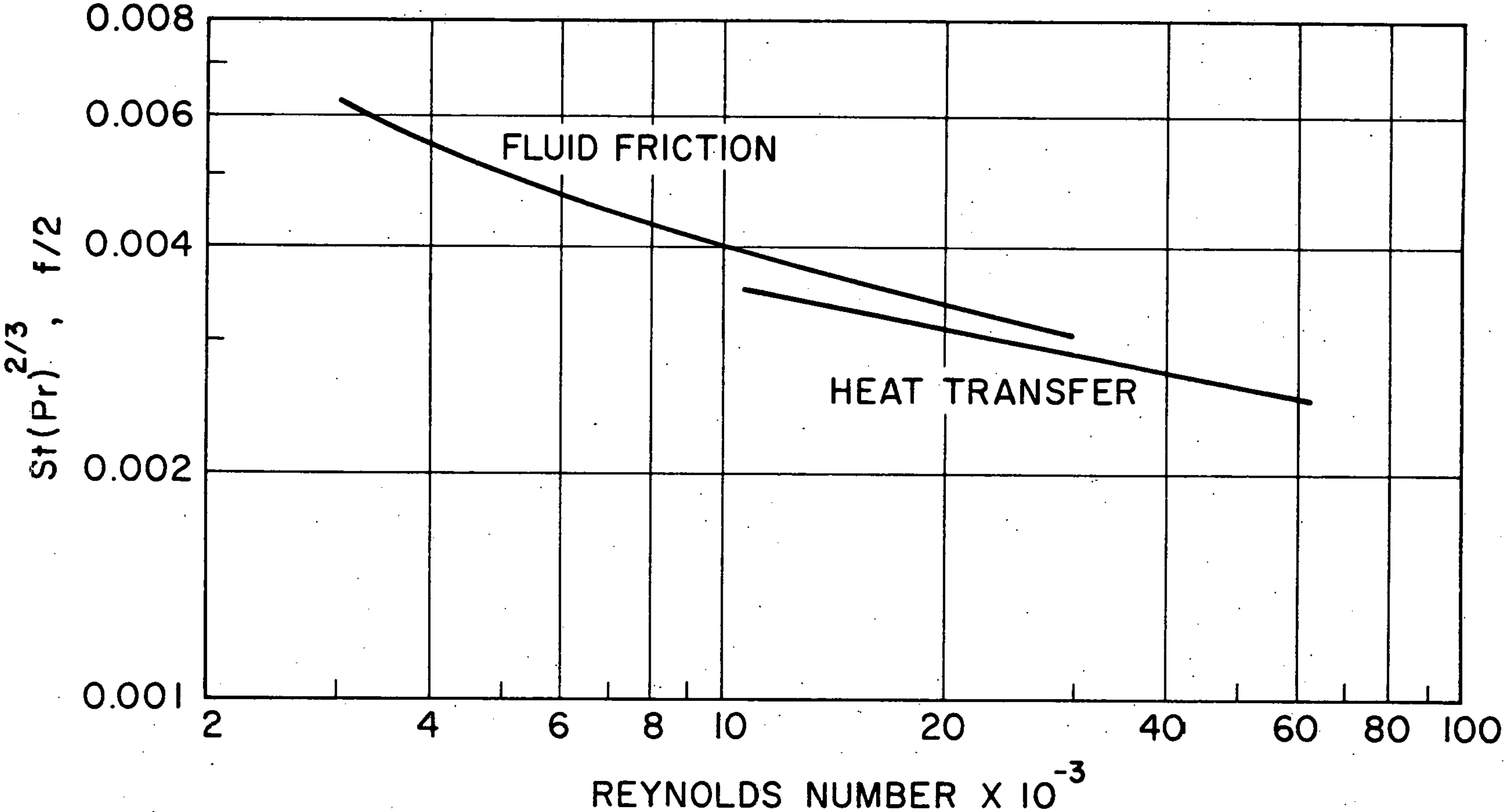


Fig. 6--Comparison of fluid-friction and average heat-transfer data

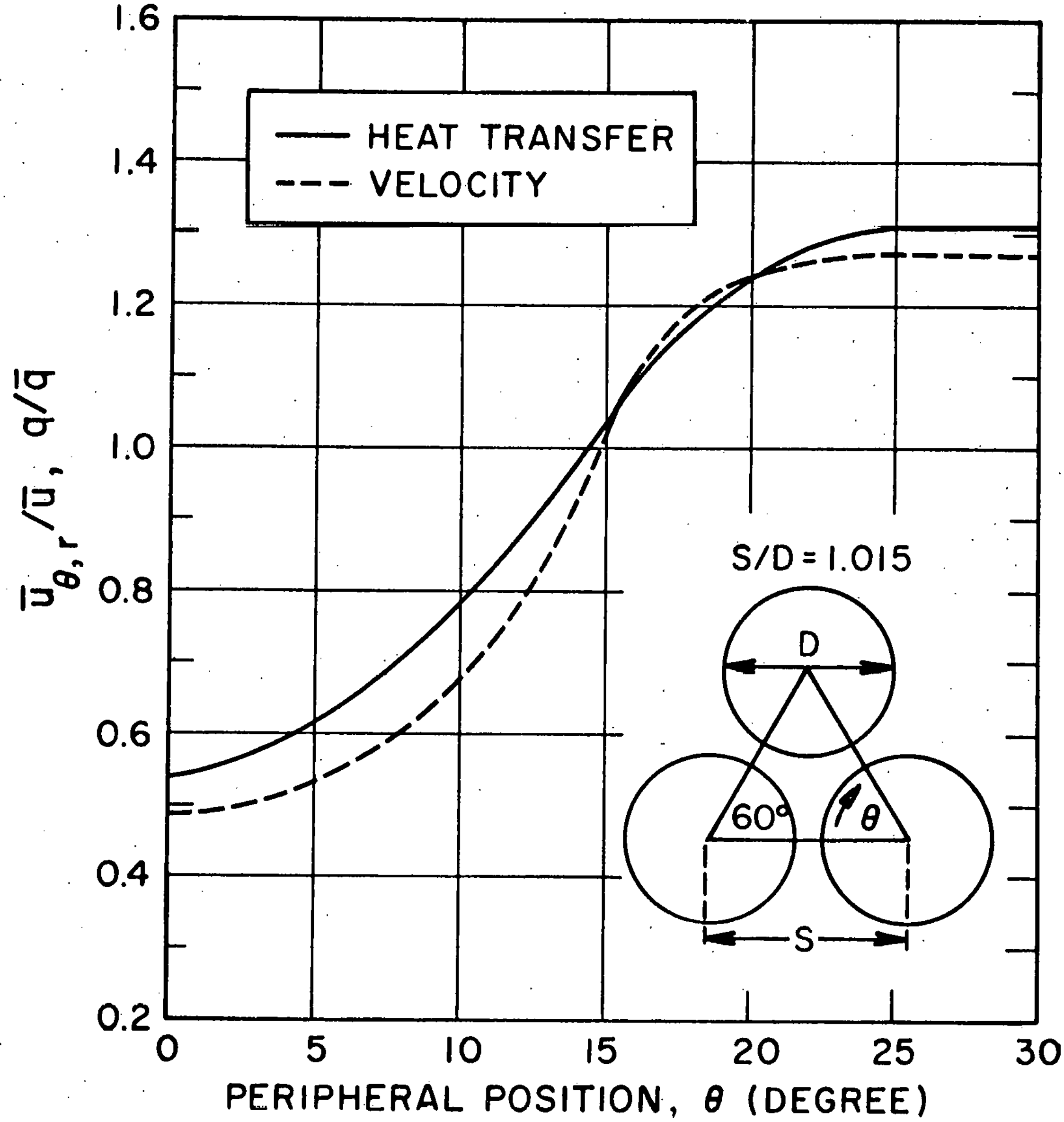


Fig. 7--Comparison of local heat-transfer film coefficients and local mean velocities at a Reynolds number of 20, 000

1. Conventional correlations may be used to predict the average heat-transfer coefficients and pressure losses.
2. The local heat-transfer coefficient varies by a factor of approximately three for a Reynolds number of 20, 000, the factor becoming progressively less as the air flow rate is increased.

ACKNOWLEDGMENTS

The authors wish to express their appreciation to Dr. P. Fortescue and Mr. C. Rickard for their encouragement and advice during the course of the investigation.

NOTATION

Letters

A = Heat-transfer area, ft^2

c_p = Specific heat, Btu/(lb)($^{\circ}$F)

D = Rod diameter, ft

D_e = Equivalent diameter (4 A/wetted perimeter)

f = Fanning friction factor, defined by $f = \frac{\Delta p}{\rho(\bar{u}_B)^2} \frac{D_e g}{2L}$

g = Gravitational constant, ft/hr^2

$\bar{h}$ = Mean heat-transfer film coefficient, Btu/(hr)(ft^2)($^{\circ}$F)

k = Thermal conductivity, Btu/(hr)(ft^2)($^{\circ}$F)/ft

L = Length, ft

m = Mass flow rate, lb/hr

n = Dimensionless distance from rod surface (perpendicular distance from rod ÷ perpendicular distance from rod to line of symmetry between rods)

Δp = Differential pressure, lb/ft^2

q = Local rate of heat transfer, Btu/hr

$\overline{q}$ = Mean heat-transfer rate, Btu/hr

S = Rod pitch spacing, ft

$t_{m,in}$ = Mixed mean inlet fluid temperature, ^{o}F

$t_{x,X}$ = Rod local surface temperature, ^{o}F

u = Local velocity, ft/sec

$\overline{u}_{r,\theta}$ = Local mean velocity, ft/sec

$\overline{u}_B$ = Bulk or average velocity, ft/sec

x = Distance from beginning of heated section, ft

X = Total length of heated section, ft

θ = Angular position, degrees

μ = Fluid dynamic viscosity, $lb_m/(hr)(ft)$

ν = Fluid kinematic viscosity, ft^2/hr

ρ = Fluid density, lb_m/ft^3

Dimensionless Numbers

Nu = Nusselt number, $\overline{h}D_e/k$

Pr = Prandtl number, $\mu c_p/k$

Re = Reynolds number, $D_e\overline{u}/\nu$, $D_e\overline{u}_B/\nu$

St = Stanton number, Nu/Re Pr = $\overline{h}/c_p\overline{u}_B\rho$

BIBLIOGRAPHY

1. "Analysis of Axial Turbulent Flow and Heat Transfer Through Banks of Rods or Tubes," by Robert G. Deissler and Maynard F. Taylor, Reactor Heat Transfer Conference of 1956: Collected Papers and Reports, John E. Viscardi (comp.), TID-7529 (Pt. 1), Book 2, November 1957, p. 416 ff.

2. "Longitudinal Laminar Flow Between Cylinders Arranged in Regular Array," by E. M. Sparrow and A. L. Loeffler, Jr., American Institute of Chemical Engineers Journal, Vol. 5, No. 3, 1959, p. 325.

3. "Heat Transfer from Parallel Rods in Axial Flow," by David A. Dingee, Wayne B. Bell, Joel W. Chastain, and Sherwood L. Fawcett, Battelle Memorial Institute, BMI-1026, August 5, 1955.

4. "Heat Transfer from Parallel Rods in Axial Flow," by D. A. Dingee and J. W. Chastain, Reactor Heat Transfer Conference of 1956: Collected Papers and Reports, John E. Viscardi (comp.), TID-7529 (Pt. 1), Book 2, November 1957, p. 462 ff.

5. "Heat Transfer to Water Flowing Parallel to a Rod Bundle," by Philip Miller, James J. Byrnes, and David M. Benforado, Paper presented at the Nuclear Engineering and Science Congress, held in December 1955, Preprint by the American Institute of Chemical Engineers, New York.

6. "Pressure Drop Through Parallel Rod Subassemblies Having a 1.12 Equilateral Triangular Pitch," by B. W. Le Tourneau, R. E. Grimble, and J. E. Zerbe, Westinghouse Electric Corp., Atomic Power Division, Report WAPD-TH-118, August 29, 1955.

7. "Measurement of Heat-Transfer and Friction Coefficients for Flow of Air in Noncircular Duct at High Surface Temperatures," by Warren H. Lowdermilk, Walter F. Weiland, Jr., and John N. B. Livingood, National Advisory Committee for Aeronautics (N. A. C. A.), Research Memorandum E53J07, January 25, 1954.

8. "Heat Transmission," by William H. McAdams, 3rd Edition, McGraw-Hill Book Company, Inc., New York, N. Y., 1954, p. 156.

9. "Measurements of Average Heat-Transfer and Friction Coefficients for Subsonic Flow of Air in Smooth Tubes at High Surface and Fluid Temperatures," by Leroy V. Humble, Warren H. Lowdermilk, and Leland G. Desmon, N. A. C. A., Report 1020, 1951.

10. "Correlation of Forced-Convection Heat-Transfer for Air Flowing in Smooth Platinum Tubes with Long Approach Entrance at High Surface and Inlet-Air Temperatures," by L. G. Desmon and E. W. Sams, N. A. C. A., Research Memorandum E50H23, 1950.

11. "Remarks on the Analogy Between Heat Transfer and Momentum Transfer," by L. M. K. Boelter, R. C. Martinelli, and Finn Jonassen, Trans. A.S.M.E., Volume 63, No. 5, 1941, pp. 447-455.

ADDENDUM

DESCRIPTION OF TYPICAL HEAT-TRANSFER ROD AND SURFACE HEATER

Drawings of a typical heat-transfer rod and a surface heater are shown in Figs. 1 and 2. A photograph of the test lattice during assembly is shown in Fig. 3. Engineering drawings of the heat transfer and hydrodynamic systems may be obtained by contacting the authors.

METHOD OF CALIBRATION OF THE SMALL SURFACE HEATER

Measurements of the rate of heat transfer from the inner wall of an annulus to air were made in order to determine the validity and resolution of the technique used in measuring the local and average heat-transfer rates within the rod-bundle test lattice. The central rod of the tube bundle was centered within a tube of 7-5/8-in. inside diameter. The annulus formed, thus, replaced the tube bundle. There were no other changes in the flow system. The heat transfer measurements were made under the following conditions:

Reynolds number	12,000 to 60,000
Inner-wall heat flux	370 to 1,150 Btu/(hr)(ft^2)
Rod surface temperature. . . .	93° to 130°F

The data for heat transfer in annuli can be correlated if the usual variables plus the ratios of the inner to outer wall diameters are included. Davis* has used a dimensionless grouping to relate several different sizes of annuli and types of fluids; the data obtained in the annulus of this experiment were compared with the Davis correlation (see Fig. 4).

It was found necessary to guard-heat the small surface heater in order to equalize the heat flux from the heater surface to that from the adjacent rod

---

*"Heat Transfer and Pressure Drop in Annuli," E. S. Davis, Transactions of the A.S.M.E., Vol. 65, No. 7, 1943, pp. 755-759.

surface. This guard-heating prevented an apparent heat loss of about 10%. The amount of guard-heating necessary for flow conditions of interest was determined, and this information was used in the heat-transfer measurements within the tricuspid channels of the test lattice.

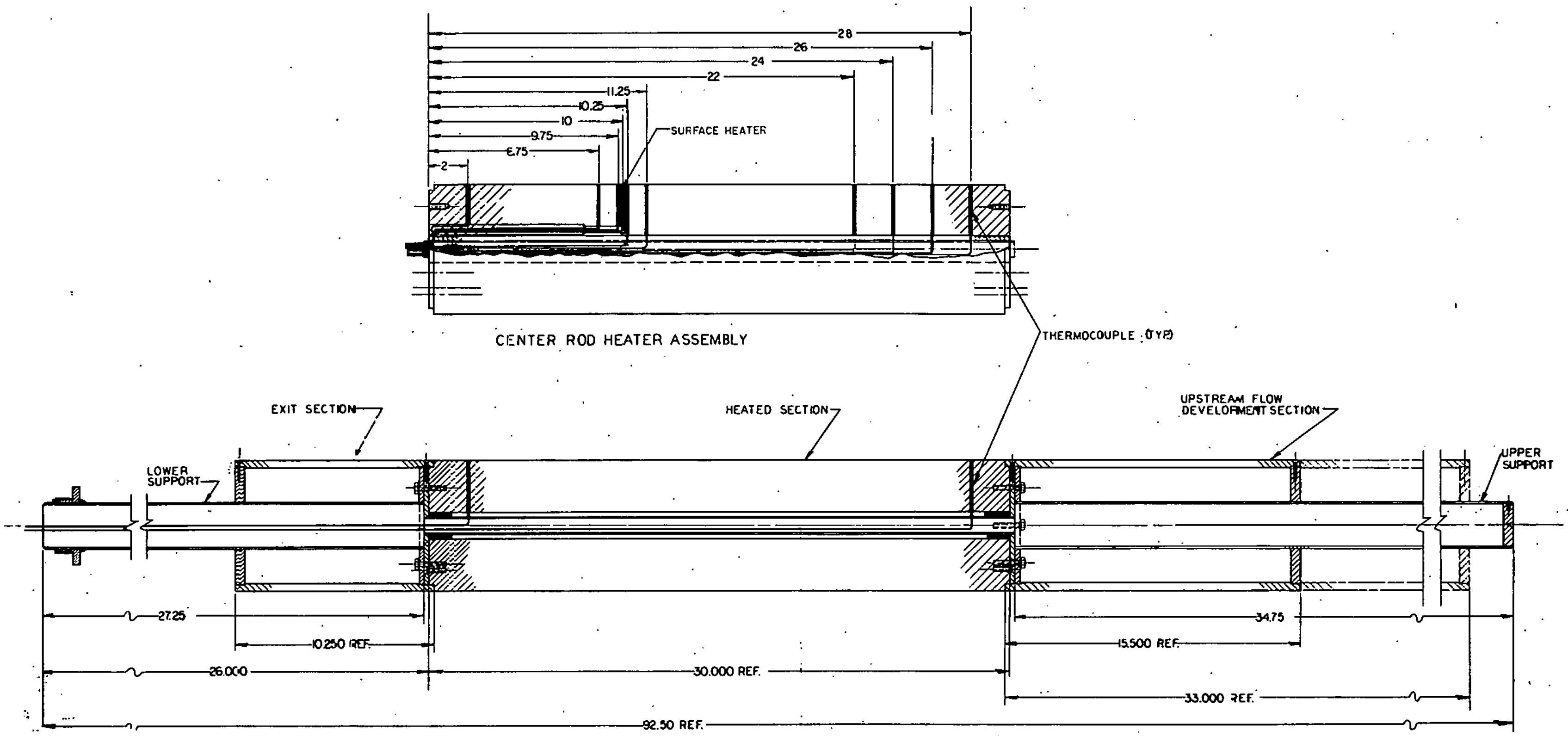
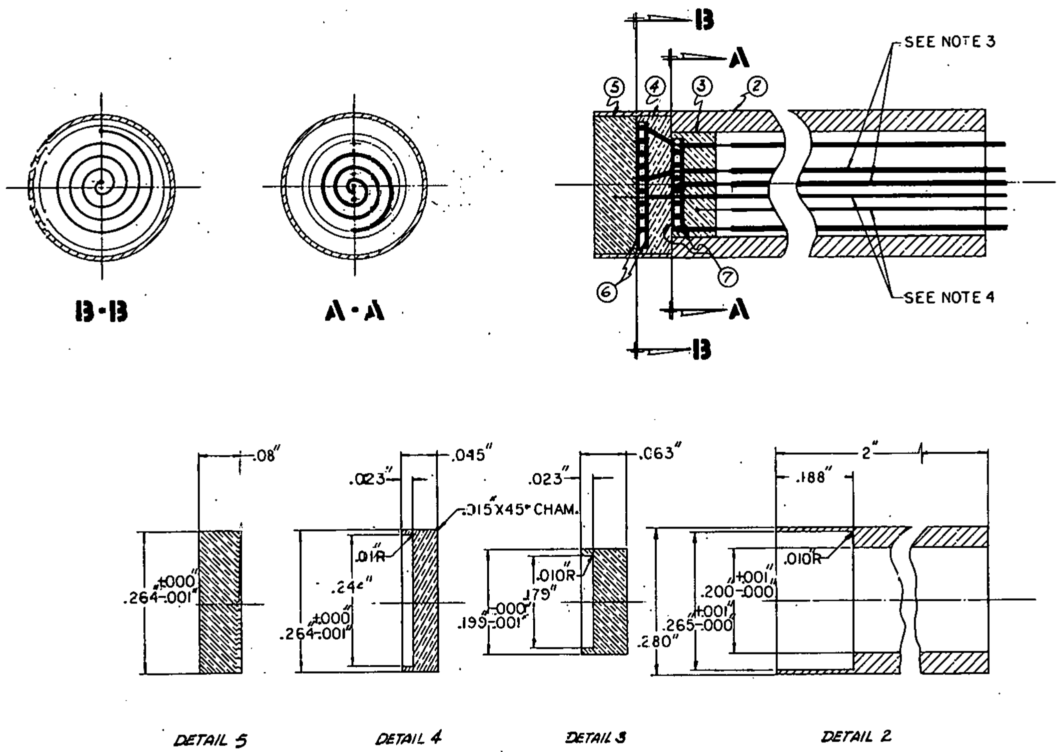


Fig. 1--Drawing of a heat-transfer test rod



NOTES -

1. REMOTE ALL BURRS & SHARP EDGES
2. ITEMS 3 & 5 TO BE GLUED IN PLACE
3. HEATING COILS TO BE 1 OHM. OR APPROX. 1" (IN LENGTH) OF 32 GA. NICHROME WIRE COILED & INSTALLED AS SHOWN. HOLES FOR ENTRANCE OF WIRE TO BE 1/64 DIA. & DRILLED AT ASSEMBLY.
4. THERMOCOUPLES TO BE MADE FROM 36 GA. COPPER-CONSTANTAN WIRE & INSTALLED APPROX. AS SHOWN. WIRE INSULATION DIA. TO DETERMINE DIA. OF HOLES (TO BE DRILLED AT ASSEMBLY) FOR ENTRANCE
5. THERMOCOUPLE AND HEATING COIL LEADS TO BE APPROX. 15 FT. IN LENGTH TO PERMIT ASSEMBLY IN ELEMENT DWG. NO. 32-SK-1295

ITEM	PART NO.	DESCRIPTION	MATL.
7		.244 DIA X .010	MICA
6		.179 DIA X .010	MICA
5		.264 DIA X .08 LG	AL. AL.
4		.264 DIA X .045 LG	AL. AL.
3		.199 DIA X .063 LG	AL. AL.
2		.280 DIA X 2" LG	FORMICA
1		ASSEMBLY	

Fig. 2--Drawing of the surface heater

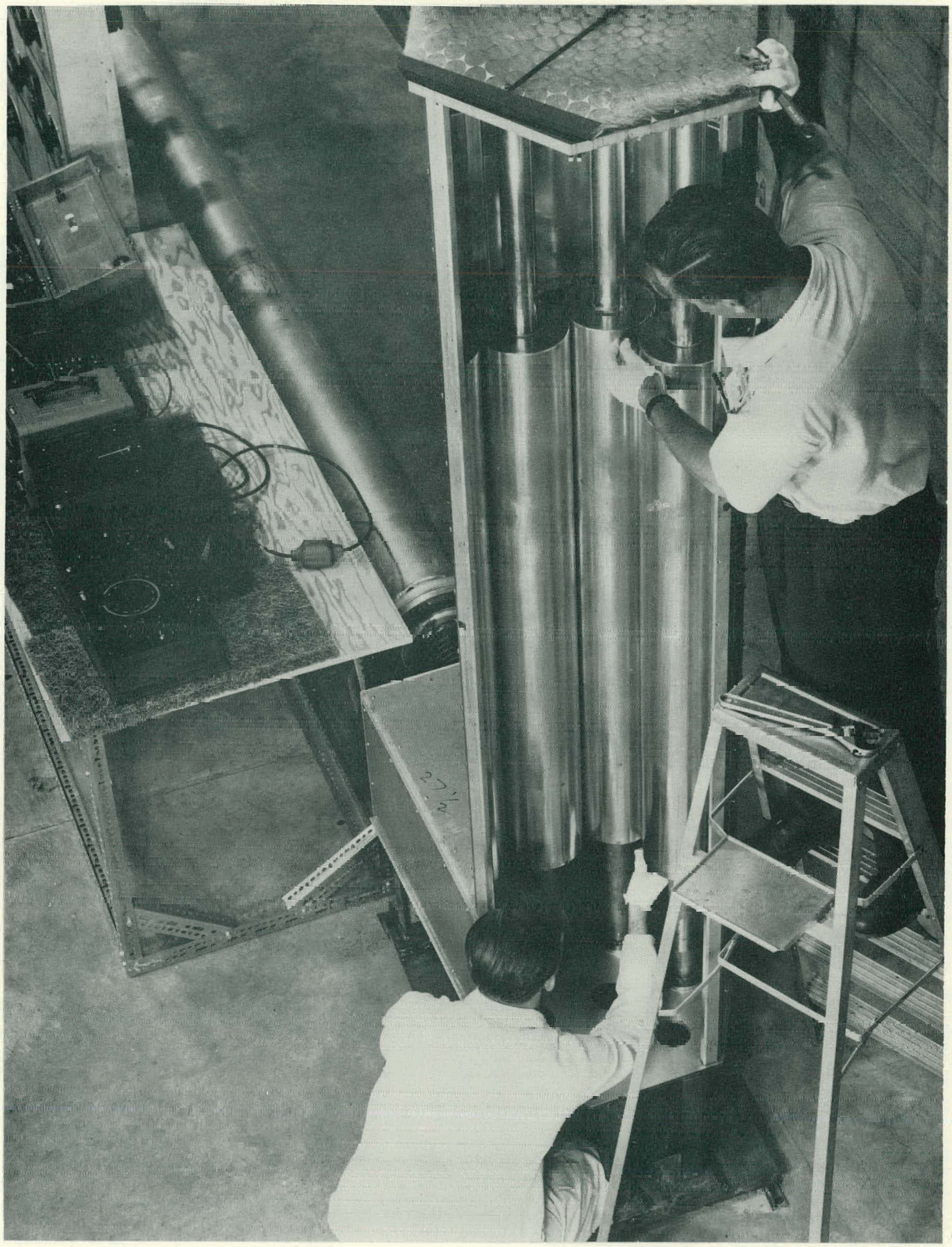
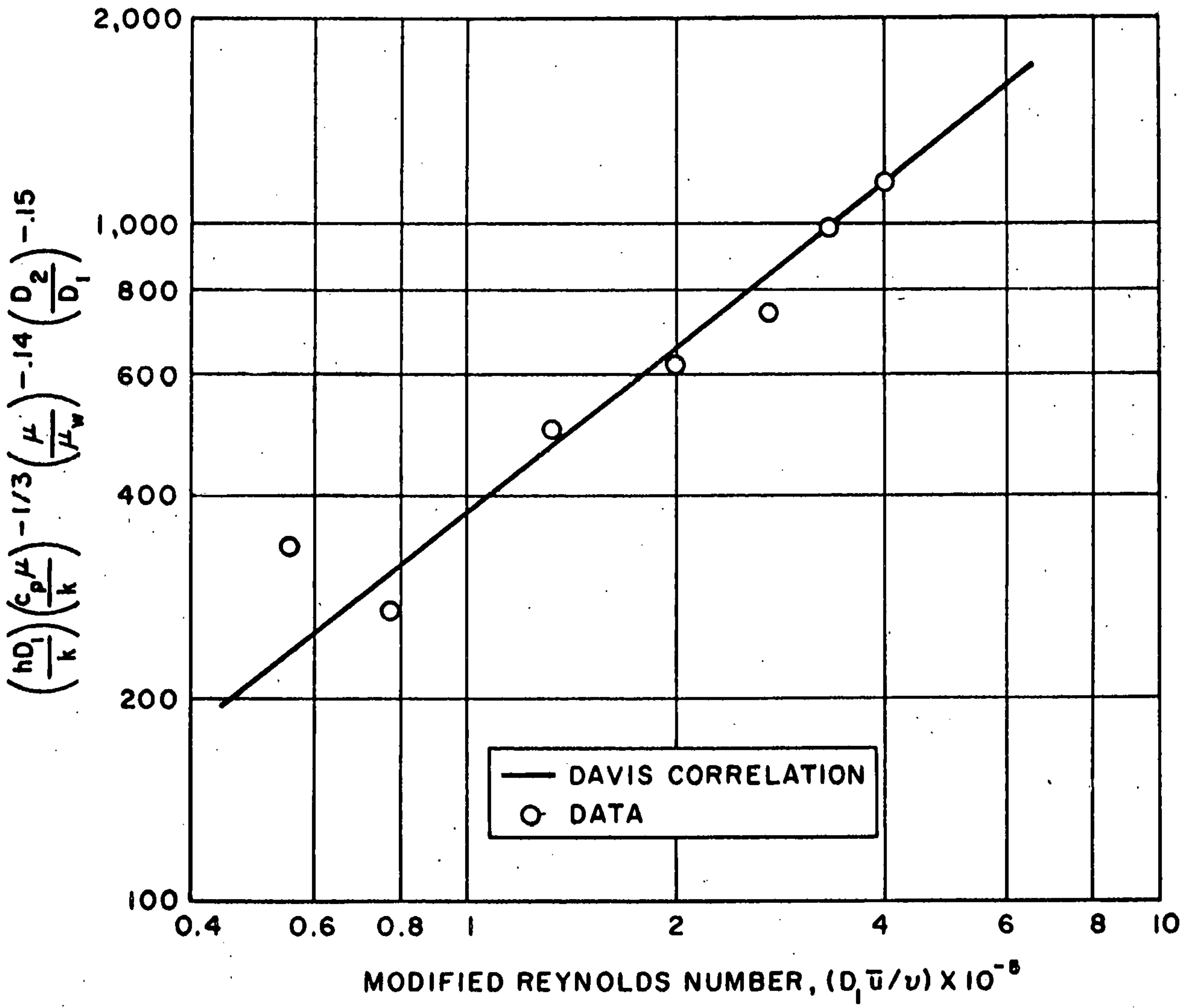


Fig. 3--Heat transfer test lattice



Here, c_p Specific heat, Btu/(lb) ($^{\circ}$F)

D_1 Inner diameter of annulus, ft

D_2 Outer diameter of annulus, ft

h Mean heat transfer film coefficient, Btu/(hr) (ft^2) ($^{\circ}$F)

k Thermal conductivity, Btu/(hr) (ft^2) ($^{\circ}$F)/ft

$\bar{u}$ Average velocity, ft/sec

μ Air dynamic viscosity, lb/(hr) (ft)

μ_w Dynamic viscosity of air at wall, lb/(hr) (ft)

ν Air kinematic viscosity, ft^2/hr

Fig. 4--Comparison of annular heat-transfer data to Davis correlation for turbulent flow