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AN EXPERIMENTAL SURVEY OF FLOW ACROSS BANKS OF ELLIPTICAL AND POINTED TUBES

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ADVANCE RESTRICTED REPORT

AN EXPERIMENTAL SURVEY OF FLOW ACROSS BANKS OF ELLIPTICAL AND POINTED TUBES

By Upshur T. Joyner and Carl B. Palmer

SUMMARY

An experimental survey of the details of the flow of fluids across banks of streamline tubes has been conducted as a continuation of previous work. Information that clarifies the picture of the flow has been obtained by surveys of total, dynamic, and static pressure, and by hot-wire cooling surveys of the unheated tubes.

When the tubes were of such shape and were arranged in such manner as to allow the fluid to flow across the bank in well-defined streams, there was little pressure drop due to breakaway from the tube surface and the formation of vortices. The concept of air flowing through a uniform passage between the tube wall and a sheet of low-velocity air has been further justified for tubes with high form drag.

INTRODUCTION

The work herein reported is a continuation of the work reported in reference 1 in which the subject of air flow across banks of circular tubes was investigated with regard to friction losses and heat transfer. The present study is a preliminary investigation to determine a tube shape suitable for use in a bank of tubes with the object of reducing the pressure losses due to separation of the flow from the rear part of the tube. The value of a detailed investigation of fluid flow through tube banks lies in its application to the design of heat exchangers, where the major problem is to obtain adequate heat transfer with minimum power loss. It is hoped that the detailed information presented on flow through tube banks will be useful in the design of more efficient tubular intercoolers.

In extending the work that has been reported in reference 1, the same general test methods have been used. Tubes of various cross-sectional shapes have been used in an effort to reduce the high form drag and the stagnation areas and to eliminate the reduced cooling areas that were encountered on the downstream side of the round tube. The pressure-drop and friction-factor data given are quantitative, but the cooling results are qualitative only, being based on hot-wire flow surveys.

SYMBOLS

a minor axis of tube cross section, feet

b height of duct, feet

C_D total drag coefficient

D_h hydraulic diameter of passage, feet $\left(\frac{4 \text{ cross-sectional area}}{\text{wetted perimeter}}\right)$

D_t hydraulic diameter of tube, feet

f friction factor $\left(\frac{\Delta p}{q_t} \frac{D_h}{4L}\right)$

I current to Wheatstone bridge, amperes

I_o current to Wheatstone bridge when no air flows over hot wire, amperes

L length of passage, feet

Δp pressure drop, inches of alcohol

q_t dynamic pressure in bank, inches of alcohol $\left(\frac{1}{2}\rho V_t^{2}\right)$

s lateral spacing, feet

V velocity of air in duct, feet per second

V_t average velocity of air at minimum section between tubes in bank, feet per second

ρ mass density of air, slugs per cubic foot

μ absolute viscosity of air, slugs per second per foot

The hydraulic diameter of the air passage D_h is calculated for the passage formed by the low-velocity wake, the tube wall at the minor axis (traverse III in figs. 8 and 9), and the top and bottom of the duct; that is,

$$D_h = \frac{4(S/2 \times b)}{2(S/2 + b)}$$

APPARATUS AND METHODS

Measurements were made in the duct shown in figure 1. The banks were made up of wooden tubes 8 inches long, with cross sections as shown in figure 2. The tubes were always arranged in the bank with succeeding rows staggered and were overlapped enough to make the separation between adjacent tubes in succeeding rows just half the lateral spacing between tubes in the same row. This arrangement is used because it allows a minimum of expansion and contraction in the air passage. Several banks are illustrated in figure 3.

One A tube and one C tube had surface orifices spaced at half-inch intervals around one side of the tube for determining the pressure distribution over the surface of the tube.

Within the tube bank, static- and total-pressure data were obtained by means of a probe consisting of a 0.030-inch tube closed at the lower end and with a 0.004-inch hole drilled in the side. (See fig. 4.) This tube extended 4 inches into the duct from the top and could be moved laterally along a slit in the top of the duct and rotated to place the drilled hole in any desired position. Total pressure was measured at the position of maximum pressure; the probe was rotated 80° from this position to obtain the static pressure. Surveys with the probe were made at four positions, as shown in figures 5, 6, and 7.

Velocity and pressure surveys were made at several positions behind the bank of elliptical tubes A. (See fig. 8.) Similar surveys for round tubes, taken from reference 1, are shown in figure 9. As is to be expected, the increased turbulence behind the bank caused the dynamic-pressure measurements to be larger behind the bank than in front of the bank.

Hot-wire data for flow near the tube surface was obtained by mounting a 1-inch length of 0.003-inch platinum wire 0.014 inch from the tube surface at various points about the tube. This wire was used as one arm of a

Wheatstone bridge of which the other three arms were fixed constantan resistances, and enough current was supplied to the bridge to maintain the platinum wire at constant temperature. Bridge current required to keep the bridge balanced is a measure of the air velocity in the vicinity of the wire and is a direct measure of the heat-transfer coefficients.

RESULTS AND DISCUSSION

Pressure data.- The data showing total and dynamic pressures within the bank, for round as well as streamline tubes A and B, are shown in figures 5 to 7. The low total-pressure wake behind tube A is not so broad as for the round tubes, and the stagnation area in front of the following tube is narrower and blankets less tube surface than with round tubes.

Total, static, and dynamic pressures at distances of 8 inches and 23 inches behind the tube bank were taken with a pitot-static tube. Data (fig. 8) indicated that static pressure was uniform across the duct. The large variations in dynamic pressure immediately behind the bank, due to geometric arrangement of the tubes, gradually disappeared farther from the bank, as was to be expected. Comparison with data for round tubes (fig. 9) shows that, for a given pressure drop over the bank, a much greater air flow is obtained with the elliptical tubes.

A comparison of the distributions of dynamic pressure for tubes A and B having the same minor axis, spacing, and position in the bank shows little difference; however, figures 10 and 11 indicate that, at $s/a = 1$ and Reynolds number of 100,000, the pressure drop per row for tube B is only three-fourths of the pressure drop per row for tube A.

At Reynolds numbers below 60,000, the bank of tubes B was more sensitive to change in Reynolds number than at higher Reynolds numbers because laminar flow was maintained deeper into the bank than for round or elliptical tubes. Figure 12 corroborates this conclusion, the curve for the first $1\frac{1}{2}$ rows having a typical laminar-flow slope and the curve for the last $1\frac{1}{2}$ rows being typical for turbulent flow. The over-all pressure drop is determined experimentally from duct pressures ahead of and behind the bank.

The nonuniform static-pressure distribution over the surface of tube A is in sharp contrast to the essentially uniform static-pressure distribution on tube C. (See figs. 13 and 14.)

<u>Spacing and friction</u>.- The total-drag coefficient for a single A tube is shown in figure 15. At large spacings the form drag of an elliptical tube in a bank approaches that of an elliptical tube in a free stream; at small spacings, friction enters as the important factor. Because, as far as friction is concerned, using smaller spacings is equivalent to decreasing the hydraulic diameter of the passageway, the friction factor increases as the spacing decreases for the same air velocity in the bank. The net effect is to make the frictional drag a greater percentage of the total drag as the spacing decreases.

In figures 16 to 20 the friction factor for flow through the banks of streamline tubes as a function of Reynolds number is shown at four different spacings of the tubes, and the friction factor for flow in smooth pipes as a function of Reynolds number between 5000 and 200,000 (reference 2) is shown as a dashed line. Within the experimental error, the rate of variation of the friction factor with respect to changing Reynolds number is the same for all spacings used and for all streamline tubes.

For close spacings of the tubes A, the experimental points fall quite precisely on the theoretical line for flow through smooth tubes. This agreement is not surprising because the fluid is considered as flowing in a passage of uniform width, the sides of which are either smooth tube wall or stagnation space.

Behind tubes B, C, and D the low-velocity wake is much narrower than behind round or elliptical tubes and therefore attains higher velocities before reaching the next tube downstream.

The concept of flow bounded by a tube wall as one boundary and a low-velocity wake as the other boundary, which works well in the case of round or elliptical tubes, is not strictly representative of conditions in a bank with small wake (tubes B, C, and D). This concept is applied to all tubes tested, however, for the sake of uniformity and ease of comparison even though it sometimes leads to results which indicate, on superficial examination, that the friction factor of flow through banks of tubes B, C, and D is lower than for flow through smooth tubes.

Hot-wire cooling surveys.- Hot-wire data on air flow near the surface of tubes A (fig. 21) shows that breakaway occurs at about 1/4 to 1/2 inch from the rear of the tube. Tubes B near the front of the bank (fig. 22) have cooling qualities considerably different from tubes A. Deep in the bank of tubes B, however, the flow follows the contours of the tube more closely (fig. 23) and gives a cooling effect much like that for the elliptical tube. Values of $I^2 - I_0^2$ in figure 23 are low because a heavier hot wire is used.

Air flow about tubes C (fig. 24) does not reach as high values as for the thicker tubes, but there is no appreciable drop-off at the ends. Tubes D (fig. 25) show an air-flow distribution similar to that of tubes B but with the high velocity maintained over nearly the full width of the tube.

CONCLUDING REMARKS

The experimental methods of reference 1, used with various streamline tubes, show that, when the tubes are of such shape and are arranged in such manner as to allow the fluid to flow across the bank in well-defined streams, there is little pressure drop due to breakaway from the tube surface and the formation of vortices. When streamline tubes are used instead of round tubes, the flow follows the tube wall closely; as a result, a larger percentage of the total wall surface is in contact with moving fluid and therefore is effective in cooling.

The concept of air flowing through a uniform passage between the tube wall and a sheet of low-velocity air has been further justified for tubes with high form drag.

A streamline tube of the type herein called tube C appears to be best suited for use in banks, on the basis of heat transfer and the consistently low friction factor for all spacings used. Manufacturing methods have not been considered in choosing tube C for use in banks.

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2. McAdams, William H.: Heat Transmission. McGraw-Hill Book Co., Inc., 1933, p. 111.

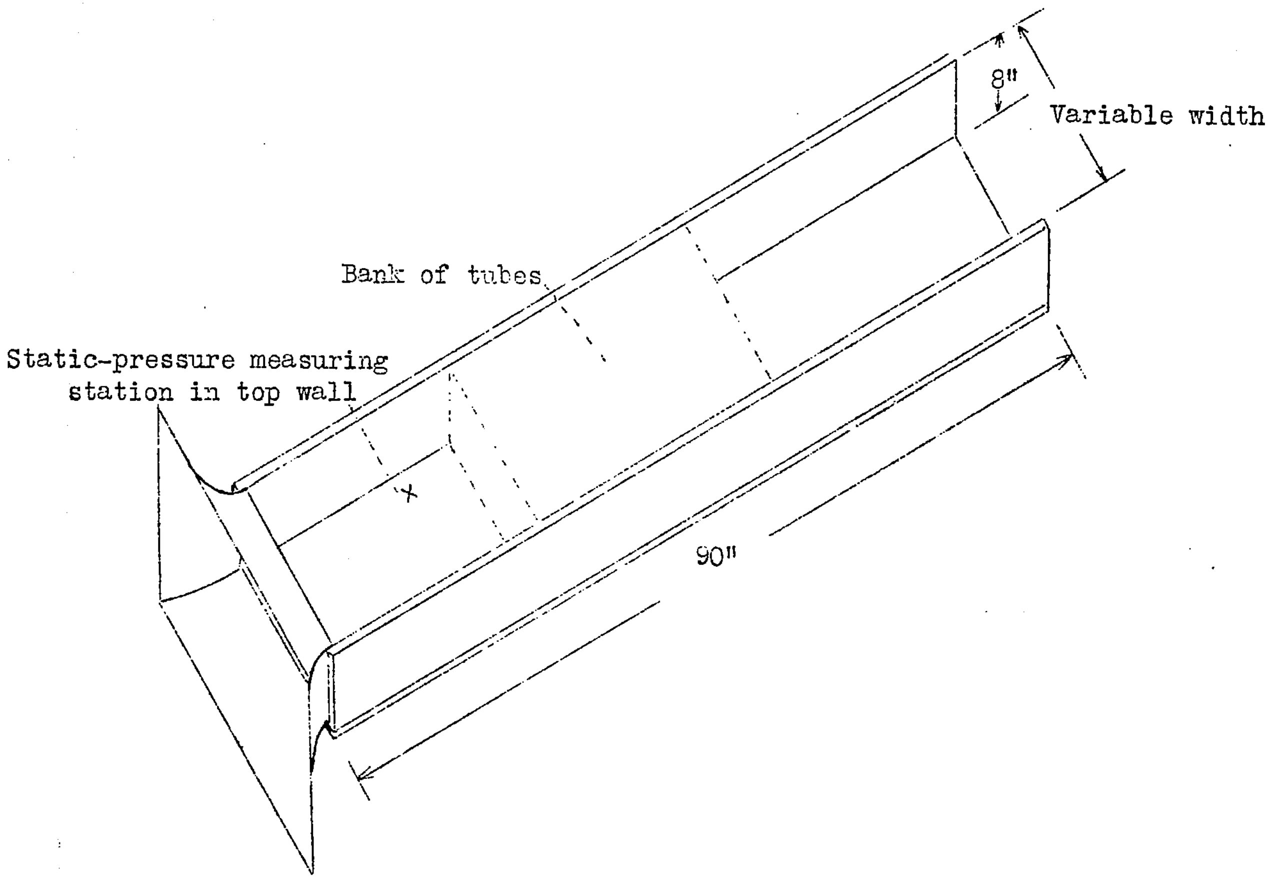


Figure 1.- Experimental duct showing position of bank of tubes. From reference 1.

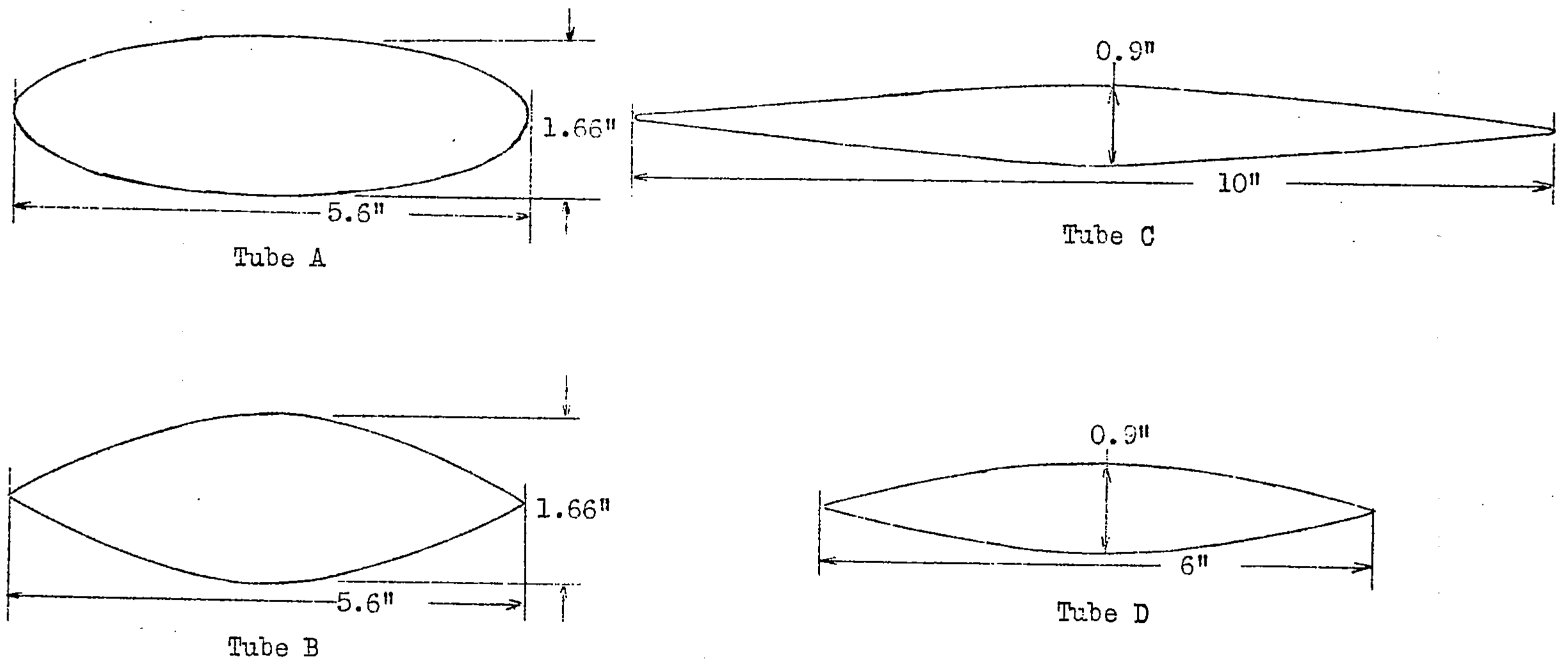


Figure 2.- Cross sections of tubes.

Tube bank arrangement.

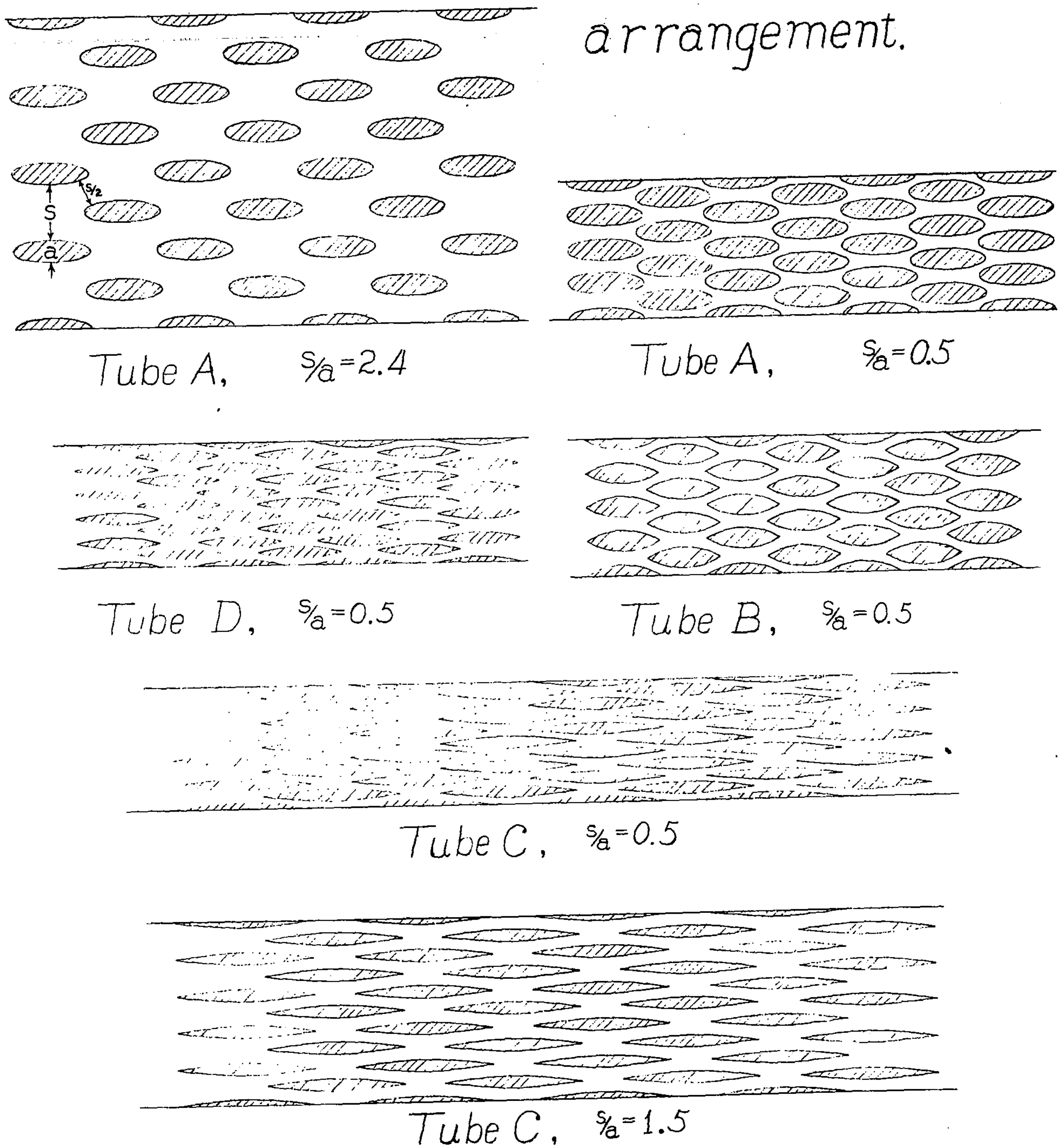


Figure 3

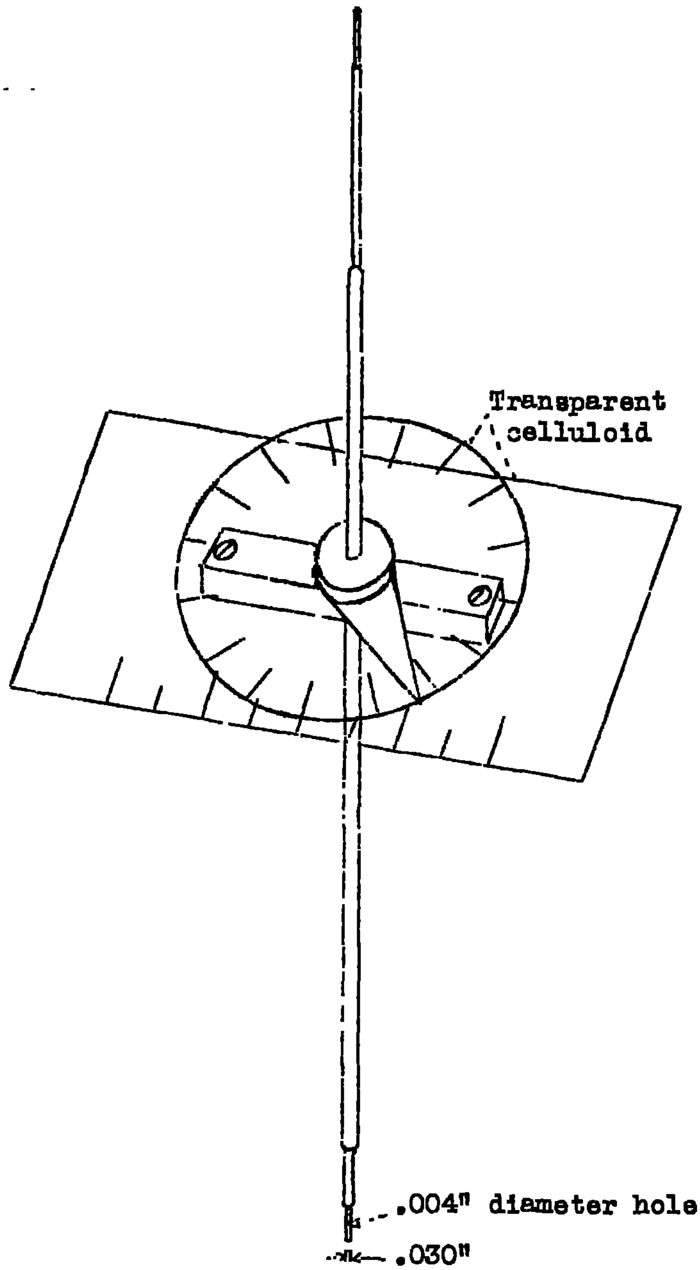


Figure 4.- Apparatus for pressure surveys inside tube bank. From ref. 1.

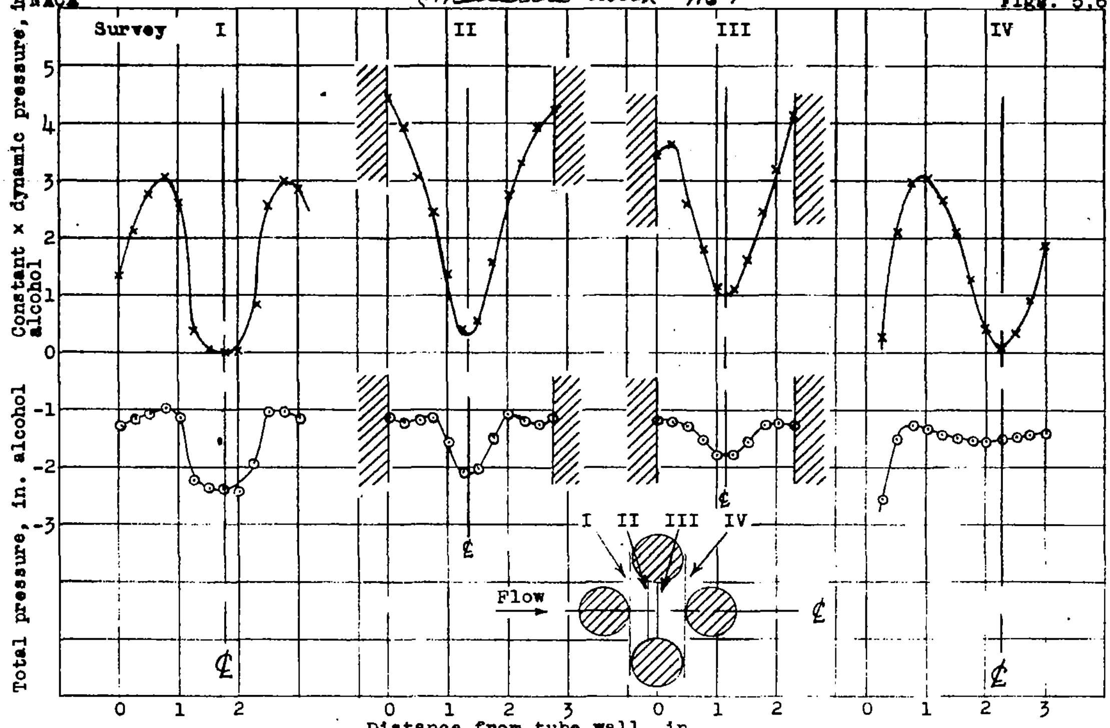


Figure 5.- Total-pressure and dynamic-pressure surveys in bank of circular tubes at $\rho V_t D_t/\mu$ = 82,000. From reference 1.

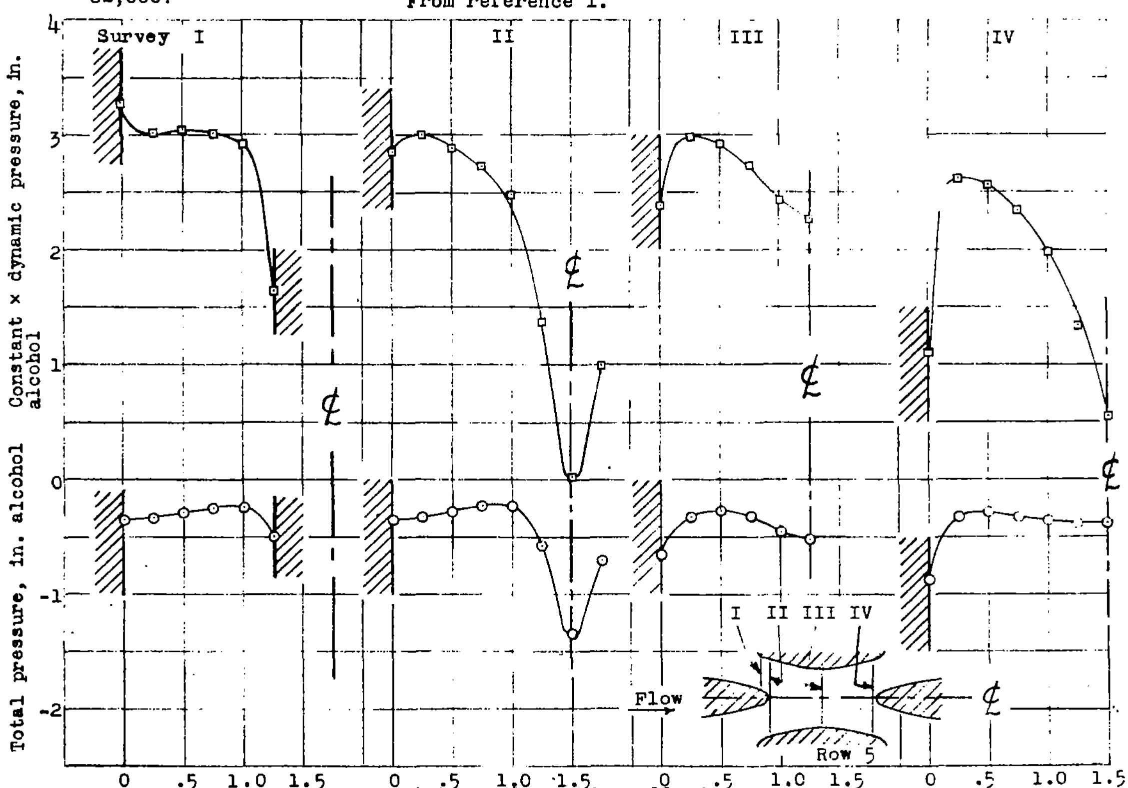


Figure 6.- Total-pressure and dynamic-pressure surveys in bank of elliptical tubes A. s, 0.2075 feet (2.49 in.); $\frac{\rho V_t a}{\mu}$, 55,000.

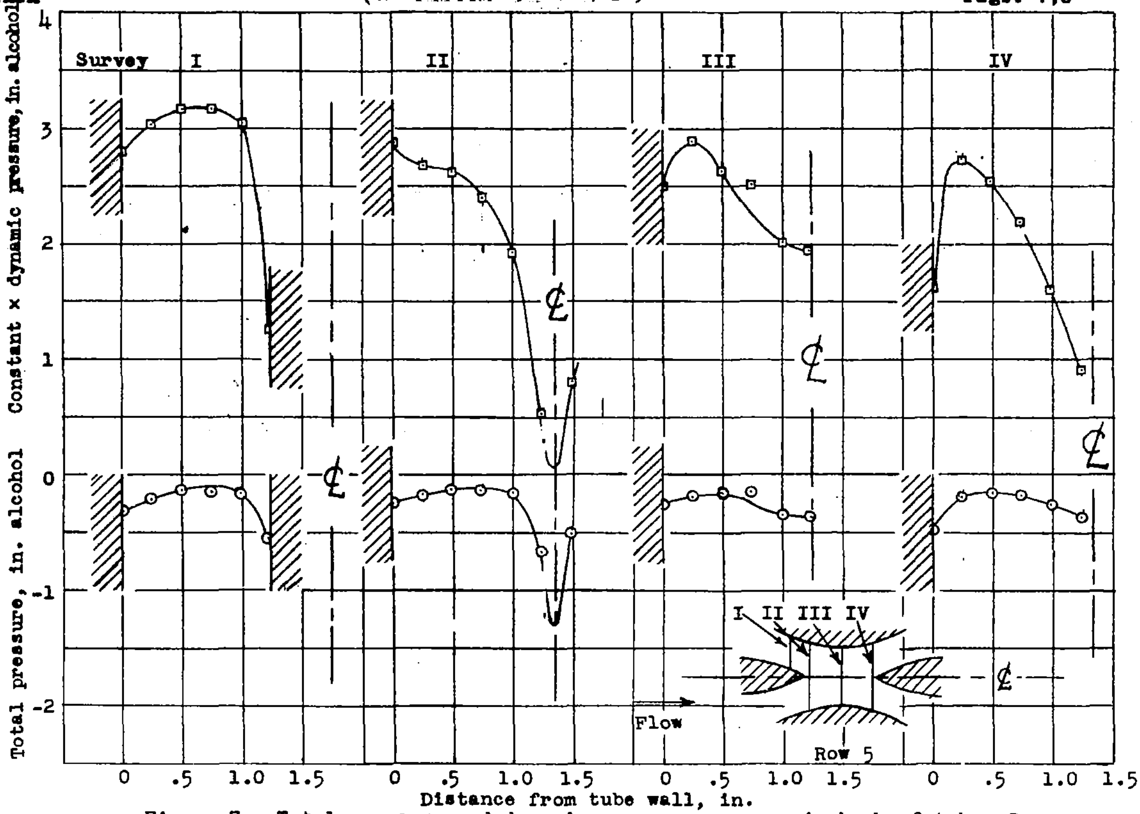
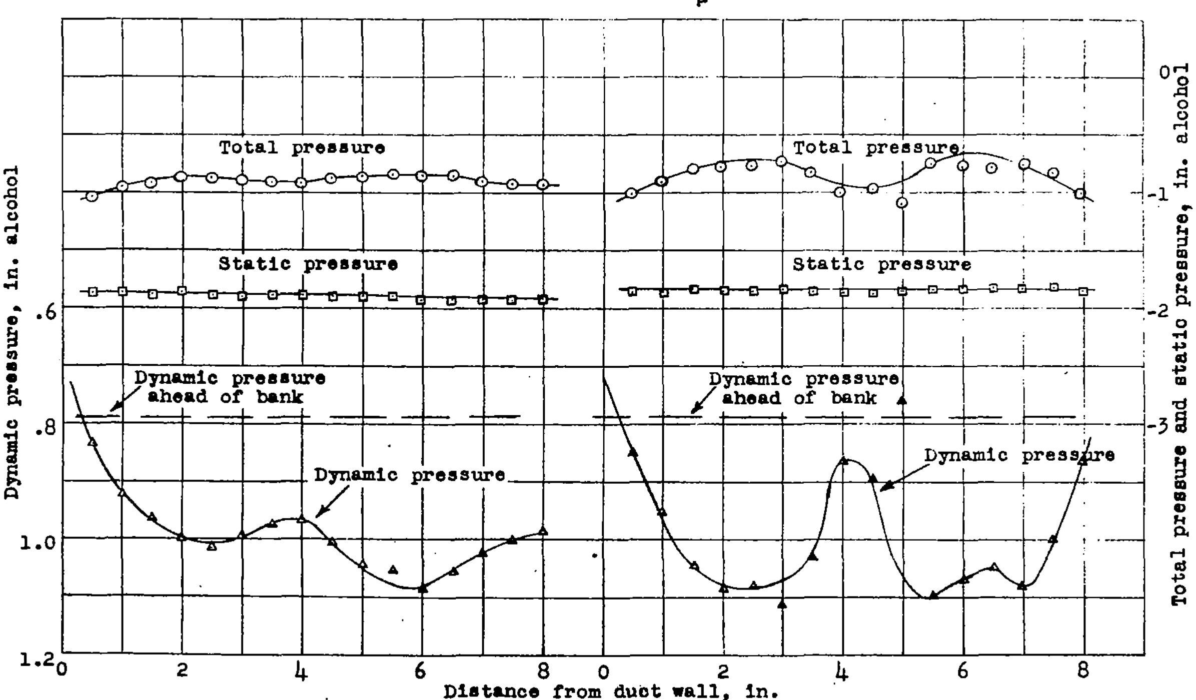


Figure 7.- Total-pressure and dynamic-pressure surveys in bank of tubes B. s, 0.2075 feet (2.49 in.); $\frac{\rho V t^a}{\mu}$, 58,000.



(a) 23 inches behind bank. (b) 8 inches behind bank.

Figure 8.- Survey behind bank of elliptical tubes A. s, 0.2075 feet (2.49 in.); D_t, 0.1875 feet (2.25 in.).

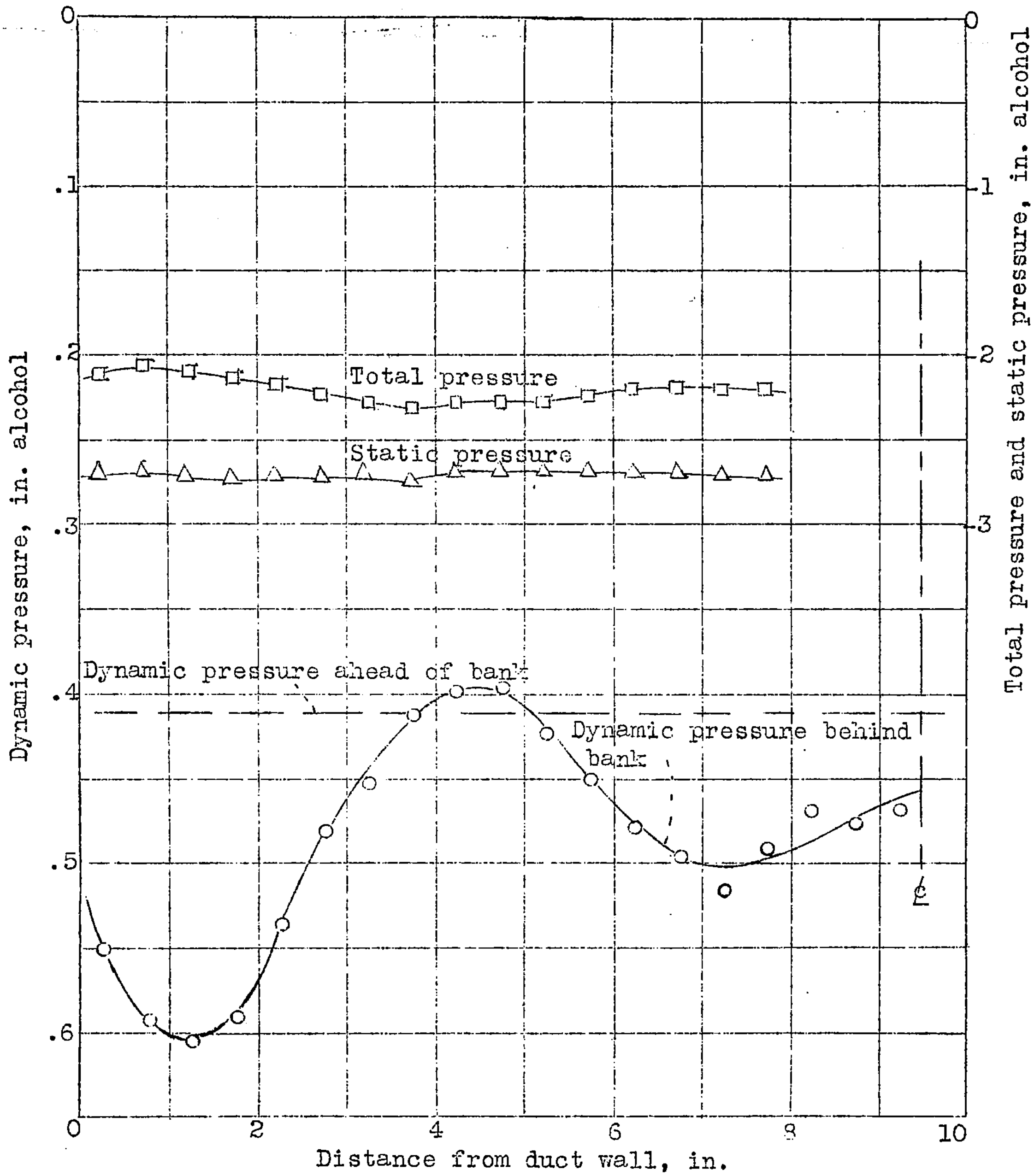


Figure 9.- Survey 12 inches behind bank of circular tubes. s, 0.2075 feet (2.49in.); D_t, 0.1875 feet (2.25 in.). From reference 1.

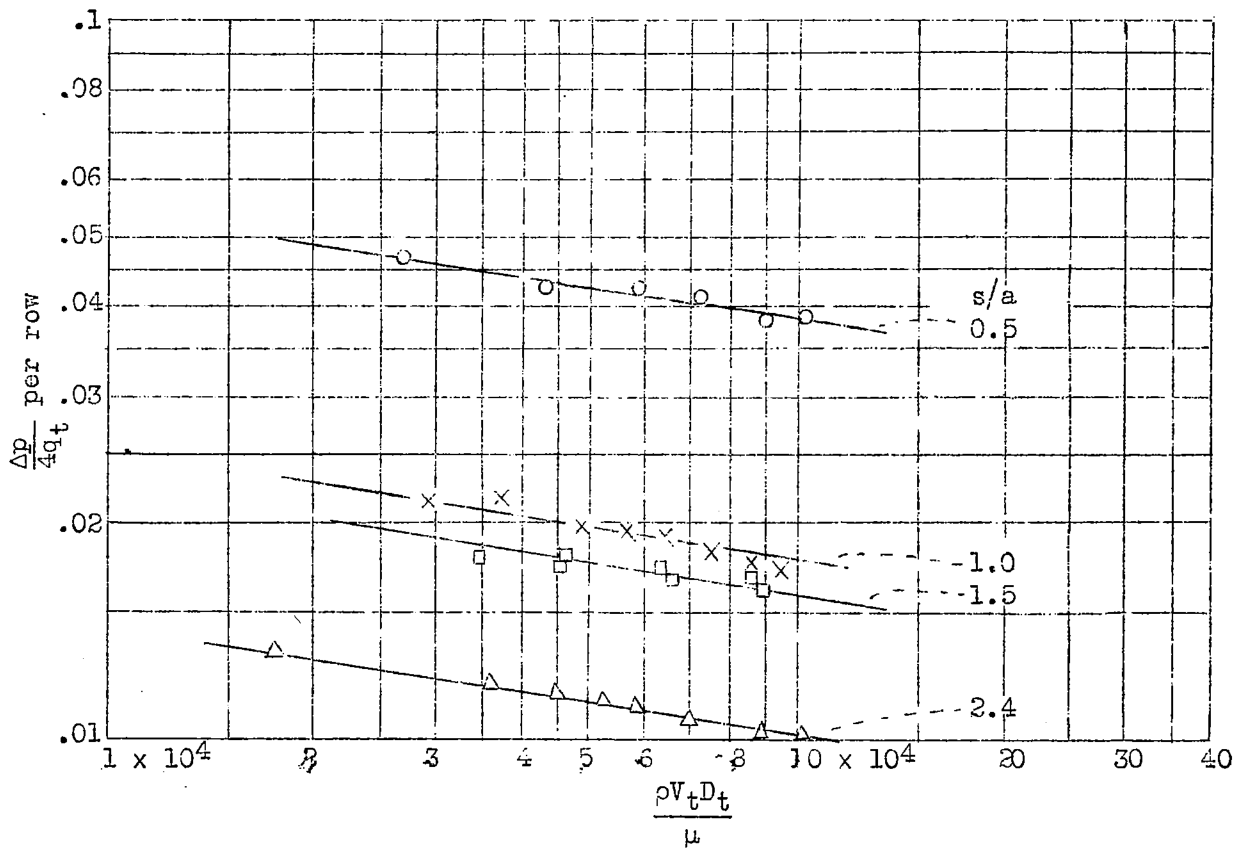


Figure 10.- Pressure-drop data for bank of elliptical tubes A.

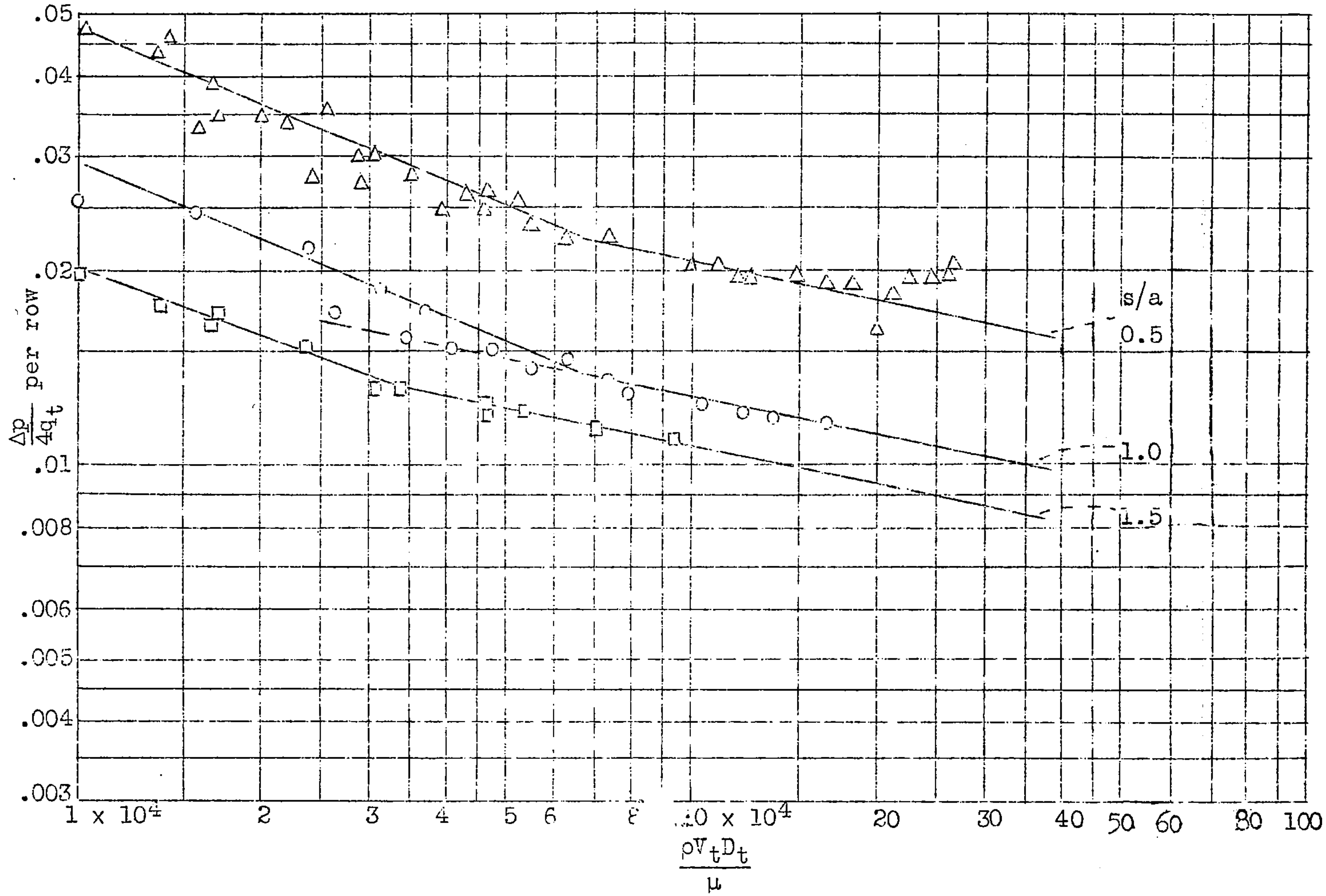


Figure 11.- Pressure-drop data for bank of tubes B.

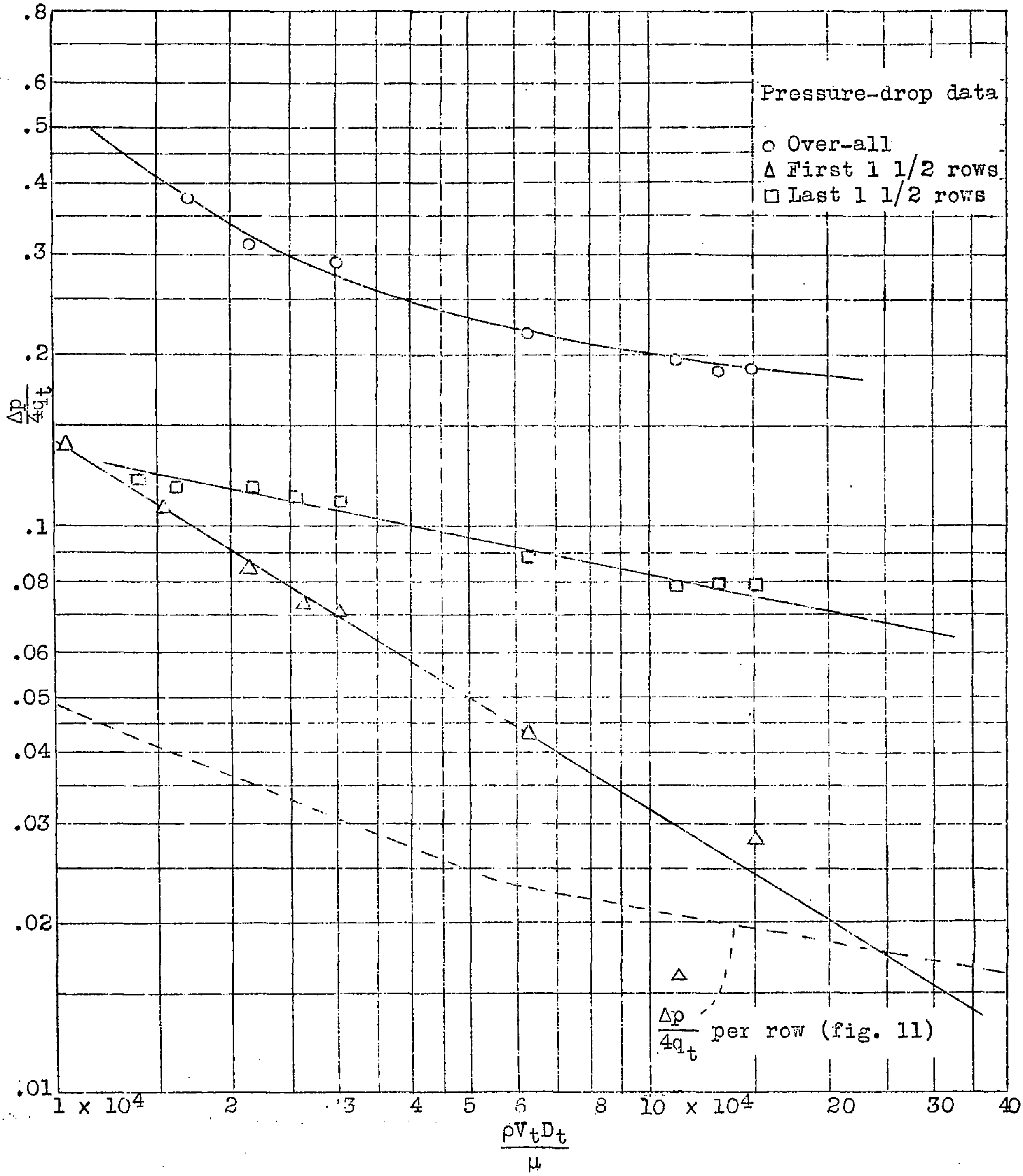
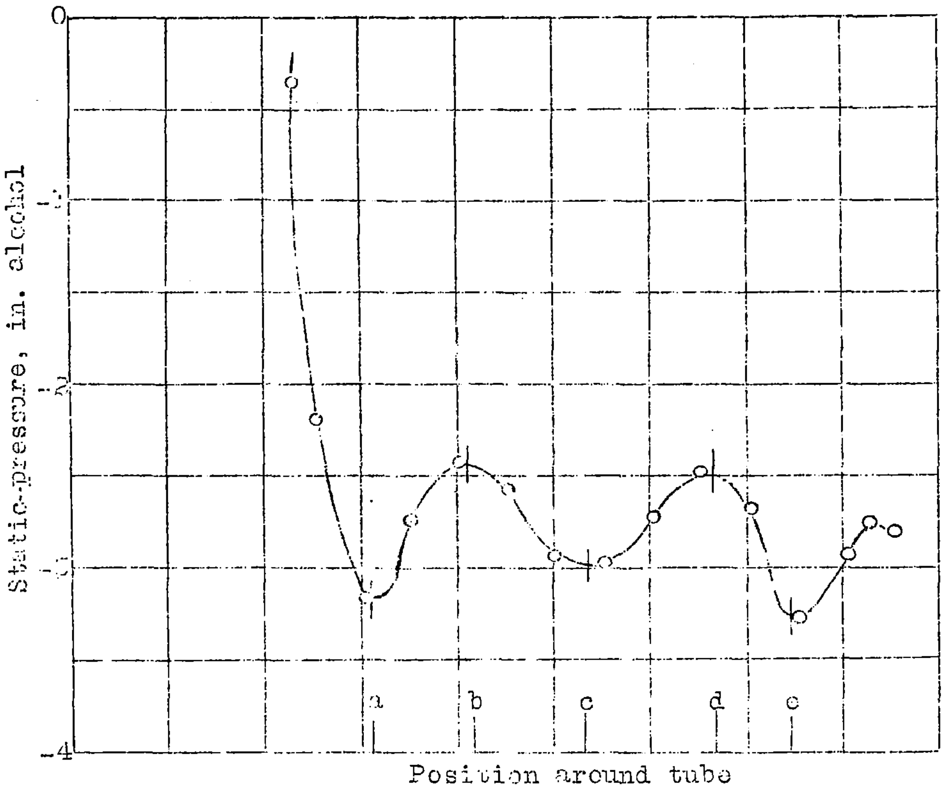


Figure 12.- Analysis of entrance effect for bank of tubes B. s, 0.069 feet (0.83 in.); s/a, 0.5.



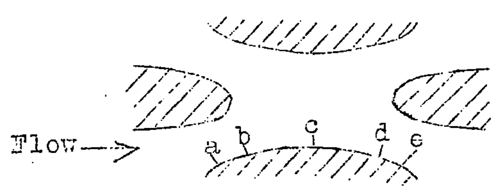


Figure 13.- Static-pressure distribution over surface of elliptical tube A in row 3.

$$\frac{\rho V_t D_t}{\mu}, \ 161,000$$

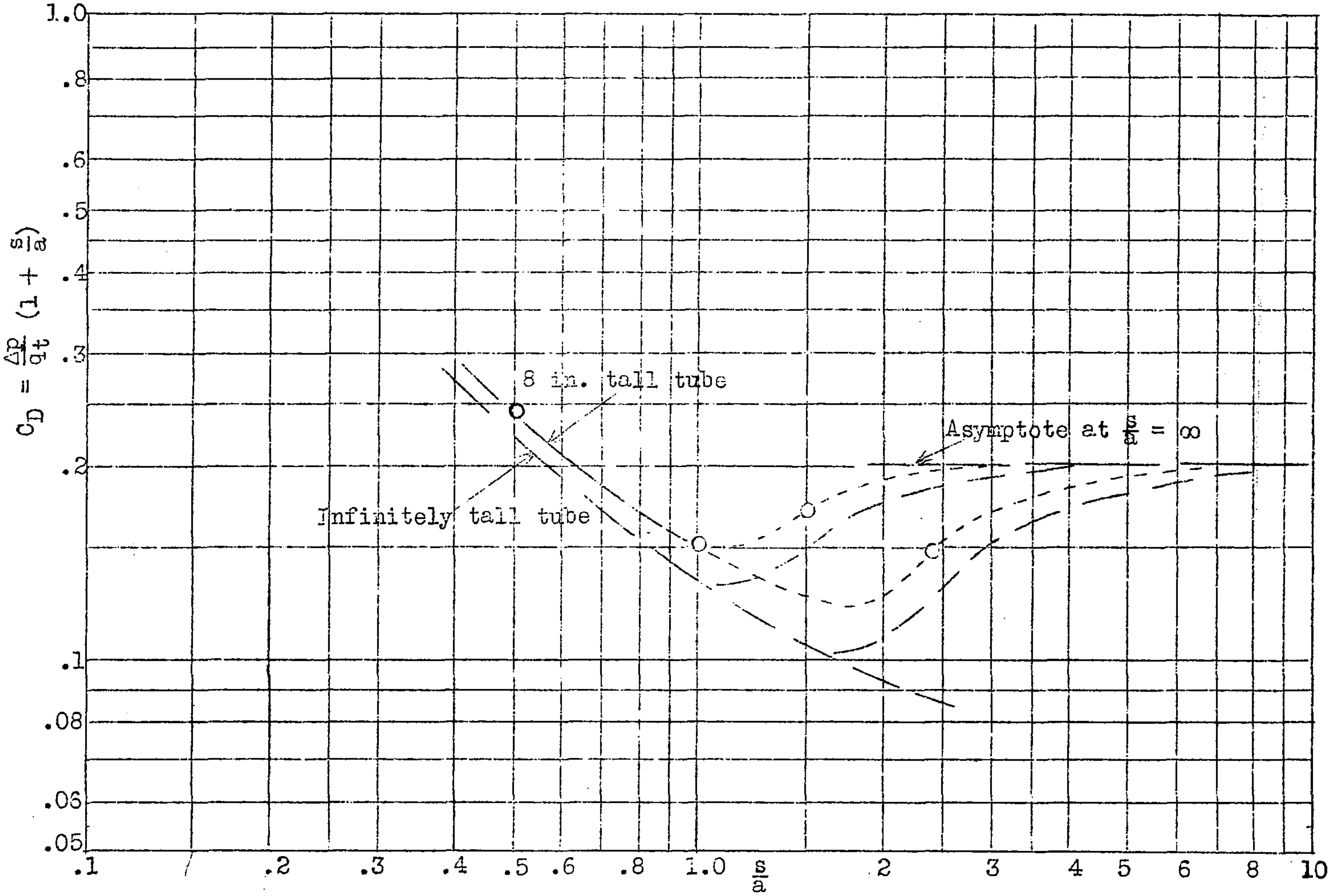


Figure 15.- Total-drag coefficient of tube A as a function of spacing. $\frac{\rho V_t a}{\mu}$, 50,000.

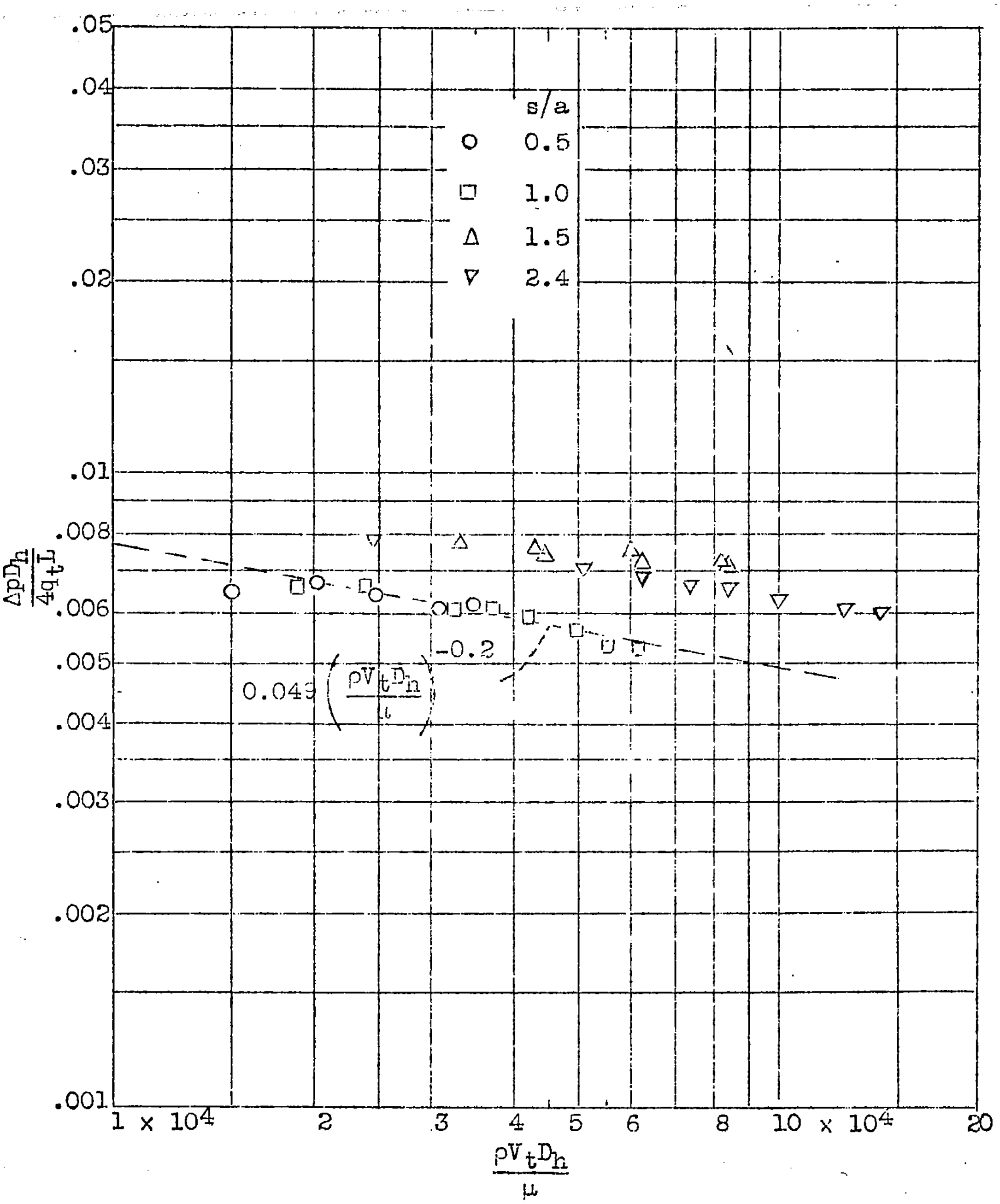


Figure 16.- Friction data for bank of tubes A.

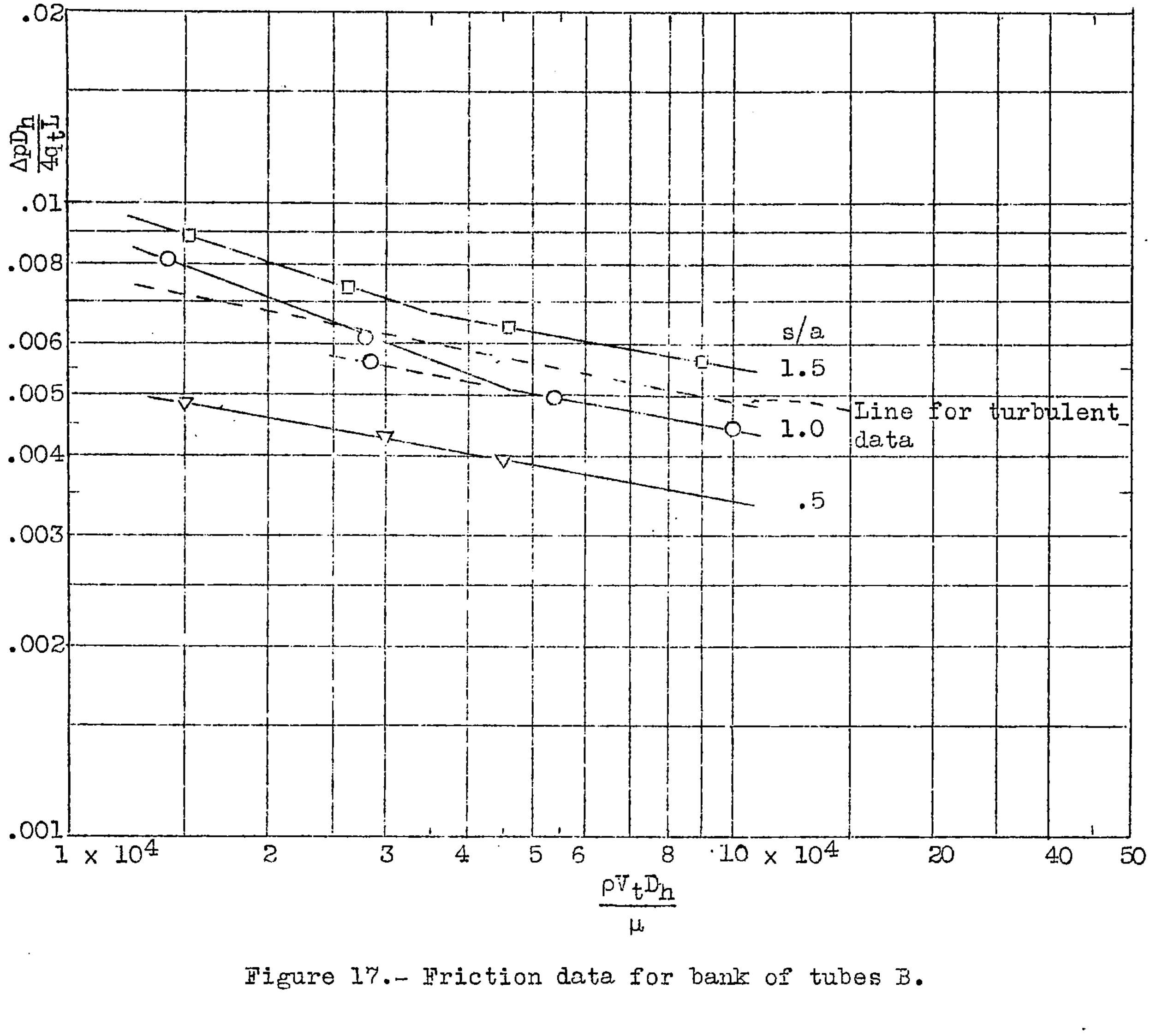


Figure 17.- Friction data for bank of tubes B.

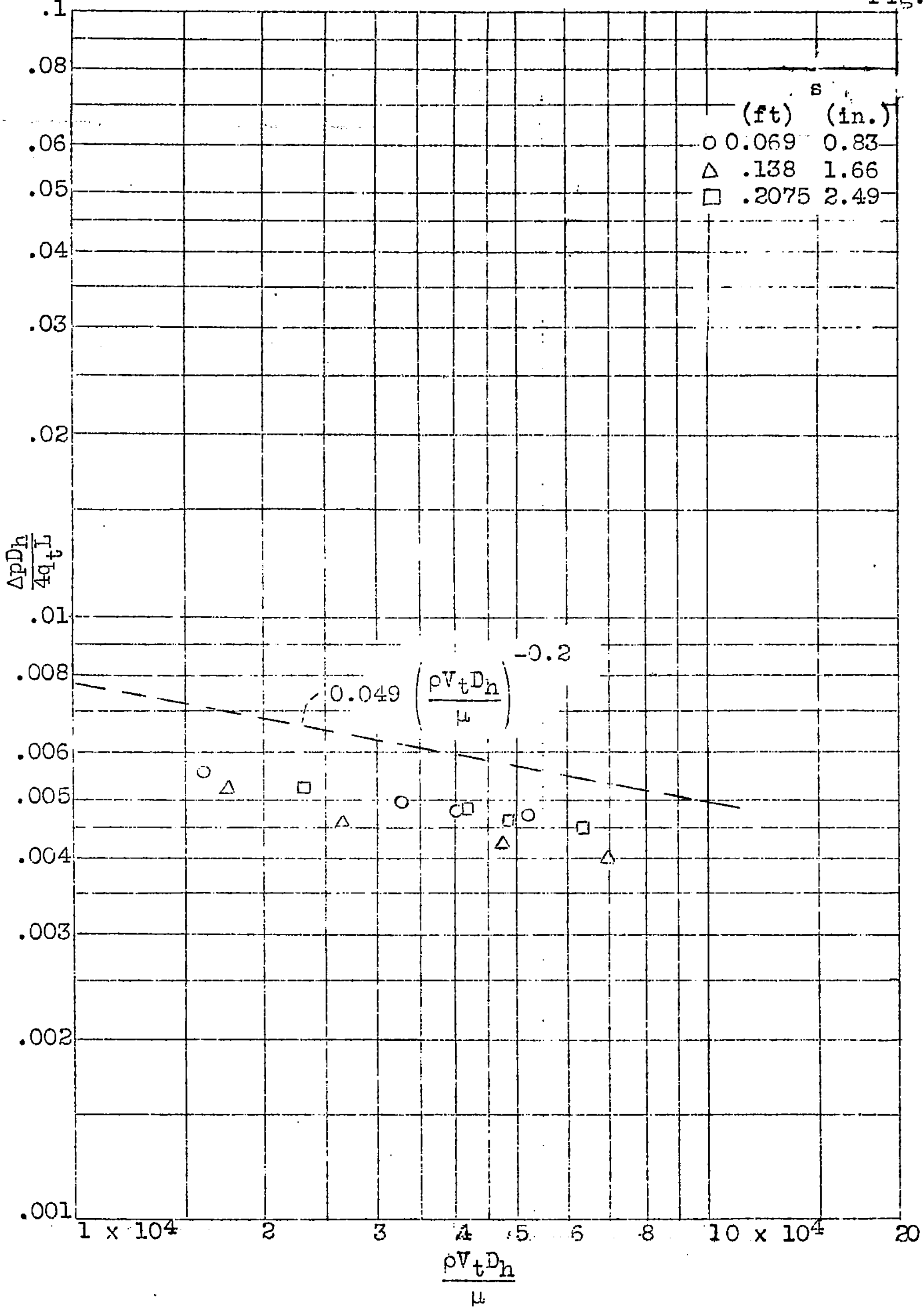


Figure 18.- Friction data for bank of tubes C.

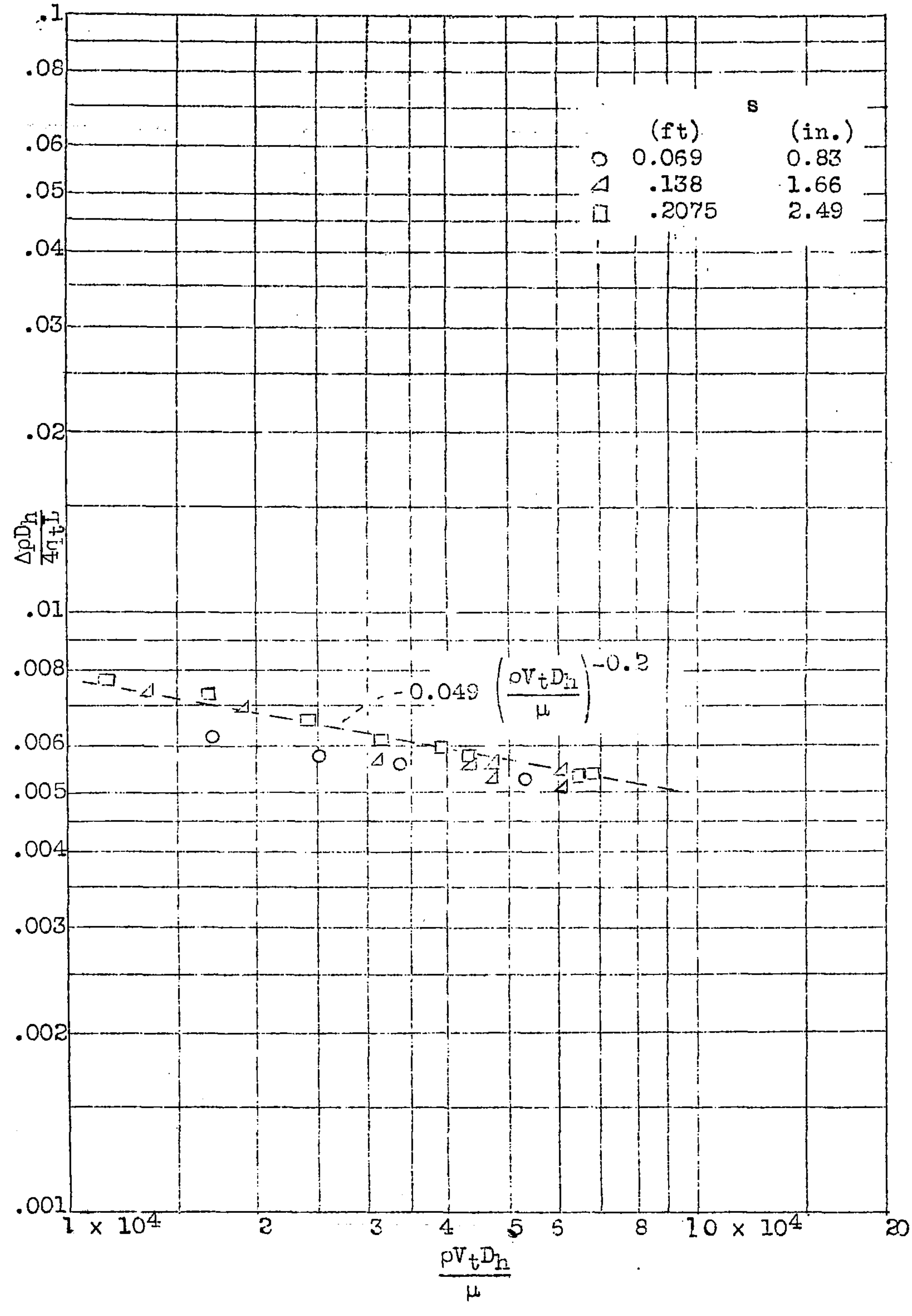


Figure 19.- Friction data for bank of tubes D.

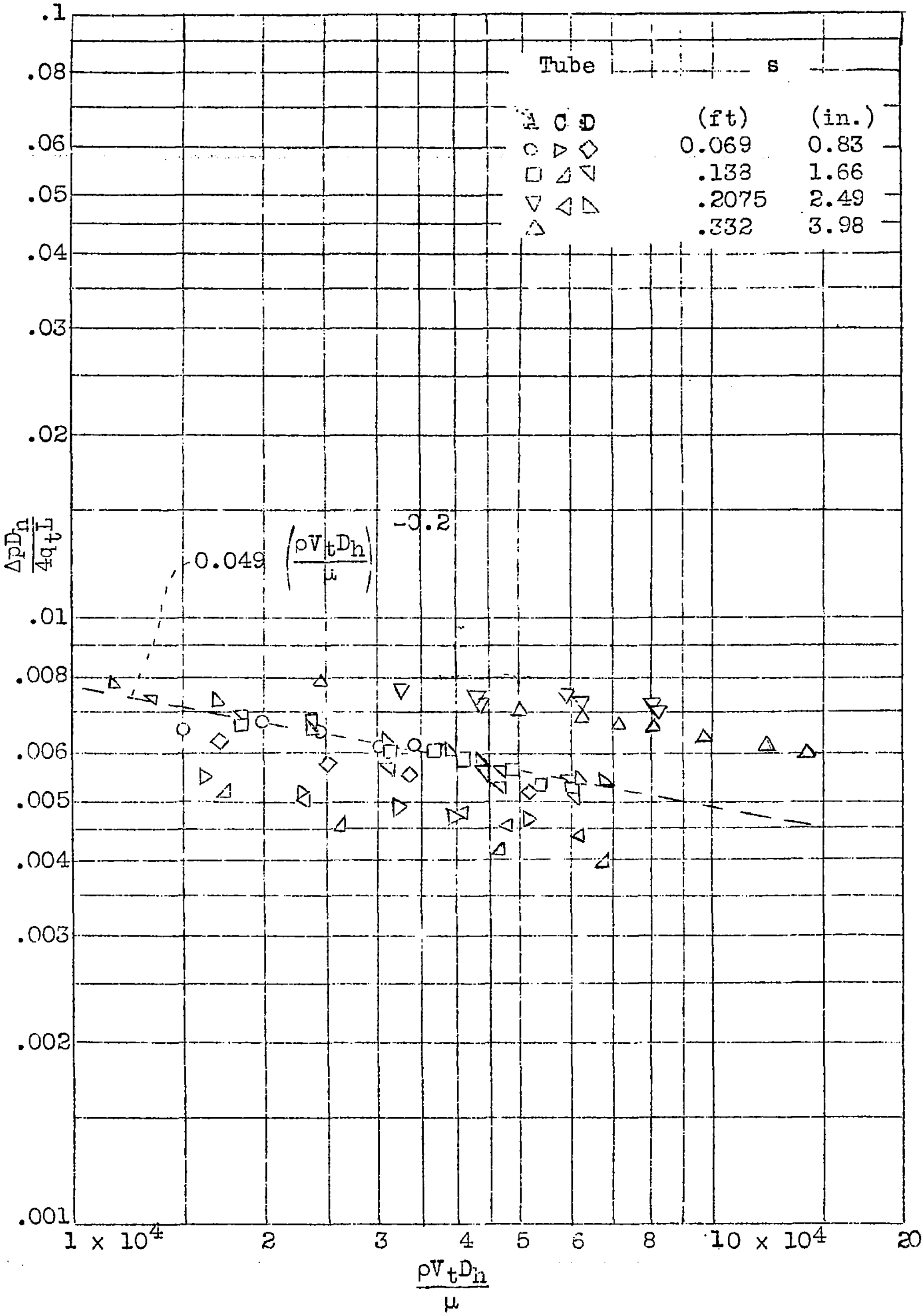


Figure 20.- Summary of friction data.

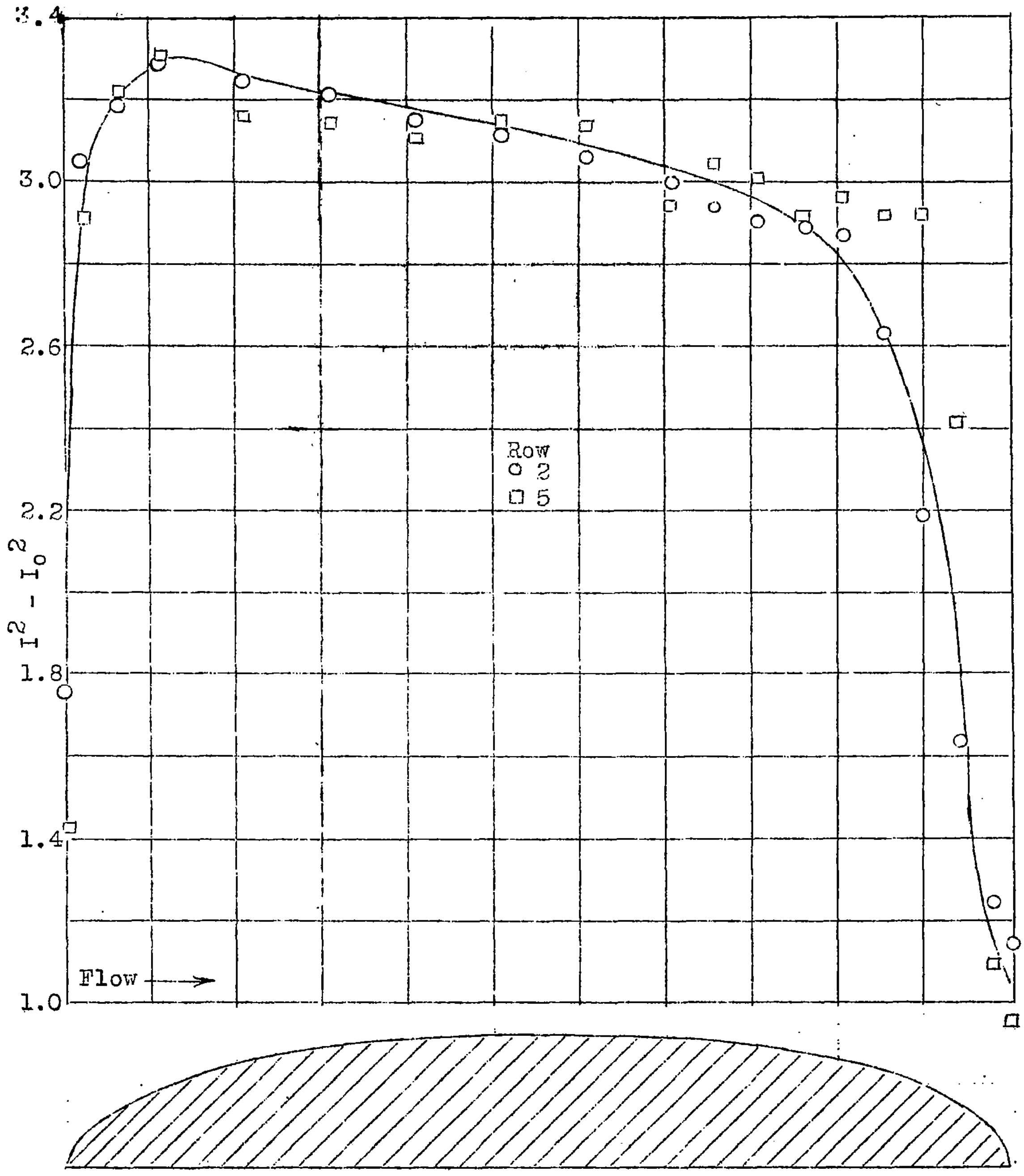


Figure 21.- Hot-wire survey of surface of elliptical tube A. s, 0.2075 feet (2.49 in.); $\frac{\rho V_t a}{\mu}$, 54,600.

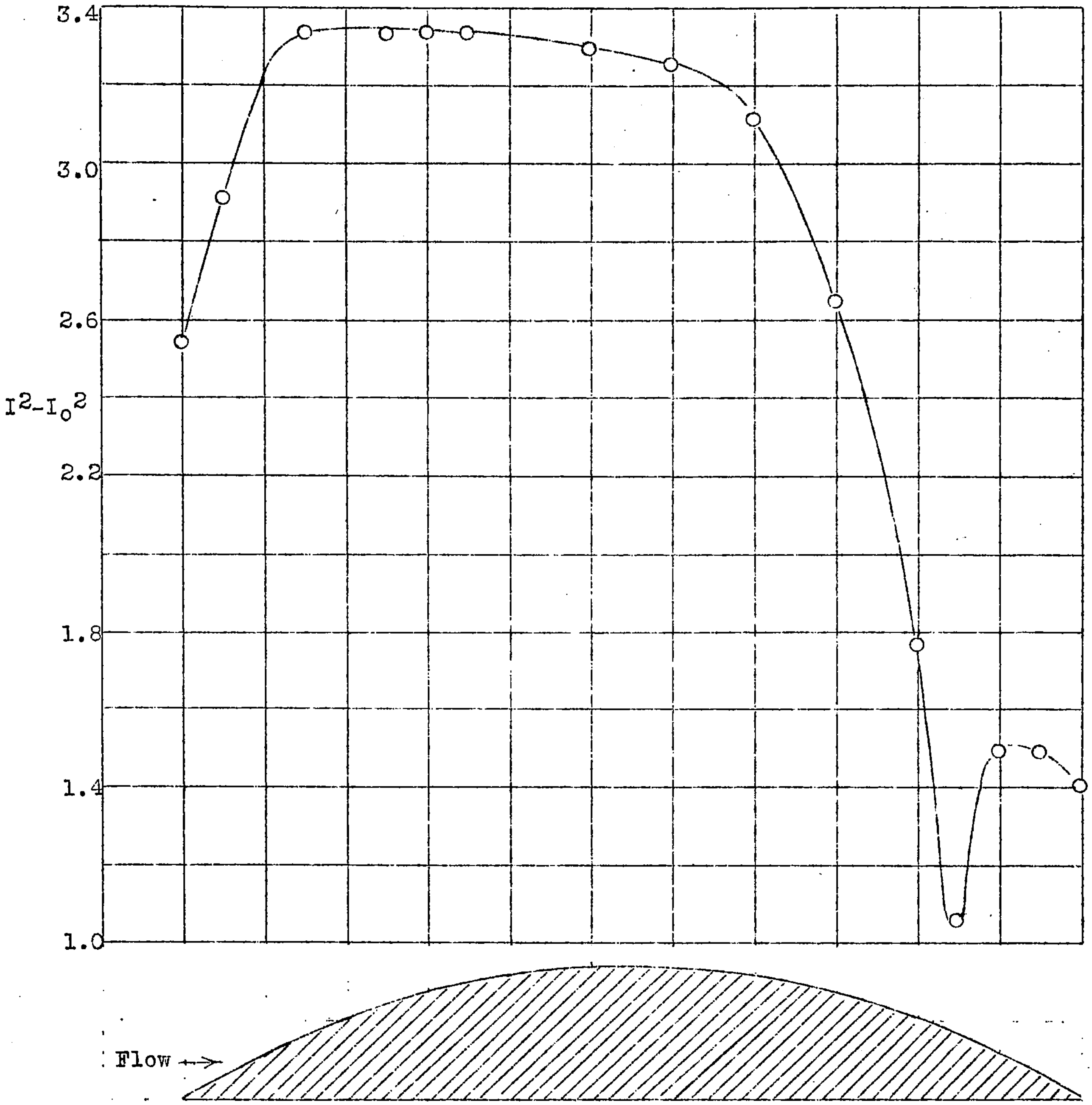


Figure 22.- Hot-wire survey of surface of tube in row 2 of bank of tubes B. s, 0.2075 feet (2.49 in.); $\frac{\rho V_t a}{\mu}$, 58,000.

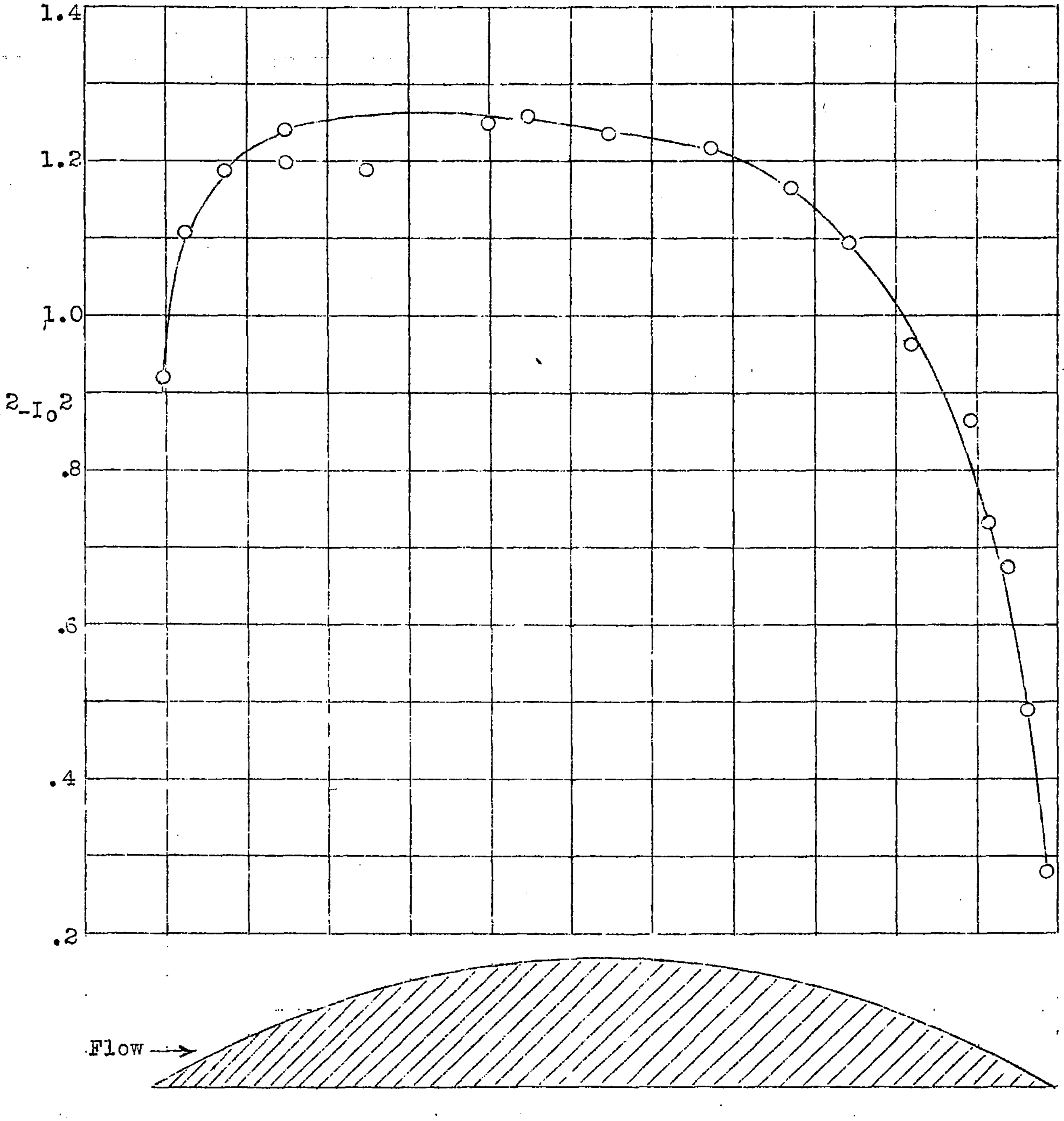


Figure 23.- Hot-wire survey of surface of tube in row 5 of bank of tubes B. s, 0.2075 feet (2.49 in.); $\frac{\rho V_t a}{\mu}$, 58,000.

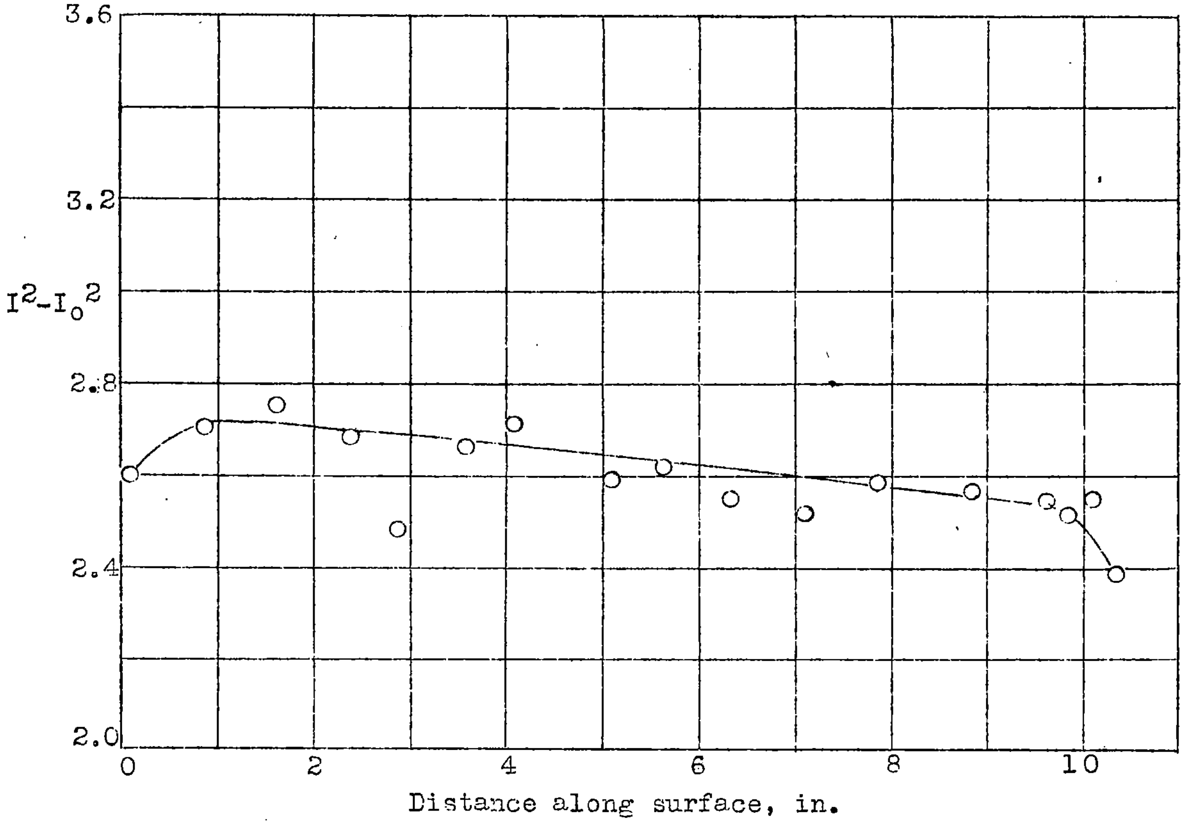


Figure 24.- Hot-wire survey of surface of tube in row 5 of bank of tubes C. s, 0.2075 feet (2.49 in.); $\frac{\rho V_t a}{\mu}$, 27,400.

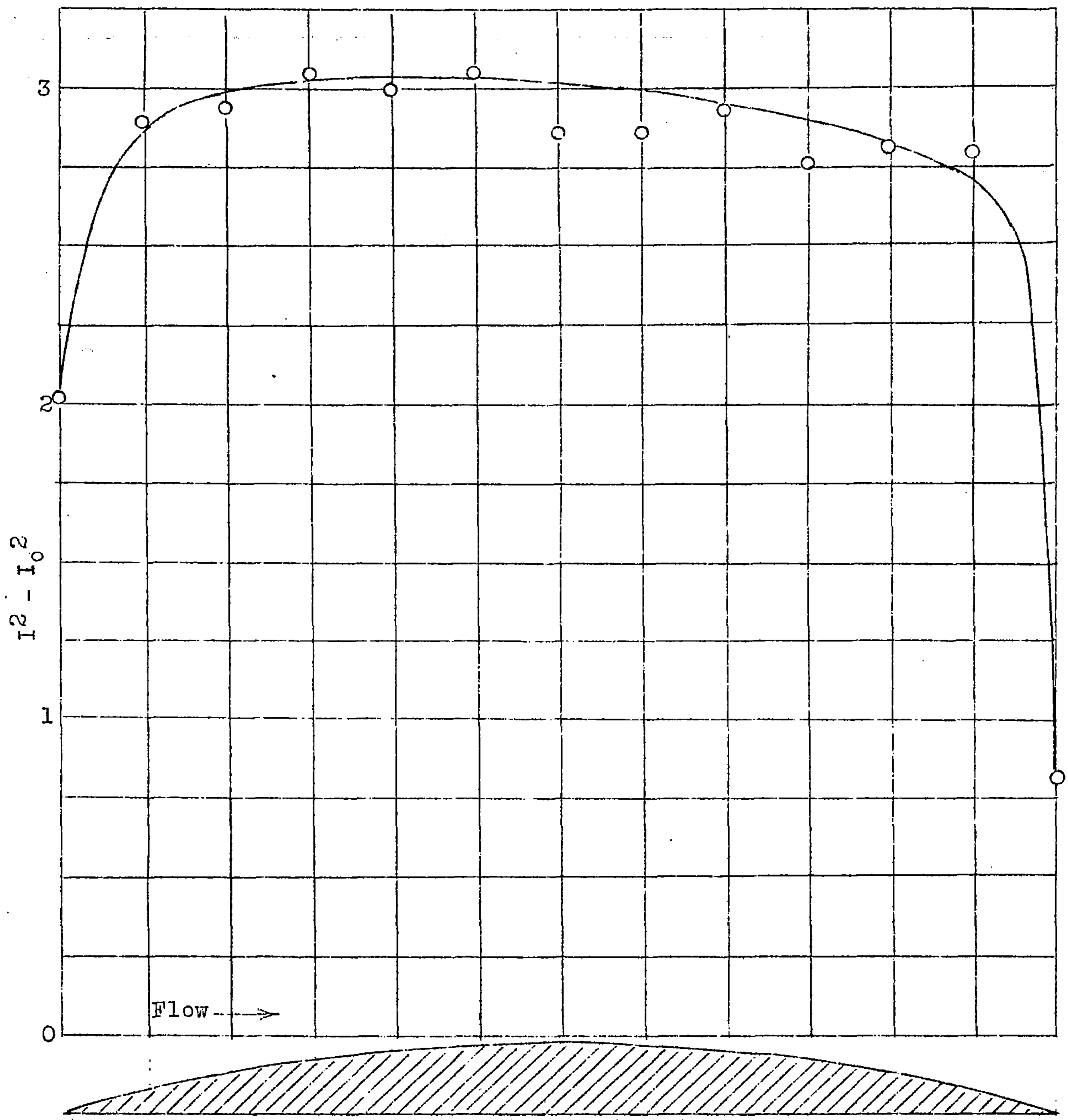


Figure 25.- Hot-wire survey of surface of tube in row 5 of bank of tubes D. s, 0.2075 feet (2.49 in.); $\frac{\rho V_t a}{\mu}$, 34,000.

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