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EXPERIMENTAL INVESTIGATION OF ENTRANCE-REGION HEAT-TRANSFER COEFFICIENTS

By Upshur T. Joyner

Langley Memorial Aeronautical Laboratory
Langley Field, Va.

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SUMMARY

Experimental results of tests made at the Langley Memorial Aeronautical Laboratory are presented to show how heat-transfer coefficients can be increased by a method utilizing the high rate of heat transfer known to exist on any heat-transfer surface in the region adjacent to the edge on which the cooling or heating fluid impinges.

The results show that, for the same pressure drop, the average surface heat-transfer coefficient can be increased 50 to 100 percent when a cooling surface having a length of four inches in the direction of fluid flow is cut to form twenty fins with a length of 0.2 inch in the direction of fluid flow and the fins are sharpened and staggered in the air stream. The percentage of increase in the surface heat-transfer coefficient obtained as a result of shortening the length of the cooling surface varies with the pressure drop of the cooling fluid in passing the surface, the increase being largest when small pressure drop is used and smallest when high pressure drop is used.

INTRODUCTION

The problem of designing heat-transfer surfaces for use in airplanes is complicated by the necessity for low weight and small volume of the cooling unit. There are a number of ways of achieving low weight and small volume by increasing the local heat-transfer coefficient and consequently decreasing the cooling surface required. For example, the velocity V of the fluids passing over the heat-transfer surfaces may be made very high. Because the heat-transfer coefficient varies approximately as $V^{0.8}$ and the pressure drop varies approximately as $V^{2.0}$, this solution leads to excessive pressure-drop requirements and large power expenditures for pumping fluids through the heat exchanger. This method is, nevertheless, invariably used where heat transfer in an existing unit is increased. Another way of increasing the heat-transfer coefficient is to insert turbulence producers in the passages of the heat exchanger. Colburn (reference 1) has reported that such turbulence producers are beneficial at low-velocity flow through the exchanger but that, at higher speeds, the ratio of heat transferred to power required becomes smaller than in a plain passage. This method is not suitable for use in aircraft heat exchangers, where the velocity of flow of fluid through the heat exchanger is always high.

The local heat-transfer coefficient may be increased by decreasing the hydraulic radius of the flow passages through the heat exchanger. Manufacturing difficulties increase greatly, however, as passageways are made smaller and some system of further improving heat transfer for any given practicable size of passageway is of interest. Such a system designated the multiple-entrance system is reported herein. Nusselt (reference 2) showed many years ago how the heat-transfer coefficient varies with passage length, but little practical use has evidently been made of his analysis. Norris and Spofford corroborated the theoretical analysis of Nusselt in an extensive investigation presented in reference 3 and the tests presented of the multiple-entrance system further confirm this theory.

SYMBOLS

f friction factor $\left(\frac{\Delta p}{q}\,\frac{D}{4x}\right)$

Nu Nusselt number $\left(\frac{hD}{k}\right)$

R Reynolds number $\left(\frac{\rho VD}{\mu}\right)$

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

h surface heat-transfer coefficient, Btu per second per square foot per $^{\circ}$F

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k thermal conductivity of air, Btu per second per square foot per °F per foot

L length of individual heat-transfer passage, feet

q dynamic pressure of air flowing through heat-transfer passage, pounds per square foot $\left(\frac{1}{2}\rho V^2\right)$

V average velocity of air flowing through heat-transfer passage, feet per second

x over-all length of finned surface tested, feet (4 in. in these tests)

Δp over-all pressure drop across finned surface tested, pounds per square foot

μ viscosity of air, slugs per foot per second

ρ mass density of air, slugs per cubic foot

TEST METHODS AND ANALYSIS

All the tests described herein were made on flat, electrically heated finned surfaces. Models were constructed in the NACA physical research laboratory. The spacing between fins on all models was 1/8 inch, the fin thickness was 1/32 inch, and the fin width, 1 inch. Five fin lengths varying from 4 inches, which gave an L/D of 18, to 0.2 inch, which gave an L/D of 0.9, were used. The fins were made of copper. Figure 1 shows the multiple-entrance system with fin lengths of 0.5 inch. In all models except the one with the 4-inch fin length, the succeeding rows of fins were staggered as shown in figure 1.

The total heat transferred, the fin-base temperatures, air temperatures, pressure drop, and quantity of air flow were measured for each of the five fin lengths. From these measurements, local heat-transfer coefficient, Nusselt number, Reynolds number, and friction factor were calculated.

According to Nusselt's analysis (reference 2), the heat-transfer coefficient is very large at the entrance of a heat-transfer passage and decreases with increasing distance down the tube to the final value for fully established flow. The reason for this large heat transfer at entrance is that the temperature gradient at the wall is large near the entrance.

The multiple-entrance system for improving heat transfer takes advantage of the thin boundary layer and the large temperature gradient at the wall and the consequent high rate of heat transfer known to exist in the entrance region of fluid flow over a surface. As the fluid progresses along the surface, the layers next to the surface are retarded by surface friction and a thick boundary layer develops through which all heat transferred must travel by conduction. In established turbulent flow, there is an exchange of fluid and, hence, of heat between the boundary layer and the main fluid stream. This exchange of fluid accounts for the fact that the heat-transfer coefficient is higher in turbulent flow than in laminar flow. In the multiple-entrance system for heat transfer, the heat is transferred, partly by turbulence and partly by conduction, through the boundary layer, and the boundary layer is kept very thin by using short lengths of cooling surface. Inasmuch as each small surface is displaced relative to the one ahead of it, the coolest fluid available is brought in contact with the surface and, consequently, the maximum heat is transferred.

When any arrangement that gives very high heat-transfer coefficients (for example, the multiple-entrance system) is used, the length of path of the fluid flow must be kept reasonably small in order that the fluid may not heat excessively and a large temperature difference may be kept available for cooling. The cooling effectiveness of the surfaces must be kept large and the surfaces should be shaped to allow reasonably undisturbed flow of the air.

RESULTS

The results of the tests are given in figures 2, 3, and 4. In figure 2 is given the variation of Nusselt number with Reynolds number. The results in this figure show the Nusselt number (and heat-transfer coefficient) to be increased as fin lengths are decreased. The large increase in heat transfer shown in figure 2 is accompanied by a corresponding increase in friction factor (fig. 3). Figure 4 is obtained from the same data as figures 2 and 3 but is presented in a form that shows the increase in heat-transfer coefficient to be derived by decreasing the value of L/D while pressure drop is kept constant.

In figure 4, large gains in the local heat-transfer coefficient, of the order of 50 to 100 percent, are seen to be attained by the relatively simple means of reducing the currently used L/D of 20 or more to an L/D of about 1. Uniform temperature over a surface may be obtained by so adjusting the value of L/D at each point that the required value of local heat-transfer coefficient is obtained.

APPLICATIONS

The multiple-entrance heat-transfer arrangement can be advantageously applied in numerous instances. For use in heating or cooling any solid surface by a fluid, such as a cylinder of an air-cooled aircraft engine, the heat-transfer surfaces would be in the form of indirect cooling surfaces, such as fins attached to the outside of the cylinder. The ratio of length to thickness of a cross section of these indirect cooling surfaces should be not less than 5 or 6 and the length of fluid path through the fins should be kept small by providing several places for entrance and exit of the cooling air.

This arrangement may also be applied in a heat-transfer unit of which both the cooled and the cooling fluids are gases. In this case, an arrangement such as that shown in figure 5 could be used. The principal heat-transfer surfaces would be of the indirect cooling type and would be made of thin sheet metal, such as is at present used in this type of construction.

In a heat-transfer unit of which one of the fluids is a liquid and the other a gas, nearly all of the resistance to heat flow is between the gas and the heat-transfer surface when the liquid has high thermal conductivity and low viscosity. In this case, the multiple-entrance arrangement of heat-transfer surfaces would probably be applied only to the passages through which the gas travels, as in the method indicated for the gas to gas heat exchanger. If, however, a liquid, such as oil, has low thermal conductivity and high viscosity and if a considerable part of the over-all thermal resistance is consequently between the liquid and the heat-transfer surface, the proposed arrangement could be advantageously used in all fluid passages.

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CONCLUSION

The results show that the value of the local heat-transfer coefficient for a given hydraulic radius may be increased 50 to 100 percent by use of a large number of short-length staggered surfaces instead of one large surface. It was shown by Nusselt and has been proved by these and other tests that a surface which is to all practical purposes operating as an entrance region will have a thin boundary layer and consequently low resistance to heat transfer.

The multiple-entrance system of heat-transfer surfaces is not suggested as the remedy for all cooling troubles, but it is thought that designers will find many places in which this type of surface can be used to advantage.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va.

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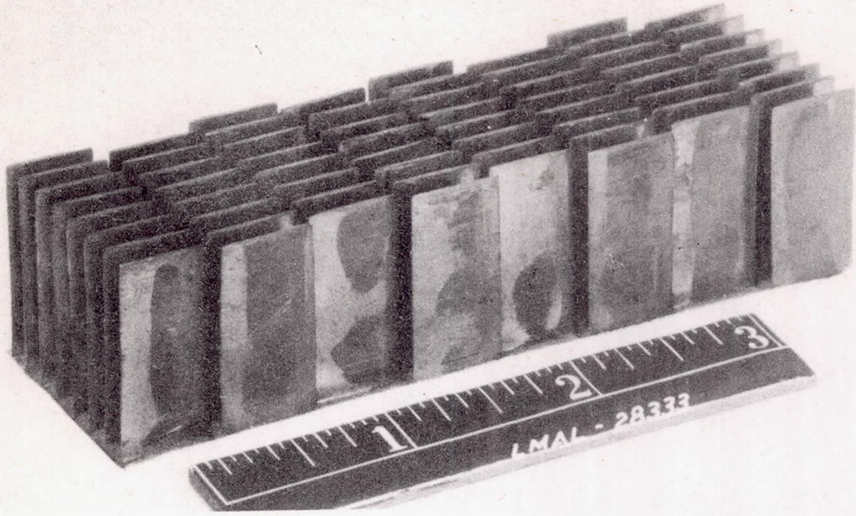

Figure 1.- Multiple-entrance heat-transfer system with 0.5 inch fins.

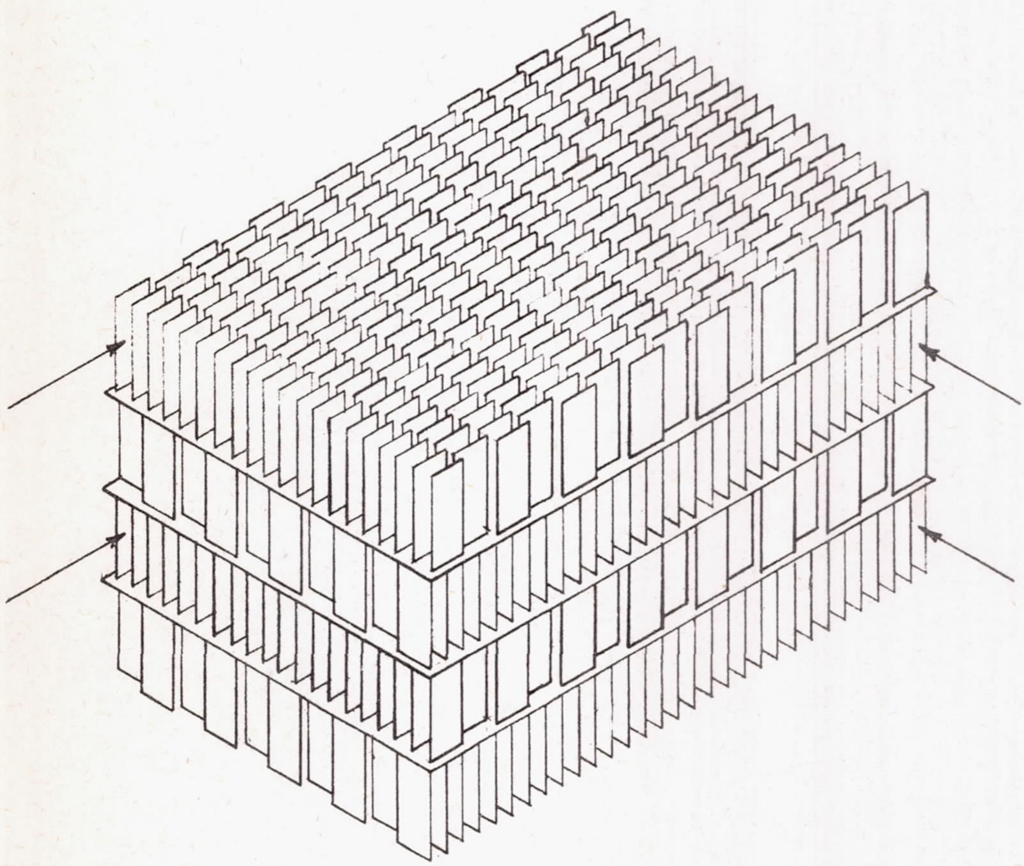

Figure 5.- Method of using staggered heat-transfer surfaces in a cross-flow heat exchanger.

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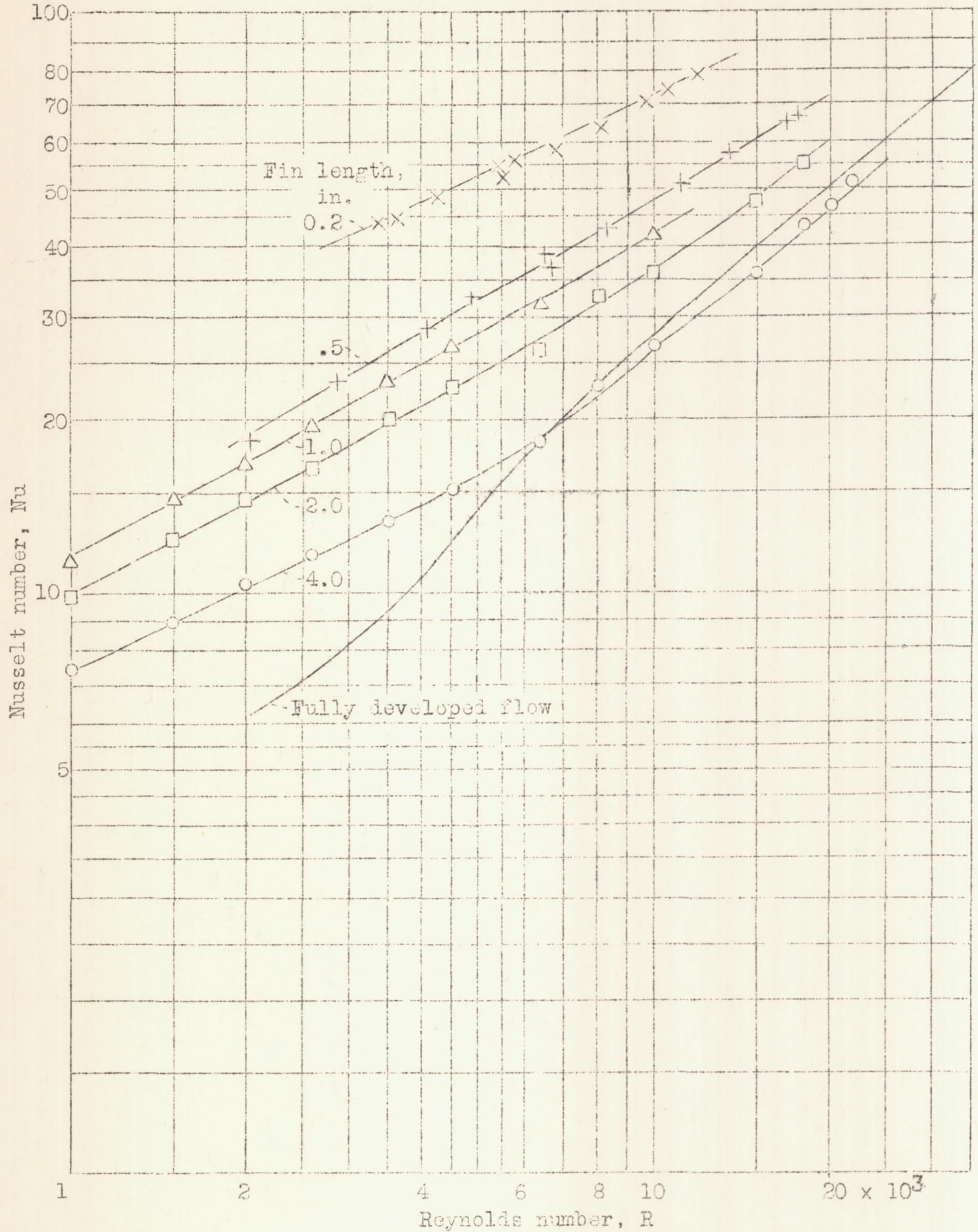

Figure 2.- Variation of Nusselt number with Reynolds number for the fin arrangements tested.

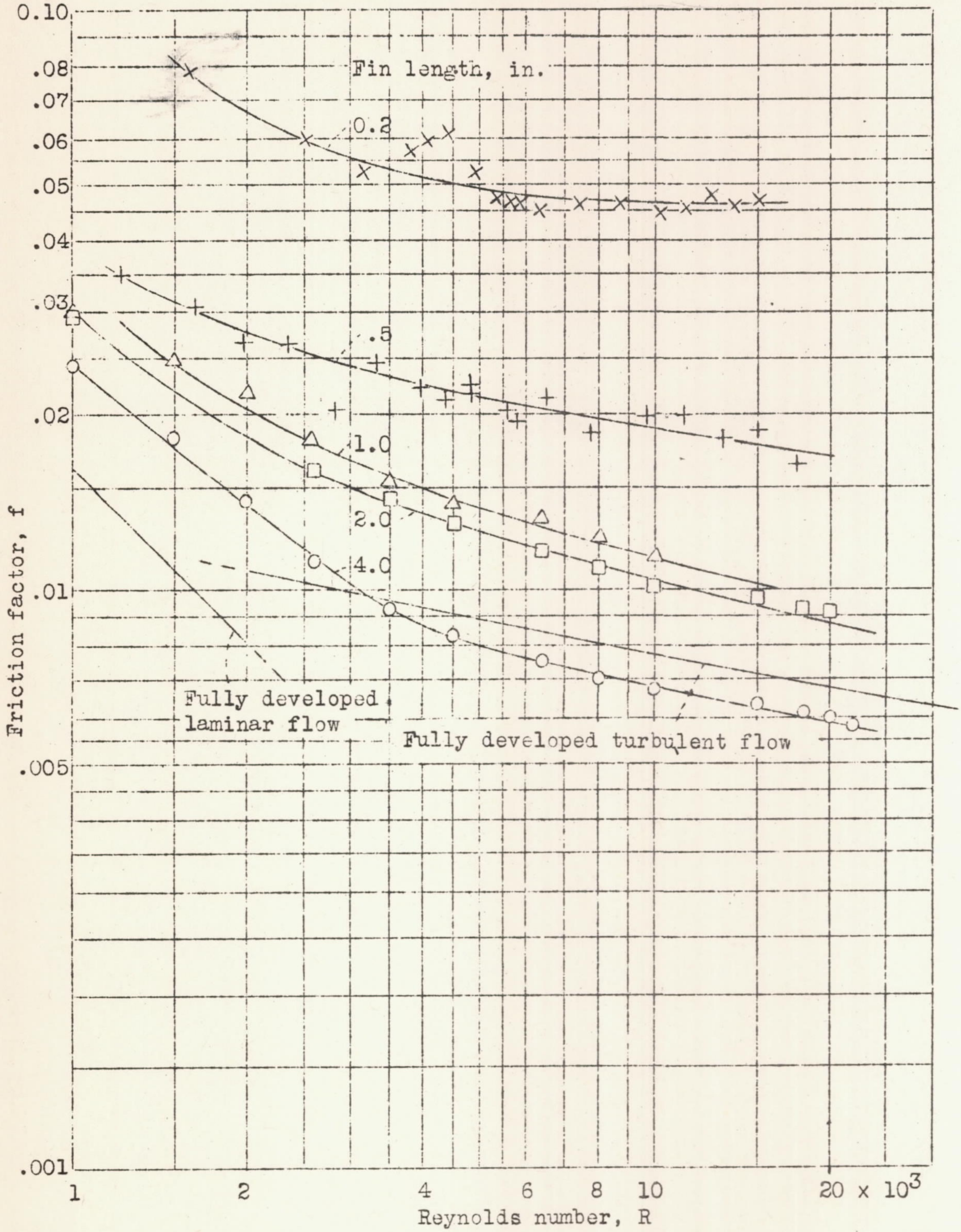

Figure 3.- Variation of friction factor with Reynolds number for the fin arrangements tested.

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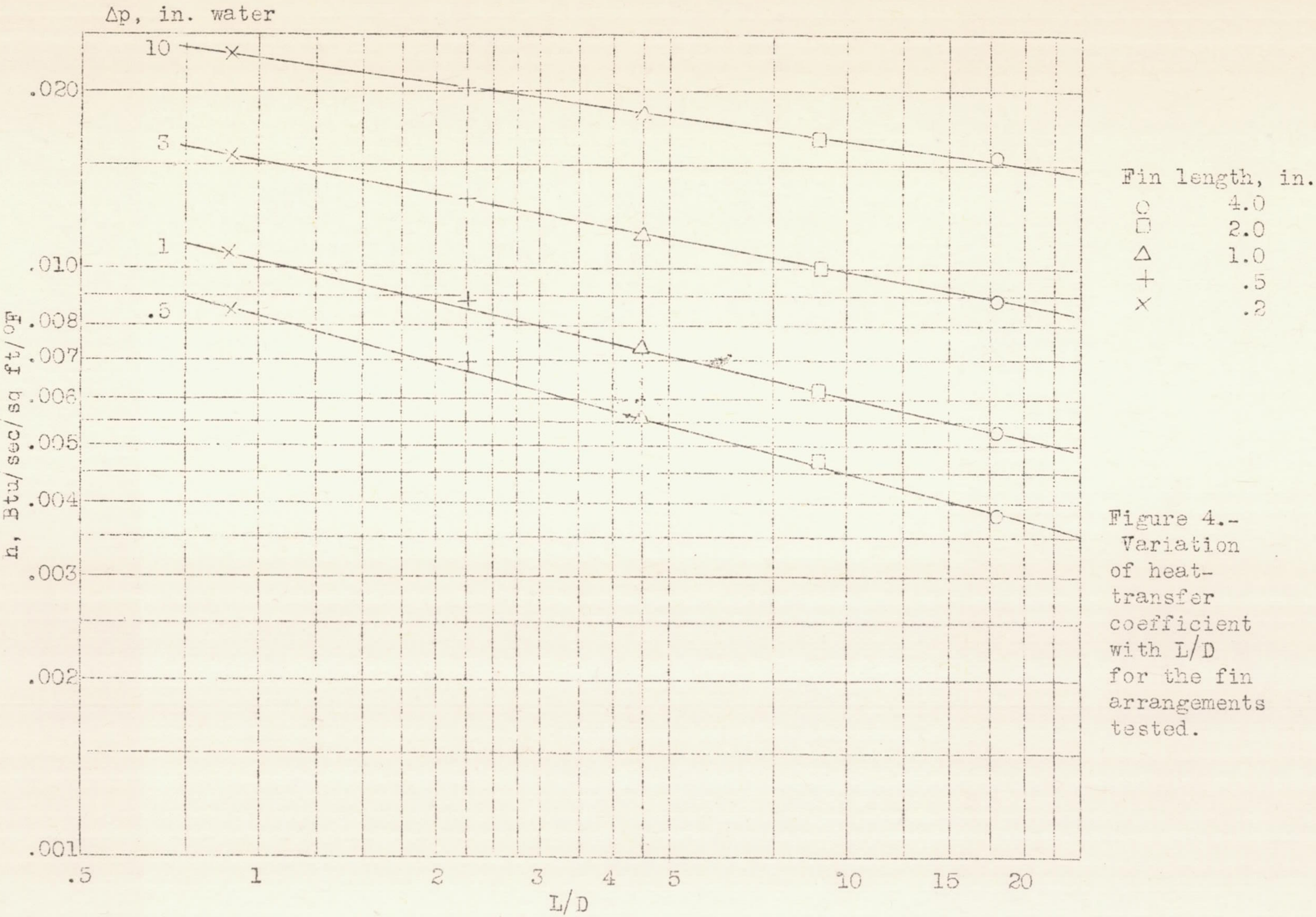

Figure 4.- Variation of heat-transfer coefficient with L/D for the fin arrangements tested.