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RESEARCH MEMORANDUM

MEASUREMENTS OF HEAT-TRANSFER AND FRICTION COEFFICIENTS FOR AIR FLOWING IN A TUBE OF LENGTH-DIAMETER RATIO OF 15 AT HIGH SURFACE TEMPERATURES

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NATIONAL ADVISORY COMMITTEE
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MEASUREMENTS OF HEAT-TRANSFER AND FRICTION COEFFICIENTS FOR AIR FLOWING IN A TUBE OF LENGTH-DIAMETER RATIO OF 15 AT HIGH SURFACE TEMPERATURES

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SUMMARY

Measurements of average heat-transfer and friction coefficients were obtained with air flowing through a smooth, electrically heated tube with a bellmouth entrance, and with a length-to-diameter ratio of 15 for a range of average surface temperature from 875° to 1735° R, Reynolds number from 2200 to 300,000, exit Mach numbers up to unity, and heat fluxes up to 230,000 Btu per hour per square foot of heat-transfer area.

The results indicate that in the turbulent range of Reynolds numbers based on tube diameter, good correlations of the average heat-transfer and friction coefficients are obtained when the physical properties and density of the air are evaluated at a reference film temperature midway between the surface and fluid bulk temperature. The average heat-transfer coefficients correlated with those obtained previously from the same test equipment with longer tubes (length-to-diameter ratios of 30, 60, and 120) at high surface temperatures and heat fluxes on the basis that the average heat-transfer coefficient varies as the -0.1 power of the length-to-diameter ratio. The average friction coefficients were the same as the values obtained with longer tubes for Reynolds numbers above approximately 30,000.

In the transition from the laminar to the turbulent range of Reynolds numbers, the reference film temperature did not give good correlation and the average heat-transfer and friction coefficients increased with increasing ratio of surface-to-fluid bulk temperature when the fluid properties and density were evaluated at the fluid film temperature.

INTRODUCTION

In an experimental investigation of average heat-transfer and friction coefficients for the turbulent flow of air in smooth tubes (ref. 1), it was found that the ratio of surface-to-fluid bulk temperature has an appreciable effect on heat-transfer and friction correlations. A similar effect was obtained in an analytical and experimental investigation (ref. 2) of local coefficients for fully developed turbulent flow with variable fluid properties (which precluded any possible effect of flow development near the tube entrance that may be included in the average coefficients of ref. 1). However, for the case of fully developed laminar flow, an analytical investigation (ref. 3) indicates that the effect of the ratio of surface-to-fluid bulk temperature is nearly negligible, when air properties are evaluated at the bulk temperature. 2627

From these investigations it might be expected that in the transition region from laminar to turbulent flow the effect of the ratio of surface-to-fluid bulk temperature varies between the two limits. Moreover, in the entrance region of a tube in which the flow is not fully developed the effect of the ratio of surface-to-fluid bulk temperature is uncertain.

In view of this uncertainty and of the current interest in short-tube heat-exchanger applications utilizing the higher heat-transfer rates attending flow development in the entrance region, the scope of the investigation of reference 1 is extended herein to a tube having a length-diameter ratio of 15. The data are presented in the form of average heat-transfer and friction coefficients and are compared with the data of reference 1 and with the data of references 4 and 5 obtained for ratios of surface-to-fluid bulk temperatures near unity.

APPARATUS

The experimental setup is essentially the same as described in reference 1. For convenience, however, the apparatus is briefly reviewed.

A schematic diagram of the test section and associated equipment is shown in figure 1. Compressed air at about 100 pounds per square inch was supplied through a pressure-regulating valve into a surge tank. From the surge tank the air passed through a second pressure-regulating valve into a bank of rotameters and then into a mixing tank, which consisted of three concentric passages so arranged that the air made three passages through the tank before entering the test section. Baffles were provided in the central passage to promote thorough mixing of the air before it entered the test section. From the test section the air flowed through a second mixing tank and was then discharged to the atmosphere. The test section, mixing tanks, and adjoining piping were thermally insulated.

The temperature of the air entering and leaving the test section was measured by pairs of iron-constantan thermocouples located downstream of the mixing baffles in the entrance and exit mixing tanks, respectively.

Test Section

The test section, as shown in figure 2, consisted of an Inconel tube having an inside diameter of 0.402 inches, a wall thickness of 0.049 inches, a length of 6 inches, and a length-diameter ratio of approximately 15. Steel flanges were welded to the ends of the tube for electrical connections. A bellmouth nozzle having a throat diameter equal to the inside diameter of the tube served as an entrance to the tube.

Pressure taps were installed in the flanges at the ends of the tube, to measure the static pressure drop.

The outside wall temperature was measured at 18 locations by means of chromel-alumel thermocouples and a self-balancing, indicating-type potentiometer. One thermocouple was located on each flange and two thermocouples 180° apart at eight stations along the tube.

Electrical System

Power was supplied to the heater tube from a 208-volt, 60-cycle supply line through an autotransformer and a power transformer. The low-voltage leads of the power transformer were connected to the steel flanges on the test section by means of copper cables.

An ammeter connected to the load through a 240:1 instrument current transformer was used to measure the current through the test section. The voltage drop across the test section was measured by means of a voltmeter connected to the steel flanges of the tube.

The capacity of the electrical equipment was 15 kilovolt-amperes.

Range of Conditions

Heat-transfer and associated pressure-drop data were obtained over a range of surface temperatures from 875° to 1735° R, inlet-air temperature of 535° R, surface-to-air temperature ratio from 1.6 to 2.8, heat flux up to 230,000 Btu per hour per square foot, Reynolds number from 2200 to 300,000, and exit Mach number up to unity.

SYMBOLS

The following symbols are used in this report:

c_p	specific heat of air at constant pressure, Btu/(lb)(°F)
D	inside diameter of test section, ft
E	voltage drop across test section, volts
f_f	modified average friction coefficient 2627
G	mass flow per unit cross-sectional area, lb/(hr)(sq ft)
g	acceleration due to gravity, 4.17×10^8 ft/hr^2
h	average heat-transfer coefficient, Btu/(hr)(ft^2)(°F)
I	current flow through test section, amperes
k	thermal conductivity of air, Btu/(hr)(sq ft)(°F/ft)
L	heat-transfer length of test section, ft
p	absolute static pressure, lb/sq ft
Δp_{fr}	friction static-pressure drop across test section, lb/sq ft
Q	rate of heat transfer to air, Btu/hr
Q_{EI}	electrical heat input, Btu/hr
Q_a	rate of heat transfer to air, $W\, c_p(T_2 - T_1)$, Btu/hr
Q_L	rate of heat lost to surroundings, Btu/hr
R	gas constant for air, 53.35 ft-lb/(lb)(°F)
S	heat-transfer area of test section, sq ft
T	total or stagnation temperature, °R
T_b	average bulk temperature, defined by $(T_1+T_2)/2$, °R
T_f	average film temperature defined by $(T_s+T_b)/2$, °R
T_s	average inside surface temperature of test section, °R

t static temperature, $^{\circ}R$

V velocity, ft/hr

W air flow, lb/hr

γ ratio of specific heats of air

μ absolute viscosity of air, lb/(hr)(ft)

ρ density of air, lb/(cu ft)

$c_p\mu/k$ Prandtl number

$\rho VD/\mu$ Reynolds number

hD/k Nusselt number

Subscripts:

b bulk (when applied to properties, indicates evaluation at average bulk temperature T_b)

f film (when applied to properties, indicates evaluation at average film temperature T_f)

1 test-section entrance

2 test-section exit

RESULTS AND DISCUSSION

Axial-Wall-Temperature Distribution

Representative axial outside-wall temperature distributions are shown in figure 3. The outside wall temperature is plotted against the distance from the tube entrance for each temperature level for a high flow rate. Also included for the highest temperature level is a wall-temperature distribution obtained during heat-loss runs in which there was no air flowing through the tube. The rate of heat transfer to the air, the air flow, the temperature rise of the air, and the average inside tube wall temperature are tabulated.

For the case of no air flow through the tube, the heat generated in the tube wall is lost by means of conduction through the electric connector flanges and cables and thus the wall temperature is affected along the entire length of the tube by the end losses. The portion of tube along which the wall temperature is affected by end losses dimenished as the flow rate increased.

Heat Balance

The heat balance is shown in figure 4 where the ratio of the electrical heat input minus heat loss to the heat transferred to the air, as determined by flow rate and temperature measurements, is plotted against the heat transferred to the air. The heat balances obtained at low heat inputs and correspondently low flow rates were very poor. In this region the outlet air temperature could not be measured accurately because the extremely low velocities in the outlet mixing tank prevented the attainment of equilibrium conditions in the mixing tank.

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The heat balance improved rapidly with increase in flow rate and for values corresponding to turbulent flow in the test section the unaccountable heat loss diminished to 7 to 10 percent of the heat input. For the higher flow rates the flow rate and temperature-rise measurements were believed to be somewhat more dependable than the electrical and heat-loss measurements. The reverse is true at low flow rates because of the inaccuracies in measuring the outlet air temperature as stated previously. Therefore, the heat-transfer coefficients are calculated herein from the flow rate and the temperature rise of the air in the high-flow-rate region (Reynolds number greater than 10,000) and from the electrical heat input and heat-loss measurements in the low-flow-rate region.

Correlation of Heat-Transfer Coefficients

The average heat-transfer coefficient h was computed from the experimental data by the relation

$$h = \frac{Q}{S(T_s - T_b)}$$

$$Q = W\, c_{p,b}(T_2 - T_1) \text{ or } (3.415\, EI - Q_L) \tag{1}$$

where the bulk temperature of the air T_b was taken as the arithmetic mean of the total temperatures at the entrance T_1 and the exit T_2 of the test section. The average surface temperature T_s was taken as an integrated average of the local outside wall temperature less the temperature drop through the wall.

The physical properties of air used in calculating the Nusselt, Reynolds, and Prandtl numbers are the same as those used in reference 1 wherein the viscosity and specific heat were based on values reported in reference 6 and the thermal conductivity was assumed to vary as the square root of temperature.

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The results presented in reference 1 for turbulent flow in tubes having length-to-diameter ratios of 30 to 120 indicate that the average Nusselt number decreases progressively as the ratio of surface-to-fluid bulk temperature increases when the fluid properties are evaluated at the fluid bulk temperature. The effect of the ratio of surface-to-bulk temperature was eliminated by evaluating the properties of the air, including the density term in the Reynolds number at the film temperature, defined as the arithmetic average of the surface and bulk temperatures, and the data for Reynolds numbers greater than 10,000 were well represented by the following relation:

$$\frac{hD}{k_f} = 0.034 \left(\frac{\rho_f V_b D}{\mu_f}\right)^{0.8} \left(\frac{c_{p,f}\ \mu_f}{k_f}\right)^{0.4} \left(\frac{L}{D}\right)^{-0.1} \tag{2}$$

The average Nusselt numbers obtained herein for a range of Reynolds number from 2200 to 300,000 and ratios of surface to bulk temperature from 1.6 to 2.8 are shown correlated accordingly in figure 5. A line representing equation (2) is included for comparison. The data for Reynolds numbers larger than 10,000 agree very well with the reference line and show very little variation with ratio of surface-to-bulk temperature, which indicates that the effect of flow development on the average Nusselt number is adequately represented by the length-to-diameter ratio raised to the -0.1 power over the range of length-to diameter ratio of 15 to 120. Also, the effect of the ratio of surface-to-bulk temperature on the average Nusselt number is nearly the same as that obtained with long tubes, and that flow development has little influence on this effect for tubes exceeding 15 diameters in length.

The results of an analytical investigation presented in reference 3 for the case of laminar flow in tubes with fluid properties variable along the radius indicate that the effect of the ratio of the surface-to-fluid bulk temperature is negligible when the fluid properties are evaluated at a temperature near the fluid bulk temperature. Hence, it may be expected that in the transition from laminar to turbulent flow (Reynolds number from 1000 to 10,000) the effect of the ratio of surface-to-fluid bulk temperature varies with Reynolds number in the transition region and could be eliminated by evaluating the fluid physical properties and density at a reference temperature which varies from the fluid bulk to the fluid film temperature as Reynolds number is increased.

This effect, together with the poor heat balance obtained at low Reynolds numbers, accounts for the increased scatter of the data obtained for Reynolds numbers less than 10,000. 2627

A comparison of the average heat-transfer coefficients obtained in reference 1 for a tube having a length-to-diameter ratio of 120 and the local heat-transfer coefficients obtained in reference 2 near the end of a tube 87 diameters in length (which precludes any effects of flow development on the local heat-transfer coefficient) indicates that the average and local heat-transfer coefficients have the same value. Hence, for tubes longer than 120 diameters the average heat-transfer coefficient remains constant with increase in length-to-diameter ratio and equation (2), by the substitution of $L/D = 120$, becomes

$$\frac{hD}{k_f} = 0.021\left(\frac{\rho_f\ V_b D}{\mu_f}\right)^{0.8}\left(\frac{c_{p,f}\ \mu_f}{k_f}\right)^{0.4} \tag{2a}$$

An alternate method of correcting for the effect of the ratio of length-to-diameter on the average heat transfer is presented in reference 7 in which correlation of the data of references 4 and 5 and various other investigators was obtained by use of the parameter $(1 + L/D^{-0.7})$. This method has the advantage that a single relation can be used for all length-to-diameter ratios. The present data and some typical data of reference 1 are replotted accordingly in figure 6 for representative low- and high-heat-flux conditions. Included for comparison is the line represented by the following expression:

$$\frac{hD}{k_f} = 0.021\left(\frac{\rho_f\ V_b D}{\mu_f}\right)^{0.8}\left(\frac{c_{p,f}\ \mu_f}{k_f}\right)^{0.4}\left(1 + \left(\frac{L}{D}\right)^{-0.7}\right) \tag{3}$$

The effect of length-to-diameter ratio on the average heat-transfer coefficient is further illustrated in figure 7 wherein the mean values of the present data and those of reference 1 for Reynolds numbers above 10,000 are compared with the data of references 4 and 5 obtained at low heat fluxes. Included are the broken, dashed, and solid lines representing equations (2), (2a), and (3), respectively. The present data and those of reference 1 are best represented by equation (2), while the data of references 4 and 5 are best fitted by equation (3). For the entire range of length-to-diameter ratio, equation (3) best represents all the data and is the most advantageous one.

Correlation of Friction Coefficient

The method of calculating the average friction coefficient is essentially the same as described in reference 1 wherein

$$f_f = \frac{\Delta p_{fr}}{4 \frac{L}{D} \frac{\rho_f V_b^2}{2g}} \tag{4}$$

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where

$$\Delta p_{fr} = (p_1 - p_2) - \frac{G^2 R}{g}\left(\frac{t_2}{p_2} - \frac{t_1}{p_1}\right)$$

$$\rho_f = \left(\frac{\rho_1 + \rho_2}{2}\right)\frac{t_b}{t_f}$$

and

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$$t = -\frac{\gamma g}{(\gamma-1)R}\left(\frac{p}{G}\right)^2 + \sqrt{\left[\frac{\gamma g}{(\gamma-1)R}\left(\frac{p}{G}\right)^2\right]^2 + 2T\frac{\gamma g}{(\gamma-1)R}\left(\frac{p}{G}\right)^2}$$

The subscripts 1 and 2 refer to positions within the tube, located 1/8 inch from the entrance and exit ends of the tube, respectively.

The average friction coefficients as calculated previously are shown in figure 8 correlated by the method proposed in reference 1 for high-heat-flux conditions. Included for comparison is the line representing the von Kármán-Nikuradse relation for turbulent flow in pipes corrected for the effect of heat flux on friction factor, which is

$$\frac{1}{\sqrt{8\frac{f_f}{2}}} = 2 \log\left(\frac{\rho_f V_b D}{\mu_f}\sqrt{8\frac{f_f}{2}}\right) - 0.8 \tag{5}$$

For Reynolds numbers above 30,000, the friction coefficients are in reasonable agreement with the reference line and no effects of length-to-diameter ratio are noticeable, which was unexpected in view of the observed effect of length-to-diameter ratio on the average heat-transfer coefficients. In the lower Reynolds number region, the friction coefficients deviate considerably from the reference line. The data obtained without heat addition fall below the reference line and indicate that the transition from laminar to turbulent flow extends up to a

Reynolds number of approximately 30,000. The data also vary with ratio of surface-to-bulk temperature, indicating that the effect of heat flux in the transition region is less than that obtained in the fully turbulent region, and hence equation (5) is not applicable in this region.

SUMMARY OF RESULTS

The results of this investigation of heat transfer and pressure drop for air flowing through a smooth tube having a length-diameter ratio of 15 for a range of surface temperature from 875° to 1735° R and corresponding surface-to-bulk temperature ratio from 1.6 to 2.8, Reynolds number from 2200 to 300,000, and heat flux up to 230,000 Btu per hour per square foot may be summarized as follows: 2627

1. The effect of the ratio of surface-to-bulk temperature on correlations of the average heat-transfer and friction coefficients for turbulent-flow Reynolds numbers above 10,000 was the same as that obtained in a previous investigation for longer length-to-diameter ratio tubes, and was eliminated by evaluating the physical properties and density of the air at a film temperature halfway between the bulk and surface temperatures.

2. In the transition from the laminar to the turbulent range of Reynolds numbers based on tube diameter, the average heat-transfer and friction coefficients increased with increasing ratio of surface-to-bulk temperature and indicated that the reference film temperature is not applicable in this region.

3. The data of the investigation correlated with the data obtained in previous investigations with longer tubes on the basis that the heat-transfer coefficient varies as the -0.1 power of the length-to-diameter ratio.

4. No effect of length-to-diameter ratio on correlation of the average friction coefficients was observed for Reynolds numbers above 30,000.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, March 20, 1953

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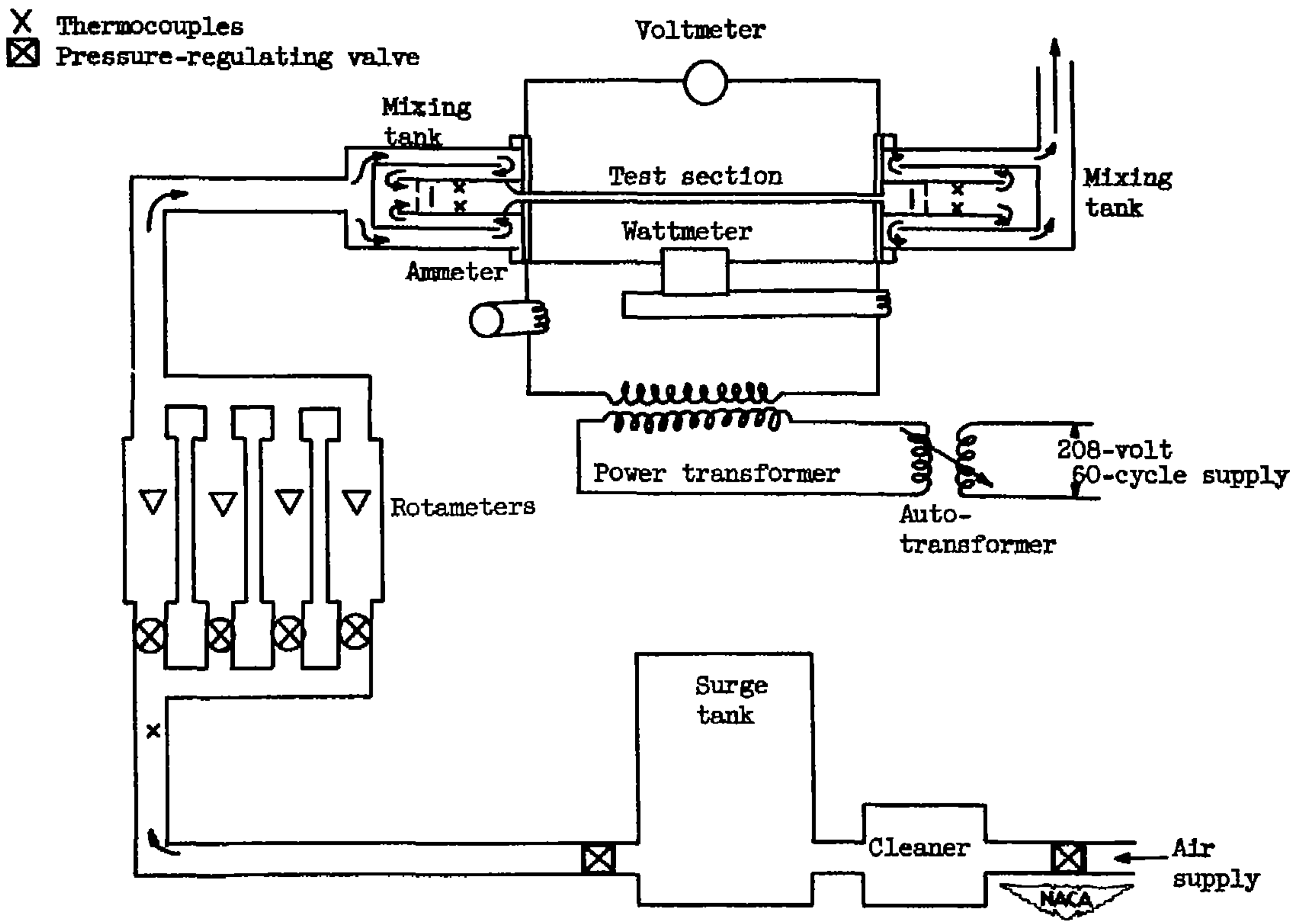

Figure 1. - Schematic diagram showing arrangement of apparatus.

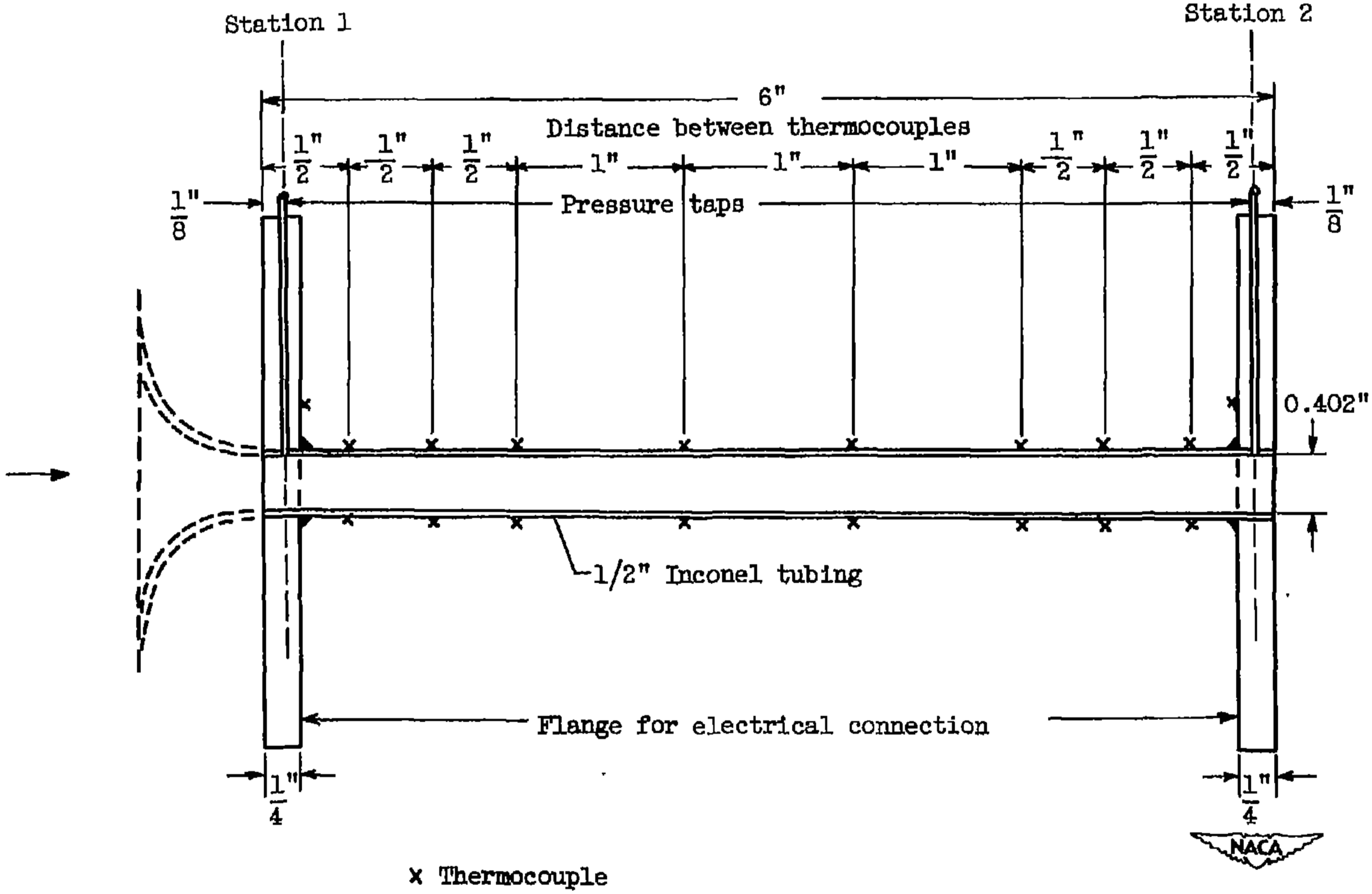

Figure 2. - Test section showing thermocouple and pressure-tap location.

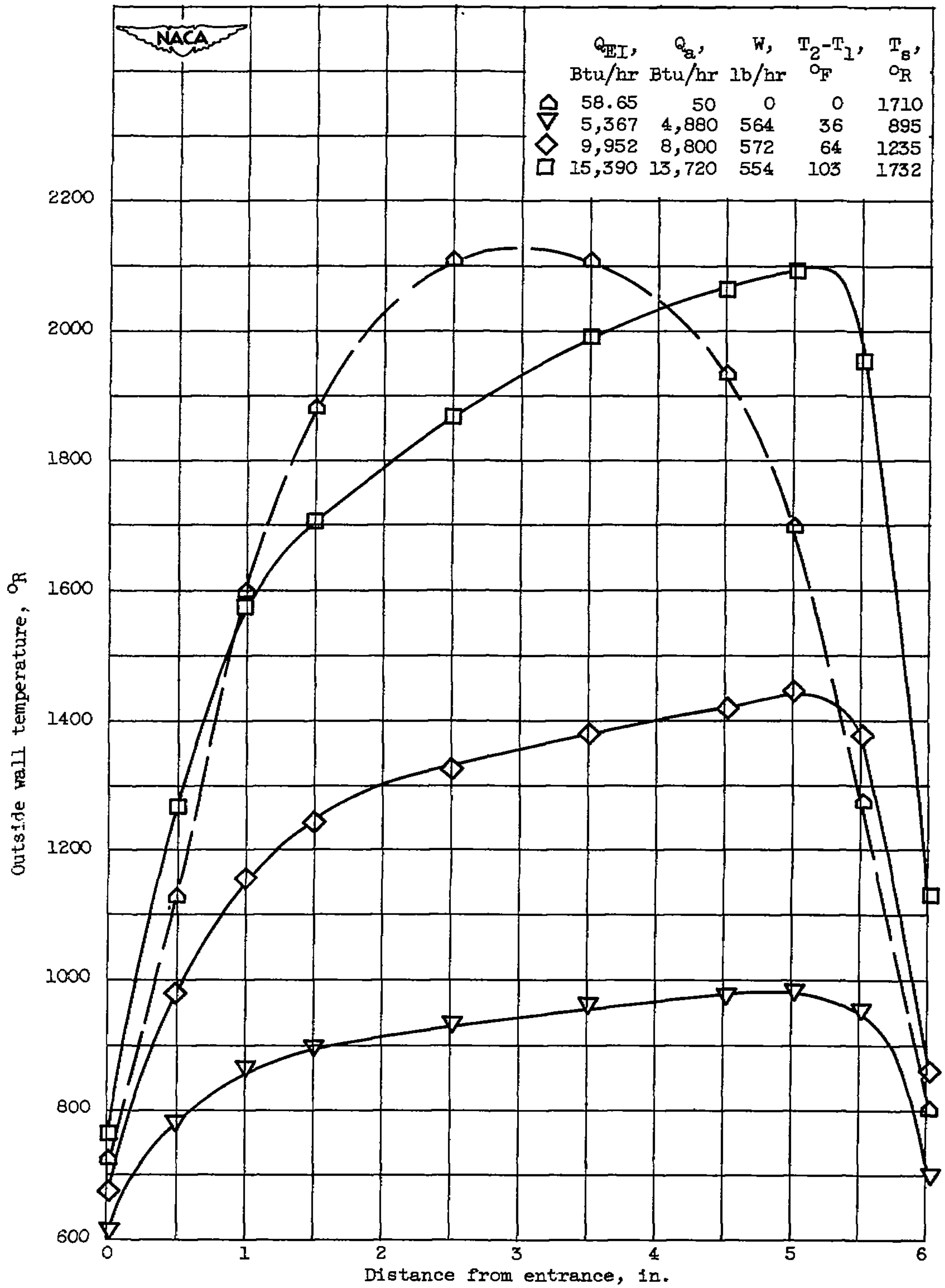

Figure 3. - Representative wall-temperature distributions.

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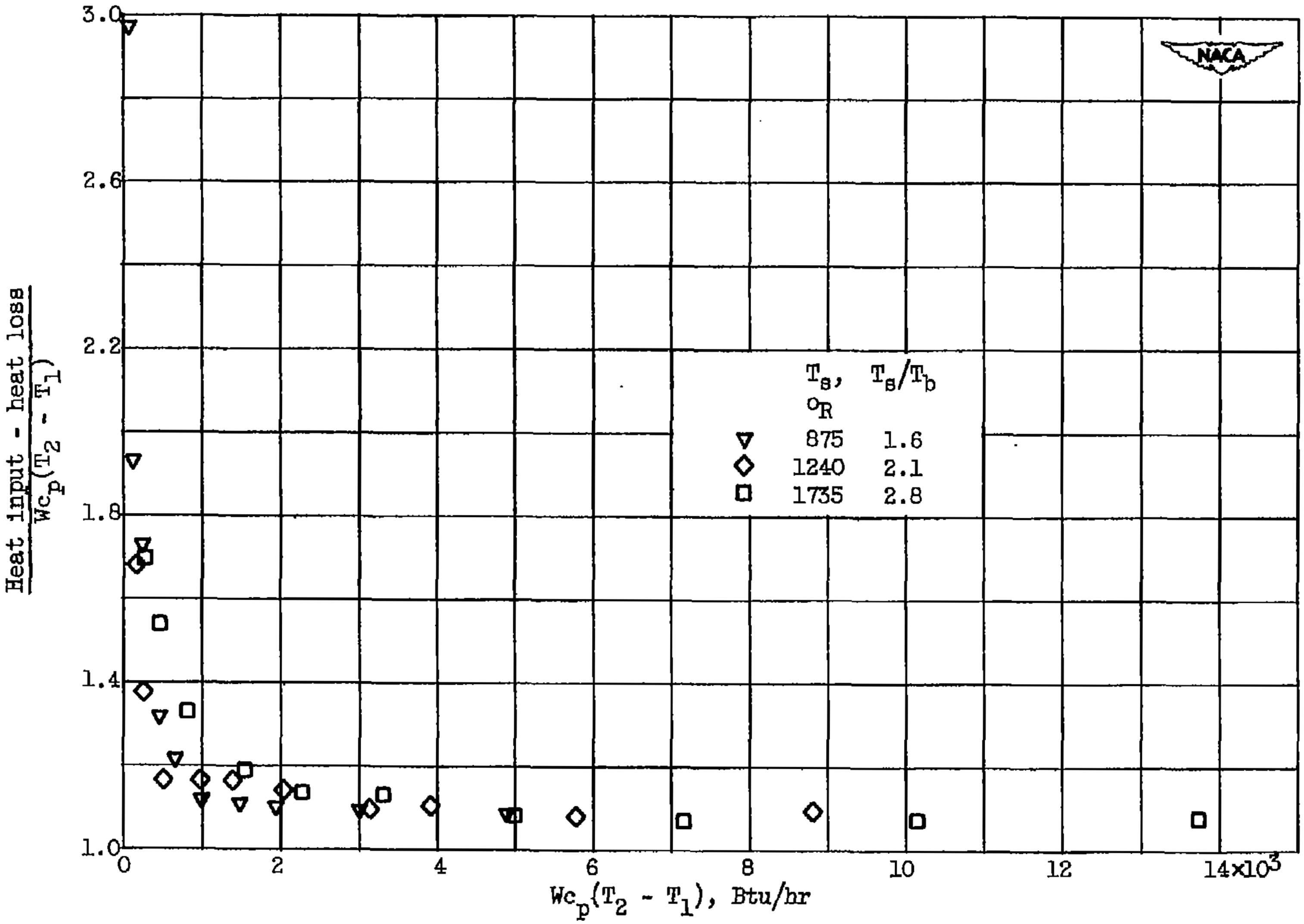

Figure 4. - Heat balance.

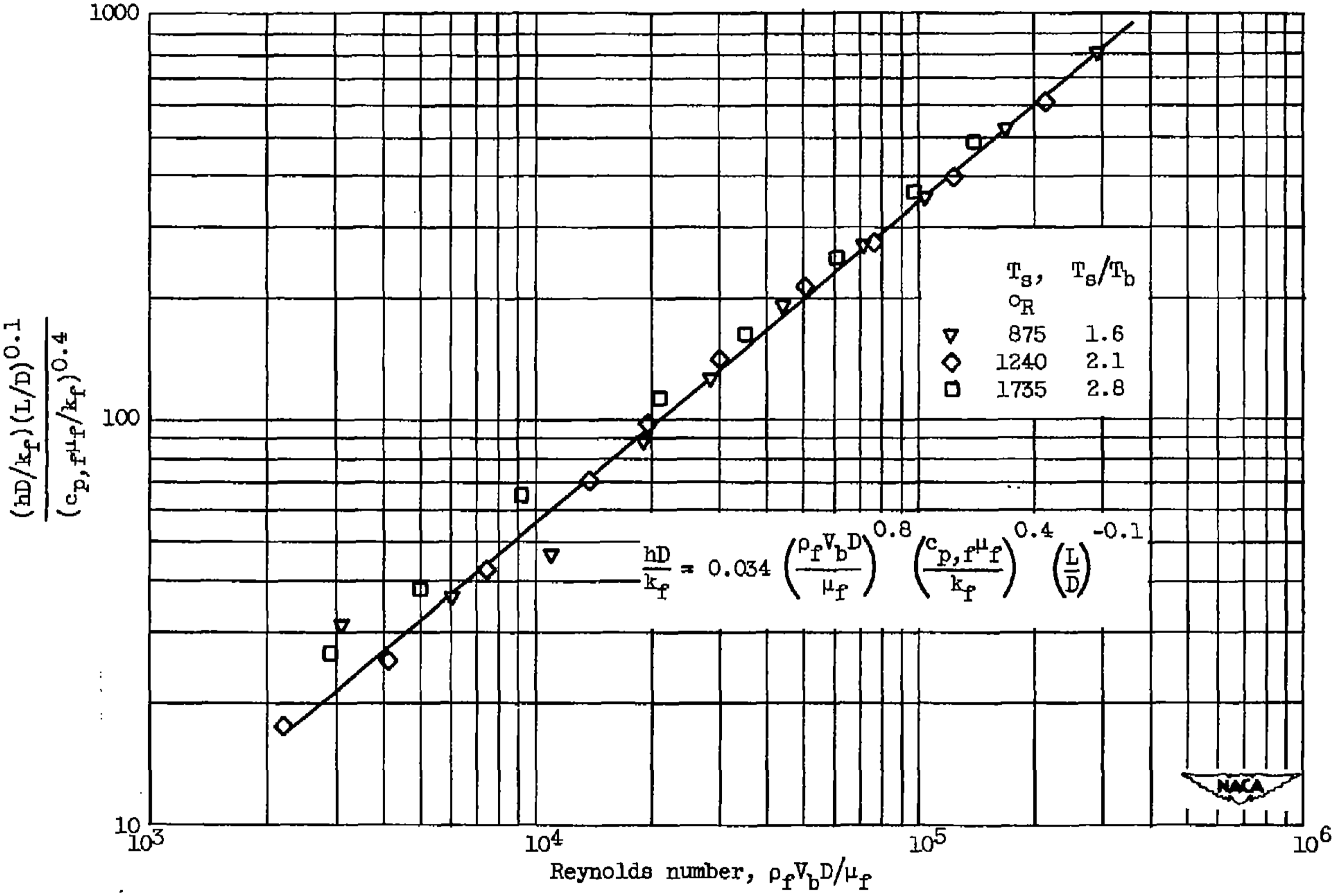

Figure 5. - Correlation of heat-transfer coefficients with variable heat flux. Length-diameter ratio, 15; bellmouth entrance, inlet temperature, 535° R; properties of air evaluated at film temperature.

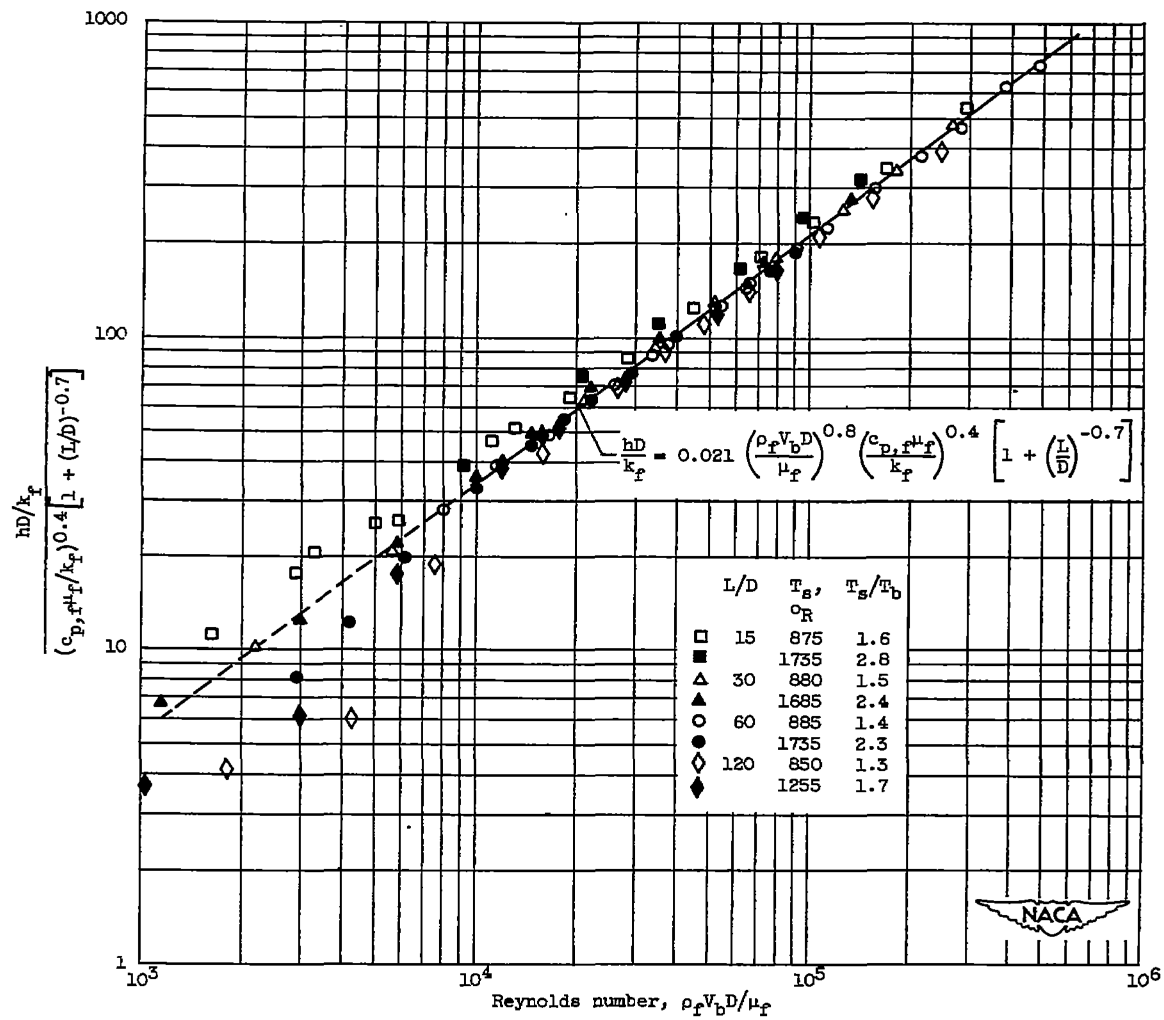

Figure 6. - Alternate method of correlating heat-transfer coefficients with variation in length to diameter ratio. Bellmouth entrance; inlet temperature, 535° R; properties of air evaluated at film temperature.

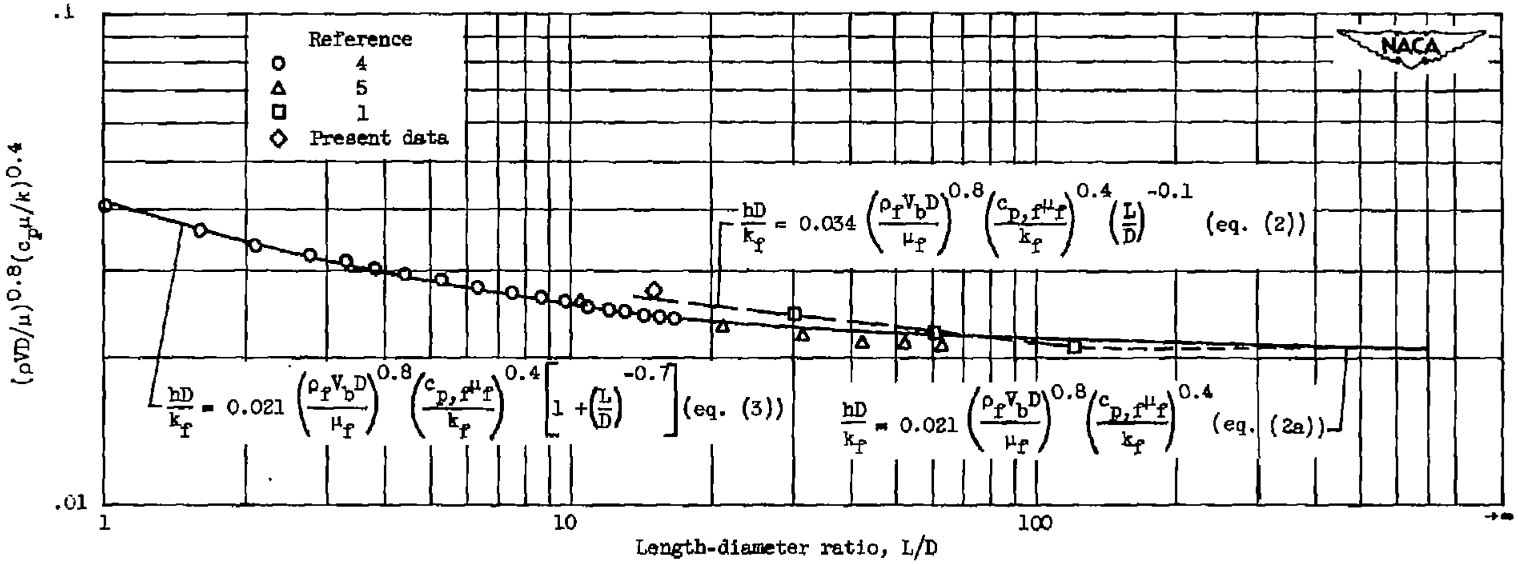

Figure 7. - Comparison of variations of average heat-transfer coefficients with length-to-diameter ratio. Bellmouth entrance; Reynolds number > 10,000.

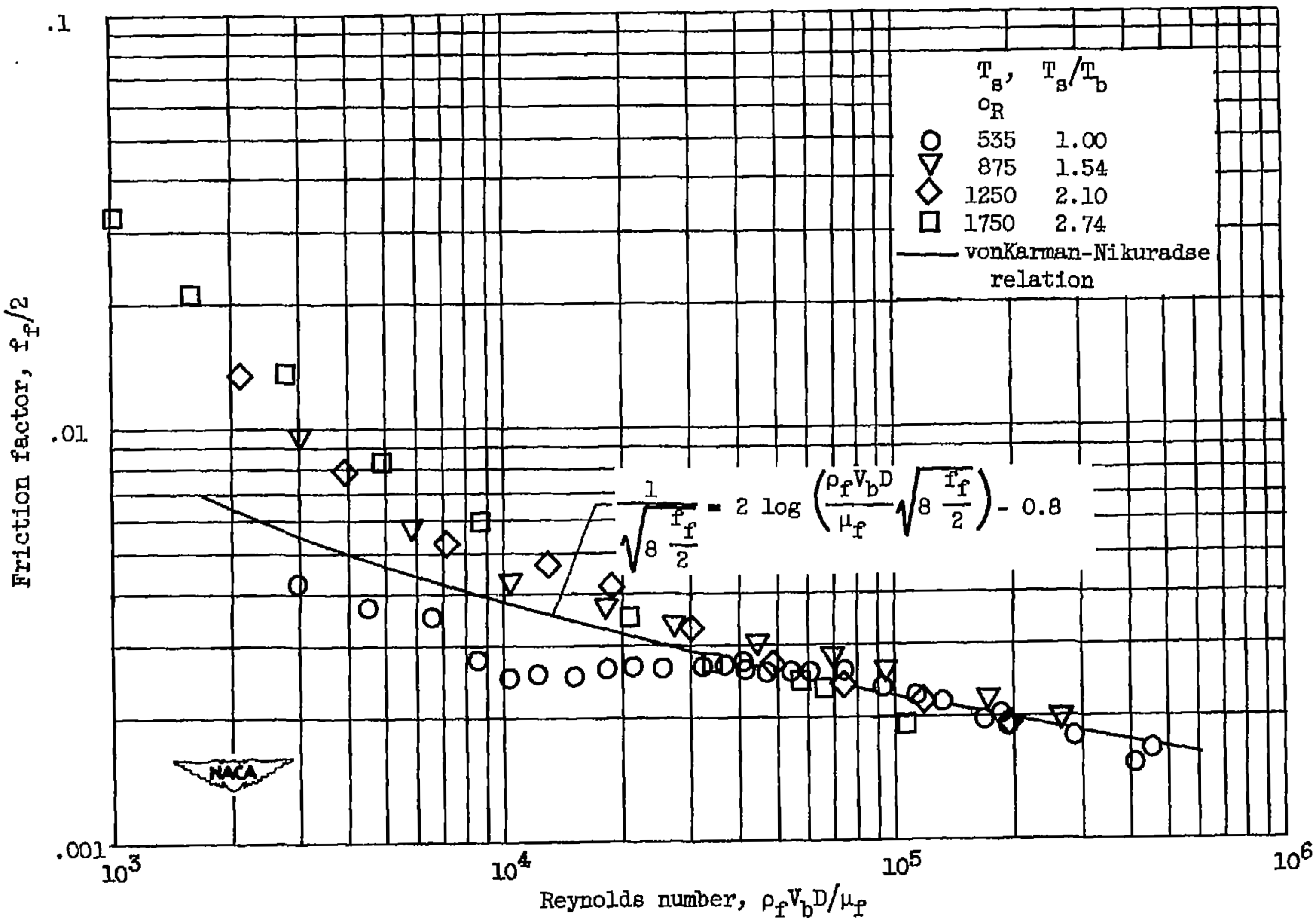

Figure 8. - Correlation of average friction factors. Viscosity and density evaluated at film temperature T_f.