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RESEARCH MEMORANDUM

PRELIMINARY INVESTIGATION OF HEAT TRANSFER TO WATER FLOWING

IN AN ELECTRICALLY HEATED INCONEL TUBE

By Samuel J. Kaufman and Francis D. Isely

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Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS
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SUMMARY

A preliminary heat-transfer investigation was conducted with
water flowing through an electrically heated Inconel tube with an
inside diameter of 0.204 inch and a length-diameter ratio of 50 for
ranges of Reynolds number up to 100,000 and of entrance pressure up
to 200 inches of mercury gage. Runs were conducted with the tube
in a horizontal position and with the tube in a vertical position
where up and down flows were individually investigated.

For conditions in which no boiling occurred, good correlation
of average heat-transfer coefficients was obtained by use of the
familiar Nusselt relation, wherein the physical properties of the
water were evaluated at an average bulk temperature. Runs made in
the nucleate-boiling region, however, gave higher values of heat-
transfer coefficient than would be predicted by the Nusselt relation.

INTRODUCTION

An experimental program has been undertaken at the NACA Lewis
laboratory to obtain surface-to-fluid heat-transfer information over
a wide range of inside-tube-wall temperatures and heat-flux densities.
One phase of the heat-transfer program is being conducted with water
flowing through an electrically heated Inconel tube. With constant
power input, a change in the type of boiling of the water may cause
a runaway condition that results in burnout of the tube. Two types
of surface boiling may occur: (a) nucleate, and (b) film. In
nucleate boiling, which generally starts when the surface tempera-
ture is only slightly in excess of the fluid saturation temperature,
bubbles form on the surface and then break away carrying large amounts
of heat into the main fluid stream. With nucleate boiling, the heat-
transfer coefficient may be appreciably larger than predicted by the
conventional Nusselt type equation. In film boiling, which occurs

when the wall temperature is increased above the fluid saturation
temperature, the bubbles form faster than they break away and result
in an insulating vapor film on the wall; the vapor film causes a
decrease in heat transfer.

Nucleate boiling, film boiling, and heating without change of
phase all represent different heat-transfer mechanisms; the heat
transfer incurred by these mechanisms cannot be predicted by the
same type of equation. The transition from nucleate to film boiling
causes the fluid flow to become unstable and results in simultaneous
surging of the flow and rapid rise in wall temperature, which, if
unchecked, leads to eventual tube burnout. The two resulting prob-
lems are therefore: (a) the prediction of heat-transfer coefficient,
and (b) the determination of the limit of stability.

Runs were made with flow through the heater tube (inside diam-
eter, 0.204 in.; effective heat-transfer length, 10 in.) in a hori-
zontal position and with flow up and down through the tube in a
vertical position. Running conditions include a range of Reynolds
numbers from 2000 to 100,000, heat fluxes up to 11,000 Btu per
minute per square foot, and entrance pressures up to 200 inches of
mercury gage.

The preliminary investigation reported herein contains data
for the heated water and the nucleate-boiling region. Average heat-
transfer coefficients are correlated in accordance with the familiar
Nusselt relation. The correlation based on these preliminary results
are subject to further investigation and extension of the range of
data.

APPARATUS

A schematic diagram of the electrically heated Inconel tube
and the associated equipment used in the runs is shown in figure 1.

Heater Tube

The heater tube (fig. 2) consists of an Inconel tube having an
inside diameter of 0.204 inch, a wall thickness of 0.055 inch, and
a total length of 10.75 inches. Electric power is dissipated in the
Inconel tube, which acts as a resistance element and is cooled by
the water in forced convection. Inconel cones are welded to each
end of the heater tube to provide electric contact with Inconel
flanges, which are in turn connected by copper bus bars and cables

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to the electric power supply. The Inconel cones are used in order
to lessen heat losses from the ends of the heater tube by reducing
the area of the flange in contact with the ends of the tube.

The heater tube is thermally insulated by a covering of insu-
lating material, which is surrounded by two concentric radiation
shields with insulating cement in the space between the shields.

Outside-wall temperatures of the heater tube are measured at
26 locations (two thermocouples located 180° apart at each of
13 stations) by chromel-alumel thermocouples (flexible glass-
insulated 24-gage wire) and a self-balancing indicating-type poten-
tiometer. The thermocouples were spot-welded to the tube and each
thermocouple junction is located in a plane normal to the axis of
the tube.

Water System

Water at a maximum pressure of slightly over 200 inches of
mercury gage is supplied by a 5-gallon-per-minute pump and flows
through a filter, a rotameter, around or through a steam heater,
and into a calming tank before entering the heater tube. After
passing through the tube, the water flows through a mixing tank,
through a heat exchanger, and then to the storage tank from which
it is recirculated.

Water pressure and flow were controlled by a by-pass valve
arrangement that permits pressures from 10 to 200 inches of mercury
gage at the entrance to the heater tube.

The calming and mixing tanks are made of three concentric
cylinders so arranged that the water makes three axial passes
before entering and after leaving the heater tube. Thus, a blanket
of water thermally insulates the tanks to assure accurate tempera-
ture readings. A honeycomb in the calming tank straightens the
flow of water entering the tube and a set of baffles in the mixing
tank assures thorough mixing of the water leaving the heater tube.

The temperature of the water entering and leaving the tube
is measured by thermocouples placed after the honeycomb in the
calming tank and after the baffles in the mixing tank. The tempera-
ture rise of the water is also measured by these same thermocouples
connected to give a differential reading on a very sensitive man-
ually operated potentiometer.

Static-pressure taps located at the entrance and exit of the heater tube were used to measure the entrance pressure, exit pressure, and pressure drop.

Electrical System

Power is supplied to the heater tube from a 208-volt 60-cycle supply line through an autotransformer and a 20:1 power transformer. The low-voltage leads of the power transformer are connected to the heater tube by copper cables, as previously described.

The power input to the heater tube is measured by an ammeter in conjunction with a 240:1 instrument current transformer and a voltmeter connected across the heater tube.

The capacity of the electrical equipment is 15 kilowatts; the maximum capacity used thus far is 9 kilowatts.

PROCEDURE

Calibration of Heat Losses

In order to establish a heat balance, the heat loss from the heater tube was obtained for a range of tube-wall temperatures by supplying various amounts of power to the heater tube with no water in the tube. After equilibrium conditions were maintained for approximately 1/2 hour, the power input and the tube-wall temperatures were recorded. The power input for a given average wall temperature with no flow was considered to be the external heat loss for the same average wall temperature with water flowing in the tube.

Heat-Transfer Data

Runs were conducted to obtain heat-transfer data for as large a range of wall temperatures as could be obtained with stable flow of the water in the tube, Reynolds numbers from about 2000 to 100,000, and water exit temperatures as high as possible. The water entrance temperature for most of the runs was kept at approximately 85° F. The entrance pressure was varied from 10 to 200 inches of mercury gage, and the power input to the heater tube was varied from 1 to 9 kilowatts.

The procedure for most of the runs consisted in setting the power input and the entrance pressure to the tube at constant pre-determined values. The flow rates were then reduced in steps from a maximum to an unsteady flow condition. Thus, a series of about six steps was made for each pressure and power input. At each step after equilibrium had been obtained, the power, pressures, temperatures, and the flow rate were recorded. This procedure was repeated until the entire range of power and entrance pressures had been covered. The same procedure was used for the vertical tube with the water flowing in the up and the down direction.

SYMBOLS

The following symbols are used in the calculations:

c_p specific heat of water at constant pressure, $(Btu/(lb)({}^\circ F))$

D inside diameter of heater tube, (ft)

G mass velocity of water, $(lb/(hr)(sq\ ft))$

h heat-transfer coefficient, $(Btu/(hr)(sq\ ft)({}^\circ F))$

k thermal conductivity of water, $(Btu/(hr)(sq\ ft)({}^\circ F/ft))$

k_m thermal conductivity of Inconel, $(Btu/(hr)(sq\ ft)({}^\circ F/ft))$

L effective heat-transfer length of heater tube, (ft)

q rate of heat transfer to water, (Btu/hr)

r_1 inner radius of heater tube, (ft)

r_o outer radius of heater tube, (ft)

S heat-transfer area of heater tube, 0.0478 (sq ft)

T_b average bulk total temperature of water, $(T_1 + T_2)/2$, $({}^\circ F)$

T_{bp} saturation temperature of water, $({}^\circ F)$

T_o average outside-wall temperature of heater tube, $({}^\circ F)$

T_s average inside-wall temperature of heater tube, $({}^\circ F)$

T_1 total water temperature at heater-tube entrance, (°F)

T_2 total water temperature at heater-tube exit, (°F)

W   water-flow rate, (lb/hr)

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

$c_p\mu/k$ Prandtl number

DG/μ Reynolds number

hD/d Nusselt number

METHOD OF CALCULATION

The average outside-tube-wall temperature T_o was obtained by plotting curves of temperature against tube length, measuring the area under the curve, and dividing the area by the tube length. The plotted values of the temperature for each point was the average of two thermocouples at the same station.

The average inside-tube-wall temperature T_s was calculated from the average outside-tube-wall temperature and the dimensions of the tube by the following equation, which can be derived with the assumption that heat is generated uniformly across the tube wall and that heat flow is radially inward (reference 1):

$$T_s = T_o - \frac{q}{2\pi \, Lk_m(r_o^2 - r_1^2)} \left[r_o^2 \log_e\left(\frac{r_o}{r_1}\right) - \frac{r_o^2 - r_1^2}{2} \right]$$

Substituting the heater-tube dimensions in the preceding equation results in

$$T_s = T_o - 0.042 \, q/k_m$$

A plot of thermal conductivity of Inconel k_m against temperature is shown in figure 3.

The average bulk total temperature of water T_b was taken as the mean of the total temperatures at the heater-tube entrance T_1 and exit T_2.

Average heat-transfer coefficient h was calculated from the following equation:

$$h = \frac{Wc_p(T_2 - T_1)}{S(T_s - T_b)}$$

The physical properties of water, specific heat at constant pressure c_p, thermal conductivity k, and viscosity μ, are plotted as functions of temperature in figure 4.

RESULTS AND DISCUSSION

Temperature Distribution

The axial distribution of outside-tube-wall temperature along the tube length for the condition of no water in the tube is shown in figure 5 for various amounts of electric heat input. The temperature drop at each end of the heater tube, which is due to conduction losses to the end flanges, is especially large for the high tube-wall temperatures.

Representative temperature-distribution curves with water flowing in the tube are shown in figure 6. The curves shown are for a constant power input of approximately 7 kilowatts (heat flux, 8330 Btu/(min)(sq ft)), a water entrance pressure of 200 inches of mercury gage, and a range of water flow from 194 to 795 pounds per hour (corresponding Reynolds number range from 14,000 to 37,000). The curves are flat when water flows through the heater tube; the temperature drop at the heater-tube entrance is probably due to an excess amount of silver-solder between the cone and the heater tube, which conducted enough heat to cause the temperature drop shown.

Heat Balance

The external heat loss from the heater tube, as obtained from figure 5, is plotted against average outside-tube-wall temperature in figure 7. Heat balances obtained by calculating the heat transferred to water $Wc_p(T_2 - T_1)$ plus heat loss (fig. 7) and plotted against electric heat input shows a maximum deviation of ± 5 percent except where severe boiling was obtained. For the series of runs reported herein, a maximum outside-tube-wall temperature of

approximately 500° F was obtainable with a corresponding heat loss
of less than 30 watts or a maximum loss of 3 percent with 1 kilo-
watt power input.

Correlation of Heat-Transfer Coefficients

Forced-convection heat-transfer data are generally correlated
by the familiar Nusselt relation (reference 2) where Nusselt number
divided by Prandtl number raised to the 0.4 power $(hD/k)/(c_p\mu/k)^{0.4}$
is plotted against Reynolds number DG/μ with the physical proper-
ties of the water evaluated at average bulk temperatures.

The results of the entire set of data taken with the heater
tube in a horizontal position for a constant water entrance tempera-
ture of approximately 85° F are presented in figure 8. The correla-
tion line (solid) is drawn through the data and the transition line
(dashed) between laminar and turbulent flow lies in the range of
Reynolds numbers between 3000 and 9000. The data shown are for power
input values of 1 to 9 kilowatts (heat flux, 1190 to 10,710 Btu/(min)
(sq ft)) and entrance pressures of 10 to 200 inches of mercury gage.
All the points that lie considerably above the correlation line were
obtained when the average inside-tube-wall temperature exceeded the
boiling point of the liquid at the operating pressure and were due to
a portion of the water being in the nucleate-boiling region.

Removal of all the points for which the wall temperature is
greater than the saturation temperature results in a good correla-
tion for heating water with no boiling or change of phase (fig. 9).
The equation for the line that represents the data for Reynolds
numbers above approximately 10,000 is

$$\frac{\dfrac{hD}{k}}{\left(\dfrac{c_p\mu}{k}\right)^{0.4}} = 0.0168 \left(\frac{DG}{\mu}\right)^{0.84}$$

The line recommended by McAdams (reference 2) is included for
comparison in figure 9. Although the reference line is slightly
below the correlation line for the water data, the agreement is
considered to be satisfactory because the spread of data obtained
by other investigators is larger than the difference shown in
figure 9.

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All data obtained with the heater tube in a vertical position with the flow up and down are presented in figures 10 and 11, respectively. The correlation line through both sets of data is the same line that was drawn through the data obtained with the horizontal heater tube.

The results of one set of runs made at a constant power input of 7 kilowatts (8330 Btu/(min)(sq ft)), a water entrance pressure of 100 inches of mercury gage, and an entrance temperature of 85° F with the heater tube in a horizontal position are correlated in the conventional manner in figure 12(a) where Nusselt number divided by Prandtl number to the 0.4 power is plotted against Reynolds number. The basic correlation line for no boiling, obtained from figures 8 and 9, is included.

As the Reynolds number is decreased from its maximum value to about 16,000 by reducing the flow rate, the data fall on the non-boiling correlation line. At a Reynolds number of 16,000, the data break away, and with further reduction in Reynolds number, fall above the line. This effect can be explained as follows: For constant power input, as Reynolds number is first decreased from its maximum value, the heat-transfer coefficient decreases in the normal manner and, consequently, the difference between the temperatures of the tube wall and of the fluid must increase. The variation of heat-transfer coefficient and temperature difference between wall and fluid with Reynolds number is shown in figure 12(b) for the data of figure 12(a). As the Reynolds number (or flow rate) is decreased, the temperature rise of the fluid and hence the average fluid temperature must increase to maintain constant power input to the tube. The increases in fluid temperature and in the difference between wall and fluid temperatures result in an increase in tube-wall temperature. The variation of average wall temperature and average fluid temperature with Reynolds number is illustrated in figure 12(c). For the data shown, the tube-wall temperature is equal to the boiling temperature of the water at a Reynolds number of 16,000. At this point, the heat-transfer coefficient starts to increase, with a corresponding reduction in the difference between wall and fluid temperature (fig. 12(b)). This increase in heat-transfer coefficient is associated with nucleate-type boiling wherein small bubbles are formed on and break away from the tube surface carrying large quantities of heat into the main fluid stream. These bubbles of steam also increase the turbulence of the liquid, which tends further to increase the heat-transfer coefficient. The increase in heat-transfer coefficient in the nucleate-boiling region results in a rapid increase in the average fluid temperature and retards the rate of increase in wall temperature, as shown in figure 12(c). For

further reduction in Reynolds number below the minimum value shown,
the bubbles tend to form faster than they break away from the wall,
resulting in an insulating vapor film at the wall that decreases
the heat-transfer coefficient and causes the wall temperature to
increase at a rapid rate, leading to eventual instability of flow
and possible tube burnout. The end points on the curves of
figure 12 correspond to the limit of stable operation for the
conditions given.

In figure 13, two sets of runs are presented for constant
power inputs of 3 and 9 kilowatts. For the 3-kilowatt series (heat
flux, 3570 Btu/(min)(sq ft)) in the Reynolds number range from
80,000 to 10,000, the points lie on the correlation line; below
10,000, the points fall above the line. As previously stated, the
breakaway point on each pressure line occurs when the wall tempera-
ture of the tube is equal to the boiling temperature of the water
at the operating pressure. The entrance temperature for each of
the four pressures presented is approximately constant at 85° F;
thus, the higher the pressure, the greater percentage of heat input
that is used for heating the water to its boiling condition. For
this constant entrance temperature condition, higher wall tempera-
tures are possible as the pressure is increased. Higher wall tem-
peratures result in lower Reynolds numbers before instability of
flow occurs. Similar results are shown in figure 13(b) for the
9-kilowatt series (heat flux, 10,710 Btu/(min)(sq ft)).

The effect of pressure on boiling is illustrated in figure 14.
The entrance temperature for each pressure was increased by an
amount equal to the change in the boiling temperature of the water
with increased pressure. An entrance temperature of approximately
85° F was used for the run with an entrance pressure of 10 inches
of mercury gage. A reversed trend is shown with increased pressure
from that obtained in figure 13. If, as in this case, the differ-
ence between entrance and boiling temperatures is made nearly equal
for each of the pressures, a decrease in pressure causes a delay in
the start of nucleate boiling; increased pressure, however, gives
higher heat-transfer coefficients for equal Reynolds numbers in the
nucleate-boiling region.

A satisfactory method of correlating the data for which nucleate
boiling occurs has not been obtained; figure 15 shows, however, the
boiling data taken thus far and includes data for both the vertical
and horizontal heater tubes. The entrance temperature for all the
runs shown is approximately 85° F. Reynolds number is plotted against
excess temperature, defined as the difference between the average

inside-wall temperature and the boiling point of the water. Data
are shown for entrance pressures of 10, 50, 100, and 200 inches of
mercury gage. Separate curves are given for each value of heat
flux. For each constant power input as Reynolds number is decreased,
the excess temperature increases. This excess-temperature increase
indicates that as the wall temperature is increased the average
liquid temperature is also increased because of the reduction in
flow rate. The increase in liquid temperature is sufficient to give
a decrease in temperature difference between the wall and liquid, as
previously shown in figure 12. A dash-dot curve representing the
approximate limit of stable operation is drawn through the data for
each pressure level. Maximum values of excess temperature of about
70° were obtainable in the nucleate-boiling region. As previously
stated, unstable flow is obtained in the film-boiling region beyond
these operating limit curves.

Two principal problems encountered in the heat-transfer process
are: (1) determination of a method of predicting the heat-transfer
properties of water after boiling starts, and (2) determination of
the safe operating range in the nucleate-boiling region. Although
no satisfactory correlation of boiling data has been obtained, it is
shown in figure 15 that a safe excess temperature of about 40° F may
be assumed for the range of conditions used and with a water entrance
temperature of approximately 85° F. A plot for predicting the begin-
ning of nucleate boiling for the same operational limits as presented
in figure 15 is therefore presented in figure 16. The curves are
based on the basic correlation line obtained in the investigation and
the calculated values of inside-tube-wall temperature are plotted
against mass velocity for four constant heat-flux values. Lines repre-
senting the boiling temperature for the four pressures are superimposed
on the wall-temperature curves. The point of intersection of each
boiling temperature and heat-flux curve is the condition at which
nucleate boiling starts. The portion of the heat-flux curve below the
intersection represents no boiling; the heat-flux curves above the
intersection are in the nucleate-boiling phase.

Another method of presenting the boiling data is shown in fig-
ure 17. Experimental values of $(hD/k)/(c_p\mu/k)^{0.4}$ were divided by
corresponding values of $0.0168(DG/\mu)^{0.84}$ and plotted against the
excess temperature $T_s - T_{bp}$. Negative values of excess temperature
result in a value of unity for the ordinate. In figure 17(a), data
are presented for a constant entrance pressure of 100 inches of mer-
cury gage and a range of power inputs from 1 to 9 kilowatts. For
excess temperatures greater than zero (boiling starts at zero), the
data points lie above a value of 1.0. Values of the ordinate as high

as 2.1 are obtained before the flow becomes unstable. As the power input is increased, higher values of excess temperature are required for a given increase in the ordinate. An operating limit curve is drawn through the data to show that increases in the value of the ordinate are obtainable as the power input is reduced before instability occurs.

The effect of pressure is shown on a similar plot in figure 17(b), where data are presented for a constant power input of 7 kilowatts and a range of entrance pressures. The data show that the highest excess temperature is obtained with the lowest pressure and, from figure 17(a), with the highest power input.

SUMMARY OF RESULTS

The results of the heat-transfer investigation conducted with water flowing in a horizontal plane and up and down in a vertical plane through an electrically heated Inconel tube with an inside diameter of 0.204 inch and a length-diameter ratio of 50 over a range of Reynolds numbers from 2000 to 100,000 showed that:

1. Correlation of average heat-transfer coefficients (for non-boiling conditions) according to the familiar Nusselt relation wherein physical properties were evaluated at an average bulk temperature was obtained and was in close agreement with previous investigations.

2. Data were obtained in the nucleate-boiling region and were shown for various power inputs and entrance pressures by the use of an excess-temperature factor, which is defined as the difference between the average inside-tube-wall temperature and the boiling temperature of the liquid at the operating pressure.

3. Nucleate boiling started when the average inside-wall temperature was equal to the boiling temperature of the water. Above this temperature, higher values of heat-transfer coefficient were obtained than those predicted by the correlation equation.

4. Maximum values of excess temperature of about 70° were obtainable in the nucleate-boiling region. Instability of flow occurred at higher values but no data were obtained for this region.

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    National Advisory Committee for Aeronautics,
        Cleveland, Ohio.

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    of Aqueous Ethylene Glycol Solutions in an Electrically
    Heated Tube. NACA ARR E5F07, 1945.

2. McAdams, William H.: Heat Transmission. McGraw-Hill Book Co.,
    Inc., 2d ed., 1942, p. 168.
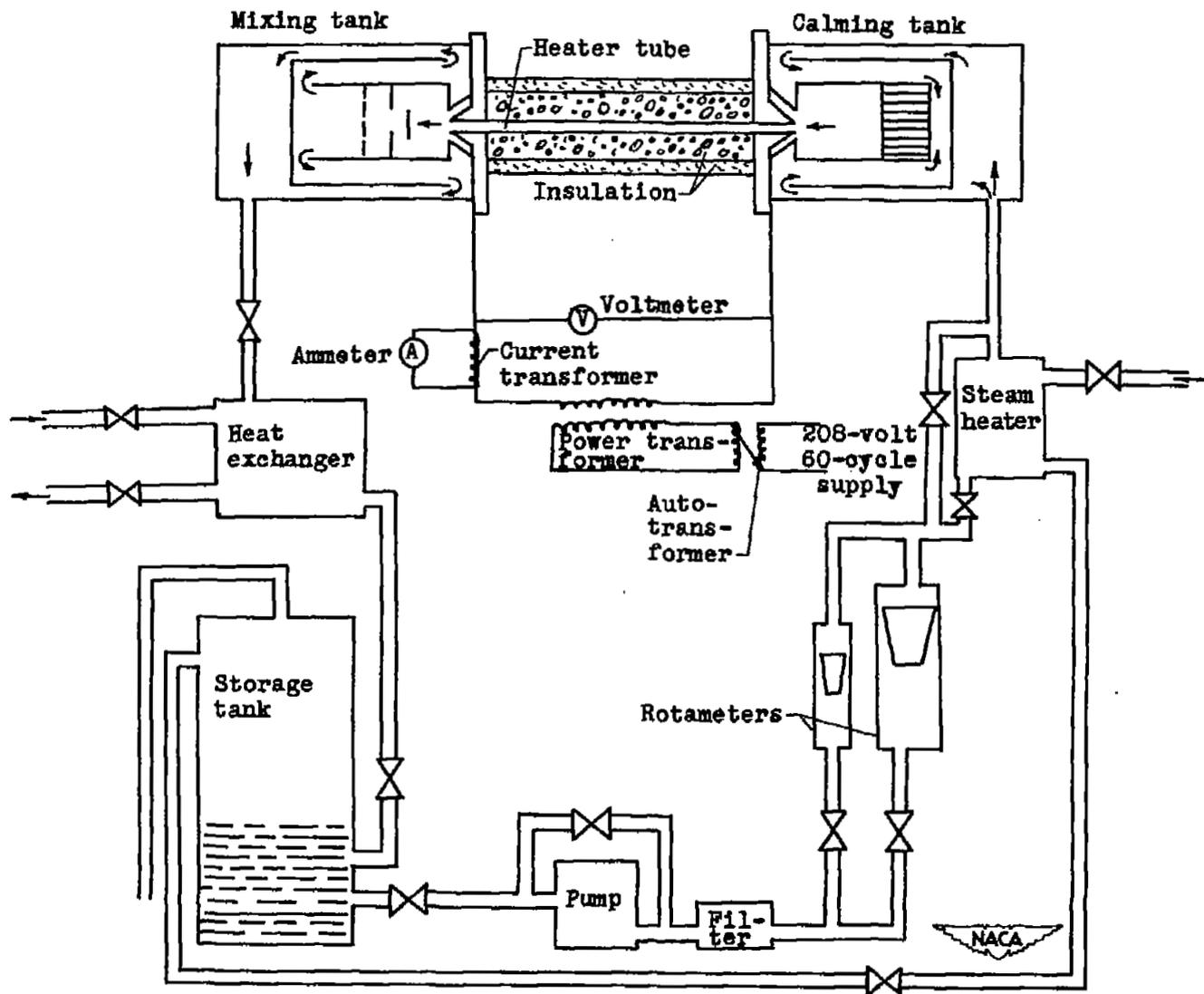
Figure 1. - Schematic diagram showing arrangement of experimental apparatus.

Voltmeter
leads

1/2" 1/2"   1"   1"

Insulation   Radiation
                    shields

x Thermocouples

NACA
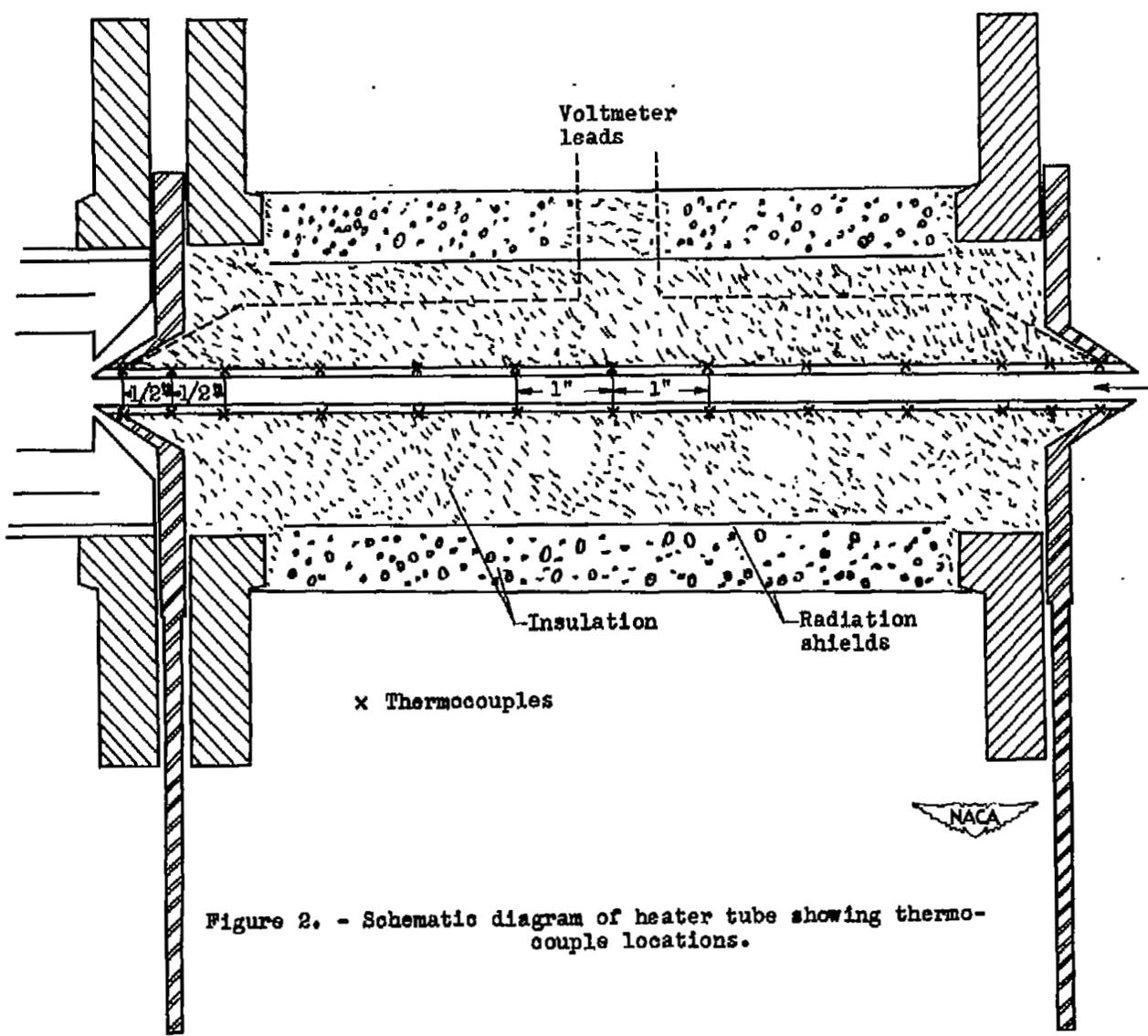
Figure 2. - Schematic diagram of heater tube showing thermo-
couple locations.
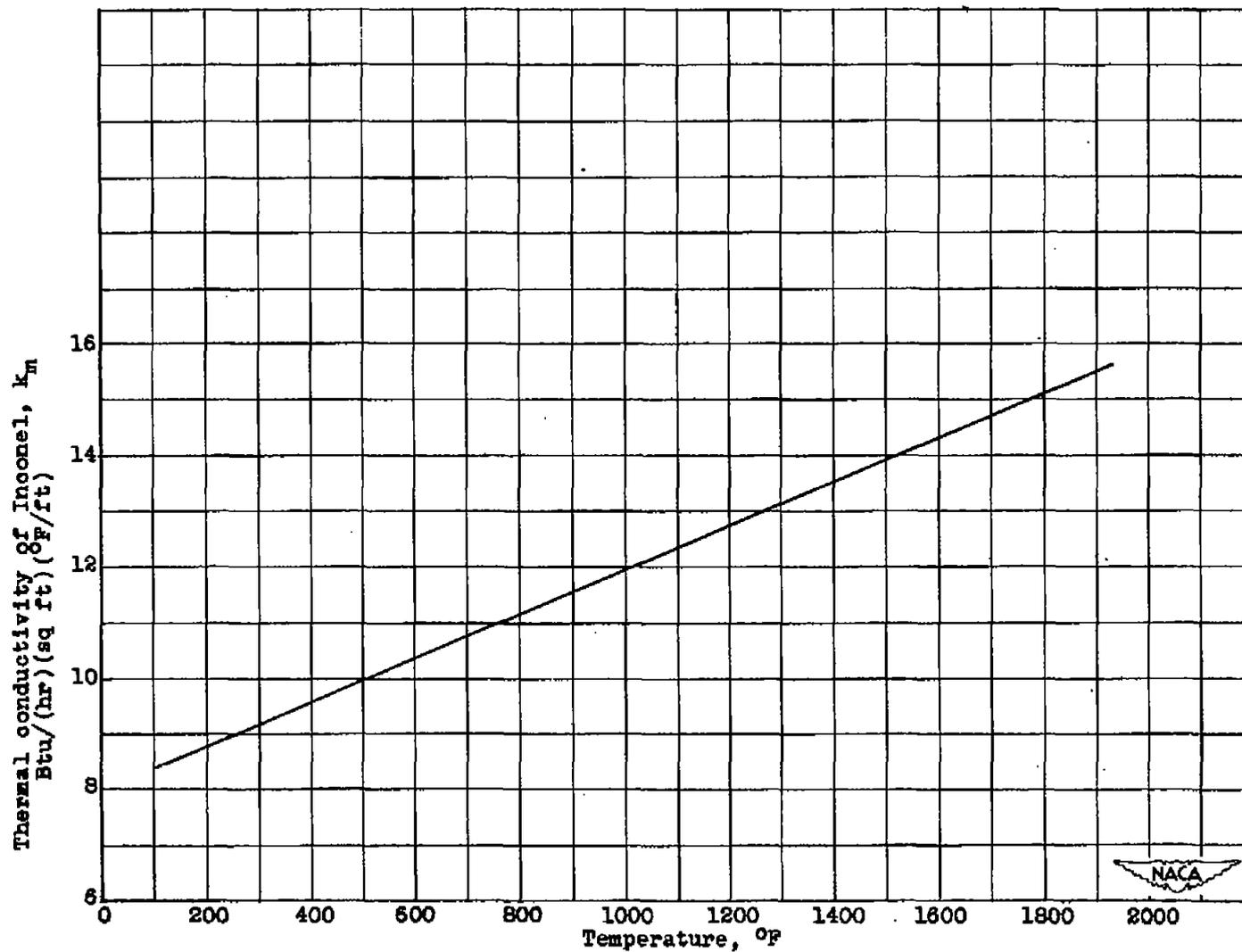
Figure 3. - Variation of thermal conductivity of Inconel with temperature.
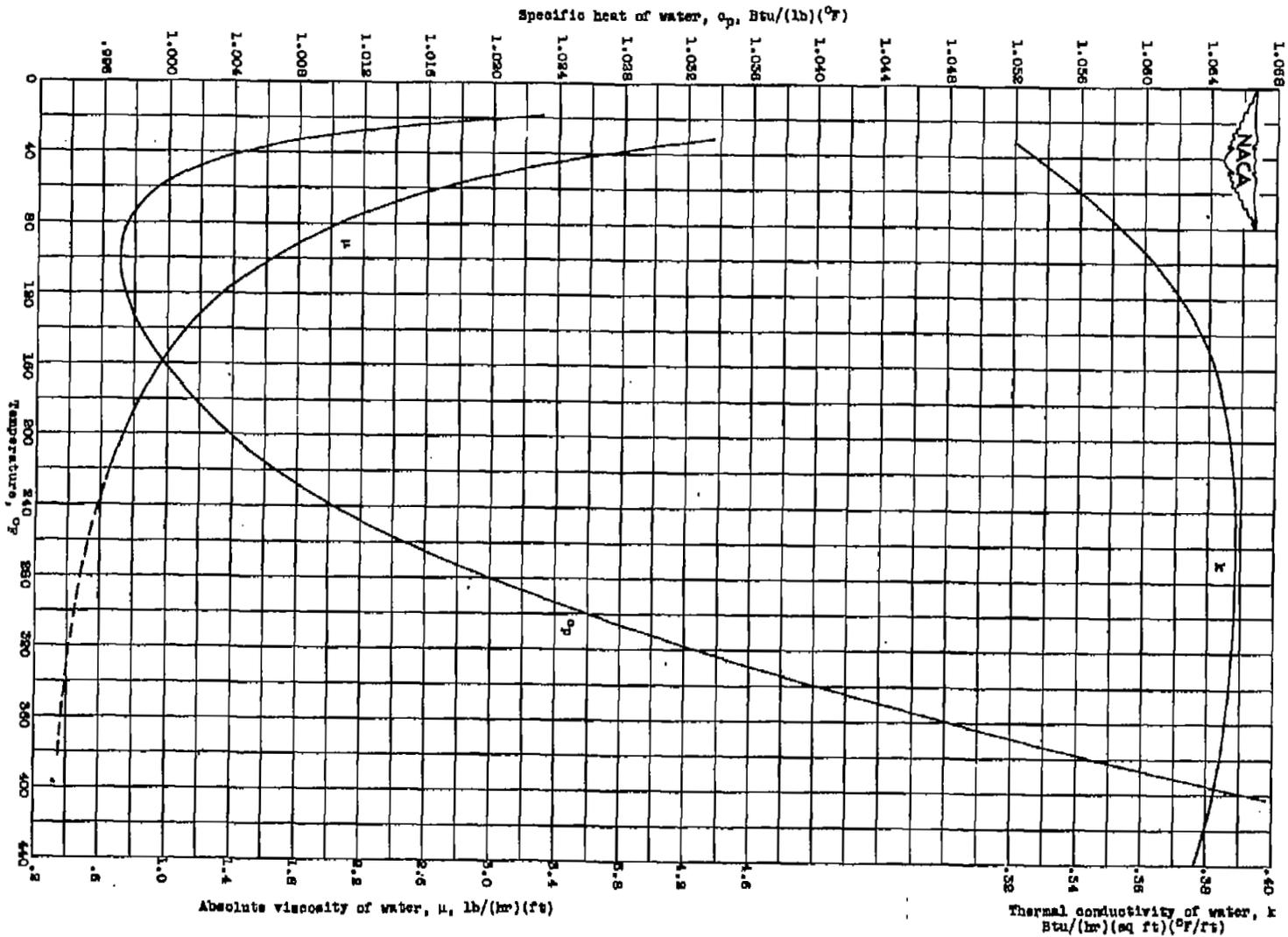
Figure 4. - Variation of physical properties of water with temperature.
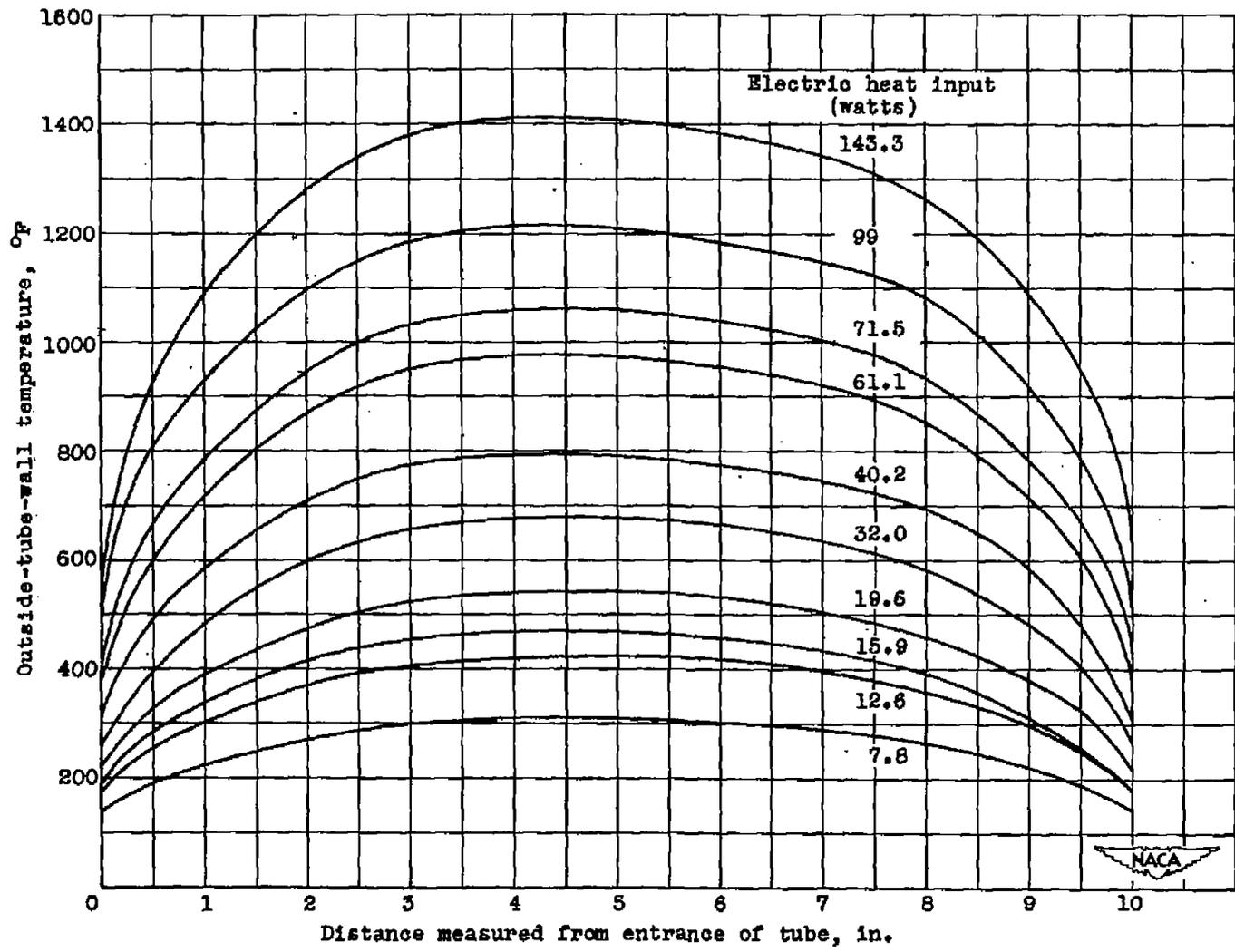
Figure 5. - Axial distribution of outside-tube-wall temperature for various amounts of electric heat input without water in tube.
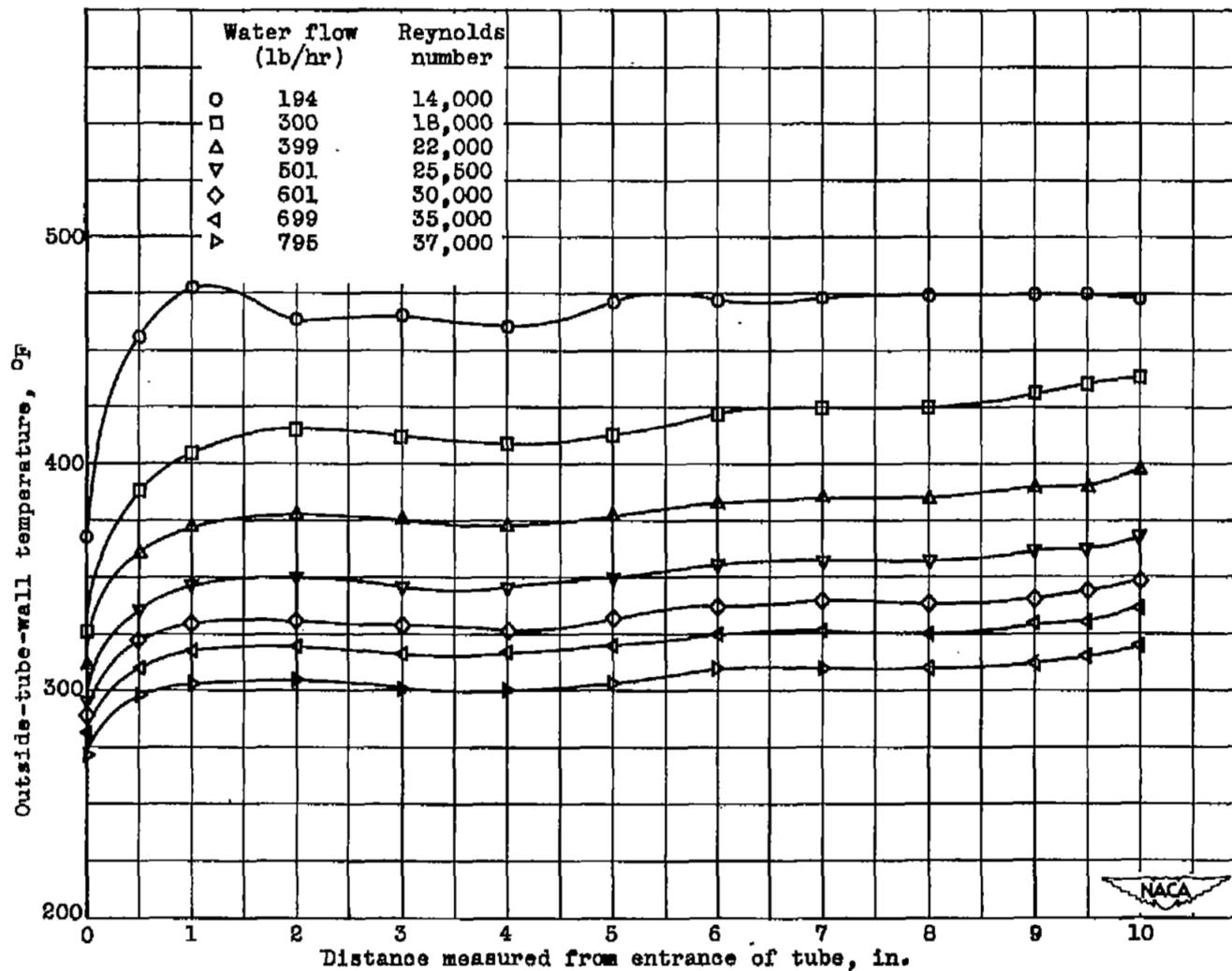
Figure 6. - Representative outside-tube-wall temperature distribution for various water flow rates. Constant power input, 7 kilowatts; inlet pressure, 200 inches of mercury gage.
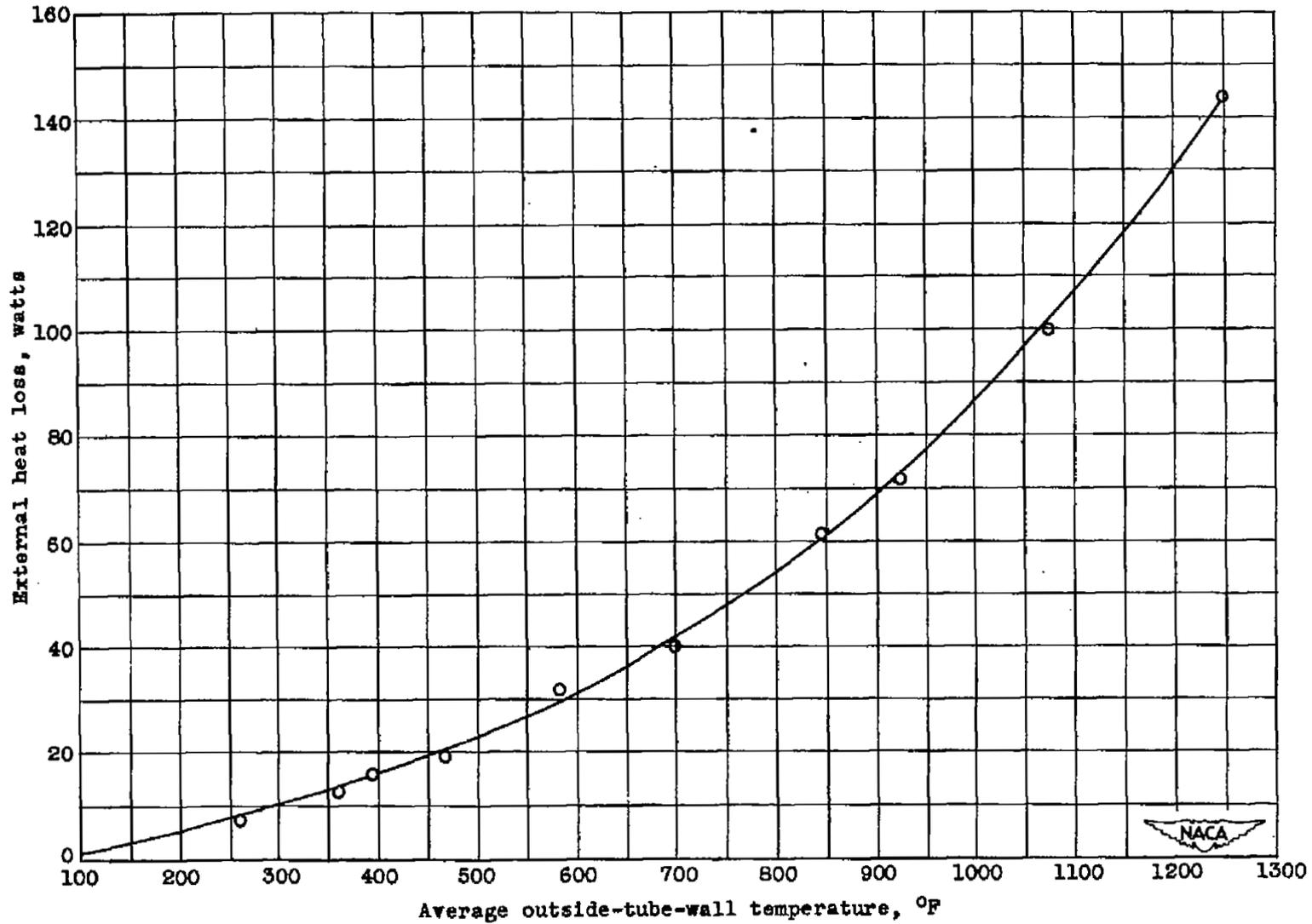
Figure 7. - Variation of external heat loss with average outside-tube-wall temperature.
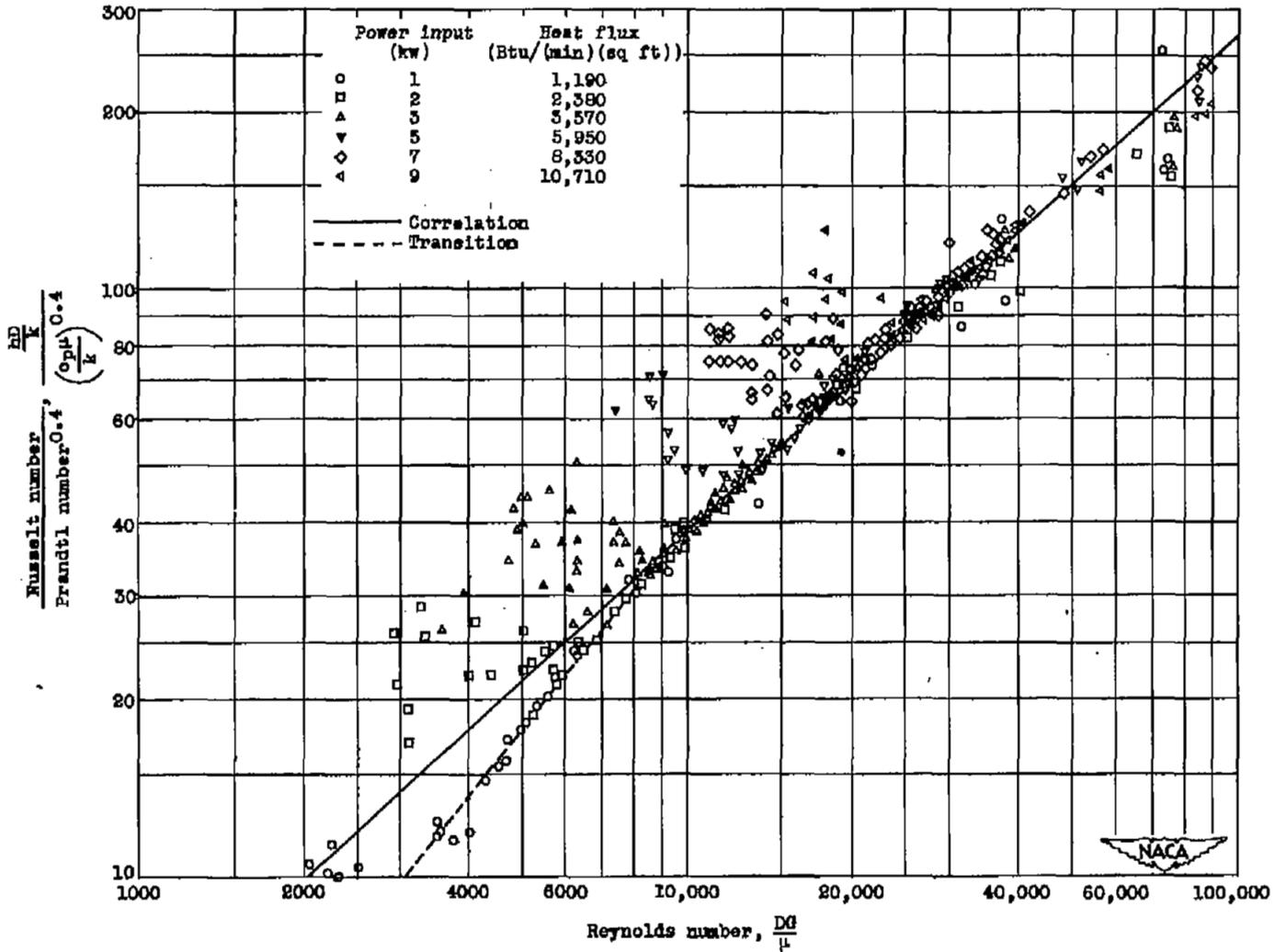
Figure 8. – Correlation of heat-transfer coefficients for horizontal tube. Physical properties of water evaluated at average bulk temperature; water entrance pressure, 10, 50, 100, and 200 inches of mercury gage.
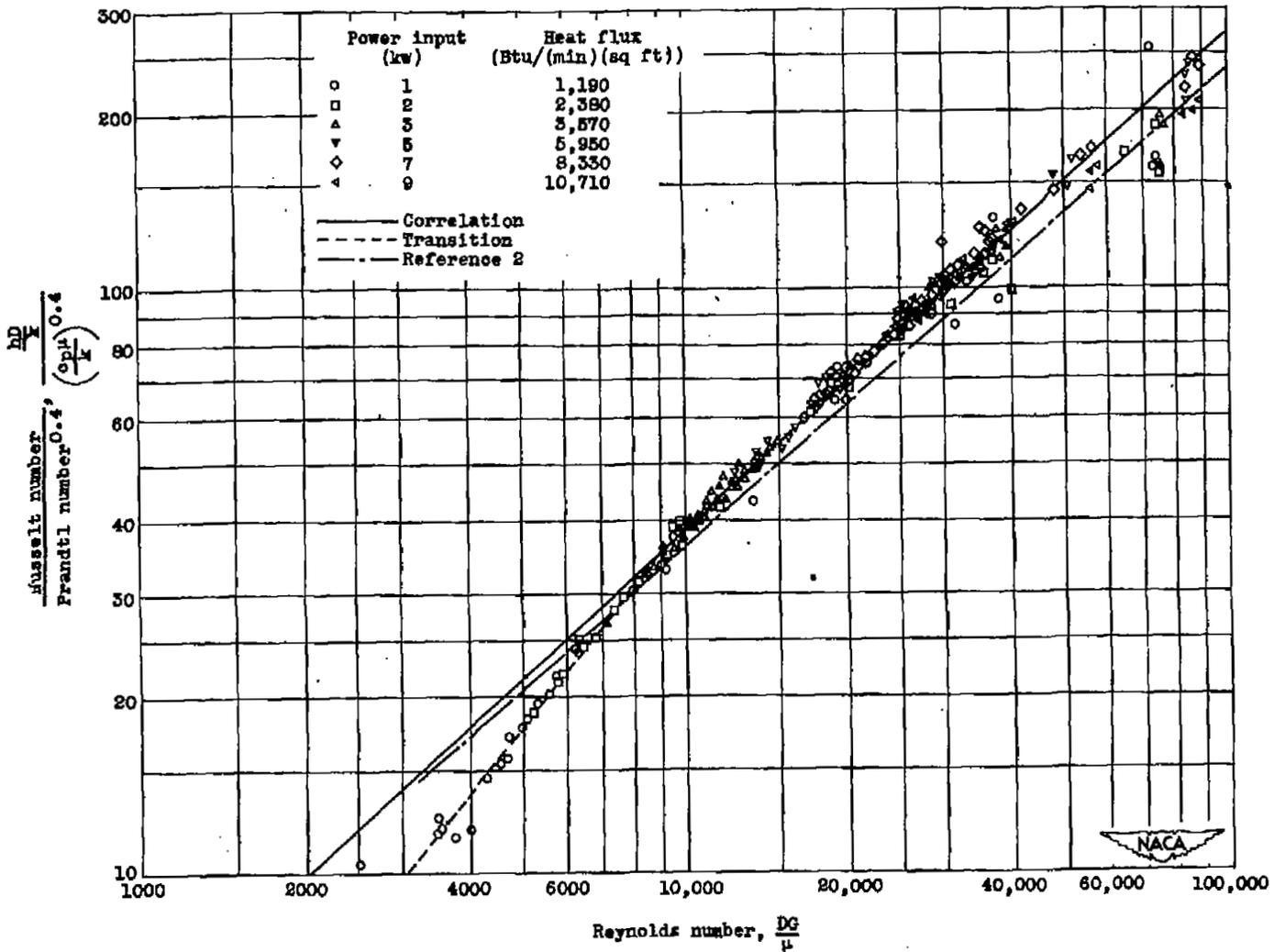
Figure 9. - Correlation of heat-transfer coefficients for horizontal tube with all nucleate-boiling points removed from plot. Physical properties of water evaluated at average bulk temperature; water entrance pressure, 10, 50, 100, and 200 inches of mercury gage.
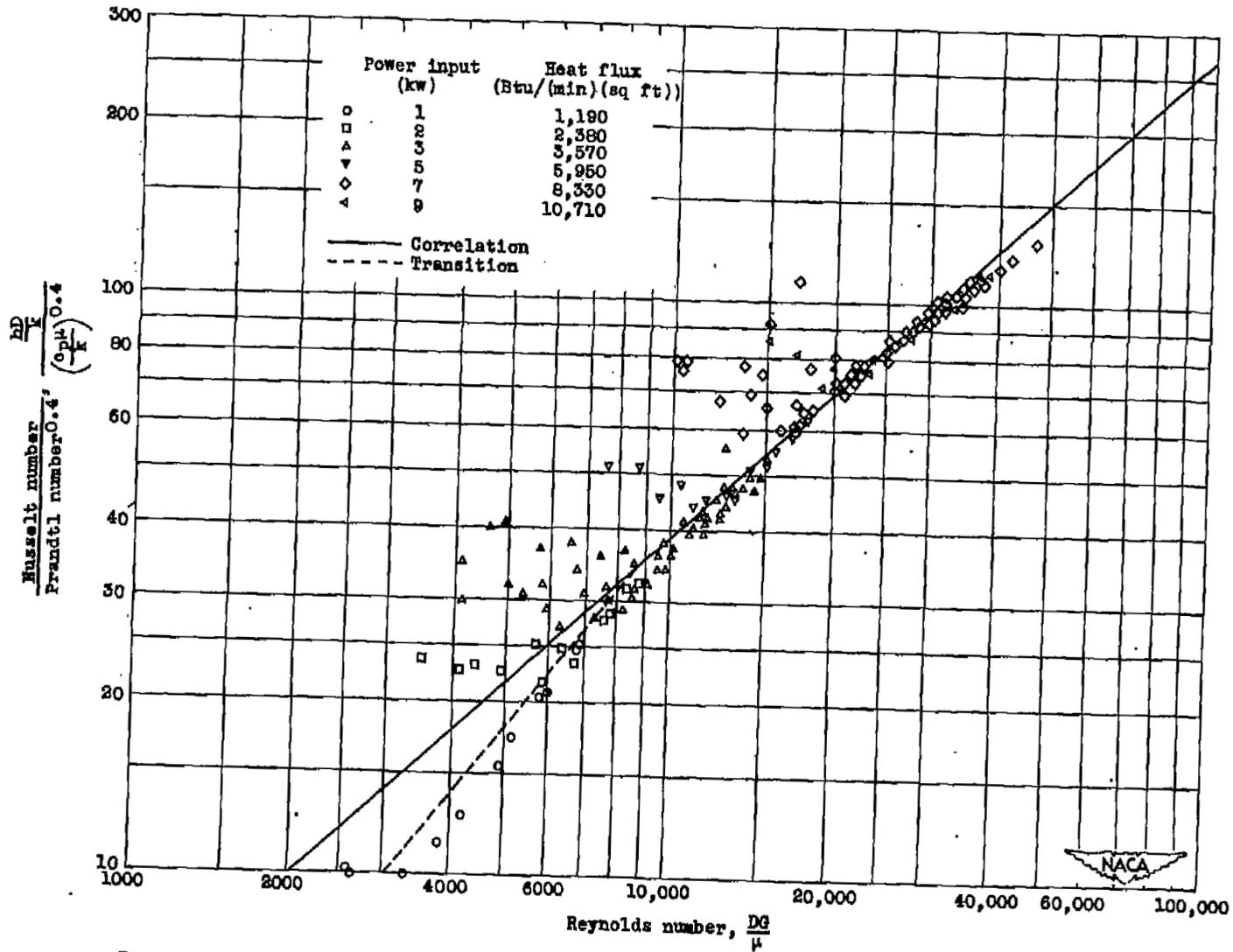
Figure 10. - Correlation of heat-transfer coefficients for vertical tube with flow up. Physical properties of water evaluated at average bulk temperature; water entrance pressure, 10, 50, 100, and 200 inches of mercury gage.
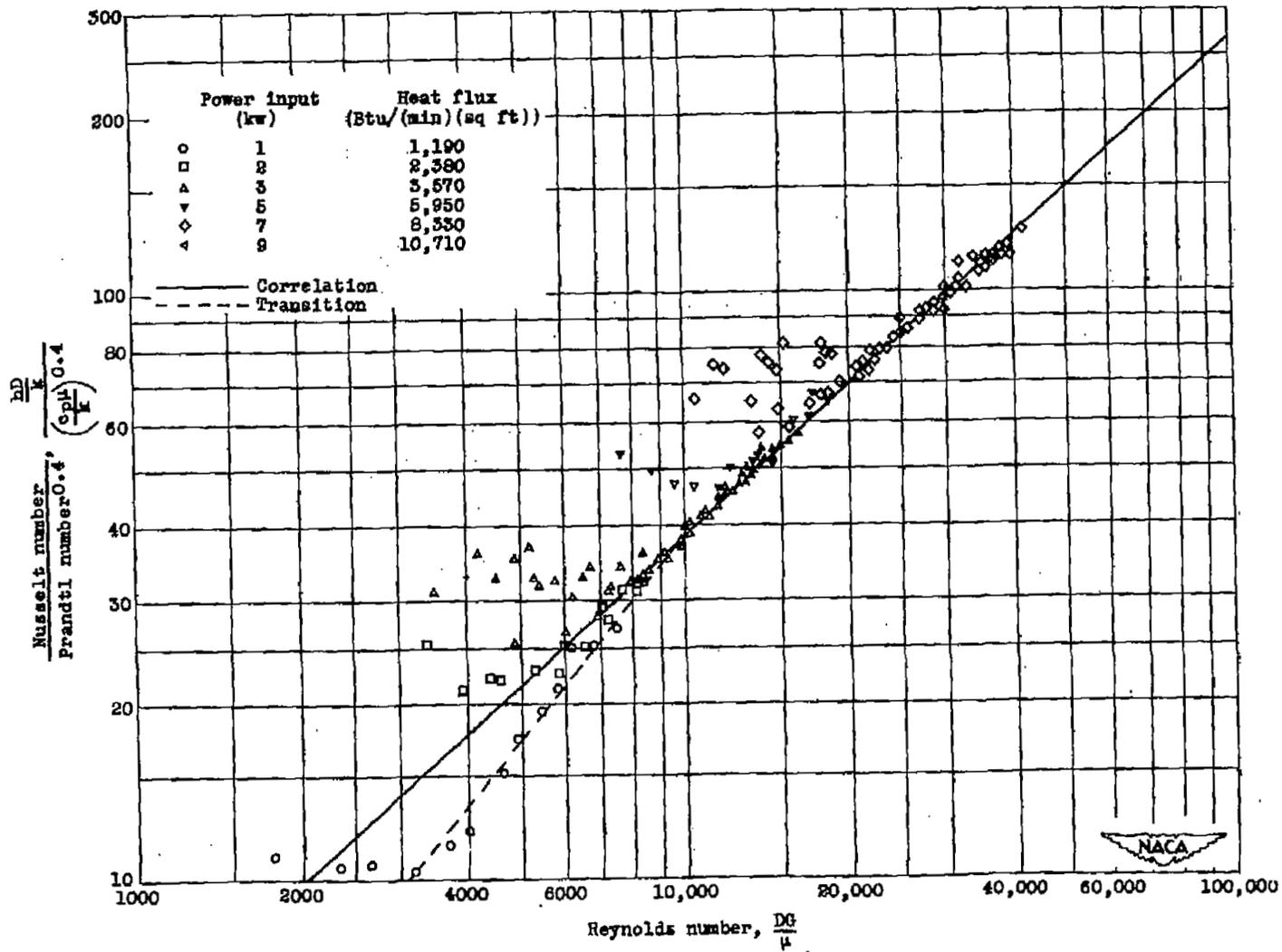
Figure 11. - Correlation of heat-transfer coefficients for vertical tube with flow down. Physical properties of water evaluated at average bulk temperature; water entrance pressure, 10, 50, 100, and 200 inches of mercury gage.
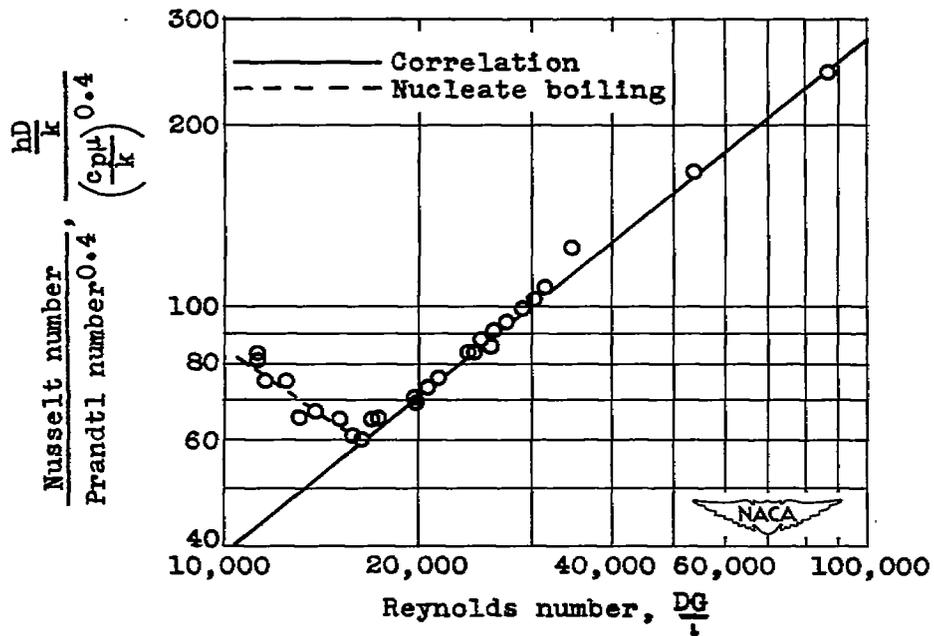
(a) Correlation of heat-transfer
     coefficients with physical properties
     of water evaluated at average bulk
     temperature.

Figure 12. - Example of horizontal-tube
     data plotted for power input of 7 kilo-
     watts and water entrance pressure of
     100 inches of mercury gage.
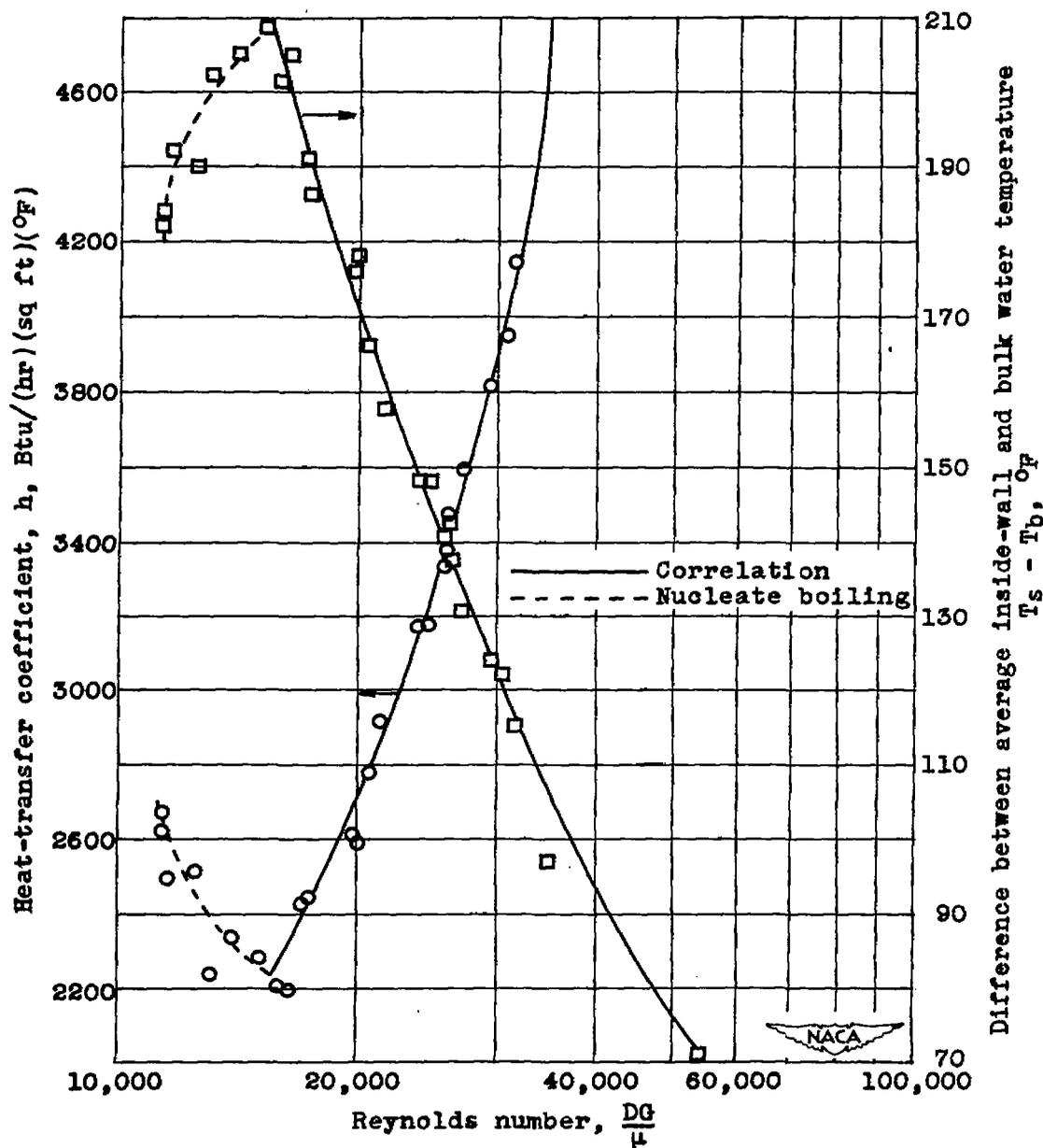
(b) Variation of heat-transfer coefficient and
    temperature difference between inside-wall and
    average bulk water temperature with Reynolds
    number.

Figure 12. - Continued. Example of horizontal-tube
    data plotted for power input of 7 kilowatts and
    water entrance pressure of 100 inches of mercury
    gage.
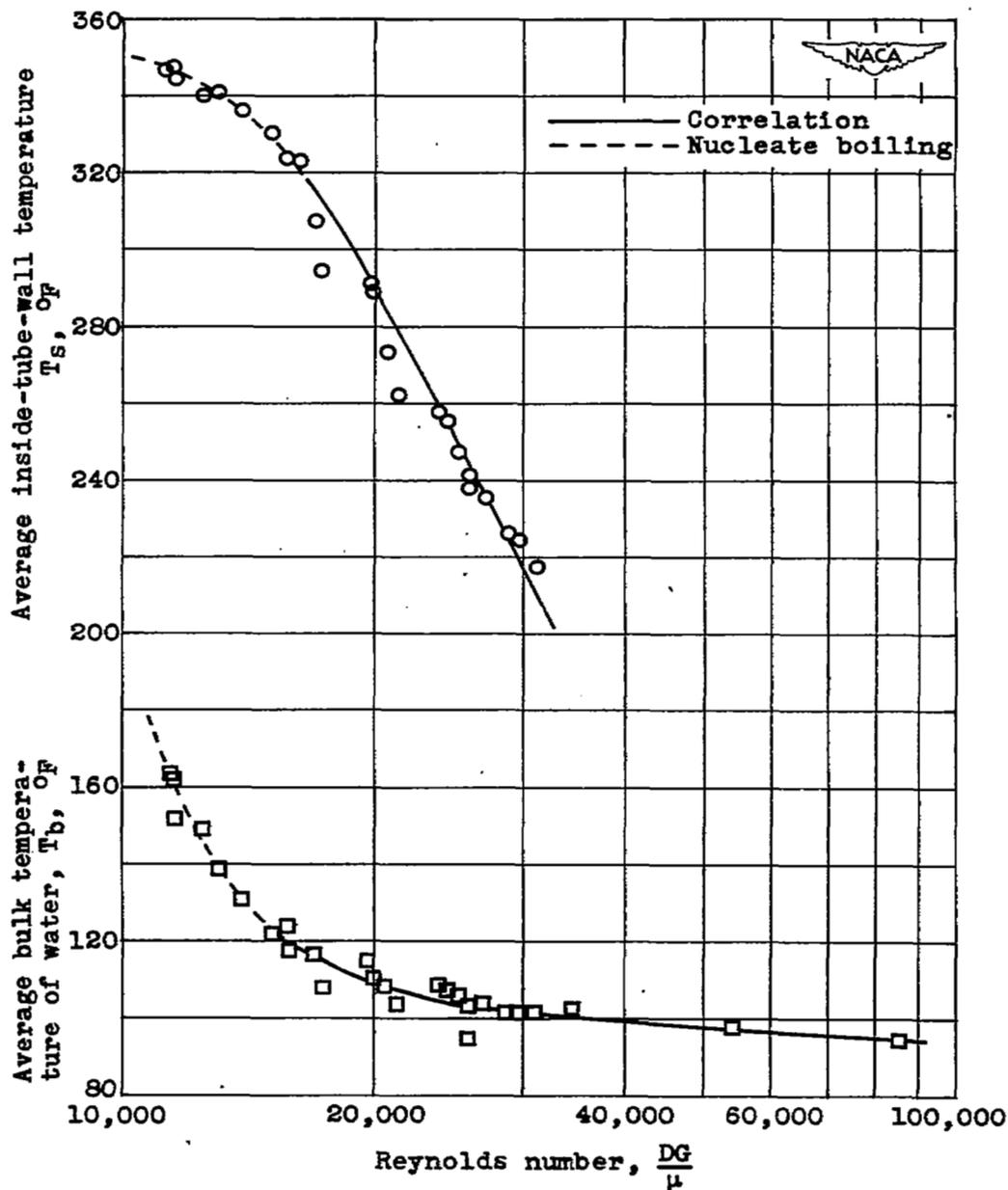
(c) Variation of inside-tube-wall temperature and average bulk water temperature with Reynolds number.

Figure 12. - Concluded. Example of horizontal tube data plotted for power input of 7 kilowatts and water entrance pressure of 100 inches of mercury gage.
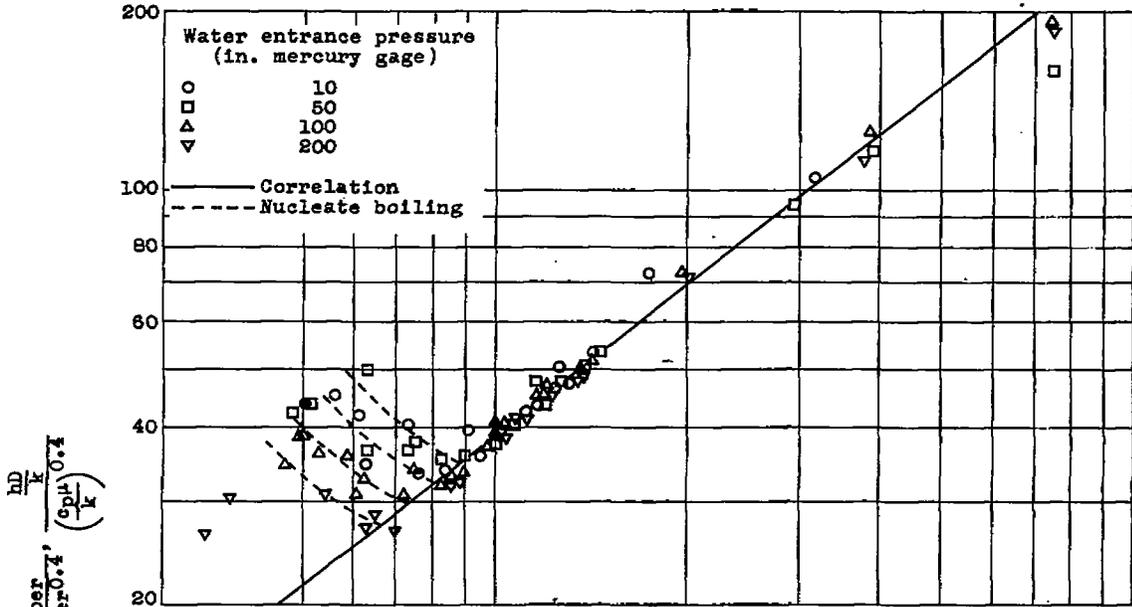
(a) Power input, 3 kilowatts (heat flux, 3570 Btu/(min)(sq ft)).
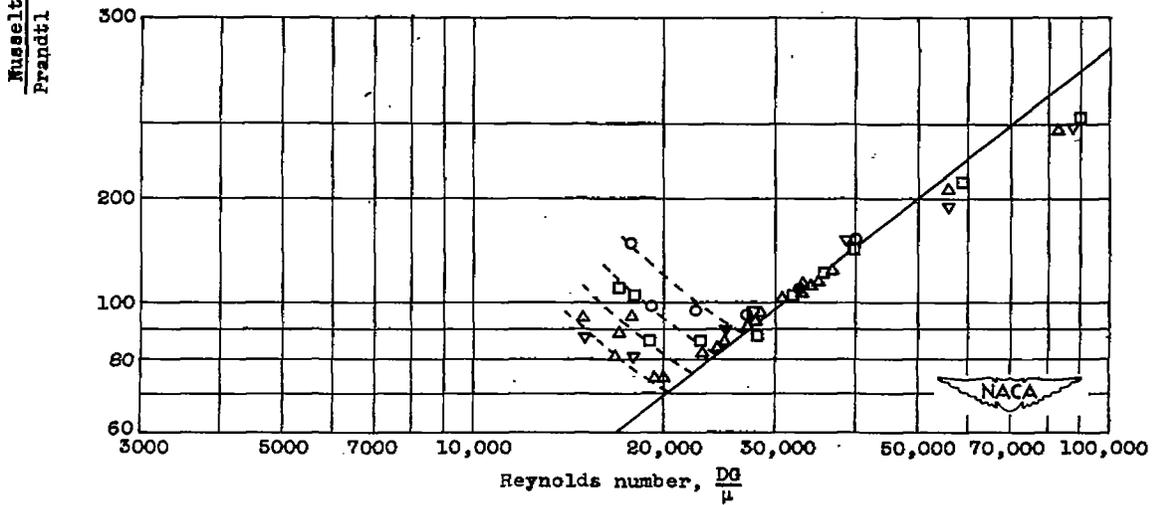
(b) Power input, 9 kilowatts (heat flux, 10,710 Btu/(min)(sq ft)).

Figure 13. - Correlation of heat-transfer coefficients for horizontal tube.
Physical properties of water evaluated at average bulk temperature.
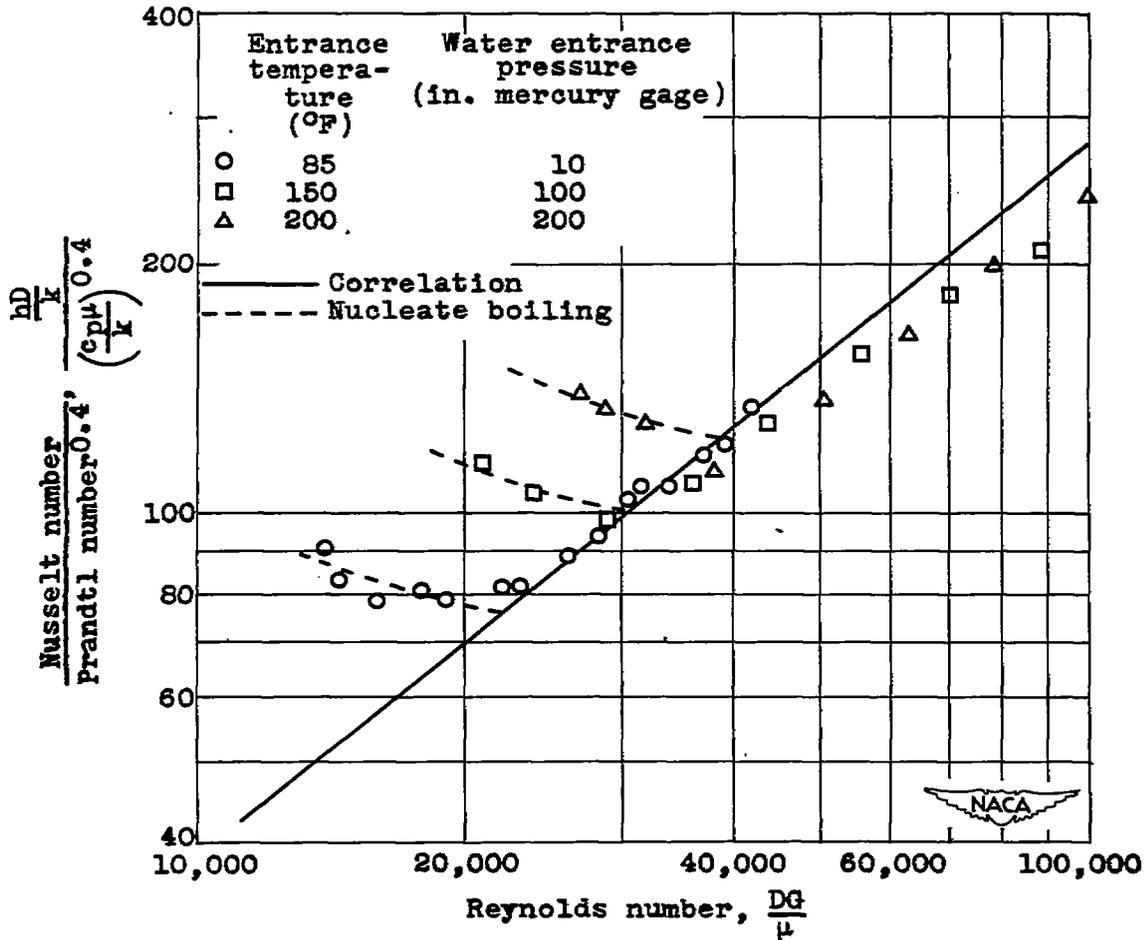
Figure 14. - Correlation of heat-transfer coefficients for various water entrance pressures where entrance temperature is increased to equal the difference in boiling temperature as the pressure is increased. Power input, 7 kilowatts; horizontal tube.
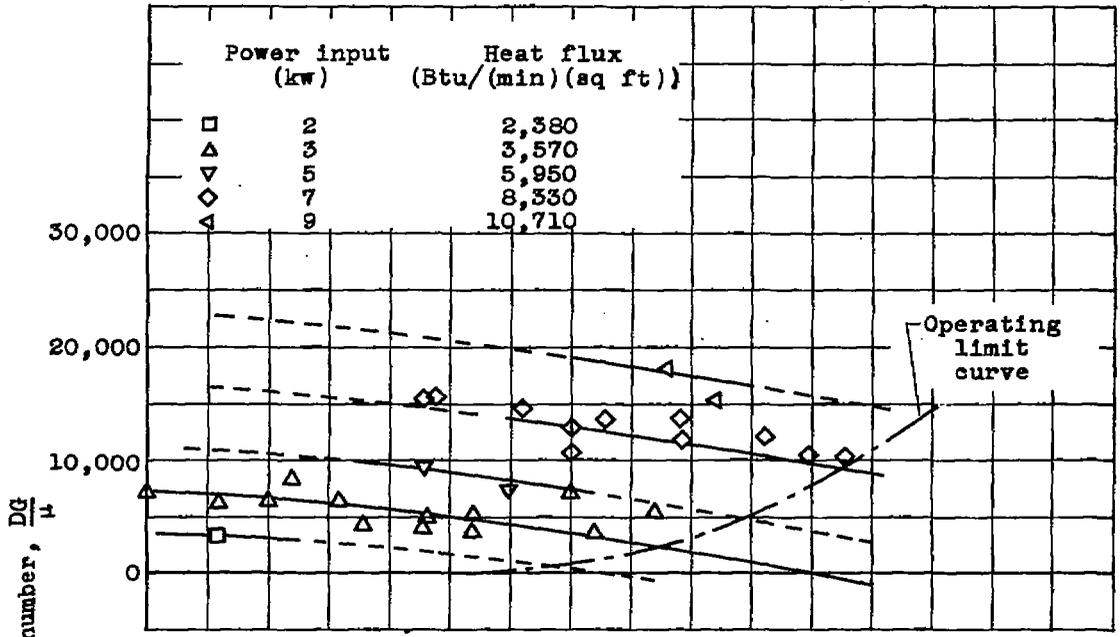
(a)   Water entrance pressure, 200 inches of mercury gage.
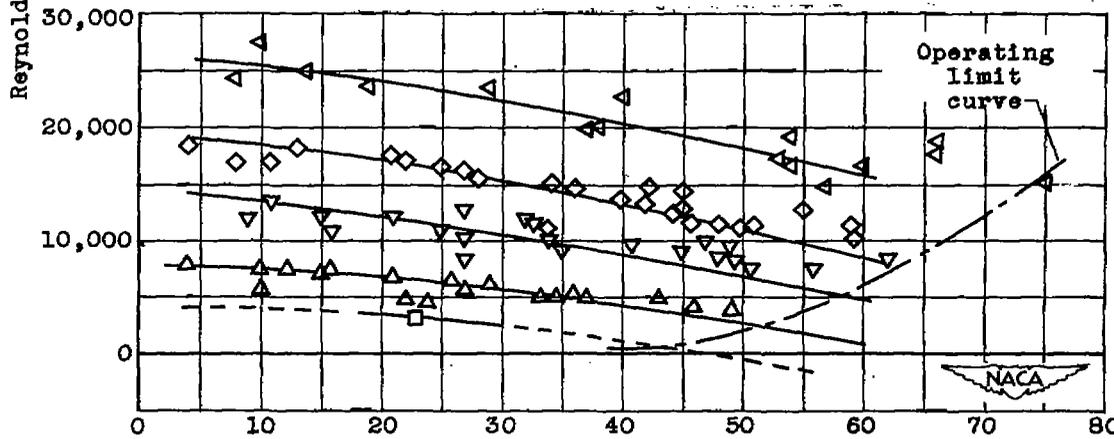
(b)   Water entrance pressure, 100 inches of mercury gage.

Figure 15. - Variation of excess temperature with Reynolds number
for various power inputs and entrance pressures.
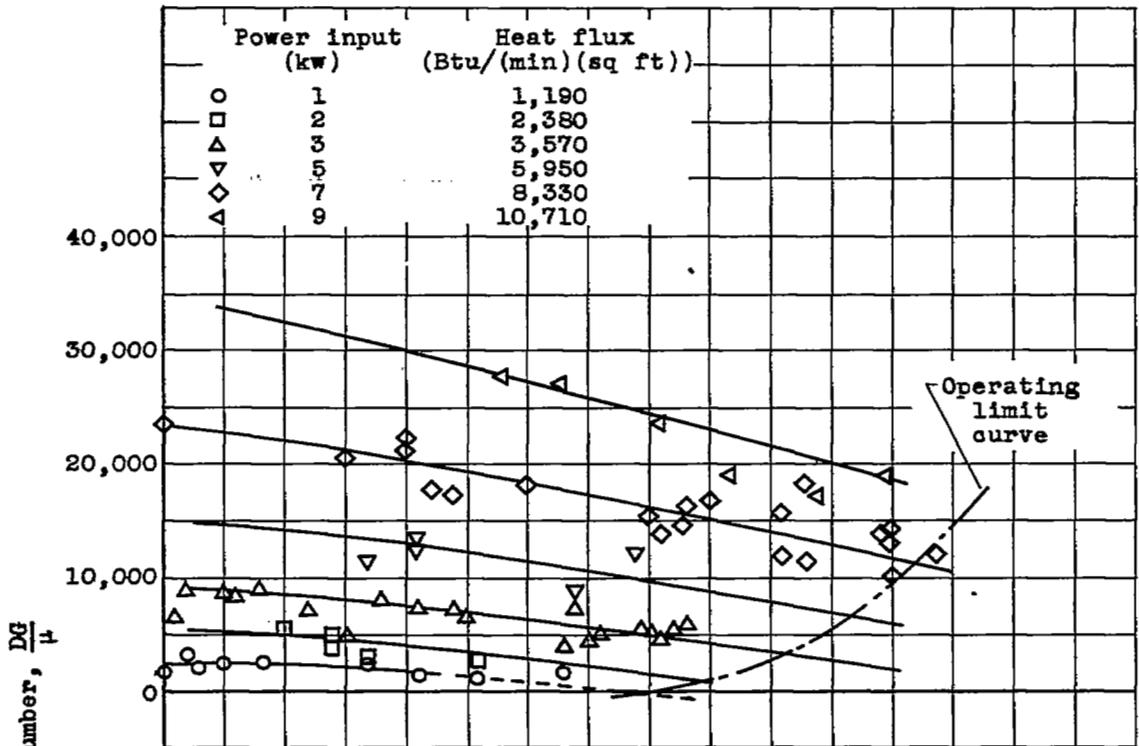
(c) Water entrance pressure, 50 inches of mercury gage.
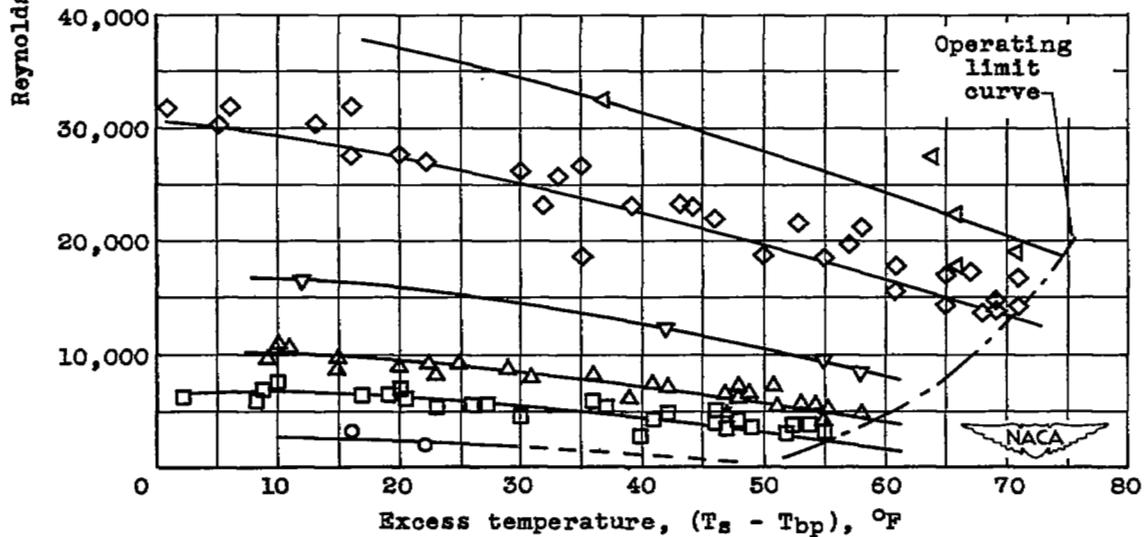
(d) Water entrance pressure, 10 inches of mercury gage.

Figure 15. - Concluded. Variation of excess temperature with
Reynolds number for various power inputs and entrance pressures.
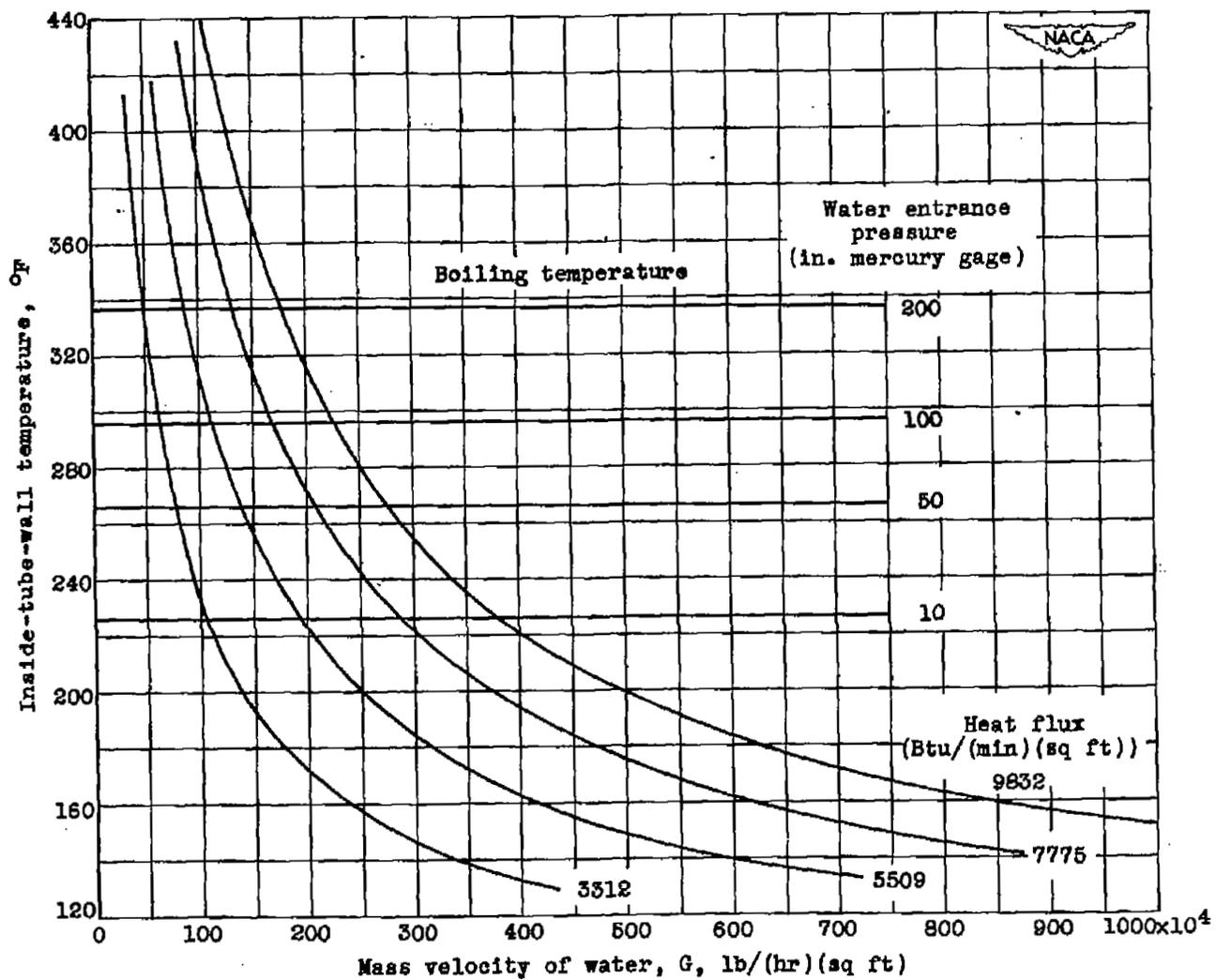
Figure 16. - Calculated variation of inside-tube-wall temperature with mass velocity for various heat-flux rates.
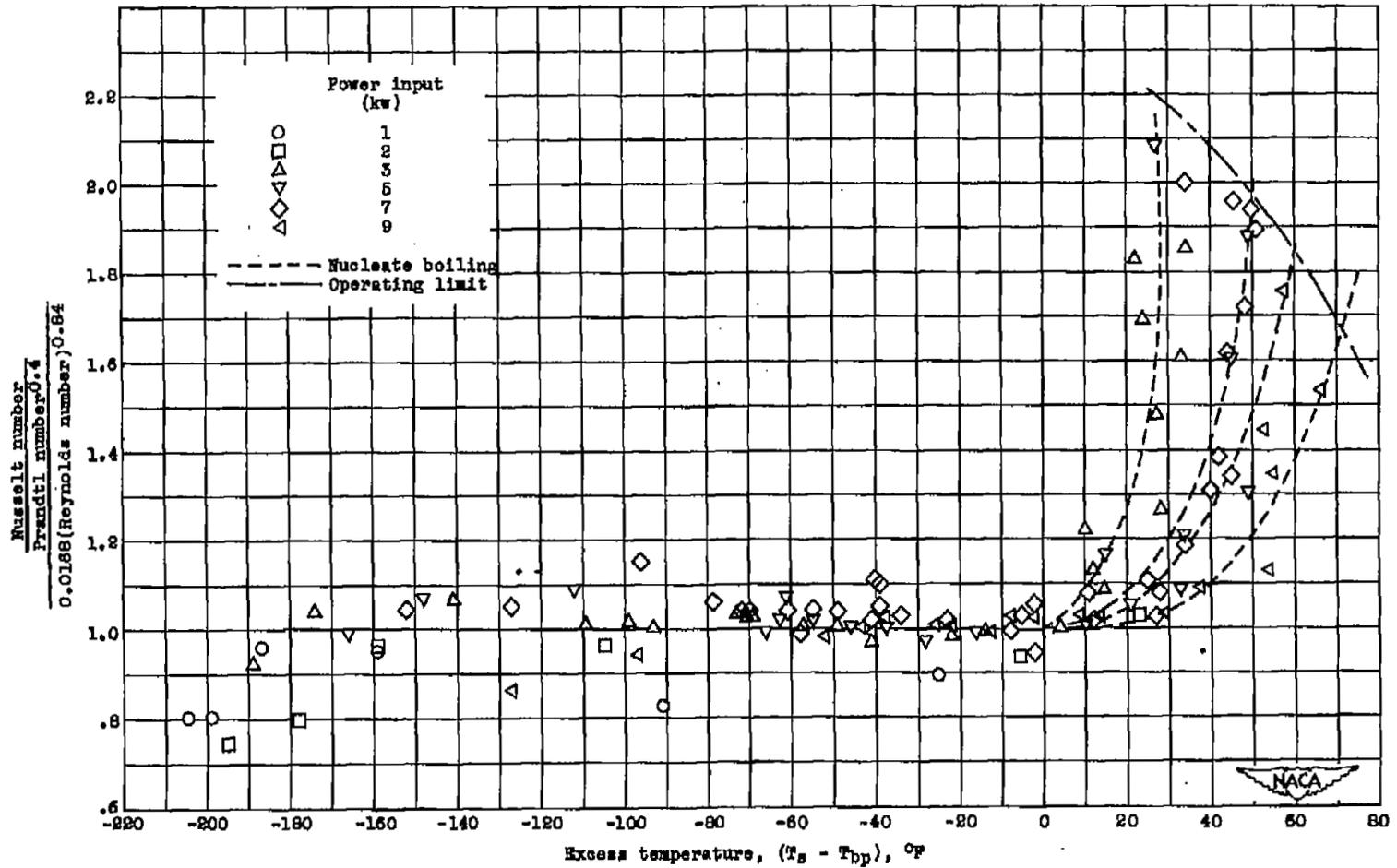
(a) Water entrance pressure, 100 inches of mercury gage.

Figure 17. - Variation of correlation equation with excess temperature in nonboiling and nucleate-boiling regions. Horizontal tube.
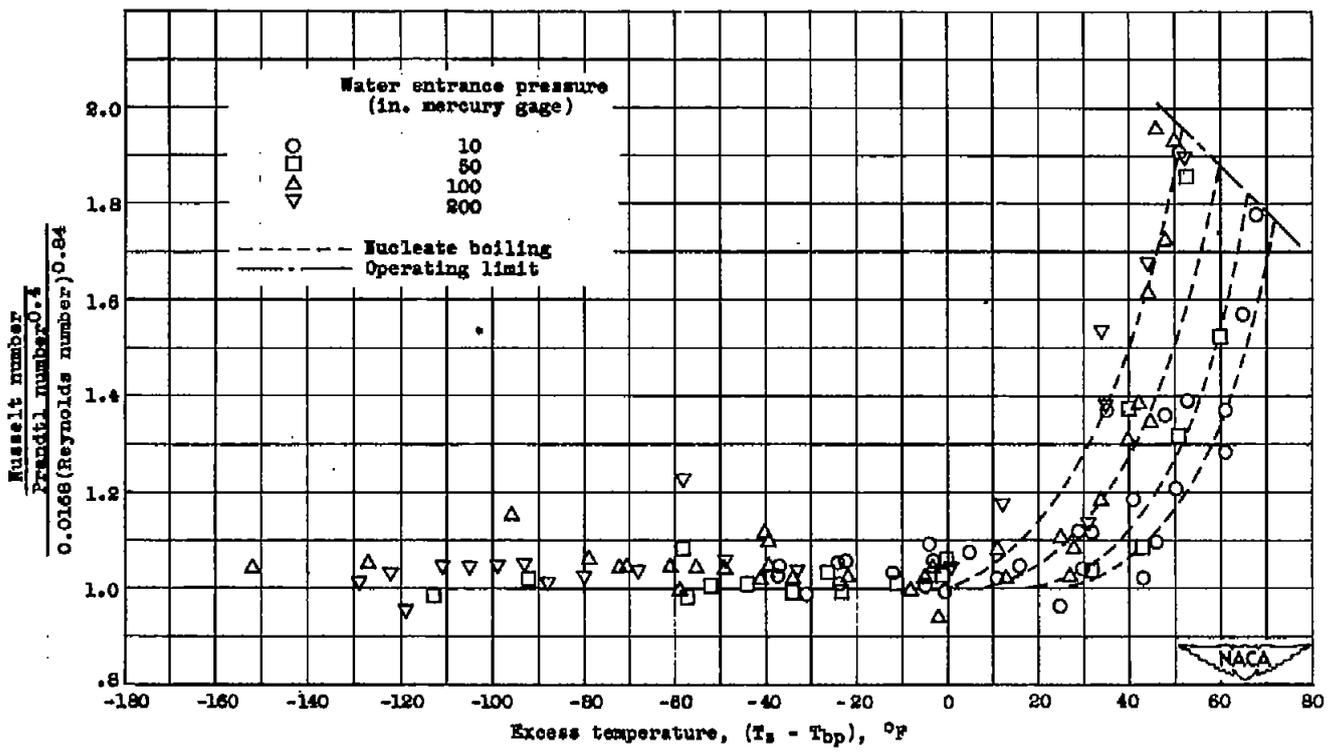
(b) Power input, 7 kilowatts.

Figure 17. — Concluded. Variation of correlation equation with excess temperature in nonboiling and nucleate-boiling regions. Horizontal tube.