RESEARCH MEMORANDUM

PRELIMINARY INVESTIGATION OF HEAT TRANSFER TO WATER FLOWING IN AN ELECTRICALLY HEATED INCONEL TUBE

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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.

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 temperature 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 problems 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 diameter, 0.204 in.; effective heat-transfer length, 10 in.) in a horizontal 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.
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 insulating 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 potentiometer. 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 temperature 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 temperature rise of the water is also measured by these same thermocouples connected to give a differential reading on a very sensitive manually 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 predetermined 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:

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$c_p$</td>
<td>specific heat of water at constant pressure, $(\text{Btu/}(\text{lb})(\text{OF}))$</td>
</tr>
<tr>
<td>$D$</td>
<td>inside diameter of heater tube, $(\text{ft})$</td>
</tr>
<tr>
<td>$G$</td>
<td>mass velocity of water, $(\text{lb/}(\text{hr})(\text{sq ft}))$</td>
</tr>
<tr>
<td>$h$</td>
<td>heat-transfer coefficient, $(\text{Btu/}(\text{hr})(\text{sq ft})(\text{OF}))$</td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity of water, $(\text{Btu/}(\text{hr})(\text{sq ft})(\text{OF/ft}))$</td>
</tr>
<tr>
<td>$k_m$</td>
<td>thermal conductivity of Inconel, $(\text{Btu/}(\text{hr})(\text{sq ft})(\text{OF/ft}))$</td>
</tr>
<tr>
<td>$L$</td>
<td>effective heat-transfer length of heater tube, $(\text{ft})$</td>
</tr>
<tr>
<td>$q$</td>
<td>rate of heat transfer to water, $(\text{Btu/hr})$</td>
</tr>
<tr>
<td>$r_i$</td>
<td>inner radius of heater tube, $(\text{ft})$</td>
</tr>
<tr>
<td>$r_o$</td>
<td>outer radius of heater tube, $(\text{ft})$</td>
</tr>
<tr>
<td>$S$</td>
<td>heat-transfer area of heater tube, $0.0478\ (\text{sq ft})$</td>
</tr>
<tr>
<td>$T_b$</td>
<td>average bulk total temperature of water, $(T_1 + T_2)/2, (\text{OF})$</td>
</tr>
<tr>
<td>$T_{bp}$</td>
<td>saturation temperature of water, $(\text{OF})$</td>
</tr>
<tr>
<td>$T_o$</td>
<td>average outside-wall temperature of heater tube, $(\text{OF})$</td>
</tr>
<tr>
<td>$T_s$</td>
<td>average inside-wall temperature of heater tube, $(\text{OF})$</td>
</tr>
</tbody>
</table>
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):

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

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

$$T_s = T_o - 0.042 \frac{q}{k_m}$$

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 $T_2$. 

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

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

$W$ water-flow rate, (lb/hr)

$\mu$ absolute viscosity of water, (lb/(ft)(hr))

$c_p \mu/k$ Prandtl number

$DG/\mu$ Reynolds number

$hD/d$ Nusselt number
Average heat-transfer coefficient \( h \) was calculated from the following equation:

\[
h = \frac{W \cdot c_p \cdot (T_2 - T_1)}{S \cdot (T_S - T_B)}
\]

The physical properties of water, specific heat at constant pressure \( c_p \), thermal conductivity \( k \), and viscosity \( \mu \), 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 \( W \cdot c_p \cdot (T_2 - T_1) \) plus heat loss (fig. 7) and plotted against electric heat input shows a maximum deviation of 15 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 kilowatt 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 \((\frac{hD}{k})/(\frac{c_p\mu}{k})^{0.4}\) is plotted against Reynolds number \(\frac{DG}{\mu}\) with the physical properties 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 temperature of approximately 85°F are presented in figure 8. The correlation 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 correlation 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

\[
\frac{hD}{k}^{0.4} = 0.0168 \left(\frac{c_p\mu}{k}\right)^{0.84}
\]

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.
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 (6330 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 temperature 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°C; 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 temperatures are possible as the pressure is increased. Higher wall temperatures 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°C 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 difference 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°C. 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°F 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 beginning 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 representing 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 figure 17. Experimental values of \( \frac{(hD/k)}{\left(c_p \nu/k\right)^{0.4}} \) were divided by corresponding values of 0.0168(D\(G/\nu\))\(^{0.84} \) 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 mercury 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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REFERENCES


Figure 1. - Schematic diagram showing arrangement of experimental apparatus.
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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.
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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, 30, 60, and 90 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 gauge.
(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 kilowatts 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.
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.
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Figure 15. - Concluded. Variation of excess temperature with Reynolds number for various power inputs and entrance pressures.
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Figure 17. Concluded. Variation of correlation equation with excess temperature in nonboiling and nucleate-boiling regions. Horizontal tube.