AECU-3579

FREE CONVECTION HEAT TRANSFER TO HORIZONTAL CYLINDERS FROM AN ORDINARY FLUID CONTAINING A VOLUME HEAT SOURCE

Jose Pineda de Guzman

M.S. Ch.E. January 1958

Contract AT (11-1) - 432

DO NOT PHOTOSTAT

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

AECU-3579

DO NOT PHOTOSTAT

I AECU-3579

FREE CONVECTION HEAT TRANSFER TO HORIZONTAL CYLINDERS FROM AN ORDINARY FLUID CONTAINING A VOLUME HEAT SOURCE

A Thesis
Submitted to the Faculty
of
Purdue University
by
Jose Pineda de Guzman

In Partial Fulfillment of the
Requirements for the Degree
of
Master of Science
in
Chemical Engineering

January, 1958

499-001

ACKNOWLEDGMENT

The author would like to express his appreciation to Dr. A. Sesonske, who proposed this investigation, for his aid and guidance throughout all phases of the work. Thanks are also due to W. N. Smith for his help in solving instrumentation problems and R. McKinley for helpful suggestions.

The author is grateful to the United States Atomic Energy Commission for the financial support they provided under Contract AT (11-1)-432 which made the work possible.

TABLE OF CONTENTS

499 - 003

LIST OF FIGURES

ABSTRACT

DeGuzman, Jose Pineda, M.S., Purdue University, January, 1958. Free Convection Heat Transfer to Horizontal Cylinders from an Ordinary Fluid Containing a Volume Heat Source. Major Professor: Alexander Sesonske

Outside heat transfer coefficients for several horizontal tubes cooling an ordinary fluid containing a volume heat source by free convection were obtained experimentally. Water was used as the coolant and a very dilute solution of hydrochloric acid was the bulk fluid.

The basic equipment included a Lucite test cell which contained the bulk fluid, copper screen grids through which an alternating current was passed to generate a volume heat source in the bulk and several horizontal 1/4" O.D. stainless steel tubes through which the coolant was pumped. Experiments were conducted using a single-tube, four-tube and five-tube geometry. The heat transfer coefficients obtained were considered accurate within 8 per cent.

With single-tube cooling, the heat transfer coefficients were correlated by

$$Nu = 0.49 \quad (Gr\ Pr)^{1/4}$$

which compares with

$$Nu = 0.54 \quad (Gr\ Pr)^{1/4}$$

recommended by McAdams (12).

With four cooling tubes, the data were correlated by

$$Nu = 0.59 \quad (Gr\ Pr)^{1/4}$$

while with five cooling tubes, the value of the coefficients in the correlation equation was

$$Nu = 0.64 \quad (Gr \quad Pr)^{1/4}$$

INTRODUCTION

AECU-3579

The homogeneous type of nuclear power reactor where the fuel is dissolved in the moderator has attractive features, namely, continuous fuel reprocessing and the avoidance of heat transfer complications inherent in solid fuel elements. The fuel solution in such a reactor would act as a volume heat source in the critical, heat generating zone of the reactor.

If a homogeneous reactor has been operating for some time at a power of the order of megawatts, the activity of fuel solution after shutdown will continue to generate many kilowatts from the fission products which continue to decay. Unless this heat is removed, temperatures will increase with possible resulting damage to the reactor. Should a shutdown result in fuel circulating pump failure, a free convection regime would be set up since the heat generating fuel solution would be in contact with cold coolant surfaces.

If steady state heat removal could be accomplished without the complications of a circulating pump in a corrosive system, the homogeneous reactor concept would be more attractive.

In both applications, horizontal tubes have practical design advantages over other geometries. The measurement of heat transfer rates to horizontal cooling tubes from a volume-heated fluid under conditions of free convection was, therefore, considered important.

To date, very limited experimental and theoretical investigations have been carried out to determine free convection film coefficients with a volume heat source. The investigation described in this thesis was one phase of a study of free convection heat transfer rates to horizontal cooling tubes from an ordinary fluid containing a volume heat source. In this particular work, outside film coefficients were obtained using a single-tube, a four-tube, and a five-tube geometry. These values were compared with the various empirical and theoretical equations in the literature applying to both volumetric and non-volumetric heat sources.

PREVIOUS INVESTIGATIONS

Non-Volumetric Heat Source

In 1937, Squire (3) used the boundary layer equations for momentum and heat flow to calculate the heat transfer in free convection for a vertical plate. He assumed ideal velocity and temperature profiles for a laminar boundary layer which were substituted into the integrated equations with the result:

$$Nu = 0.508\, Pr^{\frac{1}{2}} (0.952 + Pr)^{-\frac{1}{4}} (Gr)^{\frac{1}{4}}$$

where

$Nu = \frac{hx}{k}$

$Pr = \frac{C_p \mu}{k}$

$Gr = \frac{\rho^2 g \beta \theta x^3}{\mu^2}$

$h =$ point heat transfer coefficient

$\beta =$ coefficient of volumetric expansion

$\theta =$ temperature differential, ambient fluid to wall

$x =$ characteristic length (or diameter)

$\mu =$ viscosity of ambient fluid

$\rho =$ density of ambient fluid

$g =$ acceleration due to gravity

$k =$ thermal conductivity of ambient fluid

$C_p =$ heat capacity of ambient fluid

Squire's equation was in good agreement with the results of

Schmidt and Beckmann (3) who carried out experiments on free convection heat transfer from vertical plates.

It is interesting to note that possibly the first analytical solution of convective heat transfer from a hot vertical plate to still air was given by Lorenz (11) in 1881 with his equation:

$$h = 0.548 \left[\frac{C_p\, g\, k^3 \rho^2 \Delta t}{\mu L T}\right]^{\frac{1}{4}}$$

Nusselt (13) in 1915 applied dimensional analysis, which he called "consideration of similarity", to the fundamental differential equations of fluid flow and heat conduction, and derived for the case of natural convection the formula:

$$h = \left[\frac{ak}{d}\right]\left[\frac{C_p \mu}{k}\right]^m \left[\frac{d^3 \rho^2 \beta g \Delta t}{\mu^2}\right]^n$$

Subsequent efforts have improved Lorenz's work, the most extensive of which were carried out by Schmidt and Beckmann (16) with mathematical assistance from Pohlhausen (9).

Eckert (2) mentions the classical experiments of Schmidt where the boundary layer and the temperature field around a horizontal tube and a vertical plate in free convection were shown in schlieren and interference photographs.

Hermann (7), in 1936, solved the boundary layer equations for free convection heat transfer from a horizontal cylinder to air. He obtained the following relationship

$$\left[Nu_D\right]_{avg} = 0.40\ (Gr\ Pr)^{\frac{1}{4}}$$

where

$Nu_D = \frac{hD}{k}$

$Gr = \frac{\rho^2 g \beta \theta x^3}{\mu^2}$

D = outside diameter of the cylinder

McAdams (12) mentions a similar form of the above equation, differing only in the value of the constant:

$$Nu = 0.53 \, (Gr \, Pr)^{\frac{1}{4}}$$

This equation describes a good deal of experimental data.

The nuclear energy program has encouraged research on free convection systems. Hyman and Bonilla (8) have studied natural convection processes in mercury and lead-bismuth eutectic systems as well as several ordinary fluids. They found that for streamline flow, the data for liquid metals for all cylinders larger than wire, can be represented by:

$$Nu = 0.53 \, (Gr \, Pr^2)^{\frac{1}{4}}$$

or

$$h = 0.53 \left[\frac{\rho^2 g \beta \Delta t \, C_p{}^2 k^2}{D} \right]^{\frac{1}{4}}$$

They also investigated the transition from laminar flow at low heat transfer rates, to turbulent flow at the top of the cylinder at high rates for the case of heating non-liquid metals.

Bonilla and Collins (1) have studied natural convection heating and cooling by horizontal cylinders with ordinary fluids. Their results indicated that the usual correlation for heat transfer by natural convection,

$$Nu = 0.54 \, (Gr \, Pr)^{\frac{1}{4}}$$

applies equally well to heating or cooling and that the "film" temperature is an adequate temperature at which to evaluate the physical properties of the fluid.

In 1955, Levy (10) presented an evaluation of the range of application, accuracy and usefulness of integral methods in natural convection flow. By using integral methods, he obtained approximate solutions to free-convection problems for bodies of arbitrary geometries. For a horizontal cylinder, the solution was:

$$\left[Nu_D\right]_{avg} = 0.44 \, (Gr \, Pr)^{\frac{1}{4}}$$

Levy's results are 10 per cent greater than Hermann's and about 15 per cent below the accepted experimental correlation for horizontal cylinders given by McAdams,

$$Nu = 0.53 \, (Gr \, Pr)^{\frac{1}{4}}$$

Volumetric Heat Source

Ostrach (14), in 1952, analyzed the laminar natural convection flow and heat transfer of fluids with heat sources between two long parallel plane surfaces with constant wall temperatures. He found that frictional heating and heat sources increase the velocities and temperatures in the channel.

In 1954, Hamilton, et. al. (5) published a theoretical analysis of free convection for pipe and parallel plate systems having a large ratio of length to diameter and oriented with the long axis vertical. They developed solutions for the Navier-Stokes equations and the heat conduction equation for a moving system.

Hamilton and Lynch (6), in 1955, investigated a free convection system consisting of two channels formed by three parallel and equally spaced vertical plates. They found that the free convection circuit in such systems exists as one long cell equal to the height of the system. In the upper end region, the velocity structure had the appearance of the free convection boundary layer near the top of a cooled vertical plate immersed in a fluid. The velocity profile in the middle region appeared fully established and was similar to that given by the laminar theory (5). The transition from laminar to turbulent boundary layer occurred at a Grashof modulus of about 5×10^9.

In 1955, Hallman (4), determined the effect of a volume heat source in a vertical tube geometry. His work appeared similar to Hamilton's analysis of the vertical channel.

Recently, Randall (15) analytically studied the effect on the heat transfer coefficient of a volume heat source within an ordinary fluid being cooled in free convection by a horizontal cylinder. He solved the problem for a vertical flat plate cooling the fluid containing the volume heat source and transformed his results to a horizontal cylinder assuming the heat source existed only in the boundary layer around the cylinder. His calculations indicated the heat transfer coefficient with a volume heat source to be slightly higher than its value without a heat source. His results revealed an increase in the ratio of heat transfer coefficients with increasing value of the volume heat source for a given cylinder diameter. He also found this ratio decreased with increasing temperature difference across the boundary layer, while the incidence of turbulence occurred at smaller values of this temperature difference as the cylinder diameter increased.

EXPERIMENTAL

The purpose of the experimental part of the investigation was to obtain outside heat transfer coefficients for several horizontal tubes cooling an ordinary fluid containing a volume heat source by free convection. Water was used as the coolant and a very dilute solution of hydrochloric acid was the bulk fluid.

Experiments were first conducted using a single-tube geometry to study the different operating variables involved in the measurement of outside film coefficients. Then, experiments were carried out using four horizontal tubes and later five cooling tubes to determine the relative importance of a new variable, tube number and arrangement, which may be useful for industrial applications.

The basic equipment used included a Lucite tank which contained the bulk liquid, copper screen grids through which an alternating current was passed to generate heat in the bulk fluid, and several horizontal stainless steel tubes through which the coolant passed. A coolant pump, rotameter, piping, instrumentation, and some accessory equipment completed the installation.

The heat transferred through the tube wall was determined from the coolant mass flow rate and the coolant temperature increase. The outside film coefficients were obtained by dividing this heat transferred value by the product of the

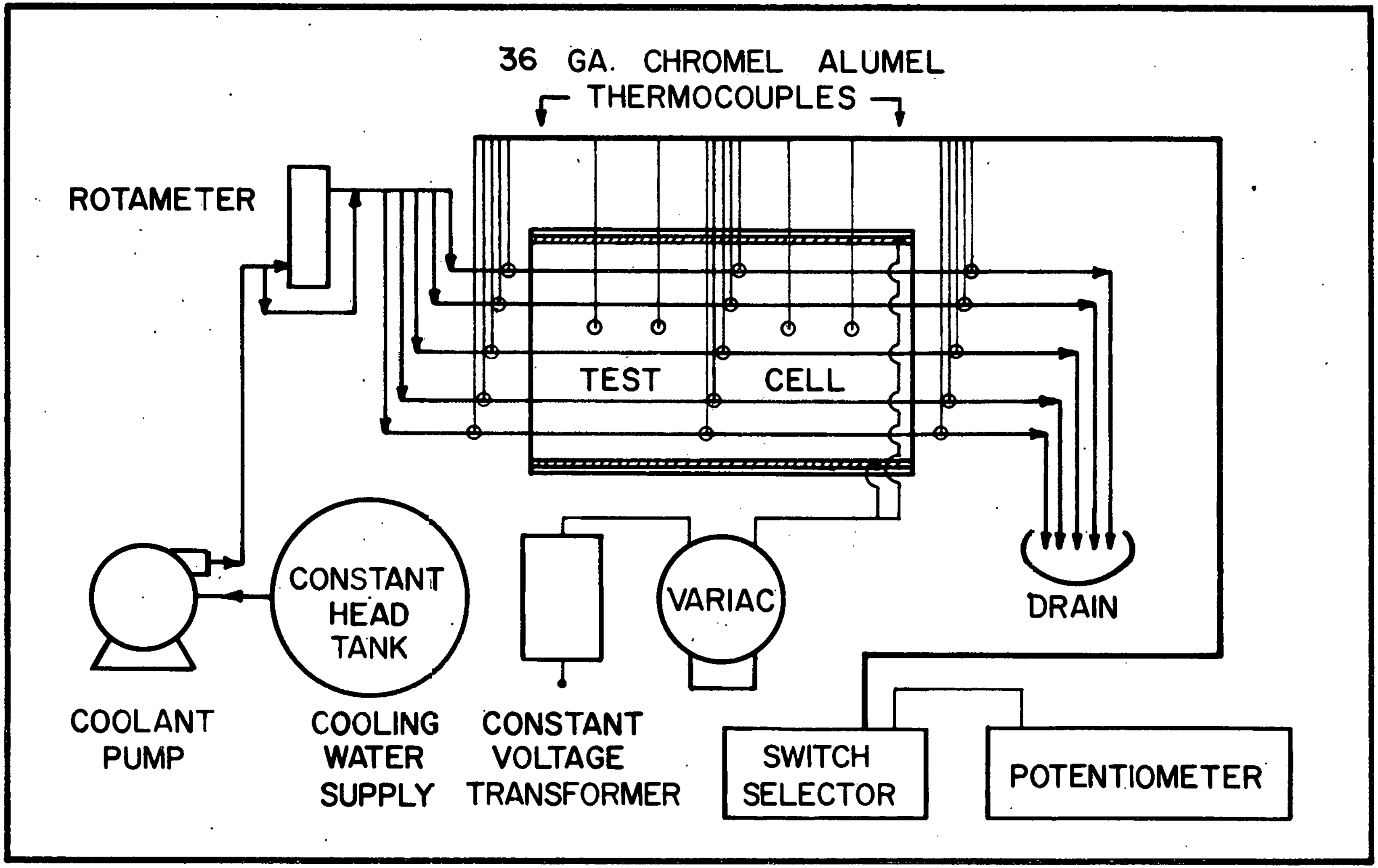


FIGURE I
SCHEMATIC DIAGRAM OF APPARATUS

bulk to tube wall temperature differential and the outside area of heat transfer.

A brief description of the equipment, discussion on temperature measurement, and experimental procedures follow:

Equipment

A Plexiglass-walled tank, 24" by 12" and 16" high served as the test cell. Lucite tube racks, 24" x 3" and 1/2" thick held the 1/4" O.D. 20 gage stainless steel (304) tubes horizontally at each end inside the tank permitting adjustment of any tube to the desired height in the electrolyte as shown in Figure 2. A uniform volume heat source was produced by passing an alternating current of about 3.5 amperes at 80 volts between two vertical 35 mesh copper wire screens, 22" long and 15" high situated outside of the tubes in the solution. A Variac was used for controlling the voltage. Heat removal was accomplished by pumping cooling water through the horizontal tubes. A 1/4 HP centrifugal pump drew its supply of cooling water from a 30-gallon constant-head tank. The coolant flow rate was roughly determined with a Brooks Full-View rotameter (cap: 0-5 GPM) and accurately measured during each run using a stop watch and a weighing scale.

Temperature Measurement

Chromel-Alumel thermocouple wire, 36 gage, with Teflon-Fiberglass insulation, Thermoelectric Co. type TG-36-CT, was

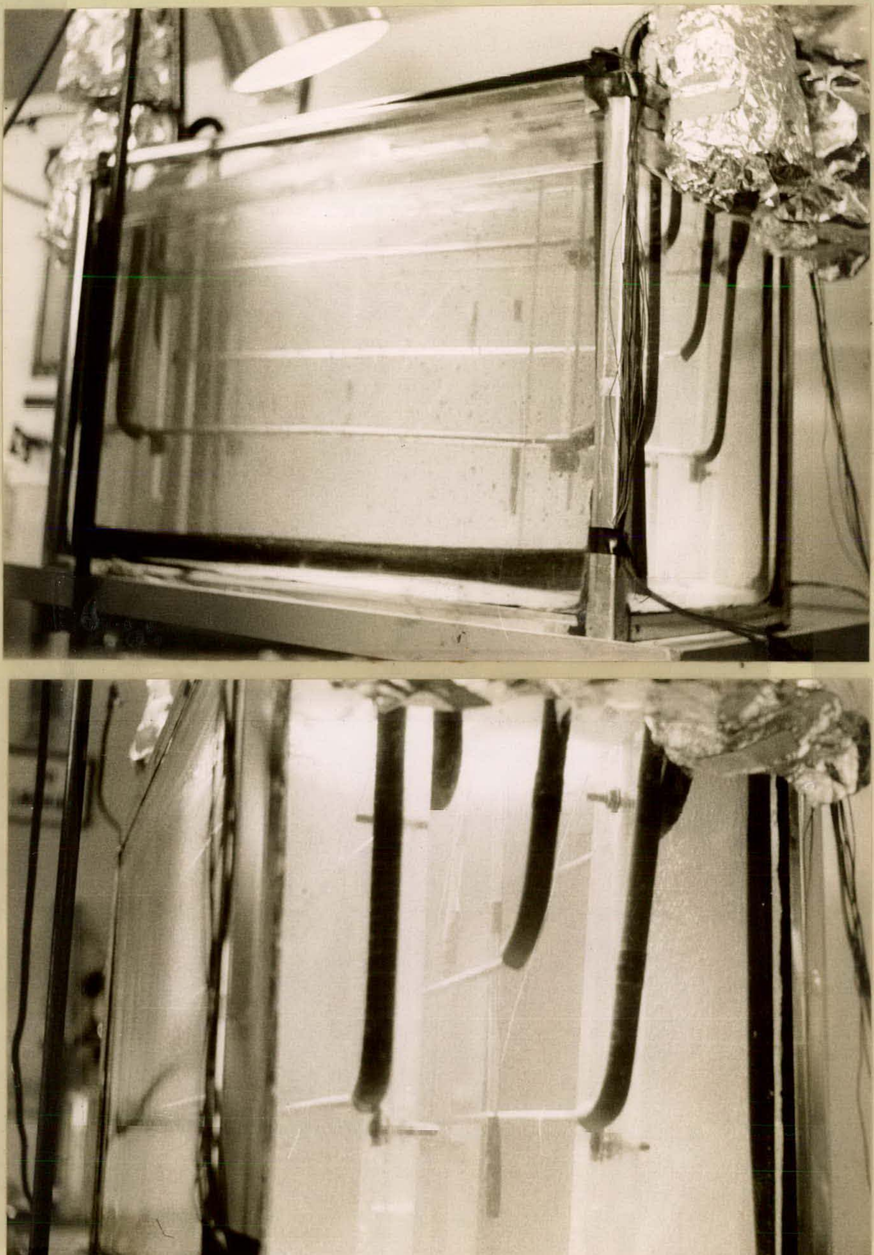


FIGURE 2
PHOTOGRAPHS OF TEST CELL SHOWING LUCITE TUBE RACKS IN A FIVE—TUBE GEOMETRY

used in all thermocouple installations.

The coolant inlet and outlet temperatures were measured by a thermocouple immersed in a mercury-filled glass capillary which was suspended in a 60 cc. bottle as shown in Figure 3. The insulated bottle served as a mixing chamber so that the coolant mixed mean temperature was actually measured.

The outside tube wall temperature was measured by a thermocouple whose junction was silver soldered to the tube wall with a carefully controlled non-oxidizing flame. A smooth soldered joint was produced as shown in Figure 3 so that the pattern of convection currents in the vicinity of the installation was not disturbed.

A series of suspended thermocouples were used to measure the bulk fluid temperatures at different known positions in the cell as shown in Figure 11.

The readings of the different thermocouples were taken with a Rubicon Co. Type B High Precision Potentiometer, Serial No. 90494. Two selector switches in a junction box provided a systematic method of reading the various temperatures.

All original temperature data were taken in millivolts. The coolant temperature differential was obtained between the individual readings for coolant inlet and outlet temperatures. In the second, a direct potential difference reading was obtained between the two thermocouples with no cold junction used. The second method was considered more reliable

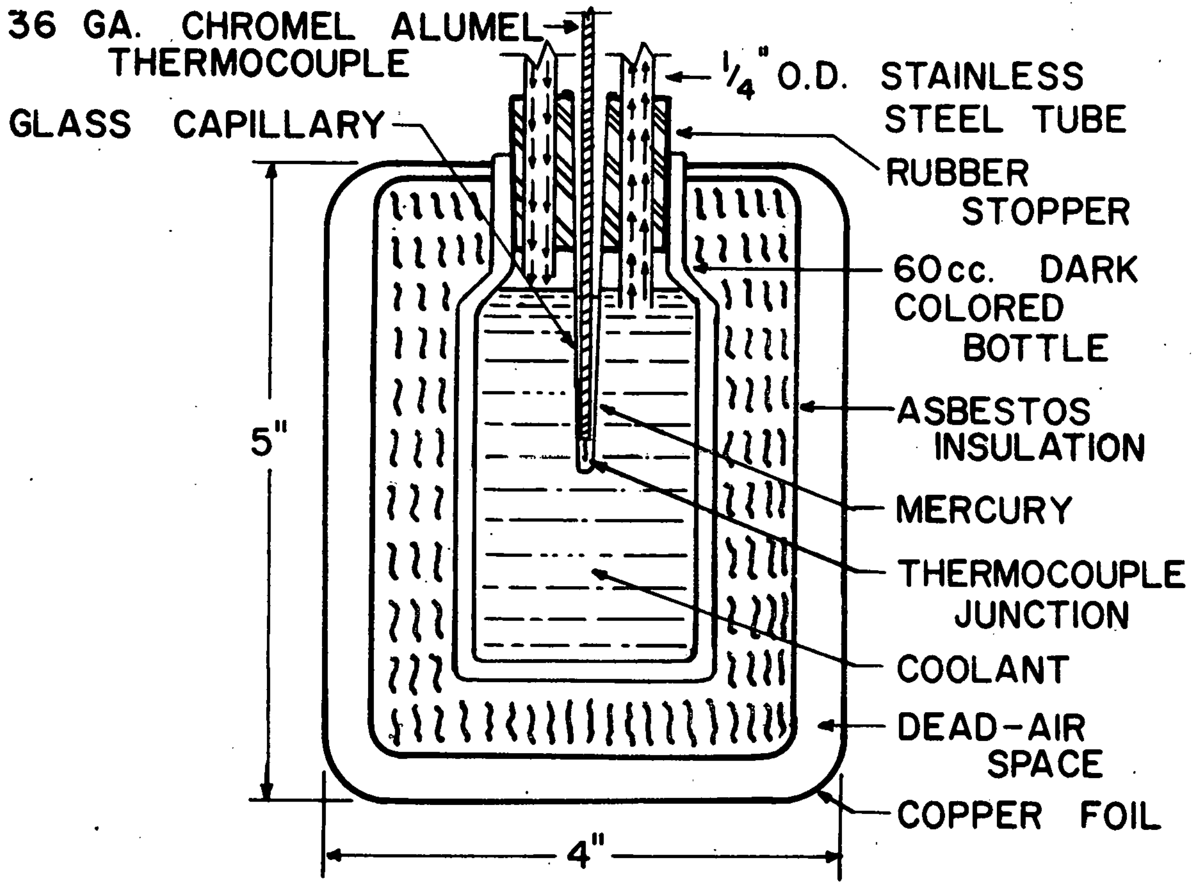
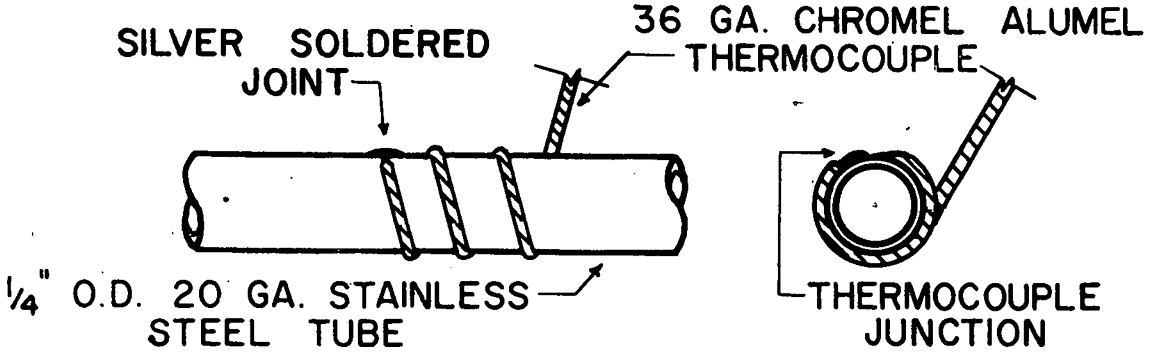


FIGURE 3
THERMOCOUPLE INSTALLATION FOR COOLANT INLET AND OUTLET TEMPERATURES



THERMOCOUPLE INSTALLATION FOR TUBE—WALL TEMPERATURE although results obtained by the two methods for a single measurement checked within one per cent.

Procedure

To start each run, the coolant pump and the Variac were switched on and the desired coolant flow rate was set with control valves. The system was then allowed to adjust itself for a period of about two hours. Readings were then taken of the coolant inlet and outlet temperatures, tube-wall temperature, bulk temperature and room temperature. Voltage across the grids was adjusted until the tank temperature remained close to room temperature. The heat transfer system was then allowed to come to steady state which usually required a period of about three hours or more. During this time, readings were taken every hour. Readings were again recorded at one hour intervals after steady state was reached.

Although a rotameter was used, the coolant flow rate through the cooling tubes were actually measured individually. The rate of flow through each tube was determined by weighing a timed amount of the coolant. This was done for every run made.

RESULTS

Presentation of Results

Single Horizontal Cooling Tube

A series of 28 runs were made to obtain data under different heat rates ranging from 60 to 249 Btu/hr and bulk to tube wall temperature differential ranging from 9.9 to 22.7°F. From these data, the heat transfer coefficients were calculated.

When the logarithm of the resultant Nusselt number was plotted against the logarithm of the product of the Grashof and Prandlt moduli, reasonable agreement was obtained with predicted values for non-volumetric heat sources. The average of the heat transfer coefficients with a volume heat source appeared only approximately 10% lower than those without a heat source as shown in Figure 4.

Four Horizontal Cooling Tubes

A total of 72 runs were made with four horizontal cooling tubes under different heat rates ranging from 89 to 352 Btu/hr and bulk to tube wall temperature differential ranging from 7.2 to 21.6°F. From these data, the outside heat transfer coefficients, the resultant Nusselt number, the Grashof modulus and the Prandlt number were calculated.

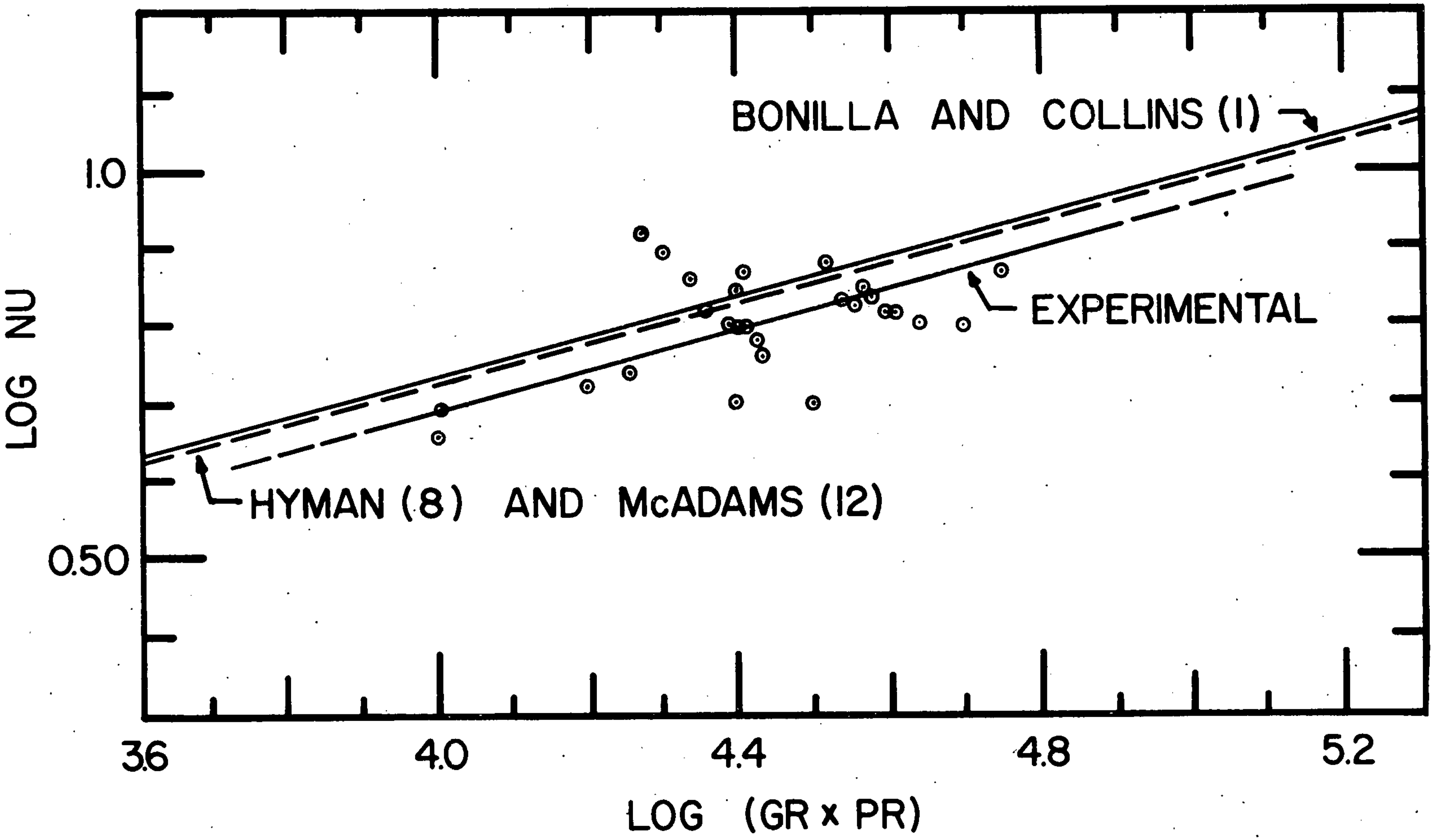


FIGURE 4

LOG NU VS. LOG (GR x PR) FOR A SINGLE HORIZONTAL COOLING TUBE

When the logarithm of the Nusselt number was plotted against the logarithm of the product of the Grashof and Prandlt moduli for the total number of runs, the volume heat source results were approximately 10% higher than that predicted by McAdam's (12) correlation for a non-volumetric heat source as shown in Figure 5.

Five Horizontal Cooling Tubes

A total of 60 runs were made with five horizontal cooling tubes. These were also made under different heat rates ranging from 136 to 340 Btu/hr and bulk to tube wall temperature differential ranging from 9 to 22.9°F. From these data, the outside heat transfer coefficients, the resultant Nusselt number, the Grashof modulus and the Prandlt number were again calculated.

When the logarithm of the Nusselt number was plotted against the logarithm of the product of the Grashof and Prandlt moduli for all of the runs, the results were 10 per cent higher than the results for the four-tube geometry. The heat transfer coefficients for each of the five horizontal cooling tubes with a volumetric heat source appeared to be approximately 20 per cent higher than that predicted by McAdams' (12) correlation for a non-volumetric heat source as shown in Figure 6.

Correlation for a Volume Heat Source

A correlation for free convection heat transfer with a volume heat source has been given by Dr. E. M. Sparrow (18)

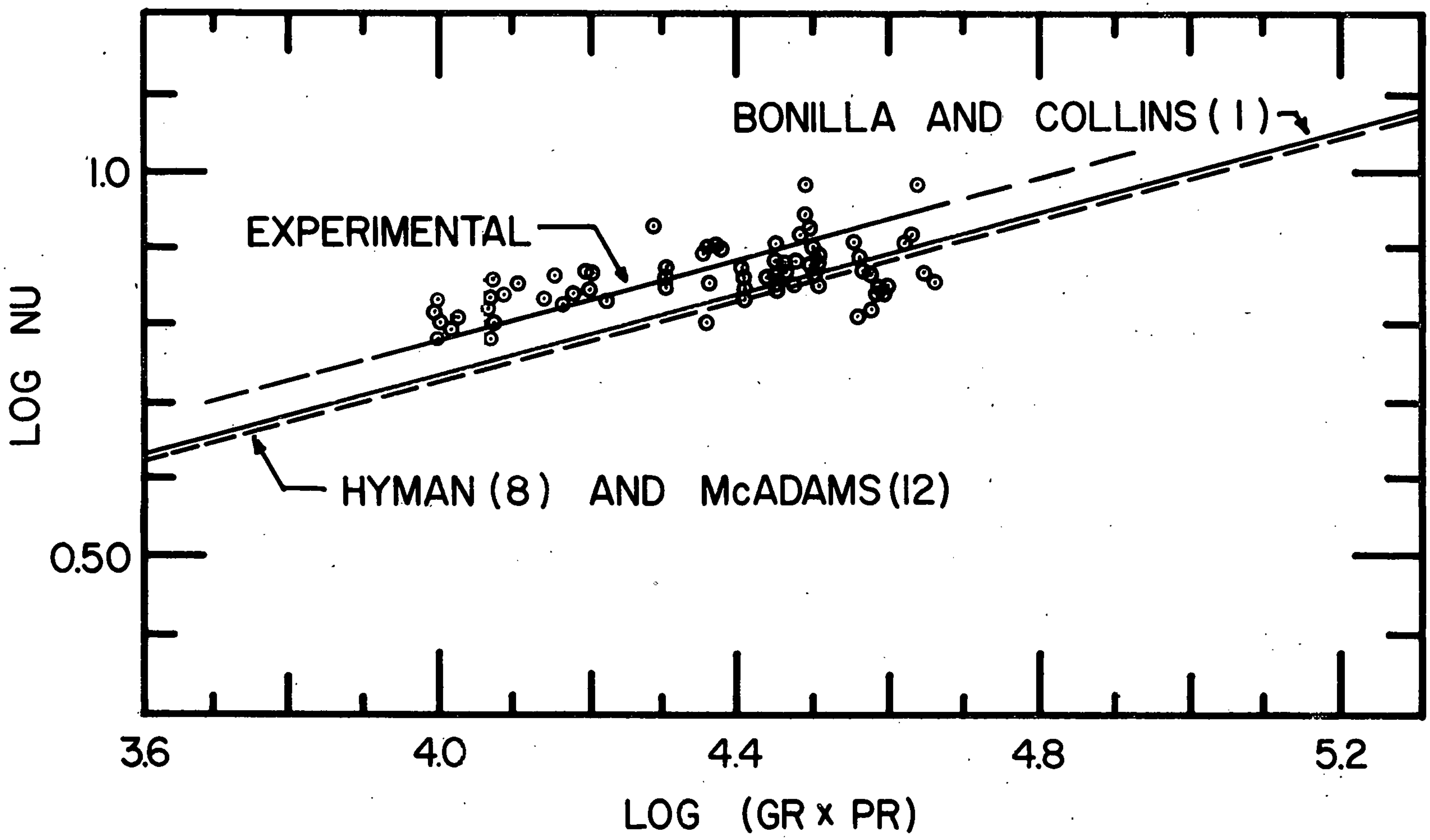


FIGURE 5
LOG NU VS. LOG (GR x PR) FOR
FOUR HORIZONTAL COOLING TUBES

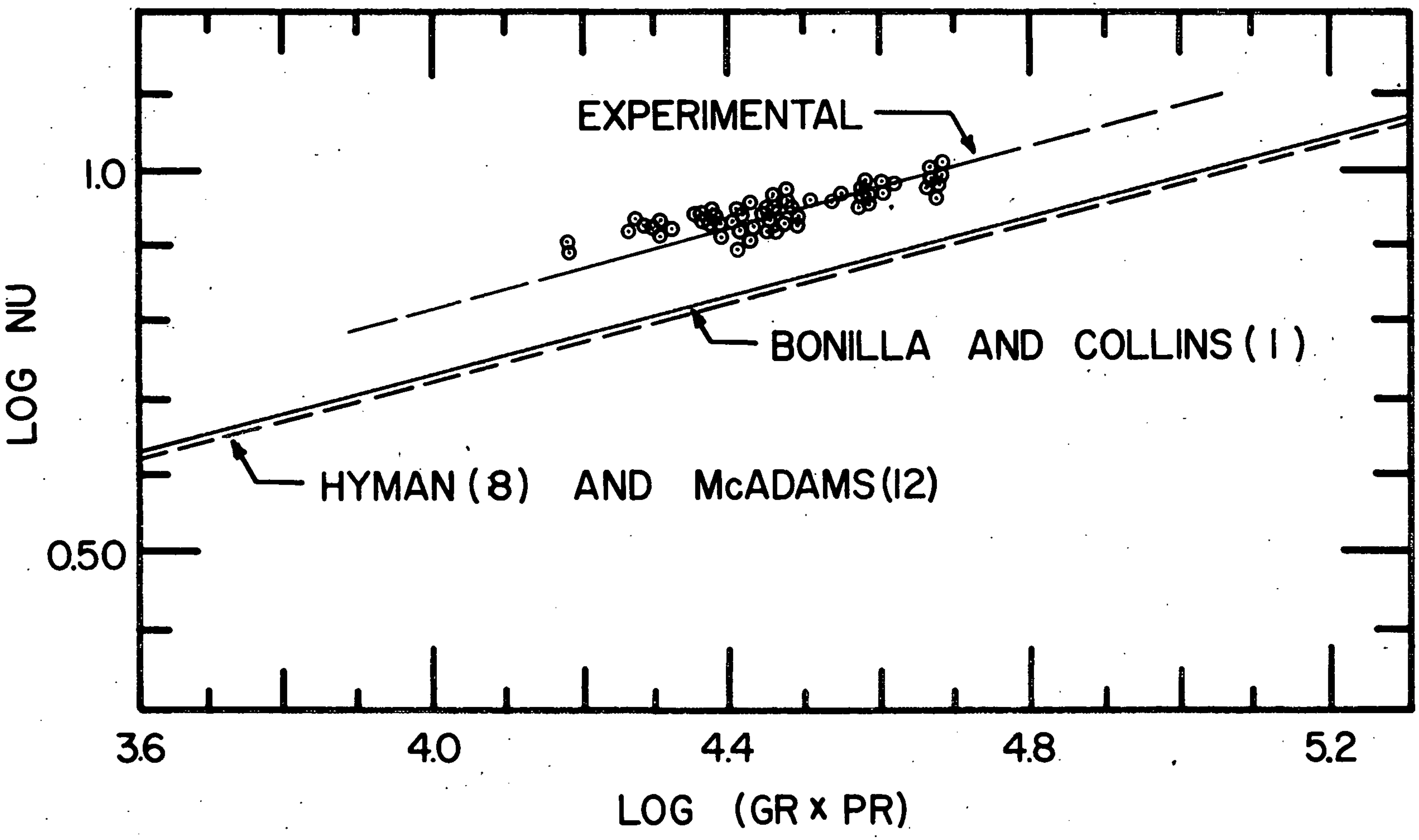


FIGURE 6
LOG NU VS. LOG (GR x PR) FOR
FIVE HORIZONTAL COOLING TUBES

who suggested a plot of the Nusselt number versus $\frac{q'''}{\Delta t} \cdot \frac{1}{Gr^{\frac{1}{4}}}$ for constant Prandlt number and tube diameter where,

$q''' =$ volume heat source, $\frac{\text{Btu}}{\text{hr x ft}^3}$

$\Delta t =$ bulk to tube wall temperature differential, °F

$Gr =$ Grashof modulus

With single-tube cooling, when the results were plotted using the suggested correlation, a straight-line relationship was obtained as shown on Figure 7. However, when the results with four cooling tubes were also plotted, there was some scattering of the data although a straight line relationship was also obtained as seen in Figure 8.

Description of Flow Patterns

A study of the nature of the flow around the cooling tubes was necessary to confirm the assumption of laminar flow expected in the Grashof Number range studied and to determine the location of the "wake" from the tube.

Visual observations of the convection flow patterns were made with the aid of minute red droplets of Meriam Unity Instrument Oil dispersed in the bulk fluid by a hypodermic syringe.

The pattern of convection currents for a single tube, four-tube and five-tube geometry are shown graphically on Figure 9, 10 and 11. The dotted arrows indicate the path of the oil droplets which seemed to quite accurately indicate

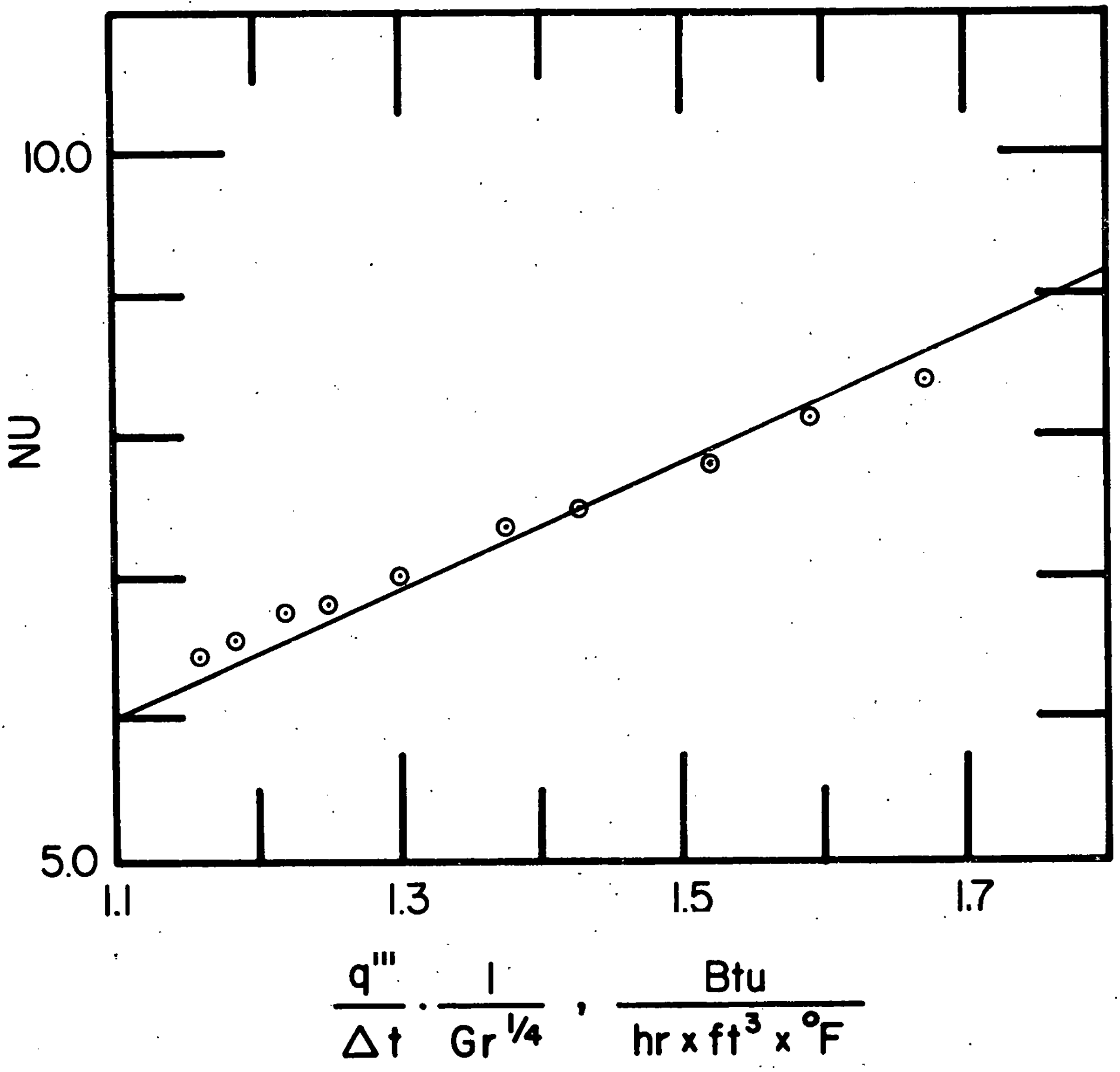


FIGURE 7

NUSSELT NUMBER VS. $\frac{q'''}{\Delta t} \cdot \frac{1}{Gr^{1/4}}$ FOR A SINGLE HORIZONTAL COOLING TUBE

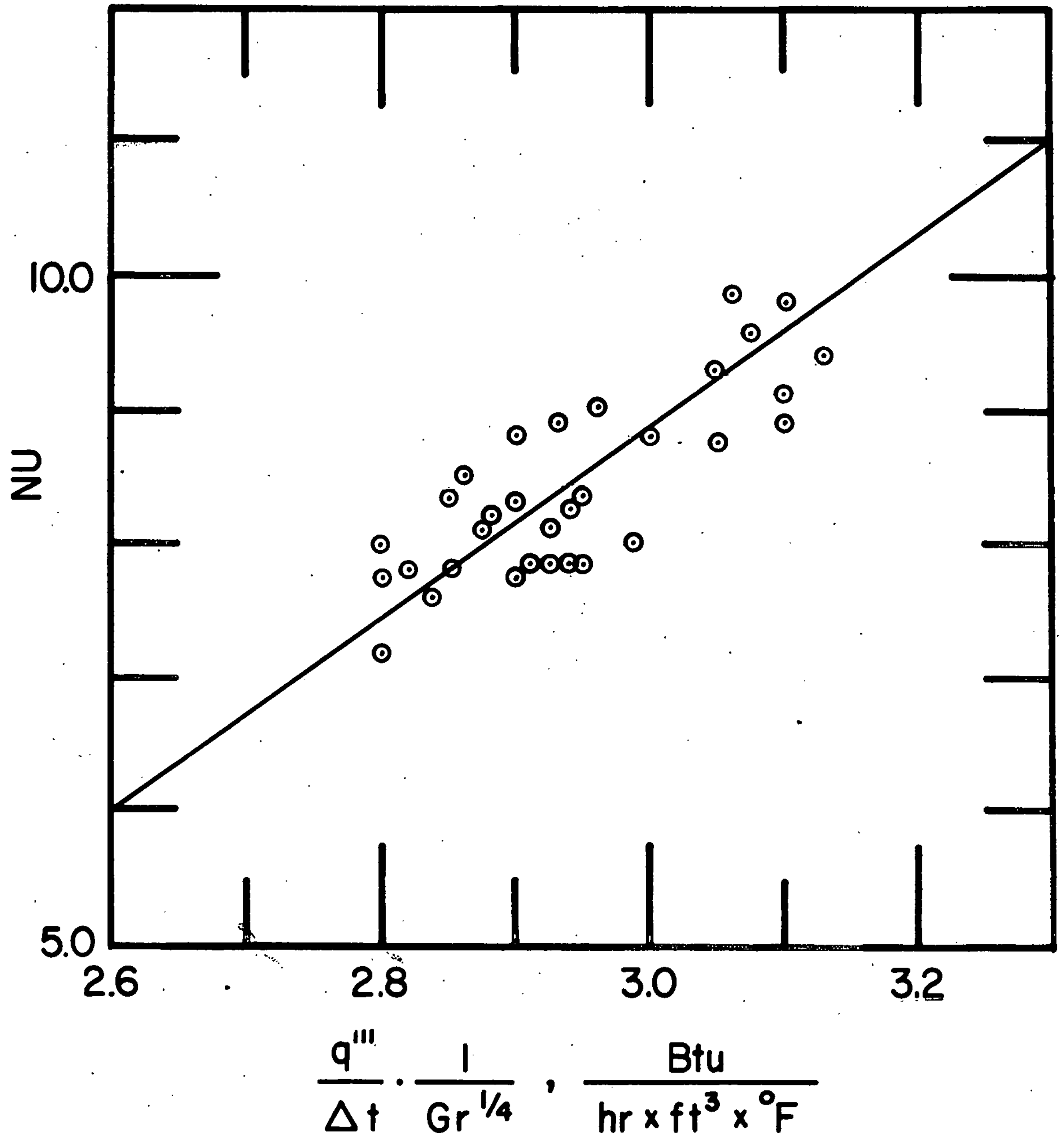


FIGURE 8

NUSSELT NUMBER VS. $\frac{q'''}{\Delta t} \cdot \frac{1}{Gr^{1/4}}$ FOR FOUR HORIZONTAL COOLING TUBES

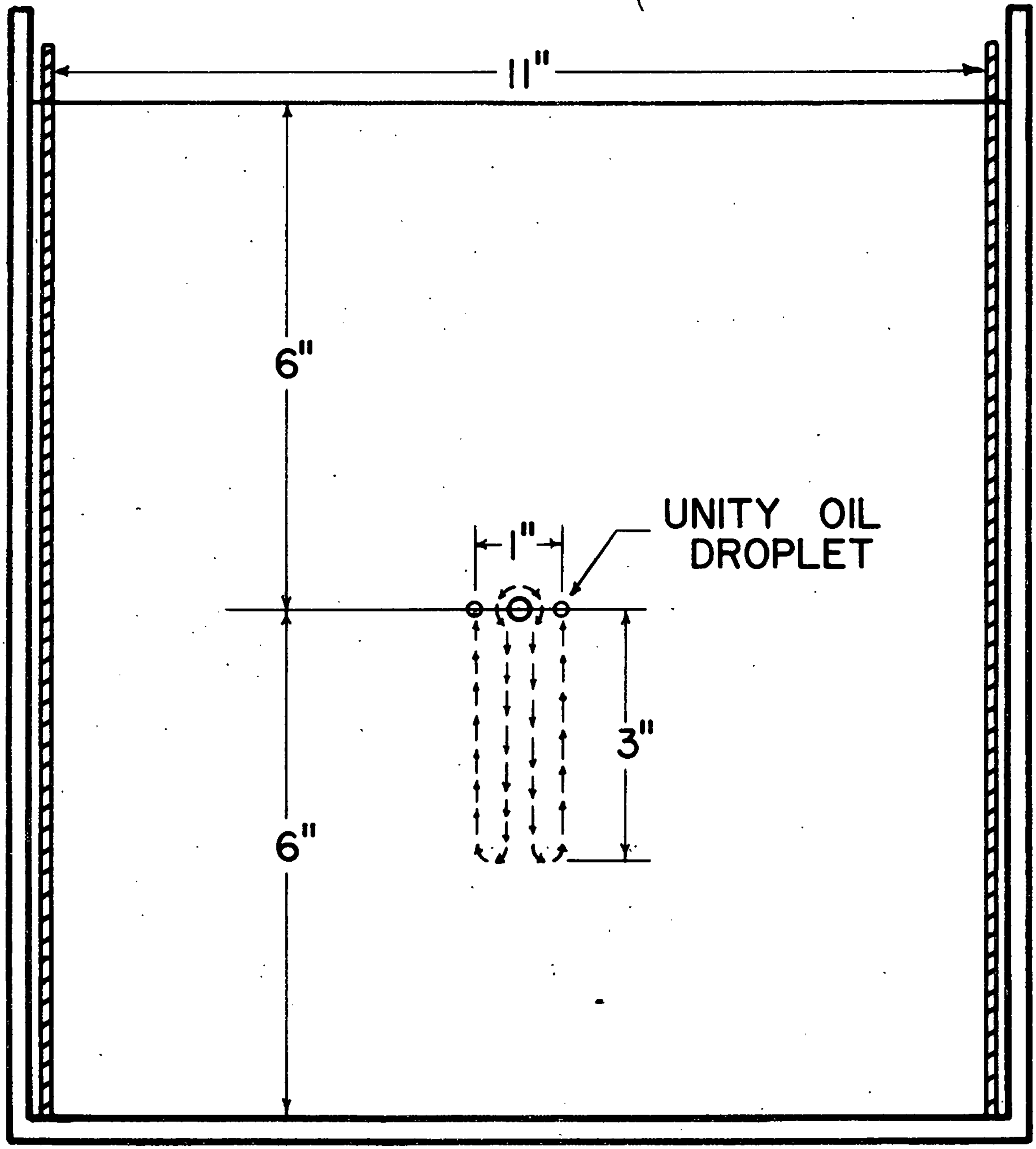


FIGURE 9

CONVECTION PATTERN IN A SINGLE HORIZONTAL COOLING TUBE ARRANGEMENT

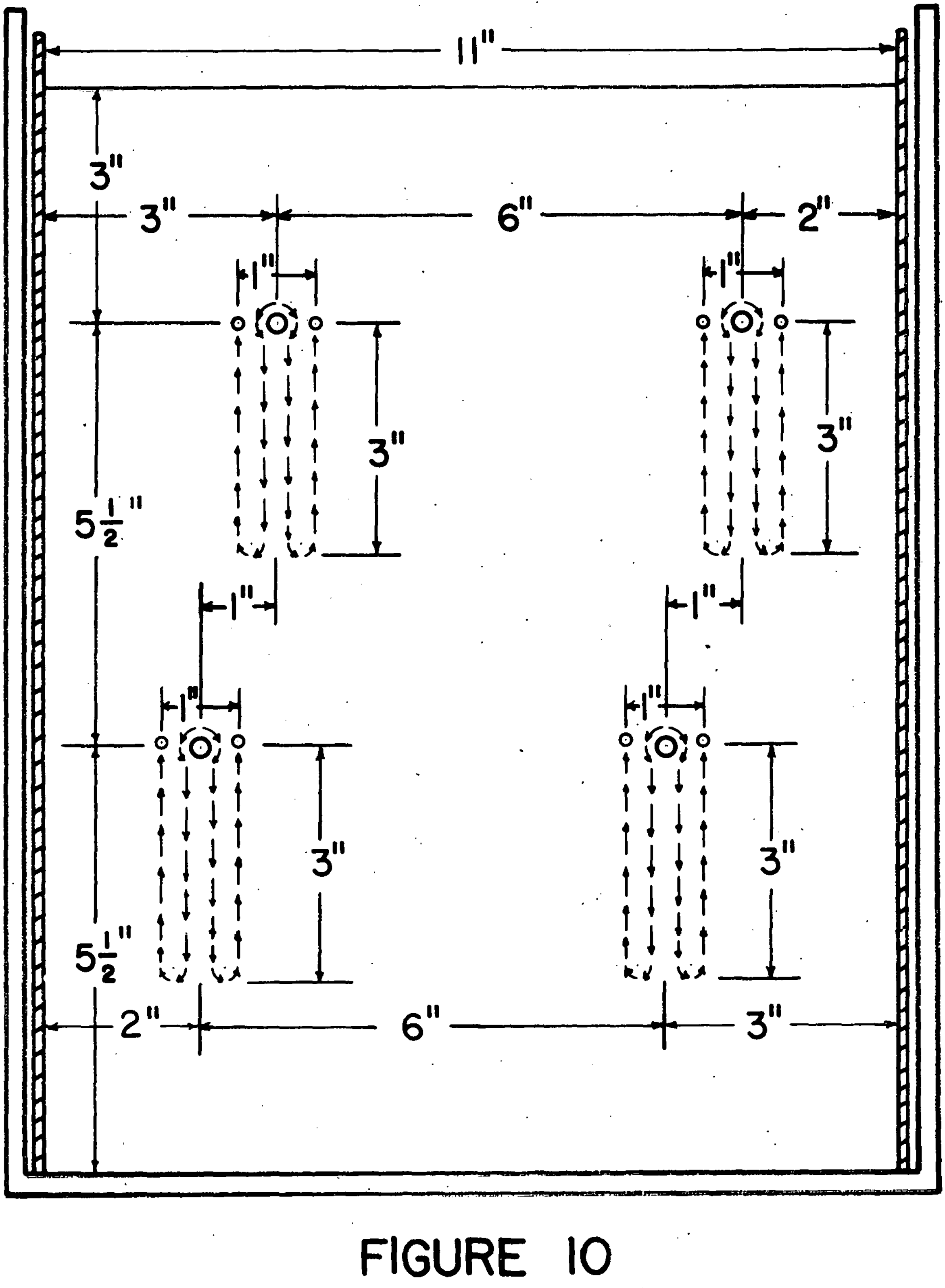


FIGURE 10
CONVECTION PATTERNS IN A FOUR HORIZONTAL COOLING TUBE ARRANGEMENT

26

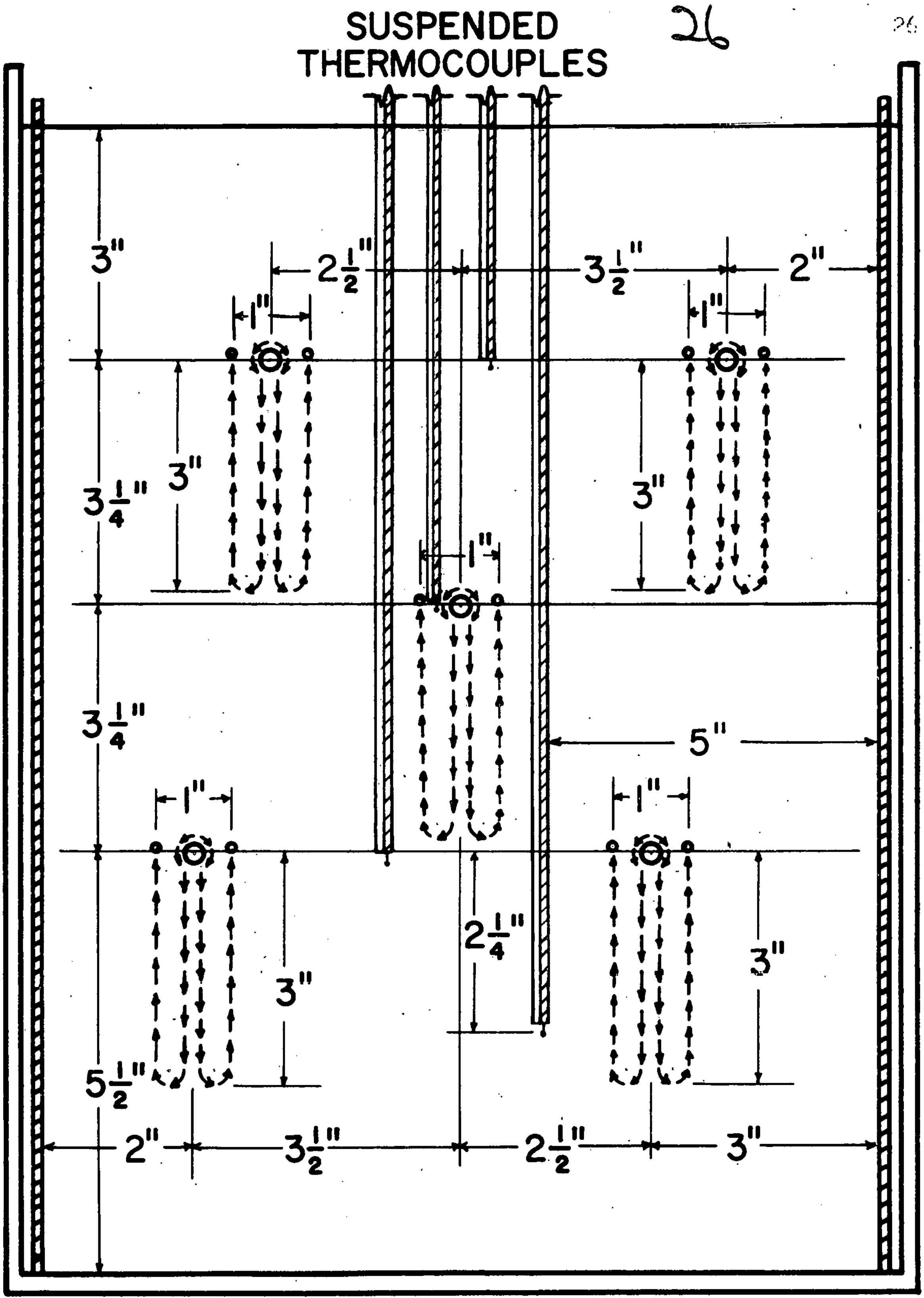


FIGURE 11
CONVECTION PATTERNS IN A FIVE HORIZONTAL COOLING TUBE ARRANGEMENT

the nature of the flow patterns around the cooling tubes. In all cases, boundary-layer behavior was noted.

After long-time observations of the flow patterns, it was found that the oil droplets in the vicinity of the wake dropped straight down from the tube for a distance of about three inches. Then the oil droplets would branch off to either side and start to rise. It appeared that the same liquid stayed in the current with the rest of the bulk fluid remaining stagnant. No attempt was made to measure accurately the velocity of the convection currents. It was observed, however, that it took the oil droplets approximately 30 to 40 seconds to travel the path indicated on Figures 9, 10 and 11 during a typical experimental run. The speed of the currents increased with an increase in the value of the volume heat source since in order to maintain steady state, the bulk to tube wall temperature differential would have to increase. As a result of this increase, there was a consequent increase in the density driving force for convective flow. Increasing the coolant flow rate also increased the velocity of the convection currents for the same reason since the average wall temperature tended to decrease.

Evaluation of Errors

Before reaching conclusions on the effect of a volume heat source using the results obtained, it was necessary to evaluate all sources of error. The data were corrected for thermal conduction effects which could be easily calculated. Other errors were examined and their effect on the heat transfer coefficient calculated. These included changes in bulk properties with temperature, errors in the measurement of the coolant temperature, coolant flow rate, errors due to inaccuracies of thermocouples and heat conduction errors.

Changes in Bulk Properties with Temperature

In the calculations, properties of the solution were assumed independent with temperature. This was, in fact, not the case. Changes in both electrical and physical properties of the electrolyte are discussed in the following paragraphs.

A. Electrical Properties

The 0.001 N HCL in distilled water solution used as the electrolyte was assumed to have the physical properties of water. At atmospheric pressure, all the physical properties of water except the viscosity and expansion coefficient do not change greatly with temperature variations.

The electrical properties of this electrolyte were measured by Hamilton and Lynch (5). Their plot of the variation of the resistivity of the electrolyte with temperature is reproduced on Figure 12 and was used to analyze the effect of this variation on the uniformity of the volume heat source.

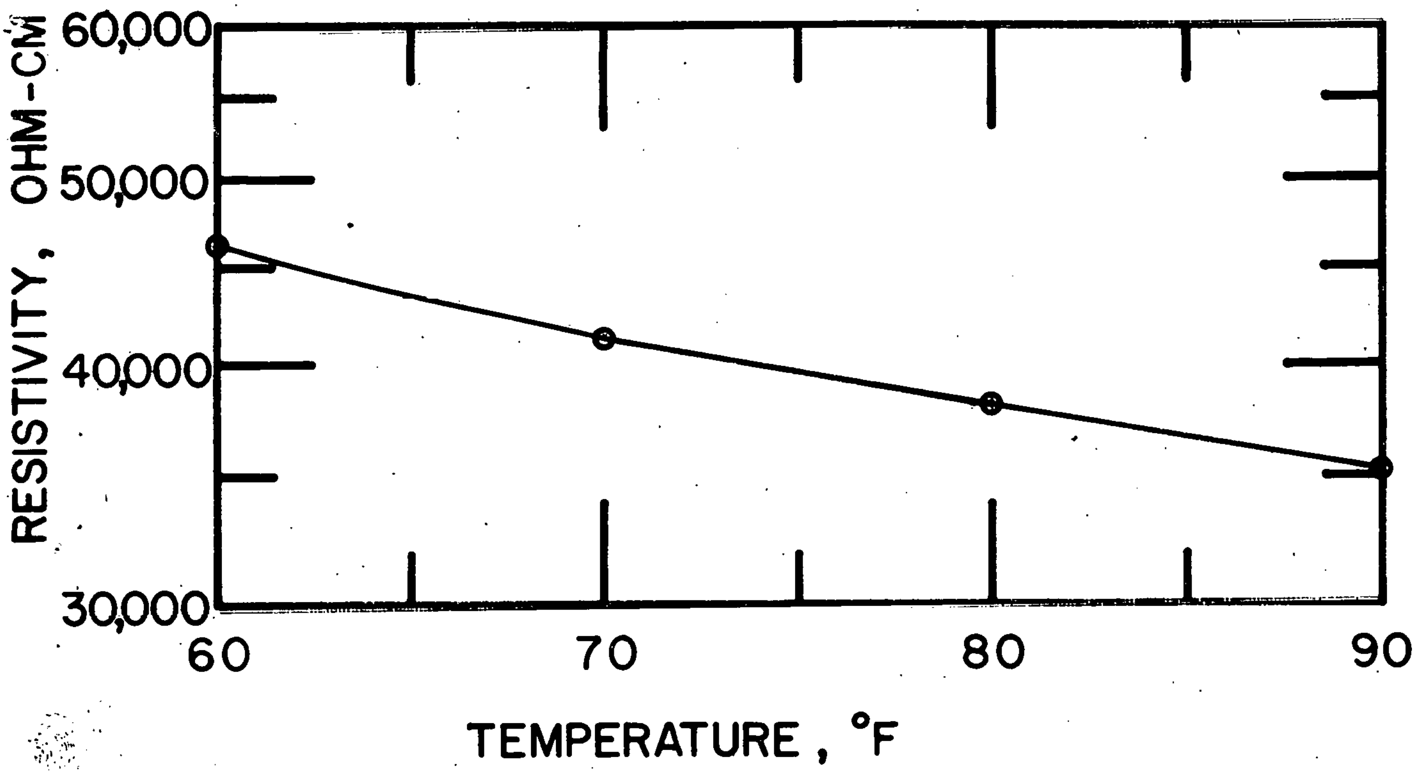


FIGURE 12

ELECTRICAL RESISTIVITY OF 0.001 N HCL IN DISTILLED WATER AS A FUNCTION OF TEMPERATURE

If the bulk fluid is divided into two layers, the heat generated in each layer is inversely propertional to the respective resistivity of the electrolyte in each layers. So, for a uniform volume heat source to exist in the bulk fluid, the electrical resistivity of both layers must be the same, i.e., the electrolyte resistivity must be independent of temperature.

With single-tube cooling, the temperature profile as shown in Figure 13, indicated the bulk temperature varied from 81°F at the lower layer to 85.8°F at the upper layer. From Figure 12, the corresponding variation in the electrolyte resistivity was from 37,800 ohm-cm at the bottom layer to 36,600 ohm-cm at the top layer. It was, therefore, evident that the electrolyte resistivity was temperature sensitive and that the volume heat source was not uniform. Due to the higher value of the electrolyte resistivity in the lower colder layer, less heat is being generated in this layer as compared with that in the upper, warmer layer. As time elapses, the lower layer becomes colder and the upper layer becomes warmer, due to the decreasing variation of the electrolyte resistivity with increasing temperature or vice versa. Consequently, density stratification developed in the bulk liquid with non-uniformity in the volume heat source.

With four cooling tubes, the temperature profile indicated the bulk temperature varied from 84.2°F at the bottom to 86°F at the top with a height of the bulk liquid

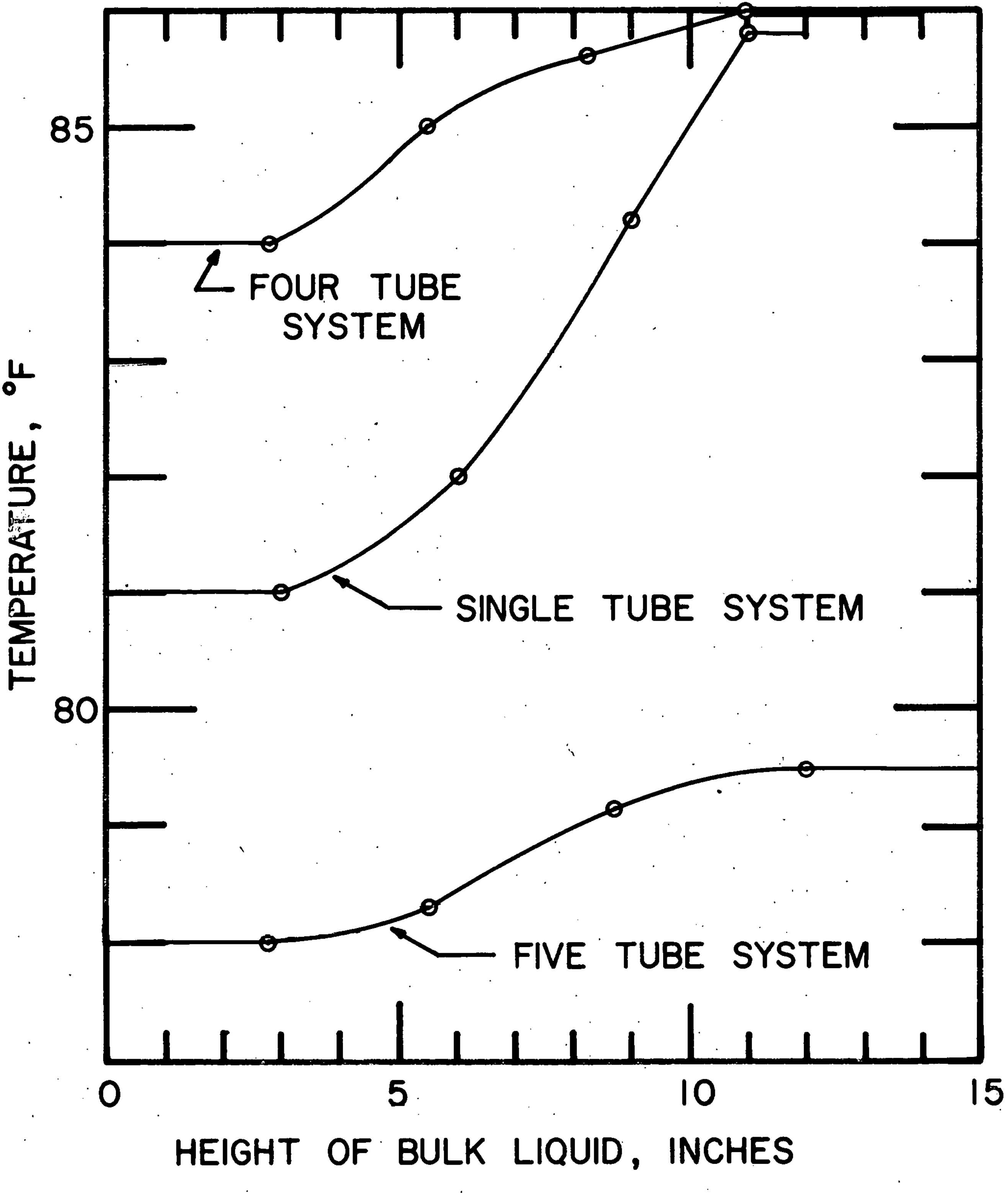


FIGURE 13

BULK TEMPERATURE PROFILE

of 14 inches. The corresponding variation in the electrolyte resistivity was from 36,870 ohm-cm at the bottom layer to 36,560 ohm-cm at the top layer. This showed only a slight variation in the electrolyte resistivity as compared with that of the single-tube case with the volume heat source quite uniform.

With five cooling tubes, the temperature profile indicated the bulk temperature varied from 78°F at the bottom to 79.5°F at the top with a height of the bulk liquid of 15 inches. The corresponding variation in the electrolyte resistivity was from 38,450 ohm-cm at the bottom layer to 38,150 at the top layer. This also indicated a slight variation in the electrolyte resistivity with the volume heat source in the bulk fluid again quite uniform.

B. Other Bulk Properties

The usual practice in free convection experiments is to evaluate the properties used in correlation equations at the "film" temperature. The validity of this method for volume heat source cases has not been explored but was used for want of a better technique. Therefore, the error involved by using the "film" temperature could not be determined.

Among the different physical properties of the electrolyte, the coefficient of volumetric expansion, β, and the viscosity seemed to vary a good deal with temperature change. These variations were assumed to have introduced an error in the value of the heat transfer coefficients of about one per cent.

Conduction Errors

Tube Wall Temperature Measurements

Possible heat conduction along the thermocouple lead wires due to the temperature difference between the bulk liquid and the tube wall would introduce an error in the tube-wall temperature measurements. This error may be negligible for small temperature differentials but may become significant as the temperature differentials increase. This error was partly minimized in the thermocouple installations made by wrapping part of the lead wires around the tube as shown in Figure 3.

For high temperature differentials, however, calculations were carried out to determine the magnitude of error in the tube wall temperature.

To estimate the thermocouple conduction error, use was made of the approximate equations of Schneider (17) which he derived for the same situation as encountered in this work. (Refer to Appendix C). A typical thermocouple conduction error in this particular case was calculated to be 0.18°F. Since this represented a calculable conduction effect, the measured temperatures were appropriately corrected.

One check on this calculation was to obtain the average tube wall temperature from the measured mixed mean inlet and outlet coolant temperatures and calculate the temperature change across the tube wall. It should be noted that the wall temperature actually measured represented the hottest part of the tube wall. Since the other parts of the tube wall were colder due to an increasing thickness of the boundary layer from the top of the tube, it was believed that the tube temperature would not be uniform. Since the actual thermocouple measurement was found to be within 0.40°F of the calculated average tube wall temperature, this was felt to provide at least "an order of magnitude" indication of the temperature variation around the periphery of the tube. This was equivalent to a heat transfer coefficient variation of around 4%.

Coolant Temperature Measurements

In the measurement of the coolant temperature differential, the discrepancy between the differential obtained by individual readings of the coolant inlet and outlet temperatures and that obtained by a direct difference reading by balancing the two thermocouples against each other without using a cold junction was less than one per cent. The former readings were found to be higher than the latter readings by less than one per cent. The latter readings were considered to be more reliable and were used as data.

Since the coolant temperature differential that was being measured was small, small errors involved in these measurements would still affect the heat transfer results. It was believed that these errors would be intoduced by the inaccuracies of the potentiometric readings as well as those of the thermocouples themselves. The potentiometer was readable to 0.001 mv or 0.043°F while the coolant temperature differentials ranged from 0.90°F to 1.8°F.

Accuracy of Thermocouples

The 36 gage Chromel-Alumel thermocouple wire used in all temperature measurements was calibrated using an ice bath and a constant boiling apparatus. Within the range of temperatures used in the experiments, the resultant calibration curve was found to be within less than one per cent of the Leeds and Northrup Co. Standard 31031 Conversion Tables for Chromel-Alumel thermocouple at a reference junction of 0°C. The L and N calibration curve was originally used to convert millivolt readings to degrees Fahrehheit.

The accuracy of the thermocouple readings were further checked when the coolant temperature increase was obtained by two methods. The individual readings of the coolant outlet and inlet thermocouples gave a difference which was within less than one per cent of the direct difference reading obtained by balancing the two thermcouples against each other without using a cold junction. This indicated that the transient changes in the individual inlet and outlet thermoccuple measurements as well as cold junction errors were negligible.

Experiments were also conducted to determine the effect of a passing alternating current on the readings of a suspended thermocouple in the bulk liquid. Although the findings of the experiments indicated the effect to be quite negligible, it was felt that there was actually some effect on the readings of the bulk thermocouple. If such an effect existed, the current would set up an e.m.f. along the suspended thermocouples which would tend to give higher bulk temperature readings. These higher readings result in a lower value of the heat transfer coefficients due to an increase in the bulk to tube wall temperature differential. The temperature data were appropriately corrected for this error.

Measurement of Coolant Flow Rate

The flow rate through the cooling tubes were actually measured individually for each run made. A stop watch was used to time the coolant flow and a Toledo weighing scale was used to measure the weight of the coolant. The accuracy of the two separate measurements was such that the error involved was considered to be about two per cent. Since the flow rate was directly proportional to the heat transfer coefficient, this introduced an error of about two per cent in the heat transfer coefficient.

Heat Conduction Through Insulation

Heat conducted into the two vertical ends of the tubes due to poor insulation would increase the coolant temperature differential, and hence, the value of the heat transfer coefficient. Soft rubber, 0.20" thick was used as insulation. To find out how effective this was as an insulator, calculations were made to determine the heat input into the insulated part of the tube (refer tc Appendix D). The calculations revealed that about 7.5 per cent of the total heat flux was due to the heat conducted through the insulation.

This effect was significant so the data were appropriately corrected.

Maximum Random Errors

On the basis of the different estimated random errors involved in the heat transfer measurements, the maximum value of the random errors was calculated using Sherwood and Reed's method for estimation of errors in a function of several variables as shown in Appendix D.

The calculations revealed that the experimental heat transfer coefficients were subject to a random error of abcut ± 8 per cent.

Reproducibility of Results

The reproducibility of the heat transfer results was checked by calculating the standard deviations of the results of the different runs. For the single tube case, the standard deviation was 7.5 while those for the four-tube and five-tube geometries were 4.6 and 3.3, respectively.

DISCUSSION OF RESULTS

Single Horizontal Cooling Tube

With single-tube cooling, the heat transfer coefficients with a volume heat source were slightly lower than those without a heat source. For the same case, Randall (12) predicted film coefficients with a volume heat source to be slightly higher than those without a heat source. Randall explained that the inclusion of the heat source raises the temperature of the fluid at every point in the boundary layer resulting in a higher temperature profile across this layer. This higher temperature profile resulted in a smaller value of the boundary layer thickness, δ, compared with δ_o without a heat source. In other words, $\delta_o > \delta$. Now since

$$\frac{h_o}{h_{nv}} = \frac{\frac{2K}{\delta}}{\frac{2K}{\delta_o}} = \frac{\delta_o}{\delta}$$

then, $\dfrac{h_o}{h_{nv}} > 1$

where

h_o = heat transfer coefficient with a volume heat source

h_{nv} = heat transfer coefficient without a heat source

For a 1/4" O.D. horizontal cylinder with a bulk to tube wall temperature differential of 20°F and a volume heat source of 60 $\frac{\text{Btu}}{\text{sec x ft}^3}$, Randall found by graphical integration that

$$\left[\frac{h_o}{h_{nv}}\right]_{avg} = 1.022$$

whereas experimental results gave

$$\left[\frac{h_o}{h_{nv}}\right]_{avg} = 0.90$$

Since temperature across the thermal boundary layer varied from a minimum at the tube wall to the bulk temperature at the outer thickness, electrolyte resistivity varied and less heat was generated at the colder region than at the warmer region. As a result, there will exist a gradient in the volume heat source across the boundary layer. But Randall's (12) analysis which assumed a uniform heat source across the boundary layer predicted that the presence of this heat source would give slightly higher film coefficients than those without a heat source. For a non-uniform heat source in the boundary layer, therefore, heat transfer coefficients would be predicted which were lower than for the same value with uniform heat source but higher than for no heat source.

Bonilla and Collins (1) in their experiments with a single horizontal tube without a volume heat source encountered the problem of density stratification in the bulk liquid which gave them lower heat transfer coefficients than expected.

Although the best line through the data fell below McAdam's correlation, scatter was such that no significant deviation from the correlation is apparent.

Four Horizontal Cooling Tubes

With four cooling tubes, the heat transfer coefficients with a volume heat source were slightly higher than those with no heat source present. This would be in agreement with Randall's analysis as mentioned for the single tube case.

Since the effect of the boundary layer temperature gradient would still apply in this case, and other conditions were presumably the same, reasons for the increase on the value of the heat transfer coefficients are not clearly apparent.

For the four cooling tubes, however, the heat transfer coefficients were not the same for each tube. The coefficients varied from five to fifteen per cent of each other. This indicated that there was some effect due to the geometry of the tube arrangement on the heat transfer coefficient. The variation may have been due to the setting up of small currents in the bulk fluid as discussed in the following section.

Five Horizontal Cooling Tubes

The heat transfer coefficients for each of the five horizontal cooling tubes with a volume heat source appeared to be somewhat higher than the usual correlation.

The differences in results between the single tube and the multiple tube cases could perhaps be explained by the setting up of small currents in the bulk fluid in the latter cases. Although long-time visual observations made in the multiple tube case did not reveal any small currents being set up in the bulk, it is possible that these did exist but were not noticeable. It is also likely that these would be different for each geometry.

CONCLUSIONS

1. With single-tube cooling, the heat transfer coefficients with a volume heat source were correlated by

   $$Nu = 0.49\ (Gr.Pr)^{\frac{1}{4}}$$

2. With four cooling tubes, the heat transfer coefficients were correlated by

   $$Nu = 0.59\ (Gr\ Pr)^{\frac{1}{4}}$$

3. With five cooling tubes, the heat transfer coefficients were correlated by

   $$Nu = 0.64\ (Gr\ Pr)^{\frac{1}{4}}$$

RECOMMENDATIONS

1. Single Horizontal Tube

It may be noted from Figure 4 that the experimental data is quite scattered. This is attributed to the fact that steady state was not reached in some of the runs even after a long time has elapsed.

It appears that the installation of a thermostatically-controlled closed cooling loop would help the system reach steady state sooner. This would ensure a more constant bulk to wall temperature differential. Also, if the coolant could be refrigerated, larger temperature differentials could be attained without increasing the tank temperature above room temperature.

As discussed under "Changes in Bulk Properties with Temperature," there existed a non-uniform volume heat source with single-tube cooling due to density stratification in the bulk fluid. This situation could be remedied by providing a compensating heat source, around the lower, colder region of the bulk fluid.

It was noted by Bonilla and Collins (1), however, that the location of this compensating heat source, when it was concentrated, had considerable effect on the measured coefficient, for a non-volumetric heat source. During a cooling run in which Bunsen burners were used, a 40% increase in heat transfer coefficient was observed when the four burners were moved two inches to one side of the points directly below the test cylinder. In the light of this result, the burners were replaced with hot plates which provided the requisite heat at low fluxes over a much wider area.

The problem could possibly be solved by positioning the heating grids such that they are inclined as a "V". In this position, more heat would be generated at the lower, colder layer due to lower resistance of the bulk in this layer as compared with the upper layer. This effect, then, would tend to neutralize the density stratification in the bulk. A somewhat more complicated electrode arrangement whereby the applied voltage could be increased in the cooler regions might also be successful.

Multiple Tube System

It was mentioned that the setting up of small currents in the bulk for the multiple tube case could perhaps explain the higher coefficients obtained in this case. No definite conclusions were reached, however, since these currents were not noticeable in the observations made. It is, therefore, recommended that further work should include development of better visualization techniques to determine whether such small currents existed in the bulk.

It was also noted that there were some variation in the heat transfer coefficients for each tube with position in the electrolyte. These results indicated that the geometry of tube arrangement in a multiple tube case has some effect on the heat transfer coefficient. Since only one geometry was studied for each of the four-tube and five-tube arrangements, it is recommended that this variable be investigated. Pertinent to this study, the effect of the "wake" in multiple tube systems should also be investigated.

NOMENCLATURE

A_o	Outside area of heat transfer, ft^2
C_p	Heat capacity at constant pressure, Btu/lb mass x ^oF
D_o	Outside tube diameter, ft.
g	Acceleration due to gravity, $\frac{ft}{hr^2}$
Gr	Grashof modulus $= \frac{\rho^2 g \beta \Delta t D_o^3}{\mu^2}$, dimensionless
h_o	Outside heat transfer coefficient with volume heat source, $\frac{Btu}{hr \times ft^2 \times {}^oF}$; h_{nv}, without volume heat source
k	Thermal conductivity, $\frac{Btu}{hr \times ft \times {}^oF}$
Nu	Nusselt modulus $= \frac{h_o D_o}{k}$, dimensionless
Pr	Prandlt modulus $= \frac{C_p \mu}{k}$, dimensionless
q	Heat rate through tube wall, Btu/hr
q'	Corrected heat rate through tube wall, Btu/hr
q'''	Volume heat source, $\frac{Btu}{hr \times ft^3}$
t	Temperature at any point; t_B, bulk temperature; t_w, tube-wall temperature, t_R, room temperature, ^oF
W	Coolant flow rate, lb/hr
Δt	Bulk to tube wall temperature differential, ^oF
Δt_{c-i}	Coolant temperature increase, ^oF
β	Coefficient of volumetric expansion $= - \frac{d\rho}{dt} \cdot \frac{1}{\rho}$, $1/^oF$
δ	Boundary layer thickness; δ_o, without volume heat source, ft

μ	Viscosity, lb mass/hr x ft
ρ	Density, lb mass/ft^3

Subscripts

b	Bulk
f	Film
w	Wall

Superscripts

Prime (') Corrected measurements and results

BIBLIOGRAPHY

1. Bonilla, C. F. and Collins, J. P., "Natural Convection Heating and Cooling by Horizontal Cylinders," NYO - 6217 (1953).

2. Eckert, E.R.G., "Introduction to the Transfer of Heat and Mass," 158-171, McGraw-Hill, New York, 1950.

3. Goldstein, S., "Modern Developments in Fluid Dynamics," Vol. II, First Edition, 638-643, Oxford Press, London, 1938.

4. Hallman, T. M., Paper 55-A-72, Am. Soc. Mech. Engrs., Chicago, 1955.

5. Hamilton, D. C., Poppendiek, H. F., Redmond, R. F., and Palmer, L. D., ORNL, 1769 (1954).

6. Hamilton, D. C., Lynch, F. E., "Free Convection Theory and Experiment in Fluids Having a Volume Heat Source," ORNL, 1888, (1955).

7. Hermann, R., Natl. Advisory Comm. Aeronaut., Tech. Memorandum 1366, (1954).

8. Hyman, S. C., Bonilla, C. F., "Heat Transfer by Natural Convection from Horizontal Cylinders to Liquid Metals; Final Report for July 1, 1949 to June 30, 1950," NYO - 560 (1950).

9. Jacob, M., "Heat Transfer," Vol. 1, 444, Wiley, New York, 1949.

10. Levy, S., J. Appl. Mech. 22, 515-522 (1955).

11. Lorenz, L., Annalen der Physik, 13, 582 (1881).

12. McAdams, W. H., "Heat Transmission," 175-180, McGraw-Hill, New York, 1954.

13. Nusselt, W., Gesundh. Ing., 38, 477, 490, (1915)

14. Ostrach, S., Natl. Advisory Comm. Aeronaut. Tech. Note 2863 (1952).

15. Randall, I. E., "Free Convection Heat Transfer to a Horizontal Cylinder from an Ordinary Fluid Containing a Volume Heat Source," M.S. Thesis, Purdue University, Lafayette, Ind., August 1957.

16. Schmidt, E., Beckmann, W., Tech. Mech. Thermodynam. 1, 1-24, 341, 391, (1930).

17. Schneider, P. J., "Conduction Heat Transfer", 173-181, Addison-Wesley, Cambridge, Mass., 1955.

18. Sparrow, E. M., Personal Communication, August, 1957.

APPENDIX A

CALCULATIONS

I. Method of Calculations

The original temperature data taken in millivolts was converted to degrees Fahrenheit by the use of a conversion table plot. The coolant temperature differential was obtained in degrees Fahreheit by dividing the millivolt difference, which was slight, by 0.023 mv/°F, the value given in the conversion tables for Chromel-Alumel thermocouple in the range of $32^{o}F$ to $212^{o}F$.

Heat transfer coefficients were calculated using two equations which follow:

$$q = W \ C_p \ \Delta t_{o-i} \tag{1}$$

$$q = h_o A_o \ \Delta t_{ov} \tag{2}$$

where

q = rate of heat transferred through the tube wall, Btu/hr

W = coolant flow rate, lbs/hr

Cp= heat capacity of coolant, Btu/lb x °F

Δt_{o-i} = temperature differential, inlet and exit coolant °F

h_o= outside heat transfer coefficient, $\frac{Btu}{hr \times ft^2 \times {}^oF}$

A_o = outside surface area of tube, ft^2

Δt_{ov} = temperature differential, bulk to tube wall, °F

Solving equation (1) and (2) for h_o, we get

$$h_o = \frac{W \ Cp \ \Delta t_{o-i}}{A_o \ \Delta t_{ov}} \tag{3}$$

II. Correlation of Data (Free Convection Heat Transfer)

A. Grashof modulus, N_{Gr} or Gr

$$Gr = \frac{D_o^3 \rho_f^2 \, g \beta_f \, \Delta t}{\mu_f^2}$$

where

D_o = outside diameter of cooling tube, ft.

g = acceleration due to gravity, ft/sec^2

β_f = coefficient of volumetric expansion, $(^{o}F)^{-1}$ at tf

ρ_f = density of bulk fluid at tf, lb/ft^3

μ_f = viscosity of bulk fluid at tf, lb/hr x ft

Δt = temperature differential, bulk to wall, ^{o}F

Properties were evaluated at the "film" temperature, the arithmetic mean between the tube wall and bulk fluid temperatures.

B. Prandlt number, Pr or N_{Pr}

$$Pr = \frac{C_p \mu_f}{k_f}$$

where

C_p = heat capacity of bulk fluid, Btu/lb x ^{o}F

μ_f = viscosity of bulk fluid, $\frac{lb}{hr \times ft}$

k_f = thermal conductivity of bulk fluid, Btu/hr x ft x ^{o}F

C. Nusselt number - Nu or N_{Nu}

$$Nu = \frac{h_o \, D_o}{k_f}$$

where

h_o = outside heat transfer coefficient, Btu/hr x ft^2 x ^{o}F

D_o = outside diameter of cooling tube, ft

k_f = thermal conductivity of bulk fluid, $\frac{\text{Btu}}{\text{hr x ft x }^o\text{F}}$

The usual correlation for heat transfer by natural convection (12) is:

$$Nu = 0.54 \left[Gr \times Pr \right]^{0.25}$$

III. Sample Calculations

Sample calculations which follow illustrate the methods used and indicate the assumptions made in obtaining the results.

DATA

Run No. 29

Coolant flow rate: 175 lb/hr

Coolant temperature increase: 1.44^{o}F

Corrected tube wall temperature: 65.0 ^{o}F

Corrected bulk temperature: 83.0 ^{o}F

Room temperature: 87.3^{o}F

Outside area of heat transfer: 0.0953 ft^2

The heat rate is fiven by the equation:

$$q = W\, C_p \Delta t_{o-i}$$

where

W = 175 lb/hr

Cp = 1.0 Btu/lb x ^{o}F

Δt_{o-i} = 1.44^{o}F

Solving for q,

$$q = 175\ (1)\ (1.44) = 252 \text{ Btu/hr}$$

The correction for heat conduction through the insulation is then applied:

$$q' = 252 - 19 = 233 \text{ Btu/hr}$$

then:

$$q' = h_o A_o \Delta t$$

where

h_o = outside coefficient of heat transfer, Btu/hr x ft^2 x ^{o}F

A_o = 0.0953 ft^2

$\Delta t = t_B - t_W = 83.0 - 65.0 = 18.0\ ^oF$

Solving for h_o,

$$h_o = \frac{q'}{A_o\ \Delta t} = \frac{233}{0.0953\ (18.0)} = 136 \quad \frac{\text{Btu}}{\text{hr x ft}^2 \text{ x } ^oF}$$

<u>Grashof and Prandlt Moduli</u>

$$t_f = \text{"film" temperature} = \frac{t_B + t_W}{2}$$

$$t_f = \frac{83.0 + 65.0}{2} = 74.0\ ^oF$$

From chart of $\left[\frac{\rho_f^2 \beta g}{\mu_f^2}\right] \left[\frac{C_p \mu_f}{k_f}\right]$ versus temperature, ^{o}F, especially prepared to simplify calculations (see Figure 14):

$$\alpha = 2.05 \times 10^8, \ \frac{1}{ft^3 \times {}^oF} \quad \text{at} \quad t_f = 74.0\ ^oF$$

$$N_{Gr} \cdot N_{Pr} = \alpha \times D^3 \times \Delta t$$

where

N_{Gr} = Grashof modulus

N_{Pr} = Prandlt number

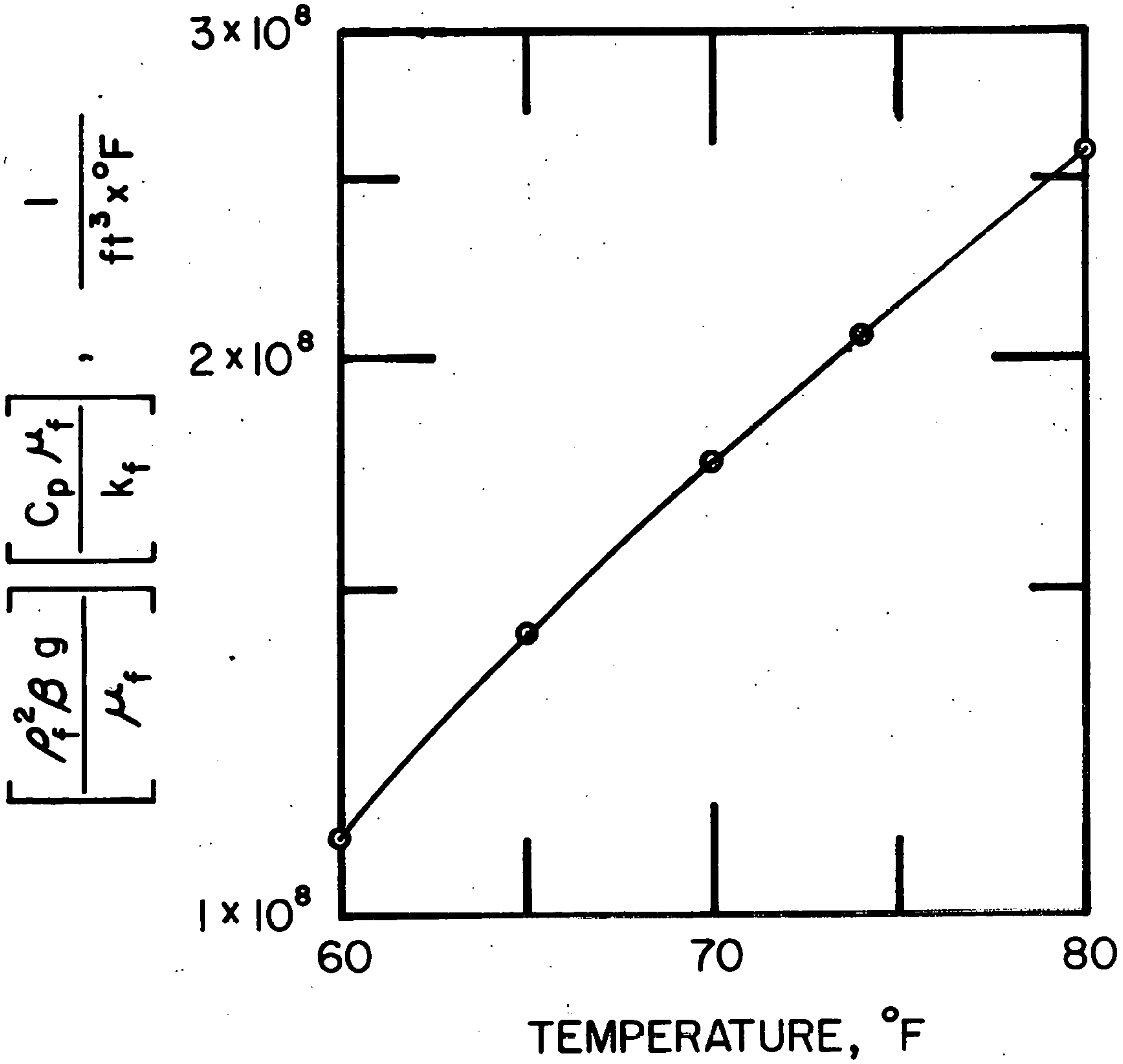


FIGURE 14

RELATIONSHIP BETWEEN FREE-CONVECTION HEAT TRANSFER PHYSICAL PROPERTIES OF WATER AND TEMPERATURE AT SATURATION PRESSURE

D^3 = outside diameter cubed = 9×10^{-6} ft^3

$\Delta t = t_s - t_W = 18.0^\circ F$

$$\alpha = \left[\frac{\rho_f^2 \beta\ g}{\mu_f^2}\right] \left[\frac{Cp\,\mu_f}{k_f}\right], \quad \frac{1}{ft^3 \times {}^\circ F}$$

Therefore,

$N_{Gr} \cdot N_{Pr} = 2.05 \times 10^8 \times 9 \times 10^{-6} \times 18.0$

$N_{Gr} \cdot N_{Pr} = 33,000$

But $N_{Pr} = 6.75$ (McAdam's Fig. A-7, p. 470) where $t_f = 74.0^\circ F$

So

$$N_{Gr} = \frac{33,000}{6.75}$$

$N_{Gr} = \underline{\underline{4880}}$

Nusselt Number

$$Nu = \frac{h_o\,D_o}{k_f} = \frac{136\ (0.0208)}{0.337}$$

where

$k_f = 0.337$ at $74^\circ F$

$Nu = \underline{\underline{8.40}}$

APPENDIX B

I. DATA

Tube No.	D_o	L	A	D_i
1	0.25	17.5	0.0953	0.1875
2	0.25	18.75	0.102	0.1875
3	0.25	16.75	0.091	0.1875
4	0.25	18.75	0.102	0.1875
5	0.25	20.75	0.113	0.1875

Legend:

D_o = outside tube diameter, inches

D_i = inside tube diameter, inches

L = tube length, inches

A = outside area of heat transfer, sq.ft.

W = coolant flow rate, lb/hr

t_B' = corrected bulk temperature, °F

t_W' = corrected wall temperature, °F

$\Delta t'$ = corrected bulk to wall temperature differential, °F

Δt_{o-i} = coolant temperature increase, °F

q' = heat transferred through the tube wall, corrected for heat conduction through insulation, Btu/hr

q''' = volume heat source, Btu/hr x ft^3

t_R = room temperature, °F

h_o' = corrected outside heat transfer coefficient, Btu/hr x ft^2 x °F

Nu = Nusselt number = $\frac{h_o D_o}{K}$

N_{Gr} = Grashof Modulus

N_{Pr} = Prandlt Modulus

DATA

Single Horizontal Cooling Tube

Run No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
1	168	78.0	65.2	12.8	0.967	146.2	103.0	78.0
2	102	78.5	58.3	20.2	1.890	173.7	229.0	78.5
3	102	78.5	59.4	19.1	1.975	181.8	229.0	78.5
4	102	78.5	60.3	18.2	1.930	177.3	229.0	78.5
5	102	78.0	61.9	16.6	1.400	128.7	229.0	78.5
6	102	78.0	64.3	13.7	1.010	92.7	229.0	78.5
7	102	78.0	70.3	7.8	0.440	60.0	229.0	78.5
8	192	71.0	61.1	9.9	0.483	83.4	229.0	77.0
9	168	78.0	63.8	14.2	1.140	172.8	103.0	78.0
10	250	78.0	60.5	17.5	0.660	148.5	103.0	78.0
11	225	78.0	61.6	16.4	0.612	123.7	103.0	78.0
12	222	78.1	65.0	13.1	0.880	176.4	158.5	77.0
13	222	78.1	64.0	14.1	0.880	176.4	158.5	77.0
14	222	78.1	62.2	15.9	0.880	176.4	158.5	77.0
15	222	78.1	61.7	16.4	0.880	176.4	158.5	77.0
16	222	78.1	61.2	16.9	0.880	176.4	158.5	77.0
17	222	78.1	61.5	16.6	0.880	176.4	158.5	77.0
18	222	78.1	60.8	17.3	0.880	176.4	158.5	77.0
19	264	77.5	63.6	13.9	0.833	208.0	229.0	77.5
20	264	77.5	61.6	15.9	0.880	208.8	229.0	77.5
21	264	78.0	61.5	16.6	0.880	208.8	229.0	77.5
22	264	78.0	58.9	19.1	0.967	229.5	229.0	77.5
23	264	78.0	58.2	19.8	0.967	229.5	229.0	77.5
24	264	78.0	57.3	20.7	1.050	249.3	229.0	77.5
25	264	78.5	57.2	21.3	1.050	249.3	229.0	77.5
26	264	78.5	56.7	21.8	1.050	249.3	229.0	77.5
27	264	78.5	56.5	22.0	1.050	249.3	229.0	77.5
28	264	78.5	56.8	22.7	1.050	249.3	229.0	77.5

Four Horizontal Cooling Tubes

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
29	1	175	83.0	65.0	18.0	1.44	233.0	412.0	87.3
	2	175	83.0	65.8	17.3	1.44	233.0	412.0	87.3
	3	175	83.0	68.6	14.4	1.44	233.0	412.0	87.3
	4	175	83.0	67.4	15.7	1.44	233.0	412.0	87.3
30	1	175	83.0	65.9	17.1	1.26	198.0	412.0	87.4
	2	175	83.0	66.8	16.2	1.26	198.0	412.0	87.4
	3	175	83.0	65.0	18.0	1.44	227.0	412.0	87.4
	4	175	83.0	66.1	16.9	1.44	227.0	412.0	87.4
31	1	175	83.0	66.1	16.9	1.26	198.0	412.0	87.5
	2	175	83.0	67.0	16.1	1.26	198.0	412.0	87.5
	3	175	83.0	65.0	18.0	1.26	198.0	412.0	87.5
	4	175	83.0	66.3	16.8	1.26	198.0	412.0	87.5
32	1	175	83.0	66.1	16.9	1.80	283.0	412.0	87.5
	2	175	83.0	66.3	16.7	1.80	283.0	412.0	87.5
	3	175	83.0	66.6	16.4	1.80	283.0	412.0	87.5
	4	175	83.0	67.7	15.5	1.80	283.0	412.0	87.5
33	1	220	83.0	62.0	21.0	1.26	249.0	508.0	87.0
	2	220	83.0	62.7	20.3	1.26	249.0	508.0	87.0
	3	220	83.0	63.0	20.0	1.26	249.0	508.0	87.0
	4	220	83.0	63.6	19.4	1.26	249.0	508.0	87.0
34	1	220	83.0	61.4	21.6	1.26	249.0	508.0	87.0
	2	220	83.0	61.7	21.3	1.26	249.0	508.0	87.0
	3	220	83.0	61.7	21.3	1.26	249.0	508.0	87.0
	4	220	83.0	62.5	20.5	1.26	249.0	508.0	87.0
35	1	220	83.0	61.7	21.3	1.80	356.0	508.0	87.1
	2	220	83.0	62.3	20.7	1.80	356.0	508.0	87.1
	3	220	83.0	62.5	20.5	1.80	356.0	508.0	87.1
	4	220	83.0	63.0	20.0	1.80	356.0	508.0	87.1
36	1	220	83.0	61.7	21.3	1.44	285.0	508.0	87.1
	2	220	83.0	62.3	20.7	1.44	285.0	508.0	87.1
	3	220	83.0	62.8	20.2	1.44	285.0	508.0	87.1
	4	220	83.0	63.5	19.5	1.44	285.0	508.0	87.1
37	1	125	83.0	70.0	13.1	1.44	162.0	323.0	88.0
	2	125	83.0	70.4	12.6	1.44	162.0	323.0	88.0
	3	125	83.0	70.8	12.2	1.44	162.0	323.0	88.0
	4	125	83.0	71.1	11.9	1.44	162.0	323.0	88.0

Four Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
38	1	125	83.0	70.0	13.1	1.26	142.2	323.0	88.1
	2	125	83.0	70.4	12.6	1.26	142.2	323.0	88.1
	3	125	83.0	70.8	12.2	1.26	142.2	323.0	88.1
	4	125	83.0	71.1	11.9	1.26	142.2	323.0	88.1
39	1	125	83.0	69.3	13.7	1.62	183.0	323.0	88.1
	2	125	83.0	69.9	13.1	1.62	183.0	323.0	88.1
	3	125	83.0	69.1	13.9	1.44	183.0	323.0	88.1
	4	125	83.0	69.6	13.5	1.44	183.0	323.0	88.1
40	1	125	83.0	68.1	14.9	1.62	183.0	323.0	88.2
	2	125	83.0	68.8	14.2	1.62	183.0	323.0	88.2
	3	125	83.0	68.6	14.4	1.80	183.0	323.0	88.2
	4	125	83.0	69.5	13.5	1.80	183.0	323.0	88.2
41	1	125	83.0	68.3	14.8	1.80	202.0	323.0	88.2
	2	125	83.0	68.6	14.4	1.80	202.0	323.0	88.2
	3	125	83.0	68.6	14.4	1.80	202.0	323.0	88.2
	4	125	83.0	68.6	14.4	1.80	202.0	323.0	88.2
42	1	125	83.0	68.9	14.1	1.62	183.0	323.0	88.1
	2	125	83.0	68.2	13.9	1.62	183.0	323.0	88.1
	3	125	83.0	69.3	13.7	1.62	183.0	323.0	88.1
	4	125	83.0	69.5	13.5	1.62	183.0	323.0	88.1
43	1	125	82.0	68.3	13.7	1.62	218.7	398.0	87.2
	2	125	82.0	66.0	16.0	1.62	218.7	398.0	87.2
	3	125	82.0	66.1	15.9	1.62	218.7	398.0	87.2
	4	125	82.0	66.9	15.1	1.62	218.7	398.0	87.2
44	1	125	82.0	66.9	15.1	1.26	170.0	398.0	87.4
	2	125	82.0	65.3	16.7	1.26	170.0	398.0	87.4
	3	125	82.0	65.3	16.7	1.26	170.0	398.0	87.4
	4	125	82.0	66.2	15.9	1.26	170.0	398.0	87.4
45	1	125	82.0	66.6	15.5	1.62	218.7	398.0	87.4
	2	125	82.0	64.8	17.3	1.62	218.7	398.0	87.4
	3	125	82.0	64.8	17.3	1.62	218.7	398.0	87.4
	4	125	82.0	64.8	17.3	1.62	218.7	398.0	87.4
46	1	150	82.0	64.0	18.0	1.80	243.0	398.0	87.4
	2	150	82.0	64.4	17.7	1.80	243.0	398.0	87.4
	3	150	82.0	64.8	17.3	1.80	243.0	398.0	87.4
	4	150	82.0	65.1	16.9	1.80	243.0	398.0	87.4

Four Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
47	1	150	82.0	65.3	16.8	1.62	218.7	398.0	87.2
	2	150	82.0	65.6	16.4	1.62	218.7	398.0	87.2
	3	150	82.0	66.4	15.7	1.62	218.7	398.0	87.2
	4	150	82.0	66.6	15.5	1.62	218.7	398.0	87.2
48	1	187	82.0	63.6	18.4	1.62	274.0	508	87.0
	2	187	82.0	64.0	18.0	1.62	274.0	508	87.0
	3	187	82.0	64.9	17.1	1.62	274.0	508	87.0
	4	187	82.0	65.4	16.6	1.62	274.0	508	87.0
49	1	200	82.0	61.8	20.2	1.44	259.0	508	87.1
	2	200	82.0	62.2	19.8	1.44	259.0	508	87.1
	3	200	82.0	62.9	19.1	1.44	259.0	508	87.1
	4	200	82.0	63.3	18.7	1.44	259.0	508	87.1
50	1	200	82.0	62.4	19.6	1.26	230.4	412	87.1
	2	200	82.0	62.9	19.1	1.26	230.4	412	87.1
	3	200	82.0	63.3	18.7	1.26	230.4	412	87.1
	4	200	82.0	63.8	18.2	1.26	230.4	412	87.1
51	1	200	82.0	63.1	18.9	1.08	194.4	412	87.1
	2	200	82.0	63.7	18.4	1.08	194.4	412	87.1
	3	200	82.0	64.0	18.0	1.08	194.4	412	87.1
	4	200	82.0	64.3	17.7	1.08	194.4	412	87.1
52	1	200	82.0	63.1	18.9	1.26	230.4	412	87.1
	2	200	82.0	63.7	18.4	1.26	230.4	412	87.1
	3	200	82.0	64.4	17.6	1.26	230.4	412	87.1
	4	200	82.0	64.6	17.5	1.26	230.4	412	87.1
53	1	200	82.0	63.1	18.9	1.26	230.4	412	87.1
	2	200	82.0	63.7	18.4	1.26	230.4	412	87.1
	3	200	82.0	64.2	17.8	1.26	230.4	412	87.1
	4	200	82.0	64.6	17.5	1.26	230.4	412	87.1
54	1	187	83.0	64.4	18.6	1.26	212.4	412	87.1
	2	187	83.0	64.4	18.6	1.26	212.4	412	88.1
	3	187	83.0	64.8	18.2	1.26	212.4	412	88.1
	4	187	83.0	65.8	17.3	1.26	212.4	412	88.1
55	1	171	83.0	64.7	18.4	1.44	222.0	412	88.1
	2	171	83.0	64.7	18.4	1.44	222.0	412	88.2
	3	171	83.0	65.9	17.1	1.44	222.0	412	88.2
	4	171	83.0	65.9	17.1	1.44	222.0	412	88.2

Four Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
56	1	171	83.0	64.65	18.4	1.26	194.4	412.0	88.2
	2	171	83.0	64.65	18.4	1.26	194.4	412.0	88.2
	3	171	83.0	65.9	17.1	1.26	194.4	412.0	88.3
	4	171	83.0	65.9	17.1	1.26	194.4	412.0	88.3
57	1	162	83.0	64.3	18.7	1.44	210.6	412.0	88.3
	2	162	83.0	64.3	18.7	1.44	210.6	412.0	88.3
	3	162	83.0	65.4	17.7	1.44	210.6	412.0	88.3
	4	162	83.0	65.4	17.7	1.44	210.6	412.0	88.3
58	1	162	83.0	64.7	18.4	1.44	210.6	412.0	88.3
	2	162	83.0	64.7	18.4	1.44	210.6	412.0	88.3
	3	162	83.0	65.7	17.3	1.44	210.6	412.0	88.3
	4	162	83.0	65.7	17.3	1.44	210.6	412.0	88.3
59	1	162	83.0	64.7	18.4	1.44	210.6	412.0	88.3
	2	162	83.0	64.7	18.4	1.44	210.6	412.0	88.3
	3	162	83.0	65.6	17.5	1.44	210.6	412.0	88.2
	4	162	83.0	65.6	17.5	1.44	210.6	412.0	88.2
60	1	162	83.0	64.7	18.4	1.44	210.6	412.0	88.2
	2	162	83.0	64.7	18.4	1.44	210.6	412.0	88.2
	3	162	83.0	65.6	17.5	1.44	210.6	412.0	88.2
	4	162	83.0	65.6	17.5	1.44	210.6	412.0	88.2
61	1	150	82.5	69.7	12.8	1.26	170.0	183.0	87.5
	2	120	82.5	69.7	12.8	1.26	170.0	183.0	87.5
	3	150	82.5	71.0	11.5	1.26	170.0	183.0	87.5
	4	150	82.5	71.0	11.5	1.26	170.0	183.0	87.5
62	1	135	82.5	70.3	12.2	1.08	131.0	183.0	87.5
	2	135	82.5	70.3	12.2	1.08	131.0	183.0	87.5
	3	135	82.5	71.5	11	1.08	131.0	183.0	87.5
	4	135	82.5	71.5	11	1.08	131.0	183.0	87.5
63	1	125	82.5	70.6	11.9	1.26	142.2	183.0	87.5
	2	125	82.5	70.6	11.9	1.26	142.2	183.0	87.5
	3	125	82.5	71.7	10.8	1.26	142.2	183.0	87.5
	4	125	82.5	71.7	10.8	1.26	142.2	183.0	87.5
64	1	125	82.5	70.8	11.9	1.26	142.2	183.0	87.5
	2	125	82.5	70.8	11.9	1.26	142.2	183.0	87.6
	3	125	82.5	71.5	11.5	1.26	142.0	183.0	87.6
	4	125	82.5	71.5	11.5	1.26	142.0	183.0	87.6

Four Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
65	1	125	82.5	70.8	11.7	1.26	142.2	183.0	87.6
	2	125	82.5	70.8	11.7	1.26	142.2	183.0	87.6
	3	125	82.5	72.0	10.5	1.26	142.2	183.0	87.6
	4	125	82.5	72.0	10.5	1.26	142.0	183.0	87.6
66	1	125	82.5	70.7	11.9	1.44	162.0	183.0	87.6
	2	125	82.5	70.7	11.9	1.44	162.0	183.0	87.6
	3	125	82.5	71.7	10.8	1.44	162.0	183.0	87.6
	4	125	82.5	71.7	10.8	1.44	162.0	183.0	87.6
67	1	126	83	72.9	10.1	1.26	143.0	250.0	87.0
	2	126	83	72.9	10.1	1.26	143.0	250.0	88.0
	3	126	83	73.8	9.2	1.26	143.0	250.0	88.0
	4	126	83	73.8	9.2	1.26	143.0	250.0	88.0
68	1	126	83	71.7	11.3	1.26	143.0	250.0	88.0
	2	126	83	71.7	11.3	1.26	143.0	250.0	88.1
	3	126	83	72.6	10.4	1.26	143.0	250.0	88.1
	4	126	83	72.6	10.4	1.26	143.0	250.0	88.1
69	1	126	83	71.1	11.9	1.26	143.0	250.0	88.1
	2	126	83	71.1	11.9	1.26	143.0	250.0	88.1
	3	126	83	72.2	10.8	1.26	143.0	250.0	88.1
	4	126	83	72.2	10.8	1.26	143.0	250.0	88.1
70	1	126	83	71.3	11.7	1.26	143.0	250.0	88.1
	2	126	83	71.3	11.7	1.26	143.0	250.0	88.1
	3	126	83	72.2	10.8	1.26	143.0	250.0	88.0
	4	126	83	72.2	10.8	1.26	143.0	250.0	88.0
71	1	110	83	72.4	10.6	0.90	89.0	250.0	88.0
	2	110	83	72.4	10.6	0.90	89.0	250.0	88.0
	3	110	83	73.4	9.5	0.90	89.0	250.0	88.1
	4	110	83	73.4	9.5	0.90	89.0	250.0	88.1
72	1	110	83	74.0	9.0	0.90	89.0	250.0	88.1
	2	110	83	74.0	9.0	0.90	89.0	250.0	88.1
	3	110	83	74.7	8.3	0.90	89.0	250.0	88.3
	4	110	83	74.7	8.3	0.90	89.0	250.0	88.3
73	1	110	83	72.9	10.1	0.90	107.0	250.0	88.3
	2	110	83	72.9	10.1	0.90	107.0	250.0	88.3
	3	110	83	74.0	9.0	0.90	107.0	250.0	88.3
	4	110	83	74.0	9.0	0.90	107.0	250.0	88.3

Four Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
74	1	110	83.0	72.8	10.3	0.90	107	250.0	88.3
	2	110	83.0	72.8	10.3	0.90	107	250.0	88.3
	3	110	83.0	74.0	9.0	0.90	107	250.0	88.3
	4	110	83.0	74.0	9.0	0.90	107	250.0	88.3
75	1	110	83.0	72.8	10.3	1.62	161	250.0	88.2
	2	110	83.0	72.8	10.3	1.62	161	250.0	88.2
	3	110	83.0	73.8	9.2	1.62	161	250.0	88.2
	4	110	83.0	73.8	9.2	1.62	161	250.0	88.2
76	1	110	83.0	72.8	10.3	1.26	125	250.0	88.2
	2	110	83.0	72.8	10.3	1.26	125	250.0	88.2
	3	110	83.0	73.8	9.2	1.26	125	250.0	88.2
	4	110	83.0	73.8	9.2	1.26	125	250.0	88.2
77	1	87	84.0	73.9	10.1	1.44	113	102.5	88.2
	2	87	84.0	73.9	10.1	1.44	113	102.5	88.2
	3	87	84.0	75.0	9.0	1.44	113	102.5	88.5
	4	87	84.0	75.0	9.0	1.44	113	102.5	88.5
78	1	87	84.0	74.8	9.2	1.44	113	102.5	88.5
	2	87	84.0	74.8	9.2	1.44	113	102.5	88.5
	3	87	84.0	75.5	8.5	1.44	113	102.5	88.5
	4	87	84.0	75.5	8.5	1.44	113	102.5	88.5
79	1	87	84.0	75.2	8.8	1.44	113	102.5	88.5
	2	87	84.0	75.2	8.8	1.44	113	102.5	88.6
	3	87	84.0	76.3	7.7	1.44	113	102.5	88.6
	4	87	84.0	76.3	7.7	1.44	113	102.5	88.6
80	1	87	84.0	76.1	7.9	1.44	113	102.5	88.6
	2	87	84.0	76.1	7.9	1.44	113	102.5	88.6
	3	87	84.0	76.8	7.2	1.44	113	102.5	88.6
	4	87	84.0	76.8	7.2	1.44	113	102.5	88.6
81	1	87	84.0	76.1	7.9	1.26	99	102.5	88.6
	2	87	84.0	76.1	7.9	1.26	99	102.5	88.6
	3	87	84.0	76.8	7.2	1.26	99	102.5	88.5
	4	87	84.0	76.8	7.2	1.26	99	102.5	88.5
82	1	87	84.0	76.4	7.6	1.26	99	102.5	88.5
	2	87	84.0	76.4	7.6	1.26	99	102.5	88.5
	3	87	84.0	76.8	7.2	1.26	99	102.5	88.5
	4	87	84.0	76.8	7.2	1.26	99	102.5	88.5

Four Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B	t_W	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
83	1	100	83	68.6	14.4	1.80	162.0	380.0	87.9
	2	100	83	68.6	14.4	1.80	162.0	380.0	87.9
	3	100	83	69.5	13.5	1.80	162.0	380.0	87.9
	4	100	83	69.5	13.5	1.80	162.0	380.0	87.9
84	1	100	83	67.9	15.1	1.80	162.0	380.0	87.9
	2	100	83	67.9	15.1	1.80	162.0	380.0	87.9
	3	100	83	68.4	14.6	1.80	162.0	380.0	88
	4	100	83	68.4	14.6	1.80	162.0	380.0	88
85	1	100	83	67.7	15.3	1.80	162.0	380.0	88
	2	100	83	67.7	15.3	1.80	162.0	380.0	88.1
	3	100	83	68.4	14.6	1.80	162.0	380.0	88.1
	4	100	83	68.4	14.6	1.80	162.0	380.0	88.1
86	1	100	83	66.8	16.2	1.80	162.0	380.0	88.1
	2	100	83	66.8	16.2	1.80	162.0	380.0	88.1
	3	100	83	67.7	15.3	1.80	162.0	380.0	88.1
	4	100	83	67.7	15.3	1.80	162.0	380.0	88.1
87	1	100	83	67.0	16.0	1.80	162.0	380.0	88.1
	2	100	83	67.0	16.0	1.80	162.0	380.0	88.1
	3	100	83	67.9	15.1	1.80	162.0	380.0	88.1
	4	100	83	67.9	15.1	1.80	162.0	380.0	88.1
88	1	100	83	66.8	16.2	1.98	178.0	380.0	88.1
	2	100	83	66.8	16.2	1.98	178.0	380.0	88.1
	3	100	83	67.5	15.5	1.98	178.0	380.0	88.0
	4	100	83	67.5	15.5	1.98	178.0	380.0	88.0
89	1	140	84	68.0	16.0	1.80	226.8	380.0	88.5
	2	140	84	68.0	16.0	1.80	226.8	380.0	88.5
	3	140	84	68.7	15.3	1.80	226.8	380.0	88.5
	4	140	84	68.7	15.3	1.80	226.8	380.0	88.5
90	1	140	84	67.8	16.2	1.80	226.8	380.0	88.5
	2	140	84	67.8	16.2	1.80	226.8	380.0	88.5
	3	140	84	68.7	15.3	1.80	226.8	380.0	88.6
	4	140	84	68.7	15.3	1.80	226.8	380.0	88.6
91	1	140	84	68.0	16.0	1.80	226.8	380.0	88.6
	2	140	84	68.0	16.0	1.80	226.8	380.0	88.6
	3	140	84	68.4	15.7	1.80	226.8	380.0	88.6
	4	140	84	68.4	15.7	1.80	226.8	380.0	88.6

Four Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
92	1	140	84	67.4	16.6	1.80	226.8	380.0	88.6
	2	140	84	67.4	16.6	1.80	226.8	380.0	88.6
	3	140	84	68.5	15.5	1.80	226.8	380.0	88.6
	4	140	84	68.5	15.5	1.80	226.8	380.0	88.6
93	1	140	84	67.4	16.6	1.80	226.8	380.0	88.6
	2	140	84	67.4	16.6	1.80	226.8	380.0	88.6
	3	140	84	68.5	15.5	1.80	226.8	380.0	88.6
	4	140	84	68.5	15.5	1.80	226.8	380.0	88.6
94	1	140	84	67.4	16.6	1.80	226.8	380.0	88.6
	2	140	84	67.4	16.6	1.80	226.8	380.0	88.6
	3	140	84	68.5	15.5	1.80	226.8	380.0	88.6
	4	140	84	68.5	15.5	1.80	226.8	380.0	88.6
95	1	165	84	69.3	14.7	1.62	240.0	380.0	88.7
	2	165	84	69.3	14.7	1.62	240.0	380.0	88.7
	3	165	84	70.15	13.9	1.62	240.0	380.0	88.7
	4	165	84	70.15	13.9	1.62	240.0	380.0	88.7
96	1	165	84	68.9	15.1	1.44	214.0	380.0	88.8
	2	165	84	68.9	15.1	1.44	214.0	380.0	88.8
	3	165	84	69.8	14.2	1.44	214.0	380.0	88.8
	4	165	84	69.8	14.2	1.44	214.0	380.0	88.8
97	1	165	84	68.9	15.1	1.26	187.0	380.0	88.8
	2	165	84	68.9	15.1	1.26	187.0	380.0	88.8
	3	165	84	69.8	14.2	1.26	187.0	380.0	88.8
	4	165	84	69.8	14.2	1.26	187.0	380.0	88.8
98	1	165	84	67.6	16.4	1.62	240.0	380.0	88.7
	2	165	84	67.6	16.4	1.62	240.0	380.0	88.7
	3	165	84	68.3	15.7	1.62	240.0	380.0	88.7
	4	165	84	68.3	15.7	1.62	240.0	380.0	88.7
99	1	165	84	67.6	16.4	1.62	240.0	380.0	88.6
	2	165	84	67.6	16.4	1.62	240.0	380.0	88.6
	3	165	84	68.3	15.7	1.62	240.0	380.0	88.6
	4	165	84	68.3	15.7	1.62	240.0	380.0	88.6
100	1	165	84	67.6	16.4	1.62	240.0	380.0	88.5
	2	165	84	67.6	16.4	1.62	240.0	380.0	88.5
	3	165	84	68.3	15.7	1.62	240.0	380.0	88.5
	4	165	84	68.3	15.7	1.62	240.0	380.0	88.5

Five Horizontal Cooling Tubes

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
101	1	180	78	64.0	14.0	1.98	320.0	535.0	87.5
	2	198	78	63.0	15.0	1.98	340.0	535.0	87.5
	3	204	78	63.0	15.0	1.98	340.0	535.0	87.5
	4	204	78	62.0	16.0	1.98	340.0	535.0	87.5
	5	180	78	65.0	13.0	1.98	320.0	535.0	87.5
102	1	180	78	64.5	14.5	1.98	340.0	535.0	87.5
	2	198	78	62.8	15.2	1.98	340.0	535.0	87.5
	3	204	78	62.8	15.2	1.98	340.0	535.0	87.5
	4	204	78	62.5	15.5	1.98	340.0	535.0	87.5
	5	180	78	65.0	13.0	1.98	320.0	535.0	87.5
103	1	180	78	63.0	15.0	1.98	320.0	535.0	88.7
	2	198	78	62.6	15.4	1.98	340.0	535.0	88.7
	3	204	78	62.6	15.4	1.98	340.0	535.0	88.7
	4	204	78	62.5	15.5	1.98	340.0	535.0	88.7
	5	180	78	64.9	13.1	1.98	320.0	535.0	88.7
104	1	180	78	62.9	15.1	1.98	320.0	535.0	89.6
	2	198	78	62.5	15.5	1.98	340.0	535.0	89.6
	3	204	78	62.5	15.5	1.98	340.0	535.0	89.6
	4	204	78	62.4	15.6	1.98	340.0	535.0	89.6
	5	180	78	64.0	14.0	1.98	320.0	535.0	89.6
105	1	180	78	62.7	15.3	1.98	320.0	535.0	89.6
	2	198	78	62.4	15.6	1.98	340.0	535.0	89.6
	3	204	78	62.4	15.6	1.98	340.0	535.0	89.6
	4	204	78	62.3	15.7	1.98	340.0	535.0	89.6
	5	180	78	64.0	14.0	1.98	320.0	535.0	89.6
106	1	180	78	62.6	15.4	1.98	320.0	535.0	89.6
	2	198	78	62.3	15.7	1.98	340.0	535.0	89.6
	3	204	78	62.3	15.7	1.98	340.0	535.0	89.6
	4	204	78	62.2	15.8	1.98	340.0	535.0	89.6
	5	180	78	63.9	14.1	1.98	320.0	535.0	89.6
107	1	180	78	62.6	15.4	1.98	320.0	535.0	89.6
	2	198	78	62.3	15.7	1.98	340.0	535.0	89.6
	3	204	78	62.3	15.7	1.98	340.0	535.0	89.6
	4	204	78	62.2	15.8	1.98	340.0	535.0	89.6
	5	180	78	63.9	14.1	1.98	320.0	535.0	89.6

Five Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
108	1	110	80	67.0	13.0	2.3	232.0	535.0	86.0
	2	110	80	67.0	13.0	2.3	232.0	535.0	86.0
	3	115	80	66.5	13.5	1.8	187.0	535.0	86.0
	4	115	80	66.5	13.5	1.8	187.0	535.0	86.0
	5	115	80	66.8	13.2	2.7	279.0	535.0	86.0
109	1	110	80	66.5	13.5	2.3	232.0	535.0	86.3
	2	110	80	66.5	13.5	2.3	232.0	535.0	86.3
	3	115	80	66.3	13.7	1.8	187.0	535.0	86.3
	4	115	80	66.3	13.7	1.8	187.0	535.0	86.3
	5	115	80	66.6	13.4	2.7	279.0	535.0	86.3
110	1	110	80	66.3	13.7	2.3	232.0	535.0	86.3
	2	110	80	66.3	13.7	2.3	232.0	535.0	86.3
	3	110	80	66.2	13.8	2.3	187.0	535.0	86.3
	4	110	80	66.2	13.8	2.3	187.0	535.0	86.3
	5	110	80	66.5	13.5	2.7	279.0	535.0	86.3
111	1	110	80	66.1	13.9	2.3	232.0	535.0	86.7
	2	110	80	66.1	13.9	2.3	232.0	535.0	86.7
	3	110	80	66.1	13.9	2.3	187.0	535.0	86.7
	4	110	80	66.1	13.9	2.3	187.0	535.0	86.7
	5	110	80	66.3	13.7	2.7	279.0	535.0	86.7
112	1	110	80	65.9	14.1	2.3	232.0	535.0	87.5
	2	110	80	65.9	14.1	2.3	232.0	535.0	87.5
	3	110	80	66.0	14.0	2.3	187.0	535.0	87.5
	4	110	80	66.0	14.0	2.3	187.0	535.0	87.5
	5	110	80	66.1	13.9	2.7	187.0	535.0	87.5
113	1	110	80	65.8	14.2	2.3	232.0	535.0	87.8
	2	110	80	65.8	14.2	2.3	232.0	535.0	87.8
	3	110	80	65.9	14.1	2.3	187.0	535.0	87.8
	4	110	80	65.9	14.1	2.3	187.0	535.0	87.8
	5	110	80	66.0	14.0	2.7	279.0	535.0	87.8
114	1	110	80	65.8	14.2	2.3	232.0	535.0	87.8
	2	110	80	65.8	14.2	2.3	232.0	535.0	87.8
	3	110	80	65.9	14.1	2.3	187.0	535.0	87.8
	4	110	80	65.9	14.1	2.3	187.0	535.0	87.8
	5	110	80	66.0	14.0	2.7	279.0	535.0	87.8

Five Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B	t_W	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
115	1	140	88	70.0	18.0	2.3	214.0	620.0	87.9
	2	140	88	70.0	18.0	2.3	214.0	620.0	87.9
	3	140	88	69.8	18.2	2.3	214.0	620.0	87.9
	4	140	88	69.8	18.2	2.3	214.0	620.0	87.9
	5	140	88	70	18	2.7	340.0	620.0	87.9
116	1	135	88	70.2	17.8	2.3	284.0	620.0	88.7
	2	135	88	70.2	17.8	2.3	284.0	620.0	88.7
	3	135	88	70.0	18.0	2.3	284.0	620.0	88.7
	4	135	88	70.0	18.0	2.3	284.0	620.0	88.7
	5	135	88	70.1	17.9	2.7	327.6	620.0	88.7
117	1	135	88	70.4	17.6	2.3	284.0	620.0	88.7
	2	135	88	70.4	17.6	2.3	284.0	620.0	88.7
	3	135	88	70.2	17.8	2.3	284.0	620.0	88.7
	4	135	88	70.2	17.8	2.3	284.0	620.0	88.7
	5	135	88	70.3	17.7	2.7	327.6	620.0	88.7
118	1	135	88	70.6	17.4	2.3	284.0	620.0	88.7
	2	135	88	70.6	17.4	2.3	284.0	620.0	88.7
	3	135	88	70.4	17.6	2.3	284.0	620.0	88.7
	4	135	88	70.4	17.6	2.3	284.0	620.0	88.7
	5	135	88	70.5	17.5	2.7	327.6	620.0	88.7
119	1	135	88	71.0	17.0	2.3	284.0	620.0	88.7
	2	135	88	71.0	17.0	2.3	284.0	620.0	88.7
	3	135	88	70.7	17.3	2.3	284.0	620.0	88.7
	4	135	88	70.7	17.3	2.3	284.0	620.0	88.7
	5	135	88	70.9	17.1	2.7	327.6	620.0	88.7
120	1	135	88	71.3	16.7	2.3	284.0	620.0	88.7
	2	135	88	71.3	16.7	2.3	284.0	620.0	88.7
	3	135	88	71.0	17.0	2.3	284.0	620.0	88.7
	4	135	88	71.0	17.0	2.3	284.0	620.0	88.7
	5	135	88	71.2	16.8	2.7	327.6	620.0	88.7
121	1	175	89	67.4	21.6	2.2	340.0	620.0	89.6
	2	175	89	67.4	21.6	2.2	340.0	620.0	89.6
	3	180	89	65.6	23.4	1.9	320.0	620.0	89.6
	4	180	89	65.6	23.4	1.9	320.0	620.0	89.6
	5	175	89	68.1	20.9	2.2	340.0	620.0	89.6

Five Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
122	1	175	89	67.9	21.1	2.2	340.0	630.0	89.6
	2	175	89	76.9	21.1	2.2	340.0	630.0	89.6
	3	180	89	66.1	22.9	1.9	320.0	630.0	89.6
	4	180	89	66.1	22.9	1.9	320.0	630.0	89.6
	5	175	89	68.5	20.5	2.2	340.0	630.0	89.6
123	1	175	89	68.5	20.6	2.2	340.0	630.0	89.6
	2	175	89	68.5	20.6	2.2	340.0	630.0	89.6
	3	180	89	66.6	22.4	1.9	320.0	630.0	89.6
	4	180	89	66.6	22.4	1.9	320.0	630.0	89.6
	5	175	89	68.9	20.1	2.2	340.0	630.0	89.6
124	1	175	89	68.4	20.6	2.2	340.0	630.0	89.6
	2	175	89	68.4	20.6	2.2	340.0	630.0	89.6
	3	180	89	66.6	22.4	1.9	320.0	630.0	89.6
	4	180	89	66.6	22.4	1.9	320.0	630.0	89.6
	5	175	89	68.9	20.1	2.2	340.0	630.0	89.6
125	1	175	89	68.3	20.7	2.2	340.0	630.0	89.6
	2	175	89	68.3	20.7	2.2	340.0	630.0	89.6
	3	180	89	66.5	22.5	1.9	320.0	630.0	89.6
	4	180	89	66.5	22.5	1.9	320.0	630.0	89.6
	5	175	89	68.8	20.2	2.2	340.0	630.0	89.6
126	1	175	89	68.6	20.4	2.2	340.0	630.0	89.6
	2	175	89	68.6	20.4	2.2	340.0	630.0	89.6
	3	180	89	66.9	22.1	1.9	320.0	630.0	89.6
	4	180	89	66.9	22.1	1.9	320.0	630.0	89.6
	5	175	89	69.0	20.0	2.2	340.0	630.0	89.6
127	1	175	89	68.8	20.2	2.2	340.0	630.0	89.6
	2	175	89	68.8	20.2	2.2	340.0	630.0	89.6
	3	180	89	67.2	21.8	1.9	320.0	630.0	89.6
	4	180	89	67.2	21.8	1.9	320.0	630.0	89.6
	5	175	89	69.2	19.8	2.2	340.0	630.0	89.6
128	1	120	85	73.1	11.9	1.4	156.0	580.0	89.8
	2	120	85	73.1	11.9	1.4	156.0	580.0	89.8
	3	120	85	74.2	10.8	1.3	136.0	580.0	89.8
	4	120	85	74.2	10.8	1.3	136.0	580.0	89.8
	5	110	85	72.8	12.25	2.5	245.0	580.0	89.8

Five Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
129	1	120	85	72.3	12.7	1.8	194.0	580.0	90.0
	2	120	85	72.3	12.7	1.8	194.0	580.0	90.0
	3	120	85	70.6	14.4	1.6	175.0	580.0	90.0
	4	120	85	70.6	14.4	1.6	175.0	580.0	90.0
	5	110	85	72.4	12.6	2.5	249.0	580.0	90.0
130	1	120	85	73.9	11.1	1.4	156.0	580.0	90.0
	2	120	85	73.9	11.1	1.4	156.0	580.0	90.0
	3	120	85	75.0	10.0	1.3	136.0	580.0	90.0
	4	120	85	75.0	10.0	1.3	136.0	580.0	90.0
	5	110	85	73.2	11.8	2.5	249.0	580.0	90.0
131	1	120	85	74.4	10.6	1.4	156.0	580.0	90.0
	2	120	85	74.4	10.6	1.4	156.0	580.0	90.0
	3	120	85	75.5	9.5	1.3	136.0	580.0	90.0
	4	120	85	75.5	9.5	1.3	136.0	580.0	90.0
	5	110	85	73.5	11.5	2.5	249.0	580.0	90.0
132	1	120	85	74.9	10.1	1.4	156.0	580.0	90.0
	2	120	85	74.9	10.1	1.4	156.0	580.0	90.0
	3	120	85	76.0	9.0	1.3	136.0	580.0	90.0
	4	120	85	76.0	9.0	1.3	136.0	580.0	90.0
	5	110	85	73.9	11.1	2.5	249.0	580.0	90.0
133	1	125	85	70.4	14.6	1.8	156.0	580.0	90.0
	2	125	85	70.4	14.6	1.8	156.0	580.0	90.0
	3	125	85	70.4	14.6	1.6	136.0	580.0	90.0
	4	125	85	70.4	14.6	1.6	136.0	580.0	90.0
	5	120	85	70.4	14.6	2.5	249.0	580.0	90.0
134	1	125	85	71.0	14.0	1.8	202.0	580.0	90.0
	2	125	85	71.0	14.0	1.8	202.0	580.0	90.0
	3	125	85	71.0	14.0	1.6	182.0	580.0	90.0
	4	125	85	71.0	14.0	1.6	182.0	580.0	90.0
	5	120	85	71.0	14.0	2.5	283.0	580.0	90.0
135	1	125	85	71.4	13.6	1.8	202.0	580.0	88.0
	2	125	85	71.4	13.6	1.8	202.0	580.0	88.0
	3	125	85	71.4	13.6	1.6	182.0	580.0	88.0
	4	125	85	71.4	13.6	1.6	182.0	580.0	88.0
	5	120	85	71.4	13.6	2.5	283.0	580.0	88.0

Five Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
136	1	125	85	71.9	13.1	1.8	202.0	580.0	88
	2	125	85	71.9	13.1	1.8	202.0	580.0	88
	3	125	85	71.9	13.1	1.6	182.0	580.0	88
	4	125	85	71.9	13.1	1.6	182.0	580.0	88
	5	120	85	71.9	13.1	2.5	283.0	580.0	88
137	1	125	85	72.3	12.7	1.8	202.0	580.0	88
	2	125	85	72.3	12.7	1.8	202.0	580.0	88
	3	125	85	72.3	12.7	1.6	182.0	580.0	88
	4	125	85	72.3	12.7	1.6	182.0	580.0	88
	5	120	85	72.3	12.7	2.5	283.0	580.0	88
138	1	125	85	72.6	12.4	1.8	202.0	580.0	88
	2	125	85	72.6	12.4	1.8	202.0	580.0	88
	3	125	85	72.6	12.4	1.6	182.0	580.0	88
	4	125	85	72.6	12.4	1.6	182.0	580.0	88
	5	120	85	72.6	12.4	2.5	283.0	580.0	88
139	1	125	85	72.9	12.1	1.9	202.0	580.0	88
	2	125	85	72.9	12.1	1.9	202.0	580.0	88
	3	125	85	72.9	12.1	1.8	182.0	580.0	88
	4	125	85	72.9	12.1	1.8	182.0	580.0	88
	5	120	85	72.9	12.1	2.5	283.0	580.0	88
140	1	145	84	66.2	17.8	1.9	158.0	580.0	88
	2	145	84	66.2	17.8	1.9	158.0	580.0	88
	3	145	84	66.3	17.6	1.8	235.0	580.0	88
	4	145	84	66.3	17.6	1.8	235.0	580.0	88
	5	145	84	66.5	17.5	2.5	328.0	580.0	88
141	1	145	84	66.0	18.0	1.9	158.0	580.0	87
	2	145	84	66.0	18.0	1.9	158.0	580.0	87
	3	145	84	67.2	16.8	1.8	235.0	580.0	87
	4	145	84	67.2	16.8	1.8	235.0	580.0	87
	5	145	84	66.2	17.8	2.5	328.0	580.0	87
142	1	145	84	66.6	18.3	1.9	158.0	580.0	87
	2	145	84	66.6	18.3	1.9	158.0	580.0	87
	3	145	84	66.9	17.1	1.8	235.0	580.0	87
	4	145	84	66.9	17.1	1.8	235.0	580.0	87
	5	145	84	66.0	18.0	2.5	328.0	580.0	87

Five Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W	'Δt'	Δt_{o-i}	q'	q'''	t_R
143	1	145	84	65.4	18.6	2.6	282.0	580.0	87
	2	145	84	65.4	18.6	2.6	282.0	580.0	87
	3	145	84	66.2	17.8	1.9	258.0	580.0	87
	4	145	84	66.2	17.8	1.9	258.0	580.0	87
	5	145	84	65.7	18.3	2.7	351.0	580.0	87
144	1	145	84	65.1	18.9	2.6	282.0	580.0	87
	2	145	84	65.1	18.9	2.6	282.0	580.0	87
	3	145	84	65.8	18.2	1.9	258.0	580.0	87
	4	145	84	65.8	18.2	1.9	258.0	580.0	87
	5	145	84	65.5	18.5	2.7	351.0	580.0	87
145	1	155	84	67.8	16.2	1.8	252.0	580.0	87
	2	155	84	67.8	16.2	1.8	252.0	580.0	87
	3	155	84	66.5	15.5	1.4	201.0	580.0	87
	4	155	84	66.5	15.5	1.4	201.0	580.0	87
	5	155	84	66.0	16.0	2.3	326.0	580.0	87
146	1	155	84	65.9	16.1	1.8	252.0	580.0	87
	2	155	84	65.9	16.1	1.8	252.0	580.0	87
	3	155	84	66.6	15.4	1.4	201.0	580.0	87
	4	155	84	66.6	15.4	1.4	201.0	580.0	87
	5	155	84	66.1	15.9	2.3	326.0	580.0	87
147	1	155	84	66.0	16.0	1.8	252.0	580.0	87
	2	155	84	66.0	16.0	1.8	252.0	580.0	87
	3	155	84	66.7	15.3	1.4	201.0	580.0	87
	4	155	84	66.7	15.3	1.4	201.0	580.0	87
	5	155	84	66.2	15.8	2.3	326.0	580.0	87
148	1	155	84	66.1	15.9	1.8	252.0	580.0	87
	2	155	84	66.1	15.9	1.8	252.0	580.0	87
	3	155	84	66.8	15.2	1.4	201.0	580.0	87
	4	155	84	66.8	15.2	1.4	201.0	580.0	87
	5	155	84	66.3	15.7	2.3	326.0	580.0	87
149	1	155	84	66.2	15.8	1.8	252.0	580.0	87
	2	155	84	66.2	15.8	1.8	252.0	580.0	87
	3	155	84	66.9	15.1	1.4	201.0	580.0	87
	4	155	84	66.9	15.1	1.4	201.0	580.0	87
	5	155	84	66.9	15.1	1.4	201.0	580.0	87

Five Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
150	1	160	83.0	65.0	18.0	1.9	252.0	580.0	85
	2	160	83.0	65.0	18.0	1.9	252.0	580.0	85
	3	160	83.0	65.3	17.6	1.8	201.0	580.0	85
	4	160	83.0	65.3	17.6	1.8	201.0	580.0	85
	5	160	83.0	65.0	18.0	2.3	326.0	580.0	85
151	1	160	83.0	64.8	18.2	1.9	284.0	580.0	85
	2	160	83.0	64.8	18.2	1.9	284.0	580.0	85
	3	160	83.0	65.1	17.9	1.8	259.0	580.0	85
	4	160	83.0	65.1	17.9	1.8	259.0	580.0	85
	5	160	83.0	64.8	18.2	2.3	337.0	580.0	85
152	1	160	83.0	64.6	18.4	1.9	284.0	580.0	85
	2	160	83.0	64.6	18.4	1.9	284.0	580.0	85
	3	160	83.0	64.9	18.1	1.8	259.0	580.0	85
	4	160	83.0	64.9	18.1	1.8	259.0	580.0	85
	5	160	83.0	64.6	18.4	2.3	337.0	580.0	85
153	1	160	83.0	64.7	18.3	1.9	284.0	580.0	85
	2	160	83.0	64.7	18.3	1.9	284.0	580.0	85
	3	160	83.0	65.0	18.0	1.8	259.0	580.0	85
	4	160	83.0	65.0	18.0	1.8	259.0	580.0	85
	5	160	83.0	64.7	18.3	2.3	337.0	580.0	85
154	1	160	83.0	64.9	18.1	1.9	284.0	580.0	85
	2	160	83.0	64.9	18.1	1.9	284.0	580.0	85
	3	160	83.0	65.2	17.8	1.8	259.0	580.0	85
	4	160	83.0	65.2	17.8	1.8	259.0	580.0	85
	5	160	83.0	64.9	18.1	2.3	337.0	580.0	85
155	1	160	83.0	65.1	17.9	1.9	284.0	580.0	85
	2	160	83.0	65.1	17.9	1.9	284.0	580.0	85
	3	160	83.0	65.4	17.6	1.8	259.0	580.0	85
	4	160	83.0	65.4	17.6	1.8	259.0	580.0	85
	5	160	83.0	65.1	17.9	2.3	337.0	580.0	85
156	1	175	82.0	64.0	18.0	1.8	284.0	580.0	85
	2	175	82.0	64.0	18.0	1.8	284.0	580.0	85
	3	175	82.0	64.9	17.1	1.6	259.0	580.0	85
	4	175	82.0	64.9	17.1	1.6	259.0	580.0	85
	5	175	82.0	64.7	17.3	1.9	337.0	580.0	85

Five Horizontal Cooling Tubes (Continued)

Run No.	Tube No.	W	t_B'	t_W'	$\Delta t'$	Δt_{o-i}	q'	q'''	t_R
157	1	175	82.0	64.2	17.8	1.8	283.0	580.0	85
	2	175	82.0	64.2	17.8	1.8	283.0	580.0	85
	3	175	82.0	65.0	17.0	1.6	256.0	580.0	85
	4	175	82.0	65.0	17.0	1.6	256.0	580.0	85
	5	175	82.0	64.9	17.1	1.9	311.0	580.0	85
158	1	175	82.0	64.4	17.6	1.8	283.0	580.0	85
	2	175	82.0	64.4	17.6	1.8	283.0	580.0	85
	3	175	82.0	65.2	16.8	1.6	256.0	580.0	85
	4	175	82.0	65.2	16.8	1.6	256.0	580.0	85
	5	175	82.0	65.0	17.0	1.9	311.0	580.0	85
159	1	175	82.0	64.5	17.5	1.8	283.0	580.0	85
	2	175	82.0	64.5	17.5	1.8	283.0	580.0	85
	3	175	82.0	65.1	16.9	1.6	256.0	580.0	85
	4	175	82.0	65.1	16.9	1.6	256.0	580.0	85
	5	175	82.0	65.0	17.0	1.9	311.0	580.0	85
160	1	175	82.0	64.5	17.5	1.8	283.0	580.0	85
	2	175	82.0	64.5	17.5	1.8	283.0	580.0	85
	3	175	82.0	65.1	16.9	1.6	256.0	580.0	85
	4	175	82.0	65.1	16.9	1.6	256.0	580.0	85
	5	175	82.0	65.0	17.0	1.9	311.0	580.0	85

II. TABULATED RESULTS

Single Horizontal Cooling Tube

Run No.	h_o'	Nu	NGr-NPr	Grashof	Log(Gr.Pr)	Log Nu
1	113.4	6.62	23,000	3380	4.361	0.821
2	83.2	4.85	58,200	9400	4.765	0.686
3	126.4	7.38	55,000	8870	4.740	0.868
4	97.2	5.67	49,200	7800	4.692	0.754
5	77.4	4.52	35,800	5520	4.554	0.655
6	67.2	3.92	23,400	3440	4.369	0.593
7	76.2	4.45	9,770	1350	4.000	0.648
8	83.4	4.87	12,500	1740	4.097	0.688
9	120.1	7.00	25,600	3770	4.408	0.844
10	84.1	4.90	31,500	4630	4.498	0.690
11	74.1	4.32	26,600	3880	4.425	0.635
12	133.2	7.78	20,000	2900	4.300	0.890
13	124.2	7.25	21,500	3070	4.333	0.860
14	110.7	6.45	24,300	3520	4.386	0.809
15	107.1	6.23	25,000	3630	4.398	0.794
16	103.5	6.05	26,600	3870	4.425	0.782
17	105.3	6.15	26,100	3800	4.416	0.789
18	100.8	5.83	27,200	3960	4.434	0.766
19	141.3	8.25	18,700	2670	4.272	0.916
20	130.5	7.60	32,900	4800	4.517	0.880
21	125.0	7.30	26,100	3800	4.416	0.863
22	118.0	6.88	34,400	5070	5.536	0.837
23	114.7	6.70	35,600	5275	4.551	0.826
24	121.5	7.10	37,300	5570	4.572	0.851
25	118.5	6.92	38,200	5700	4.582	0.840
26	114.0	6.65	39,200	5850	4.593	0.822
27	112.9	6.60	41,600	6250	4.418	0.820
28	108.9	6.37	43,000	6470	4.634	0.804

Four Horizontal Cooling Tubes

Run No.	h_o'	Nu	NGr-NPr	Grashof	Log(Gr.Pr)	Log Nu
29	144.0	8.40	30,800	4880	4.488	0.924
30	128.2	7.50	32,200	4850	4.508	0.875
31	120.1	7.00	32,000	4820	4.505	0.845
32	167.1	9.90	30,300	4520	4.481	0.988
33	125.0	7.30	43.500	6690	4.638	0.863
34	120.6	7.03	45,800	7070	4.661	0.847
35	167.0	9.73	42,700	6520	4.630	0.988
36	143.0	8.33	42,300	6470	4.626	0.920
37	134.0	7.82	22,100	3250	4.344	0.894
38	108.0	6.30	22,500	3310	4.352	0.800
39	123.3	7.20	23,100	3400	4.364	0.857
40	136.2	7.93	23,500	3460	4.371	0.900
41	137.7	8.02	23,200	3410	4.366	0.904
42	137.0	7.98	22,300	3280	4.348	0.902
43	145.0	8.45	30,000	4480	4.477	0.926
44	126.0	7.35	25,200	3790	4.401	0.864
45	131.4	7.67	27,400	4120	4.438	0.884
46	136.0	7.93	31,000	4630	4.491	0.900
47	140.0	8.17	27,500	4100	4.440	0.912
48	150.0	8.75	30,500	4560	4.484	0.942
49	138.0	8.06	42,000	6470	4.623	0.906
50	126.0	7.35	37,500	5680	4.574	0.866
51	110.0	6.42	36,200	5480	4.558	0.807
52	141.0	8.23	35,700	5420	4.553	0.915
53	131.0	7.63	35,900	5430	4.555	0.882
54	121.0	7.06	39,300	6030	4.594	0.848
55	128.0	7.47	38,400	5920	4.584	0.873
56	113.4	6.62	38,400	5920	4.584	0.820
57	119.0	6.93	39,200	6025	4.593	0.840
58	118.0	6.88	38,500	5930	4.585	0.838
59	120.6	7.02	38,700	5950	4.587	0.846
60	120.6	7.02	38,700	5950	4.587	0.846
61	147.6	8.60	19,100	2790	4.281	0.934
62	117.0	6.83	16,700	2420	4.222	0.835
63	129.0	7.52	16,000	2320	4.204	0.876
64	129.0	7.52	15,600	2290	4.193	0.876
65	121.5	7.09	15,600	2270	4.193	0.850
66	120.0	7.00	15,000	2140	4.176	0.845
67	118.5	6.93	12,150	1730	4.085	0.841
68	117.5	6.86	13,650	1950	4.135	0.836
69	116.5	6.77	14,300	2040	4.155	0.830
70	127.0	7.42	14,150	2020	4.150	0.870
71	125.5	7.33	12,600	1800	4.100	0.865

Four Horizontal Cooling Tubes (Continued)

Run No.	h_o'	Nu	NGr-NPr	Grashof	Log(Gr.Pr)	Log Nu
72	113.0	6.61	10,500	1480	4.022	0.820
73	119.5	6.97	11,550	1630	4.063	0.843
74	114.5	6.68	11,700	1660	4.068	0.825
75	124.0	7.25	11,800	1680	4.072	0.860
76	110.0	6.42	11,800	1680	4.072	0.807
77	104.5	6.10	11,600	1635	4.065	0.785
78	108.5	6.33	10,300	1450	4.013	0.801
79	111.0	6.47	9,700	1360	3.986	0.810
80	113.5	6.63	8,900	1235	3.949	0.822
81	116.0	6.77	8,900	1235	3,949	0.830
82	103.0	6.03	8,900	1235	3.949	0.780
83	127.0	7.41	27,600	4180	4.441	0.870
84	130.0	7.58	29,500	4470	4.470	0.880
85	123.0	7.17	29,600	4450	4.471	0.855
86	127.5	7.43	31,200	4700	4.494	0.871
87	128.5	7.50	30,900	4650	4.490	0.875
88	130.0	7.60	31,400	4720	4.497	0.881
89	121.5	7.08	25,300	3720	4.403	0.850
90	116.0	6.77	25,500	3750	4.406	0.830
91	124.0	7.25	27,100	3990	4.433	0.860
92	120.0	7.00	27,500	4030	4.439	0.845
93	122.0	7.10	27,500	4030	4.439	0.851
94	126.0	7.33	27,500	4030	4.439	0.865
95	130.0	7.58	19,800	2890	4.296	0.880
96	128.0	7.45	19,800	2890	4.296	0.872
97	121.0	7.08	19,800	2890	4.296	0.850
98	129.5	7.57	28,800	4230	4.459	0.879
99	126.5	7.38	28,800	4230	4.459	0.868
100	124.0	7.25	28,800	4230	4.459	0.860

Five Horizontal Cooling Tubes

Run No.	h_c'	Nu	NGr-NPr	Grashof	Log(Gr.Pr)	Log Nu
101	153.0	8.92	26,300	3980	4.420	0.950
102	156.0	9.12	28,400	4330	4.453	0.960
103	158.0	9.23	29,700	4630	4.473	0.965
104	155.0	9.02	29,800	4660	4.474	0.955
105	149.0	8.70	30,000	4680	4.477	0.940
106	142.5	8.32	30,500	4770	4.484	0.920
107	144.5	8.42	30,500	4770	4.484	0.925
108	143.0	8.36	25,300	3830	4.403	0.922
109	146.0	8.52	25,700	3960	4.410	0.930
110	139.0	8.12	26,000	4000	4.415	0.910
111	142.0	8.29	26,400	4070	4.421	0.918
112	151.0	8.82	27,000	4160	4.431	0.945
113	145.0	8.43	27,400	4220	4.438	0.926
114	140.0	8.20	27,400	4220	4.438	0.914
115	165.0	9.66	38,800	6320	4.589	0.985
116	158.0	9.22	38,100	6250	4.580	0.965
117	153.5	8.95	37,900	6270	4.579	0.952
118	160.0	9.33	37,700	6280	4.576	0.970
119	157.5	9.17	37,800	6300	4.578	0.962
120	163.0	9.53	37,600	6270	4.575	0.979
121	165.0	9.66	47,500	7800	4.676	0.985
122	164.0	9.55	47,100	7720	4.672	0.980
123	162.0	9.46	46,700	7650	4.669	0.976
124	169.5	9.89	47,000	7830	4.672	0.995
125	171.0	10.00	46,900	7820	4.672	1.000
126	158.0	9.22	46,500	7880	4.668	0.965
127	162.0	9.44	46,300	7850	4.666	0.975
128	144.0	8.42	20,000	2970	4.301	0.925
129	142.0	8.29	22,700	3370	4.356	0.918
130	141.0	8.20	19,400	2820	4.288	0.914
131	142.5	8.32	19,000	2770	4.279	0.920
132	144.0	8.40	18,500	2700	4.267	0.924
133	139.5	8.13	25,600	3820	4.408	0.910
134	136.0	7.95	24,300	3630	4.385	0.900
135	142.0	8.30	24,000	3560	4.380	0.919
136	145.0	8.46	23,700	3570	4.375	0.927
137	149.0	8.70	23,300	3430	4.367	0.939
138	146.0	8.51	22,900	3370	4.360	0.930
139	146.0	8.50	22,500	3310	4.352	0.929
140	153.0	8.93	33,500	5230	4.525	0.951
141	156.5	9.12	34,500	5430	4.538	0.960
142	150.0	8.73	36,400	5770	4.561	0.941
143	158.5	9.22	39,100	6210	4.592	0.965

Five Horizontal Cooling Tubes (Continued)

Run No.	h_o'	Nu	NGr-NPr	Grashof	Log(Gr.Pr)	Log Nu
144	163.5	9.55	40,500	6380	4.608	0.980
145	149.5	8.71	27,400	4070	4.438	0.940
146	146.0	8.51	26,800	3970	4.428	0.930
147	137.0	7.97	26,300	3900	4.420	0.901
148	138.0	8.07	25,900	3840	4.414	0.907
149	152.5	8.90	25,500	3780	4.406	0.950
150	154.0	9.00	32,400	4920	4.510	0.954
151	142.5	8.32	30,000	4420	4.477	0.920
152	139.5	8.12	28,000	4120	4.447	0.910
153	135.0	7.86	26,000	3800	4.415	0.895
154	141.5	8.25	23,500	3410	4.371	0.916
155	137.5	8.02	20,000	2900	4.301	0.904
156	135.0	7.87	26,000	3770	4.415	0.896
157	142.5	8.32	21,000	3000	4.322	0.920
158	141.0	8.22	18,000	2570	4.256	0.915
159	136.5	7.95	15,000	2140	4.176	0.900
160	133.0	7.77	15,000	2140	4.176	0.890

APPENDIX C

ESTIMATION OF ERRORS IN TUBE-WALL TEMPERATURE MEASUREMENTS

Calculations for the average tube wall temperature and for the thermocouple conduction error involved in tube wall measurements are contained here. As a check on the direct measurement, an attempt was first made to calculate the average tube wall temperature using the following equations and data:

$$q = \frac{(t_{W_{Ave}} - t_{C_{Ave}})}{R} = W\ Cp\ \Delta t_{oi}$$

where

q = heat transferred through the tube wall, Btu/hr

$t_{W_{Ave}}$ = average tube wall temperature, ^{o}F

$t_{C_{Ave}}$ = average coolant temperature = $63.8^{o}F$

W = coolant mass flow rate = 175 lb/hr

Δt_{oi} = coolant temperature increase = $1.44^{o}F$

Cp = Heat capacity of coolant = 1 Btu/lb x ^{o}F

R = resistance of stainless steel tube wall

$$R = \frac{\ln \frac{D_2}{D_1}}{2\pi k_{ss} L}$$

where

D_2 = outside tube diameter = 0.0208 ft

D_1 = inside tube diameter = 0.0156 ft

k_{ss} = stainless steel thermal conductivity = 9.4 Btu/hr x ft x ^{o}F

L = length of tube = 22.0" or 1.835 ft

Solving for R,

$$R = \frac{\ln \frac{0.0208}{0.0156}}{2\pi(9.4)(1.835)} = \underline{\underline{0.00267}}$$

Solving for q,

$$q = W\ Cp\,\Delta t_{oi} = 175\ (1)(1.44) = 252 \quad \text{Btu/hr}$$

Applying correction for conduction through insulation:

$$q' = 252 - 19 = 233 \quad \text{Btu/hr}$$

Solving for $t_{W_{Ave}}$:

$$t_{W_{Ave}} = t_{C_{Ave}} + q'\ (R)$$

$$t_{W_{Ave}} = 63.8 + 233\ (0.00267)$$

$$t_{W_{Ave}} = 64.4^{\circ}F$$

The tube wall temperature as read from the thermocouple was 64.8°F. The calculations therefore seemed to roughly check the thermocouple reading.

Now, to estimate the thermocouple conduction error, use is made of the approximate equations of Schneider (17) which he derived for the same situation as encountered in this work. He givesthe following equation as a close approximation for calculating thermocouple conduction errors:

$$\frac{t_o - \xi}{t_B - \xi} = \frac{1}{1 + 2\pi \frac{K\gamma}{K_1 t} \cdot \epsilon r_s\ K_1\ (\epsilon r_s) / K_o\ (\epsilon r_s)}$$

where

t_o = temperature recorded by thermocouple

ξ = wall temperature in absence of thermocouple or temperature at an infinite distance from TC junction

K_1^t = internal conductance of thermocouple leads

K = thermal conductivity of tube wall

γ = tube wall thickness

$\left.\begin{matrix} K_1 \\ K_0 \end{matrix}\right\}$ modified Bessel functions of the second kind

t_B = temperature of bulk fluid

$r_s = \sqrt{2}\ r_\tau$ where r_τ = TC wire radius

$\epsilon = \sqrt{\dfrac{h_1 h_2}{K\gamma}}$

where

h_1 = outside film coefficient (bulk-side)

h_2 = inside film coefficient (coolant-side)

The internal thermocouple conductance, K_1^t, for the two thermocouple leads in parallel can be expressed thus:

$$K_1^t = (H_T P_T K_T A_T)^{\frac{1}{2}} + (H_{T'} P_{T'} K_{T'} A_{T'})^{\frac{1}{2}}$$

where

H_T = average unit surface conductance for flow over Chromel wire

$H_{T'}$ = average unit surface conductance for flow over Alumel wire

P_T = perimeter of Chromel wire $= 2\pi r_T$

$P_{T'}$ = perimeter of Alumel wire $= 2\pi r_{T'}$

A_T = cross-sectional area of Chromel wire $= \pi r_T^2$

$A_{T'}$ = cross-sectional area of Alumel wire $= \pi r_{T'}^2$

Since $H_T = H_{T'}$, $A_T = A_{T'}$ and $P_T = P_{T'}$, expression for internal thermocouple conductance reduces to:

$$K_1^{t} = \pi W r_T \quad (2H_T r_T)^{\frac{1}{2}} \text{ -------------- } 2$$

where

$W = K_T^{\frac{1}{2}} + K_{T'}^{\frac{1}{2}}$

$K_T =$ thermal conductivity of chromel wire

$K_{T'} =$ thermal conductivity of Alumel wire

Applying equation 2 into equation 1:

$$\frac{t_o - \xi}{t_B - \xi} = \frac{1}{1 + K_1 \nu(\epsilon r_s)/K_o(\epsilon r_s)} \text{ ------ } 3$$

where

$$\nu = \frac{2}{W} \left[K_{ss} \gamma (h_1 + h_2)/H_T r_T \right]^{\frac{1}{2}}$$

$$\left.\begin{matrix} K_1 \\ K_o \end{matrix}\right\}$$ modified Bessel functions of the 2nd kind

Schneider plots $\dfrac{t_o - \xi}{t_B - \xi}$ versus $\dfrac{K_{ss}\gamma}{K_1^{t}}$ for parametric values of ϵr_s from 1.0 to 0.001.

Now, to calculate the thermocouple conduction error, the unit surface conductance over each insulated lead, h_T, is assumed to be 180 Btu/hr ft^2 °F.

The average unit surface conductance, H_T is:

$$H_T = \frac{1}{\frac{1}{h_T} + \frac{i}{K_i}}$$

$\gamma_i =$ insulation thickness $= \frac{0.0147}{12}$ ft

$K_i =$ thermal conductivity of insulation

$K_1 = 0.025$ Btu/hr ft ^oF

$$H_T = \frac{1}{\frac{1}{180} + \frac{0.0147}{0.025(12)}} = 18.3$$

Solving for $K_1{}^t$ in equation 2 when:

$W = K_T^{\frac{1}{2}} + K_{T'}^{\frac{1}{2}}$

$W = (10)^{\frac{1}{2}} + (27.9)^{\frac{1}{2}} = 8.44$

$r_T = 0.0025"$ or 0.000208 ft

$K_T = 10$ Btu/hr ft ^oF

$K_{T'} = 27.9$ Btu/hr ft ^oF

$K_1{}^t = \pi W r_T (2H_T r_T)^{\frac{1}{2}}$

$K_1{}^t = \pi (8.44)(0.000208)\left[2(18.3)(0.000208)\right]^{\frac{1}{2}}$

$K_1{}^t = 4.82 \times 10^{-4}$

To calculate ϵ, assume $h_1 = 100$ Btu/hr ft^2 ^oF

$h_2 = 1000$ Btu/hr ft^2 ^oF

Solving for ϵ:

$$\epsilon = \sqrt{\frac{h_1 + h_2}{k_{ss}}}$$

$K_{ss} = 9.4$ Btu/hr ft ^oF

$\gamma = 0.03125"$

$$\epsilon = \sqrt{\frac{1100\ (12)}{9.4(0.03125)}} = \underline{212}$$

$r_s = \sqrt{2}\quad r_T = 1.414\ (0.000208) = \underline{2.94 \times 10^{-4}}$ ft

Value of the parameter ϵr_s is:

$\epsilon r_s = 212\ (2.94 \times 10^{-4}) = \underline{0.0623}$

Value of $\frac{K_{ss}\gamma}{K_1 t}$ is:

$$\frac{K_{ss}\gamma}{K_1 t} = \frac{9.4(0.03125)}{4.82 \times 10^{-4}\ (12)} = 51$$

Using Schneider's plot (Figure g-5, p. 178:

when

$$\epsilon r_s = 0.0623$$

$$\frac{K_{ss}\gamma}{K_1 t} = 51$$

by extrapolation, $\frac{t_o - \xi}{t_B - \xi} = 0.010$

Since $t_o = 64.8\,^{o}F$ and $t_B = 82.7\,^{o}F$,

$$\xi = t_o - 0.010\ (t_B - \xi)$$

or

$$\xi = 64.8 - 0.010\ (82.7 - \xi) \text{ --------------- } 4$$

The true wall temperature, ξ, can be solved by trial and error.

Assume: $\xi = 64.7\,^{o}F$

Substituting values in equation (4):

$$\xi = 64.8 - 0.010\ (82.7 - 64.7) = 64.8 - 0.18$$

$$\xi = 64.62\,^{o}F \text{ (close to assumed value)}$$

The thermocouple conduction error in this case was, therefore, 0.18°F. The measured tube-wall temperature data were appropriately corrected to take into account this conduction error.

APPENDIX D

I. Estimation of Errors in a Function of Several Variables

The random error in the value of the heat transfer coefficient obtained was estimated using the following development:

$$Q = W\ Cp\ \Delta t_{o-i} = h_o\ A_o\ \Delta t_{ov} \qquad (1)$$

where

W = coolant flow rate

Δt_{o-i} = coolant temperature increase

Δt_{ov} = bulk to tube wall temperature differential

$Cp = 1.0$ Btu/lb x ^{o}F

A_o = outside heat transfer area 0.101 ft^2

Solving for h_o in equation 1:

$$h_o = \frac{WCp\ \Delta t_{o-i}}{A_o\ \Delta t_{ov}} = \frac{W\ \Delta t_{o-i}}{0.101\ \Delta t_{ov}}$$

By partial differentiation, we get

$$\frac{\partial h}{\partial W} = \frac{\Delta t_{o-i}}{0.101(\Delta t_{ov})}; \quad \frac{\partial h}{\partial(\Delta t_{o-i})} = \frac{W}{0.101\ (\Delta t_{ov})}; \quad \frac{\partial h}{\partial(\Delta tov)} = - \frac{W\ (\Delta t_{oi})}{0.101\ (\Delta tov)^2}$$

or

$$\Delta\ h = \frac{\Delta t_{o-i}\ (\Delta_e W)}{0.101(\Delta tov)} + \frac{W\ (\Delta_e t_{o-i})}{0.101(\Delta tov)} - \frac{W(\Delta t_{o-i})\ (\Delta_e tov)}{0.101\ (\Delta tov)^2}$$

where Δh = error in the value of heat transfer coefficient

$\Delta_e t_{ov}$ = estimated error in the bulk to tube-wall temperature differential, °F

$\Delta_e t_{o-i}$ = estimated error in the coolant temperature increase, °F

$\Delta_e W$ = estimated error in the coolant flow rate

From previous discussion, the estimated error in the coolant flow rate was 2 per cent. Using the same data as in Sample calculations:

$W = 175$ lb/hr

$\Delta_e W = 0.02\ (175) = \pm 3.5$ lb/hr

The error in the bulk to tube-wall temperature was estimated to be about 0.50°F after taking into account the accuracy of the thermocouples and the uncertainties in the individual bulk and tube-wall temperatures.

The estimated error in the coolant temperature differential was 0.050°F after considering errors in potentiometric readings and the accuracy of the thermocouples.

The estimated errors are summarized thus:

$\Delta_e W = \pm 3.5$ lb/hr

$\Delta_e t_{ov} = \pm 0.50°$F

$\Delta_e t_{o-i} = \pm 0.050°$F

To obtain the maximum value of the estimated random error in the heat transfer coefficient, Δh is evaluated taking the signs of $\Delta_e W$ and $\Delta_e t_{o-i}$ as positive and the sign of $\Delta_e t_{ov}$ as negative.

Solving for Δh:

$$\Delta h = \frac{1.44(3.5)}{0.0953\ (18)} + \frac{175\ (0.050)}{0.0953(18)} + \frac{175(1.44)(0.50)}{0.0953\ (18)^2}$$

$$\Delta h = 2.94 + 5.10 + 4.07 = 12.11$$

The approximate value of h is found to be:

$$h = \frac{W\ Cp\ \Delta t_{o-i}}{A\ (\Delta t_{ov})} = \frac{175(1)(1.44)}{0.0953\ (18)} = 147\ \text{Btu/hr x ft}^2\ \text{x}\ ^{\circ}\text{F}$$

$$\text{Maximum Random Error} = \pm \frac{12.11}{147} \times 100 = \pm \underline{\underline{8.23\%}}$$

II. Calculation of Heat Input Through Insulation

The heat input through the insulated part of the tubes can be calculated using the equation:

$$\Delta q = \frac{\Sigma \Delta t}{\Sigma R} = \frac{t_{soln} - t_{coolant}}{R_1 + R_2 + R_3}$$

where

R_1 = resistance due to rubber insulation

$$R_1 = \frac{\ln \frac{D_1}{D_2}}{2 \pi K_R\ L}$$

R_2 = resistance due to stainless steel tube wall

$$R_2 = \frac{\ln \frac{D_2}{D_3}}{2 \pi K_{ss}\ L}$$

R_3 = resistance due to outside film coefficient

$$R_3 = \frac{1}{h_o\ A_o}$$

Using the following data:

$D_1 = 0.625"$

$D_2 = 0.25"$

$D_3 = 0.1875$

$K_{ss} = 9.4$

$K_R = 0.10$

$L = 0.667$ ft

$t_{coolant} = 66.0^\circ F$

$t_{soln} = 83.0^\circ F$

$h_o = 100$ Btu/hr ft^2 $^\circ F$

$A_o = 0.109\ ft^2$

Solving for Δq,

$$\Delta q = \frac{(83.0 - 66.0)}{\frac{\ln \frac{0.625}{0.25}}{2\pi(0.1(0.667)} + \frac{\ln \frac{0.250}{0.1875}}{2\pi(9.4)(0.667)} + \frac{1}{100\ (0.109)}}$$

$$\Delta q = \frac{17}{0.792 + 0.00736 + 0.0917} = \frac{17}{0.891}$$

$\Delta q = 19$ Btu/hr

The heat input through the insulation was, therefore, 19 Btu/hr, the correction applied to the measured heat rate.