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AN EXPERIMENTAL INVESTIGATION
OF HEAT TRANSFER TO
SUPERHEATED STEAM IN ROUND AND
RECTANGULAR CHANNELS

by

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PREFACE

This investigation was initiated to obtain data and correlations which predict convection coefficients applicable to superheated steam in conventional and nuclear steam generators. In particular, heat transfer in a thin rectangular channel was in question because of some experimental evidence which reported much lower coefficients than those found in a round channel of the same equivalent diameter.

This work was performed under the auspices of the Reactor Engineering Division of Argonne National Laboratory. The invaluable assistance of the Heat Engineering Section, in particular H. R. Niemoth, A. M. Stogsdill, and J. P. O'Grady, who assisted in the design and construction of the apparatus, and especially P. L. Zaleski, who performed many tedious calculations, is gratefully acknowledged.

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NOMENCLATURE

<u>Symbol</u>	<u>Description</u>	<u>Unit</u>
a	thickness of flat plate	in. or ft.
A	surface heat transfer area	ft^2
A_a	metal cross-sectional area normal to axis of tube	ft^2
A_f	flow area	ft^2
c_p	specific heat	$\text{B lb}^{-1}\text{F}^{-1}$
D	diameter	in. or ft.
D_e	equivalent diameter, equal to $4 A_f/P$	in. or ft.
E	potential	volts
F_a	thermal expansion factor	dimensionless
G	mass velocity	$\text{lb ft}^{-2}\text{sec}^{-1}$
h	convection coefficient	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
H	enthalpy of fluid	B lb^{-1}
ΔH	enthalpy difference	B lb^{-1}
I	current	amperes
$I_0(x)$	modified Bessel function of the first kind, zero order	
$I_1(x)$	modified Bessel function of the first kind, first order	
k	conductivity	$\text{B hr}^{-1}\text{ft}^{-1}\text{F}^{-1}$
K	orifice coefficient	dimensionless
$K_0(x)$	modified Bessel function of the second kind, zero order	
$K_1(x)$	modified Bessel function of the second kind, first order	
L	heated length of tube	in. or ft.
$\dot{m}$	mass flow rate	lb hr^{-1}
N_{Nu}	Nusselt number, equal to $h D_e/k$	dimensionless
p	pressure	psi

NOMENCLATURE (Cont'd)

<u>Symbol</u>	<u>Description</u>	<u>Unit</u>
Δp	pressure difference	psi, in. of hg, or in. of H ₂ O
P, P_i	inside perimeter	in. or ft.
N_{Pr}	Prandtl number, equal to $C_p \mu / k$	dimensionless
q_x	total heat generated from $x=0$ to x	B hr ⁻¹
q_L	heat loss	B hr ⁻¹
q_x''	local heat flux per unit area	B hr ⁻¹ ft ⁻²
q_x'''	local volumetric heat generation	B hr ⁻¹ ft ⁻³
q_0'''	volumetric heat generation at $x=0$	B hr ⁻¹ ft ⁻³
r	radius	in. or ft.
r_i	inside radius	in. or ft.
r_o	outside radius	in. or ft.
N_{Re}	Reynolds number, equal to $G D_e / \mu$	dimensionless
t	temperature	F
t_a	ambient temperature	F
t_c	temperature at thermocouple position	F
t_f	average film temperature	F
t_i	inside surface temperature	F
t_o	outside wall temperature	F
t_s	fluid temperature	F
Δt	temperature difference	F
Δt_{sup}	degrees of superheat	F
V	velocity	ft sec ⁻¹
x	axial distance from start of heated length	in. or ft.
Y	orifice compressibility factor	dimensionless

NOMENCLATURE (Cont'd)

<u>Greek Symbols</u>	<u>Description</u>	<u>Unit</u>
α	temperature coefficient of resistivity	F^{-1}
β	square root of the temperature coefficient of volumetric heat generation, equal to $\sqrt{\alpha q'_{o'}/k}$	ft^{-1}
γ	temperature increase of fluid per unit length	$F\ ft^{-1}$
θ	temperature excess, equal to $(t-t_a)$	F
ρ	electrical resistivity	microhm - cm
ρ_1	fluid density	$lb\ ft^{-3}$

Subscripts

f	refers to average film temperature
p	refers to conditions at a given power level
r	refers to radial heat flow in tubular section
t	refers to transverse heat flow in rectangular section

CHAPTER I

INTRODUCTION

Purpose of the Investigation

Within the past thirty years, a voluminous amount of experimental information concerning single phase convection heat transfer has appeared on the scientific scene. However, the major portion of this information has been derived from studies which embrace regions of temperature and pressure in which fluid properties exhibit small variation, and further, the studies, in general, do not encompass the influence of flow geometry. In particular, with the advent of proposals for superheating nuclear reactors the need for more specific insight into the mechanism of heat transfer to flowing superheated steam becomes apparent as soon as the literature is scanned. As steam is a relatively poor heat transfer medium, the surface temperature of the confining channel wall can become excessively high at practical rates of heat flux. Since total fuel volume and surface area are at a premium in a reactor core and, further, since operating fuel temperature is an important parameter in reactor performance, the precise prediction of the convection coefficient is imperative. Furthermore, since flat plate fuel elements, which are arranged to form thin rectangular channels, are desirable practically, the influence of flow channels with high aspect ratios must be carefully investigated. In this respect, there is some experimental evidence that the convection coefficient in rectangular channels of high aspect ratio is considerably lower than the coefficient in circular channels of the same equivalent diameter. As shown by the literature review, there is but one previous superheated steam experiment which encompasses an applicable range of temperature and pressure, and this research is limited to flow in an annulus to a Reynolds number 40,000.

In view of the previous considerations, the experimental intent here becomes two-fold.

1. The corroboration and extension to higher Reynolds number of the existing experimental information concerning convection heat transfer to superheated steam at intermediate pressures.

2. The influence exerted upon the convection coefficient by a geometric change from a circular channel to a rectangular channel of high aspect ratio.

Scope of the Research

Two test sections were constructed of 304 stainless steel and heated by alternating current. The tubular section had an inside diameter of 0.333 inches, an outside diameter of 0.376 inches, a heated length of 12.00 inches and was cooled by superheated steam in the following range of variables:

Steam Pressure: 300 to 1500 psia

Steam Temperature: 550 to 900°F

Steam Superheat: 73 to 142 F°

Surface Temperature: 650 to 1290°F

Heat Flux: 50,000 to 287,000 B hr⁻¹ ft⁻²

Steam Mass Velocity: 144,000 to 792,000 lb hr⁻¹ ft⁻²

Reynolds Number: 60,000 to 370,000.

The rectangular section was a channel of 1.250 inches by 0.047 inches and was heated along a 12.00 inch length. This particular aspect ratio (26.6:1) was chosen to simulate the channel proposed for the Borax V superheating reactor core.

The rectangular channel experiment encompassed the following range of variables:

Steam Pressure: 300 to 1000 psia

Steam Temperature: 470 to 850°F

Steam Superheat: 5 to 160 F°

Surface Temperature: 520 to 1125°F

Heat Flux: 23,000 to 96,000 B hr⁻¹ ft⁻²

Steam Mass Velocity: 197,000 to 553,000 lb hr⁻¹ ft⁻²

Reynolds Number: 25,000 to 100,000.

Literature Review

There are few experiments and theories which appear applicable to the specific problem of heat transfer to superheated steam. Most studies of transfer to turbulent gases have been performed at relatively low pressure and with gases which obey the perfect gas law (or some modification thereof) to a considerable extent. In many theoretical studies, the influence of the Prandtl number has been neglected altogether, since for many common gases, N_{Pr} is close to 1. Within the scope of this investigation, N_{Pr} varies from 0.50 to 1.40, so that its influence will account for a considerable variation in the final correlation.

The most common method of correlating forced convection, single-phase heat transfer data is that of the Dittus - Boelter type, incorporating three dimensionless groups:

$$N_{Nu} = C (N_{Re})^n (N_{Pr})^m \quad (1)$$

Early investigators found this equation quite satisfactory for regimes of low heat flux and nearly constant fluid properties. However, as the investigations proceeded to regions of higher power density, difficulties arose due to the attendant higher temperature differences between the heat transfer surface and the bulk or "mixed-mean" temperature of the convective medium. As experimenters and theorists have determined, the mechanism of transfer depends, to a large extent, upon the properties and velocity and temperature distributions in the thin boundary layer adjacent to the solid wall. As Jakob⁽²⁾ and others point out, Eq. 1 is applicable to all classes of single-phase fluids only if the principle of similarity holds geometrically and for fluid property variation. It is seen that similarity does not hold between gases and liquids in the case of viscosity, for example. In general, viscosity

decreases for liquids and increases for gases with increasing temperature. Geometrically, Eq. 1 suffices only for one finite characteristic dimension. For example, in the case of a round tube, D is the only variable dimension; the length of the tube is considered infinite. It is therefore reasonable to introduce one or more additional dimensionless groups to account for finite geometries and variations of properties with pressure and temperature. If Eq. 1 is written in terms of individual definitions,

$$\frac{h D_e}{k} = C \left(\frac{G D_e}{\mu} \right)^n \left(\frac{c_p \mu}{k} \right)^m \quad (2)$$

it is seen that some allowance must be made for variations in specific heat, viscosity, and conductivity with temperature. Since the heat flow is a function of the steep temperature gradient at the heat transfer surface, it has been found that quite satisfactory results are obtained by evaluating these properties at an "average" film temperature. As a practical consideration, the arithmetic mean temperature defined by

$$t_f = \frac{t_i + t_s}{2} \quad (3)$$

has been used and found to suffice for most cases of a fluid being heated. By using t_f as the "property" temperature, the influence of potential (or film temperature difference) is accounted for without the practical disadvantage of introducing absolute temperature ratios to exactly account for property variation. Mikheyev⁽³⁾ has been able to correlate large amounts of data for different fluids, both heated and cooled, by introducing the dimensionless parameter, (Pr_f/Pr_w) , which apparently accounts with good accuracy for the effects of both heat flux (potential) and direction of heat flow. For a finite tube geometry, the introduction of a parameter, $(L/D)^r$, is necessary to account for the fact that the Nusselt number decreases with increasing tube length, as shown by Jakob.⁽²⁾ Since this experiment is concerned with the heating of superheated steam for a considerable range of film temperature

differences and a consideration of the establishment of the temperature profile in the boundary layer, the following modification of Eq. 1 is expected to suffice for correlation.

$$\frac{h D_e}{k_f} = C \left(\frac{G D_e}{\mu_f} \right)^n \left(\frac{c_p \mu}{k} \right)_f^m \left(\frac{L}{D_e} \right)^r \quad (4)$$

At this point it is important to note that the dimensionless equation to be used has an effect on experimental procedure. In the case of Eq. 4, the four constants C , n , m , and r must be determined; theoretically, four experiments will suffice. Although one may introduce other ratios at the data processing stage in order to effect a better correlation, the experiment will not have been designed to investigate the effect of each additional dimensionless group. Consequently Eq. 4 allows more freedom in experimental procedure and results in simpler control problems for the variables used in the experiment.

MacAdams, Kennel, and Addoms⁽⁴⁾ have published what appears to be the only experiment performed using superheated steam as the heat transfer medium at any pressure above one atmosphere. This experiment was performed using an annulus ($D_e = 0.131$ inches) heated on one side, with corrections for radiation to the cool outer wall. The data covers a Reynolds number range from 6,000 to 40,000 and a pressure range from 115 to 3500 psia. The data was evaluated first in 1949 using the measurements of steam conductivity as determined by Timroth and Vargaftik⁽⁵⁾ (1939). The resulting correlation was

$$\frac{h D_e}{k_f} = 0.0126 \left(\frac{G D_e}{\mu_f} \right)^{0.89} \left(\frac{c_p \mu}{k} \right)_f^{1/3} \left(\frac{L}{D_e} \right)^{-0.13} \quad (5)$$

However this initial evaluation was followed shortly by a new table of conductivity values submitted by Keyes and Sandell.⁽⁶⁾ These new values differed considerably from the Russian data, and MacAdams re-evaluated the heat transfer data in the light of the new measurements. This resulted in the more satisfactory equation:

$$\frac{h D_e}{k_f} = 0.0214 \left(\frac{G D_e}{\mu_f} \right)^{0.80} \left(\frac{c_p \mu}{k} \right)_f^{1/3} \left(1 + \frac{2.3}{L/D_e} \right) \quad (6)$$

A recent extensive review of water and water vapor properties by Nowak and Grosh⁽⁷⁾ in 1959 provided evidence that the Russian conductivity data are the more reliable. They report that Vargaftik and Smirnova⁽⁸⁾ carried out new experiments (1956) to clarify the reason for the discrepancy between the above two sets of measurements (1939 and 1950). Vargaftik and Smirnova found very good agreement between their new findings and the 1939 data. On this basis, and after a study of experimental methods involved in all three investigations, Nowak and Grosh concluded that the various Russian values of thermal conductivity of steam are to be preferred. Table 1 was constructed from this reference⁽⁷⁾ and Fig. 1, which was derived from the various Russian values and the best curve through the data.

Table 1. Thermal Conductivity,¹ k, of Superheated Steam, B hr⁻¹F⁻¹ft⁻¹

Press. psia	Temp. °F							
	500	550	600	650	700	800	900	950
300	0.0258	0.0268	0.0281	0.0297	0.0316	0.0356	0.0400	0.0424
600	0.0301	0.0300	0.0361	0.0317	0.0332	0.0370	0.0410	0.0431
1000	-	0.0344	0.0339	0.0344	0.0353	0.0390	0.0428	0.0447
1500	-	-	0.0396	0.0393	0.0398	0.0421	0.0450	0.0466

¹Measurements of Vargaftik,⁽¹⁷⁾ Timroth and Vargaftik,⁽⁵⁾
Vargaftik and Smirnova,⁽⁸⁾ all reported in reference 7.

In the same publication, Nowak and Grosh recommend the water vapor dynamic viscosity values of Schugajew,⁽⁹⁾ Schugajew et al.,⁽¹⁰⁾ Sigwart,⁽¹¹⁾ and Timroth.⁽¹²⁾ The pressure dependence of viscosity is less than 6 per cent between 1 atmosphere and 1320 psia and much less disagreement between investigators is evident than for conductivity. Figure 2 is derived from reference (7) in accordance with the above recommendation and furnishes the values listed in Table 2.

FIG. 1 - THERMAL CONDUCTIVITY OF SUPERHEATED STEAM

FIG. 2 - DYNAMIC VISCOSITY OF SUPERHEATED STEAM

Table 2. Dynamic Viscosity,² μ , of Superheated Steam, $\text{lb ft}^{-1}\text{sec}^{-1}\times 10^6$

Press. psia	Temp. °F							
	500	550	600	650	700	800	900	950
300	12.6	13.2	13.8	14.5	15.2	16.7	18.3	19.1
600	12.8	13.4	14.0	14.7	15.4	16.9	18.5	19.5
1000	-	13.6	14.3	15.0	15.6	17.2	18.7	19.5
1500	-	-	14.8	15.4	16.1	17.5	19.1	19.9

²Measurements of Schugajew,⁽⁹⁾ Schugajew et al.,⁽¹⁰⁾ Sigwart,⁽¹¹⁾ and Timroth,⁽¹²⁾ all reported in reference 7.

Specific heats are values derived from quite accurate pressure-volume-temperature measurements, and the values used for this experiment are taken from Keenan and Keyes.⁽¹³⁾

Boelter, Young, and Iversen⁽¹⁴⁾ ran experiments in 1948 which show that the unit thermal conductance (or convection coefficient) in the "entrance section of a heated tube is highly dependent upon the type of flow section which precedes the heated length of a circular tube. The entrance section is defined as the length of heated tube required to establish the temperature profile in turbulent flow. Local coefficients in this entrance section are higher for a "short" ($L/D_e = 2.8$) starting section than for a "long" ($L/D_e = 11.2$) starting section; the difference lies in the stage of development of the velocity profile for turbulent flow at the start of the heated length. The starting sections used in the present study are in the "long" category since this condition more nearly approximates the inactive fuel zone portion of a reactor core fuel element. Deissler, in a later publication,⁽¹⁵⁾ shows experimentally that the establishment of a fully developed velocity profile in adiabatic turbulent flow requires 45 diameters for a sharp-edged entrance condition for Reynolds number from 46,000 to 580,000 and for a rounded entrance, a slight development still exists at 100 diameters. Furthermore, the velocity gradient decreases with increasing tube length. One therefore expects a decreasing convective effect with increasing tube

length in non-adiabatic flow. This fact is definitely borne out by the work performed by Boelter, Young, and Iversen,⁽¹⁴⁾ previously discussed. More support for these observations is obtained through the results of the present study.

Levy, Fuller, and Niemi⁽¹⁶⁾ in 1959 reported that experiments performed with turbulent water flow in a thin rectangular channel (0.1 x 2.5 inches) yielded convection coefficients approximately 40 per cent below the values obtained for round tubes having the same equivalent diameter.

CHAPTER II

EXPERIMENTAL APPARATUS AND PROCEDURE

The apparatus included four major components as follows:

1. Water Treatment Plant
2. Boiler-Superheater
3. General Loop and Instrumentation
4. Test Sections and Instrumentation

Figure 3 is a schematic diagram of the flow loop and attendant instrumentation. The water treatment plant is shown at the upper left of the diagram, the steam generator at upper right, the general loop and instrumentation on the lower right and the test section at lower left. Flow is approximately clockwise around the diagram. Figure 4 is a photographic view of the demineralizer, boiler and loop area, left, right and center of the picture, respectively.

Water Treatment Plant

Tap water was introduced to a mixed-bed ion-exchanger (Illco-Way, Model MB590). Effluent purity was maintained such that ionic solids produced a resistivity which was not allowed to fall below one megohm-cm and was usually on the order of eight to ten megohm-cm. After regeneration, total demineralizing capacity was approximately 7500 gallons. Resistivity was monitored continuously by a Solu-bridge (Model RE83-Y1) which was connected to an alarm bell, which warned of approaching bed exhaustion.

The demineralizer effluent was fed directly to a carbon steel deaerating tank which removed dissolved gasses by steam heating the water to saturation at 2 to 5 psig. A small cooler at the tank exit reduced the water temperature a few degrees so as to prevent flashing at the suction of the reciprocating boiler feed pump.

FIG. 3 - FLOW LOOP AND INSTRUMENTATION DIAGRAM

FIG. 4 - GENERAL EXPERIMENTAL AREA

Boiler-Superheater

A once-through gas-fired steam generator, built by the Bessler Corp., provided superheated steam at pressures from 300 to 2200 psig, superheat to 200°F at all pressures, and a maximum flow of 2000 lb/hr. The boiler feed pump provided a maximum of 4 gpm through the 316 and 321 stainless steel boiler tubes in the firebox. The proper combustion mixture was obtained by manual adjustment of a mechanical linkage between the inlet gas butterfly valve and the damper in the forced draft air system. Ignition was effected by a 10,000 volt alternating current spark from a 440V - 10,000V transformer. A complex pressure-temperature interlock controlled outlet steam conditions for a particular flow setting. Control was difficult, but a steady operating point could be obtained by setting the feedwater regulating valve such that exactly the proper amount of water was introduced to the coils for evaporation without allowing the pressure and temperature to oscillate due to over-control by the temperature regulator. A high pressure relief valve and a high temperature fire control interlock were integral with the boiler unit for safe operation.

General Loop and Instrumentation

The flow loop was constructed of 1-1/4% Cr - 1/2% Mo steel with the exception of three 304 stainless steel valves and a set of 304 stainless steel orifice flanges. Regulation of flow through the test loop was effected with a bypass valve immediately following the boiler stop valve. The by-pass steam and the test section steam were discharged to the atmosphere.

A 0.2342 inch orifice in a set of 0.957 inch orifice flanges was calibrated with steam at approximately 600 psig. The results of this calibration are presented in Appendix A. Four 2000 psi, 60-inch, single-leg manometers (Meriam Instrument Co., Model 30FA200) were used to measure the orifice differential pressure and various differentials at the test section. Mercury was used as the measuring fluid.

A 1500 psi Heise gage was used for measurement at the test section inlet and orifice inlet. Calibration with a dead weight tester showed a maximum deviation of 2 psi from true pressure in the entire range of the gage. A 1500 psi Mercoid pressure switch (Type DA21-3) was interlocked with the main power supply so that the test sections would not be overheated in the event of a flow failure.

The 220 volt bus was connected to three 28 ampere variacs in parallel, where manual power control was provided. A Sorgel 10 KVA transformer (220V/12.5V), with a maximum current rating of 800 amperes through the secondary, fed power directly to the test section. Test section voltage was measured with a Weston Model 433 voltmeter having a scale range of 0/10/20 volts and a full-scale accuracy of 0.25 per cent. A Weston Model 432 wattmeter (0/375/750 w scale range) was used to indicate roughly the total power applied to the test section so that heat flux variations could be made at fairly regular intervals without computing the power from voltage and current readings.

All temperature measurements were made using AWG No. 30 Chromel-Alumel thermocouples. Each pair of leads were wired to a common ice junction through an insulated switchbox. Switches used were made by Leeds and Northrup, Type 8240. The cold junction was placed in a glass tube immersed in a distilled water-ice mixture. The thermal electromotive force was read by one of two instruments, or both. The most convenient to use was a self-balancing potentiometer (Minneapolis-Honeywell, Brown Electronik, Model No. SY156X15-VH), which had a total scale range of 0-70 millivolts, a readability of one microvolt, a sensitivity of one microvolt, and an accuracy of three microvolts. An error of 5 microvolts corresponds to a temperature error of 0.2°F. In addition, a portable potentiometer (Leeds and Northrup) was provided. Some of the thermocouples experienced alternating current pickup, which affected the amplifier in the self-balancing

potentiometer. Since the portable potentiometer sees only the direct current component generated by the thermocouple (except for an occasional jump when the circuit is first closed), readings could be obtained without the use of the Brown instrument. The self-balancing potentiometer was equipped with a re-transmitting slide wire which was wired to a strip chart follower (Minneapolis-Honeywell, Brown Elektronik, Model No. 143X10). This instrument provided observation of any one time-temperature curve for determination of steady-state conditions.

Tubular Test Section

Figure 5 is a detailed sketch of the tubular test section, terminal connections, and mixing chamber assembly for measurement of inlet and outlet steam temperature and pressure. Figures 6 and 7 are photographs of the un-insulated and insulated section in place in the test loop. All parts were constructed of 304 stainless steel with the exception of the Durabla gaskets, which provided both electrical insulation and pressure seals. The heated length of the 0.33 inch (inside diameter) section was 12.00 inches and an unheated calming section of 16.3 diameters was installed. Note that the mixing chamber is a radiation shield, and its flow area provides a low velocity section for pressure measurement but not so low that an appreciable amount of heat is conducted along the axial length of the thermocouple well; i.e., the convection coefficient is high enough to yield a good temperature measurement. The outside of the tube was coated with a thin ceramic (Al_2O_3) for electrical insulation of the section from the thermocouple beads. This precaution proved quite worthwhile, as it later was found that pickup at the amplifier was observed only from those thermocouples which were in direct contact with the current carrying portions of the section. The thermocouple leads were taped half-way around the tube periphery in an isothermal plane to prevent conduction away from the bead through the lead wires.

FIG. 5 - TUBULAR TEST SECTION AND
MIXING CHAMBER ASSEMBLY.

FIG. 6 - TUBULAR TEST SECTION IN PLACE

FIG. 7 - INSULATED TUBULAR TEST SECTION

Thin Rectangular Test Section

Figure 8 is a detailed sketch of the section, terminal connections, and transition pieces. Figures 9, 10, and 11 are photographs of the disassembled parts, the uninsulated section in place, and the insulated section in place, respectively. The transition pieces permitted re-use of the aforementioned mixing chambers. The rectangular piece was fabricated of 304 stainless steel and machine welded from two "L" sections of 0.0156 inch plate. This formed an inside channel, 1.250 by 0.047 inches, and a heated length of 12.00 inches. The unheated entrance section was 13.9 equivalent diameters in length, assuming that the equivalent diameter is equal to four times the flow area divided by the wetted perimeter of the channel. The test section and the electrodes were insulated from the back-up plates by 0.020 inch Mica, which was slotted at points where the 0.020 inch thermocouple lead wires were placed for temperature measurement. This 0.020 inch space on each side of the section was compensated for by metal spacers between the back-up plates outside the width of the section. Thus when the inside channel was pressurized, no bulges or dents were created due to non-uniform pressure on the outside surface. Again the lead wires were placed along approximate isotherms to minimize temperature errors from heat conduction.

Experimental Procedure

Before the steam generator was started, the entire operating system was flushed with demineralized water from the deaerating tank, and manometer and pressure lines were bled to purge the system of impurities and air. The boiler was fired after purging and brought up to the proper pressure and flow by regulation of the fire control mechanism, the by-pass valve and the test section outlet valve. Setting a specific amount of superheat at the test section inlet was difficult, since the best operating range for the generator was approximately 100 F° above saturation and the heat loss between the boiler and the

FIG. 8 - RECTANGULAR TEST SECTION ASSEMBLY

FIG. 9 - DISASSEMBLED RECTANGULAR TEST SECTION

FIG. 10 - RECTANGULAR TEST SECTION IN PLACE

FIG. 11 - INSULATED RECTANGULAR TEST SECTION

test section was highly variable from run to run. The amount of superheat at the test section inlet was in the range of 50 to 100 F° for most runs. Once the generator was settled to a steady output of pressure, temperature, and flow, the system and test section could be brought to steady temperature in 45 to 90 minutes, dependent upon flow rate.

When steady state was attained at zero power, all thermocouples were read for calibration to determine the corrections applicable to the temperatures at the outside wall of the section. The method of applying these zero power corrections is shown in Appendix B.

Following the zero power readings, power was applied to the section with the Variac control previously described. For each power setting, approximately 30 to 60 minutes were required for steady state conditions to prevail. Occasionally the main power supply would fluctuate or the steam generator would begin a slight oscillation, thus disturbing the steady point attained. In this case, system adjustments were made until steady state again prevailed and readings were taken. The maximum power setting for any particular flow rate was limited by either the capacity of the main transformer or a test section surface temperature of 1250°F, whichever condition was first attained.

For each run, the following quantities were recorded:

1. Thermal emf from fifteen thermocouples on the outside wall of the heated length - mv.
2. Thermal emf from four thermocouples on the electrodes near the heated section and two thermocouples at the orifice - mv.
3. Thermal emf from four thermocouples measuring inlet and outlet steam temperature - mv.
4. Voltage, current, and power from three meters - volts, amperes, watts.

5. Pressure differentials at the orifice and from test section inlet to outlet - inches of Hg.
6. Pressure at the test section inlet and at the orifice - psig.
7. Atmospheric pressure and temperature - in.of Hg, mv.

CHAPTER III

DATA ANALYSIS, RESULTS AND DISCUSSION

General

The determination of the local convection coefficient depends upon measurement and calculation of the quantities in the equation

$$h = \frac{q''}{t_i - t_s} \quad (7)$$

There are differences between the two geometries considered, but the general methods apply to both. The three desired quantities, heat flux, surface temperature, and fluid temperature (based on energy input to the local channel point) are somewhat interdependent so that one or two iterations are necessary to produce h . Appendix E contains tabulations of all the quantities involved in the h calculations.

Determination of Local Surface Temperature, t_i

The thermocouples for wall temperature measurement were located outside of the thin ceramic coating described in Chapter II. The zero power corrections were added to the readings as described in Appendix B to obtain the outside metal wall temperature, t_0 . Near the ends of the heated length, the axial temperature profile becomes severely depressed, due to the heavy electrode assemblies, which act as heat sinks. The general two-dimensional heat conduction problem with variable heat generation with temperature is discussed in Appendix C. The two one-dimensional cases (axial and radial) are also discussed. The axial case provides both an estimate of heat losses to the electrodes and a prediction of the axial temperature profile. The shape of this profile was borne out by experiment as shown in Fig. 12, which is a typical plot of measured outside wall temperature versus heated length. On the basis of these observations, it was decided that convection coefficients would be computed only for temperature positions

FIG. 12 - TYPICAL MEASURED AXIAL TEMPERATURE
PROFILE - TUBULAR TEST SECTION.

between $x = 2$ inches and $x = 10$ inches, the portion of the heated length in which the axial temperature profile is essentially linear. If one considers the general two-dimensional conduction equation for a hollow cylinder,

$$\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial r^2} + \frac{1}{r} \frac{\partial t}{\partial r} + \beta^2 t = - \frac{q_0'''}{k_w} \quad (8)$$

it is seen that $\frac{\partial^2 t}{\partial x^2}$ vanishes in this linear region, $2 < x < 10$. Consequently Eq. 8 reduces to the one-dimensional radial case,

$$\frac{d^2 t}{dr^2} + \frac{1}{r} \frac{dt}{dr} + \beta^2 t = - \frac{q_0'''}{k_w} \quad (9)$$

In this experiment the temperature difference through the wall is small, so that radial variation of heat generation may be neglected. Now Eq. 9 reduces to

$$\frac{d^2 t}{dr^2} + \frac{1}{r} \frac{dt}{dr} = - \frac{q_0'''}{k_w} \quad (9a)$$

Using the boundary conditions

$$t = t_0, \quad \frac{dt}{dr} = 0 \text{ at } r = r_0; \quad t = t_i \text{ at } r = r_i \quad (9b)$$

the solution for the temperature difference through the wall is

$$t_0 - t_i = \frac{q'''}{2k_w} \left[r_0^2 \ln \frac{r_0}{r_i} - \frac{r_0^2 - r_i^2}{2} \right] \quad (10)$$

For the rectangular channel, t_0 was measured at the center line of the wide face and by reasoning similar to the above, the temperature difference for the plane wall is

$$t_0 - t_i = \frac{a^2 q'''}{2k_w} \quad (11)$$

The inside surface temperature, t_i , was calculated by Eqs. 10 or 11. The values of heat generation and conductivity were taken at an approximate average wall temperature for any local axial point of measurement. This implies that the axial variation of q and k_w were

accounted for, but not the radial or transverse variation. The foregoing method calculates t_i independent of the axial heat losses to the electrodes but not the radial or transverse loss through the insulation. This effect is small relative to the transfer to the fluid as shown by the error analysis in Appendix D.

Determination of Local Heat Flux, q''

Because of the linearity discussed in the previous section the net heat flow in the axial direction at any point in the interval $2 < x < 10$ is zero. Thus all the heat generated by the current at any of these axial points flows in the radial or transverse direction. Because the external boundary is insulated, essentially all this heat is carried away by the convective medium. The power generated in a differential piece of section can be written

$$d q = I^2 \rho(x) \frac{d x}{A_a} \quad (12)$$

The radial or transverse heat flux into the fluid is

$$q'' = \frac{d q}{d A} = \frac{I^2 \rho(x) d x}{A_a P_i d x} = \frac{I^2 \rho(x)}{A_a P_i} \quad (13)$$

For any particular run, I , A_a , and P_i are constant so that Eq. 13 reduces to a linear relationship between q'' and $\rho(x)$,

$$q'' = c_1 \rho(x) \quad (14)$$

Local heat fluxes were calculated by Eq. 14; $\rho(x)$ was evaluated at the local $t(x)$, an average radial wall temperature estimated from Eq. 10. The variation of resistivity with temperature is nearly linear in the range of experimentation, and this curve is shown in Appendix E. Of course, Eq. 10 requires a preliminary estimate of q'' from Eq. 14. Since the wall drop was small compared to $t_i - t_s$, only one iteration was required to produce accurate values of both $t_0 - t_i$ and q'' . It should be noted that q'' is also calculated independent of axial heat losses to the electrodes. The heat flux can be approximated by two

other methods, each based upon an independent calculation of power input. The first is calculated by the EI product and the q_{avg}'' is found by

$$q_{avg}'' = 3.413 \frac{EI}{A} \quad (15)$$

The second is computed from the flow rate, as measured by the orifice, and from the enthalpy difference by pressure and temperature measurements in the mixing chambers. Heat flux is found by

$$q_{avg}'' = \frac{\dot{m} \Delta H}{A} \quad (16)$$

The average heat fluxes as given by Eqs. 15 and 16 were used as a check on the average heat flux computed from Eq. 13, in which a mean resistivity, ρ_m , was used as determined by a mean wall temperature, t_m . This quantity, t_m , was found by planimeter under the curve of Fig. 12. The heat balances were good; neither Eq. 15 nor Eq. 16 gave values which were more than eight per cent different from those found by Eq. 13. The heat input obtained by Eq. 16 was almost always lower than that obtained by the other two methods. This is to be expected since heat losses should appear in this comparison.

Determination of Local Fluid Temperature, t_s

The fluid temperature, t_s , is assumed to be the temperature attained by the absorption of energy from the entrance of the heated section to the local point under consideration. For purposes of computation it was assumed that this temperature varied linearly with tube length from the inlet of the heated length. The temperatures measured in the mixing chambers were used as inlet and outlet points. This assumption is justified by the error analysis outlined in Appendix D, which indicates that the error introduced is quite negligible, amounting to a maximum of one per cent of the film temperature difference, $t_i - t_s$.

Calculation of Mass Velocity, G

The mass velocity was computed from the orifice differential and the test section flow area, according to the following equation:

$$G = \frac{K Y F_a \sqrt{(\Delta p) \rho_1}}{(182.6) A_f} \frac{\text{lb}}{\text{sec-ft}^2} \quad (17)$$

The orifice flow coefficient, K, was determined by the calibration shown in Appendix A.

Correlation and Results for Tubular Channel

Because of the apparent regular variation of the convection coefficient with tube length, it was decided to compute dimensionless ratios according to Eq. 4 at L/D_e ratios of 6, 12, 18, 24, and 30, even though intermediate values were available. The local Nusselt, Prandtl, and Reynolds numbers were computed for each of these tube positions, and each incorporated the property values discussed in the introduction. The tables in Appendix E present all of the data used in these calculations. Thirty-two runs were made, all of which appeared valid according to the heat balances. A total of 157 data points were evaluated and the number of points for each pressure range is listed below.

<u>Pressure Range</u> <u>psia</u>	<u>No. of</u> <u>Data Points</u>
310-348	40
593-652	67
904-1024	30
1404-1470	20

For each L/D_e ratio, a three-variable correlation was obtained and was of the form

$$\frac{h D_e}{k_f} = C \left(\frac{G D_e}{\mu_f} \right)^n \left(\frac{c_p \mu}{k} \right)_f^m \quad (18)$$

The values of C, n, and m derived by a "least squares" fit for each L/D_e are listed in Table 3. It should be noted that the value of the

Prandtl number exponent, m , was not derived experimentally. To do so requires that one of the other variables, either Nusselt or Reynolds number, be held constant while the Prandtl number is varied. In the experiment performed, this was practically impossible. Furthermore, the Prandtl number is sufficiently close to one so that the results of a great many experiments which recommend $m = 0.333$ could be used without compunction.

Table 3. Constants for Equation 18 per L/D_e

L/D_e	C	n	m
6	0.1513	0.8363	0.3333
12	0.1348	0.8441	0.3333
18	0.1351	0.8424	0.3333
24	0.1500	0.8323	0.3333
30	0.1332	0.8414	0.3333

Equation 18 may be written

$$N_{Nu} \left(N_{Pr} \right)^{-1/3} = C \left(N_{Re} \right)^n \quad (19)$$

If the left hand side of Eq. 19 is used as an ordinate and plotted against Reynolds number for each L/D_e , as in Fig. 13, a regular variation of decreasing ordinate with increasing L/D_e is noted, although each line has a slightly different slope as noted in Table 3. For a final correlation, an average slope of 0.84 was used for "n." A "least squares" fit for the equation,

$$N_{Nu} \left(N_{Pr} \right)_f^{-1/3} \left(N_{Re} \right)^{0.84} = C \left(\frac{L}{D_e} \right)^x \quad (20)$$

resulted in the final correlation which may be written as

$$N_{Nu} \left(N_{Pr} \right)_f^{-1/3} \left(\frac{L}{D_e} \right)^{0.04} = 0.0157 \left(N_{Re} \right)^{0.84} \quad (21)$$

FIG. 13 - DATA CORRELATION FOR TUBULAR CHANNEL AT VARIOUS L/D_e .

Figure 14 is a plot of the final correlation and data points using the left hand side of Eq. 21 as the ordinate. The maximum deviation in the ordinate is 12 per cent, the average deviation is 4 per cent and 96 per cent of the points lie within 9 per cent of the correlation line.

Correlation and Results for the Rectangular Channel

The same procedure was followed for data analysis as for the tubular channel, using the centerline temperatures of the broad side of the test section for surface temperature computation. In addition, some temperature profiles across the 1.250 inch width were measured to detect any serious differences in convection properties near the corners of the heated perimeter. While some higher temperatures were noted directly at the corners, the slight increases in temperature noted at points one-sixth of the broad channel width from the edge amounted to a maximum of five per cent of the centerline coefficient. These edge temperatures were recorded only for the data taken in the 300 psi range.

Figure 15 is a typical convection coefficient vs. length plot. The peak, found in all the runs to a maximum L/D_e of approximately 60, is due to the vena contracta formed in the entrance section by the sharp-edged entrance in the transition piece preceding the heated length. This phenomenon is in agreement with similar velocity profile distortions found by Deissler.⁽¹⁵⁾ These distortions were noted over a considerably greater length of tube than found by Deissler in his round tube data and support evidence that a commensurately greater length of thin rectangular channel is required to establish a fully-developed velocity profile. No particular significance is attached to the slight increase in the coefficient near the end of the channel. It is conjectured that the one-dimensional temperature corrections at the outside wall apply with less accuracy at these points than at the central channel points.

It is apparent that no simple L/D_e relationship can be derived from the rectangular data. However, as pointed out in the discussion,

FIG. 14 - FINAL CORRELATION FOR SUPERHEATED STEAM FLOWING IN A TUBULAR CHANNEL.

FIG. 15 - TYPICAL COEFFICIENT VS. LENGTH
PLOT FOR THE RECTANGULAR CHANNEL

some valuable conclusions and observations can be extracted from Fig. 16. Figure 16 depicts the round tube correlation line (Eq. 21) at an L/D_e of 60. The thin rectangular data is plotted on the same coordinates to effect a comparison. Data is shown only for $66 < L/D_e < 121$, since this is the area of minimum h and supposedly close to the fully-developed flow region. Also shown is the Dittus-Boelter equation for further comparison purposes.

Twenty-four runs were made on the rectangular test sections and calculations are shown in Appendix E.

Discussion

Because of the lack of similar data, it is not possible to make direct comparisons using the correlations and data shown in Figs. 14 and 15. However, the correlation of McAdams, Kennel, and Addoms⁽⁴⁾ may be extended to the Reynolds number range included in the present study, and several interesting facts appear. As seen by Eq. 5, which uses the Timroth and Vargaftik⁽⁵⁾ thermal conductivity data, the influence of Reynolds number and L/D_e ratio is slightly different than that found in Eq. 21.

The two equations are repeated here:

$$\frac{h D_e}{k_f} = 0.0126 \left(\frac{G D_e}{\mu_f} \right)^{0.89} \left(\frac{c_p \mu}{k} \right)_f^{1/3} \left(\frac{L}{D_e} \right)^{-0.13} \quad (\text{McAdams})$$

$$\frac{h D_e}{k_f} = 0.0157 \left(\frac{G D_e}{\mu_f} \right)^{0.84} \left(\frac{c_p \mu}{k} \right)_f^{1/3} \left(\frac{L}{D_e} \right)^{-0.04} \quad (\text{Fig. 14})$$

The most pertinent comparison to be made indicates the difference in respective convection coefficients for a given heat transfer situation. Essentially the same steam properties are used in both equations; consequently the ratio of the two coefficients is instructive and can be calculated from

$$\frac{h_a}{h_m} = 1.246 \left(N_{Re} \right)^{-0.05} \left(\frac{L}{D_e} \right)^{0.09} \quad (22)$$

FIG. 16 - COMPARISON OF RECTANGULAR CHANNEL DATA WITH EQUA. 21.

where h_a denotes the coefficient of Fig. 14, and h_m denotes McAdams' coefficient. Figure 17 is a set of curves for Eq. 22 at various Reynolds numbers. Although the curves extend to an L/D_e of 60, it should be noted that the McAdams correlation is recommended for L/D_e between 13 and 57, and the present correlation only for L/D_e between 6 and 36. The fact that the two correlations are within ± 5 per cent of each other near the end of the recommended L/D_e range indicates that the establishment of the velocity profile is nearly complete and obviates the necessity for an L/D_e parameter beyond a certain undetermined number of diameters for either correlation. The discrepancy in the low L/D_e range supports the remarks made in the introduction concerning the importance of the unheated entrance section. In McAdams' experiment, the apparatus was constructed such that heating of the fluid began immediately after a 90-degree, approximately round bend in the flow path. The present experimental apparatus incorporated a "long" starting section, 16.3 diameters in length for the round tube. The experiments run by Boelter, Young, and Iversen⁽¹⁴⁾ show that 15 diameters downstream of the start of the heated length, the local coefficient for their unheated "long" calming section ($L/D_e = 11.2$) is about 87 per cent of the coefficient for a 90-degree round bend calming section. This case is for a Reynolds number of approximately 50,000. Figure 17 shows that the h_a/h_m ratio at 15 diameters is 92 per cent for a Reynolds number of 50,000. Recalling that the calming section used in the present study is 16.3 diameters in length, it is entirely reasonable to expect this 8 per cent discrepancy between McAdams and the present study.

The above discussion presents allowances which indicate that there is reasonable agreement between the two studies, although the exponent of the Reynolds number in both cases is open to some question. It may be that a constant slope, even in the fully developed convection region, is not feasible for complete coverage of the Reynolds number range, 6,000 to 370,000, which range includes both

FIG. 17 - COMPARISON OF CONVECTION COEFFICIENTS

experiments. Also the question of accurate conductivity values is not yet fully answered. The important thing to note here is that if Eq. 21 is used to predict convection coefficients for superheated steam, it must incorporate the conductivity and viscosity values found in Tables 1 and 2. Furthermore, careful attention must be given to the type of unheated entrance section used, and the correlation should not be extrapolated outside of the range of variables shown in Fig. 14, except that the L/D_e range may be safely extended to 60.

From Fig. 16, it can be postulated that some bias in the round channel data could have produced a slope of 0.84 rather than the conventional 0.80. If so, it can be concluded that the Dittus-Boelter equation is fairly well substantiated beyond an L/D_e of 60, which should certainly suffice as an upper limit for fully established flow in the round tube. Furthermore, except for the vena contracta peak, there is no observable difference between convective transfer in the round tube and similar transfer in the thin rectangular tube at an L/D_e of 60. As previously mentioned, there is no reason to expect any unusual behavior beyond an L/D_e of 60 in either geometry, so that beyond this point, Eq. 21 should suffice for either geometry and a corresponding constant coefficient should be used as follows:

$$\frac{h D_e}{k_f} = 0.0133 \left(\frac{G D_e}{\mu_f} \right)^{0.84} \left(\frac{c_p \mu}{k} \right)_f^{1/3} \quad (23)$$

It should be noted that a small change in the exponent of the Reynolds number results in a considerable change in the coefficient. For example, if the slope is changed from 0.84 to the conventional 0.80 at $N_{Re} = 10^5$, the equation becomes

$$\frac{h D_e}{k_f} = 0.02115 \left(\frac{G D_e}{\mu_f} \right)^{0.80} \left(\frac{c_p \mu}{k} \right)_f^{1/3} \quad (23a)$$

a value approximately eight per cent lower than the Dittus-Boelter equation. Thus the older correlation is fairly well corroborated if proper account is made for entrance effects.

CHAPTER IV

SUMMARY AND CONCLUSIONS

Forced convection heat transfer to superheated steam in turbulent flow was experimentally investigated, utilizing two channel geometries: a round tube, 0.333 inch I.D. x 12.00 inches, and a thin rectangular channel, 0.047 inches wide, aspect ratio, 26.6:1, and 12.00 inches in length. The experiments encompassed a Reynolds number range from 20,000 to 370,000, pressure range from 300 to 1500 psia, inlet superheat from 5 to 160 F°, and film temperature differences from 30 to 550 F°.

Concerning the round channel experiments, the following conclusions may be drawn:

1. Convection coefficients may be predicted within ± 10 per cent, using the following equations and the properties found in Tables 1 and 2:

$$\frac{h D_e}{k_f} = 0.0157 \left(\frac{G D_e}{\mu_f} \right)^{0.84} \left(\frac{c_p \mu}{k} \right)_f^{1/3} \left(\frac{L}{D_e} \right)^{-0.04}$$

$$\text{for } (6 < \frac{L}{D_e} < 60)$$

and

$$\frac{h D_e}{k_f} = 0.0133 \left(\frac{G D_e}{\mu_f} \right)^{0.84} \left(\frac{c_p \mu}{k} \right)_f^{1/3}$$

$$\text{for } (L/D_e > 60)$$

2. Particular attention should be paid to the type of unheated entrance section preceding the heated length of channel. The first of the above equations is applicable when this precedent section is

16 diameters in length. It may be safely used (with respect to limiting surface temperatures) for shorter unheated sections and should closely apply for a more fully developed velocity profile (unheated $L/D_e > 16$), because of the relatively weak influence of L/D_e .

3. Making allowances for the unheated entrance section, the data substantiates the experiments of McAdams, Kennel, and Addoms⁽⁴⁾ within ± 5 per cent, even to a Reynolds number of 370,000.

4. The Dittus-Boelter equation,

$$\frac{h D_e}{k_f} = 0.023 \left(\frac{G D_e}{\mu_f} \right)^{0.80} \left(\frac{C_p \mu}{k} \right)_f^{1/3},$$

when compared to the present study, yields values of the coefficient which are 11.5 per cent high in the lower Reynolds range (25,000) and 1.8 per cent high in the upper range (370,000). It is conjectured that this is due, in part, to the unsettled question of correct steam conductivity values.

Concerning the thin rectangular channel experiments, the following observations are made:

1. The vena contracta caused by the sharp-edged entrance condition influences the heat transfer coefficient to a length of approximately 75 equivalent diameters, including the unheated entrance length of 13 diameters.

2. There is no observable difference between the rectangular data and the round channel correlation in the L/D_e range from 60 to 121. Even if it is postulated that the flow is not fully established in this region, there is no evidence that any serious decrease in coefficient will occur beyond this range, even according to the data of Levy, Fuller, and Niemi,⁽¹⁶⁾ whose data run to an L/D_e of approximately 180.

APPENDIX A

ORIFICE CALIBRATION

The 0.2342 inch orifice was machined and mounted according to the ASME standards.⁽¹⁸⁾ The calibration was made in place in the test loop, and steam at approximately 600 psia was passed through the loop to a temporary condenser and weighing apparatus. The differential pressure was measured by manometer, using mercury as the measuring fluid. The flow coefficient, K , which includes the velocity of approach factor, was calculated for each point by

$$K = \frac{182.6 \dot{m}}{Y F_a \sqrt{\Delta p} \rho_1} \quad (24)$$

The results are plotted in Fig. 18, which includes comparison lines from reference 18.

FIG. 18 - CALIBRATION OF 0.2342 inch ORIFICE

APPENDIX B

ZERO POWER THERMOCOUPLE CALIBRATION

Because of heat losses through the insulation surrounding the heated length of the test sections, some temperature difference was present between the point of temperature measurement and the true outside wall temperature, t_0 . This was due to the thin ceramic coating on the tubular section and the 0.001 inch mica between the thermocouple bead and the metal wall of the rectangular section. When the apparatus was operating at zero power, and steady state conditions prevailed, readings were taken along the wall and compared to the steam temperature inside the channel. These differences were used as a basis for correcting each reading at power.

Figure 19 is a sketch of the temperature profiles at zero power and with wall heat generation. For the one-dimensional conduction case, valid only for points sufficiently far from the electrodes, one can write for the radial or transverse heat loss

$$q_L = C_1 (t_0 - t_a) \quad (25)$$

where C_1 is a function of the geometry, the various conductivities, and the free convection coefficient in the ambient air. Also for the transfer through the insulation, exclusive of the ceramic, one can write

$$q_L = C_2 (t_c - t_a) \quad (26)$$

where C_2 is a function of geometry and the conductivity of the insulation, and t_c is the temperature as seen by the thermocouple bead. By an evaluation of C_1 in Eq. 25, an estimate of the insulation heat loss can be obtained. A maximum value of $t_0 - t_a$ is 600 F°, and C_1 is approximately 0.232 B/hr - °F, using a mean temperature for conductivities. In this case,

$$q_L = (0.232) 600 = 139 \text{ B/HR} \quad (27)$$

FIG. 19 - CONDUCTION REGIONS AND TEMPERATURE POINTS FOR ZERO POWER CORRECTIONS.

A simple calculation shows that the temperature difference through the wall amounts to a maximum of 0.2 F° at zero power. Therefore, it is reasonable to assume that the steam temperature, t_s , is equal to the outside metal wall temperature, t_0 . The correction is therefore $t_s - t_c$ at zero power. However, at power, this temperature correction is increased by the increase in heat loss. Denoting power conditions with the subscript "p", the heat loss at power is

$$q_{Lp} = C_1 (t_{op} - t_a) \quad (28)$$

and

$$q_{Lp} = C_2 (t_{cp} - t_a) \quad (29)$$

Combining Eqs. 25, 26, 28 and 29 yields

$$\frac{q_{Lp}}{q_L} = \frac{t_{op} - t_a}{t_0 - t_a} = \frac{t_{cp} - t_a}{t_c - t_a} \quad (30)$$

From Eq. 30,

$$t_{op} = t_a + (t_0 - t_a) \frac{t_{cp} - t_a}{t_c - t_a} \quad (31)$$

Equation 31 was applied to all the t_c measurements; it is recognized that the corrections made near the electrode junctures are not as accurate as those between $L/D_e = 6$ and $L/D_e = 30$, since the one-dimensional transfer problem does not apply near these heat sinks.

APPENDIX C
STEADY STATE CONDUCTION PROBLEM
FOR TEST SECTION CONFIGURATIONS

The differential equation expressing the steady state conduction of heat with internal heat generation in the walls of a hollow cylinder (tubular test section) can be written in cylindrical coordinates as

$$\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial r^2} + \frac{1}{r} \frac{\partial t}{\partial r} + \beta^2 t = - \frac{q_0'''}{k} \quad (32)$$

This equation includes heat generation variable with temperature, and constant conductivity. Since one presumably measures the outside wall temperature profile and the temperatures at the electrode juncture, and further, since the outside wall is well insulated, the following boundary conditions are applicable and will determine a unique solution:

$$t = t_a \text{ at } x = 0, t = t_b \text{ at } x = L \quad (33)$$

$$\frac{\partial t}{\partial r}(x, r_0) = 0 \quad (34)$$

$$t(x, r_0) = f(x) \quad (35)$$

Assuming a solution of the form,

$$t(x, r) = X(x) R(r) + \Psi(x) \quad (36)$$

and applying the boundary conditions, a solution is evolved:

$$\begin{aligned} t(x, r) = & \sum_{n=1}^{\infty} A_n I_0(\lambda_n r) + B_n K_0(\lambda_n r) \sin \frac{n\pi x}{L} \\ & + t_a + (t_b - t_a) \frac{\sin \beta x}{\sin \beta L} \\ & + \left(t_a + \frac{q_0'''}{\beta^2 k} \right) \left[\frac{\cos \beta \left(\frac{1}{2} L - x \right)}{\cos \frac{1}{2} \beta L} - 1 \right] \end{aligned} \quad (37)$$

where

$$A_n = \frac{b_n K_1(\lambda_n r_0)}{I_0(\lambda_n r_0) K_1(\lambda_n r_0) + I_1(\lambda_n r_0) K_0(\lambda_n r_0)} \quad (38)$$

$$B_n = \frac{b_n I_1 (\lambda_n r_0)}{I_0 (\lambda_n r_0) K_1 (\lambda_n r_0) + I_1 (\lambda_n r_0) K_0 (\lambda_n r_0)} \quad (39)$$

and b_n are the coefficients in the Fourier sine series,

$$g(x) = \sum_{n=1}^{\infty} b_n \sin \frac{n\pi x}{L} \quad (40)$$

$$b_n = \frac{2}{L} \int_0^L g(x') \sin \frac{n\pi x'}{L} dx' \quad (40a)$$

$$\text{and } g(x) = f(x) - \Psi(x) \quad (41)$$

In this case,

$$\lambda_n^2 = \frac{n^2 \pi^2}{L^2} - \beta^2 \quad (42)$$

and we deduce that β must be less than or equal to π/L , if Eq. 37 is to yield real values of $t(x, r)$. It is noted that

$$\beta^2 = \alpha \frac{q_0'''}{k} \quad (43)$$

and, in the present experiment, q_0 is much too large to satisfy the real value requirement. The problem of non-variable resistivity (heat generation) can be solved in the same manner as above, thus eliminating the difficulty of determining the λ_n . However, there still remains the practical difficulty of forming an $f(x)$ to conform with the measured temperature profile at r_0 . This can be done by fitting a function to the shape, but the subsequent integration required to find the b_n is laborious. Since the main interest here is one involving variable heat generation, it was decided to proceed directly to the one-dimensional axial case outlined below.

Assuming that the temperature drop across the wall in the radial direction is small compared to the gradient near the electrode junctions, the ordinary differential equation is derived for heat generation variable with temperature, as

$$\frac{d^2 \theta}{d x^2} - (m^2 - \beta^2) \theta = - \frac{q_0'''}{k} - m^2 \gamma x \quad (44)$$

where

$$\theta = t - t_a \quad (45)$$

$$\text{and } m = \sqrt{\frac{h P}{k A_a}} \quad (46)$$

The boundary conditions are

$$\theta = 0 \text{ at } x = 0 \quad (47)$$

$$\theta = t_b - t_a \text{ at } x = L \quad (48)$$

Assuming h and k are constant along the tube length, the solution is

$$\theta(x) = M \exp(-\sqrt{m^2 - \beta^2} x) + N \exp(\sqrt{m^2 - \beta^2} x) + \theta_p \quad (49)$$

where

$$M = - \left[N + \frac{q_0'''/k}{m^2 - \beta^2} \right] \quad (50)$$

$$N = \frac{(t_b - t_a) + \left[\frac{q_0'''/k}{m^2 - \beta^2} \right] \left[\exp(-\sqrt{m^2 - \beta^2} L) - 1 \right] - \frac{m^2 \gamma L}{m^2 - \beta^2}}{\exp(\sqrt{m^2 - \beta^2} L) - \exp(-\sqrt{m^2 - \beta^2} L)} \quad (51)$$

$$\theta_p = \frac{q_0 + m^2 \gamma x}{m^2 - \beta^2} \quad (52)$$

At this point it is instructive to choose some typical values of the constants involved to illustrate to what degree the electrodes affect the axial temperature profile. Figure 20 is a set of curves of θ versus length for various values of h and the following typical values.

$$q_0''' = 4.00 \times 10^7 \text{ B hr}^{-1} \text{ ft}^{-3}$$

$$k = 13.0 \text{ B hr}^{-1} \text{ ft}^{-1} \text{ } ^\circ\text{F}^{-1}$$

$$t_b - t_a = 240 \text{ F}^\circ$$

$$\alpha = 4.00 \times 10^{-4} \text{ } ^\circ\text{F}^{-1}$$

$$\gamma = 20 \text{ F}^\circ - \text{in.}^{-1}$$

$$P = 0.0872 \text{ ft}^2 \text{ (tubular test section)}$$

$$A_a = 1.665 \times 10^{-4} \text{ (tubular test section)}$$

FIG. 20 - THEORETICAL TEMPERATURE PROFILES

In these curves, only a very slight influence is noted at $x = 1$ ($L/D_e = 3$) and $x = 11$ ($L/D_e = 33$). Figure 12 of the text shows that the measured profiles exhibit a somewhat more pronounced influence up to $L/D_e = 2$ and beyond $L/D_e = 30$. It is postulated that heat losses to the insulation and a variable h account for the small differences in the regions $0 < x < 2$ and $10 < x < 12$.

The one-dimensional radial and transverse conduction cases for calculation of $t_0 - t_i$ have been derived and discussed in the main text.

Although the above discussion was concentrated on the cylindrical test section, the same analysis and remarks apply to the rectangular section, except that the general two-dimensional problem is derived in Cartesian coordinates. Also, because of the highly variable coefficient, h , in the entrance section, the experimental curves do not agree with those of Fig. 20 in the thermal entry length.

APPENDIX D

ERROR ANALYSIS

The introduction of experimental error arises from laboratory measurements of power, flow, pressures, and temperatures, and from the subsequent assumptions required for calculation of q'' , t_0 , and t_i . All the following observations are made using maximum possible error values.

The maximum error in temperature readings due to inaccuracy of the chromel-alumel thermocouples is 1.5°F by calibration of a piece of wire from the spool from which all the thermocouples were made. Combined instrument and reading error is 0.2°F . This 1.7°F error combined with the error introduced by assumptions in the zero power calibration technique will yield the direct error in t_0 . The ratio of the maximum axial temperature gradient (about $300^\circ\text{F}/\text{ft}$) in the central portion of the test section to the attendant radial or transverse temperature gradient (about $20,000^\circ\text{F}/\text{ft}$) yields an error in non-linearity of two per cent of the correction applied to yield the true t_0 . This is a maximum error of 0.3°F , based upon the highest temperature run. The ratio of mean conductivities between zero power and the highest power run adds a further error of approximately five per cent of the same correction or 0.8°F . The total maximum error in t_0 is therefore 2.8°F . The assumption of perfect insulation and the error in wall thickness leads to a maximum error of 0.3°F . Therefore t_i may be in error by 3.1°F or a maximum of 2.0 per cent of the minimum film drop.

Errors in $q''(x)$ arise from the inherent errors in $\rho(x)$ and the measurement of current and tube cross-sectional area. A 3.0°F error in temperature results in a 0.2 per cent discrepancy in resistivity and the resistivity-temperature curve may be in error by three per cent in the high temperature range. The ammeter introduces a 0.50 per cent error, which amounts to a one per cent difference in calculated heat

flux. Room temperature measurements of test section resistance indicated a three per cent agreement with the measured cross-sectional area of the sections. Therefore a probable error of 7.2 per cent exists in heat flux calculation.

Errors in fluid temperature, t_s , arise from the measurement of inlet and outlet bulk temperatures, heat loss between the mixing chambers and the heated length, axial heat transfer in the tube near the electrodes, and compressible flow and friction effects. Variable heat generation also affects the enthalpy increase in the fluid. In the tubular test section, the total pressure drop between mixing chambers was never greater than 7.8 psi, so that compressible flow and frictional losses contribute a negligible error. By the zero power measurements and an estimate of the heat losses in the heated length, the maximum error incurred at the inlet by heat loss through the inlet flanges and piping was 1.0 F°; the same applies at the outlet. Variable heat generation accounts for a small difference in enthalpy addition from the heated entrance. This can be shown by plotting $q''(x)$ vs x and planimentering the area under the curve point by point. The maximum error occurs up to the first point used for calculation ($x = 2$ inches) and amounts to three per cent of the assumed linear temperature-length line between $x = 0$ and $x = 2$. For the maximum heat input this amounted to an error of 1.0 F° in fluid temperature. The fact that there is axial conduction to the electrodes further distorts this assumed linearity, and again the maximum error occurs at $x = 2$. This heat loss can be approximated by the use of the one-dimensional axial problem derived in Appendix C. Differentiation of Eq. 49 yields

$$\begin{aligned} \frac{d\theta}{dx} = & -\sqrt{m^2 - \beta^2} M \exp(-\sqrt{m^2 - \beta^2} x) \\ & + \sqrt{m^2 - \beta^2} N \exp(\sqrt{m^2 - \beta^2} x) + \frac{m^2 \gamma}{m^2 - \beta^2} \end{aligned} \quad (53)$$

Some values of the various constants were chosen from a high temperature run (Table 23) to evaluate the axial conduction losses. For this case, the following constants apply.

$$\begin{aligned}m^2 &= 12,090 \text{ ft}^{-2} \\ \beta^2 &= 2,224 \text{ ft}^{-2} \\ q_0''' &= 7.311 \times 10^7 \text{ B} \cdot \text{hr}^{-1} \cdot \text{ft}^{-3} \\ k &= 13.0 \text{ B} \cdot \text{hr}^{-1} \cdot \text{ft}^{-1} \cdot ^\circ\text{F}^{-1} \\ \gamma &= 170 \text{ F}^\circ \cdot \text{ft}^{-1} \\ t_b - t_a &= 170 \text{ F}^\circ\end{aligned}$$

Substitution into Eq. 53 yields

$$\left. \frac{d\theta}{dx} \right|_{x=0} = 56,260 \text{ F}^\circ/\text{ft}$$

and

$$\left. \frac{d\theta}{dx} \right|_{x=L} = 55,110 \text{ F}^\circ/\text{ft}$$

Therefore

$$q_a \Big|_{x=0} = -k A_a \left. \frac{d\theta}{dx} \right|_{x=0} = 121.6 \text{ B/hr}$$

$$q_a \Big|_{x=L} = -k A_a \left. \frac{d\theta}{dx} \right|_{x=L} = 119.0 \text{ B/hr}$$

This loss amounts to a total 1.70 per cent of the heat generated in the tube, and 0.86 per cent of the heat generated is not added to the fluid from $x = 0$ to $x = 2$ inches. This amounts to a 0.7 F° error at the point of maximum error, $x = 2$ inches. Non-linearity of specific heat in the region of investigation introduces a further error in temperature of 0.7 F° at $x = 2$ and 2.1 F° at $x = 6$. However the total error at $x = 6$ from the previously discussed sources amounts to only 1.0 F° , so that $x = 2$ is still the point of maximum error. The maximum error in fluid temperature at the highest film temperature difference and fluid temperature rise is therefore 3.4 F° . This is a 0.7 per cent discrepancy in film temperature difference.

The total error in the convection coefficient as calculated by Eq. 7 is 11.0 per cent. Differences in average film temperature due to the above errors have a negligible effect on property evaluation. The fact that G , as used in the Reynolds number, is determined independently by orifice introduces a one per cent deviation in Reynolds number, which amounts to a 0.8 per cent deviation in the ordinates of Figs. 13 and 15. Therefore, the total theoretical error in these ordinates is 11.8 per cent. The error in Fig. 15 (rectangular data) may be slightly larger, since a somewhat greater discrepancy in fluid temperature is present due to a larger pressure drop than that found in the circular tube experiments.

APPENDIX E
TABULATED DATA FOR CALCULATION OF
THE LOCAL CONVECTION COEFFICIENT

Figure 21 is a set of curves used for the determination of the thermal conductivity and the electrical resistivity of 304 stainless steel.⁽¹⁹⁾

Tables 4 through 35 present the data used to compute the local coefficient in the circular channel. Tables 36 through 59 present similar data applicable to the thin rectangular channel.

In Tables 4 through 59, t_f is the bulk temperature of the fluid, equivalent to t_s in the main text.

FIG. 21 - THERMAL CONDUCTIVITY AND ELECTRICAL RESISTIVITY OF 304 S.S.

Table 4. Round Channel. Run No. 1; Inlet Pressure, 606.4 psia; $\dot{m}$, 323.8 lb/hr; Inlet Δt_{sup} , 106.1 F°; Power Setting, 2.059 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	no zero power calibration for t ₀							
1	3								
2	6								
3	9								
4	12								
5	15								
6	18	746.4	99.7	8.048	6.0	740.4	609.0	131.4	612.5
7	21	750.4	99.9	8.064	6.0	744.4	611.9	132.5	608.6
8	24	754.7	100.1	8.080	6.0	748.7	614.7	134.0	602.9
9	27	758.7	100.3	8.096	6.0	752.7	617.5	135.2	598.8
10	30	762.8	100.5	8.112	6.0	756.8	620.2	136.6	593.9
11	33						622.9		
12	36						625.6		

Table 5. Round Channel. Run No. 2; Inlet Pressure, 615.3 psia; $\dot{m}$, 290.7 lb/hr; Inlet Δt_{sup} , 103.4 F°; Power Setting, 1.546 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	639.1					592.4		
1	3	699.3					594.4		
2	6	705.6	99.62	5.865	4.4	701.2	596.5	104.7	559.6
3	9	708.7	99.74	5.872	4.4	704.3	598.6	105.7	555.5
4	12	711.7	99.86	5.879	4.4	707.3	600.7	106.6	551.4
5	15	714.9	99.98	5.886	4.4	710.5	602.8	107.7	546.5
6	18	718.1	100.1	5.893	4.4	713.7	604.9	108.8	541.6
7	21	721.2	100.22	5.900	4.4	716.8	607.0	109.8	537.3
8	24	724.3	100.34	5.907	4.4	719.9	609.0	110.9	532.6
9	27	727.4	100.46	5.914	4.4	723.0	611.2	111.8	528.9
10	30	730.6	100.58	5.921	4.4	726.2	613.3	112.9	524.4
11	33	732.5					615.4		
12	36	691.8					617.6		

Table 6. Round Channel. Run No. 3; Inlet Pressure, 620.4 psia; $\dot{m}$, 283.7 lb/hr; Inlet Δt_{sup} , 104.4 F°; Power Setting, 4.12 kw.

x	L/D _e	t ₀	$\rho(x)$	$q''_R(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B}}{\text{hr}}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	719.7					594.2		
1	3	890.0	103.8	1.600	11.0	880.0	600.6	279.4	572.5
2	6	901.0	104.2	1.606	11.0	890.0	606.9	283.1	565.5
3	9	911.0	104.6	1.612	11.0	900.0	613.2	286.8	561.0
4	12	921.0	105.0	1.618	11.0	910.0	619.6	290.4	555.5
5	15	932.0	105.4	1.624	11.0	921.0	626.0	295.0	551.0
6	18	942.0	105.8	1.630	11.0	931.0	632.3	298.7	545.0
7	21	952.0	106.2	1.636	11.0	941.0	638.6	302.4	541.0
8	24	962.0	106.6	1.642	11.0	951.0	645.0	306.0	536.0
9	27	973.0	107.0	1.648	11.0	962.0	651.3	310.7	531.0
10	30	983.0	107.4	1.654	11.0	972.0	657.7	314.3	527.0
11	33	993.0	107.8	1.660	11.0	981.9	664.1	317.8	522.0
12	36	909.4					670.3		

Table 7. Round Channel. Run No. 4; Inlet Pressure, 624.4 psia; $\dot{m}$, 277.1 lb/hr; Inlet Δt_{sup} , 105.7 F°; Power Setting 6.50 kw.

x	L/D _e	t ₀	$\rho(x)$	$q''_R(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B}}{\text{hr}}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	798.5					596.5		
1	3	1059.8					606.8		
2	6	1089.0	109.7	24.87	16.4	1072.6	617.0	455.6	545.8
3	9	1105.5	110.4	25.03	16.4	1089.1	627.0	462.1	541.6
4	12	1122.0	111.0	25.16	16.4	1105.6	637.3	468.3	537.3
5	15	1138.5	111.7	25.32	16.4	1122.1	647.6	474.5	533.6
6	18	1155.0	112.3	25.46	16.4	1138.6	658.0	480.6	529.7
7	21	1171.5	112.9	25.59	16.4	1155.1	668.4	486.7	525.8
8	24	1188.5	113.6	25.75	16.4	1172.1	678.7	493.4	521.9
9	27	1205.0	114.2	25.89	16.4	1188.6	689.2	499.4	518.4
10	30	1221.5	114.9	26.05	16.4	1205.1	699.4	505.7	515.0
11	33	1238.5	115.5	26.18	16.4	1222.1	710.0	512.1	511.3
12	36	1064.7					720.8		

Table 8. Round Channel. Run No. 5; Inlet Pressure, 652.4 psia; $\dot{m}$, 253.7 lb/hr; Inlet Δt_{sup} , 96.8 F°; Power Setting 1.765 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	647.9					592.1		
1	3	723.8	97.4	6.740	5.0	718.8	594.9	123.9	541.7
2	6	728.3	97.6	6.753	5.0	723.3	597.6	125.7	534.9
3	9	732.8	97.8	6.766	5.0	727.8	600.3	127.5	530.6
4	12	737.3	98.0	6.780	5.0	732.3	603.1	129.2	524.7
5	15	741.8	98.1	6.787	5.0	736.8	605.8	131.0	518.0
6	18	746.3	98.3	6.800	5.0	741.3	608.6	132.7	512.4
7	21	750.8	98.5	6.814	5.0	745.8	611.3	134.5	506.6
8	24	755.3	98.6	6.821	5.0	750.3	614.1	136.2	500.8
9	27	759.8	98.8	6.835	5.1	754.7	616.8	137.9	495.6
10	30	764.3	99.0	6.849	5.1	759.2	619.6	139.6	490.6
11	33	768.8	99.2	6.863	5.1	763.7	622.3	141.4	485.3
12	36	715.1					625.1		

Table 9. Round Channel. Run No. 6; Inlet Pressure, 644.3 psia; $\dot{m}$, 246.2 lb/hr; Inlet Δt_{sup} , 96.8 F°; Power Setting 3.240 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	689.7					590.7		
1	3	827.0					595.8		
2	6	840.5					601.0		
3	9	849.8	102.4	11.76	9.6	840.2	606.3	233.9	502.8
4	12	858.1	102.6	11.79	9.6	848.5	611.6	236.9	497.7
5	15	866.5	102.9	11.83	9.6	856.9	616.9	240.0	492.9
6	18	874.8	103.2	11.87	9.6	865.2	622.2	243.0	488.5
7	21	883.1	103.5	11.91	9.6	873.5	627.5	246.0	484.1
8	24	891.4	103.8	11.95	9.7	881.7	632.8	248.9	480.1
9	27	899.8	104.2	12.00	9.7	890.1	638.2	251.9	476.4
10	30	908.2	104.5	12.04	9.7	898.5	643.5	255.0	472.2
11	33	916.6	104.8	12.08	9.7	906.9	649.0	257.6	468.9
12	36	820.1					654.5		

Table 10. Round Channel. Run No. 7; Inlet Pressure, 643.4 psia; $\dot{m}$, 239.3 lb/hr; Inlet Δt_{sup} , 96.5 F°; Power Setting, 4.798 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_R''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	747.3					590.3		
1	3	968.5	104.0	18.21	12.5	956.0	599.0	357.0	510.1
2	6	994.0	105.0	18.39	12.5	981.5	607.7	373.8	491.8
3	9	1012.7	105.6	18.49	12.5	1000.2	616.5	383.7	481.9
4	12	1027.9	106.3	18.61	12.5	1015.5	625.3	390.2	477.0
5	15	1041.2	106.8	18.70	12.5	1028.7	634.1	394.6	473.8
6	18	1054.5	107.3	18.79	12.5	1042.0	642.9	399.1	470.8
7	21	1067.8	107.8	18.88	12.6	1055.2	651.7	403.5	467.8
8	24	1081.1	108.3	18.96	12.6	1068.5	660.5	408.0	464.7
9	27	1094.4	108.8	19.05	12.6	1081.8	669.4	412.4	462.0
10	30	1107.7	109.4	19.16	12.6	1095.1	678.3	416.8	459.6
11	33	1121.0	109.9	19.24	12.6	1108.4	687.1	421.3	456.8
12	36	968.8					696.2		

Table 11. Round Channel. Run No. 8; Inlet Pressure, 640.4 psia; $\dot{m}$, 234.5 lb/hr; Inlet Δt_{sup} , 96.8 F°; Power Setting, 5.773 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_R''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	786.6					590.1		
1	3	1071.5	110.1	21.83	14.5	1057.0	601.3	455.7	479.0
2	6	1102.5	111.2	22.05	14.5	1088.0	612.1	475.9	463.3
3	9	1124.0	112.2	22.25	14.5	1109.5	623.4	486.1	457.7
4	12	1141.0	112.8	22.37	14.5	1126.5	634.6	491.9	454.8
5	15	1158.0	113.4	22.49	14.6	1143.4	645.5	497.9	451.7
6	18	1175.0	114.1	22.62	14.6	1160.4	656.7	503.7	449.0
7	21	1192.0	114.8	22.75	14.6	1177.4	667.9	509.5	446.6
8	24	1209.0	115.4	22.87	14.6	1194.4	679.0	515.4	443.7
9	27	1226.0	116.0	22.99	14.6	1211.4	690.1	521.3	441.0
10	30	1243.0	116.7	23.13	14.6	1228.4	701.3	527.1	438.8
11	33	1260.0	117.3	23.25	14.6	1245.4	712.5	532.9	436.3
12	36	1069.7					724.3		

Table 12. Round Channel. Run No. 9; Inlet Pressure, 606.4 psia; $\dot{m}$, 183.3 lb/hr; Inlet Δt_{sup} , 94.9 F°; Power Setting, 1.399 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_R''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	634.7					582.2		
1	3	710.6	97.0	5.571	4.0	706.6	585.4	121.2	459.7
2	6	724.7	97.5	5.600	4.0	720.7	588.6	132.1	423.9
3	9	731.3	97.7	5.611	4.0	727.3	591.8	135.5	414.1
4	12	735.6	97.9	5.623	4.0	731.6	595.0	136.6	411.6
5	15	740.0	98.0	5.628	4.0	736.0	598.2	137.8	408.4
6	18	744.4	98.2	5.640	4.0	740.4	601.4	139.0	405.8
7	21	748.8	98.4	5.651	4.0	744.8	604.6	140.2	403.1
8	24	753.2	98.5	5.657	4.0	749.2	607.8	141.4	400.1
9	27	757.5	98.7	5.669	4.0	753.5	611.0	142.5	397.8
10	30	761.9	98.9	5.680	4.0	757.9	614.2	143.7	395.3
11	33	762.9					617.4		
12	36	705.9					620.6		

Table 13. Round Channel. Run No. 10; Inlet Pressure, 604.4 psia; $\dot{m}$, 170.7 lb/hr; Inlet Δt_{sup} , 92.0 F°; Power Setting, 2.749 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_R''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	685.1					579.0		
1	3	847.5	102.2	10.42	7.5	840.0	585.8	254.2	410.1
2	6	873.8	103.3	10.54	7.5	866.3	592.6	273.7	385.0
3	9	889.2	104.3	10.64	7.5	881.7	599.4	282.3	376.9
4	12	898.8	104.6	10.67	7.5	891.3	606.3	285.0	374.4
5	15	908.5	105.0	10.71	7.5	901.0	613.1	287.9	372.0
6	18	918.2	105.4	10.75	7.5	910.7	619.9	290.8	369.7
7	21	927.9	105.8	10.79	7.5	920.4	626.7	293.7	367.4
8	24	937.5	106.1	10.82	7.5	930.0	633.6	296.4	365.1
9	27	947.2	106.5	10.86	7.5	939.7	640.4	299.3	362.9
10	30	956.8	106.8	10.89	7.5	949.3	647.3	302.0	360.7
11	33	960.1					654.3		
12	36	843.4					661.3		

Table 14. Round Channel. Run No. 11; Inlet Pressure, 636.3 psia; $\dot{m}$, 189.0 lb/hr; Inlet Δt_{sup} , 82.8 F°; Power Setting, 4.326 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_R''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	748.3					575.7		
1	3	1033.5	109.2	17.24	11.6	1021.9	586.5	435.4	396.0
2	6	1057.3	110.1	17.38	11.6	1045.7	597.3	448.4	387.7
3	9	1080.0	111.0	17.53	11.6	1068.4	608.1	460.3	380.8
4	12	1100.8	111.8	17.65	11.6	1089.2	618.9	470.3	375.3
5	15	1120.5	112.6	17.79	11.6	1108.9	629.7	479.2	371.0
6	18	1138.3	113.3	17.89	11.6	1126.7	640.5	486.2	367.9
7	21	1154.7	113.9	17.98	11.6	1143.1	651.3	491.8	365.7
8	24	1169.8	114.5	18.08	11.7	1158.1	662.1	496.0	364.5
9	27	1183.5	115.0	18.16	11.7	1171.8	672.9	498.9	364.0
10	30	1195.5	115.5	18.24	11.7	1183.8	683.7	500.1	364.7
11	33	1206.0	115.9	18.30	11.7	1194.3	694.5	499.8	366.2
12	36	1025.8					705.3		

Table 15. Round Channel. Run No. 12; Inlet Pressure, 617.4 psia; $\dot{m}$, 134.6 lb/hr; Inlet Δt_{sup} , 141.6 F°; Power Setting 1.331 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_R''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	700.3					630.0		
1	3	808.0	103.0	5.022	4.2	803.8	634.1	169.7	295.9
2	6	817.4	103.4	5.042	4.2	813.2	638.3	174.9	288.3
3	9	824.6	103.7	5.056	4.2	820.4	642.8	177.6	284.7
4	12	831.7	104.0	5.071	4.2	827.5	647.1	180.4	281.1
5	15	838.5	104.2	5.081	4.2	834.3	651.4	182.9	277.8
6	18	845.2	104.5	5.095	4.2	841.0	655.7	185.3	275.0
7	21	851.7	104.8	5.110	4.2	847.5	660.2	187.3	272.8
8	24	857.5	105.0	5.120	4.2	853.3	664.4	188.9	271.0
9	27	861.8	105.2	5.130	4.2	857.6	668.8	188.8	271.7
10	30	864.5	105.3	5.134	4.3	860.2	673.2	187.0	274.5
11	33	855.9					677.7		
12	36	801.0					682.2		

Table 16. Round Channel. Run No. 13; Inlet Pressure, 609.4 psia; $\dot{m}$, 130.5 lb/hr; Inlet Δt_{sup} , 105.3 F°; Power Setting, 2.521 kw,

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	726.1					593.2		
1	3	936.0	106.1	9.568	6.7	929.3	602.3	327.0	292.6
2	6	958.7	107.0	9.649	6.7	952.0	611.4	340.6	283.3
3	9	974.6	107.6	9.703	6.7	967.9	620.5	347.4	279.3
4	12	987.7	108.1	9.750	6.7	981.0	629.6	351.4	277.4
5	15	1001.0	108.6	9.793	6.7	994.3	638.7	355.6	275.4
6	18	1013.1	109.1	9.839	6.7	1006.4	647.8	358.6	274.4
7	21	1025.1	109.6	9.884	6.7	1018.4	656.9	361.5	273.4
8	24	1036.4	110.0	9.920	6.7	1029.7	666.0	363.7	272.7
9	27	1047.0	110.4	9.956	6.7	1040.3	675.1	365.2	272.6
10	30	1057.3	110.8	9.992	6.7	1050.6	684.2	366.4	272.7
11	33	1066.3	111.1	10.02	6.7	1059.6	693.3	366.3	273.5
12	36	933.7					702.4		

Table 17. Round Channel. Run No. 14; Inlet Pressure, 593.4 psia; $\dot{m}$, 123.2 lb/hr; Inlet Δt_{sup} , 107.6 F°; Power Setting 3.500 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	776.6					592.6		
1	3	1078.0	111.2	13.70	9.4	1068.6	605.4	463.2	295.8
2	6	1116.0	112.7	13.88	9.4	1106.6	618.2	488.4	284.3
3	9	1142.7	113.7	14.01	9.4	1133.3	631.0	502.3	278.9
4	12	1163.5	114.5	14.11	9.4	1154.1	643.9	510.2	276.6
5	15	1182.5	115.2	14.19	9.4	1173.1	656.7	516.4	274.8
6	18	1200.5	115.9	14.28	9.4	1191.1	669.5	521.6	273.8
7	21	1218.5	116.6	14.36	9.4	1209.1	682.3	526.8	272.7
8	24	1236.5	117.3	14.45	9.4	1227.1	695.1	532.0	271.6
9	27	1253.0	117.9	14.52	9.4	1243.6	707.9	535.7	271.1
10	30	1265.5	118.4	14.59	9.4	1256.1	720.8	535.3	272.6
11	33	1275.0	118.8	14.64	9.4	1265.6	733.6	532.0	275.1
12	36	1085.9					746.4		

Table 18. Round Channel. Run No. 15; Inlet Pressure, 344.4 psia; $\dot{m}$, 272.5 lb/hr; Inlet Δt_{sup} , 132.2 F°; Power Setting, 1.963 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	628.0					562.4		
1	3	714.6	96.8	7.622	5.7	708.9	565.9	143.0	533.0
2	6	719.7	97.0	7.638	5.7	714.0	569.4	144.6	528.2
3	9	724.8	97.2	7.654	5.7	719.1	572.9	146.2	523.5
4	12	729.9	97.4	7.669	5.7	724.2	576.4	147.8	518.9
5	15	735.0	97.6	7.685	5.7	729.3	579.9	149.4	514.4
6	18	740.2	97.8	7.700	5.7	734.5	583.3	151.2	509.3
7	21	745.4	98.0	7.716	5.7	739.7	586.7	152.0	507.7
8	24	750.5	98.2	7.732	5.7	744.8	590.2	154.6	500.1
9	27	755.7	98.4	7.748	5.7	750.7	593.7	157.0	493.5
10	30	760.9	98.6	7.764	5.7	755.2	597.2	158.0	491.4
11	33	765.2	98.8	7.780	5.7	760.5	600.7	159.8	486.8
12	36	702.1					604.2		

Table 19. Round Channel. Run No. 16; Inlet Pressure, 348.4 psia; $\dot{m}$, 244.2 lb/hr; Inlet Δt_{sup} , 132.0 F°; Power Setting, 3.543 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	685.8					563.3		
1	3	848.2	101.9	13.68	9.9	838.3	570.2	268.6	509.5
2	6	858.6	102.3	13.74	9.9	848.7	577.2	271.5	506.0
3	9	869.2	102.8	13.81	9.9	859.3	584.1	275.2	501.7
4	12	879.8	103.2	13.86	9.9	869.9	591.1	278.8	497.1
5	15	890.4	103.6	13.91	9.9	880.5	598.0	282.5	492.5
6	18	901.0	104.0	13.97	9.9	891.1	605.0	286.1	488.2
7	21	911.6	104.4	14.00	9.9	901.7	612.0	289.7	484.0
8	24	923.2	104.8	14.07	9.9	913.3	618.9	294.4	478.0
9	27	933.8	105.2	14.13	9.9	923.9	625.9	298.0	474.1
10	30	944.4	105.7	14.20	9.9	934.5	632.8	301.7	470.5
11	33	955.0	106.1	14.25	9.9	945.1	639.8	305.3	466.7
12	36	835.4					646.7		

Table 20. Round Channel. Run No. 17; Inlet Pressure, 341.3 psia; $\dot{m}$, 235.1 lb/hr; Inlet Δt_{sup} , 131.4 F°; Power Setting, 6.079 kw.

x	L/D _e	t ₀	$\rho(x)$	$q''_R(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B}}{\text{hr}}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	780.9					560.7		
1	3	1083.5	111.3	23.26	15.1	1068.4	573.9	494.5	494.4
2	6	1101.3	112.1	23.43	15.1	1086.2	587.1	499.1	469.4
3	9	1120.2	112.8	23.58	15.1	1105.1	600.3	504.8	467.1
4	12	1139.0	113.5	23.72	15.1	1123.9	613.5	510.4	464.7
5	15	1157.9	114.3	23.89	15.1	1142.8	626.7	516.1	462.9
6	18	1176.7	115.0	24.04	15.1	1161.6	640.0	521.6	460.8
7	21	1195.5	115.7	24.18	15.1	1180.4	653.2	527.2	458.6
8	24	1214.4	116.5	24.35	15.1	1199.3	666.4	532.9	456.9
9	27	1233.2	117.2	24.49	15.1	1218.1	679.6	538.5	454.8
10	30	1252.1	117.9	24.64	15.1	1237.0	692.8	544.2	452.8
11	33	1272.0	118.7	24.81	15.1	1257.0	706.0	551.0	450.3
12	36	964.7					719.2		

Table 21. Round Channel. Run No. 18; Inlet Pressure, 334.4 psia; $\dot{m}$, 168.2 lb/hr; Inlet Δt_{sup} , 115.0 F°; Power Setting, 1.581 kw.

x	L/D _e	t ₀	$\rho(x)$	$q''_R(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B}}{\text{hr}}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	621.9					542.4		
1	3	737.3	97.8	6.615	4.9	732.4	546.9	185.5	356.6
2	6	743.5	98.0	6.629	4.9	738.6	551.4	187.2	354.0
3	9	749.7	98.3	6.649	4.9	744.8	555.9	188.9	352.0
4	12	755.9	98.5	6.662	4.9	751.0	560.4	190.6	349.6
5	15	762.1	98.8	6.683	4.9	757.2	564.9	192.3	347.5
6	18	768.2	99.0	6.696	4.9	763.3	569.4	193.9	345.4
7	21	774.4	99.2	6.710	4.9	769.5	573.9	195.6	343.0
8	24	780.6	99.5	6.730	4.9	775.7	578.2	197.5	340.8
9	27	786.8	99.7	6.744	4.9	781.9	582.7	199.2	338.5
10	30	793.0	100.0	6.764	5.0	788.0	587.2	200.8	336.9
11	33	799.4	100.2	6.778	5.0	794.4	591.7	202.7	334.4
12	36	719.8					596.2		

Table 22. Round Channel. Run No. 19; Inlet Pressure, 322.4 psia; $\dot{m}$, 152.2 lb/hr; Inlet Δt_{sup} , 115.8 F°; Power Setting, 3.126 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	693.3					539.8		
1	3	925.3	105.2	12.04	8.4	916.9	549.4	367.5	327.5
2	6	939.5	105.8	12.12	8.4	931.1	559.0	372.1	325.8
3	9	953.7	106.3	12.46	8.4	945.3	568.7	376.6	330.9
4	12	967.9	106.9	12.23	8.4	959.5	578.3	381.2	320.8
5	15	982.1	107.4	12.29	8.4	973.7	587.9	385.8	318.5
6	18	996.3	108.0	12.36	8.4	987.9	597.6	390.3	316.6
7	21	1010.5	108.6	12.42	8.4	1002.1	607.2	394.9	314.6
8	24	1024.7	109.1	12.48	8.4	1016.3	616.8	399.5	312.4
9	27	1038.9	109.7	12.55	8.4	1030.5	626.5	404.0	310.6
10	30	1053.1	110.4	12.63	8.4	1044.7	636.1	408.6	309.1
11	33	1067.3	111.0	12.70	8.4	1058.9	645.7	413.2	307.3
12	36	905.0					655.4		

Table 23. Round Channel. Run No. 20; Inlet Pressure, 311.4 psia; $\dot{m}$, 150.9 lb/hr; Inlet Δt_{sup} , 115.6 F°; Power Setting, 4.370 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	747.9					536.4		
1	3	1075.0	110.8	15.69	10.4	1064.6	550.6	514.0	305.2
2	6	1094.5	111.6	15.80	10.4	1084.1	564.8	519.3	304.3
3	9	1114.1	112.3	15.90	10.4	1103.7	579.0	524.7	303.1
4	12	1133.6	113.1	16.01	10.4	1123.2	593.2	530.0	302.2
5	15	1153.2	113.8	16.11	10.4	1142.8	607.4	535.4	301.0
6	18	1172.8	114.6	16.23	10.4	1162.4	621.5	540.8	300.1
7	21	1192.4	115.4	16.34	10.4	1182.0	635.8	546.2	299.2
8	24	1212.0	116.1	16.44	10.4	1201.6	650.0	551.6	298.0
9	27	1231.5	116.9	16.55	10.4	1221.1	664.2	556.9	297.2
10	30	1251.0	117.6	16.65	10.4	1240.6	678.4	562.2	296.2
11	33	1270.5	118.4	16.77	10.4	1260.1	692.6	567.5	295.4
12	36	1051.1					706.7		

Table 24. Round Channel. Run No. 21; Inlet Pressure, 317.4 psia; $\dot{m}$, 90.8 lb/hr; Inlet Δt_{sup} , 78.9 F°; Power Setting, 1.616 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	611.1					501.5		
1	3	792.5	100.0	5.750	4.8	787.7	510.2	277.5	207.2
2	6	812.0	100.8	5.796	4.8	807.2	518.9	288.3	201.0
3	9	827.0	101.4	5.830	4.8	822.2	527.6	294.6	197.9
4	12	839.0	101.8	5.854	4.8	834.2	536.3	297.9	196.5
5	15	850.5	102.2	5.876	4.8	845.7	545.0	300.7	195.4
6	18	861.5	102.6	5.900	4.8	856.7	553.7	303.0	194.7
7	21	873.0	103.0	5.922	4.8	868.2	562.4	305.8	193.7
8	24	885.0	103.4	5.946	4.8	880.2	571.1	309.1	192.3
9	27	896.7	103.8	5.968	4.8	891.9	579.8	312.1	191.2
10	30	908.5	104.2	5.992	4.8	903.7	588.5	315.2	190.1
11	33	917.5	104.6	6.014	4.8	912.7	597.2	315.5	190.6
12	36	767.9					605.8		

Table 25. Round Channel. Run No. 22; Inlet Pressure, 315.4 psia; $\dot{m}$, 87.41 lb/hr; Inlet Δt_{sup} , 78.0 F°; Power Setting, 2.749 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	695.3					500.0		
1	3	999.0	107.5	10.29	7.1	991.9	515.5	476.4	216.1
2	6	1035.3	108.9	10.43	7.1	1028.2	531.0	497.2	209.7
3	9	1055.6	109.7	10.50	7.1	1048.5	546.4	502.1	209.2
4	12	1075.9	110.4	10.57	7.1	1068.8	561.9	506.9	208.5
5	15	1096.2	111.2	10.65	7.1	1089.1	577.4	511.7	208.1
6	18	1116.5	112.0	10.72	7.1	1109.4	592.9	516.5	207.6
7	21	1136.8	112.8	10.80	7.1	1129.7	608.3	521.4	207.1
8	24	1157.1	113.6	10.88	7.1	1150.0	623.8	526.2	206.7
9	27	1177.4	114.3	10.94	7.1	1170.3	639.3	531.0	206.1
10	30	1197.8	115.1	11.02	7.1	1190.7	654.8	535.9	205.6
11	33	1214.2					670.2		
12	36	960.4					685.7		

Table 26. Round Channel. Run No. 23; Inlet Pressure, 929.4 psia; $\dot{m}$, 383.0 lb/hr; Inlet Δt_{sup} , 122.8 F°; Power Setting, 1.903 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	702.1					658.6		
1	3	758.7	98.7	7.402	5.4	753.3	660.7	92.6	799.3
2	6	762.1	98.8	7.409	5.4	756.7	662.8	93.9	789.0
3	9	764.7	98.9	7.416	5.4	759.3	664.9	94.4	785.6
4	12	767.3	99.0	7.424	5.5	761.8	667.0	94.8	783.1
5	15	769.9	99.1	7.432	5.5	764.4	669.1	95.3	779.8
6	18	772.5	99.2	7.439	5.5	767.0	671.2	95.8	776.5
7	21	775.1	99.3	7.446	5.5	769.6	673.3	96.3	773.3
8	24	777.8	99.4	7.454	5.5	772.3	675.4	96.9	769.2
9	27	780.2	99.5	7.462	5.5	774.7	677.5	97.2	767.6
10	30	782.0	99.6	7.469	5.5	776.5	679.5	96.9	770.8
11	33	781.8					681.7		
12	36	742.0					683.7		

Table 27. Round Channel. Run No. 24; Inlet Pressure, 942.4 psia; $\dot{m}$, 380.9 lb/hr; Inlet Δt_{sup} , 126.0 F°; Power Setting, 4.102 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	749.9					663.4		
1	3	860.7	100.1	14.36	10.2	850.5	667.9	182.6	786.6
2	6	869.8	100.5	14.42	10.2	859.6	672.3	187.3	769.0
3	9	875.7	100.7	14.45	10.2	865.5	676.8	188.7	765.8
4	12	881.7	101.0	14.49	10.2	871.5	681.2	190.3	761.6
5	15	887.7	101.2	14.52	10.2	877.5	685.7	191.8	757.2
6	18	893.4	101.4	14.55	10.2	883.2	690.1	193.1	753.5
7	21	899.4	101.6	14.58	10.2	889.2	694.6	194.6	749.2
8	24	905.3	101.8	14.61	10.2	895.1	699.0	196.1	744.9
9	27	910.0	102.1	14.65	10.3	899.7	703.5	196.2	746.8
10	30	912.7	102.3	14.68	10.3	902.4	707.9	194.5	754.8
11	33	914.0					712.4		
12	36	834.9					717.0		

Table 28. Round Channel. Run No. 25; Inlet Pressure, 904.4 psia; $\dot{m}$, 379.1 lb/hr; Inlet Δt_{sup} , 129.0 F°; Power Setting, 7.005 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_R''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	817.3					661.6		
1	3	1031.0	109.3	27.63	16.5	1014.5	669.6	344.9	801.1
2	6	1049.0	110.0	27.81	16.5	1032.5	677.6	354.9	783.5
3	9	1062.0	110.5	27.93	16.5	1045.5	685.7	359.8	776.4
4	12	1075.0	111.0	28.06	16.5	1058.5	693.7	364.8	769.2
5	15	1088.5	111.5	28.19	16.5	1072.0	701.7	370.3	761.2
6	18	1101.0	112.0	28.31	16.5	1084.5	709.8	374.7	755.6
7	21	1114.5	112.5	28.44	16.5	1098.0	717.8	380.2	748.0
8	24	1127.5	113.0	28.57	16.5	1111.0	725.8	385.2	741.6
9	27	1137.0	113.4	28.67	16.5	1120.5	733.9	386.6	741.5
10	30	1143.0	113.6	28.72	16.5	1126.5	741.9	384.6	746.7
11	33	1145.0	113.7	38.74	16.5	1128.5	749.9	378.6	759.2
12	36	997.4					758.0		

Table 29. Round Channel. Run No. 26; Inlet Pressure, 1024.4 psia; $\dot{m}$, 286.1 lb/hr; Inlet Δt_{sup} , 104.5 F°; Power Setting, 1.663 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_R''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	700.3					652.1		
1	3	763.1	98.83	6.962	4.9	758.2	654.3	103.9	670.0
2	6	768.4	99.04	6.976	4.9	763.5	656.5	107.0	652.0
3	9	772.0	99.18	6.986	4.9	767.1	658.7	108.4	644.5
4	12	775.6	99.32	6.996	5.0	770.6	660.9	109.7	637.7
5	15	779.2	99.46	7.006	5.0	774.2	663.1	111.1	630.6
6	18	782.8	99.60	7.016	5.0	777.8	665.3	112.5	623.6
7	21	786.4	99.74	7.026	5.0	781.4	667.5	113.9	616.8
8	24	790.0	99.88	7.036	5.0	785.0	669.7	115.3	610.2
9	27	793.6	100.02	7.045	5.0	788.6	671.9	116.7	603.7
10	30	797.2	100.16	7.055	5.0	792.2	674.1	118.1	597.4
11	33	800.8	100.30	7.065	5.0	795.8	676.3	119.5	591.2
12	36	748.6					678.4		

Table 30. Round Channel. Run No. 27; Inlet Pressure, 1019.4 psia; $\dot{m}$, 284.0 lb/hr; Inlet Δt_{sup} , 104.6 F°; Power Setting, 3.704 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	759.0					651.0		
1	3	906.5	104.01	15.23	9.3	897.2	656.4	240.8	632.4
2	6	919.5	104.52	15.30	9.3	910.2	661.8	248.4	616.0
3	9	929.3	104.89	15.36	9.3	920.0	667.1	252.9	607.2
4	12	939.0	105.26	15.41	9.3	929.7	672.5	257.2	599.1
5	15	948.5	105.63	15.46	9.3	939.2	677.9	261.3	591.8
6	18	957.7	106.00	15.52	9.3	948.4	683.2	265.2	585.2
7	21	967.3	106.37	15.57	9.3	958.0	688.6	269.4	578.0
8	24	977.0	106.74	15.63	9.3	967.7	694.0	273.7	570.9
9	27	985.7	107.08	15.68	9.3	976.4	699.3	277.1	565.7
10	30	993.0	107.36	15.72	9.3	983.7	704.7	279.0	563.4
11	33	997.3	107.53	15.74	9.3	988.0	710.1	277.9	566.5
12	36	881.9					715.4		

Table 31. Round Channel. Run No. 28; Inlet Pressure, 1008.4 psia; $\dot{m}$, 282.0 lb/hr; Inlet Δt_{sup} , 106.5 F°; Power Setting, 6.083 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	832.6					652.2		
1	3	1092.5	111.7	23.26	15.3	1077.2	661.2	416.0	559.0
2	6	1113.8	112.5	23.42	15.3	1098.5	670.2	428.3	546.8
3	9	1129.7	113.1	23.55	15.3	1114.4	679.2	435.2	541.1
4	12	1145.6	113.7	23.67	15.4	1130.2	688.3	441.9	535.7
5	15	1161.5	114.3	23.80	15.4	1146.1	697.3	448.8	530.2
6	18	1177.4	114.9	23.92	15.4	1162.0	706.3	455.7	527.7
7	21	1193.3	115.5	24.05	15.4	1177.9	715.3	462.6	519.9
8	24	1209.3	116.1	24.17	15.4	1193.9	724.4	469.5	514.8
9	27	1225.2	116.7	24.30	15.4	1209.8	733.4	476.4	510.0
10	30	1239.8	117.3	24.42	15.4	1224.4	742.4	482.0	506.7
11	33	1247.8	117.6	24.48	15.4	1232.4	751.4	481.0	509.0
12	36	1049.7					760.5		

Table 32. Round Channel. Run No. 29; Inlet Pressure, 1458.4 psia; $\dot{m}$, 477.4 lb/hr; Inlet Δt_{sup} , 85.6 F°; Power Setting, 3.141 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	737.6					678.2		
1	3	809.7	100.65	12.26	8.4	801.3	680.6	120.7	1015.7
2	6	813.7	100.80	12.28	8.4	805.3	683.0	122.3	1003.9
3	9	817.7	100.95	12.30	8.4	809.3	685.4	123.9	992.4
4	12	821.7	101.10	12.31	8.4	813.3	687.9	125.4	982.0
5	15	825.7	101.25	12.33	8.4	817.3	690.3	127.0	971.0
6	18	829.7	101.40	12.35	8.4	821.3	692.7	128.6	960.4
7	21	833.7	101.55	12.37	8.4	825.3	695.2	130.1	950.7
8	24	837.7	101.70	12.39	8.4	829.3	697.6	131.7	940.6
9	27	841.7	101.85	12.40	8.4	833.3	700.0	133.3	930.6
10	30	845.7	102.00	12.42	8.4	837.3	702.4	134.9	920.9
11	33	849.7	102.15	12.44	8.4	841.3	704.8	136.5	911.5
12	36	794.3					707.2		

Table 33. Round Channel. Run No. 30; Inlet Pressure, 1452.4 psia; $\dot{m}$, 476.6 lb/hr; Inlet Δt_{sup} , 86.3 F°; Power Setting, 6.027 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	796.4					678.2		
1	3	947.0	106.00	23.78	16.5	930.5	682.6	247.9	959.2
2	6	959.0	106.34	23.85	16.5	942.5	687.0	255.5	933.5
3	9	967.7	106.68	23.93	16.5	951.2	691.4	259.8	921.0
4	12	976.5	107.02	24.00	16.5	960.0	695.8	264.2	908.6
5	15	985.2	107.36	24.08	16.5	968.7	700.2	268.5	896.9
6	18	994.0	107.70	24.16	16.5	977.5	704.7	272.8	885.5
7	21	1002.7	108.14	24.26	16.5	986.2	709.1	277.1	875.3
8	24	1011.5	108.48	24.33	16.5	995.0	713.5	281.5	864.4
9	27	1020.2	108.82	24.41	16.5	1003.7	717.9	285.8	854.0
10	30	1029.0	109.16	24.48	16.5	1012.5	722.3	290.2	843.7
11	33	1037.7	109.50	24.56	16.5	1021.2	726.7	294.5	834.0
12	36	925.7					731.2		

Table 34. Round Channel. Run No. 31; Inlet Pressure, 1417.4 psia; $\dot{m}$, 272.0 lb/hr; Inlet Δt_{sup} , 73.1 F°; Power Setting, 3.219 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B}}{\text{hr}}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\frac{\text{B}}{\text{hr}}\text{ft}^{-2}\text{F}^{-1}$
0	0	743.3					661.7		
1	3	874.0	103.12	12.55	8.9	865.1	665.7	199.4	629.4
2	6	883.3	103.48	12.59	8.9	874.4	669.7	204.7	615.2
3	9	890.5	103.76	12.63	8.9	881.6	673.6	208.0	607.1
4	12	897.6	104.04	12.66	8.9	888.7	677.6	211.1	599.8
5	15	904.8	104.32	12.70	8.9	895.9	681.5	214.4	592.2
6	18	911.9	104.60	12.73	8.9	903.0	685.5	217.5	585.3
7	21	919.1	104.88	12.76	8.9	910.2	689.4	220.8	578.0
8	24	926.2	105.16	12.80	8.9	917.3	693.4	223.9	571.6
9	27	933.4	105.44	12.83	8.9	924.5	697.3	227.2	564.7
10	30	940.5	105.72	12.87	8.9	931.6	701.3	230.3	558.7
11	33	947.7	106.00	12.90	8.9	938.8	705.2	233.6	552.2
12	36	845.8					709.2		

Table 35. Round Channel. Run No. 32; Inlet Pressure, 1404.4 psia; $\dot{m}$, 283.5 lb/hr; Inlet Δt_{sup} , 85.7 F°; Power Setting, 5.985 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_r''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B}}{\text{hr}}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\frac{\text{B}}{\text{hr}}\text{ft}^{-2}\text{F}^{-1}$
0	0	829.9					673.2		
1	3	1083.0	111.09	22.93	15.2	1067.8	680.8	387.0	592.5
2	6	1110.0	112.14	23.14	15.2	1094.8	688.4	406.4	569.5
3	9	1124.8	112.72	23.26	15.2	1109.6	696.1	413.5	562.6
4	12	1139.6	113.30	23.38	15.2	1124.4	703.7	420.7	555.9
5	15	1154.4	113.88	23.50	15.2	1139.3	711.3	428.0	549.2
6	18	1169.2	114.40	23.61	15.2	1154.0	719.0	435.0	542.8
7	21	1184.0	114.98	23.73	15.2	1168.8	726.6	442.2	536.7
8	24	1198.9	115.56	23.85	15.2	1183.7	734.2	449.5	530.6
9	27	1213.7	116.14	23.97	15.2	1198.5	741.9	456.6	525.0
10	30	1228.5	116.72	24.09	15.2	1213.3	739.5	463.8	519.4
11	33	1243.3	117.30	24.21	15.2	1228.1	757.1	471.0	514.0
12	36	1041.4					764.8		

Table 36. Rectangular Channel. Run No. 1; Inlet Pressure, 325.8 psia;
 $\dot{m}$, 129.7 lb/hr; Inlet Δt_{sup} , 153.3 F°; Power Setting, 1.644 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	617.6					577.3		
1	11.0	650.4	94.6	2.568	1.5	648.9	581.5	67.4	381.0
2	22.1	635.1	94.0	2.552	1.5	633.6	585.8	47.8	533.8
3	33.1	628.6	93.7	2.544	1.5	627.1	590.1	37.0	687.4
4	44.2	628.0	93.7	2.544	1.5	626.5	594.6	31.9	797.3
5	55.2	635.8	93.9	2.549	1.5	634.3	598.8	35.5	718.0
6	66.2	651.0	94.7	2.571	1.5	649.5	603.2	46.3	555.2
7	77.3	661.1	95.0	2.579	1.5	659.6	607.6	52.0	495.9
8	88.3	668.8	95.3	2.587	1.5	667.3	612.0	55.3	467.8
9	99.4	675.1	95.5	2.592	1.5	673.6	616.7	56.9	455.6
10	110.4	680.7	95.7	2.598	1.5	679.2	621.2	58.0	447.9
11	121.4	685.5	95.8	2.600	1.5	684.0	625.5	58.5	444.5
12	132.5	689.0					630.0		

Table 37. Rectangular Channel. Run No. 2; Inlet Pressure, 323.8 psia;
 $\dot{m}$, 127.3 lb/hr; Inlet Δt_{sup} , 158.0 F°; Power Setting, 2.951 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	663.5					582.2		
1	11.0	754.2	98.6	4.598	2.6	751.6	593.3	158.3	290.4
2	22.1	736.5	97.8	4.561	2.6	733.9	604.5	129.4	352.4
3	33.1	728.9	97.6	4.551	2.6	726.3	615.6	110.7	411.1
4	44.2	736.5	97.9	4.565	2.6	733.9	626.7	107.2	425.9
5	55.2	750.1	98.5	4.593	2.6	747.6	637.8	109.8	418.3
6	66.2	763.2	99.0	4.617	2.6	760.6	649.0	111.6	413.7
7	77.3	775.1	99.5	4.640	2.6	772.5	660.1	112.4	412.8
8	88.3	786.3	99.8	4.654	2.6	783.7	671.3	112.4	414.0
9	99.4	796.7	100.1	4.668	2.6	794.1	682.4	111.7	417.9
10	110.4	806.2	100.3	4.677	2.6	803.6	693.5	110.1	424.8
11	121.4	815.0	100.5	4.687	2.6	812.4	704.7	107.7	435.2
12	132.5	813.5					715.8		

Table 38. Rectangular Channel. Run No. 3; Inlet Pressure, 320.8 psia;
 $\dot{m}$, 124.7 lb/hr; Inlet Δt_{sup} , 157.4 F°; Power Setting, 4.682 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	712.1					580.9		
1	11.0	852.2	102.5	7.389	4.0	848.2	600.0	248.2	297.7
2	22.1	822.8	101.2	7.296	4.0	818.8	619.1	199.7	365.3
3	33.1	819.2	101.1	7.288	4.0	815.2	638.2	177.0	411.8
4	44.2	829.8	101.5	7.317	4.0	825.8	657.3	168.5	428.3
5	55.2	855.0	102.5	7.389	4.0	851.0	676.4	174.6	423.2
6	66.2	877.5	103.4	7.454	4.0	873.5	695.4	178.1	418.5
7	77.3	896.2	104.1	7.505	4.0	892.2	714.5	177.7	422.3
8	88.3	913.2	104.8	7.555	4.0	909.2	733.6	175.6	430.2
9	99.4	928.5	105.4	7.598	4.0	924.5	752.7	171.8	442.3
10	110.4	944.0	106.0	7.642	4.0	940.0	771.8	168.2	454.3
11	121.4	959.0	106.6	7.685	4.0	955.0	790.9	164.1	468.3
12	132.5	974.2					810.0		

Table 39. Rectangular Channel. Run No. 4; Inlet Pressure, 344.8 psia;
 $\dot{m}$, 88.6 lb/hr; Inlet Δt_{sup} , 121.2 F°; Power Setting, 1.624 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	614.7					551.4		
1	11.0	681.6	95.7	2.534	1.4	680.2	559.9	120.3	210.6
2	22.1	667.4	95.2	2.521	1.4	666.0	568.4	89.1	282.9
3	33.1	662.0	95.0	2.516	1.4	660.6	576.9	83.7	300.5
4	44.2	663.8	95.1	2.518	1.4	662.4	585.4	77.0	327.0
5	55.2	674.5	95.5	2.529	1.4	673.1	593.9	79.2	319.3
6	66.2	686.7	95.8	2.537	1.4	685.3	602.4	82.9	306.0
7	77.3	696.4	96.4	2.553	1.4	695.0	610.9	84.1	303.5
8	88.3	704.7	96.6	2.558	1.4	703.4	619.4	84.0	304.5
9	99.4	711.4	96.9	2.566	1.4	710.0	627.9	82.1	312.5
10	110.4	717.2	97.1	2.571	1.4	715.8	636.4	79.4	323.8
11	121.4	722.0	97.3	2.576	1.4	720.6	644.9	75.7	340.3
12	132.5	726.0					653.4		

Table 40. Rectangular Channel. Run No. 5; Inlet Pressure, 345.8 psia;
 $\dot{m}$, 87.0 lb/hr; Inlet Δt_{sup} , 118.4 F°; Power Setting 3.196 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	674.2					548.9		
1	11.0	803.8	100.2	4.973	2.7	801.1	567.0	234.1	212.4
2	22.1	784.8	99.8	4.950	2.7	782.1	585.1	197.0	251.3
3	33.1	777.2	99.5	4.935	2.7	774.5	603.2	171.3	288.1
4	44.2	780.5	99.7	4.945	2.7	777.8	621.3	156.5	316.0
5	55.2	799.5	100.3	4.975	2.7	796.8	639.4	157.4	316.0
6	66.2	820.0	101.1	5.015	2.7	817.3	657.4	159.9	313.6
7	77.3	838.0	101.9	5.054	2.7	835.3	675.5	159.8	316.3
8	88.3	854.0	102.5	5.084	2.7	851.3	693.6	157.7	322.4
9	99.4	869.0	103.1	5.114	2.7	866.3	711.7	154.6	330.8
10	110.4	882.5	103.6	5.139	2.7	879.8	729.8	150.0	342.6
11	121.4	895.8	104.1	5.164	2.7	893.1	747.9	145.2	355.6
12	132.5	907.7					766.0		

Table 41. Rectangular Channel. Run No. 6; Inlet Pressure, 345.8 psia;
 $\dot{m}$, 85.5 lb/hr; Inlet Δt_{sup} , 118.3 F°; Power Setting 4.947 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	737.1					548.8		
1	11.0	938.0	105.7	7.832	4.0	934.0	577.6	356.4	219.8
2	22.1	907.0	104.6	7.751	4.0	903.0	606.4	296.6	261.3
3	33.1	897.0	104.2	7.721	4.0	893.0	635.2	257.8	299.5
4	44.2	906.0	104.5	7.743	4.0	902.0	664.0	238.0	325.4
5	55.2	941.5	105.8	7.840	4.0	937.5	692.8	244.7	320.4
6	66.2	974.0	107.2	7.944	4.0	970.0	721.6	248.4	319.8
7	77.3	998.5	108.1	8.010	4.0	994.5	750.4	244.1	328.2
8	88.3	1022.0	109.0	8.077	4.0	1018.0	779.3	238.7	338.8
9	99.4	1044.0	109.9	8.144	4.0	1040.0	808.1	231.9	351.2
10	110.4	1065.0	110.3	8.173	4.0	1061.0	836.9	214.1	381.7
11	121.4	1085.0	111.5	8.262	4.0	1081.0	865.7	215.3	383.7
12	132.5	1106.2					894.5		

Table 42. Rectangular Channel. Run No. 7; Inlet Pressure, 603.4 psia;
 $\dot{m}$, 153.8 lb/hr; Inlet Δt_{sup} , 5.6 F°; Power Setting, 1.462 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	519.0					492.4		
1	11.0	527.8	89.8	2.293	1.4	526.4	494.9	31.5	727.9
2	22.1	527.2	89.8	2.293	1.4	525.8	497.4	28.4	807.4
3	33.1	525.1	89.7	2.290	1.4	523.7	499.9	23.8	962.2
4	44.2	537.7	90.1	2.300	1.4	536.3	502.3	34.0	676.5
5	55.2	544.4	90.4	2.308	1.4	543.4	504.8	38.6	597.9
6	66.2	548.8	90.6	2.313	1.4	547.4	507.3	40.1	576.8
7	77.3	552.2	90.7	2.316	1.4	550.8	509.8	41.0	564.8
8	88.3	554.8	90.8	2.318	1.4	553.4	512.3	41.1	564.0
9	99.4	556.8	90.9	2.321	1.4	555.4	514.8	40.6	571.7
10	110.4	558.5	91.0	2.323	1.4	557.1	517.3	39.8	583.7
11	121.4	559.8	91.1	2.326	1.4	558.4	519.7	38.7	601.0
12	132.5	558.7					522.1		

Table 43. Rectangular Channel. Run No. 8; Inlet Pressure, 604.4 psia;
 $\dot{m}$, 142.1 lb/hr; Inlet Δt_{sup} , 5.0 F°; Power Setting, 3.236 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	573.8					492.1		
1	11.0	602.0	92.6	5.044	2.9	599.1	500.2	98.9	510.0
2	22.1	596.8	92.5	5.038	2.9	593.9	508.4	85.5	589.2
3	33.1	586.5	92.1	5.017	2.9	583.6	516.5	77.1	650.7
4	44.2	621.6	93.4	5.088	2.9	618.7	524.7	94.0	541.3
5	55.2	637.7	94.0	5.120	2.9	634.8	532.8	102.0	502.0
6	66.2	649.8	94.5	5.147	2.9	646.9	541.0	105.9	486.0
7	77.3	659.5	94.7	5.158	2.9	656.6	549.1	107.5	479.8
8	88.3	667.8	95.1	5.180	2.9	664.9	557.3	107.6	481.4
9	99.4	675.2	95.5	5.202	2.9	672.3	565.4	106.9	486.6
10	110.4	682.2	95.8	5.218	2.9	679.3	573.6	105.7	493.7
11	121.4	688.9	96.0	5.229	2.9	686.0	581.7	104.3	501.3
12	132.5	685.0					589.9		

Table 44. Rectangular Channel. Run No. 9; Inlet Pressure, 604.4 psia;
 m, 141.6 lb/hr; Inlet Δt_{sup} , 9.2 F°; Power Setting 5.138 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	632.5					496.3		
1	11.0	677.3	95.6	8.004	4.5	672.8	510.8	162.0	494.1
2	22.1	672.5	95.4	7.988	4.5	668.0	525.3	142.7	559.8
3	33.1	651.0	94.6	7.921	4.5	646.5	539.8	106.7	742.4
4	44.2	708.5	96.8	8.105	4.5	704.0	554.3	149.7	541.4
5	55.2	739.3	98.0	8.205	4.5	734.8	568.8	166.0	494.3
6	66.2	762.3	98.9	8.281	4.5	757.8	583.3	174.5	474.6
7	77.3	780.7	99.6	8.339	4.5	776.2	597.8	178.4	467.4
8	88.3	796.5	100.2	8.390	4.5	792.0	612.3	179.7	466.9
9	99.4	810.7	100.8	8.440	4.5	806.2	686.8	179.4	470.5
10	110.4	824.0	101.3	8.482	4.5	819.5	641.3	178.2	476.0
11	121.4	836.7	101.7	8.515	4.5	832.2	655.8	176.4	482.7
12	132.5	824.5					670.4		

Table 45. Rectangular Channel. Run No. 10; Inlet Pressure, 592.4 psia;
 m, 199.3 lb/hr; Inlet Δt_{sup} , 24.2 F°; Power Setting 1.634 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	536.9					509.0		
1	11.0	548.8	90.6	3.577	2.1	546.7	510.7	36.0	993.6
2	22.1	547.0	90.6	3.577	2.1	544.9	512.5	32.4	1104.0
3	33.1	541.0	90.4	3.569	2.1	538.9	514.2	34.7	1028.5
4	44.2	556.1	91.0	3.593	2.1	554.0	516.0	38.0	945.5
5	55.2	562.5	91.2	3.600	2.1	560.4	517.7	42.7	843.1
6	66.2	567.1	91.4	3.608	2.1	565.0	519.5	45.5	793.0
7	77.3	570.6	91.5	3.612	2.1	568.5	521.2	47.3	763.6
8	88.3	573.4	91.6	3.616	2.1	571.3	523.0	48.3	748.7
9	99.4	575.7	91.7	3.620	2.1	573.6	524.7	48.9	740.3
10	110.4	577.7	91.8	3.624	2.1	575.6	526.5	49.1	738.1
11	121.4	579.4	91.8	3.624	2.1	577.3	528.2	49.1	738.1
12	132.5	574.8					530.0		

Table 46. Rectangular Channel. Run No. 11; Inlet Pressure, 593.4 psia;
 $\dot{m}$, 193.5 lb/hr; Inlet Δt_{sup} , 21.3 F°; Power Setting, 3.510 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	575.0					506.3		
1	11.0	596.7	92.5	5.487	3.2	593.5	512.2	81.3	674.9
2	22.1	591.4	92.3	5.476	3.2	588.2	518.1	70.1	781.2
3	33.1	576.7	91.8	5.446	3.2	573.5	524.0	49.5	1100.2
4	44.2	612.7	93.1	5.523	3.2	609.5	529.8	79.7	692.9
5	55.2	626.6	93.6	5.553	3.2	623.4	535.7	87.7	633.2
6	66.2	636.0	94.0	5.576	3.2	632.8	541.6	91.2	611.4
7	77.3	643.3	94.2	5.588	3.2	640.1	547.5	92.6	603.5
8	88.3	648.8	94.5	5.606	3.2	645.6	553.4	92.2	608.0
9	99.4	653.4	94.6	5.612	3.2	650.2	559.3	90.9	617.4
10	110.4	657.1	94.8	5.624	3.2	653.9	565.1	88.8	633.3
11	121.4	660.4	9.5	5.636	3.2	657.2	571.0	86.2	653.8
12	132.5	658.2					576.9		

Table 47. Rectangular Channel. Run No. 12; Inlet Pressure, 594.4 psia;
 $\dot{m}$, 190.9 lb/hr; Inlet Δt_{sup} , 20.0 F°; Power Setting, 5.243 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	619.6					505.2		
1	11.0	654.0	93.9	8.141	4.6	649.4	515.4	134.0	607.5
2	22.1	647.3	94.4	8.184	4.6	642.7	525.6	117.1	698.9
3	33.1	621.0	93.4	8.098	4.6	616.4	535.8	80.6	1004.7
4	44.2	672.0	95.4	8.271	4.6	667.4	546.0	121.4	681.3
5	55.2	699.2	96.5	8.366	4.6	694.6	556.2	138.4	604.5
6	66.2	715.4	97.1	8.418	4.6	710.8	566.4	144.4	583.0
7	77.3	727.5	97.6	8.462	4.6	722.9	576.6	146.3	578.4
8	88.3	737.0	98.0	8.497	4.6	732.4	586.8	145.6	583.6
9	99.4	745.0	98.2	8.514	4.6	740.4	597.0	143.4	593.7
10	110.4	751.8	98.5	8.540	4.6	747.2	607.2	140.0	610.0
11	121.4	758.3	98.8	8.566	4.7	753.7	617.4	136.3	628.5
12	132.5	755.7					627.5		

Table 48. Rectangular Channel. Run No. 13; Inlet Pressure, 598.4 psia;
 $\dot{m}$, 126.8 lb/hr; Inlet Δt_{sup} , 21.7 F°; Power Setting, 1.616 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	561.4					517.6		
1	11.0	575.4	91.7	2.586	1.5	573.9	521.6	52.3	494.5
2	22.1	575.3	91.7	2.586	1.5	573.8	525.6	48.2	536.5
3	33.1	563.6	91.2	2.571	1.5	562.1	529.6	32.5	791.1
4	44.2	580.7	91.9	2.591	1.5	579.2	533.5	45.7	567.0
5	55.2	590.6	92.2	2.600	1.5	589.1	537.5	51.6	503.9
6	66.2	596.7	92.5	2.608	1.5	595.2	541.5	53.7	485.7
7	77.3	601.5	92.7	2.614	1.5	600.0	545.5	54.5	479.6
8	88.3	605.3	92.8	2.617	1.5	603.8	549.5	54.3	482.0
9	99.4	608.3	92.9	2.619	1.5	606.8	553.4	53.4	490.4
10	110.4	610.7	93.0	2.622	1.5	609.2	557.4	51.8	506.2
11	121.4	612.7	93.1	2.625	1.5	611.2	561.4	49.8	527.1
12	132.5	611.0					565.4		

Table 49. Rectangular Channel. Run No. 14; Inlet Pressure, 600.4 psia;
 $\dot{m}$, 125.0 lb/hr; Inlet Δt_{sup} , 33.7 F°; Power Setting, 3.444 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	630.6					520.0		
1	11.0	666.6	95.1	5.362	3.0	663.6	530.9	132.7	404.0
2	22.1	662.7	95.0	5.357	3.0	659.7	541.8	117.9	454.3
3	33.1	636.5	94.0	5.300	3.0	633.5	552.7	80.8	655.9
4	44.2	669.5	95.3	5.374	3.0	666.5	563.6	102.9	522.2
5	55.2	692.9	96.2	5.424	3.0	689.9	574.6	115.3	470.4
6	66.2	711.0	96.9	5.464	3.0	708.0	585.5	122.5	446.0
7	77.3	725.4	97.5	5.498	3.0	722.4	596.4	126.0	436.3
8	88.3	737.3	98.0	5.526	3.0	734.3	607.3	127.0	435.1
9	99.4	747.6	98.4	5.548	3.0	744.6	618.2	126.4	438.9
10	110.4	756.9	98.8	5.571	3.0	753.9	629.1	124.8	446.4
11	121.4	765.9	99.1	5.588	3.0	762.9	640.0	122.9	454.7
12	132.5	761.0					651.0		

Table 50. Rectangular Channel. Run No. 15; Inlet Pressure, 602.4 psia;
 $\dot{m}$, 121.7 lb/hr; Inlet Δt_{sup} , 28.8 F°; Power Setting, 5.153 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	699.5					519.0		
1	11.0	757.0	98.7	8.052	4.4	752.6	537.0	215.6	373.5
2	22.1	755.5	98.6	8.044	4.4	751.1	555.0	196.1	410.2
3	33.1	714.0	97.0	7.913	4.4	709.6	573.0	136.6	579.3
4	44.2	754.5	98.6	8.044	4.4	750.1	591.0	151.0	532.7
5	55.2	802.5	100.5	8.199	4.4	798.1	609.0	188.1	435.9
6	66.2	832.5	101.6	8.289	4.4	828.1	627.0	201.1	412.2
7	77.3	855.7	102.5	8.362	4.4	851.3	645.0	206.3	405.3
8	88.3	872.5	103.2	8.419	4.4	868.1	663.0	205.1	410.5
9	99.4	887.0	103.8	8.468	4.4	882.6	681.0	201.6	420.0
10	110.4	900.3	104.3	8.509	4.4	895.9	699.0	196.9	432.1
11	121.4	912.4	104.8	8.55	4.4	908.	717.0	191.0	447.6
12	132.5	903.5					735.0		

Table 51. Rectangular Channel. Run No. 16; Inlet Pressure, 321.4 psia;
 $\dot{m}$, 83.7 lb/hr; Inlet Δt_{sup} , 66.8 F°; Power Setting, 1.599 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	558.3					490.5		
1	11.0	582.5	92.0	2.490	1.4	581.1	497.9	83.2	300.
2	22.1	583.1	92.0	2.490	1.4	581.7	505.3	76.4	325.9
3	33.1	567.6	91.4	2.474	1.4	566.2	512.6	53.6	461.6
4	44.2	591.5	92.3	2.498	1.4	590.1	520.0	70.1	356.3
5	55.2	611.2	93.0	2.517	1.4	609.8	527.4	82.4	305.5
6	66.2	622.8	93.5	2.531	1.4	621.4	534.7	86.7	291.9
7	77.3	632.5	93.8	2.539	1.4	631.1	542.1	89.0	285.3
8	88.3	638.4	94.1	2.547	1.4	637.0	549.5	87.5	405.4
9	99.4	644.0	94.3	2.552	1.4	642.6	556.9	85.7	297.8
10	110.4	648.7	94.5	2.558	1.4	647.3	564.2	83.1	307.8
11	121.4	652.7	94.6	2.561	1.4	651.3	571.5	79.8	320.9
12	132.5	647.6					578.8		

Table 52. Rectangular Channel. Run No., 17; Inlet Pressure, 321.4 psia;
 $\dot{m}$, 82.0 lb/hr; Inlet Δt_{sup} , 49.1 F°; Power Setting, 3.374 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	627.0					472.8		
1	11.0	683.5	95.7	5.335	3.0	680.5	491.3	189.2	282.0
2	22.1	685.0	95.9	5.346	3.0	682.0	509.8	172.2	310.4
3	33.1	653.0	94.6	5.273	3.0	650.0	528.3	121.7	433.3
4	44.2	689.0	96.0	5.351	3.0	686.0	546.9	139.1	384.7
5	55.2	738.5	98.0	5.463	3.0	735.5	565.4	170.1	321.2
6	66.2	766.7	99.1	5.524	3.0	763.7	583.9	179.8	307.2
7	77.3	787.5	99.9	5.567	3.0	784.5	602.4	182.1	305.7
8	88.3	804.3	100.5	5.602	3.0	801.3	620.9	180.4	310.5
9	99.4	818.7	101.1	5.636	3.0	815.7	639.5	176.2	319.9
10	110.4	831.0	101.5	5.658	3.0	828.0	658.0	170.0	332.8
11	121.4	842.5	102.0	5.686	3.0	839.5	676.5	163.0	348.8
12	132.5	831.0					695.0		

Table 53. Rectangular Channel. Run No. 18; Inlet Pressure, 322.4 psia;
 $\dot{m}$, 80.6 lb/hr; Inlet Δt_{sup} , 45.2 F°; Power Setting, 5.567 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	750.					469.2		
1	11.0	850.	102.3	8.666	4.6	845.4	503.5	341.9	253.5
2	22.1	857.	102.5	8.683	4.6	852.4	537.7	314.7	275.9
3	33.1	802.	100.5	8.514	4.6	797.4	572.0	225.4	377.7
4	44.2	847.	102.2	8.658	4.6	842.4	606.2	236.2	366.6
5	55.2	925.	105.2	8.912	4.6	920.4	640.5	279.9	318.4
6	66.2	977.	107.3	9.090	4.6	972.4	674.7	297.7	305.3
7	77.3	1018.	109.	9.234	4.6	1013.4	709.0	304.4	303.3
8	88.3	1050.	110.1	9.327	4.6	1045.4	743.2	302.2	308.6
9	99.4	1078.	111.2	9.420	4.6	1073.4	777.5	295.9	318.2
10	110.4	1105.	112.3	9.513	4.6	1100.4	811.7	288.7	329.5
11	121.4	1130.	113.3	9.598	4.6	1125.4	846.0	279.4	343.5
12	132.5	1106.					880.3		

Table 54. Rectangular Channel. Run No. 19; Inlet Pressure, 999.4 psia;
 $\dot{m}$, 153.2 lb/hr; Inlet Δt_{sup} , 46.3 F°; Power Setting, 1.703 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	621.4					590.8		
1	11.0	632.2	93.8	2.645	1.5	630.7	594.1	36.6	722.7
2	22.1	645.7	94.3	2.659	1.5	644.2	597.4	46.8	568.2
3	33.1	641.3	94.2	2.656	1.5	639.8	600.6	39.2	677.6
4	44.2	640.4	94.2	2.656	1.5	638.9	603.9	35.0	758.8
5	55.2	648.3	94.5	2.664	1.5	646.8	607.2	39.6	672.7
6	66.2	651.9	94.6	2.667	1.5	650.4	610.4	40.0	666.8
7	77.3	656.4	94.8	2.673	1.5	654.9	613.7	41.2	648.8
8	88.3	660.2	94.9	2.676	1.5	658.7	617.0	41.7	641.7
9	99.4	663.6	95.0	2.679	1.5	662.1	620.2	41.9	639.4
10	110.4	666.7	95.1	2.681	1.5	665.2	623.5	41.7	642.9
11	121.4	669.6	95.3	2.687	1.5	668.1	626.8	41.3	650.6
12	132.5	670.0					630.0		

Table 55. Rectangular Channel. Run No. 20; Inlet Pressure, 1001.4 psia;
 $\dot{m}$, 147.9 lb/hr; Inlet Δt_{sup} , 45.9 F°; Power Setting, 3.737 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	667.1					590.6		
1	11.0	699.0	96.4	5.782	3.2	695.8	599.6	96.2	601.0
2	22.1	720.7	97.4	5.843	3.2	717.5	608.6	108.9	536.5
3	33.1	709.6	96.8	5.807	3.2	706.4	617.7	88.7	654.7
4	44.2	717.4	97.2	5.831	3.2	714.2	626.7	87.5	666.4
5	55.2	736.7	97.9	5.873	3.2	733.5	635.7	97.8	600.5
6	66.2	751.7	98.5	5.909	3.2	748.5	644.8	103.7	569.8
7	77.3	764.2	99.0	5.939	3.2	761.0	653.8	107.2	554.0
8	88.3	775.1	99.4	5.963	3.2	771.9	662.8	109.1	546.6
9	99.4	784.6	99.8	5.987	3.2	781.4	671.9	109.5	546.8
10	110.4	793.3	100.1	6.005	3.2	790.1	680.9	109.2	549.9
11	121.4	801.2	100.4	6.023	3.2	798.0	689.9	108.1	557.2
12	132.5	802.7					699.0		

Table 56. Rectangular Channel. Run No. 21; Inlet Pressure, 1002.4 psia;
 $\dot{m}$, 144.4 lb/hr; Inlet Δt_{sup} , 45.4 F°; Power Setting, 5.608 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	706.0					590.2		
1	11.0	751.0	98.5	8.639	4.7	746.3	605.2	141.1	612.3
2	22.1	798.7	100.3	8.796	4.7	794.0	620.2	173.8	506.1
3	33.1	775.0	99.4	8.718	4.7	770.3	635.2	135.1	645.3
4	44.2	801.3	100.4	8.805	4.7	796.6	650.2	146.4	601.4
5	55.2	827.0	101.4	8.893	4.7	822.3	665.2	157.1	566.1
6	66.2	849.5	102.3	8.972	4.7	844.8	680.2	164.6	545.1
7	77.3	869.0	103.1	9.042	4.7	864.3	695.2	169.1	534.7
8	88.3	887.0	103.8	9.103	4.7	882.3	710.2	172.1	528.9
9	99.4	903.5	104.4	9.156	4.7	898.8	725.2	173.6	527.4
10	110.4	919.0	105.0	9.209	4.7	914.3	730.2	174.1	528.9
11	121.4	934.3	105.6	9.261	4.7	929.6	755.2	174.4	531.0
12	132.5	935.0					770.2		

Table 57. Rectangular Channel. Run No. 22; Inlet Pressure, 995.4 psia;
 $\dot{m}$, 221.4 lb/hr; Inlet Δt_{sup} , 64.2 F°; Power Setting, 1.717 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\text{B hr}^{-1}\text{ft}^{-2}$ $\times 10^{-4}$	F	°F	°F	F	$\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}$
0	0	633.8					608.2		
1	11.0	645.3	94.3	2.691	1.5	643.8	610.4	33.4	805.7
2	22.1	653.7	94.6	2.700	1.5	652.2	612.6	39.6	681.8
3	33.1	647.3	94.4	2.694	1.5	645.8	614.8	31.0	869.0
4	44.2	647.0	94.4	2.694	1.5	645.5	617.0	28.5	945.3
5	55.2	650.0	94.5	2.697	1.5	648.5	619.2	29.3	920.5
6	66.2	655.3	94.7	2.703	1.5	653.8	621.3	32.5	831.7
7	77.3	659.7	94.9	2.708	1.5	658.2	623.5	34.7	780.4
8	88.3	663.3	95.0	2.711	1.5	661.8	625.7	36.1	751.0
9	99.4	666.3	95.1	2.714	1.5	664.8	627.9	36.9	735.5
10	110.4	668.9	95.2	2.717	1.5	667.4	630.1	37.3	728.4
11	121.4	671.2	95.3	2.720	1.5	669.7	632.3	37.4	727.3
12	132.5	672.7					634.4		

Table 58. Rectangular Channel. Run No. 23; Inlet Pressure, 992.4 psia;
m, 224.3 lb/hr; Inlet Δt_{sup} , 65.5 F°; Power Setting, 4.190 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	671.2					609.3		
1	11.0	701.3	96.5	6.550	3.6	697.7	615.9	81.8	800.7
2	22.1	713.3	97.0	6.584	3.6	709.7	622.5	87.2	755.0
3	33.1	695.4	96.3	6.536	3.6	691.8	629.1	62.7	1042.4
4	44.2	704.5	96.6	6.557	3.6	700.9	635.7	65.2	1005.7
5	55.2	720.4	97.3	6.604	3.6	716.8	642.3	74.5	886.4
6	66.2	733.2	97.8	6.638	3.6	729.6	648.9	80.7	822.5
7	77.3	743.9	98.2	6.665	3.6	740.3	655.5	84.8	786.0
8	88.3	753.2	98.6	6.692	3.6	749.6	662.1	87.5	764.8
9	99.4	761.3	98.9	6.713	3.6	757.7	668.7	89.0	754.3
10	110.4	768.5	99.2	6.733	3.6	764.9	675.3	89.6	751.4
11	121.4	775.0	99.4	6.747	3.6	771.4	681.9	89.5	753.8
12	132.5	779.2					688.5		

Table 59. Rectangular Channel. Run No. 24; Inlet Pressure, 991.4 psia;
m, 225.7 lb/hr; Inlet Δt_{sup} , 66.3 F°; Power Setting, 5.622 kw.

x	L/D _e	t ₀	$\rho(x)$	$q_t''(x)$	Δt_w	t _i	t _f	t _i -t _f	h
in.	--	°F	$\mu\Omega\text{-cm}$	$\frac{\text{B hr}^{-1}\text{ft}^{-2}}{\times 10^{-4}}$	F	°F	°F	F	$\frac{\text{B hr}^{-1}\text{ft}^{-2}\text{F}^{-1}}$
0	0	699.7					610.0		
1	11.0	735.5	98.0	8.833	4.8	730.7	619.5	111.2	794.3
2	22.1	757.7	98.7	8.896	4.8	752.9	629.0	123.9	718.0
3	33.1	733.6	97.8	8.815	4.8	728.8	638.4	90.4	975.1
4	44.2	741.9	98.1	8.842	4.8	737.1	647.9	89.2	991.2
5	55.2	768.7	99.2	8.941	4.8	763.9	657.4	106.5	839.5
6	66.2	787.8	100.	9.013	4.8	783.0	666.8	116.2	775.6
7	77.3	802.8	100.5	9.058	4.8	798.0	676.3	121.7	744.3
8	88.3	815.1	101.	9.103	4.8	810.3	685.8	124.5	731.2
9	99.4	825.3	101.3	9.130	4.8	820.5	695.2	125.3	728.6
10	110.4	834.2	101.6	9.157	4.8	829.4	704.7	124.7	734.3
11	121.4	841.8	102.	9.193	4.8	837.0	714.2	122.8	738.6
12	132.5	848.0					723.5		

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