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**COMBINED TURBULENT
CONVECTION HEAT TRANSFER
TO NEAR CRITICAL WATER**

by

**R. F. Touba
P. W. McFadden**

January 1966

RELEASED FOR ANNOUNCEMENT
IN NUCLEAR SCIENCE ABSTRACTS

PURDUE UNIVERSITY
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Lafayette, Indiana

PURDUE RESEARCH FOUNDATION
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TO

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Research reported herein was taken from the Ph. D. Thesis of R. F. Touba.

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ABSTRACT

Touba, Riza Feredj, Ph.D., Purdue University, June 1965.

Combined Turbulent Convection Heat Transfer to Near Critical Water.

Major Professors: Richard J. Grosh and Peter W. McFadden.

The experimental study reported herein deals with an investigation of heat transfer to near critical water flowing through a heated vertical tube. Turbulent free and forced convection flow was present.

The study utilized the facilities of the High Flux Heat Transfer Research of the School of Mechanical Engineering, Purdue University, sponsored by the United States Atomic Energy Commission. Two AISI type 304 seamless stainless steel tubes of 0.245 and 0.370 inch inside diameter served as joulean heated test sections.

The range of variables covered was as follows:

Test Section Bulk Fluid Temperature	610-716 F
Pressure	3220-3400 psia
Mass Flux	$0.849-3.86 \times 10^6 \frac{\text{lb}_m}{\text{hr ft}^2}$
Heat Flux	$176,000-523,000 \frac{\text{B}}{\text{hr ft}^2}$
Wall Temperature	621-835 F
Wall to Bulk Fluid Temperature Differences	1.28-117 F

Local heat transfer and Stanton numbers were computed from the data. The data was checked against the Dittus-Boelter equation and good agreement was obtained for temperatures below 650 F or Prandtl number, based on the bulk fluid temperature, not exceeding 1.20.

The new correlation of Swenson, Carver, and Kakarala was also tested, which proved satisfactory up to a temperature of around 690 F. Beyond that range agreement deteriorated rapidly with proximity to the peak temperature. The peak temperature was defined as the temperature at which the isobaric specific heat curve peaked.

The effect upon heat transfer of the proximity of both the wall and/or fluid temperatures to the peak temperature was tested. However, the range of the equipment did not allow testing with the bulk fluid temperature above the peak temperature. Very large heat transfer coefficients were obtained for conditions where the wall and/or fluid temperatures approached the peak temperature.

Significant fluctuations were detected in the pressure drop across the test section. These perturbations were present during all isothermal and heating operations at all temperature and pressure levels. The amplitude of the fluctuations magnified at the near critical operations. The cause and the effect of the vibrations could not be determined accurately.

The data is presented in the form of the Stanton number modified to remove the difficulty with the specific heat term. It was possible to correlate the near critical data in the form of

$$St_e Re_b^a = c e^{\frac{H_b}{H_p}}$$

to within ± 16 per cent.

Data below t_b/t_p of 0.93 could be correlated in the form of

$$\overline{St} = \text{constant}$$

for limited ranges of mass and heat fluxes.

INTRODUCTION

The engineer's everlasting desire for greater efficiencies and economy has tempted him towards higher and higher temperatures. This thrust has taxed both science and technology. From the heat transfer point of view, the recent interests in the developments of supercritical steam power plants, nuclear reactor technology, cooling of computers, and rocket engines have been most demanding. Efficient methods of heat transfer are being sought. One such method is the use of fluids in the proximity of their critical point, where investigations have revealed very sharp increases in the heat transfer rates.

Complexity of the process and the characteristics of the fluids in the near critical region pose a great difficulty to both the analytical worker and the experimentalist. The basic equations that describe such a process are vastly complicated, and present the analyst challenge and many hardships. The experimental investigations are hampered not only by the very difficult task of obtaining reliable information, but also by the unavailability of vast and accurate information on thermodynamic and transport properties.

The critical pressure and temperature specify a limit above which a liquid and saturated vapor can not coexist together in thermodynamic equilibrium. Thus, above the critical point there is no distinction between a liquid and vapor. The critical pressure, temperature,

and enthalpy for water are believed to be 3208 psia, 705.5 F, and 900 B/lb_m respectively.

The correlation of the experimental data for heat transfer has been another major difficulty with near critical fluids. From a practical standpoint it is essential to obtain simple yet general correlations. Many such attempts have been unsuccessful in the critical region partly due to the large changes in the fluid properties. The properties also present a massive obstacle to the theoretical studies. Besides obtaining experimental data, this work was also an attempt to find a simple method of correlation of data.

The facilities of the High Flux Heat Transfer Loop of the School of Mechanical Engineering, sponsored by the United States Atomic Energy Commission, were used. The study concerned itself primarily with turbulent forced convection heat transfer to near critical water, exploring a region where previously little or no data had been available. Figure 1 describes the area of interest based on the bulk fluid properties. In the succeeding sections of this thesis a literature survey, a description of the test equipment, an analysis of the problem, and the results of the investigations are presented.

FIGURE 1. REGION OF INTEREST TO THIS INVESTIGATION

LITERATURE SURVEY

The literature dealing with the subject of heat transfer to near critical fluids is scarce. This can be attributed primarily to the complexity of both analytical and experimental investigations in this region. While the solution of the most simplified equations, roughly describing the process of heat transfer near the critical region of a fluid, involves laborious numerical methods, the experimentalist faces the most difficult task of obtaining reliable data and processing it satisfactorily. Briefly, it is the complex nature of the physical process that has denied the investigators a full understanding into this mode of heat transfer.

The high-lights of several analytical and experimental investigations in the regions below and above the critical point are described in the following. The problem of heat transfer to near critical fluids under the effect of combined forced and free turbulent convection is far from solved and understood.

1. Analytical Investigations

Deissler (1)* has generalized a previous analysis for fully developed turbulent variable property flow and heat transfer in smooth tubes to make it applicable to air and supercritical water.

---

*Underlined numbers in parentheses designate References in the Bibliography.

Large variations with temperature of the physical properties of supercritical water were recognized and the following assumptions were made;

1. The eddy diffusivity for momentum, ϵ , and heat transfer, ϵ_h , were assumed equal. For $Re > 15,000$, the ratio of ϵ_h to ϵ was nearly constant.
2. The variation across the tube of the shear stress and the heat flux was considered to have a negligible effect on the velocity and temperature distributions.
3. The static pressure across the tube was considered constant.

$$4. \quad \frac{\epsilon}{\nu_w} = \kappa^2 \frac{\left(\frac{du^+}{dy^+}\right)^3}{\left(\frac{u^+}{y^+}\right)^2} \quad * \quad \text{away from the wall, and} \quad \frac{\epsilon}{\nu_w} = n^2 u^+ y^+$$

close to the wall.

$$5. \quad \frac{\mu}{\mu_w} = a \left(\frac{T}{T_w}\right)^b, \text{ where } a \text{ and } b \text{ are constants.}$$

The following equations for the computation of Reynolds number, Nusselt number, and friction factor were then obtained.

$$Nu_w = 2 r_o \frac{Pr_w}{T_b^+} \quad (1)$$

$$Re_w = 2 \bar{U} r_o^+ \quad (2)$$

$$f_w = \frac{2}{\bar{U}^+} \quad (3)$$

---

* For description of symbols refer to Appendix A.

A plot of Nusselt number against Reynolds number for a pressure of 5000 psia and a wall temperature of 900 F with $\Omega = \frac{q_w \sqrt{\tau_w / \rho_w}}{c_{p_w} g \tau_w T_w}$ as a parameter was presented. The variation of Nusselt number with Ω proved to be rather complex, and a single reference temperature which could be used in evaluating the properties (independent of Ω) could not be obtained. Thus, a plot of the reference temperature T_{ref} versus $\frac{T_w}{T_b}$ with T_w as a parameter was given for use in the determination of Nusselt and Reynolds parameters. A similar plot gave the reference temperature for use in the determination of the friction factor. For better results it was recommended that Nu_{ref} determined from a plot of Nu_{ref} versus Re_{ref} with $Pr = 1$ as a parameter be multiplied by $Pr_w^{0.45}$.

A similar analysis for air showed substantial agreement between experimental results and analytically predicted heat transfer and friction correlations.

E. R. G. Eckert (2), in discussing the above work of Deissler, gave a plot of $\sigma = \frac{t_{ref} - t_b}{t_w - t_b}$ versus $\frac{t_p - t_b}{t_w - t_b}$ with t_w as a parameter. The parameter t_w became quite insignificant as all curves for different t_w values aligned well along a single curve. Eckert showed that for

$$\frac{t_p - t_b}{t_w - t_b} \leq 0, \quad \sigma \approx 0,$$

hence $t_{ref} \approx t_b$; and for

$$\frac{t_p - t_b}{t_w - t_b} \geq \text{unity}, \quad \sigma \approx \text{unity},$$

hence $t_{ref} \approx t_w$.

It was reported that the influence of t_p on the location of $t_{ref}(\text{friction})$ to be used in the calculation of friction was weaker than for heat transfer.

In his closure, Deissler concluded that the properties in the Nusselt and Reynolds number could be calculated at t_p as long as t_p remained between t_w and t_b . If t_p was higher than t_w the properties should be evaluated at t_w . If t_p was lower than t_b then they should be evaluated at approximately t_b . The Prandtl number was to be evaluated at t_w rather than t_{ref} , i.e.,

$$Nu_{ref} = C Re_{ref}^a Pr_w^b \quad (4)$$

Goldmann (3, 4) has presented a method of analysis to predict heat transfer and pressure drop characteristics for fluids with temperature dependent properties in fully developed turbulent flow. It was a further extension of the Reynolds analogy between turbulent momentum exchange and heat transfer. Results were shown to be in good agreement with experimental data available for air under high heat fluxes. Heat transfer and pressure drop characteristics for water at 5000 psia as predicted by this method were given in graphical form.

The essential difference between Goldmann's and Deissler's analyses was that Goldmann avoided the assumption of the constancy of κ and n . Instead, the assumption was made that the turbulent mixing process at any point was a function of fluid properties at that point but was not affected by small property changes in the neighborhood of that point.

Plots of the heat flux parameter $\left(\frac{q_w'' D^{0.2}}{G^{0.8}}\right)$ versus wall temperature with bulk temperature as a second parameter, and shear stress parameter $\left(\frac{\tau_w (g \rho_b) D^{0.3}}{G^{1.7}}\right)$ versus bulk temperature with wall temperature as a second parameter, were presented in references (3, 4) for water at 5000 psia.

Such a correlation followed from a consideration that for convective heat transfer the following relations are usually employed:

$$\frac{hD}{k} = C \left(\frac{GD}{\mu}\right)^m (Pr)^b \quad (5)$$

and

$$h \equiv \frac{q_w''}{t_w - t_b} \quad (6)$$

For isothermal shear stress the following relation is applied:

$$\frac{\tau_w}{\rho_u^2} = \frac{f}{2} = \frac{C}{\left(\frac{GD}{\mu}\right)^{1-n}} \quad (7)$$

These relations have led to

$$\frac{q_w'' D^{1-m}}{G^m} = f(t_w, t_b), \quad m = 0.8 \quad (8)$$

and

$$\frac{\tau_w D^{1-n}}{G^{1+n}} = g(t_w, t_b), \quad n = 0.7 \quad (9)$$

Usually m and n are functions of Reynolds and Prandtl numbers.

Eckert (2) gave the following theoretically founded equation for forced convection heat transfer in a tube with turbulent flow:

$$St = \frac{Nu}{Re Pr} = \frac{0.0384(Re)^{-0.25}}{1 + A(Re)^{-0.125} (Pr-1)} \quad (10)$$

The value of A may be obtained from a curve by W. Bühne presented in reference (2), page 207 or calculated from the following equation given by E. Hoffman (2, page 207):

$$A = 1.5 \text{ Pr}^{-\frac{1}{6}} \quad (11)$$

The property values to be used in the above equations must be computed at a reference temperature

$$t_{\text{ref}} = t_b + \frac{0.1 \text{ Pr} + 40}{\text{Pr} + 72} (t_w - t_b) \quad (12)$$

Hsu and Smith (5) have presented a study in which the effect of large density changes (in the radial direction) upon heat transfer within a tube was investigated analytically. Deissler's work did not take into account the influence of density variations on the radial transfer of momentum and heat between adjacent fluid elements, and the radial variation of the force of gravity on the fluid. Hsu and Smith attempted this objective, and employed Deissler's expressions for the eddy diffusivity in solving the equations for heat and momentum transfer. The effect of free convection on heat transfer coefficient for turbulent flow was predicted.

The results indicated that the effect was significant only for relatively high Grashof numbers and low Reynolds numbers. The ratio $\frac{\text{Gr}}{\text{Re}_w^3}$, obtained in the analysis, showed the influence of variable density upon the shear stress profile. $\text{Gr} \ll \text{Re}_w^3$ signified the negligible effect of body force.

Velocity and temperature profiles were computed. The results for heating showed that including the density variation flattened both

the temperature and velocity curves and increased the heat transfer coefficient at a given Reynolds number.

The computed heat transfer results for near critical CO₂ at 1200 psia were compared with the experimental determinations of Bringer (6). It was observed that taking into account the affect of density variations improved the agreement with the experimental data. However, the Grashof number at Bringer's operating conditions was too low for natural convection to be significant.

Recently, Ojalvo (7) has presented an analytical study of turbulent heat transfer in a vertical circular tube under the conditions of combined forced and free convection with uniform wall heat flux. Three linear integro-differential equations which contained Prandtl, Rayleigh, and friction Reynolds parameters were obtained from the basic conservation equations. A numerical solution yielded fully developed velocity profile, temperature profile and pressure drop for fixed values of the above parameters. Nusselt and Reynolds parameters together with mixed-mean-to-wall temperature difference were also computed.

Ojalvo's assumptions which are of interest in this work are presented below:

1. All fluid properties were assumed constant except density in the expression for body force. A mean density was used for all other density terms.
2. Viscous dissipation and axial heat conduction were considered negligible compared to the heat conduction in the radial direction.
3. The eddy diffusivities of momentum and heat were assumed to be in constant proportion, and given by an equation presented by Lykoudis (8) in his study of heat transfer to liquid metals, viz.

$$\epsilon_h = \frac{6\epsilon}{\pi^2} \sum_{n=1}^{\infty} \frac{1}{n^2 \exp(0.01 \frac{n^2}{Pr})} \quad (13)$$

4. The equation giving the eddy diffusivity of momentum was a modification of Reichardt's equation and was based on various available experimental data. Thus,

$$\frac{\epsilon}{\nu} = 0.0667 \text{ Re}^*(0.5 + \eta^2)(1 - \eta^2) \quad (14a)$$

$$\text{for } 0 \leq \eta \leq \left(1 - \frac{10}{\text{Re}^*}\right)$$

$$\frac{\epsilon}{\nu} = 0 \quad \text{for } \left(1 - \frac{10}{\text{Re}^*}\right) \leq \eta \leq 1 \quad (14b)$$

Certainly some of the assumptions of the analytical works presented above may not be valid for heat transfer near the critical region. Therefore, such solutions must be regarded as limited in scope and valid only within the domain of their assumptions. In the section reviewing experimental investigations the inadequacy of some analytical results is presented.

2. Experimental Investigations

McAdams, et al. (9) have conducted experimental investigations on the problem of heat transfer to superheated steam at high pressures, and have determined local coefficients of heat transfer from surfaces to superheated steam at pressures up to 3500 psia and bulk temperatures from 430 to 1000 F. Their experimental apparatus consisted of a vertical annulus with electrically heated inner wall and unheated outer wall. Steam flowed upward, with mass velocity restricted to 55,000 - 165,000 $\frac{\text{lb}_m}{\text{hr ft}^2}$. Wall and bulk fluid temperature differences were from 100 to 625 F.

The following correlation which must not be extrapolated outside the range of data was recommended:

$$\left(\frac{hD_e}{k_b}\right) \left(\frac{c_p \mu}{k}\right)_b^{-\frac{1}{3}} \left(\frac{L}{D_e}\right)^{0.13} = 0.0126 \left(\frac{D_e G}{\mu_b}\right)^{0.89} \quad (15)$$

Based on thermal conductivity extrapolated from newer measurements of Keyes and Sandell (9)* the above equation became:

$$\left(\frac{hD_e}{k_b}\right) \left(\frac{c_p \mu}{k}\right)_b^{-\frac{1}{3}} \left(1 + \frac{2.3}{L/D_e}\right)^{-1} = 0.0214 \left(\frac{D_e G}{\mu_b}\right)^{0.80} \quad (16)$$

$$\text{for } 6000 \leq \frac{D_e G}{\mu_b} \leq 50,000$$

It was reported that for a given Reynolds number the maximum deviation from measured values was 28 per cent, and the average deviation was seven per cent.

Kreith and Summerfield (10) have conducted an experimental study of heat transfer to water at high fluxes with and without surface boiling. Although they operated at low pressures, their results were worth noting. Surface coefficients of heat transfer, at heat fluxes up to $1.04 \times 10^6 \frac{\text{B}}{\text{hr ft}^2}$, were obtained for water flowing in stainless steel tubes at mass fluxes up to $2.8 \times 10^6 \frac{\text{lb}_m}{\text{hr ft}^2}$. The non-boiling forced convection data were correlated by the Colburn equation

$$\text{Nu}_f = 0.023 \text{Re}_f^{0.8} \text{Pr}_f^{1/3} \quad (17)$$

within plus or minus five per cent in the Reynolds number range from 50,000 to 350,000. It was also concluded that when heat was transferred

---

* Personal communications of McAdams.

to water in turbulent flow without surface boiling in a tube, the friction coefficient at equal Reynolds numbers decreased according to the following equation:

$$\frac{f(\text{heat transfer})}{f(\text{isothermal})} = \left(\frac{\mu_w}{\mu_b}\right)^{0.13} \quad (18)$$

The authors also reported that the Colburn equation could not predict the heat transfer coefficient between a metal surface and water flowing in a tube when the surface temperature exceeded the boiling point of the fluid.

The Dittus-Boelter equation (2) for heat flow has been given as

$$Nu = 0.0243 (Re)^{0.8} (Pr)^{0.4} * \quad (19)$$

where properties may be evaluated at the bulk fluid temperature.

McAdams has proposed a rule⁺ by which heat transfer in mixed (i.e., combined free and forced convection) flow through a vertical tube could be determined as follows: The heat transfer coefficient is calculated from forced convection and free convection relations and the larger value is used. Eckert reported that measured values did not deviate by more than 25 per cent from the values calculated by McAdams' rule.

Russian investigators Miropolskii and Shitsman (11), working with a steel tube of 0.354/0.276 inch diameter heated in an electrical radiation furnace, covered the following ranges of variables:

* (2), p. 211.

+ Ibid, p. 332.

Pressure = 60 - 4200 psia

Fluid temperature = 36.5 - 770 F

Surface to fluid temperature diff. = 16.2 - 387 F

$$\text{Mass flux} = 0.125 \times 10^6 - 2.21 \times 10^6 \frac{\text{lb}_m}{\text{hr ft}^2}$$

$$\text{Heat flux} = 0.369 \times 10^5 - 7.38 \times 10^5 \frac{\text{B}}{\text{hr ft}^2}$$

The fluid temperature at intermediate distances along the test section were determined by linear interpolation from inlet and outlet temperatures. Hydrodynamic stability was established before the inlet to the heated section.

Experimental results were compared with certain relations frequently used for heat transfer computations in conjunction with turbulent flow through a tube. It was concluded that the following relations,

$$\text{Nu}_b = 0.023 \text{Re}_b^{0.8} \text{Pr}_b^{0.4} \quad (20)$$

$$\text{Nu}_b = 0.021 \text{Re}_b^{0.8} \text{Pr}_b^{0.43} \left(\frac{\text{Pr}_b}{\text{Pr}_w} \right)^{0.25} \quad (21)$$

$$\text{Nu}_b = 0.027 \text{Re}_b^{0.8} \text{Pr}_b^{0.33} \left(\frac{\mu_b}{\mu_w} \right)^{0.14} \quad (22)$$

did not give satisfactory results. For heating superheated steam at given inlet temperature, pressure, and mass flux, equations (21) and (22) gave an increase in the calculated values of h , while the experimental results showed that heat transfer coefficients decreased with an increase in the wall-to-bulk fluid temperature difference. Also, the

application of the above equations in the temperature ranges where the values of Prandtl number varied in connection with changes in the specific heat did not give satisfactory results.

Miropolskii and Shitsman proposed an equation containing the known parameters Nu , Re , Pr for the evaluation of local heat transfer coefficients to water and steam. They believed this to be advantageous for temperatures of the heat transfer medium above 300 F. Their equation is:

$$Nu_b = 0.023 Re_b^{0.8} Pr^{0.8} \quad (23)$$

The bulk temperature of the fluid is assumed to be the determining factor in the calculation of Nu and Re , but Pr is determined according to the following rules:

1. Use the wall temperature if $Pr_b > Pr_w$.
2. Use the fluid temperature if $Pr_b < Pr_w$.

The authors reported that, for water at subcritical pressures of 25-220 atm., their data was predicted within plus or minus ten per cent by their proposed equation. The accuracy dropped down to ± 15 per cent for pressures of 230-280 atm. It was also reported that the standard correlations, equations (20) to (22), were suitable for the range of temperatures and pressures where Pr variations were associated with changes in the dynamic viscosity.

Dickinson and Welch (12), working with a horizontal, electrically heated AISI Type 304 stainless steel tube having an inside diameter of 0.300 inch, covered the following range of variables:

Pressure	4500 psia - some 3500 psia
Fluid bulk temperature	220 - 1000 F
Mass flux	$1.6 - 2.5 \times 10^6 \frac{\text{lb}_m}{\text{hr ft}^2}$
Heat flux	$28 - 58 \times 10^4 \frac{\text{B}}{\text{hr ft}^2}$
Wall-to-steam temperature diff.	> 40 F

Data points having bulk temperatures below 660 F were compared with equation (20), viz.

$$\text{Nu}_b = 0.023 \text{Re}_b^{0.8} \text{Pr}_b^{0.4} \quad (20)$$

and were found to be approximately ten per cent below the values obtained from the above equation. Due to unavailability of adequate property values, no attempt was made to correlate the data using dimensionless parameters completely. Instead, film coefficients were plotted against surface temperature for various values of mass flux. The authors reported that cross plots of the data indicated that the heat transfer coefficient varied approximately as $G^{0.8}$ for bulk temperatures below 660 F and as $G^{1.0}$ for bulk temperatures above 800 F.

It was concluded that for surface temperatures within 800 to 1100 F, the data could reliably be represented in the form of a constant Stanton number of 0.00189 for the range of mass fluxes tested. That is,

$$\text{St.} = \frac{h}{c_p G} = 0.00189 \pm 10 \text{ per cent.} \quad (24)$$

where c_p is evaluated at the wall temperature. For the range of 660-800 F heat transfer coefficients higher than those predicted by theoretical studies of Deissler and Goldmann were obtained.

Touba (13) investigated the characteristics of heat transfer to near critical water (pressure-and-temperature-wise) in turbulent upward flow through an electrically heated Type 347 seamless stainless steel tube. The range of variables covered was as follows:

Pressure	3230 - 3300 psia
Fluid bulk temperature	685 - 712 F
Mass flux	$1.87 - 2.69 \times 10^6 \frac{\text{lb}_m}{\text{hr ft}^2}$
Heat flux	$5 \times 10^5 \frac{\text{B}}{\text{hr ft}^2}$
Wall-to-fluid temperature diff.	2 - 37.1 F.

Local and average heat transfer coefficients for subcritical and supercritical (temperature-wise) water were determined. Dimensional analysis based on prevailing conditions in the test section led to the Stanton number as a major parameter, which was redefined to suit the needs of the work.

Heat transfer coefficients of more than 50,000 Btu per hour per square foot per F were reported. Large values occurred close to the temperature at which the specific heat corresponding to a given pressure reached its peak.

Plots of Stanton number and the heat transfer coefficient versus bulk and wall temperatures were presented. A plot of Stanton number versus $\frac{t_b}{t_p}$ was given. This correlation helped reduce the scatter of data. The Dittus-Boelter equation failed to give adequate results.

Recently Swenson, Kakarala, and Carver (14) published the results of their investigations in heat transfer to supercritical water in smooth-bore tubes. The range of variables covered was:

Pressure	3300 - 6000 psia
Fluid bulk temperature	167 - 1068 F
Mass flux	$4 - 15.85 \times 10^5 \frac{\text{lb}_m}{\text{hr ft}^2}$
Heat flux	$6.5 - 57.8 \times 10^4 \frac{\text{B}}{\text{hr ft}^2}$
Wall-to-fluid temperature diff.	11 - 513 F

An electric-resistance welded AISI Type 304 stainless steel tube 116 inches long, having a nominal 1/2-inch outer diameter and 0.371 inch inside diameter was used as the test section. The test section was installed vertically with the flow upward, and heated electrically along 72 inches of its length. An open-circuit heat transfer test apparatus was employed.

Local forced convection heat transfer coefficients for supercritical water were obtained. It was shown that conventional forced convection correlations for fully developed turbulent flow were inadequate and did not fit the data. A satisfactory equation, which correlated only the data from the downstream half of the tube, was obtained by modifying the conventional forced convection heat transfer relation. This relation is of the form:

$$\text{Nu}_w = a \text{Re}_w^b \text{Pr}_w^c \left(\frac{v_b}{v_w}\right)^d \quad (25)$$

where

$$a = 0.00459$$

$$b = 0.923$$

$$c = 0.613$$

$$d = 0.231$$

All properties in the non-dimensional parameters are evaluated at the inside wall temperature, and the specific heat term in the Prandtl number is defined as

$$c_{p_h} = \frac{H_w - H_b}{t_w - t_b}.$$

The authors reported a good fit, indicating that 95 per cent of their data lay within ± 15 per cent of the correlation. It was also found that the same equation correlated the supercritical heat transfer data of carbon dioxide, and predicted heat transfer coefficients in the regions where the constant property assumptions were valid. Furthermore, it was concluded that in the supercritical region heat transfer coefficient is strongly affected by heat flux - specifically, it decreases with increasing heat flux.

For the sake of completeness, a plot presented by Swenson, Kakarala and Carver in an attempt to compare their results with the analytical studies of Deissler (1) and Goldmann (3, 4) is reproduced in Figure 2. Since the property values used in the three works represented in this plot were not the same, the reader must be cautious as to the extent of his conclusions. It has already been stated that Hsu and Smith (5) modified Deissler's analysis, and obtained a better agreement with experimental data for carbon dioxide.

It is now appropriate to point out and survey some of the important experimental investigations dealing with the subject of heat transfer to supercritical CO₂. Similar studies were reported for Hydrogen by Hess and Kunz (15), and for Oxygen by Powell (16). Powell's

FIGURE 2. COMPARISON OF EQUATION 25 WITH
ANALYTICAL STUDIES
(After Swenson et.al.)

data showed a minimum for the heat transfer coefficient near the critical temperature.

Bringer and Smith (6) reported measurements of heat transfer coefficients for carbon dioxide in turbulent flow. The bulk temperature was varied from 70 to 120 F at a pressure of 1200 psia.* The heat transfer coefficients ranged from $300 - 2600 \frac{\text{B}}{\text{hr ft}^2 \text{ F}}$ over a Reynolds number range of $3 \times 10^4 - 30 \times 10^4$. The test section was installed vertically with upward flow. Inadequacy of existing empirical and semitheoretical correlations was observed. Numerical computations for carbon dioxide based on Deissler's analytical method were found to fit the experimental data. A simplified procedure was proposed for estimating heat transfer coefficients in the critical region. The authors replaced Deissler's $\Omega = 0$ solution for water at 5000 psia by a conventional type heat transfer equation - namely,

$$\text{Nu}_{\text{ref}} = 0.0266 \text{Re}_{\text{ref}}^{0.77} \text{Pr}_w^{0.55} \quad (26)$$

and proposed to evaluate the property values in Nusselt and Reynolds parameters according to the rule given by Deissler (1). Table 1 below restates this rule in a tabular form.

TABLE 1

Eckert-Deissler Rule for Reference Temperature

$\frac{t_p - t_b}{t_w - t_b}$	$t_{\text{reference}}$
0 to 1.0	t_p
> 1.0	t_w
< 1.0	t_b

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* CO₂ critical conditions are: $P_c = 1071 \text{ psia}$, $t_c = 87.8 \text{ F}$.

It was reported that the method worked satisfactorily when compared with the computed heat transfer coefficients of Deissler for supercritical water, but showed 30 per cent deviation when compared with the carbon dioxide results. The discrepancy was attributed to the fact that while the CO_2 was very close to the critical point (reduced pressure = 1.1), the water was further removed (reduced pressure = 1.6). The coefficient in equation (26) could be increased to obtain a better fit with an average deviation of ± 16 per cent.

Russian investigators Petukhov, Krasnoschekov, and Protopopov (17) have investigated experimentally the problem of heat transfer to supercritical carbon dioxide in turbulent flow through a horizontal tube. Measurements were carried out at pressures of 90, 100, and 110 Kg/cm^2 and temperature differences from 4 to 50 C. Reynolds number ranged from 5×10^4 to 3×10^5 .

An equation correlating the available experimental data on heat transfer to carbon dioxide and water in the supercritical region was proposed. It was reported that the majority of their data was correlated within ± 15 per cent, and only 6 per cent diverged slightly more than 20 per cent. The equation is:

$$\text{Nu}_b = \text{Nu}_b^* \left(\frac{\mu_b}{\mu_w} \right)^{0.11} \left(\frac{k_b}{k_w} \right)^{-0.33} \left(\frac{c_{p_h}}{c_{p_b}} \right)^{0.35} \quad (27)$$

where $c_{p_h} = \frac{H_w - H_b}{t_w - t_b}$, as was defined previously.

Nu_b^* was reported to have been derived by Petukhov and Kirillov, by a theoretical computation, for heat transfer to turbulent pipe flow of

fluids with constant physical properties.

$$Nu_b = \frac{\frac{N}{8} Re_b Pr_b}{12.7 \sqrt{\frac{N}{8}} (Pr_b^{2/3} - 1) + 1.07} \quad (28a)$$

where

$$N = \frac{1}{(1.82 \log Re_b - 1.64)^2} \quad (28b)$$

Koppel and Smith (18) have carried out investigations on heat transfer to near critical carbon dioxide. A horizontal, electrically heated test section was employed. The range of the experiments was:

Pressure	1071 - 1100 psia
Heat flux	$2 \times 10^4 - 20 \times 10^4 \frac{B}{hr \ ft^2}$
Fluid temperature	65 - 120 F
Reynolds number	$3 \times 10^4 - 30 \times 10^4$

Unusual results were observed. One was a decrease in the wall temperature in the section of the tube immediately after the entrance. The investigators reported that:

1. the phenomenon would not occur if the bulk fluid temperature at the tube inlet was above the peak temperature;
2. the transition was gradual with the wall temperature profile at the entrance first flattening and finally dipping as the flow rate was increased;
3. the flow rate necessary to cause the transition was an increasing function of the wall heat flux.

Furthermore, they concluded that:

1. normal heat transfer results were obtained for fluid inlet temperatures above the peak temperature;
2. at inlet temperatures below the peak temperature unusual axial variations of the heat transfer coefficient were observed, which behavior was most pronounced at the critical pressure;
3. at low or moderate wall-to-fluid temperature differences, the fluid exhibited increasingly better heat transfer characteristics as the peak temperature was approached.

An effort to correlate the data according to equation (23) and the rule proposed by Miropolskii and Shitsman (11), proved unsatisfactory. The authors indicated that a different approach was necessary to explain heat transfer near the critical point.

Recently Wood and Smith (19) measured temperature and velocity profiles and local heat transfer coefficients for turbulent flow of carbon dioxide in a tube at 1075 psia ($p_c = 1071$). Their results showed a maximum in the velocity profile between the wall and tube axis, a peak in the heat transfer coefficient, and an acute flattening of the temperature profile as the bulk fluid temperature passed through the peak temperature.

A closed loop circuit was used. The test section was a seamless Inconel tube. Inconel was selected for its very small temperature coefficient of resistivity. The tube outside diameter was 1.000 inch, the inside diameter 0.902 inch, and the total length of the tube 56.5 inches. The tube was installed vertically and heated along 30.1

inches of its length by AC power. The length of the tube upstream from the heated section was 21 inches. Temperature and velocity profiles were measured at 27.7 inches downstream from the beginning of the heated section.

The temperature and velocity profiles were obtained from separate runs as nearly similar as practically possible. The authors indicated that since the reproducibility was good, the measurements from two separate runs could be used as one.

Agreement between the electrical heat input and the measured fluid energy rise was reported to have been within two per cent at temperatures well below the peak temperature. However, for temperatures close to t_p variations up to 20 per cent were reported as common. It was also noted that when the temperature profile included t_p , i.e., density was changing rapidly, the agreement between flow rates from the integrated velocity profile and the venturi meter was poor.

The authors concluded that:

1. velocity profiles with maxima at positions other than center were possible and existed;
2. the temperature profiles flattened acutely when the bulk temperature passed through t_p ;
3. the maximum in the heat transfer coefficient was properly associated with the bulk fluid temperature and not the wall temperature;
4. at constant bulk temperature the heat transfer coefficient decreased with increasing heat flux.

The authors attempted to account for the general effects by considering a two resistance concept, consisting of a region next to the wall where conduction was the dominant heat transfer mechanism and the turbulent core where the heat capacity or the product of the heat capacity and density was the significant variable. It was reasoned that:

1. As the fluid temperature approached the peak temperature, c_p increased rapidly with temperature, heat transfer due to mixing was improved and temperature gradients in the core were lowered, resulting in the flattening of the temperature profile;
2. the peak in the heat transfer coefficient near the peak temperature resulted from better mixing in the core and a large increase in the thermal conductivity near the wall;
3. the sudden decrease in the heat transfer coefficient above the peak temperature was due to a large decrease in the fluid thermal conductivity accompanied by an increase in the core resistance;
4. the increase in the heat flux caused higher wall temperatures, thus lowering the thermal conductivity which in turn aggravated the situation resulting in lower heat transfer coefficients.

In concluding this section the following points must be recalled. Most of the works reported here dealt with the far supercritical region. In general only the investigations of Koppel and Smith (18), Wood and Smith (19), and Touba (13) were specifically intended to study the

characteristics of heat transfer to the near critical fluids where the reduced pressure ($\frac{P}{P_c}$) was less than 1.05. Of these the first two dealt with carbon dioxide and the last with water. Some investigations have only approached this region within the course of other work. Presently there are no correlations that were devised for the near critical region. The analytical work of Hsu and Smith (5) was the only study intended for the near critical region, while the works of Deissler (1) and Goldmann (3, 4) belong to the far supercritical region. Basically, investigations have been carried out under three different test section arrangements - namely, horizontal, vertical, and vertical annulus (9). In the near critical region the results for the horizontal and vertical tubes may not agree since there is a sharp contrast in the direction of the force fields.

Regarding the results, there is general agreement that the heat transfer coefficient increases appreciably as the peak temperature is reached. A sharp contrast was the results of Powell (16) which indicated a minimum near the critical temperature. Although the two resistance concept of Wood and Smith substantiated their experimental results, it depends to some extent upon the controversial fact that the fluid thermal conductivity undergoes a sharp increase near the peak temperature. Such is not true of the commonly used thermal conductivity values for water.

ANALYSIS

1. Introduction

The problem of heat transfer to critical water in steady flow through a heated vertical tube under the effects of combined forced and free convection is treated in the following. Emphasis is placed mainly on the presentation and definition of the problem and the variables involved therein.

The mixed or combined flow region is defined as the region where the effects of free and forced convection are of the same order of magnitude. It may also be defined as the region where heat transfer differs by more than ten per cent* from the value obtained with forced or free convection relations respectively.

It has been the general practice to express heat transfer results for convection in terms of certain non-dimensional parameters and/or combinations thereof. As long as the thermodynamic properties employed in these parameters remained constant or experienced very slow changes no major trouble with these correlations was encountered.

The relations and methods proposed for the supercritical region far from the critical environment cannot be expected to give proper

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* (4), p. 332.

results for near critical conditions. Since in the far-supercritical states the thermodynamic properties no longer experience the large variations characteristic of the near critical region, the use of conventional type correlations may prove to be adequate.

Also, the concept of a "reference temperature" has been introduced in an effort to make valid the use of conventional correlations in the regions where thermodynamic property changes may be present.

However, the validity of such rules for supercritical fluids has not been properly established. Also, it is not certain that these rules could be applied successfully in the immediate vicinity of the critical region where more severe conditions prevail. One such rule is given by Deissler where the fluid properties should be evaluated at the peak temperature.

2. Analytical Approach

The essential features which must be considered in the analysis of the problem undertaken in this work are:

1. Forced and free convection single phase upward flow through a tube of circular cross section.
2. Highly variable fluid thermodynamic and transport properties.
3. Turbulent flow.
4. Uniform wall heat flux.

Turbulent fluctuations may be introduced in the momentum, energy, and continuity equations in the conventional manner - i.e., by substituting for a turbulent variable its mean and fluctuating components, for example, $u = \bar{u} + u'$. The resulting equations could then be simplified on the basis of various assumptions.

Whether any use could be made of the conventional boundary layer assumptions and techniques to simplify the equations further is doubtful. The difficulty is due to two facts. First, the common assumption for curved walls that the boundary layer thickness is small compared with the radius of curvature of the wall can not be satisfied for tubes of rather small diameter used in experimental investigations and in the practice. Second, no orders of magnitude can be easily assigned to the perturbation terms.

Presently, very little is known about such perturbation terms as ρ' and μ' and their behavior. Also, the data available on velocity perturbation terms has been obtained mainly under isothermal flow conditions.

It can also be shown that in the case of variable fluid properties the eddy diffusivity for momentum, as it is known for constant property fully developed flow, can not basically satisfy the many requirements. The same conclusion is reached for eddy diffusivity for heat transfer.

Another point to be considered is the question whether fully developed flow can be achieved in heat transfer to a fluid with rapidly varying thermodynamic properties. The continuity equation, after the application of various common assumptions, reads:

$$\frac{1}{r} \frac{\partial}{\partial r} (r \bar{\rho} \bar{u}_r) + \frac{\partial}{\partial z} (\bar{\rho} \bar{u}_z) + \frac{1}{r} \frac{\partial}{\partial r} (\overline{r \rho^0 u_r^0}) = 0 \quad (29)$$

If by fully developed flow it is meant that the velocity and temperature profiles are developed and have become invariant, then that is not possible here. Contrary to constant property flow, $\frac{\partial}{\partial z} (\bar{u}_z)$ can not be set equal to zero since $\bar{u}_z$ varies not only in the radial direction, but also continuously in the axial direction as the fluid density is changed. Axial velocity is of course further influenced by the free convection effects which are also a function of density. The work of Wood and Smith (19) has clearly illustrated the dependence of temperature and velocity profiles upon bulk fluid temperature. It is quite possible that the heat flux or wall temperature also has significant influence upon the velocity and temperature profiles. The dependence of heat transfer coefficient on the heat flux, as reported by many investigators, substantiates this point.

Certainly the inability of many analytical findings to confirm experimental results for heat transfer to near critical fluids is partially rooted in the above mentioned difficulties.

3. Semi-Empirical Approach

An efficient method of making heat transfer data available to the designer of equipment has been the use of so-called non-dimensional parameters.

Equations of the form

$$Nu = f' (Re, Pr) \quad \text{Forced Convection} \quad (30)$$

$$Nu = g' (Gr, Pr) \quad \text{Free Convection} \quad (31)$$

have proved very successful in heat transfer work. However, Hallman (20), studying combined forced and free convection in a vertical tube in the laminar region, reported that the Rayleigh number, defined as

$$Ra \equiv \frac{\rho^2 g \beta c_p D^4}{16 \mu k} \frac{\partial t}{\partial z} \quad (32)$$

was significant as a measure of the influence of the free convection upon the forced convection. $\frac{\partial t}{\partial z}$ represents the axial temperature gradient. For upward flow with heating the fluid is hotter downstream, and thus the Rayleigh number is positive.

It is important to recognize that the Rayleigh number, as defined above, was no different than the product of Grashof, Prandtl, and

$$\frac{D}{(t_w - t_b)} \frac{\partial t}{\partial z}.$$

Sparrow et al. (21) in a paper on combined forced and free convection underlined the significance of the ratio $\frac{Gr_b}{Re_b^2}$ as a measure of the influence of free convection on forced convection flow. They indicated that the ratio would approach unity as the affects of free and forced convection became equal in magnitude.

Hence, for combined forced and free convection it would seem logical to express the heat transfer results in the following form (or one similar to it);

$$Nu = f(Re, Pr, Gr, \frac{D}{(t_w - t_b)} \frac{\partial t}{\partial z}) \text{ Combined Convection} \quad (33)$$

Relations of this form have been used by many authors in presenting their analytical findings. In the case of near critical fluids the evaluation of these parameters becomes very difficult.

Another school, among them Goldmann (3, 4), favor correlations of the form given by equation (8), viz.

$$\frac{q_w'' D^{1-m}}{G^m} = f(t_w, t_b) \quad (8)$$

Although such a correlation entirely removes the difficulty with the thermodynamic and transport properties, it restricts the validity of the results to the range of the experiments. Thus, not only the results of experiments with one fluid may not be applied to other fluids, but perhaps rather extensive experimentations with one fluid are required to cover all possible variables for that fluid.

The difficulty lies in the fact that equation (8) is highly empirical. The so-called heat flux parameter $\frac{q_w'' D^{1-m}}{G^m}$ has complicated dimensions and is also a function of another dimensional quantity. For near critical fluids this correlation is pressure dependent.

Therefore, of the two methods considered here it may be said that: the one employing non-dimensional parameters is quite general, but **strongly** dependent on the fluid properties; the other, rather simple and not dependent on fluid properties, lacks necessary breadth. The last part of the present section is devoted to a brief development of a different method of correlation.

4. Thermodynamic and Transport Properties of Critical Fluids

From the heat transfer standpoint the behavior of thermodynamic and transport properties of fluids near their critical region is most interesting. It is to the rather large changes of these properties in the critical region that a significant increase in the heat transfer coefficient may be attributed.

The critical region, as used in this work, is the area in close proximity to the critical point. - more specifically, the region that is associated with most rapid and large changes in the properties. Thus, the above definition would exclude far-supercritical states such as $P = 5000$ psia where the property changes are, relatively, neither rapid nor large. In more concrete terms, the area of interest here may be given as follows:

$$1.00 \leq \frac{P}{P_c} \leq 1.05 \rightarrow 1.10 *$$

$$0.95 \leq \frac{T_b}{T_p} \leq 1.05$$

The critical pressure and temperature specify a limit above which a liquid and a saturated vapor cannot coexist together in thermodynamic equilibrium. In other words, above the critical point there is no distinction between a liquid and a vapor.** Mathematically stated,

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* Recall that the present interest does not include two phase problems.

** (22), pp. 54-55.

$$\begin{aligned}\frac{\partial P}{\partial v} \Big|_{t_c} &= 0 \\ \frac{\partial^2 P}{\partial v^2} \Big|_{t_c} &= 0\end{aligned}\tag{34}$$

From general thermodynamic relations it follows that at the critical point

$$\begin{aligned}c_p &\rightarrow \infty \\ \beta &\rightarrow \infty\end{aligned}$$

A large change in c_p in turn causes a rapid increase in the enthalpy through the critical temperature range. Little may be said of the exact behavior of viscosity and thermal conductivity, since a knowledge of thermodynamic properties is not sufficient for the determination of the transport properties.

At pressures above the critical pressure $\frac{\partial P}{\partial v} \Big|_t$ is finite, but the inflection point still occurs, though at respectively higher temperatures. For the lack of a better designation, this region is referred to (unproperly or properly) as the pseudo-critical or transposed-critical region. In the vicinity of the inflection point, where $\frac{\partial^2 P}{\partial v^2} \Big|_t = 0$ and $\frac{\partial P}{\partial v} \Big|_t$ attains its least magnitude, the specific heat and the volume expansion coefficient along isobars reach their maxima.

Thus, the interests of the present work are limited to a narrow region on both sides of the peak temperature. As the pressure is increased the magnitude of $\frac{\partial P}{\partial v} \Big|_t$ around the peak temperature increases. This helps lower the peak attained by the β and c_p curves, and also diminishes their rate of change with temperature. In other words, at higher pressures, the thermodynamic properties cease to exhibit the

large variations characteristic of the near critical pressures. The maxima attained by the isobaric specific heat and volume expansion coefficient decrease rather rapidly with accompanying broadening of the curves.

Figures 3 and 4 give the thermodynamic and transport properties of water at 3300 psia. The specific volume, enthalpy, and volume expansion coefficient are shown in Figure 3, while the specific heat, viscosity, and thermal conductivity appear in Figure 4. Large variations in the properties within a narrow band containing the peak temperature are observed. Due to the rapid changes in the properties their experimental determination has posed exceedingly difficult problems. Hence, most of the present plots are based upon derived values, for example curves presented for β and c_p were adapted from (23).

In accordance with the Principle of Corresponding States it is expected that fluids will behave similarly at the same reduced states. Figure 5, adapted from Koppel and Smith (18), presents the thermodynamic and transport properties of carbon dioxide near the peak temperature corresponding to 1100 psia. The reduced pressures in Figures 3, 4 and 5 are 1.029 for water and 1.027 for carbon dioxide. Hence, the two fluids are in corresponding states. The transport properties are seen to be different. This is the real reason for presenting Figure 5.

Thermal conductivity has been subject to many controversies. The presence of tremendous free convection currents near the peak temperatures close to the critical region have prevented its determination with certainty. The newer measurements for carbon dioxide

FIGURE 3. SPECIFIC VOLUME, ENTHALPY, AND VOLUME EXPANSION COEFFICIENT FOR WATER NEAR THE PEAK TEMPERATURE AT 3300 PSIA PRESSURE

FIGURE 4. SPECIFIC HEAT, VISCOSITY, AND THERMAL CONDUCTIVITY FOR WATER NEAR THE PEAK TEMPERATURE AT 3300 PSIA PRESSURE

THERMODYNAMIC PROPERTIES OF
CARBON DIOXIDE

TRANSPORT PROPERTIES OF CARBON DIOXIDE

FIGURE 5 THERMODYNAMIC AND TRANSPORT
PROPERTIES OF CARBON DIOXIDE
AT 1100 PSIA

show that the thermal conductivity at the peak temperature increases rapidly and then decreases, somewhat like the specific heat curve. On the contrary, the recent recommendations for the thermal conductivity of near critical water specify a behavior as presented in Figure 4.

Nowak and Grosh made a complete survey and study of the thermodynamic properties of water in the critical region. Their findings and recommendations were published in a series of reports (24, 25, 26, 27), which at the time of publication represented the best available thermodynamic data for this region. Table II gives the extent of uncertainties in each of the thermodynamic properties derived, smoothed, and recommended by the authors. The data has been extracted from the above reports.

Fritsch and Grosh (23, 28) surveyed the experimental investigations covering the thermal conductivity and dynamic viscosity of water. It was reported that no measurements for thermal conductivity were available very close to the critical point. Where available, experimental uncertainties in measurements were found to be around three per cent for thermal conductivity and six per cent for dynamic viscosity. Appropriate correlations for the critical range were suggested. The uncertainty in the values obtained from these correlation would probably be around fifteen per cent on the average.

The above review helps show that the thermodynamic and transport properties present perhaps the biggest obstacle to a successful correlation of the data for heat transfer in the near critical region. In a later section in conjunction with data reduction the actual values of the properties used in this work are discussed and presented.

TABLE II

Possible Uncertainties in the Thermodynamic Properties
for Water in the Critical Region
(Based on Nowak-Grosh Recommendations)

Property	Uncertainty	Comments
Specific Volume	1-10/10,000	Experimental uncertainty of recent measurements is estimated at 20/10,000.
Volume Expansion Coefficient	5% 5-30%	690-750 F range $\beta_{\max}$ region.
Specific Heat	1-5% 20%* within 40%	Majority of measurements. Near the critical point and the peak temperatures. Derived values.
Enthalpy	2-3/1000 within 1/1000 1-2/1000	Maximum deviations in measurements. Average deviation. Derived values.

5. Formulation of a Proposed Method of Correlation

In a situation where meager experimentally determined property values are available and most of the present plots are based upon derived values, presentation of the results of experiments in terms of parameters which depend on a number of rapidly varying properties would be hard to justify. Even if the thermodynamic and transport properties in the critical region were known precisely, temperature measurements

and calculations would need to be accurate within a few tenths of a degree or better in order to stay within reasonable bounds of error in computing the values of the dimensionless parameters involved. Figures 3 to 5 support this conclusion. With the type of experimental set-ups discussed in the section on literature survey such an absolute accuracy is not realizable. In almost every case the inside wall temperature of a tube is calculated from measurements of the outside wall temperature. This involves an accurate knowledge of the tube dimensions together with pertinent thermophysical data - namely, electrical and thermal conductivities and their temperature coefficients. The tube wall thickness is subject to some variations along the axial and peripheral directions. The measured thermal conductivity of stainless steels, commonly used in high pressure and temperature experiments, is truly chaotic. Figure 6, prepared by the author in 1962*, shows as an example the various measurements for the thermal conductivity of Type 347 stainless steel. The scatter is undoubtedly partly due to the quantities of various alloying elements in the specimen tested. At high heat flux levels considerable error may be introduced into the computations through faulty knowledge of such data. Appendix C discusses the effect of possible error in the tube parameters upon the inside wall temperature computations.

Hence, it is very possible that the value of such parameters as Prandtl, Grashof, and Rayleigh numbers, which contain several thermodynamic and transport properties subject to relatively appreciable

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* Data was obtained from the Thermophysical Properties Research Center (TPRC), Purdue University, Lafayette, Indiana.

uncertainties in the critical region, may be erroneously computed.

To clarify this point, let a unit uncertainty be associated with each property. The uncertainties associated with each of the above parameters and the Reynolds number will then be:

Reynolds Number	1 units
Prandtl Number	3
Grashof Number	5
Rayleigh Number	6

In the above it is assumed that the Reynolds number is computed from $\frac{GD}{\mu}$ where G is known with more certainty.

Briefly the method that is presented below is an attempt to accomplish the following:

1. simplicity and convenience of use in the near critical region;
2. satisfactory generality and range;
3. evaluation of the properties at a temperature which may be determined with higher accuracy.

The rate of heat transfer between a unit element of surface of a tube of circular cross section and the fluid flowing past that surface, at a location sufficiently removed from the inlet, is determined by four factors, viz.

1. the flow field;
2. the temperature field;
3. the state of the fluid; and
4. the tube diameter.

All the effects are interrelated or coupled. The thermodynamic state of the fluid may be described by two thermodynamic properties.** However, this is not sufficient to describe the transport properties, which are also a function of the molecular structure and the molecular field forces. The work of Fritsch and Grosh (28) reported that it was possible to correlate the thermal conductivity and viscosity data for water in terms of the specific volume and temperature.

The specific heat, viscosity, temperature, and the peak temperature were selected to define the state of the fluid. The peak temperature serves to provide a reference state. The effects of the flow field and the temperature field may be expressed by the heat transfer coefficient, mass flux, and the state of the fluid. Therefore,

$$\Phi(h, G, \mu_b, c_{p_b}, t_b, t_p, D, g) = 0 \quad (35)$$

In applying dimensional analysis techniques to the above equation the method of Buckingham's Pi theorem was employed*. The basic dimensions consisted of the dimensions of temperature, mass, time, length, and heat. The result is as follows:

$$\frac{h}{c_{p_b} G} = f(Re_b, \frac{t_b}{t_p}) \quad (36)$$

It is recognized that the term $\frac{h}{c_{p_b} G}$ is the well-known Stanton number. Although the above functional relationship seems to be a simple one, the inherent difficulty in it is the evaluation of c_{p_b} .

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* (29), pp. 42-55.

**Thermodynamic equilibrium is assumed.

The reader may correctly question the justifiability of using the thermodynamic property c_p in the above. The reasons are two fold. Firstly, specific heat plays an important role in heat transfer in the near critical region; secondly, it is an important element of the Prandtl number used in many correlations. In any case, the difficulty can be overcome in three ways, two of which follow from the definition of the heat transfer coefficient, h . Conventionally the heat transfer coefficient is defined as

$$h \equiv \frac{q_w''}{t_w - t_b} \quad (6)$$

Another possible definition is to use the enthalpy difference, as has been done in the high speed heat transfer problems, instead of the temperature difference. The new coefficient h_e which has the dimensions of $\frac{h}{c_p}$ is given by

$$h_e \equiv \frac{q_w''}{H_w - H_b} \quad (37)$$

The two heat transfer coefficients as defined above are related through an average specific heat function; thus

$$\frac{h}{h_e} \equiv \frac{H_w - H_b}{t_w - t_b} = c_{ph}$$

It may be recalled that Swenson, et al. used c_{ph} in their Prandtl number.

In a previous work (13) the author suggested that the Stanton number appearing in equation (36) may be modified either by introducing h_e to replace $\frac{h}{c_p}$ - viz. $St_e = \frac{h_e}{G}$, or using an averaged Stanton number

between two points on the heated section. This required that the average values of h and c_p be used. These were defined in the following way:

$$\bar{h} \equiv \frac{\bar{q}_w^*}{\bar{t}_w - \bar{t}_b} \quad (6a)$$

$$\bar{c}_p \equiv \frac{\Delta H_b}{\Delta t_b} = \frac{H_{b_{z+\Delta z}} - H_{b_z}}{t_{b_{z+\Delta z}} - t_{b_z}}$$

The superscript ($\bar{}$) indicates averaged quantities within an interval Δz .

A third method of disposing with the difficulty is to evaluate c_p from the smoothed enthalpy data. This is done by computing the slope of an isobaric (Hxt) versus H curve, since

$$\left. \frac{\partial (Hxt)}{\partial H} \right|_p = t + \frac{H}{c_p} \quad (38)$$

and using it to determine c_p from the equation

$$c_p = \frac{H}{\left. \frac{\partial (Hxt)}{\partial H} \right|_p - t} \quad (39)$$

The curve Hxt versus H is sectionally nearly straight. If accurate isobaric enthalpy-temperature data is available, then the computation may be readily performed on a digital computer.

Hence, the relation (36) may be either used in its original form or in the form of one of the two modifications. The three forms are:

$$St = f(Re_b, \frac{t_b}{t_p}) \quad (36a)$$

$$St_e = f(Re_b, \frac{t_b}{t_p}) \quad (36b)$$

$$\overline{St} = f(\overline{Re}_b, \frac{\overline{t}_b}{\overline{t}_p}) \quad (36c)$$

The specific heat used in (36a) may be obtained from appropriate tables or from a method outlined above and given by equation (39).

The Stanton number in (36b), defined by $St_e = \frac{h_e}{G}$, would probably be accurate in most regions. However, in the immediate vicinity of the peak temperature where enthalpy has a very sharp temperature derivative good accuracy in the determination of the wall temperature is required. On the contrary, the Stanton number as used in equation (36c) is immune from such danger. The enthalpy difference is actually the amount of heat input to the fluid in the interval Δz , and the fluid temperature can be evaluated from the measured (computed) fluid enthalpy. The process of calculating the temperature from the enthalpy in the region of rapid enthalpy variations is usually more accurate than the reverse process. It must be also noted that, while c_{ph} used in equation (36b) has been limited in magnitude for all practical purposes by the very nature of its definition, there is nothing in equation (39) that may cause such a limitation.

Before closing, several interesting features of equation (36) should be discussed. The recent correlations of forced convection flow in the subcritical and supercritical regions, using some form of the basic Nusselt equation, viz.

$$Nu = C Re^a Pr^b$$

have given values of "a" higher than 0.8. McAdams found that a value of 0.89 best fitted his results and Swenson et al. used a value of 0.923 in their correlation. This indicates that the Stanton number in equation (36) may be somewhat independent of the Reynolds number, or for restricted Reynolds number range

$$St = g \left(\frac{t_b}{t_p} \right) \quad (36d)$$

With $a = 0.923$, an order of magnitude change in the Reynolds number may be reflected as 19.4 per cent in equation (36d). Equation (36) is also compatible with other equations of the Nusselt type since it can be written as

$$St Re^{1-a} = f' \left(\frac{t_b}{t_p} \right) \quad (40)$$

where $f' \left(\frac{t_b}{t_p} \right)$ replaces other property dependent functions in conventional correlations.

Another correlation where $f' \left(\frac{t_b}{t_p} \right)$ was replaced by $g' \left(\frac{H_b}{H_p} \right)$ was tried. The results appeared satisfactory. However, the main difficulty was the determination of H_p , the isobaric peak enthalpy corresponding to the peak temperature. This and other points are discussed in a later section.

In the next section a general description of the experimental apparatus is given.

DESCRIPTION OF APPARATUS

The high flux heat transfer loop of the School of Mechanical Engineering sponsored by the United States Atomic Energy Commission was used for the acquisition of the data reported in this work. In order to render this work self sufficient a general description of the equipment is given in the following. However, for additional details on various pieces of equipment the reader should consult references (13, 30, 31, 32, and 33). The system is best described when various subsystems are treated separately.

1. The Main Loop

The essential constituents of the main circuit of the loop were a canned-motor centrifugal pump, obtained from the Chempump Corporation, a pump bypass, two venturi-meters, an electrically heated preheater, the test section proper, and an air-cooled heat exchanger-condenser which was connected to the main loop through a three way Annin valve. The pump rotor and bearings were immersed in the test fluid for cooling and lubrication. In order to maintain the motor windings and the gaskets below a maximum operating temperature of 450 F, cooling was provided through a shell in the pump housing. Figure 7 shows a schematic arrangement of the high flux heat transfer loop.

The pump outflow was fed into either of the venturi-meters for flow rate measurements. The smaller venturi was used for flow rates

FIGURE 7. SCHEMATIC ARRANGEMENT OF THE HIGH FLUX HEAT TRANSFER LOOP

less than $0.8 \frac{\text{lb}_m}{\text{sec.}}$. Each venturi-meter was equipped with a valve which was used for regulating the flow rate or shutting off the flow through it. The valves did not provide good regulation since they were only effective near the closed position. Better flow regulation was obtained by manipulating the pump bypass valve together with the venturi-meter valves.

Leaving the venturis, the flow entered the preheater where it was heated to the required test section inlet temperature. The old preheater, designed in 1957 and used in the earlier experiments described by Dix (30), Van Putte (31), Tokar (32), and Touba (13), was inadequate. The present preheater, fully described in (33), has its cartridge heaters inserted into wells that are immersed in the test fluid. The metal separating each cartridge from the fluid is only 1/16 inch thick. However, the contact resistance between a cartridge and its well, despite the close tolerances, was still large enough so that only about 15 kw could be used without endangering the cartridge heaters. This design also offered a short response time. At high flow rates the preheater could raise the system temperature to 360 F only.

From the outlet of the preheater the flow was directed to the test section which was connected to the loop piping through a pair of electrically insulated, water-cooled, heavy flanges. The lower flange assembly is shown in Figure 8. The flanges were designed to provide electrical insulation between the test section and the loop which was grounded. One of the flanges was welded to the system piping and the other, connected to the test section, was bolted to the first flange

FIGURE 8. LOWER FLANGE ASSEMBLY

with a sheet of "Duralba" high temperature asbestos gasket material inserted between them. The inner faces of the flanges in contact with the gasket were serrated to provide a strong grip. Insulation material was also inserted around the bolts and under the washers. To minimize heat loss from the hot test fluid to the heavy pressure flanges a standpipe was welded to the flange on the test section side. The test section tubing was then inserted into the standpipe and welded in concentric arrangement. The space between the tubes was filled with loose insulation. When assembled, the standpipe projected into the loop piping allowing a volume of stagnant water to form around it. This provided a further thermal insulation between the test section and the flange.

In his trial runs, Van Putte (31) had encountered a severe problem with the flanges. During periods of warm-up, due to uneven thermal expansion rates of the flanges and their bolts, thermal stresses caused a failure in the gasket material between flanges leading to dangerous and damaging short circuits. Very long warm-up periods of around 12 hours were required to eliminate the difficulty. To remedy this problem, water cooling was incorporated into the flange design and the standpipe arrangement mentioned above was used to reduce heat losses.

Any vertical test section, properly designed and instrumented, may be installed between the upper and lower flanges. The loop design also provides for easy conversion for use in conjunction with horizontal test sections. The details of the test section are discussed in part 4 of the present section.

Downstream of the test section a three-way valve was provided for the purpose of regulating the pump and the test section inlet temperatures. Any portion of the test fluid could be channeled through the air cooled heat exchanger-condenser. The outflow of the heat exchanger joined the fluid which bypassed it via the valve, and the combined streams were directed into the pump inlet. Due to this arrangement relatively good control over the pump inlet temperature could be obtained. However, the operation of the heat exchanger-condenser presented other problems which are mentioned in the next section.

The heat exchanger was cooled by a motor driven blower capable of driving up to 10,000 CFM of air past its coils. Adjustable vanes and shutters were provided at the air intake and exhaust. At times, a fan placed close to the blower intake was used instead. The blower was rarely used.

The main loop also accommodated two bladder type accumulators of $2\frac{1}{2}$ and 5 gallons capacity. The water side of one or both accumulators could be connected to the pump outlet or any other location in the loop. The gas side was charged with nitrogen to a pressure at least 25 per cent of the operating pressure. Pressurization on the nitrogen side could also be used to increase the system pressure. The line connecting the accumulator(s) to the pump outlet or system was made long in order that the hot fluid from the loop would reach it at a safe temperature. Otherwise, the Buna N synthetic rubber bag could be damaged. For this reason the water line to the accumulator(s) was also connected to the cold test fluid in the auxiliary loop which is

discussed next. A valve arrangement allowed switching to the cold fluid when rapid heating was in progress, while for steady state operations the connection to the pump outlet was preferred.

2. The Auxiliary Loop

In order to maintain the high purity of the test fluid an ion exchange circuit was incorporated into the system. This helped prevent scale formation and assures reproducibility of data. Approximately five per cent of the system flow continuously passed through an ion exchange bed. Water resistivity measurements could be taken at the inlet and outlet of the ion exchange bed.

The system water, bled at the circulating pump outlet, was passed through a regenerative heat exchanger which transferred heat from the hot water leaving the main loop to the deionized fluid returning to the pump inlet. After further cooling by a single-pass counterflow tap water heat exchanger, the fluid passed through a remote-controlled valve into the ion-exchange bed. This system was made necessary by the fact that the ion-exchange resin operating temperature was limited to 140 F. A Gardsman controller was used to guard against temperature excesses. If needed, the Gardsman unit, sensing the water temperature through a thermocouple located at the ion-exchange inlet, could automatically engage an air operated valve which would stop all flow through the ion exchanger.

Rupture disks were provided in both the hot and cold loops. Pressures exceeding the system design pressure were thus relieved. Finally, the loops were mainly constructed of one inch, schedule 160, type 347 stainless steel pipe, with proper consideration for thermal expansion.

3. The Filling and Vacuum Subsystems

The purpose of the filling subsystem was two fold: (1) to remove the impurities in the tap water which cause scale formation and corrosion, and (2) to remove the dissolved gases from the water. Figure 9 shows a schematic of the filling system.

Two ion-exchange beds were provided. The function of each was to absorb the metallic and non-metallic ions present in tap water liberating the equivalent in hydrogen and hydroxyl ions respectively. Tap water entered the larger bed packed with a lower-melting point resin* (105 F) and flowed into a 30-gallon stainless steel reservoir which was heated by a natural-gas flame.

The water to be admitted into the system was boiled for about two hours. It then passed through a tap-water cooled heat exchanger and into a smaller ion exchange bed with a melting point of around 140 F**. This was to insure further purity and avoid possible mixing of degassed water with undegassed water in the larger ion-exchanger if the latter were used for the second purification pass.

Another feature of the filling subsystem was an air-driven positive displacement pump. The pump was used for initial pressurization of the system, and for system pressure control during the experimental runs. The water from the second deionizer entered the system via the positive displacement pump.

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* Monobed MB-3, Enley Products Inc., N. Y.

** Monobed MB-1, Enley Products Inc., N. Y.

FIGURE 9. SCHEMATIC OF THE FILLING SYSTEM

To obtain higher purity water an air-driven recirculation pump was incorporated into the filling subsystem. The pump was operated at all times except when water was being boiled. The water from the reservoir was drawn into the pump through the heat exchanger, and forced over the larger ion-exchange bed. Under this procedure, the resistivity of the water in the reservoir consistently measured at better than 3,000,000 ohm-cm.

Prior to filling, the system was evacuated for about three hours. A vacuum pump was connected to the highest point in the system through a reservoir which received two lines from the system piping. One line (1/2 inch OD) was an integral part of the system and served both the vacuum subsystem and a rupture disk. The other line, an ordinary vacuum hose, was connected to a "Y-like" joint, installed at the highest point in the system, which also received the 1/2 inch stainless steel line.

This arrangement accomplished two objectives: (1) it provided a sufficiently large passage to the vacuum pump, and (2) it insured against air pockets within the operational system. A satisfactory evacuation and filling manifested itself in the fact that only very few strokes of the positive displacement pump were needed to raise the system pressure above atmospheric.

The vacuum pump was allowed to run while water was being admitted into the system. Rising water level in the reservoir indicated that the system and the $\frac{1}{2}$ inch stainless steel line were full. The valve connecting this line to the pump was then shut off, while the pump operation through the vacuum hose continued. An instant later the hose was also disconnected and the vacuum pump was switched off. It should be noted that since the system could never be drained completely a vacuum of better than 29.3 inches of mercury was rarely obtained.

4. The Test Section

For the purposes of the experimental study reported here two test sections were prepared and instrumented. Table III supplies some pertinent information on these test sections. Tube wall thickness was measured by cutting axially through a six inch length of the tube, cut from one end of the test section tubing, and taking micrometer readings on each half.

TABLE III

Test Sections Used in This Work

	Small	Large
Code Name	XE	XF
Outside Diameter	0.3762 inch	0.5010 inch
Inside Diameter	0.2446 "	0.3698 "
Wall Thickness	0.0658 "	0.0656 "
Heated Length	52.0 "	56.0 "
Distance between pressure taps	71.0 "	72.0-74.0"
Material	SS. Type 304	SS. Type 304
Manufactured	Drawn Seamless	Drawn Seamless

Circular current flanges were silver-soldered to the test section. The current flanges measured 7 inches in diameter and had two symmetrically located wings to which heavy flexible power leads were bolted. Two circular guard heaters, producing 1250 watts together, were concentrically clamped to the outer face of each flange. Short pieces of Nichrome wire were used for power connection to the heaters since copper wires became badly corroded at elevated temperatures.

In early operations of the high flux heat transfer loop the test section was an integral part of the heavy, water cooled, isolation flanges. This meant that every test section change required a reconstruction of the heavy flanges, the cooling system, the pressure taps, and the immersed thermocouples used for bulk fluid temperature measurements. Since in conjunction with the above a number of silver-soldered joints were also used, many frustrating leaks developed. During the work of (13) a new and efficient design was adopted and carried to the present work.

The water cooled flanges, the primary function of which was to provide electrical insulation between the test section and the rest of the loop, together with the pressure taps and the immersed thermocouples were installed on a permanent basis.

The above arrangement served two important functions as follows:

1. The test section could be dismounted or mounted readily;
2. different test sections could be prepared and installed without any welding or soldering.

Welded joints were used throughout the system since most soldered joints failed after several hours of operation giving rise to many difficulties. It is likely that this was due to the formation of some

hydrogen atoms which penetrated the weak points of the solder and caused the formation of miniature holes.

Figure 10 allows a comparison of an early test section and the small diameter test section (XE) of this investigation. The older test section is seen with heavy, uncooled isolation flanges, pressure taps, large square current flanges, and guard heaters installed on both faces of the current flanges. It is bulky and end-heavy. The test section XE is seen with its current flanges and thermocouples installed. The dark spots on the current flanges indicate the location of the guard heaters. The center thermocouple was removed for post-testing calibration.

A special fitting which housed a 1/16 inch Conoclad stainless steel sheathed chromel-alumel thermocouple and a pressure tap was installed at the test section inlet. Inlet pressure and temperature data were obtained through this unit. Another pressure tap was installed immediately at the test section outlet. However, in order to obtain a better mixing cup temperature, the test section exit temperature was measured at a distance of about twelve inches from the exit. To stimulate mixing a coiled length of stainless steel wire was inserted in the tube.

Figure 11 shows the test section outlet. The guard heaters clamped in position on the current flange, the outlet pressure tap, the immersed thermocouple fitting, and the water-cooled isolation flanges are seen. Part of the vacuum and electrical systems are pictured to the left.

FIGURE 10. COMPARISON OF OLD AND NEW TEST SECTION DESIGNS

FIGURE II. TEST SECTION OUTLET

The system and the test section were insulated with double layers of Johns Manville high temperature insulation. Glass wool insulation was used around guard heaters and elsewhere.

5. Instrumentation

The instrumentation of the system may be grouped under three broad categories of power supply and control, the safety interlock circuits, and the measurement devices. Thermometry is discussed in part 6.

Two independent power supplies were available. Single phase, 60 cycle, 2400 volt electrical power was supplied to the test area and stepped down to 240 volts by a power distribution transformer. This voltage was delivered to a water-cooled welding transformer, capable of supplying the test section with voltage from 4.2 to 50.7 volts via a saturable core reactor-transformer. The reactor-transformer was controlled by a magnetic amplifier with which the output could be varied from 10 to 95 per cent of the line voltage.

The saturable core reactor-magnetic amplifier combination, used in conjunction with a Leeds and Northrup Type H Speedomax Recorder, was used to maintain constant power delivery to the test section. The controller worked through a thermal converter to which the test section voltage and current were delivered via appropriate potential and current transformers. The actual working was as follows. The thermal converter signal was evaluated by the Speedomax Recorder which fed a Current Adjusting Type Controller a signal proportional to the difference between the actual and the "set" powers. A DC signal from the

controller to the magnetic amplifier was amplified and delivered to the reactor-transformer. Therefore, the test section power was controlled by the DC current which determined the degree of core saturation in the reactor-transformer. Power available for test section use was 167 KVA.

The preheater was supplied with 230 volt, 3 phase power. Maximum available preheater power was 22.5 KVA. A separate control system similar to the test section power regulation was also available. This, however, was designed to control the test section inlet temperature. The essential difference between the test section and the preheater controls was the fact that the thermocouple in the preheater control system replaced the thermal converter used in the test section power control.

Both systems operated satisfactorily, but the lack of preheater power and the temperature limit on its cartridge heaters did not allow extensive use of the preheater controller. This system, in conjunction with a joulean heated preheater, would give very satisfactory results. Copper buss bars and heavy braided copper leads connected the welding transformer to the test section. The power input to the main saturable core reactor-transformer was via a 2 pole, 800 ampere, 50,000 amperes interrupting, air circuit breaker.

The safety interlock system was designed to prevent injury to personnel and damage to the test apparatus. It consisted of an electrical circuit which had a set of relays in series with the holding coils on the preheater and test section power supply lines. When any relay in the interlock system was opened, both power supply circuits were

disconnected and a light on the panel turned off to indicate the trouble spot. The components of the system were as follows.

1. A Gardsman* high limit controller, activated by a thermocouple, monitored the test section outside wall temperature. If the temperature exceeded a preset value, test section power was interrupted.
2. The power lines in the circulating pump were wired with two relays in the interlock system. Thus, the test section power is disconnected in the case of pump power failure or pump motor temperature excess. The relays could be disconnected for pure free convection experiments.
3. A micro-switch, activated by a vane in the air cooled heat exchanger and wired to a relay, provided safety against failure in the blower. When air flow was not required, a switch on the instrument panel disarmed this safety.
4. A Weston 705 relay was available for use in conjunction with the preheater. This device could be set to interrupt the system power when cartridges exceeded 1000 F. Instead, in the present investigation a chromel-alumel thermocouple installed in a groove cut into a cartridge was connected to a Micro-Max recorder and the cartridge temperature was monitored by the operators. This eliminated power interruptions under maximum safe preheater power which resulted in cartridge temperatures of a little over 1000 F.

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* West Instrument Corporation.

5. A high-low pressure limit gage* wired to a relay interrupted power when the system pressure reached a preset maximum or minimum allowable pressure. This gage could be used to prevent overpressures above the system design pressure, or underpressure which could lead to burn-out.
6. A big, red switch on the instrument panel served as a "panic button". It, too, was wired to a relay which interrupted all heating power.

A six channel Micro-Max recorder was wired to provide constant monitoring of certain critical temperatures. Three channels, every alternate one, were connected to a thermocouple immersed in the test fluid near the pump inlet. Other channels were assigned to lower and upper water-cooled insulation flanges, and to the pump motor casing. These insured against accidental cooling-water failure to these areas.

The measurement devices may be divided into power (current and voltage), pressure, and temperature measuring equipment. Below, a short account of related devices used in this investigation is presented.

Previously, power measurements were made by GE P3 type volt, ampere and watt meters. The test section power supply Speedomax Recorder readings could also be used for comparison. However, when during the work of reference (13) the test section power voltage wave form was photographed, it was discovered that the input voltage was not sinusoidal. This could be attributed to the presence of the saturable core reactor-transformer in the power circuit. A Fourier analysis of

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* Obtained from the United States Gage Co.

photographs taken at a power input of 45.7 kw revealed that the voltage wave contained 10.4 per cent third harmonic and 19.0 per cent fifth harmonic with respect to the first harmonic. It was concluded that the meters designed for sinusoidal wave forms and for a frequency range of up to 133 cps could be in error.

Later a Greibach Model 700 AC-DC volt-ammeter was installed which was guaranteed to give true RMS voltage and current readings within one half of one per cent. Comparison of the measurements taken from the older meters and the new indicated that the voltmeter was the only source of error.

The Greibach volt-ammeter was connected across the test section for voltage measurements and across a 240/1 ratio Weston Model 347 current transformer for current measurements. The P3 volt and watt meters and the Speedomax Recorder were connected across the test section through a potential transformer which allowed meters to be operated within their midscales. Current was supplied by a separate 240/1 ratio current transformer. A current transformer of 600/1 ratio was available for higher current measurements.

For pressure measurements two large pressure gages, two differential pressure transducers, and a specially designed high pressure manometer were used.

A Crosby 0-4000 psia Hydrostatic test gage was connected to the gas side of the accumulator(s). The accumulator pressure could then be always monitored whether or not it was connected to the loop. A 0-5000 psia Heise gage with a least count of 10 psia was connected

to the system at the circulation pump output line and the test section inlet. A valve arrangement allowed selection of either location.

Two Statham* $\pm$ 40 psi differential transducers were used in the system. One was permanently connected across the test section. The other, and an especially designed mercury manometer, could be connected across both the large and small venturi-meters and the test section through an arrangement of Aminco high pressure needle valves. The transducers were operated by Model BAM-1 Bridge Amplifier Meter supplied by Ellis Associates. Bridge output was monitored by its own meter or auxiliary equipment - including a two channel Beckman Offner Type RS Dynograph or a Type 502 dual beam Tektronix oscilloscope equipped with Polaroid camera attachment. Pressure lines were connected to the test section pressure taps through two isolation flanges using mica sheets.

The high pressure manometer, designed and fabricated at Purdue University, employed stainless steel tubes which contained the mercury. The mercury level was determined by locating a steel ball that floated on the mercury column. For this a special centering coil energized by a Hewlett Packard Model 200AB Audio-Oscillator was used. A Hewlett Packard Model 400D Vacuum Tube Voltmeter indicated the output. The manometer scale was engraved at 0.01 inch intervals.

The pressure gages and one transducer were calibrated with a dead weight tester. The test section pressure drop transducer was checked against the calibrated transducer. The transducers were also

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* Operational pressure: 0-5000 psi; temperature compensation: 0-200 F.

calibrated against the manometer. Excellent agreement to better than 0.03 inch of mercury existed. The calibration of all equipment was checked regularly, and all exhibited near perfect stability. The Heise gage scale permitted readings with an accuracy of 5 psi.

For temperature measurements a Leeds and Northrup Type K3 Universal Potentiometer in conjunction with a Leeds and Northrup D.C. Null Detector and 099034 Constant Voltage Supply by Dynage was used. The thermocouples were connected to the potentiometer by Leeds and Northrup switches.

6. Thermometry

Each test section was instrumented with a number of thermocouples. Test sections XE and XF had respectively 29 and 27 thermocouples attached to their heated span. Furthermore, two thermocouples were mounted on the collar of each current flange. These were used to regulate the guard heaters. All thermocouples were 28 gage chromel-alumel wires with double fiberglass insulation, covered by a braided tinned-copper shield, and jacketed in Nylon. Premium grade Thermo-Electric wires were used.

The thermocouples were installed by forming each wire around the tube wall and carefully spot welding it to the tube metal. Therefore, the junction of the thermocouple was made through the metal and heat conduction was eliminated by the isothermal length. Previous experience had shown that this was a safe method. The junction was then wrapped with glass tape and the wires were led out to a wooden bar to which they were fastened. See Figure 10.

Additional wires were installed on the two isothermal sections of the tube at the inlet and outlet. These were used to provide a check for the immersed thermocouples; 1/16 inch conoclad stainless steel sheathed chromel-alumel thermocouples were also installed at the inlet and outlet of the preheater. Other thermocouples were attached to the large venturi outer wall.

All thermocouples were properly shielded and led into an isothermal junction box. From the box to the thermocouple switches copper wire was used. All wires were shielded and all shields were grounded.

A cold junction was made by inserting a spot welded thermocouple wire into kerosene in a thin glass tube. The tube was placed in a continuously stirred ice bath.

Figures 12 and 13 show an overall view of test sections XF and XE respectively. The location and number of thermocouples is properly indicated. Figure 12 also illustrates the cooled flanges and the special fluid temperature and pressure measuring units at inlet and outlet. The details of the wiring of a thermocouple are shown in Figure 14.

Thermocouples were calibrated against the freezing points of tin, lead, and zinc. The metals were standard samples obtained from the National Bureau of Standards. They were placed in graphite crucibles and were covered by a layer of graphite powder. A bored graphite rod inserted through a hole in the lid contained the thermocouple. The crucible was heated in an electrical radiation furnace.

Two test section thermocouples together with preheater and test section inlet and outlet immersed thermocouples were calibrated. The

FIGURE 12. OVERALL VIEW OF 1/2" O.D. (XF) TEST SECTION

FIGURE 13. OVERALL VIEW OF 3/8" O.D. (XE)
TEST SECTION

FIGURE 14 THERMOCOUPLE WIRING DETAIL

latter were found to be indicating up to 6.85 F higher than the true temperature. The test section thermocouples were off by less than 1 F.

Finally, Figure 15 gives a view of the instrument and control panel, and Figure 16 shows the loop with the test section installed. In Figure 16 the installation of the test section, the thermocouples, pressure taps, and the water-cooled isolation flanges may be seen. Power supply equipment is to the left. In the background the air-cooled heat exchanger, and the large flanges and the tube of the system ion-exchanger are visible.

FIGURE 15. VIEW OF THE INSTRUMENT AND CONTROL PANEL

**FIGURE 16. VIEW OF THE MAIN LOOP AND TEST
SECTION**

EXPERIMENTAL PROCEDURE

Experiments were conducted to determine the characteristics of heat transfer to near critical water flowing turbulently through an electrically heated tube. Upward forced convection flow was produced by the system circulation pump. Free convection effects were expected to be present in the regions where the density gradient with temperature became large.

At the start of a series of tests, the system was evacuated and filled with de-ionized, degassified water. The accumulator was charged with nitrogen to a pressure ranging from 1500 to 2000 psia depending upon the supply pressure available. The system pressure was then raised to around 1500 psia by the positive displacement pump.

Before power was activated, all system components such as the safety relay circuits, recorders, thermocouples, and the interlock system were carefully checked and calibrated where necessary. A strict start-up and shut-down procedure was followed. Every step on a special check list was carried out. These checks included:

1. The test section resistance to ground was measured to insure against possible shorts.
2. All electrical equipment and auxiliaries were turned on to allow a sufficient warm-up period.
3. All cooling water was turned on.

4. The system circulation pump was started, and the flow rate and test section pressure drop were measured using the appropriate pressure transducers.
5. Current transformer leads were checked.
6. All equipment was standardized.
7. The panic button was reset, and the preheater was activated and adjusted to a safe initial power.
8. Finally, the test section power was activated, the magnetic amplifier was turned on, and the automatic control system was switched on to maintain a constant power input to the test section.

All personnel participating in the test runs were trained and instructed as to the conduct of the test and the necessary action to be taken in the event of an emergency.

The system pressure was usually maintained at a value above the saturation pressure of the prevailing highest test section temperature to prevent possible burn-out. Only when it was desired to get into the boiling range to check thermocouples was the pressure lowered to the saturation value. The system temperature was raised gradually in order to avoid leaks in certain units such as the valves. However, the problem of leaks was more serious during the shut-down phase. The three way valve almost always let off steam when temperatures were being lowered.

The power input to the test section was gradually increased until the pressure reached the test range. The needle valve of the line connecting the cooler section of the auxiliary loop and the accumulator was then adjusted so that excess water due to volumetric expansion

flowed into the accumulator while the system pressure was maintained at an almost constant value. When the temperature approached the desired range, the flow through the air cooled heat exchanger and the power input at the preheater were adjusted to obtain the desired fluid temperature at the test section inlet. Heat flux could be adjusted easily; however, regulation of the flow rate and the test section inlet temperature was not as easy. The system pressure was maintained by occasional bleeding of water from the system or pumping water into the system by the positive displacement pump. The controls for these units were located near the Heise pressure gage. System pressure control by bleeding from or adding nitrogen to the accumulator was not practical for the reason that commercial nitrogen cylinders are pressurized to only about 2300 psi, and that no such fine pressure regulation was needed.

An effort was made to minimize bleeding of water from the system at elevated temperatures. This would not only save the pure water in the system, but would also help the positive displacement pump to maintain the system pressure in the event of rapid temperature decreases. This could be realized by starting with a partially pressurized accumulator and system such that very little bleeding would be necessary as the system reached its operating temperature. Further small increases in the system pressure could be adjusted by additional cooling in the heat exchanger and heating in the preheater. The scheme was satisfactory only when such adjustments could be made with rather limited preheater power. However, the regulation of the system pressure did not actually present much difficulty.

The main difficulties were the regulation of the flow rate through the test section, and the test section inlet temperature. Due to the construction of the venturi valves very little flow regulation was possible. However, a combined operation of the circulation pump bypass valve and the venturi valves could produce the desired effect.

The fluid inlet temperature at the test section was controlled by several variables - namely, the rate of flow through the heat exchanger, changes in the ambient temperature, drafts in the test area, and the power capacity of the preheater. The heat exchanger with its large surface area and exposed exterior was extremely sensitive to drafts and ambient temperature changes. A small decrease in the room temperature or a draft caused rapid pressure drops as the density of the fluid in the heat exchanger increased. This was also followed by a simultaneous drop in the test section inlet temperature. The test area was located in the vicinity of a gate and very close to traffic. During the winter months attaining of a steady state in operations was extremely difficult since an opening of the gate caused an inrush of cold air and subsequent rapid drop in system pressure and temperature. This ruined many otherwise good experimental runs. At other times, steady state was difficult to maintain during some period after the sunset.

With the heat exchanger blower in operation steady state was difficult to maintain. Also, the blower produced too much cooling. Hence, the heat exchanger was operated with natural air circulation, or rather by using a small fan to produce a steady draft.

Due to the insufficient capacity of the preheater the test section heat flux was governed by the inlet temperature desired. For this reason heat fluxes lower than about $150,000 \frac{\text{B}}{\text{hr ft}^2}$ could not be permitted since lower heat fluxes would not give sufficiently high inlet temperatures. Another element in favor of high heat fluxes was the necessity to obtain sufficiently large surface-to-fluid temperature differences such that the percentage of error would be decreased. This point is taken up in Appendices C and J.

The system reached steady state conditions at near the critical point around three to four hours after start-up. For reasons discussed before, faster heating was avoided. The heat flux was set for automatic regulation, and the flow rate was adjusted. The operators then undertook to attain the desired inlet temperature by manipulation of the preheater. The heat exchanger was not used unless its use became necessary. The system pressure was also regulated at the desired level.

A data run was initiated when the existence of steady state conditions was assured through the following observations:

1. The test section inlet and outlet temperatures remained constant for at least five minutes.
2. The venturi pressure drop remained constant with time.
3. The fluid temperature at preheater exit and the pump inlet remained steady.

In the absence of a rapid potential measuring device, a Wollensak tape recorder equipped with foot-controls was used to speed data acquisition. This also afforded considerable savings in part time

help, and decreased confusion by reducing the number of operators to two. Another feature was that verbal information describing the condition of various variables during the test runs could be fed into the tape, to be used in a later evaluation of the test. Operator A was assigned only the duty of reading the thermocouples and operating the tape recorder in conjunction with that. Operator B was in charge of maintaining the system pressure, and making power, flow rate, and pressure drop measurements. In between data runs operator A regulated the test section guard heaters.

When the conditions were favorable the data run was started by reading into the magnetic tape the test run number, date and the time. Operator A then followed the following sequence in measuring and recording the thermocouple potentials:

1. all thermocouples associated with the run, but not needed for computations;
2. the venturi, preheater and test section inlet and outlet thermocouples;
3. all thermocouples installed on the heated span of the test section;
4. repeated section 2 and recorded the finish time.

In the meantime operator B measured and recorded in writing the test section and preheater voltage and current, Speedomax power reading, preheater outlet temperature as it appeared on the recorder, the test section pressure drop, guard heater settings, system pressure at the test section inlet, and the venturi pressure drop as indicated by the

transducer. If the venturi pressure drop was less than 10 psi*, it was also measured by the manometer. Additionally, later it became necessary to obtain a time record of the test section and venturi pressure drops indicated by the transducers. For this the Offner recorder was used. This subject is introduced in the section entitled Experimental Observations.

The average time required for a data run was around 17 minutes. If the system had not maintained its steady state condition during this period of time the data were discarded. Otherwise, the system parameters were changed and a new steady state was awaited. One half to one hour or more was required, depending upon the many variables which determined the process.

Altogether 73 sets of data were collected, 31 of which were taken with the XE test section and 42 with the XF. In order to check the measurements against the well established correlations some data were taken at 3000 psia and a bulk fluid temperature range of 400-530 F. Separate experiments were conducted for isothermal friction drop measurements. Most of the data were taken at pressures ranging from 3220-3400 psia, and heat fluxes of 150,000 to 525,000 B/hr ft².

At the end of a series of data runs the system was shut down. This involved a reversal of the start-up procedures. Flow of cooling water to the welding transformer, isolation flanges, and the pump casing was maintained for about three hours after shut-down. System pressure was allowed to decrease to atmospheric pressure. The time elapsed between start-up and shut-down ranged from 10 to 22 hours.

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* The manometer did not accommodate higher pressure drops.

EXPERIMENTAL OBSERVATIONS

During the course of the work various trial experiments were performed to check the operational integrity and reliability of the system. These tests included the determination of isothermal friction factor for the test sections, heat balance tests, and low temperature (400-560 F) runs where agreement with well established heat transfer equations could be checked. Such basic tests are a prerequisite to any investigation of this kind.

The isothermal friction factor tests led to two strange effects as follows:

1. the pressure transducer connected across the test section indicated that the pressure drop across the test section was not steady;
2. the results of the friction factor measurements indicated that the friction factor for the large tube (XF) leveled off at around $Re = 1.5 \times 10^5$, a trend characteristic of rough tubes.

Similar behavior as in "1" above was also observed on the differential pressure transducer connected to the large venturi, although the amplitude of the fluctuations was smaller. Since these observations were made at around the 400 F level, it was decided to check the behavior at room temperatures with absolutely no heating, and with heating at temperatures where future experiments were to be run.

The result of these observations created great concern. It was found that the fluctuations in the pressure drops across both the test section and the venturi persisted at room temperatures and that their frequency increased with pressure. In the near critical condition the test section pressure drop fluctuations amplified several times. Figures 17 and 18 show Dynograph traces of the pressure drop fluctuations. No fluctuations were observed in the static system pressure at any time. Undoubtedly, the nature of fluctuations in the near critical region was greatly influenced by the fluid density as water became more compressible.

Several months were spent trying to determine the origin of the fluctuations and/or to remedy the difficulty. Although some insight was gained, the fluctuations could not be stopped. Appendix K is devoted to this matter where the procedures and the results of the endeavor are briefly outlined and discussed.

At this point the reader must be cautioned against hasty conclusions. Similar fluctuations may have been present in a great many investigations, and may have gone undetected. Neither the Bourdon tube pressure gage nor the mercury manometer were capable of detecting the fluctuations. Their construction did not allow for sufficient dynamic response. In investigations where such devices are used for monitoring pressure drops, fluctuations of the type reported here can not be observed. Hence, the situation is probably not unique to this investigation. In the section discussing the results it is shown that despite these perturbations there was good agreement between heat

FIGURE 17. DYNOGRAPH TRACES OF PRESSURE DROP FLUCTUATIONS AT 300 F, 1900 PSIA

RUN XE 28
 TEST SECTION
 PRESSURE DROP FLUCTUATIONS

$P = 3400 \text{ PSIA}$
 $t_{\text{inlet}} = 698 \text{ F}$
 $\dot{m} = 810.3 \text{ lbm/hr}$

FIGURE 18. TEST SECTION PRESSURE DROP FLUCTUATIONS FOR TEST XE 28

transfer measurements at 3000 psia, in the 400-560 F range, and the Dittus-Boelter equation.

Returning to the case of the friction factor measurements, at first it was thought that the leveling off could be attributed to the pressure drop fluctuations on the hypothesis that the perturbations may have caused the tube to become hydraulically rough. To check the instrumentation and the computations a separate set-up was arranged for friction coefficient measurements in a tube similar to the XE test section. The flow rate was computed from time and weight measurements. The pressure drop across the tube was obtained from both a mercury-in-glass manometer and the differential pressure transducer. These readings were always in complete agreement. Since the tap-water line pressure was not constant and could not be depended upon, a pressurized water reservoir arrangement was employed. The tank held about eight gallons of water and was pressurized with air. Two to four gallons were allowed to flow for each run. The arrangement was found to give very steady flow. Further, the pressure drop across the tube as monitored by the transducer and recorded by the Dynograph did not show any fluctuations or drift.

The results of these measurements fell in the range $Re = 1 - 5 \times 10^4$ and were in good agreement with the Prandtl universal friction factor curve for smooth tubes. Although the limited range did not allow for any conclusions to be drawn in regards to the flattening effect, the correctness and the reliability of the transducer measurements were indeed established.

Low temperature heat transfer experiments showed a deterioration in the heat balances as the flow rate was decreased. The same trend was also observed when the experimental heat transfer coefficients were compared with values computed from the Dittus-Boelter equation. The trouble was successfully linked to the Annin valve on the small venturi meter. It was found that the leak was caused by the thermal expansion of the valve and increased linearly with temperature.

With this difficulty removed, very good heat balances were obtained and the low temperature data agreed well with the Dittus-Boelter equation at all flow rates.

Expecting that the valve leakage had also influenced the computations for the tube friction factor, further tests were carried out, some with flow straighteners (described in Appendix K) installed and the lower pressure tap located on the test section proper, downstream from the straighteners. However, no improvements were noted but all measurements showed excellent reproducibility of the measured friction factors. There is, therefore, no doubt that the XF test section tube did act rough. The results of the measurements and an explanation for this behavior is presented in a later section.

Another problem encountered during the investigations was the general deterioration in the heat balances near the critical region. Table IV gives a general appraisal of the heat balances for various inlet and outlet temperatures. Up to 650 F at the inlet good heat balances, within one and one half per cent, were obtained. The situation deteriorated to ten per cent for the inlet and outlet temperatures below 675 F and 700 F, respectively. At higher temperatures the situation immediately grew worse giving very poor heat balances in the 700-710 F

TABLE IV

Discrepancy in the Heat Balances

Condition		Heat Balance Error
Inlet	Outlet	
$t_b \leq 650 \text{ F}$	$t_b \leq 675$	good (less than 1.5 per cent)
$650 \leq t_b \leq 675$	$675 \leq t_b \leq 700$	fair (within 10 per cent)
$675 < t_b$	$700 < t_b$	poor (some more than + 50 per cent)
$706 < t_b$	$715 \leq t_b$	poor (negative errors)

range for the exit fluid temperature. Some heat balances were off by more than 50 per cent, with the generated heat exceeding the product of the flow rate and the enthalpy rise between the inlet and outlet. At the inlet temperatures above approximately 706 F the discrepancy was in the opposite direction.

A general deterioration of the heat balances with the approach to the near critical region has been reported by many investigators. This seems to be another riddle associated with the problem of heat transfer to near critical fluids. Koppel's results were an exception primarily because the same equipment and set-up were used to measure the enthalpy of carbon dioxide in the critical region prior to the heat transfer tests.

There are essentially four basic factors which may influence the heat balances:

1. Error in the flow rate.
2. Thermocouple calibration error.
3. Thermocouple not measuring the correct bulk fluid temperature.
4. Discrepancy in the enthalpy tables.

The fluid specific volume appears in the venturi equation and is the only element of that equation which may possibly cause an abrupt discrepancy in the flow rate. However, for the flow rate to be, for example, twenty per cent off, the value of the specific volume must be off by over forty per cent. Even close to the steepest section of the isobaric specific volume-temperature curve, this would require 2 to 3 F error in the temperature. Since the fluid immersed thermocouples were calibrated at the lead and zinc points, such a large temperature error was improbable.

The more probable causes of the discrepancy were items 3 and 4 above. Despite the fact that a twelve inch length of tube was provided at the outlet from the heated section to insure proper fluid mixing, it was still quite possible that the centrally located thermocouple did not measure the true mixing-cup temperature. It is also possible that the presently available enthalpy data have a major discrepancy in the immediate vicinity of the critical region. Since the test section inlet temperature was known with better certainty, all computations were based on the inlet temperature. The presence of three fluid immersed thermocouples, two of which belonged to the pre-heater, together with several other thermocouples mounted on the piping

and undoubtedly better opportunity for fluid mixing contributed to this decision. Furthermore, considering the facts that fair heat balances were obtained up to 700 F at the exit, and that the fluid inlet temperature was usually less than 700 F, it was felt that the use of the inlet temperature was the safer of the two.

Certain instabilities were noted in the thermocouple readings. It appeared that the temperatures fluctuated within a few tenths of a degree from the mean value. The fluctuations in the wall temperatures were believed to be due both to the test section power fluctuations and probably to the fluid fluctuations discussed previously. The fluid temperature fluctuations were due to the latter. The fluctuations seemed to decrease as the critical region was approached.

No system pressure instabilities of the type reported for free convection studies were observed. It may be hypothesized that these instabilities have their origin in the small flow fluctuations observed in this work. However, at lower fluid flow rates, as is the case in free convection flows, there is insufficient damping and the perturbations grow. As the fluid becomes more compressible, the amplitude grows and the lower frequencies become more distinct. An example of such a behavior is given in Figure 18 where large differential pressure drop drifts may be seen. It was also observed that at higher flow rates there was a damping effect and much lower amplitudes prevailed. Hence, at low flow rates the amplitudes may become quite large.

REDUCTION OF DATA

All satisfactory experimental data obtained during the course of the investigation were reduced to obtain the system flow rate, local wall and bulk fluid temperatures and enthalpies, and local heat fluxes. From these basic data the local heat transfer coefficients and other parameters of interest were computed. In the following the equations and procedure used in the reduction of data, and other calculations of interest, are outlined.

1. Flow Rate Calculations

The small and large venturi flow rates were computed from the general venturi equation, viz.

$$\dot{m} = AKE \sqrt{\frac{2g\Delta P_v}{v}} \quad (41)$$

Upon substituting the proper constants in equation (41) and accounting for the water leg in the manometer specific equations for each venturi were obtained. Thus,

$$\dot{m}_s = 0.00435 K_s E \sqrt{\frac{\Delta z_{vs}}{v}} \quad \frac{\text{lb}_m}{\text{sec}} \quad (42a)$$

$$\dot{m}_l = 0.01489 K_l E \sqrt{\frac{\Delta z_{vl}}{v}} \quad \frac{\text{lb}_m}{\text{sec}} \quad (42b)$$

In the above equations Δz_v represents venturi pressure drop in inches of mercury, whereas v is given in cubic feet per pound of mass. When

venturi pressure drop is measured on the transducer a correction must be applied to eliminate the effect of the water leg introduced in equations (42a, 42b). It can be easily shown that

$$\Delta z_v = \frac{\Delta P_v}{0.455} \quad (43)$$

where ΔP_v represents the venturi pressure drop in "psi" as measured by the transducer.

The flow coefficient K for each venturi and the thermal expansion coefficient E are given in Figures 19 and 20 respectively. The venturis were built at Purdue University and calibrated by the Argonne National Laboratory at the time of their manufacture. In conjunction with the computer work the flow coefficients and the thermal expansion factor were approximated by simple equations which appear in the figures. The Reynolds number for each venturi was computed from the equation given below:

$$Re_s = 268300 \frac{\dot{m}}{\mu_b} \quad (44a)$$

$$Re_l = 496100 \frac{\dot{m}}{\mu_b} \quad (44b)$$

where viscosity is expressed in units of $\frac{\text{lb}_m}{\text{hr ft}}$

2. Test Section Pressure Drop

The test section pressure drop was measured with a differential transducer. However, the difference in the density of the fluid in the test section and in the cold leg of the circuit required a correction.

FIGURE 19. FLOW COEFFICIENTS FOR LARGE AND SMALL VENTURIS

FIGURE 20 VENTURI THERMAL EXPANSION FACTOR

When the fluid temperature in the test section was high and small pressure differences were being measured, the correction was quite significant and often larger than the measured quantity.

In Appendix B it is shown that the correction is given by the following equation:

$$\Delta P_f = \Delta P_t + \Delta z_p (\rho - \bar{\rho}_b) \quad (45)$$

where $\bar{\rho}_b$ is the mean fluid density between the pressure taps in the test section. ΔP_f and ΔP_t indicate test section frictional pressure drop and the pressure differential registered by the transducer, respectively.

3. Calculation of the Inside Wall Temperature

Inside wall temperatures were computed from the first four terms of a series solution given by Kreith and Summerfield (34) using the measurements of outside wall temperatures and the current flowing through the section. The equation is as follows:

$$\Delta t = t_s - t_w = \frac{M \Delta x^2}{(1 + \gamma t_s)(1 + \alpha t_s)} \left[1 + \frac{\Delta x}{3r_o} + \frac{\Delta x^2}{4r_o^2} + \frac{M \Delta x^2}{(1 + \gamma t_s)(1 + \alpha t_s)} \frac{(3\alpha + 4\alpha \gamma t_s + \gamma)}{6(1 + \gamma t_s)(1 + \alpha t_s)} \right] \quad (46)$$

where

$$M = \frac{I^2 \rho_{emz}^2}{2Jk_o \rho_{eo} Ac^2}$$

$$\rho_{emz} = \rho_{eo} \left[1 + \frac{\alpha}{r_o - r_i} \int_{r_i}^{r_o} t(r) dr \right] \approx \rho_{eo} (1 + \alpha t_m)$$

and

$$t_m = \frac{t_s + t_w}{2}$$

It is observed that an accurate determination of the inside wall temperature of the tube depends to a great extent on a precise knowledge of the variables appearing in equation (46). This subject is taken up in Appendix C where the influence of each variable upon the computed value of Δt is considered. Since the thermal conductivity of stainless steels is subject to considerable controversy (see Figure 4), an apparatus was designed and measurements were carried out with a piece from a similar tube with the same heat number as the XF test section.

The result of measurements was in fair agreement with an equation used by Dickinson and Welch (12) who gave the National Bureau of Standards for the source. The thermal conductivity, computed from an equation obtained by fitting a linear least squares curve to the experimental data, was on the average 2.5 per cent less than the value given by the N.B.S. originated equation in the 650-950 F range.

The equation given by Dickinson and Welch, viz.

$$k = 8.50 (1 + 5.17 \times 10^{-4} t) \frac{B}{hr \ ft \ F} \quad (47)$$

was used in conjunction with equation (46) for the computations of the inside wall temperature. Since time did not allow measurements with the tube material of the XE test section, the use of two different relations would have been unwise from the point of view of data correlation and comparison.

Furthermore, the value of the measurements lies in the fact that they gave some indication of the extent of uncertainty in the thermal conductivity computed from equation (47).

Comparison of the electrical resistivities of the test sections revealed that they differed by about 3.4 per cent at the 800 F temperature level. It is possible that the thermal conductivities were also different. Computations of the electrical resistivity of the XF test section from voltage and current measurements showed that the results were in excellent agreement with previous, unreported, precision measurements of the author with a Type 347 stainless steel tube. Appendix D briefly describes the method, and Figure D-3 shows a comparison with other measurements reported in the literature. A Kelvin bridge arrangement with a measurement accuracy of 10^{-6} ohm was used.

The XF test section resistivities as computed from the voltage and current readings taken during the experimental runs had rms deviations of less than 0.16 per cent in the 600-850 F range and less than 0.21 per cent in the 400-850 F range as found from a second degree least squares curve fitted to the data of the above mentioned electrical resistivity measurements. In the light of the fact that the least squares fit had a normalized absolute deviation of 0.083 per cent from the measured data the agreement is obviously quite satisfactory.

The XE test section electrical resistivity compared very well with an equation given by Dickinson and Welch (12). The rms deviation was less than 0.20 per cent in the 600-800 F range. Table V summarizes the thermal conductivities and electrical resistivities together with their linearized forms (where applicable) used for each test section and in conjunction with equation (46). Figure 21 is a graph of various electrical resistivities.

TABLE V

Summary of Electrical Resistivities and Thermal
Conductivity for the Test Sections

Thermal Conductivity*

$$k = 8.50(1 + 5.17 \times 10^{-4} t) \quad \text{XE and XF} \quad (47)$$

Electrical Resistivity **

$$\rho_e = 30.18(1 + 3.71 \times 10^{-4} t) 10^{-6} \quad \begin{array}{c} \text{XE} \\ 600 \leq t \end{array} \quad (48)$$

$$\rho_e = 28.46(1 + 6.49 \times 10^{-4} t - 1.52 \times 10^{-7} t^2) 10^{-6} \quad \text{XF} \quad (49)$$

$$\text{linearized } \rho_e = 31.75(1 + 3.42 \times 10^{-4} t) 10^{-6} \quad \begin{array}{c} \text{XF} \\ 650 < t < 1000 \end{array} \quad (50)$$

$$\text{linearized } \rho_e = 29.45(1 + 4.84 \times 10^{-4} t) 10^{-6} \quad \begin{array}{c} \text{XF} \\ 400 < t < 600 \end{array} \quad (51)$$

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* Units are in B/hr ft F.

** Units are in Ohm-inches.

FIGURE 21. ELECTRICAL RESISTIVITIES OF TEST SECTIONS XE AND XF

4. Calculation of Local Wall Heat Flux

The heat generation in a differential length of the test section is given by

$$Q = \frac{I^2 \rho_{emz} \Delta z}{A_c} \times 3.4128 \frac{B}{hr} \quad (52)$$

The heat flux at the inside wall was computed by dividing the heat rate by the inside area of the differential element under consideration. Therefore,

$$\frac{Q}{A_i} = 3.4128 \frac{I^2 \rho_{emz}}{\pi D_i A_c} \quad (53)$$

where ρ_{emz} is the local mean value of the electrical resistivity. In order to account for the deviation of the actual value of the electrical resistivity from that computed from an appropriate equation, the wall heat flux was multiplied by the ratio of measured resistivity to the average calculated resistivity. Thus,

$$q_w'' = 3.4128 \frac{I^2 \rho_{emz}}{\pi D_i A_c} \cdot \frac{\bar{\rho}_e}{\rho_{em}} \frac{B}{hr ft^2} \quad (54)$$

where

$$\rho_{em} = \frac{1}{L} \int_0^L \rho_{emz} dz \quad (55)$$

ρ_{emz} was computed from equations (48) or (49) depending on the test section.

5. The Local Bulk Fluid Temperature

The bulk fluid temperature was determined from tabulated enthalpy-temperature-pressure data taken from smoothed curves. The bulk enthalpy at a point in the test section was determined by adding to the inlet bulk enthalpy the total heat delivered to the water from the entrance to the point in question. Thus,

$$H_z = H_0 + \frac{\pi D_i X}{m} \int_0^z q_w'' dz \quad (56)$$

In the above equation X is the ratio of the heat input to the fluid over that generated.

Both integrals in equations (55) and (56) were evaluated by the trapezoidal rule, viz.

$$\int_z^{z+\Delta z} y dz = \frac{\Delta z}{2} (y_z + y_{z+\Delta z})$$

The bulk fluid pressure was determined by linear interpolation of the pressure drop between the entrance and exit, which was obtained from differential pressure transducer readings. Therefore,

$$P_z = P_0 - \frac{z}{\Delta z_p} \Delta P_f \quad (57)$$

From the computed values of the wall temperature and flux, the bulk fluid enthalpy and the bulk temperature various parameters of interest were calculated.

6. Isothermal Friction Factor

The isothermal friction factor for the test section, defined by the equation

$$f \equiv \frac{\Delta P_f}{\frac{\Delta z_p \rho v^2}{D_i 2g}} \quad (58)$$

was computed for a series of isothermal pressure drop measurements with the XE, XF, and another test section. For calculation purposes equation (58) was expressed in terms of the flow rate instead of the velocity, viz.

$$f \equiv \frac{18 \pi^2 g D_i^5}{\Delta z_p} \frac{\Delta P_f}{\dot{v}^2} \quad (59)$$

The above relation also shows that the tube inside diameter must be determined with good accuracy.

7. Fluid Properties Used in This Work

For temperatures above 600 F the dynamic viscosity and thermal conductivity were evaluated from equations adapted from Fritsch and Grosh (28), viz.

$$\mu = (37.94 \times 10^{-10} x (\frac{1}{v})^{1.57} + 4.788 \times 10^{-10} t + 14.79 \times 10^{-8}) 115920 \quad (60)$$

$$k = 26.6 \times 10^{-4} x (\frac{1}{v})^{1.24} + 4.0 \times 10^{-5} (t - 550) + 0.0246 \quad (61)$$

The units are $\frac{\text{lb}_m}{\text{hr ft}}$ and $\frac{\text{B}}{\text{hr ft F}}$ respectively. The values obtained from the above equations for 3300 psia pressure and temperatures from

600 to 750 F were compared to values taken from large plots and tables made available by the Babcock and Wilcox Co. Research Center*. In general good agreement existed except at near the peak temperature where the thermal conductivity showed a large discrepancy of about 25 per cent. At other temperatures the agreement was well within four per cent. For the 400-560 F temperature region the above mentioned plots and tables were used.

Since no experimental thermal conductivity values are available for water in the near critical region, both the equations and the plots give interpolated values. The former were preferred since they were easier for machine computations.

All thermodynamic properties for the critical region were obtained from the recommended values of Nowak and Grosh. The specific heat, required in the Dittus-Boelter equation, was taken from Keenan and Keyes steam tables (36) at temperatures below those covered by (26). Otherwise, the use was limited to the modified forms presented earlier.

The specific volume was obtained from references (26), where available, and (36). Its use was in conjunction with the flow rate computations and various correlations where the specific volume, dynamic viscosity, and thermal conductivity were required.

Data from various tables were put together and cross plotted several times to obtain a set of smoothed, consistent enthalpy-temperature-pressure plots and tables for a region extending from about 600 to 830 F, and 2900 to 3400 psia. Special emphasis was placed in the

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* Personal communications.

near critical region where the data was spaced every 20 psi from 3200-3300 psia. Tables contained temperature as a function of pressure and enthalpy, which was spaced every 5.0 B/lb_m.

Data in the critical region were mostly obtained from Nowak and Grosh (27). Other data taken from references (36, 37, 38, 39, 40) were used in conjunction with, and beyond the range of (27). Isenthalpic and isobaric plots were prepared. The smoothness of the isobaric enthalpy-temperature values in the critical region was checked by plotting the product of enthalpy and temperature versus enthalpy. Indeed, the results showed very smooth, sectionally linear curves.

Appendix E presents the enthalpy-pressure-temperature tables as described above and used in this work. The enthalpy range extends from 600 to 1300 B/lb_m and is spaced every 5.0 B/lb_m. Figure 22 illustrates the isobaric enthalpy-temperature curves for water in the critical region.

The values of the peak temperature versus pressure were adapted from reference (26), and are expressed by the equation:

$$t_p = 705.50 + 0.044643 (P-3208) \quad (62)$$

which is valid up to around 3750 psia. Nowak has given the critical pressure and temperature as 3208.0 psia and 705.5 F. A plot of t_p versus pressure is given in Figure 23.

An effort was made to obtain the value of the enthalpy corresponding to the peak temperature (or the peak enthalpy), but despite the use of several methods no consistent results could be obtained. This probably presents the toughest demand from any temperature-

FIGURE 22. ISOBARIC ENTHALPY - TEMPERATURE CURVES FOR WATER IN THE CRITICAL REGION

FIGURE 23 PRESSURE AND TEMPERATURE
CORRESPONDING TO $C_{p_{\max}}$ ALONG
ISOBARS

enthalpy-pressure data for the critical region. Appendix F discusses this point a little further.

8. Computer Reduction of Experimental Data

The Purdue University IBM 7094 digital computer was programmed to reduce sets of data and to compute various parameters of interest. The program was written in FORTRAN II compiler language and is presented in Appendix G.

The program consisted of two sections, run separately. The first part was designed to compute heat balances on the experimental runs. The heat balance program performed related computations and on the output described the various parameters of importance to heat balance computations such as the flow rate, heat generated, heat absorbed, and the heat balance error.

If the heat balance results were satisfactory, the entire wall thermocouple potentials and other data needed for reduction and analysis were fed into the second section. This program computed the inside wall temperatures, iterating for the correct mean temperature for the Kreith-Summerfield equation, the local wall heat fluxes, local fluid and wall enthalpies, local fluid temperatures, and many other parameters of interest for correlations. The ratio of the measured test section electrical resistivity over the average computed value was also calculated. Test section pressure drop, though not significant in some runs, was entered into the computations as a matter of routine.

Every effort was made to reduce man-handling of data reduction to eliminate mistakes and possibly some prejudice. All punched cards were

double checked for accuracy. Thermocouple calibrations were also done on the computer.

Sample data reductions from XE and XF series are given in Appendix H together with an explanation of various computer input and output codes and symbols. Several sets of raw data are presented in Appendix I. However, space does not permit including all data.

EXPERIMENTAL RESULTS, CORRELATIONS, AND DISCUSSION

The purpose of this section is to present and discuss the experimental findings and the correlations used in this work. To establish the validity and the integrity of the results for the near critical experiments, the isothermal friction factor and low temperature heat transfer results are presented first.

Isothermal measurements of the test section pressure drop, from which the friction factor was computed, were carried out at 50 psia, 61-89 F; 2000 psia, 297-425 F; and 3000 psia, 367-437 F. The results of these tests are presented in Figure 24, together with Prandtl's universal law of friction for smooth pipes. It is observed that, for the methods employed, there was very good agreement at Reynolds numbers less than 10^5 . The small discrepancy could be easily due to a small uncertainty in the test section diameter. In the section on data reduction it was pointed out that the fifth power of the diameter entered the computation. Thus, a one per cent error in the diameter would reflect as five per cent in the friction factor. At Reynolds numbers above 10^5 the XE test section still showed good agreement, but had a tendency to flatten out at around 3×10^5 . The XF test section demonstrated the characteristics of a rough tube, its friction factor having leveled off at the value of 0.018. This should not be disturbing. Referring to the Moody chart for commercial pipes

FIGURE 24. ISOTHERMAL FRICTION FACTOR FOR TEST SECTIONS

it is seen that the above corresponds to a relative roughness of 0.0007, which upon multiplication by the diameter gives a roughness of 0.0003 inch. It is quite doubtful that the tube wall was any smoother. The measurements of the tube wall indicated that it had up to 0.0008 inch variations from the mean and that it was not axis-symmetric. In the light of the above information the flattening in the friction factor might be expected. Thus, the results that are presented in the following apply to semi-smooth tubes.

The results of the heat transfer measurements at 3000 psia and 400-560 F temperature range were compared with the Dittus-Boelter equation and showed good agreement. Figure 25 shows a typical case of the inside wall temperature, bulk fluid temperature, and the heat transfer coefficient distribution along the heated length of the tube. A large scale was employed to show the behavior of the variables. It is observed that the inside wall temperature increased smoothly except for a dip at a distance of 22 inches (or 59.5 length to diameter ratio) from the on-set of heating. The maximum variation of the local heat transfer coefficients from the mean value along the tube did not exceed 10.5 per cent. This magnitude is nearly the same as the estimated experimental uncertainty. Such data, without any smoothing, were used in the evaluation of other parameters of interest.

The low temperature data were also compared with the equation suggested by Swenson et al., viz.

FIGURE 25. LOW TEMPERATURE HEAT TRANSFER RESULTS

$$Nu_w = 0.00453 Re_w^{0.923} \left(\frac{\mu_w (H_w - H_b)}{(t_w - t_b) k_w} \right)^{0.613} \left(\frac{v_b}{v_w} \right)^{0.231} \quad (25)$$

Figure 26 shows the comparison. Most of the data fell above the correlation line, but within 15 per cent of it. This was not as good as the Dittus-Boelter correlation where the rms deviation was around seven per cent. However, the above equation was not basically intended for pressures below 3300 psia.

The agreement of the low temperature data with the above correlations indicated that the operation of the equipment was satisfactory and that higher temperature data could be obtained and reduced with some confidence. Also, it appeared that the fluid vibrations present in the system did not have a significant influence upon the heat transfer coefficient. This is certainly correct if other data used in the correlations were obtained under conditions where such perturbations were not present. Otherwise, the above conclusion is not valid.

The behavior of the wall temperature and the heat transfer coefficient along the length of the XF test section with inlet temperature of around 665 F is shown in Figure 27. It is observed that there is little difference between the shape of the heat transfer coefficient in the central portion of the tube for this run and the lower temperature run depicted in Figure 25. Except for the different pressures, the mass flux was nearly constant and the heat flux was increased by about fifteen per cent to provide the higher temperature at the inlet. However, the shape of the "h" profile close to the inlet and the outlet

FIGURE 26. COMPARISON OF LOW TEMPERATURE DATA WITH THE EQUATION OF SWENSON ET AL.

FIGURE 27. HEAT TRANSFER RESULTS FOR RUN XF 10

was somewhat different. It appeared that such a behavior was characteristic of low surface to bulk temperature differences. Figure 28 shows similar results for run XE 4 where large heat and mass fluxes were imposed. The fluid inlet temperature in reduced (t_b/t_p) form was 0.8532 as compared to 0.9407 for run XF 10. It is observed that the "h" profile was distinctly different. The wall to bulk temperature differences for run XE 4 were more than twice as much as for runs XF 31 and XF 10. The larger heat transfer coefficients in Figure 27 were undoubtedly due to the small temperature differences. The increase in the heat transfer coefficient close to the outlet seemed to be due to the wall temperature reaching closer to the peak temperature.

When the wall temperature is near the peak temperature, the fluid in contact with it becomes a better heat transfer medium and the heat transfer coefficient is increased to the extent that the wall temperature tends to flatten out. This point is well demonstrated in Figure 28 where, despite the very high heat flux, the wall temperature gradient along the tube becomes very small. This effect could not be attributed to the fluid since the bulk fluid temperature was still quite remote from the peak temperature. The reader must be reminded that this study concerned itself also with wall temperatures near the peak temperature.

A plot of the Stanton parameter St_e , previously defined as

$$St_e = \frac{q_w''}{(H_w - H_b)G}$$

FIGURE 28. HEAT TRANSFER RESULTS FOR RUN XE 4

versus tube length-over-diameter ratio is presented in Figure 29 for the runs discussed above. The very distinct behavior of XE 4 as compared with XF 10 and XF 31 is observed. The effect of wall to bulk fluid temperature is well illustrated. As $(t_w - t_b)$ is increased the Stanton number levels off. On the average the film temperature difference for the runs XF 10, XF 31, and XE 4 were 14, 20, and 41 F respectively. It must also be noted that the effect presented in Figure 29 was by no means associated with the proximity of fluid and/or surface temperature to the peak temperature since the fluid and wall temperatures for runs XE 4 and XF 10 were in the same range and XF 31 was a low temperature run. An estimate of the uncertainty in the film temperature difference, and hence in the heat transfer coefficient and the various parameters employed in this work, is presented in Appendix J.

The next series of plots show the behavior of the bulk and wall temperatures and the heat transfer coefficient under near critical operations. The effect of various parameters upon the heat transfer coefficient is illustrated. In contrast to Figure 28, Figure 30 shows a hump in the heat transfer coefficient profile around the center section of the tube. The heat flux may be considered as constant since the change in it was less than four per cent. Because in general the effect of the lower mass flux should be a decrease in the heat transfer coefficient, it may be concluded that the higher heat transfer coefficient and the hump were due to the proximity of the fluid temperature to the peak temperature. It may be hypothesized that the hump occurs where the optimum conditions for the transfer of heat from the

XE 7
 PRESSURE 3300 PSIA
 HEAT FLUX 504000 B/HR FT²
 MASS FLUX 2.66×10^6 LB_m/HR FT²
 t_{b0}/t_p 0.9639

FIGURE 30. HEAT TRANSFER RESULTS FOR RUN XE 7

wall to the fluid stream are reached. Figure 31 shows a widening of the hump as the inlet fluid temperature gets closer to the peak temperature. The hump also moved left where conditions were optimum. The widening could be also attributed to the decrease in the heat flux which allowed a slower increase in the fluid and wall temperatures. A reduction in the magnitude of the heat transfer coefficient was also observed. Since the wall temperature in Figures 30 and 31 remained virtually constant, the decrease in the heat flux may be cited as the reason. However, the flow rate may be thought to have contributed to the decrease also.

Figures 32, 33, and 34 illustrate further the effect of the ratio t_b/t_p upon the heat transfer coefficient profile along the tube. The heat flux and the mass flux were maintained nearly constant. As t_b/t_p approached unity the hump in the heat transfer coefficient curve flattened out and moved to the left. Figure 34 shows a decreasing h curve along the entire tube length. It should also be observed that the wall temperature, being now further away from the peak temperature, has a smooth increasing gradient.

Stanton number profiles for the above runs together with several other typical runs are shown in Figure 35. The conditions for each run are summarized in the accompanying table. The illustration provides for easy determination of the effect of various independent variables upon the Stanton number. A comparison of XF 42 and XF 39 revealed that a larger t_b/t_p ratio at the inlet tended to increase the Stanton number; or comparison of XF 39, XF 36, and XE 28 showed that for nearly constant t_b/t_p the Stanton number decreased non-linearly with decreasing $\frac{q_w''}{G}$ ratio.

XE 31
 PRESSURE 3300 PSIA
 HEAT FLUX 382000 B/HR FT²
 MASS FLUX 2.26×10^6 LB_m/HR FT²
 t_{b0}/t_p 0.9850

FIGURE 31. HEAT TRANSFER RESULTS FOR RUN XE 31

FIGURE 32. HEAT TRANSFER RESULTS FOR RUN XF 42

FIGURE 33. HEAT TRANSFER RESULTS FOR XF 39

FIGURE 34. HEAT TRANSFER RESULTS FOR XF 40

FIGURE 35. STANTON NUMBER PROFILES FOR RUNS XE 7, XE 28, XE 31, XF 36, XF 39, XF 40, XF 42

In concluding this part of the presentation of the results, it should be stated that the data of this investigation did not substantiate the findings of Swenson et al. (14) in regard to the general behavior of the heat transfer coefficient. Their data showed a decrease in the heat transfer coefficient along the tube length followed by continuous increase starting at 57 diameters down-stream from the on-set of heating for inlet temperatures below the peak temperature. No such behavior was observed in the present investigation.

The data were compared with the correlations of Dittus-Boelter and Swenson et al. Figure 36 shows the comparison of both the low and high temperature data with the Dittus-Boelter equation. It was found that good agreement was obtained for the low temperature data together with other test data up to a temperature of 650 F or $Pr_b = 1.20$. Beyond this limit a rather rapid disagreement was in evidence.

Figure 37 gives a comparison of the high temperature data with the correlation of Swenson et al. Good agreement existed even beyond the tested range of the correlation. However, a rapid deterioration at the near critical temperatures was observed. Figure 38 shows a plot of the Nusselt number as computed from equation (25) versus the Nusselt number obtained from the measurements. No fixed point for the beginning of the disagreement may be given in terms of the Nusselt number. Experience showed that discrepancy started at the near critical temperatures, more specifically above 690 F. Figure 38 shows deviation starting at around $Nu_w = 1500$ for bulk fluid temperatures close to the peak temperature, but the wall temperature well above the peak temperature. Larger deviations occurred at higher

FIGURE 36. COMPARISON OF DATA WITH THE DITTUS-BOELTER CORRELATION

FIGURE 37. COMPARISON OF HIGH TEMPERATURE DATA WITH THE CORRELATION OF SWENSON ET AL.

Nusselt numbers resulting from conditions where both the bulk fluid and the wall temperatures were in the proximity of the peak temperature. The data used in conjunction with the correlation of Swenson et al. were taken from the downstream half of the tube as the correlation called for.

Figures 39 and 40 show an example of the behavior of the wall temperature and the heat transfer coefficient for small wall to bulk fluid temperature differences in the critical region. It is observed that the wall temperature gradient along the tube was very small. The heat transfer coefficient showed an increasing trend as the fluid temperature approached the peak temperature. It is evident that the heat transfer process becomes more efficient when both the wall and fluid temperatures are near the peak temperature. The experimental heat transfer coefficients obtained under such conditions were generally more than one and one half times as large as the values computed from the available correlations.

The Stanton number profiles along the tube length for runs XF 18 and XE 25 are shown in Figure 41. The profile for run XF 18 may be compared with the profile for run XE 7 shown in Figure 35. Except for the heat flux all other pertinent variables were nearly constant. Once again the influence of small wall to fluid temperature differences upon the heat transfer coefficient and the Stanton number is evident.

Several correlations of the data of this investigation are presented in the following. In general the data for which the estimated uncertainty in the heat transfer coefficient exceeded 25 per cent

FIGURE 39. HEAT TRANSFER RESULTS FOR RUN XF 18

FIGURE 40. HEAT TRANSFER RESULTS FOR RUN XE 25

FIGURE 41. STANTON NUMBER PROFILES FOR RUNS XE 25, XF 18

were excluded. This included the runs having small film temperature drops.

Figure 42 illustrates one possible method of correlation proposed in this work. The average Stanton number, previously defined as

$$\overline{St} \equiv \frac{\overline{h}}{c_p G}$$

was plotted against $\frac{\overline{t_b}}{\overline{t_p}}$ which signifies the ratio of average fluid temperature between stations $z + \Delta z$ and z , and the peak temperature corresponding to the prevailing average pressure. At $\frac{\overline{t_b}}{\overline{t_p}}$ less than 0.93 the Stanton number may be considered as constant for a limited range of variables. The data scatter was within ± 10 per cent of the mean for $\frac{\overline{t_b}}{\overline{t_p}}$ not exceeding 0.98. Beyond this limit the average Stanton number decreased rapidly and larger scatter was observed. This correlation provides a simple method for presenting the near critical data. It may be improved and extended in range by adjusting the ordinate for the Reynolds number effect. This is not done here since a better method of correlation is presented later.

Another useful and simple correlation is presented in Figure 43, where the average Stanton number was plotted against $\frac{t_p - t_b}{\overline{t_w} - \overline{t_b}}$. Again, to obtain a better fit and a wider range proper adjustment for the Reynolds number effect must be provided. However, in their present forms the above correlations included most of the data within ten to fifteen per cent. The "specific heat-like" behavior in the proximity of the peak temperature, arising from the definition of $\overline{St}$, seemed to be the main disadvantage of the correlations.

FIGURE 42. AVERAGE STANTON NUMBER VERSUS REDUCED BULK FLUID TEMPERATURE

FIGURE 43. AVERAGE STANTON NUMBER VERSUS $\frac{t_p - \bar{t}_b}{\bar{t}_w - \bar{t}_b}$

Figure 44 presents two correlations for heat transfer to water in the critical region. It is seen that the upper plot, based on bulk fluid properties, correlates nearly all the data within ± 16 per cent. Some scatter at lower enthalpy values was primarily due to small film temperature drops. The lower plot was based on properties evaluated at the wall temperature, and correlated most of the data within ± 15 per cent. The correlation equations are:

$$St_e Re_b^{0.2} = 0.0068 \exp \left[0.242(H_b - 725) \times 10^{-2} \right] \quad (63)$$

$$St_e Re_w^{0.2} = 0.0080 \exp \left[0.169(H_w - 1010) \times 10^{-2} \right] \quad (64)$$

The data shown in Figure 44 are for 3300 psia pressure; however, the above equations also correlated the data of other pressures in the range 3220-3400 psia. The data from the entire length of the tube except for about eight diameters at the inlet and outlet were included. This was done to eliminate the possible effect of heat loss from the ends. At present the range of the equations is limited to the extent of data shown in Figure 44. The lower limit of $H_b = 725$ B/lb_m and $H_w = 1010$ B/lb_m must be observed.

The range and validity of the correlations may be increased somewhat by using H_p to non-dimensionalize the enthalpy term in the exponent. An estimate of the peak enthalpy may be obtained from Figure F-1. Under this transformation equation (63) changes to:

$$St_e Re_b^{0.2} = 0.0068 \exp \left[2.19 \left(\frac{H_b}{H_p} - 0.801 \right) \right] \quad (65)$$

The uncertainty in H_p was estimated at less than two per cent.

FIGURE 44. CORRELATION OF NEAR CRITICAL HEAT TRANSFER DATA

Some experimental heat transfer results for carbon dioxide taken from the works of Koppel, Wood, and Smith were compared with values computed from equation (65). Despite inadequate knowledge of H_p , good agreement (well within the limits given in Figure 44) was obtained.

An attempt was made to extend the range of the Dittus-Boelter equation which satisfactorily correlated the low temperature data where the value of the Prandtl number did not exceed 1.2. The parameter

$$\xi_b = \frac{St_e Re_b^{0.2} Pr^{0.6}}{0.0243}, \text{ where } c_{ph} \text{ was used in the bulk fluid Prandtl}$$

number, was plotted against Pr , $\frac{v_b}{v_w}$, and other pertinent variables.

No direct relationship between the above parameter and $\frac{v_b}{v_w}$ could be found, but the parameter was found to be related to the Prandtl number for $Pr > 6$, where a large increase in the value of ξ_b with increasing Prandtl numbers could be seen. Figure 45 shows a plot of ξ_b versus Pr . A slight Reynolds number dependency is evident since the data with low mass fluxes fell below the minus ten per cent line.

Concerning the heat transfer coefficient in the supercritical region, it should be stated that the results of this investigation showed a decrease in the heat transfer coefficient with increasing heat fluxes. This was in agreement with the findings of other investigators.

FIGURE 45. DEPENDENCE OF ξ_b UPON THE PRANDTL NUMBER

CONCLUSIONS AND RECOMMENDATIONS

Consideration of the experimental observations and the results led to the following conclusions:

1. Large heat transfer coefficients can be expected for heat transfer to critical water in turbulent upward flow through tubes. Very large heat transfer coefficients are obtained at small wall-to-bulk fluid temperature differences; however, uncertainty in the computation of the inside wall temperature does not allow a better understanding of heat transfer under such conditions. Such data could not be employed in the correlations and/or presented with any degree of certitude.
2. The Dittus-Boelter equation was proved to be inadequate for use with water at temperatures above 650 F or $Pr_b = 1.20$. The upper limit beyond which the equation may again be found useful can not be determined from the results of this investigation.
3. The recent correlation of Swenson et al. proved adequate up to around 690 F in general. It showed progressive deviation for temperatures closer to the peak temperature.
4. For applications up to 650 F the Dittus-Boelter equation would be preferred.
5. The failure of equations of the Dittus-Boelter type in the near critical region is associated with their inadequacy to

respond to large variations in the Prandtl number. In other words, in the critical region the Nusselt number is strongly dependent on the properties which undergo large changes rather than the Reynolds number alone.

6. The use of Stanton number, St_e , appears to be quite promising. However, more extensive data must be obtained in order to arrive at a firm conclusion. Reduced enthalpy seems to be a good parameter to be used in the correlations involving the near critical region.

7. A correlation of the type

$$St_e Re^a = c e^{b(H_b/H_p)}$$

is recommended for use in conjunction with critical water.

It is hoped that with the use of H_b/H_p the range of the above correlation may be widened.

In view of the present work the following recommendations are presented.

1. For any further investigations of this nature the High Flux Heat Transfer Loop of the School of Mechanical Engineering must be modified to provide for: 1) a much better and more flexible preheater, 2) a rapid sequential thermocouple potential measuring device, and 3) better fluid immersed thermometry.
2. An extensive investigation must be conducted to pin-point the cause and effect of the fluid vibrations present in the system.

3. With the above two items taken care of, investigations must be carried out in the region above the peak temperature.
4. The possibility of initiating extensive measurements of the temperature and velocity profiles in the near critical fluids must be considered seriously. This would provide a firm foundation for future analytical studies and a good understanding of this mode of heat transfer.
5. The possibility of additional investigations of heat transfer to near critical fluids under small wall-to-bulk fluid temperature differences must be considered.
6. Investigations should be undertaken to determine the correctness of the hypothesis that the free convection pressure instabilities have their origin in the small flow fluctuations observed in this work.

BIBLIOGRAPHY

BIBLIOGRAPHY

1. Deissler, R. G., "Heat Transfer and Fluid Friction for Fully Developed Turbulent Flow for Air and Supercritical Water with Variable Fluid Properties", ASME Trans., vol. 76, Jan. 1954, pp. 73-85.
2. Eckert, E. R. G. and Drake, R. M., Introduction to Heat and Mass Transfer, McGraw Hill Book Co., Inc., New York, Second Edition, 1959.
3. Goldmann, K., "Heat Transfer to Supercritical Water and Other Fluids with Temperature Dependent Properties", Chem. E. Progress Symposium, Series 50, 1954, pp. 105-113.
4. Goldmann, K., "Heat Transfer to Supercritical Water at 5000 psia Flowing at High Mass Flow Rates through Round Tubes", International Developments in Heat Transfer, Part III, ASME, 1961, pp. 561-568.
5. Hsu, Y. Y., and Smith, J. M., "The Effect of Density Variation on Heat Transfer in the Critical Region", ASME paper 60-HT-8, presented at the ASME-AIChE Heat Transfer Conference, August 15, 1960.
6. Bringer, R. P., and Smith, J. M., "Heat Transfer in the Critical Region", AIChE Journal, No. 3, 1957, pp. 49-55.
7. Ojalvo, M. S., "Combined Forced and Free Turbulent Convection in a Vertical Tube", Ph.D. Thesis, Purdue University, Jan., 1962.
8. Lykoudis, P. S., "Analytical Study of Heat Transfer in Liquid Metals", Ph.D. Thesis, Purdue University, Jan., 1956.
9. McAdams, W. H., Kennel, W. E., and Adams, J. N., "Heat Transfer to Superheated Steam at High Pressures", ASME Trans., No. 72, 1950, pp. 421-428.
10. Kreith, F., and Summerfield, M., "Heat Transfer to Water at High Flux Densities with or without Surface Boiling", ASME Trans., Vol. 71, 1949, pp. 805-15.
11. Miropolskii, L., and Shitsman, M. E., "Heat Transfer to Water and Steam at Variable Specific Heat in Near Critical Region", Soviet Physics, Technical Physics, Translated into English by the American Institute of Physics, Vol. 2, No. 10, Oct. 1957, pp. 2196-2208.

12. Dickinson, W. L., and Welch, C. P., "Heat Transfer in Supercritical Water", ASME Trans., Vol. 80, 1958, pp. 746-752.
13. Touba, R. F., "Heat Transfer to Critical Water Flowing through a Heated Vertical Tube Under the Effects of Combined Forced and Free Convection", M.S. Thesis, Purdue University, June, 1962.
14. Swenson, H. S., Kakarala, C. R., Carver, J. R., "Heat Transfer to Supercritical Water in Smooth Bore Tubes", ASME paper 64-WA/HT-25, presented at the Winter Annual Meeting, New York, N. Y., Nov. 29-Dec. 4, 1964.
15. Hess, H. L., and Kunz, H. R., "A Study of Forced Convection Heat Transfer to Supercritical Hydrogen", ASME paper 63-WA-205, presented at the Winter Annual Meeting, Nov., 1963.
16. Powell, W. B., "Heat Transfer to Fluids in the Region of Critical Temperature", Jet Propulsion, Vol. 27, No. 7, 1957, pp. 776-783.
17. Petukhov, B. S., Krasnoschekov, E. A., and Protopopov, V. S., "An Investigation of Heat Transfer to Fluids Flowing in Pipes Under Supercritical Conditions", International Developments in Heat Transfer, Part III, ASME, New York, 1961, pp. 569-578.
18. Koppel, L. B., and Smith, J. M., "Turbulent Heat Transfer in the Critical Region", International Developments in Heat Transfer, part III, ASME, 1961, pp. 585-590.
19. Wood, R. D. and Smith, J. M., "Heat Transfer in the Critical Region-Temperature and Velocity Profiles in Turbulent Flow", AIChE Journal, March 64, pp. 180-186.
20. Hallman, T. M., "Combined Forced and Free Convection in a Vertical Tube", Ph.D. Thesis, Purdue University, May, 1958.
21. Sparrow, E. M., Eichhorn, R., and Gregg, J. L., "Combined Forced and Free Convection in Boundary Layer Flow", The Physics of Fluids, No. 3, May-June, 1959, pp. 319-328.
22. Hawkins, G. A., Thermodynamics, Second Edition, John Wiley and Sons, Inc., New York, 1957.
23. Fritsch, C. A., "Free Convection Heat Transfer to a Supercritical Fluid - An Analytical and Experimental Study", Ph.D. Thesis, Purdue University, January 1962.
24. Nowak, E. S., Grosh, R. J., Liley, P. E., "Smoothed Pressure-Volume-Temperature Data for Water in the Critical Region Derived from Experimental Measurements", School of Mechanical Engineering, Purdue University

25. Nowak, E. S., Grosh, R. J., "The Coefficient of Volume Expansion for Water and Water Vapor in the Critical Region", Technical Report No. 6 to Atomic Energy Commission, School of Mechanical Engineering, Purdue University, July 1961.
26. Nowak, E. S., Grosh, R. J., "An Analysis of Specific Heat Data for Water and Water Vapor in the Critical Region", Technical Report No. 8 to Atomic Energy Commission, School of Mechanical Engineering, Purdue University, Aug. 1961.
27. Nowak, E. S., Grosh, R. J., "An Investigation of Enthalpy Data for water and Water Vapor in the Critical Region", Technical Report No. 10 to Atomic Energy Commission, School of Mechanical Engineering, Purdue University, Sept. 1961.
28. Fritsch, C. A., Grosh, R. J., "A Description of Thermal Conductivity and Dynamic Viscosity of Water in the Critical Region", Argonne National Laboratory, ANL-6238, Oct. 1960.
29. Focken, C. M., Dimensional Methods and Their Applications, Edward Arnold and Co., London, 1953.
30. Dix, R. C., "High Flux Heat Transfer Studies", M. S. Thesis, Purdue University, 1958.
31. Van Putte, D. A., Grosh, R. J., "Heat Transfer to Water in the Near Critical Region", Technical Report No. 4 to Argonne National Laboratory, School of Mechanical Engineering, Purdue University, August 1960.
32. Tokar, John V., "The Construction of an Experimental Heat Transfer Loop and a Review of Nucleate Boiling", M. S. Thesis, Purdue University, August 1961.
33. Wallace, R. B., "Single and Two Phase Heat Transfer to Near Critical Water Under Forced Convection in an Electrically Heated Vertical Tube", Ph.D. Thesis, Purdue University, June 1964.
34. Kreith, F., and Summerfield, M., "Investigation of Heat Transfer at High Heat Flux Densities, Experimental Study with Water-Friction Drop and Forced Convection with and without Surface Boiling in Tubes", Progress Report No. 4-68, Jet Propulsion Laboratory, California Institute of Technology, Pasadena, California.
35. National Bureau of Standards, Tables of Circular and Hyperbolic Sines and Cosines for Radian Arguments, Applied Mathematics Series 36, Issued Nov. 30, 1953.
36. Keenan, J. H., and Keyes, F. G., Thermodynamic Properties of Steam, First Edition, John Wiley and Sons, Inc., New York, N. Y., 1956.

37. Koch, Werner, Schmidt, Ernst, VDI-Wasserdampfataeln Mit einem Molier (i,s)-Diagramm bis 800°C und einem T,S-Diagramm, 6th Revised and Amplified Edition, Springer Verlag, Berlin, 1963.
38. Babcock and Wilcox Co., Atomic Energy Division, "Enthalpy, Entropy, and Specific Volume of Water and Superheated Steam as a Function of Pressure and Temperature", N. Y., 1963.
39. Vukalovich, M., Thermodynamische Eigenschaften des Wassers und des Wasserdampfes, 6th Edition, VEB Berlag Technik, Berlin, 1958.
40. Timrot, D. L., "Tables of Thermodynamic Properties of Water and Water Vapor", First Edition, Moscow, 1952, Leningrad, Translated into English by Land Turbine Engineering Division, Westinghouse Electric Corp., South Philadelphia Works, 1954.
41. Quinn, B. E., "Class Lecture Notes", ME 567-667, Dynamic Problems in Machine Design, School of Mechanical Engineering, Purdue University, Lafayette, Ind., Fall 1961 and Spring 1962.

APPENDICES

APPENDIX A

LIST OF SYMBOLS

Non-Dimensional Parameters

Gr	=	Grashof number	$\frac{\rho^2 \beta g \Delta t D^3}{2}$
Nu	=	Nusselt number	$\frac{hD}{k}$
Pr	=	Prandtl number	$\frac{\mu c_p}{k}$
Ra	=	Rayleigh number	$\frac{\rho^2 \beta g c_p D^4}{16 \mu k} \frac{\partial t}{\partial z}$
Re	=	Reynolds number	$\frac{GD}{\mu}$
S	=	Strouhal number	$\frac{f D}{\text{velocity}}$
St	=	Stanton number	$\frac{h}{c_p G}$ (basic form)

Latin Alphabet (capitals)

A	=	area
Ac	=	tube wall cross sectional area
Al	=	area along the axis
B	=	constant
D	=	tube diameter

D_e	=	hydraulic diameter
E	=	thermal expansion factor for venturis
F	=	Force
G	=	mass flux
H	=	enthalpy
I	=	current
J	=	Joulean constant
K	=	Venturi flow coefficient
K_1, K_2	=	spring constants
L	=	Length
M	=	mass
P	=	pressure
ΔP	=	differential pressure
Q	=	heat rate
R	=	gas constant
T	=	absolute temperature, or time in Appendix K
T^+	=	$(1 - \frac{T}{T_w})/\Omega$
U	=	overall heat transfer coefficient
$\bar{U}$	=	mean bulk velocity
$\bar{U}^+$	=	$\bar{U}/u^*$
V	=	volume
X	=	heat gain factor
Z_0	=	body force in the z-direction
Re^*	=	friction Reynolds number

Latin Alphabet (lower case)

a,b,c,d	=	constants
c_p	=	specific heat
c_{p_h}	=	$(H_w - H_b)/(t_w - t_b)$
f	=	frequency, cycles per second
f	=	friction factor
g	=	gravitational acceleration
h	=	heat transfer coefficient
he	=	heat transfer coefficient based on enthalpy
k	=	Thermal conductivity
l	=	length element
log	=	logarithm to base 10
ln	=	logarithm to base e (2.71828)
m,n	=	constants
$\dot{m}$	=	flow rate
p	=	pressure in thermodynamic relations
q_w''	=	wall heat flux
r	=	radius
r^+	=	u^*r/ν_w
t	=	temperature
t_s	=	surface temperature
u	=	velocity
u^+	=	u/u^*
u^*	=	$\sqrt{\tau_w / \rho_w}$
v	=	specific volume
x	=	coordinate

y	=	perpendicular distance from wall
y^+	=	u^*y/ν_w
z	=	axial direction of tube
Δz_p	=	distance between pressure taps

Greek Alphabet

α	=	temperature coefficient of electrical resistivity
β	=	volume expansion coefficient
γ	=	temperature coefficient of thermal conductivity
Δ	=	differential
ϵ	=	coefficient of eddy diffusivity for momentum
ϵ_h	=	coefficient of eddy diffusivity for heat transfer
η	=	$2r/D_i$
θ	=	angular direction, time in Appendix K
κ	=	Von Karman constant
μ	=	dynamic viscosity
ν	=	kinematic viscosity μ/ρ
π	=	3.14159
ρ	=	density
ρ_e	=	electrical resistivity
Σ	=	indicates summation
ζ	=	$\frac{t_{ref} - t_b}{t_w - t_b}$
τ	=	time, lag value in Appendix K
τ_w	=	wall shear stress

$$\Omega = q_w'' u^* / c_{p_w} g \tau_w T_w$$

$$\omega = \text{frequency, radians per second}$$

Subscripts

b	=	bulk
c	=	critical, used with the properties
f	=	friction, or film when appropriate
g	=	gas
i	=	inside
l	=	large
m	=	mean
o	=	outside, or initial
p	=	peak, in conjunction with the properties
( ) _p	=	constant pressure
r	=	r--direction
ref	=	reference
s	=	small
t	=	transducer
v	=	venturi
w	=	wall
z	=	z-direction
θ	=	θ -direction
∞	=	ambient conditions

Superscripts

-	=	averaged quantity
.	=	fluctuating quantity

APPENDIX B

DENSITY CORRECTION IN TEST SECTION PRESSURE DROP
MEASUREMENTS

Consider the transducer circuit given in Figure B-1 in conjunction with the simple derivation presented below. The work is self explanatory.

$$P_A = P_1 + \rho_{\infty} (z_1 - z_0) \quad \text{B-1}$$

$$P_B = P_2 + \rho_{\infty} (z_2 - z_0) \quad \text{B-2}$$

$$(P_A - P_B) = (P_1 - P_2) - \rho_{\infty} (z_2 - z_1) \quad \text{B-3}$$

But, $P_2 = P_1 - \Delta P_f - \bar{\rho}_b (z_2 - z_1) \quad \text{B-4}$

Substituting for P_2 and transposing,

$$\Delta P_f = (P_A - P_B) + (z_2 - z_1) (\rho_{\infty} - \bar{\rho}_b) \quad \text{B-5}$$

or $\Delta p_f = \Delta P_t + \Delta z_p (\rho_{\infty} - \bar{\rho}_b) \quad \text{B-6}$

$$\bar{\phi}_b = \frac{1}{(z_2 - z_1)} \int_{z_1}^{z_2} \phi dz$$

FIGURE B-1 SIMPLE SCHEMATIC OF TRANSDUCER CIRCUIT

APPENDIX C

POSSIBLE SOURCES OF UNCERTAINTIES AND THEIR EFFECT
ON THE CALCULATION OF INSIDE WALL TEMPERATURE

Inside wall temperatures were computed from the Kreith-Summerfield equation (46) using outside wall temperatures and the current flow through the section. Basically, there are three possible sources of error. Put in question form:

1. Are thermal conductivity and electrical resistivity for a material like stainless steel linear functions of temperature?
2. Is $\frac{t_s + t_w}{2} \stackrel{?}{=} \frac{1}{r_o - r_i} \int_{r_i}^{r_o} t(r) dr$?
3. How does error in various independent variables of the equation affect Δt ?

As far as the first question goes, indeed precision measurements have shown that the electrical resistivity and the thermal conductivity are not linear in an extended region. However, these functions may be linearized in a narrow region and used in equation (46) with good accuracy. Figure 21 attests to this. The errors which are then introduced in the process of linearization belong to the domain of the third question.

To facilitate further discussions, the form of equation (46) is modified as follows.

$$\begin{aligned}
 R &= \frac{M \Delta x^2}{(1 + \gamma t_s)(1 + a t_s)} \\
 &= \frac{I^2 \rho_{es}}{2 \pi^2 j K_s (2r_o - \Delta x)^2} \frac{(1 + a t_m)^2}{(1 + a t_s)^2} \quad \text{C-1}
 \end{aligned}$$

Since $A_c = \pi (r_o^2 - r_i^2) = \pi \Delta x (2r_o - \Delta x)$

and $\rho_{emz} = \rho_{es} \frac{(1 + a t_m)}{(1 + a t_s)}$

Equation (46) may be then written as:

$$\begin{aligned}
 t = t_s - t_w &= R \left[1 + \frac{\Delta x}{3r_o} + \frac{\Delta x^2}{4r_o^2} \right] \\
 &+ R^2 \frac{(3a + 4a \gamma t_s + \gamma)}{6(1 + \gamma t_s)(1 + a t_s)} \quad \text{C-2}
 \end{aligned}$$

The magnitude of the second term on the right hand side of equation (C-2) was found to be less than 1.6 per cent of the total at highest currents used in this work. Therefore, the magnitude of the uncertainty that this term may introduce in Δt is quite limited. In the following the effect of various uncertainties in the first term on the right hand side of (C-2) is considered. Equation C-2 has been now reduced to:

$$\Delta t = \frac{I^2 \rho_{es}}{2 \pi^2 j K_s} \frac{(1 + a t_m)^2}{(1 + a t_s)^2} \frac{1}{(2r_o - \Delta x)^2} \left[1 + \frac{\Delta x}{3r_o} + \frac{\Delta x^2}{4r_o^2} \right] \quad \text{C-3}$$

Taking logarithms and differentiating, the following expression is obtained:

$$\begin{aligned} \frac{\Delta(\Delta t)}{\Delta t} = & \frac{2\Delta I}{I} + \frac{\Delta \rho_{es}}{\rho_{es}} - \frac{\Delta k_s}{k_s} + 2 \frac{\Delta \alpha_{t_m}}{1 + \alpha_{t_m}} - 2 \frac{\Delta \alpha_{t_s}}{1 + \alpha_{t_s}} \\ & - 2 \frac{2\Delta(r_o) - \Delta(\Delta x)}{(2r_o - \Delta x)} + \frac{\Delta\left(\frac{\Delta x}{3r_o} + \frac{\Delta x^2}{4r_o^2}\right)}{\left(1 + \frac{\Delta x}{3r_o} + \frac{\Delta x^2}{4r_o^2}\right)} \end{aligned} \quad C-4$$

Equation (C-4) shows how uncertainties in the variables may reflect in Δt . Table C-I displays an estimate of various uncertainties (which are based on measurements) for both test sections.

TABLE C-I

Estimates of Uncertainty in the Variables
(based on measurements)

Variable	Uncertainty	
	XE	XF
Current, I	±0.5 per cent	±0.5 per cent
Electrical resistivity, ρ_{es}	±0.20 per cent	±0.16 per cent
Thermal conductivity, k_s	±2.5 per cent	-2.5 per cent
Tube outside radius, r_o	±0.0004 inch	±0.0002 inch
Tube wall thickness, Δx	±0.0012 inch	±0.0010 inch

The effect of uncertainty in the temperature coefficient of electrical resistivity, α , is negligible since the two terms in which it appears are nearly equal and of opposite sign.. Also very little error may be expected in α since the equations describing the electrical resistivity were in good agreement with the experimentally determined values over a sufficiently large range.

Table C-II, prepared from the values presented in Table C-I, shows the effect of uncertainty in individual variables and the sum of their absolute magnitudes upon Δt . It is observed that about 50 per cent of the total absolute uncertainty is due to the thermal conductivity.

TABLE C-II

Estimates of Possible Uncertainty Introduced
by Each Variable into $(t_s - t_w)$

Variable	XE	XF
I	1.0 per cent	1.0 per cent
P_{es}	0.20 per cent	0.16 per cent
k_s	2.50 per cent	2.50 per cent
r_o	0.55 per cent	0.19 per cent
Δx	1.05 per cent	0.63 per cent
Total absolute uncertainty	5.30 per cent	4.48 per cent

Since the total uncertainty was computed on the accumulative basis, it may be expected that the actual uncertainties would be lower.

Question two can also be answered from equation (C-3). If all variables except t_m are held constant, then

$$\frac{\Delta(\Delta t)}{\Delta t} = 2 \frac{\alpha \Delta t_m}{1 + \alpha t_m} \quad \text{C-5}$$

Consider the typical case of $t_s = 900$ F and $\Delta t = 100$ F, where $t_m = 850$. Supposing that the value of t_m was in error by 20 F, the error in Δt would be

$$\frac{\Delta(\Delta t)}{\Delta t} = \frac{2 \times 0.000371 (20)}{1 + 0.000371 (850)} = 0.0113$$

Therefore an error of about one per cent may be expected if t_m deviated by about 20 per cent from the true mean.

Summarizing this analysis it may be concluded that accumulative uncertainties of about five per cent may be present in the wall temperature drop computations. A better knowledge of the tube parameters together with better thermophysical data are required for more accurate computations. Also, future studies should be conducted to determine the deviation of t_m from the correct mean temperature. Furthermore, it should be noted that a small error in the outside wall thermocouple reading does not significantly influence Δt through the thermophysical properties. The error is directly transferred to the inside surface temperature t_s .

APPENDIX D

ELECTRICAL RESISTIVITY MEASUREMENT

An apparatus was designed and operated to obtain accurate data on the electrical resistivity of Type 347 stainless steel tube. The purpose of this appendix is: 1) to show the method of measurement; and 2) to contribute to the literature. Ironically, the data obtained in this investigation was in complete agreement with the data reduced from voltage and current measurements on the test section XF of this investigation which was of the 304 type. Thus, the measurements reported here were used in conjunction with the main experiment. Furthermore, Wallace (33) reported good agreement between his measured values for stainless steel Type 321 and this work.

A twenty inch long piece of tube (0.375 inch O.D. and 0.245 inch I.D.) was used as a test section. The ends of the tube were fitted with proper current taps. The middle eight inches of the tube were instrumented for electrical resistivity measurements, with 5-1/2 inches on each side serving as guard heaters. The tube was covered with electrical Sauereisen cement, wound with Nickrom heating wires, and covered with another layer of the cement. Thermocouples were mounted to control the guard heaters and to monitor the test section temperature. Special yokes were designed to assure point contacts at the test section. To these the potential taps were connected. Figure D-1 shows the details

FIGURE D-1. DETAILS OF TEST SECTION

of the test section construction. Proper thermal insulation with several layers of glass wool were provided. Powerstat type heater controls were used.

Instrumentation for electrical resistivity measurements consisted of a Leeds and Northrup Kelvin bridge, appropriate ratio box and two current circuits connected to three six volt heavy duty automobile batteries arranged in series. When satisfactory test section temperature control was achieved, the low current circuit carrying about three amperes was used to balance the bridge roughly. The current was then increased to ten amperes and the bridge was further balanced. Finally, to obtain a precision balance, the high current circuit was operated momentarily with currents up to 70 amperes. This procedure was adapted to avoid Joulean heating of the test section with consequences of unallowable temperature drifts. Large currents are necessary for accurate measurements of this kind, and having two different current circuits allowed prebalancing of the bridge at low currents and only a very brief use of high currents. Figure D-2 shows the complete circuit diagram.

A second degree polynomial least squares curve was fitted to the data. The normalized absolute deviation of the curve from the experimental points was less than 0.083 per cent. The data ranged all the way from room temperature to about 1220 F. Altogether 44 data points were obtained and entered in the curve fit. Unfortunately, due to a premature failure of one of the guard heaters and scarcity of time the work could not be continued at high temperatures.

FIGURE D-2. CIRCUIT FOR THE MEASUREMENT OF ELECTRICAL RESISTIVITY

Figure D-3 shows the results of this measurement together with other data taken from the literature.

FIGURE D-3

ELECTRICAL RESISTIVITY OF STAINLESS STEEL TYPE 347 $\frac{1}{8}$ TEMPERATURE

APPENDIX E

ENTHALPY-PRESSURE-TEMPERATURE TABLES

FOR NEAR CRITICAL WATER

ENTHALPY B/LB	P R E S S U R E, PSIA											
	2900.0	3000.0	3100.0	3150.0	3200.0	3220.0	3240.0	3260.0	3280.0	3300.0	3350.0	3400.0
600.00	592.20	592.45	592.76	592.89	593.00	593.04	593.07	593.11	593.14	593.16	593.22	593.25
605.00	595.90	596.15	596.51	596.67	596.80	596.85	596.89	596.94	596.98	597.01	597.09	597.15
610.00	599.55	599.90	600.21	600.36	600.50	600.55	600.61	600.66	600.71	600.76	600.88	601.00
615.00	603.25	603.60	604.00	604.18	604.35	604.41	604.48	604.54	604.59	604.65	604.78	604.90
620.00	606.85	607.30	607.72	607.90	608.05	608.11	608.16	608.21	608.25	608.29	608.38	608.45
625.00	610.50	611.00	611.30	611.45	611.60	611.66	611.72	611.78	611.84	611.90	612.05	612.20
630.00	614.00	614.55	614.92	615.09	615.25	615.31	615.37	615.43	615.49	615.54	615.68	615.80
635.00	617.45	617.95	618.42	618.62	618.80	618.87	618.93	618.99	619.04	619.09	619.21	619.30
640.00	620.90	621.40	621.89	622.10	622.30	622.37	622.44	622.51	622.58	622.64	622.78	622.90
645.00	624.30	625.00	625.38	625.57	625.75	625.82	625.89	625.97	626.04	626.11	626.28	626.45
650.00	627.55	628.25	628.76	628.99	629.20	629.28	629.36	629.44	629.51	629.58	629.75	629.90
655.00	630.75	631.45	631.96	632.21	632.45	632.54	632.64	632.73	632.82	632.91	633.13	633.35
660.00	633.95	634.65	635.22	635.49	635.75	635.85	635.95	636.05	636.15	636.24	636.48	636.70
665.00	637.20	637.95	638.48	638.74	639.00	639.10	639.20	639.31	639.41	639.51	639.75	640.00
670.00	640.40	641.05	641.64	641.92	642.20	642.31	642.42	642.53	642.63	642.74	643.00	643.25
675.00	643.50	644.20	644.82	645.11	645.40	645.51	645.62	645.73	645.84	645.94	646.20	646.45
680.00	646.55	647.25	647.90	648.20	648.50	648.60	648.70	648.80	648.95	649.05	649.25	649.50
685.00	649.50	650.25	650.95	651.25	651.60	651.70	651.80	651.90	652.00	652.15	652.35	652.60
690.00	652.40	653.05	653.75	654.10	654.45	654.55	654.70	654.80	654.95	655.10	655.35	655.65
695.00	655.15	655.85	656.60	656.90	657.25	657.35	657.50	657.60	657.75	657.90	658.20	658.50
700.00	657.85	658.70	659.45	659.75	660.05	660.20	660.30	660.40	660.55	660.70	660.95	661.25
705.00	660.45	661.35	662.20	662.55	662.90	663.00	663.15	663.25	663.40	663.50	663.80	664.10
710.00	663.00	664.05	664.80	665.20	665.55	665.75	665.80	665.95	666.10	666.25	666.60	666.95
715.00	665.55	666.45	667.30	667.70	668.15	668.35	668.45	668.60	668.80	669.00	669.30	669.70
720.00	668.00	669.00	669.90	670.40	670.80	670.85	671.00	671.25	671.40	671.55	671.85	672.85
725.00	670.40	671.30	672.35	672.85	673.35	673.30	673.50	673.75	673.85	674.20	674.55	674.85
730.00	672.65	673.60	674.75	675.25	675.55	675.65	675.90	675.15	676.25	676.55	677.00	677.55
735.00	674.70	675.70	676.80	677.35	677.85	677.95	678.25	678.45	678.60	678.70	679.25	679.85
740.00	676.65	677.60	678.90	679.45	680.05	680.25	680.55	680.75	680.85	681.00	681.55	682.15
745.00	678.50	679.90	680.95	681.45	682.25	682.45	682.70	682.90	683.05	683.25	683.85	684.45
750.00	680.25	681.50	682.90	683.50	684.35	684.45	684.70	684.95	685.15	685.40	685.95	686.55
755.00	681.95	683.75	684.75	685.50	686.25	686.45	686.70	686.90	687.15	687.45	688.00	688.65
760.00	683.60	685.50	686.55	687.35	688.15	688.35	688.60	688.85	689.10	689.50	690.05	690.75
765.00	685.15	687.05	688.25	689.05	689.95	690.15	690.40	690.65	690.90	691.50	692.10	692.85
770.00	686.65	688.55	689.85	690.70	691.60	691.85	692.10	692.40	692.70	693.30	694.05	694.80
775.00	687.85	689.85	691.35	692.25	693.20	693.50	693.75	694.10	694.40	695.00	695.85	696.55
780.00	689.05	691.10	692.75	693.75	694.75	695.00	695.35	695.60	696.10	696.65	697.35	698.20
785.00	690.10	692.25	694.15	695.10	696.15	696.55	696.85	697.15	697.60	698.15	698.80	699.65
790.00	690.26	693.25	695.45	696.45	697.50	697.90	698.25	698.55	698.95	699.50	700.20	701.05
795.00	690.26	694.10	696.65	697.65	698.70	699.15	699.50	699.85	700.25	700.70	701.50	702.45
800.00	690.26	694.85	697.70	698.80	699.25	700.25	700.65	701.05	701.45	701.85	702.75	703.70
805.00	690.26	695.35	698.55	699.70	700.65	701.25	701.70	702.05	702.53	702.95	703.90	704.95
810.00	690.26	695.35	699.25	700.45	701.45	702.10	702.60	703.00	703.50	703.90	704.95	706.10
815.00	690.26	695.35	699.75	701.00	702.05	702.85	703.35	703.80	704.30	704.80	705.95	707.10
820.00	690.26	695.35	700.10	701.50	702.60	703.45	704.00	704.45	705.00	705.60	706.80	708.00
825.00	690.26	695.35	700.32	701.85	703.10	703.95	704.55	705.00	705.50	706.10	707.55	708.75
830.00	690.26	695.35	700.32	702.15	703.45	704.40	705.00	705.50	706.10	706.80	708.15	709.40
835.00	690.26	695.35	700.32	702.40	703.80	704.75	705.35	705.85	706.55	707.20	708.60	709.95
840.00	690.26	695.35	700.32	702.55	704.05	705.05	705.65	706.20	706.85	707.55	709.05	710.45
845.00	690.26	695.35	700.32	702.55	704.30	705.25	705.90	706.45	707.15	707.85	709.40	710.85
850.00	690.26	695.35	700.32	702.55	704.50	705.40	706.10	706.70	707.40	708.10	709.70	711.25

855.00	690.26	695.35	700.32	702.55	704.70	705.50	706.25	705.90	707.60	708.35	710.00	711.60
860.00	690.26	695.35	700.32	702.55	704.85	705.60	706.35	707.05	707.80	708.55	710.30	711.95
865.00	690.26	695.35	700.32	702.55	705.00	705.70	706.45	707.15	707.95	708.70	710.50	712.25
870.00	690.26	695.35	700.32	702.55	705.10	705.75	706.55	707.30	708.10	708.85	710.75	712.55
875.00	690.26	695.35	700.32	702.55	705.10	705.80	706.60	707.35	708.20	709.00	710.95	712.85
880.00	690.26	695.35	700.32	702.55	705.10	705.85	706.68	707.45	708.25	709.15	711.15	713.10
885.00	690.26	695.35	700.32	702.55	705.10	705.89	706.72	707.50	708.35	709.25	711.30	713.30
890.00	690.26	695.35	700.32	702.55	705.10	705.92	706.77	707.60	708.45	709.35	711.40	713.45
895.00	690.26	695.35	700.32	702.55	705.10	705.95	706.80	707.65	708.50	709.40	711.55	713.55
900.00	690.26	695.35	700.32	702.55	705.10	705.99	706.85	707.70	708.55	709.50	711.65	713.75
905.00	690.26	695.35	700.32	702.55	705.10	706.00	706.90	707.80	708.62	709.60	711.75	713.90
910.00	690.26	695.35	700.32	702.55	705.10	706.02	706.95	707.85	708.70	709.70	711.85	714.05
915.00	690.26	695.35	700.32	702.55	705.10	706.04	706.98	707.90	708.75	709.75	712.00	714.20
920.00	690.26	695.35	700.32	702.55	705.10	706.06	707.02	708.00	708.85	709.85	712.10	714.35
925.00	690.26	695.35	700.32	702.55	705.10	706.09	707.06	708.05	708.90	709.95	712.20	714.50
930.00	690.26	695.35	700.32	702.55	705.12	706.12	707.10	708.10	709.00	710.00	712.30	714.65
935.00	690.26	695.35	700.32	702.55	705.14	706.16	707.17	708.20	709.10	710.10	712.45	714.80
940.00	690.26	695.35	700.32	702.55	705.18	706.20	707.22	708.25	709.20	710.20	712.55	714.95
945.00	690.26	695.35	700.32	702.55	705.20	706.25	707.30	708.35	709.30	710.25	712.70	715.10
950.00	690.26	695.35	700.32	702.55	705.25	706.30	707.35	708.45	709.40	710.35	712.85	715.25
955.00	690.26	695.35	700.32	702.55	705.30	706.37	707.45	708.55	709.50	710.45	713.00	715.40
960.00	690.26	695.35	700.32	702.55	705.37	706.45	707.55	708.65	709.63	710.60	713.20	715.60
965.00	690.26	695.35	700.32	702.55	705.45	706.55	707.65	708.76	709.77	710.75	713.40	715.80
970.00	690.26	695.35	700.32	702.60	705.55	706.65	707.80	708.90	709.90	710.90	713.60	716.00
975.00	690.26	695.35	700.32	702.70	705.70	706.80	707.90	709.05	710.08	711.10	713.80	716.20
980.00	690.26	695.35	700.32	702.85	705.80	706.95	708.10	709.22	710.28	711.30	714.05	716.45
985.00	690.26	695.35	700.32	703.05	705.95	707.15	708.25	709.42	710.50	711.50	714.30	716.70
990.00	690.26	695.35	700.40	703.25	706.20	707.35	708.50	709.65	710.70	711.80	714.55	717.00
995.00	690.26	695.35	700.50	703.50	706.40	707.55	708.75	709.90	711.00	712.05	714.85	717.35
1000.00	690.26	695.35	700.75	703.75	706.70	707.85	709.05	710.17	711.30	712.35	715.20	717.75
1005.00	690.26	695.35	701.05	704.10	707.05	708.25	709.45	710.50	711.68	712.70	715.60	718.15
1010.00	690.26	695.35	701.35	704.45	707.40	708.60	709.80	710.90	712.10	713.15	716.00	718.65
1015.00	690.26	695.35	701.75	704.85	707.85	709.00	710.25	711.35	712.52	713.60	716.45	719.15
1020.00	690.26	695.65	702.15	705.30	708.30	709.50	710.75	711.85	713.00	714.10	717.00	719.70
1025.00	690.26	696.00	702.60	705.80	708.80	710.00	711.30	712.40	713.50	714.65	717.55	720.25
1030.00	690.26	696.40	703.10	706.35	709.35	710.60	711.90	713.00	714.10	715.25	718.20	720.90
1035.00	690.26	696.85	703.65	706.90	709.95	711.25	712.55	713.65	714.72	715.85	718.85	721.60
1040.00	690.50	697.35	704.25	707.50	710.50	711.95	713.20	714.35	715.40	716.55	719.60	722.40
1045.00	691.00	697.50	704.85	708.20	711.15	712.65	713.95	715.10	716.12	717.25	720.40	723.15
1050.00	691.55	698.50	705.55	708.90	711.90	713.45	714.75	715.88	716.85	718.05	721.20	724.00
1055.00	692.15	699.15	706.35	709.70	712.70	714.25	715.55	716.70	717.67	718.90	722.10	724.90
1060.00	692.85	699.85	707.15	710.55	713.55	715.15	716.45	717.55	718.55	719.80	723.05	725.85
1065.00	693.65	700.70	708.00	711.45	714.50	716.10	717.35	718.50	719.46	720.75	724.00	726.85
1070.00	694.55	701.55	708.95	712.40	715.45	717.10	718.30	719.40	720.45	721.75	725.05	727.85
1075.00	695.45	702.50	709.90	713.45	716.50	718.15	719.35	720.40	721.50	722.85	726.10	728.95
1080.00	696.45	703.55	710.90	714.55	717.55	719.20	720.45	721.50	722.60	724.00	727.20	730.05
1085.00	697.45	704.65	712.00	715.70	718.70	720.35	721.60	722.65	723.83	725.20	728.35	731.25
1090.00	698.60	705.75	713.20	716.95	719.90	721.55	722.80	723.90	725.05	726.50	729.50	732.45
1095.00	699.80	707.00	714.45	718.25	721.15	722.85	724.05	725.15	726.35	727.80	730.75	733.75
1100.00	701.05	708.25	715.80	719.55	722.45	724.15	725.40	726.47	727.70	729.20	732.10	735.10
1105.00	702.40	709.55	717.20	721.05	723.90	725.55	726.75	727.85	729.10	730.60	733.60	736.55
1110.00	703.80	710.95	718.65	722.50	725.30	726.95	728.20	729.30	730.50	732.15	735.15	738.10
1115.00	705.40	712.45	720.15	724.00	726.75	728.35	729.65	730.75	732.00	733.75	736.80	739.75
1120.00	707.00	714.05	721.75	725.45	728.30	729.90	731.10	732.25	733.60	735.50	738.55	741.50
1125.00	708.75	715.75	723.50	727.00	729.90	731.45	732.65	733.92	735.35	737.35	740.35	743.30
1130.00	710.50	717.60	725.20	728.55	731.55	733.15	734.40	735.70	737.20	739.20	742.20	745.15
1135.00	712.35	719.60	727.00	730.25	733.35	735.00	736.25	737.65	739.25	741.10	744.05	747.00
1140.00	714.25	720.65	728.85	732.05	735.25	736.95	738.25	739.70	741.25	743.00	745.90	748.80
1145.00	716.25	723.75	730.65	734.00	737.35	739.10	740.40	741.80	743.30	744.90	747.75	750.55
1150.00	718.30	725.80	732.50	736.05	739.60	741.30	742.65	743.95	745.35	746.80	749.65	752.50
1155.00	720.50	727.85	734.60	738.40	741.85	743.55	744.95	746.10	747.45	748.75	751.50	754.60

1160.00	722.90	730.00	736.95	740.80	744.15	745.85	747.20	748.25	749.50	750.75	753.70	756.85
1165.00	725.25	732.10	739.50	743.20	746.45	748.10	749.40	750.55	751.70	752.90	756.05	759.30
1170.00	727.70	734.50	742.05	745.55	748.75	750.35	751.65	752.90	754.05	755.20	758.55	761.75
1175.00	730.10	737.15	744.60	748.00	751.20	752.80	754.20	755.40	756.55	757.70	761.05	764.35
1180.00	732.75	740.00	747.25	750.25	753.80	755.80	756.75	757.95	759.15	760.30	763.70	766.95
1185.00	735.45	742.75	749.90	752.90	756.45	757.55	759.40	760.60	761.75	762.95	766.30	769.60
1190.00	738.25	745.60	752.50	755.60	759.15	760.78	762.15	763.35	764.50	765.65	769.10	772.35
1195.00	741.10	748.50	755.25	758.45	762.00	763.60	765.00	766.15	767.40	768.55	771.95	775.15
1200.00	744.15	751.50	758.15	761.35	764.95	766.55	767.95	769.15	770.30	771.50	774.90	778.05
1205.00	747.25	754.45	761.15	764.45	767.97	769.65	771.00	772.05	773.35	774.55	777.85	780.95
1210.00	750.35	757.50	764.30	767.55	771.20	772.80	774.10	775.20	776.45	777.70	780.95	783.95
1215.00	753.45	760.75	767.55	770.90	774.40	776.05	777.30	778.40	779.75	780.95	784.20	787.15
1220.00	756.75	764.10	770.85	774.25	777.70	779.35	780.50	781.55	783.05	784.25	787.43	790.45
1225.00	760.20	767.55	774.35	777.65	781.10	782.75	783.85	785.05	786.40	787.55	790.75	793.85
1230.00	763.75	771.10	777.80	781.05	784.55	786.20	787.30	788.65	789.90	790.85	794.25	797.35
1235.00	767.45	774.65	781.40	784.60	788.05	789.75	790.90	792.20	793.40	794.40	797.80	800.90
1240.00	771.25	778.35	785.00	788.30	791.70	793.40	794.55	795.75	796.90	798.05	801.45	804.50
1245.00	775.10	782.00	788.75	792.03	795.35	797.05	798.30	799.45	800.55	801.85	805.15	808.30
1250.00	779.10	785.95	792.65	795.85	799.10	800.80	802.00	803.15	804.35	805.60	808.95	812.05
1255.00	783.00	790.00	796.60	799.70	802.95	804.65	805.80	807.10	808.25	809.50	812.75	815.80
1260.00	787.20	794.00	800.55	803.75	806.85	808.70	809.82	811.15	812.20	813.48	817.55	819.60
1265.00	791.50	798.10	804.55	807.85	811.00	812.75	813.90	815.15	816.25	817.55	820.38	823.40
1270.00	795.75	802.25	808.75	812.00	815.15	816.90	818.05	819.15	820.30	821.50	824.20	827.40
1275.00	800.05	806.65	813.05	816.35	819.45	821.00	822.20	823.25	824.35	825.55	828.45	831.60
1280.00	804.25	811.20	817.50	820.60	823.65	825.00	826.25	827.55	828.60	829.80	832.80	835.90
1285.00	808.75	815.80	822.10	824.90	828.00	829.30	830.60	831.90	833.00	834.30	837.20	840.25
1290.00	813.55	820.40	826.45	829.20	832.55	833.75	835.10	836.45	837.55	838.70	841.75	844.80
1295.00	818.50	825.00	831.05	833.90	837.15	838.55	839.85	840.95	842.05	843.25	846.35	849.40
1300.00	823.60	829.55	835.75	838.65	841.90	843.40	844.60	845.70	846.80	847.90	851.05	853.95

APPENDIX F

CALCULATIONS FOR PEAK ENTHALPY

Peak enthalpy was computed by several different methods with efforts to obtain more consistent results. Nowak and Grosh (27) reported that the inflection of the isobars of the product of enthalpy and temperature versus enthalpy were always found to occur at enthalpy values from 900 to 920 B/lb. This is very close to the value of the enthalpy at the critical point, $H_c = 900$ B/lb. In (27) it was also reported that the uncertainty in this latter value could be as high as two or three per cent. It is, therefore, quite reasonable that the computation of H_p should present difficulties.

One method of obtaining H_p was by direct reference to the enthalpy-pressure-temperature plots prepared in conjunction with the present study. A second method was to fit a least squares straight line to the H_{xt} versus H data in the range $H = 890$ to 925 B/lb_m, and to compute H_p from the coefficients obtained therefrom. Thus, if

$$H_{xt} = a + bH \quad F-1$$

then

$$H_p = \frac{a}{t_p - b} \quad F-2$$

Although the maximum deviation of the straight lines from the data was less than 0.015 per cent, with a much lower rms deviation, the results

were not smooth. Figure F-1 shows the results obtained from this and the first method. It is observed that the values are well within the range described by Nowak and Grosh.

A third possible method was to derive the value of H_p starting from another point. The change in the enthalpy may be expressed as

$$dH = \left(\frac{\partial H}{\partial P}\right)_t dp + \left(\frac{\partial H}{\partial t}\right)_p dt \quad F-3$$

which upon the substitution of the expression

$$v - T \left(\frac{\partial v}{\partial t}\right)_p = \left(\frac{\partial H}{\partial P}\right)_t \quad F-4$$

and integration becomes:

$$\Delta H = \int_{r_1}^{P_2} \left[v - T \left(\frac{\partial v}{\partial t}\right)_p \right] dp + \int_{t_1}^{t_2} c_p dt \quad F-5$$

The integration of the first term does not present any difficulties. However, the method runs into many difficulties precipitated by the specific heat term.

Another method took into account the non-linearity of the Hxt versus H curve removing the linearity restriction imposed on the second method. By successive differentiation it may be shown that:

$$\left(\frac{\partial(Ht)}{\partial H}\right)_p = t + \frac{H}{c_p} \quad F-5$$

FIGURE F-1. PRESSURE AND ENTHALPY
CORRESPONDING TO PEAK ENTHALPY ALONG
ISOBARS

$$\left(\frac{\partial^2(Ht)}{\partial H^2}\right)_p = \frac{2}{c_p} - \frac{H}{c_p^3} \left(\frac{\partial c_p}{\partial t}\right)_p$$

F-6

and so on. The fact that $\left(\frac{\partial c_p}{\partial t}\right)_p$, which has a large magnitude near the peak temperatures, enters the derivatives makes it necessary that higher degree polynomials or equations be employed. However, polynomial curve fits are known to be inadequate for thermodynamic data.

Thus, no accurate values for H_p could be obtained, and the best guess would be the approximate straight line shown in Figure F-1. This was based on the assumption that accuracy could be better at higher supercritical pressures where the magnitude of $\left(\frac{\partial c_p}{\partial t}\right)_p$ becomes smaller.

APPENDIX G

IBM

7094 FORTRAN II

COMPUTER PROGRAMS

1. Heat Balance Program

Input:

M6	Number of enthalpy-temperature values
M7	Number of pressures
P(J2)	Pressures
HOT	Starting enthalpy value
ET(J1,J2)	Temperature data spaced every 5.0 B/lb _m starting at HOT for various pressures
TMVL	Starting temperature for thermocouple tables
TMV	Entire matrix of (21, 6) giving millivolt values for temperature spaced every 5.0 F., Same order as Leeds and Northrup tables.
ZL, Z(I)	Distance between pressure taps, pressure drop lengths, in inches.
M3	Test run number
M4	00 = last set of data, 01 = continue, more to come
M8	00 = venturi pressure drop in psi, 01 = pressure drop in inches of mercury
M9	Number of times to iterate for venturi discharge coefficient
L	System pressure code corresponding to the order of enthalpy-pressure-temperature data made available
VTM	Venturi surface; preheater inlet, outlet; test section inlet, outlet temperatures in millivolts
SVV	Specific volume at venturi
AMPS	Test section current
VOLTS	Test section voltage
DPVI	Large venturi pressure drop

DPT   Test section pressure drop

PI   System pressure at test section inlet

VISV   Viscosity at venturi, will compute if VISV = 0.0

DPV2   Small venturi pressure drop, same units as DPVI

\$ID 3105*003*050*050**R.F.TOUBA --DATA REDUCTION-- XF

R*R*R*

\$EXECUTE FORTRAN

* XEQ
* BIN
* DUMP
* LABEL

C P303 DATA REDUCTION - HEAT BALANCE COMPUTATIONS

DIMENSION ET(160, 12), P(12)
DIMENSION VTWM(40), TMV(21, 6)
DIMENSION VTM(5), TM(5), HSI(2), HS(5), H(5), Z(5)
DIMENSION TWM(5), TS(5), TMI(5)

READ INPUT TAPE 5, 53, M6, M7
READ INPUT TAPE 5, 56, (P(J2), J2=1, M7)
READ INPUT TAPE 5, 55, HOT
READ INPUT TAPE 5, 56, ((ET(J1, J2), J1=1, M6), J2=1, M7)
READ INPUT TAPE 5, 55, TMVL
READ INPUT TAPE 5, 55, TMV
READ INPUT TAPE 5, 55, ZL, (Z(I), I=1, 4)

303 READ INPUT TAPE 5, 54, M3
READ INPUT TAPE 5, 54, M4, M8, M9, L
READ INPUT TAPE 5, 55, VTM
READ INPUT TAPE 5, 56, SVV, AMPS, VOLTS, DPV1, DPT, PI, VISV, DPV2

DO100 I=1, 5
VTWM(I) = VTM(I)
J=1

II=1

229 IF(VTWM(I)-TMV(II, J)) 230, 231, 232

232 J=J+1

GO TO 229

230 J=J-1

235 IF(VTWM(I)-TMV(II, J)) 233, 231, 234

234 II=II+1

GO TO 235

233 II=II-1

231 AJ=J-1

AI=II-1

TWM(I) = AJ*100.0 + (AI + (VTWM(I)-TMV(II, J))/(TMV(II+1, J)-TMV(II, J)))*5.0 + TMVL

TMI(I) = TWM(I)

100 CONTINUE

TM(1) = TMI(1)

TM(2) = TMI(2)-(5.58+0.00444*(TMI(2)-626.9))

TM(3) = TMI(3)-(5.58+0.00444*(TMI(3)-626.9))

TM(4)=TMI(4)-6.85

TM(5)=TMI(5)-6.85

IF(VISV) 237, 237, 238

237 VISV = ((37.94E-10)*(1.0/SVV)**1.57 + (0.04788E-08)*TM(2) +
1 (14.79E-08))*115920.0

238 TV = TM(1)

E = 0.9985 + 0.00002*TV

IF(M8) 210, 210, 211

210 FRS1 = 0.01489*E*SQRTE(DPV1/(0.455*SVV))

FRS2 = 0.004350*E*SQRTE(DPV2/(0.455*SVV))

GO TO 212

211 FRS1 = 0.01489*E*SQRTE(DPV1/SVV)

FRS2 = 0.004350*E*SQRTE(DPV2/SVV)

212 FRSA1 = FRS1

FRSA2 = FRS2

N=M9

200 REV1 = 268300.0*FRS1/VISV

```

REV2 = 496100.0*FTEMP/VISV
DC1 = 0.8270 + 0.07466*(REV1 + 1.0)*0.07
DC2 = 0.468 + 0.0436*LOGF(REV2 + 1.1)
FRS1 = FRSA1*DC1
FRS2 = FRSA2*DC2
N = N-1
IF(N) 201, 201, 200
201 FRH1 = FRS1*3600.0
FRH2 = FRS2*3600.0
FRH = FRH1 + FRH2
J=5
DO109 I=1, 4
TS(I) = TM(I+1)
J5= 0
DO108 K=1, 2
NJ=L-J5
IF(TS(I)-ET(J,NJ)) 220, 221, 222
220 J=J-1
IF(TS(I)-ET(J,NJ)) 220, 221, 221
222 J=J+1
IF(TS(I)-ET(J,NJ)) 223, 221, 222
223 J=J-1
221 AJ=J-1
HSI(K) = HOT + (AJ + (TS(I)-ET(J,NJ))/(ET(J+1,NJ)-ET(J,NJ)))*5.0
108 J5=J5+1
OHS(I)=HSI(2)+((PI-DPT*Z(I)/ZL)-P(L-1))/(P(L)-P(L-1))*(HSI(1)-
1 HSI(2))
H(I) = HS(I)
109 CONTINUE
QATS = FRH*(H(4)-H(3))
QGTS = AMPS*VOLTS*3.4128
HBERR = (QGTS-QATS)/QGTS*100.0
QAPH = FRH*(H(2)-H(1))
HO=H(3)
WRITE OUTPUT TAPE 6, 81
WRITE OUTPUT TAPE 6, 65, (M3)
WRITE OUTPUT TAPE 6, 82
WRITE OUTPUT TAPE 6, 83, (VTM(1), TMI(1), TM(1), I=1, 5)
WRITE OUTPUT TAPE 6, 70, (FRH, FRH1, FRH2)
WRITE OUTPUT TAPE 6, 84, (QGTS, QATS, HBERR, QAPH, PI, VISV)
WRITE OUTPUT TAPE 6, 85, (DPV1, DPT, TV, HO, SVV, DPV2)
PUNCH 93, (DPV1, DPT, TV, HO, SVV, VISV, PI, DPV2)
81 FORMAT (1H1, 7X, 28H HEAT BALANCE COMPUTATIONS//)
65 FORMAT (31X, 8H RUN XF, 12////)
82 FORMAT (10X, 6HVTM,MV, 10X, 5HTMI,F, 10X, 4HTM,F, //)
83 FORMAT (10X, F6.3, 8X, F7.2, 8X, F7.2)
70 FORMAT (////, 10X, 14H FLOW RATE = , F11.1, 9H LBS/HR//,
1 10X, 14H FRH1 = , F11.1, 9H LBS/HR//,
2 10X, 14H FRH2 = , F11.1, 9H LBS/HR//)
84 FORMAT (11X, 12HQGTS = , 4X, F7.0, 8H B/HR//,
1 11X, 12HQATS = , 4X, F7.0, 8H B/HR//,
2 11X, 12HHBERR = , 4X, F8.1, 10H PERCENT//,
3 11X, 12HQAPH = , 4X, F7.0, 8H B/HR///// ,
4 11X, 12HPRESSURE = , 4X, F8.1, 7H PSIA//,
5 11X, 12HVISCOSITY = , 5X, F9.3, 8H LB/HRFT//)
85 FORMAT (4F15.2, F15.5, F15.2)
93 FORMAT (8F10.5)
IF(M4) 206, 206, 303
206 CALL EXIT
53 FORMAT (214)
54 FORMAT (514)
55 FORMAT (7F10.0)
56 FORMAT (8F10.0)
END

```

2. Data Reduction and Analysis Program

Input:

(Includes symbols not covered previously)

- K(J) Number of intervals between thermocouples evenly spaced.
For example XE test section 8, 2, 6, 2, 8
- DZ(J) Length of the above intervals in inches
For example XE test section 0.50, 3.0, 6.0, 3.0, 0.50
- DO Tube outside diameter, inches
- DI Tube inside diameter, inches
- WT Tube wall thickness, inches
- ALFA
- BETA
- RHOEO e_o
- TKO k_o
- RHO,RH1,RH2 coefficients of second degree electrical resistivity equations
- B Iteration accuracy for t_m in Kreith Summerfield equation
- B1, B2 Thermocouple calibration constants equation (1)
- B3, B4 Thermocouple calibration constants equation (2)
- HG X, Heat gain factor
- M2 Number of test section thermocouples
- M5 Thermocouple number where calibration shifts from equation (1) to (2). Numbering starts at start of heating, inlet.
If not needed $M5 = M2$.
- VTNM(I) Measured millivolts of test section thermocouples entered from inlet up.
- DPVI,DPT,TV,HO,SVV,VISV,PI,DPV2 This card is punched out by the Heat Balance Program
- TV Venturi surface temperature, F
- HO Test section inlet enthalpy

```

$ID      3105*005*250*000**R.F.TOURA  --DATA REDUCTION--  XF      R#R#R#
$EXECUTE      FORTRAN
*      XEQ
*      BIN
*      DUMP
*      LABEL
C P302 COMPLETE DATA REDUCTION AND ANALYSIS  --  RUNS XE AND XF
  DIMENSION ET(160,12), P(12)
  DIMENSION VTWM(40), TMV(21,6)
  DIMENSION TWM(40), K(5), DZ(5)
  DIMENSION TW(40), TS(40), N(40), RHOE(40), HFLUX(40)
  DIMENSION HZ(40), Z(40), TB(40), HTC(40)
  DIMENSION HSI(2), HS(40), STE(40), DTSB(40), DESB(40)
  DIMENSION TSA(40), TBA(40), HTCA(40), STTBA(40), STEA(40),
  1HSA(40), HZA(40)
  DIMENSION TBTP(40), TSTP(40), TBATP(40), TSATP(40), ZDI(40),
  1 ZADI(40), PBOT(40), FTTP(40), APBOT(40), AFTTP(40)
  READ INPUT TAPE 5, 53, M6, M7
  READ INPUT TAPE 5, 56, (P(J2), J2=1, M7)
  READ INPUT TAPE 5, 55, HOT
  READ INPUT TAPE 5, 56, ((ET(J1,J2), J1=1, M6), J2=1, M7)
  READ INPUT TAPE 5, 55, TMVL
  READ INPUT TAPE 5, 55, TMV
  READ INPUT TAPE 5, 54, (K(J), J=1, 5)
  READ INPUT TAPE 5, 55, (DZ(J), J=1, 5), ZL
  READ INPUT TAPE 5, 55, DO, D1, WT, ALFA, BETA, RHOEO, TKO
  READ INPUT TAPE 5, 55, RH0, RH1, RH2
  READ INPUT TAPE 5, 55, B, B1, B2, B3, B4
  READ INPUT TAPE 5, 55, HG
301 READ INPUT TAPE 5, 54, M2, M3, M4, M5
  READ INPUT TAPE 5, 53, M8, M9
  READ INPUT TAPE 5, 53, L
  READ INPUT TAPE 5, 55, (VTWM(I), I=1, M2), AMPS, VOLTS
  READ INPUT TAPE 5, 56, DPV1, DPT, TV, HO, SVV, VISV, PI, DPV2
  DO100 I=1, M2
    J=1
    11=1
229 IF(VTWM(I)-TMV(11,J)) 230, 231, 232
232 J=J+1
    GO TO 229
230 J=J-1
235 IF(VTWM(I)-TMV(11,J)) 233, 231, 234
234 11=11+1
    GO TO 235
233 11=11-1
231 AJ=J-1
    AI=11-1
    TWM(I) = AJ*100.0 + (AI + (VTWM(I)-TMV(11,J))/(TMV(11+1,J)-TMV(11,
  1 J)))*5.0 + TMVL
100 CONTINUE
  WRITE OUTPUT TAPE 6, 50
  WRITE OUTPUT TAPE 6, 51
  WRITE OUTPUT TAPE 6, 65, (M3)
  WRITE OUTPUT TAPE 6, 52
  WRITE OUTPUT TAPE 6, 57, (VTWM(I), TWM(I), I=1, M2)
50 FORMAT (1H1, 53H MILLIVOLT-TEMPERATURE CONVERSION FOR THERMOCOUP
  1LFS//)
51 FORMAT (38H TYPE CHROMEL ALUMEL COUPLES///)
65 FORMAT (//, 21X, 8H RUN XF, 12 ////)
52 FORMAT (12X, 7HVTWM,MV, 10X, 5HTWM,F//)

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```

57 FORMAT (12X, F6.3, 10X, F7.2)
AC = (DO**2 - DI**2)*3.14159/4.0
A = (40.9536*AMPS**2*RHOEO)/(2.0*TKO*AC**2)
A1 = 1.0 + 2.0*WT/(3.0*DO) + (WT/DO)**2
DTA = 100.0
DO101 I=1, M2
N(I) = 0
IF(I - M5) 207, 207, 208
207 TW(I) = TWM(I) + B1*(TWM(I) - B2)
GO TO 209
208 TW(I) = TWM(I) + B3*(TWM(I) - B4)
209 A2 = (1.0 + ALFA*TW(I))*(1.0 + BETA*TW(I))
A3 = A*WT**2/A2
A4 = A3*(3.0*ALFA + 4.0*ALFA*BETA*TW(I) + BETA)/(6.0*A2)
203 N(I) = N(I) + 1
TM = TW(I) - DTA/2.0
A5 = (1.0 + ALFA*TM)**2
DT = A3*A5*(A1 + A4*A5)
IF(ABS(DT - DTA) - B) 205, 205, 204
204 DTA = DT
GO TO 203
205 TS(I) = TW(I) - DT
TM = (TW(I) + TS(I))/2.0
RHOE(I) = RHO*(1.0 + RH1*TM + RH2*TM**2)
101 CONTINUE
SUM = 0.0
I = 1
DO102 J=1, 5
I2 = I+K(J)
PSUM = (RHOE(I) + RHOE(I2))/2.0
K1 = I + 1
K2 = I+K(J)-1
DO103 I=K1, K2
103 PSUM = PSUM + RHOE(I)
SUM = SUM + DZ(J)*PSUM
102 I = I2
A6 = (144.0*3.4128/(3.14159*AC*DI))*AMPS**2
A7 = AC/ZL
RHOEA = SUM/ZL
RHOEM = A7*VOLTS/AMPS
RHOER = RHOEM/RHOEA
DO104 I=1, M2
104 HFLUX(I) = A6*RHOE(I)*RHOER
WRITE OUTPUT TAPE 6, 64
WRITE OUTPUT TAPE 6, 66, (RHOER)
WRITE OUTPUT TAPE 6, 67
OWRITE OUTPUT TAPE 6, 68, (TWM(I), TW(I), TS(I), N(I), HFLUX(I),
1 I=1, M2)
64 FORMAT (1H1, ///, 69H COMPUTATION OF INSIDE SURFACE TEMPERATURE
1AND LOCAL WALL HEAT FLUX ////)
66 FORMAT (11X, 3HRHOER = , F6.4///)
670FORMAT (11X, 5HTWM,F, 8X, 4HTW,F, 9X, 4HTS,F, 8X, 1HN, 5X, 14HHFLU
1X,P/HRSQFT/)
68 FORMAT (10X, F6.2, 7X, F6.2, 7X, F6.2, 6X, 12, 7X, F7.0)
E = 0.9985 + 0.0000236*TV
IF(MB) 210, 210, 211
210 FRS1 = 0.01489*E*SQRTE(DPV1/(0.455*SVV))
FRS2 = 0.004550*E*SQRTE(DPV2/(0.455*SVV))
GO TO 212
211 FRS1 = 0.01489*E*SQRTE(DPV1/SVV)

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```

      FRS2 = 0.004350*E*SQRT(DPV2/SVV)
210  FRSA1 = FRS1
      FRSA2 = FRS2
      N=M9
200  REV1 = 268300.0*FRS1/VISV
      REV2 = 496100.0*FRS2/VISV
      DC1 = 0.8270 + 0.07466*(REV1 + 1.0)**0.07
      DC2 = 0.468 + 0.0436*LOGF(REV2 + 1.1)
      FRS1 = FRSA1*DC1
      FRS2 = FRSA2*DC2
      N = N-1
      IF(N) 201, 201, 200
201  FRH1 = FRS1*3600.0
      FRH2 = FRS2*3600.0
      FRH = FRH1 + FRH2
      FFLUX = 576.0*FRH/(3.14159*D1**2)
      A8 = 3.14159*D1/(144.0*FRH)
      A9 = A8*HG
      I=1
      Z(I) = 0.0
      HZ(I) = H0
      DO105 J=1, 5
        K1 = I+1
        K2 = I + K(J)
        DO106 I=K1, K2
          HZ(I) = HZ(I-1) + A9*(HFLUX(I-1) + HFLUX(I))/2.0*DZ(J)
106  Z(I) = Z(I-1)+DZ(J)
105  I=K2
      DO107 I=1, M2
        A10 = (HZ(I) - HOT)/5.0
        NAA = A10
        RN = NAA
        NA = NAA + 1
        A11 = A10 - BN
        TB1 = ET(NA,L) + (ET(NA+1,L) - ET(NA,L))*A11
        TB2 = ET(NA,L-1) + (ET(NA+1,L-1) - ET(NA,L-1))*A11
        TB(I) = TB2 + ((PI-DPT*Z(I)/ZL)-P(L-1))/(P(L)-P(L-1))*(TB1-TB2)
        HTC(I) = HFLUX(I)/(TS(I)-TB(I))
107  CONTINUE
      J=5
      DO109 I=1, M2
        J5= 0
        DO108 K=1, 2
          NJ=L-J5
          IF(TS(I)-ET(J,NJ)) 220, 221, 222
220  J=J-1
          IF(TS(I)-ET(J,NJ)) 220, 221, 221
222  J=J+1
          IF(TS(I)-ET(J,NJ)) 223, 221, 222
223  J=J-1
221  AJ=J-1
      HSI(K) = HOT + (AJ + (TS(I)-ET(J,NJ))/(ET(J+1,NJ)-ET(J,NJ)))*5.0
108  J5=J5+1
      OHS(I)=HSI(2)+((PI-DPT*Z(I)/ZL)-P(L-1))/(P(L)-P(L-1))*(HSI(1)-
1      HSI(2))
      DTSB(I) = TS(I)-TB(I)
      DESB(I) = HSI(I) - HZ(I)
      STE(I) = HFLUX(I)/(DESB(I)*FFLUX)
109  CONTINUE
      DO110 I=2,M2

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```

TSA(1) = (TS(1) + TS(1-1))/2.0
TBA(1) = (TB(1) + TB(1-1))/2.0
HSA(1) = (HS(1) + HS(1-1))/2.0
HZA(1) = (HZ(1) + HZ(1-1))/2.0
HTCA(1) = (HFLUX(1) + HFLUX(1-1))/(2.0*(TSA(1)-TBA(1)))
STTBA(1) = HTCA(1)/((HZ(1)-HZ(1-1))/(TB(1)-TB(1-1))*FFLUX)
STEA(1) = (HFLUX(1) + HFLUX(1-1))/(2.0*(HSA(1) - HZA(1))*FFLUX)
110 CONTINUE
WRITE OUTPUT TAPE 6, 69
WRITE OUTPUT TAPE 6, 70, (FRH, P1)
WRITE OUTPUT TAPE 6, 71, (FFLUX)
WRITE OUTPUT TAPE 6, 72
WRITE OUTPUT TAPE 6, 73, (Z(1), TB(1), TS(1), HZ(1), HS(1),
1 HTC(1), STE(1), I=1, M2)
69 FORMAT (1H1, ///, 19X, 80H LOCAL TEMPERATURES, ENTHALPIES, HEAT TR
1ANSFER COEFFICIENTS, AND STANTON NUMBERS////)
70 FORMAT (11X, 14H FLOW RATE = , F11.1, 9H LBS/HR, 10X,
1 12H PRESSURE = , F6.1, 7H PSIA //)
71 FORMAT (11X, 14H MASS FLUX = , 1PE11.4, 13H LBS/HR SQFT////)
72 FORMAT (16X, 4HZ, 1N, 9X, 4HTB, F, 10X, 4HTS, F, 8X, 7HHB, B/LB, 6X,
1 7HHS, B/LB, 4X, 13HHTC, B/HR SQFT, 7X, 3HSTE //)
73 FORMAT (16X, F4.1, 7X, F7.2, 7X, F7.2, 7X, F6.1, 7X, F6.1, 7X,
1 F7.0, 7X, F8.6)
WRITE OUTPUT TAPE 6, 74
WRITE OUTPUT TAPE 6, 75
WRITE OUTPUT TAPE 6, 76, (Z(1), TS(1), TB(1), HS(1), HZ(1),
1 DTSB(1), DESB(1), I=1, M2)
74 FORMAT (1H1, ///, 30X, 60H LOCAL SURFACE AND BULK TEMPERATURE, AND
1ENTHALPY DIFFERENCES////)
75 FORMAT (17X, 4HZ, 1N, 9X, 4HTS, F, 10X, 4HTB, F, 8X, 7HHS, B/LB, 6X,
1 7HHB, B/LB, 7X, 6HDTSB, F, 6X, 9HDESB, B/LB //)
76 FORMAT (17X, F4.1, 7X, F7.2, 7X, F7.2, 7X, F6.1, 7X, F6.1, 7X,
1 F7.2, 7X, F7.2)
WRITE OUTPUT TAPE 6, 77
WRITE OUTPUT TAPE 6, 78
WRITE OUTPUT TAPE 6, 79, (TSA(1), TBA(1), HSA(1), HZA(1), HTCA(1),
1 STTBA(1), STEA(1), I=2, M2)
77 FORMAT (1H1, ///, 16X, 87H LOCAL AVERAGE TEMPERATURES, ENTHALPIES,
1HEAT TRANSFER COEFFICIENTS, AND STANTON NUMBERS////)
78 FORMAT (14X, 5HTSA, F, 9X, 5HTBA, F, 7X, 8HHTSA, B/LB, 6X, 8HHBA, B/LB,
1 4X, 14HHTCA, B/HR SQFT, 5X, 5HSTTBA, 10X, 4HSTEA //)
79 FORMAT (13X, F7.2, 7X, F7.2, 7X, F7.2, 7X, F7.2, 7X, F7.2, 7X,
1 F8.6, 7X, F8.6)
IF (PI-3208.0) 241, 241, 240
240 DO111 I=1, M2
ZDI(1) = Z(1)/DI
TT = 705.50 + 0.044643*((PI - DPT*Z(1)/ZL - 3208.00)
TBT(1) = TB(1)/TT
TST(1) = TS(1)/TT
PB(1) = (TT-TB(1))/DTSB(1)
FTTP(1) = (TS(1)+TB(1))/(2.0*TT)
111 CONTINUE
DO112 I= 2, M2
ZA = (Z(1)+Z(I-1))/2.0
ZADI(1) = ZA/DI
TIA = 705.40 + 0.044693*((PI-DPT*ZA/ZL) - 3206.20)
TBA(1) = TBA(1)/TIA
TSA(1) = TSA(1)/TIA
APB(1) = (TIA-TBA(1))/(TSA(1)-TBA(1))
AFTTP(1) = (TSA(1)+TBA(1))/(2.0*TIA)

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112 CONTINUE
    WRITE OUTPUT TAPE 6, 80
    WRITE OUTPUT TAPE 6, 81, ( ZDI(1), TBTP(1), TSTP(1), PBOT(1),
1   FTTP(1), HTC(1), STE(1), I=1, M2)
80 FORMAT(1H1, ///, 30X, 60H LOCAL PARAMETERS VERSUS NORMALIZED DISTA
    NCE AND TEMPERATURE, ///, 17X, 4HZ/DI, 8X, 5HTD/TP, 8X, 5HTS/TP,
    2 8X, 5HPB/DT, 8X, 5HFT/TP, 6X, 13HHTC,B/HRSQFTF, 6X, 3HSTE //)
81 FORMAT (17X, F5.1, 7X, F6.4, 7X, F6.4, 6X, F7.4, 7X, F6.4, 6X,
1   F8.0, 7X, F8.6)
    WRITE OUTPUT TAPE 6, 82
    WRITE OUTPUT TAPE 6, 83, (ZADI(1), TBATP(1), TSATP(1), APBOT(1),
1   AFTTP(1), HTCA(1), STTBA(1), STEA(1), I=2, M2)
82 FORMAT(1H1, ///, 26X, 60H AVERAGL LOCAL PARAMETERS VERSUS NORMALIZ
    ED DISTANCE AND TEMPERATURE, ///, 10X, 5HZA/DI, 7X, 6HTBA/TP, 7X,
    2 6HTSA/TP, 7X, 6HAPB/DT, 7X, 6HAFT/TP, 4X, 14HHTCA,B/HRSQFTF, 5X,
    3 5HSTTDA, 10X, 4HSTEA//)
83 FORMAT (10X, F5.1, 7X, F6.4, 7X, F6.4, 6X, F7.4, 7X, F6.4, 6X,
1   F8.0, 7X, F8.6, 7X, F8.6 )
241 IF(M4) 206, 206, 301
206 CALL EXIT
    53 FORMAT (214)
    54 FORMAT (514)
    55 FORMAT (7F10.0)
    56 FORMAT (8F10.0)
    END

```

APPENDIX H

SAMPLE COMPUTATIONS

Three sets of sample computations are presented in the following. The data shows the behavior and the magnitude of several parameters of interest under various conditions of operation. Run XF 9 had low film temperature drops. The symbols are explained below.

VTM, MV	= Thermocouple potentials for venturi wall, and preheater and test section inlet and outlet, millivolts
TMI, F	= Indicated temperature
TM, F	= True temperature
FRH 1	= Large venturi flow rate
FRH 2	= Small venturi flow rate
QGTS	= Test section heat input rate
QATS	= Fluid heat absorption rate in the test section
HBERR	= Heat balance error
QAPH	= Fluid heat absorption rate in the preheater
VTWM, MV	= Test section outside wall thermocouple potentials
TWM, F	= Indicated test section outside wall temperature (starts at the bottom)
RHOER	= Ratio of measured over average computed electrical resistivity
TW, F	= True outside wall temperature
TS, F	= Inside wall temperature
HFLUX	= Wall heat flux
Z	= Distance along the test section starting at the bottom

TB, F	=	t_b
TS, F	=	t_w
HB, B/LB	=	H_b
HS, B/LB	=	H_w
HTC	=	Heat transfer coefficient h
STE	=	Stanton number St_e
DTSB, F	=	$t_w - t_b$
DESB, B/LB	=	$H_w - H_b$
TSA, F	=	$\bar{t}_w$
TBA, F	=	$\bar{t}_b$
HBA, B/LB	=	$\bar{H}_b$
HSA, B/LB	=	$\bar{H}_w$
HTCA	=	$\bar{h}$
STTBA	=	$\bar{St}$
STEA	=	$\bar{St}_e$
Z/DI	=	Tube length to diameter ratio
TB/TP	=	$\frac{t_b}{t_p}$
TS/TP	=	$\frac{t_w}{t_p}$
FB/DT	=	$\frac{t_p - t_b}{t_w - t_b}$
FT/TP	=	$\frac{t_w + t_b}{2t_p}$

HEAT BALANCE COMPUTATIONS

RUN XE 4

VTM,MV	TMI,F	TM,F
13.219	616.21	616.21
13.369	622.46	616.90
13.365	622.29	616.73
13.130	612.27	605.42
14.738	681.17	674.32

FLCW RATE = 1212.8 LBS/HR

FRH1 = 1116.8 LBS/HR

FRH2 = 95.9 LBS/HR

CGTS = 134547. B/HR

GATS = 132901. B/HR

HBERR = 1.2 PERCENT

CAPH = -283. B/HR

PRESSURE = 3300.0 PSIA

VISCOSITY = 0.211 LB/HRFT

9.70	11.00	616.21	616.06	0.02344	0.95

COMPUTATION OF INSIDE SURFACE TEMPERATURE AND LOCAL WALL HEAT FLUX

RHOER = 1.0037

TWM,F	TW,F	TS,F	N	HFLUX,B/HR SQFT
713.00	713.03	612.22	2	474535.
758.59	758.77	658.32	2	481024.
760.67	760.85	660.42	1	481320.
755.18	755.34	654.87	1	480539.
757.27	757.44	656.99	1	480837.
761.21	761.39	660.96	1	481397.
763.21	763.40	662.99	1	481681.
766.00	766.20	665.81	1	482079.
769.29	769.50	669.14	1	482547.
770.00	770.21	669.85	1	482648.
772.91	773.13	672.79	2	483062.
779.00	779.24	678.95	1	483929.
782.75	783.00	682.74	1	484463.
784.67	784.93	684.68	1	484735.
788.75	789.02	688.81	2	485316.
792.05	792.33	692.14	1	485785.
800.00	800.31	700.18	1	486917.
801.59	801.90	701.79	1	487144.
803.64	803.96	703.85	2	487435.
803.36	803.68	703.58	1	487396.
804.18	804.50	704.41	1	487512.
805.08	805.41	705.32	1	487641.
804.95	805.28	705.19	1	487622.
803.23	803.55	703.44	1	487376.
803.18	803.50	703.39	1	487370.
806.50	806.83	706.75	1	487842.
808.75	809.09	709.02	1	488162.
809.67	810.01	709.95	1	488293.
765.13	765.32	664.92	2	481954.

LOCAL TEMPERATURES, ENTHALPIES, HEAT TRANSFER COEFFICIENTS, AND STANTON NUMBERS

FLOW RATE = 1212.8 LBS/HR

PRESSURE = 3300.0 PSIA

MASS FLUX = 3.7165E 06 LBS/HR SQFT

Z, IN	TB, F	TS, F	HB, B/LB	HS, B/LB	HTC, B/HR SQFT	STE
0.	605.42	612.22	616.1	625.4	69805.	0.013613
0.5	606.18	658.32	617.1	695.7	9226.	0.001646
1.0	606.94	660.42	618.2	699.5	9001.	0.001592
1.5	607.71	654.87	619.2	689.6	10188.	0.001836
2.0	609.47	656.99	620.2	693.4	9910.	0.001769
2.5	609.22	660.96	621.3	700.5	9304.	0.001636
4.5	612.25	662.99	625.5	704.1	9494.	0.001649
6.5	615.31	665.81	629.7	709.2	9546.	0.001631
8.5	618.30	669.14	633.9	715.3	9491.	0.001595
10.5	621.28	669.85	638.1	716.6	9936.	0.001654
12.5	624.23	672.79	642.3	722.2	9946.	0.001626
16.5	630.05	678.95	650.7	735.6	9896.	0.001534
20.5	635.66	682.74	659.2	744.0	10290.	0.001537
24.5	641.17	684.68	667.6	748.5	11139.	0.001613
28.5	646.57	688.81	676.1	758.6	11490.	0.001582
32.5	651.80	692.14	684.5	767.3	12043.	0.001578
36.5	656.72	700.18	693.0	793.6	11203.	0.001303
40.5	661.47	701.79	701.5	800.5	12082.	0.001323
42.5	663.85	703.85	705.7	810.9	12185.	0.001247
44.5	666.16	703.58	710.0	809.3	13027.	0.001320
46.5	669.47	704.41	714.2	814.3	13568.	0.001311
48.5	670.75	705.32	718.5	820.5	14107.	0.001286
49.0	671.32	705.19	719.5	819.6	14396.	0.001312
49.5	671.85	703.44	720.6	808.7	15431.	0.001488
50.0	672.37	703.39	721.6	808.5	15711.	0.001511
50.5	672.89	706.75	722.7	834.2	14409.	0.001177
51.0	673.41	709.02	723.8	905.7	13708.	0.000722
51.5	673.93	709.95	724.8	950.3	13556.	0.000583
52.0	674.43	664.92	725.9	707.7	-50689.	0.007135

LOCAL SURFACE AND BULK TEMPERATURE, AND ENTHALPY DIFFERENCES

Z, IN	TS, F	TB, F	HS, B/LB	HB, B/LB	DTSB, F	DESB, B/LB
0.	612.22	605.42	625.4	616.1	6.80	9.38
0.5	658.32	606.18	695.7	617.1	52.14	78.65
1.0	660.42	606.94	699.5	618.2	53.47	81.35
1.5	654.87	607.71	689.6	619.2	47.17	70.42
2.0	656.99	608.47	693.4	620.2	48.52	73.13
2.5	660.96	609.22	700.5	621.3	51.74	79.19
4.5	662.99	612.25	704.1	625.5	50.74	78.61
6.5	665.81	615.31	709.2	629.7	50.50	79.53
8.5	669.14	618.30	715.3	633.9	50.84	81.40
10.5	669.85	621.28	716.6	638.1	48.58	78.51
12.5	672.79	624.23	722.2	642.3	48.57	79.92
16.5	678.95	630.05	735.6	650.7	48.90	84.87
20.5	682.74	635.66	744.0	659.2	47.08	84.81
24.5	684.68	641.17	748.5	667.6	43.52	80.87
28.5	688.81	646.57	758.6	676.1	42.24	82.54
32.5	692.14	651.80	767.3	684.5	40.34	82.83
36.5	700.18	656.72	793.6	693.0	43.46	100.58
40.5	701.79	661.47	800.5	701.5	40.32	99.06
42.5	703.85	663.85	810.9	705.7	40.00	105.14
44.5	703.58	666.16	809.3	710.0	37.42	99.36
46.5	704.41	668.47	814.3	714.2	35.93	100.04
48.5	705.32	670.75	820.5	718.5	34.57	102.03
49.0	705.19	671.32	819.6	719.5	33.87	100.03
49.5	703.44	671.85	808.7	720.6	31.59	88.10
50.0	703.39	672.37	808.5	721.6	31.02	86.82
50.5	706.75	672.89	834.2	722.7	33.86	111.51
51.0	709.02	673.41	905.7	723.8	35.61	181.88
51.5	709.95	673.93	950.3	724.8	36.02	225.43
52.0	664.92	674.43	707.7	725.9	-9.51	-18.18

LOCAL PARAMETERS VERSUS NORMALIZED DISTANCE AND TEMPERATURE

Z/DI	TB/TP	TS/TP	PB/DT	FT/TP	HTC,B/HR SQ FT	STE
0.	0.8532	0.8628	15.3257	0.8580	69805.	0.013613
2.0	0.8543	0.9277	1.9836	0.8910	9226.	0.001646
4.1	0.8553	0.9307	1.9197	0.8930	9001.	0.001592
6.1	0.8564	0.9229	2.1602	0.8896	10188.	0.001836
8.2	0.8575	0.9259	2.0842	0.8917	9910.	0.001769
10.2	0.8586	0.9315	1.9396	0.8950	9304.	0.001636
18.4	0.8629	0.9344	1.9181	0.8986	9494.	0.001649
26.6	0.8672	0.9384	1.8660	0.9028	9546.	0.001631
34.8	0.8714	0.9431	1.7944	0.9072	9491.	0.001595
42.9	0.8756	0.9441	1.8163	0.9099	9936.	0.001654
51.1	0.8798	0.9483	1.7556	0.9141	9946.	0.001626
67.5	0.8881	0.9570	1.6237	0.9225	9896.	0.001534
93.8	0.8960	0.9624	1.5665	0.9292	10290.	0.001537
100.2	0.9038	0.9652	1.5675	0.9345	11139.	0.001613
116.5	0.9115	0.9711	1.4861	0.9413	11490.	0.001582
132.9	0.9189	0.9758	1.4254	0.9474	12043.	0.001578
149.2	0.9259	0.9872	1.2090	0.9566	11203.	0.001303
165.6	0.9327	0.9895	1.1844	0.9611	12082.	0.001323
173.8	0.9360	0.9925	1.1338	0.9642	12185.	0.001247
181.9	0.9393	0.9921	1.1499	0.9657	13027.	0.001320
190.1	0.9426	0.9933	1.1326	0.9680	13568.	0.001311
198.3	0.9459	0.9946	1.1109	0.9702	14107.	0.001286
200.3	0.9467	0.9944	1.1169	0.9705	14396.	0.001312
202.4	0.9474	0.9920	1.1805	0.9697	15431.	0.001488
204.4	0.9482	0.9919	1.1851	0.9700	15711.	0.001511
206.5	0.9489	0.9966	1.0703	0.9728	14409.	0.001177
208.5	0.9496	0.9999	1.0029	0.9747	13708.	0.000722
210.5	0.9504	1.0012	0.9770	0.9758	13556.	0.000583
212.6	0.9511	0.9377	-3.6478	0.9444	-50689.	0.007135

AVERAGE LOCAL PARAMETERS VERSUS NORMALIZED DISTANCE AND TEMPERATURE

ZA/DI	TBA/TP	TSA/TP	APB/DT	AFT/TP	HTCA,B/HRSQFT	STTBA	STEA
1.0	0.8537	0.8953	3.5221	0.8745	16213.	0.003175	0.002921
3.1	0.8548	0.9292	1.9510	0.8920	9112.	0.001784	0.001618
5.1	0.8559	0.9268	2.0321	0.8913	9557.	0.001872	0.001705
7.2	0.8570	0.9244	2.1213	0.8907	10047.	0.001964	0.001802
9.2	0.8580	0.9287	2.0093	0.8934	9597.	0.001864	0.001700
14.3	0.8607	0.9329	1.9286	0.8968	9398.	0.001826	0.001642
22.5	0.8650	0.9364	1.8918	0.9007	9520.	0.001864	0.001640
30.7	0.8693	0.9407	1.8298	0.9050	9518.	0.001821	0.001613
38.8	0.8736	0.9436	1.8048	0.9086	9708.	0.001854	0.001624
47.0	0.8778	0.9462	1.7856	0.9120	9941.	0.001874	0.001640
59.3	0.8840	0.9527	1.6891	0.9183	9921.	0.001845	0.001579
75.6	0.8921	0.9597	1.5953	0.9259	10089.	0.001806	0.001536
92.0	0.9000	0.9638	1.5666	0.9319	10698.	0.001876	0.001574
108.3	0.9077	0.9681	1.5270	0.9379	11312.	0.001945	0.001597
124.7	0.9152	0.9735	1.4561	0.9443	11760.	0.001957	0.001580
141.0	0.9224	0.9815	1.3128	0.9520	11607.	0.001812	0.001427
157.4	0.9293	0.9884	1.1968	0.9588	11626.	0.001752	0.001313
169.7	0.9344	0.9910	1.1588	0.9627	12133.	0.001830	0.001284
177.8	0.9377	0.9923	1.1412	0.9650	12592.	0.001846	0.001283
186.0	0.9410	0.9927	1.1410	0.9669	13292.	0.001947	0.001316
194.2	0.9443	0.9940	1.1215	0.9691	13832.	0.001993	0.001299
199.3	0.9463	0.9945	1.1134	0.9704	14250.	0.002043	0.001299
201.3	0.9471	0.9932	1.1471	0.9701	14895.	0.002036	0.001394
203.4	0.9478	0.9920	1.1822	0.9699	15569.	0.002045	0.001499
205.4	0.9485	0.9943	1.1247	0.9714	15031.	0.001975	0.001323
207.5	0.9493	0.9983	1.0353	0.9738	14049.	0.001846	0.000895
209.5	0.9500	1.0005	0.9894	0.9753	13631.	0.001791	0.000645
211.6	0.9508	0.9694	2.6344	0.9601	36596.	0.004687	0.001260

HEAT BALANCE COMPUTATIONS

RUN XF 9

VTM,MV	TMI,F	TM,F
14.437	668.21	668.21
14.572	673.88	668.09
14.650	677.27	671.47
14.540	672.50	665.65
15.046	654.42	687.57

FLCW RATE = 1896.7 LBS/HR

FRH1 = 1722.6 LBS/HR

FRH2 = 173.5 LBS/HR

CGTS = 84550. B/HR

CATS = 85045. B/HR

HBERR = -0.6 PERCENT

QAPH = 10891. B/HR

PRESSURE = 3400.0 PSIA

VISCOITY = 0.186 LB/HRFT

11.70 4.50 668.21 707.72 0.02643 1.50

COMPUTATION OF INSIDE SURFACE TEMPERATURE AND LOCAL WALL HEAT FLUX

RHCEP = 1.0033

TWM, F	TW, F	TS, F	N	HFLUX, B/HR SQFT
723.92	723.98	683.02	3	186831.
727.29	727.36	686.42	1	187031.
727.29	727.36	686.42	1	187031.
726.79	726.86	685.91	1	187001.
723.67	723.73	682.76	1	186816.
723.37	723.43	682.47	1	186798.
724.37	724.44	683.48	1	186858.
724.17	724.23	683.27	1	186845.
725.08	725.15	684.19	1	186900.
726.04	726.11	685.16	1	186957.
727.87	727.95	687.00	1	187066.
729.17	729.25	688.31	1	187142.
727.67	727.74	686.79	1	187053.
730.96	731.04	690.11	1	187249.
734.25	734.35	693.43	1	187443.
737.23	737.33	696.43	1	187619.
738.13	738.29	697.39	1	187675.
739.00	739.11	698.21	1	187723.
739.32	739.43	699.04	1	187772.
740.79	740.91	700.02	1	187829.
740.33	740.95	700.06	1	187831.
741.00	741.12	700.23	1	187841.
740.94	741.08	700.18	1	187839.
742.27	742.41	701.53	1	187917.
743.75	743.88	703.00	1	188003.
744.17	744.29	703.42	1	188027.
740.37	740.99	700.10	1	187834.

LOCAL TEMPERATURES, ENTHALPIES, HEAT TRANSFER COEFFICIENTS, AND STANTON NUMBERS

FLOW RATE = 1901.3 LBS/HR PRESSURE = 3400.0 PSIA

MASS FLUX = 2.5491E C6 LBS/HRSQFT

Z, IN	TB, F	TS, F	HB, B/LB	HS, B/LB	HTC, B/HRSQFTF	STE
0.	665.65	683.02	707.7	741.9	10758.	0.002145
0.5	665.87	686.42	708.1	749.7	9104.	0.001765
1.0	666.10	686.42	708.5	749.7	9204.	0.001782
1.5	666.32	685.91	708.9	748.5	9545.	0.001853
2.0	666.54	682.76	709.3	741.3	11517.	0.002287
2.5	666.77	682.47	709.7	740.7	11895.	0.002362
3.0	666.99	683.48	710.1	742.9	11332.	0.002234
3.5	667.21	683.27	710.5	742.4	11632.	0.002293
4.0	667.42	684.19	710.9	744.4	11144.	0.002183
7.0	668.71	685.16	713.2	746.7	11370.	0.002190
10.0	670.05	687.00	715.6	751.1	11036.	0.002065
16.0	672.95	688.31	720.3	754.2	12189.	0.002164
22.0	674.84	686.79	725.0	750.6	15649.	0.002863
28.0	677.38	690.11	729.7	758.6	14703.	0.002548
34.0	679.56	693.43	734.4	766.6	13519.	0.002288
40.0	681.73	696.43	739.2	774.8	12767.	0.002066
46.0	683.90	697.39	743.9	777.7	13914.	0.002176
49.0	684.93	698.21	746.3	780.3	14139.	0.002166
52.0	685.93	699.04	748.6	783.1	14322.	0.002135
52.5	686.09	700.02	749.0	786.6	13487.	0.001963
53.0	686.26	700.06	749.4	786.7	13608.	0.001976
53.5	686.42	700.23	749.8	787.3	13606.	0.001965
54.0	686.59	700.18	750.2	787.2	13813.	0.001994
54.5	686.75	701.53	750.6	792.0	12717.	0.001781
55.0	686.92	703.00	751.0	797.5	11691.	0.001585
55.5	687.08	703.42	751.4	799.2	11510.	0.001542
56.0	687.25	700.10	751.8	786.9	14612.	0.002100

LOCAL SURFACE AND BULK TEMPERATURE, AND ENTHALPY DIFFERENCES

Z, IN	TS, F	TB, F	HS, B/LB	HB, B/LB	DTSB, F	DESB, B/LB
0.	683.02	665.65	741.9	707.7	17.37	34.16
0.5	686.42	665.87	749.7	708.1	20.54	41.57
1.0	686.42	666.10	749.7	708.5	20.32	41.18
1.5	685.91	666.32	748.5	708.9	19.59	39.59
2.0	682.76	666.54	741.3	709.3	16.22	32.05
2.5	682.47	666.77	740.7	709.7	15.70	31.02
3.0	683.48	666.99	742.9	710.1	16.49	32.82
3.5	683.27	667.21	742.4	710.5	16.06	31.97
4.0	684.19	667.42	744.4	710.9	16.77	33.59
7.0	685.16	668.71	746.7	713.2	16.44	33.48
10.0	687.00	670.05	751.1	715.6	16.95	35.53
16.0	688.31	672.95	754.2	720.3	15.35	33.93
22.0	686.79	674.84	750.6	725.0	11.95	25.63
28.0	690.11	677.38	758.6	729.7	12.74	28.83
34.0	693.42	679.56	766.6	734.4	13.86	32.14
40.0	696.42	681.73	774.8	739.2	14.70	35.62
46.0	697.39	683.90	777.7	743.9	13.49	33.83
49.0	698.21	684.93	780.3	746.3	13.28	34.01
52.0	699.04	685.93	783.1	748.6	13.11	34.50
52.5	700.02	686.09	786.6	749.0	13.93	37.54
53.0	700.06	686.26	786.7	749.4	13.80	37.30
53.5	700.23	686.42	787.3	749.8	13.81	37.51
54.0	700.18	686.59	787.2	750.2	13.60	36.96
54.5	701.53	686.75	792.0	750.6	14.78	41.40
55.0	702.00	686.92	797.5	751.0	16.08	46.53
55.5	703.42	687.08	799.2	751.4	16.34	47.83
56.0	700.10	687.25	786.9	751.8	12.85	35.09

LOCAL PARAMETERS VERSUS NORMALIZED DISTANCE AND TEMPERATURE

Z/DI	TB/TP	TS/TP	PB/DT	FT/TP	HTC,B/HR SQ FT	STE
0.	0.9322	0.9565	2.7882	0.9443	10758.	0.002145
1.4	0.9325	0.9613	2.3461	0.9469	9104.	0.001765
2.7	0.9328	0.9613	2.3608	0.9470	9204.	0.001782
4.1	0.9331	0.9606	2.4370	0.9469	9545.	0.001853
5.4	0.9335	0.9562	2.9296	0.9448	11517.	0.002287
6.8	0.9338	0.9558	3.0118	0.9448	11895.	0.002362
8.1	0.9341	0.9572	2.8547	0.9456	11332.	0.002234
9.5	0.9344	0.9569	2.9168	0.9456	11632.	0.002293
10.8	0.9347	0.9582	2.7807	0.9464	11144.	0.002183
18.9	0.9365	0.9595	2.7569	0.9480	11370.	0.002190
27.0	0.9384	0.9621	2.5947	0.9503	11036.	0.002065
43.3	0.9425	0.9640	2.6744	0.9532	12189.	0.002164
59.5	0.9452	0.9619	3.2755	0.9535	15649.	0.002863
75.7	0.9487	0.9666	2.8736	0.9577	14703.	0.002548
91.9	0.9518	0.9713	2.4802	0.9615	13519.	0.002288
108.2	0.9549	0.9755	2.1909	0.9652	12767.	0.002066
124.4	0.9580	0.9769	2.2247	0.9674	13914.	0.002176
137.5	0.9594	0.9780	2.1813	0.9687	14139.	0.002166
140.6	0.9608	0.9792	2.1326	0.9700	14322.	0.002135
142.0	0.9611	0.9806	1.9957	0.9708	13487.	0.001963
143.3	0.9612	0.9806	2.0014	0.9710	13608.	0.001976
144.7	0.9615	0.9809	1.9890	0.9712	13606.	0.001965
146.0	0.9618	0.9808	2.0070	0.9713	13813.	0.001994
147.4	0.9620	0.9827	1.8357	0.9724	12717.	0.001781
148.7	0.9622	0.9848	1.6764	0.9735	11691.	0.001585
150.1	0.9625	0.9854	1.6401	0.9739	11510.	0.001542
151.4	0.9627	0.9807	2.0712	0.9717	14612.	0.002100

LOCAL AVERAGE TEMPERATURES, ENTHALPIES, HEAT TRANSFER COEFFICIENTS, AND STANTON NUMBERS

TSA,F	TBA,F	HSA,B/LB	HBA,B/LB	HTCA,B/HRSQFTF	STTBA	STE A
684.72	665.76	745.78	707.92	9862.	0.002203	0.001936
686.42	665.99	749.68	708.31	9154.	0.002044	0.001773
686.16	666.21	749.09	708.70	9371.	0.002093	0.001817
684.34	666.43	744.91	709.09	10438.	0.002331	0.002047
682.62	666.66	741.02	709.49	11703.	0.002614	0.002324
682.97	666.88	741.80	709.88	11607.	0.002575	0.002296
683.37	667.10	742.67	710.27	11480.	0.002473	0.002263
683.73	667.31	743.44	710.66	11383.	0.002452	0.002236
684.07	668.07	745.57	712.04	11256.	0.002425	0.002187
686.08	669.38	748.90	714.39	11200.	0.002498	0.002126
687.65	671.50	752.66	717.93	11584.	0.002792	0.002113
687.55	673.90	752.43	722.65	13703.	0.002155	0.002465
688.45	676.11	754.59	727.36	15161.	0.003195	0.002696
691.77	678.47	762.57	732.08	14086.	0.002558	0.002411
694.93	680.65	770.69	736.81	13132.	0.002363	0.002171
696.91	682.81	776.26	741.54	13316.	0.002397	0.002120
697.80	684.42	779.00	745.08	14026.	0.002407	0.002171
698.62	685.43	781.70	747.45	14230.	0.002338	0.002150
699.53	686.01	784.85	748.83	13892.	0.002282	0.002045
700.04	686.17	786.64	749.22	13547.	0.002226	0.001969
700.14	686.34	787.02	749.62	13607.	0.002235	0.001970
700.21	686.50	787.25	750.01	13709.	0.002250	0.001979
700.86	686.67	789.59	750.41	13242.	0.002171	0.001881
702.26	686.83	794.77	750.80	12182.	0.001997	0.001677
703.21	687.00	798.38	751.20	11600.	0.001902	0.001563
701.76	687.16	793.05	751.59	12876.	0.002111	0.001778

AVERAGE LOCAL PARAMETERS VERSUS NORMALIZED DISTANCE AND TEMPERATURE

ZA/CI	TBA/TP	TSA/TP	APB/DT	AFT/TP	HTCA,B/HR SQFT	STTBA	STE A
0.7	0.9324	0.9589	2.5481	0.9456	9862.	0.002203	0.001936
2.0	0.9327	0.9613	2.3529	0.9470	9154.	0.002044	0.001773
3.4	0.9330	0.9609	2.3977	0.9470	9371.	0.002093	0.001817
4.7	0.9333	0.9584	2.6595	0.9458	10438.	0.002331	0.002047
6.1	0.9336	0.9560	2.9694	0.9448	11703.	0.002614	0.002324
7.4	0.9339	0.9565	2.9307	0.9452	11607.	0.002575	0.002296
8.8	0.9342	0.9570	2.8847	0.9456	11480.	0.002473	0.002263
10.1	0.9345	0.9575	2.8467	0.9460	11383.	0.002452	0.002236
14.9	0.9356	0.9589	2.7683	0.9472	11256.	0.002425	0.002187
23.0	0.9375	0.9609	2.6740	0.9492	11200.	0.002498	0.002126
35.2	0.9405	0.9631	2.6320	0.9518	11584.	0.002792	0.002113
51.4	0.9438	0.9630	2.9368	0.9534	13703.	0.002155	0.002465
67.6	0.9470	0.9643	3.0674	0.9556	15161.	0.003195	0.002696
83.8	0.9503	0.9689	2.6678	0.9596	14086.	0.002558	0.002411
100.1	0.9534	0.9734	2.3307	0.9634	13132.	0.002363	0.002171
116.3	0.9564	0.9762	2.2064	0.9663	13315.	0.002397	0.002120
128.4	0.9587	0.9775	2.2024	0.9681	14025.	0.002407	0.002171
136.6	0.9601	0.9786	2.1563	0.9694	14230.	0.002338	0.002150
141.3	0.9610	0.9799	2.0613	0.9704	13892.	0.002282	0.002045
142.6	0.9612	0.9806	1.9978	0.9709	13547.	0.002226	0.001969
144.0	0.9614	0.9808	1.9945	0.9711	13607.	0.002235	0.001970
145.3	0.9617	0.9809	1.9971	0.9713	13709.	0.002250	0.001979
146.7	0.9619	0.9818	1.9170	0.9718	13242.	0.002171	0.001881
148.1	0.9621	0.9837	1.7520	0.9729	12182.	0.001997	0.001677
149.4	0.9624	0.9851	1.6575	0.9737	11600.	0.001902	0.001563
150.8	0.9626	0.9830	1.8292	0.9728	12876.	0.002111	0.001778

HEAT BALANCE COMPUTATIONS

RUN XE11

VTM,MV	IMI,F	IM,F
14.694	679.27	679.27
14.863	686.50	680.66
15.070	695.42	689.53
14.879	687.23	680.38
15.527	715.29	708.44

FLCW RATE = 933.1 LBS/HR

FRF1 = 846.8 LBS/HR

FRF2 = 86.3 LBS/HR

CGTS = 144794. B/HR

GATS = 81853. B/HR

HBERR = 43.5 PERCENT

QAPH = 18988. B/HR

PRESSURE = 3400.0 PSIA

VISCOSITY = 0.177 LB/HRFT

6.59	4.50	679.27	736.15	0.02771	0.90

MILLIVOLT-TEMPERATURE CONVERSION FOR THERMOCOUPLES
 TYPE CHROMEL ALUMEL COUPLES

RUN XE11

VTWM, MV	TWM, F
16.976	776.92
18.287	832.79
18.334	834.75
18.263	831.79
18.193	828.88
18.272	832.17
18.316	834.00
18.345	835.21
18.374	836.42
18.389	837.04
18.400	837.50
18.499	841.62
18.515	842.29
18.494	841.42
18.513	842.21
18.516	842.33
18.614	846.42
18.575	844.79
18.610	846.25
18.584	845.17
18.632	847.17
18.653	848.04
18.645	847.71
18.600	845.83
18.613	846.38
18.720	850.91
18.761	852.77
18.790	854.09
17.393	794.68

COMPUTATION OF INSIDE SURFACE TEMPERATURE AND LOCAL WALL HEAT FLUX

RHOER = 0.9987

TWM,F	TW,F	TS,F	N	HFLUX,B/HR SQFT
776.92	777.15	670.41	3	512107.
832.79	833.21	726.91	2	520536.
834.75	835.17	728.89	1	520831.
831.79	832.20	725.90	1	520385.
828.88	829.28	722.95	1	519945.
832.17	832.58	726.28	1	520442.
834.00	834.42	728.13	1	520718.
835.21	835.63	729.35	1	520900.
836.42	836.84	730.58	1	521083.
837.04	837.47	731.21	1	521177.
837.50	837.93	731.67	1	521246.
841.62	842.07	735.84	1	521868.
842.29	842.74	736.52	1	521969.
841.42	841.86	735.63	1	521837.
842.21	842.65	736.43	1	521957.
842.33	842.78	736.56	1	521975.
846.42	846.88	740.69	1	522591.
844.79	845.25	739.05	1	522346.
846.25	846.71	740.52	1	522566.
845.17	845.62	739.42	1	522403.
847.17	847.63	741.44	2	522704.
848.04	848.51	742.33	1	522836.
847.71	848.17	741.99	1	522786.
845.83	846.29	740.10	1	522503.
846.38	846.83	740.64	1	522585.
850.91	851.38	745.23	1	523269.
852.77	853.25	747.11	1	523550.
854.09	854.58	748.45	1	523749.
794.68	794.97	688.38	2	514787.

LOCAL TEMPERATURES, ENTHALPIES, HEAT TRANSFER COEFFICIENTS, AND STANTON NUMBERS

FLOW RATE = 933.1 LBS/HR PRESSURE = 3400.0 PSIA

MASS FLUX = 2.8594E 06 LBS/HRSQFT

Z, IN	TB, F	TS, F	HB, B/LB	HIS, B/LB	HTC, B/HRSQFT	STE
0.	680.38	670.41	736.1	716.1	-51380.	0.008946
0.5	681.05	726.91	737.6	1065.3	11350.	0.000556
1.0	681.73	728.89	739.1	1074.8	11043.	0.000543
1.5	682.40	725.90	740.6	1060.3	11964.	0.000569
2.0	683.08	722.95	742.0	1043.7	13042.	0.000603
2.5	683.76	726.28	743.5	1062.2	12240.	0.000571
4.5	686.29	728.13	749.4	1071.4	12445.	0.000566
6.5	688.77	729.35	755.3	1077.0	12834.	0.000566
8.5	691.24	730.58	761.2	1082.4	13247.	0.000567
10.5	693.65	731.21	767.1	1085.0	13878.	0.000573
12.5	695.83	731.67	773.0	1087.0	14544.	0.000581
16.5	699.57	735.84	784.8	1102.8	14387.	0.000574
20.5	702.82	736.52	796.6	1105.2	15492.	0.000592
24.5	705.70	735.63	808.4	1102.2	17432.	0.000621
28.5	707.98	736.43	820.3	1105.1	18345.	0.000641
32.5	709.56	736.56	832.1	1105.5	19331.	0.000668
36.5	710.67	740.69	843.9	1118.2	17411.	0.000666
40.5	711.54	739.05	855.8	1113.5	18991.	0.000709
42.5	711.93	740.52	861.7	1117.8	18276.	0.000713
44.5	712.27	739.42	867.6	1114.7	19237.	0.000739
46.5	712.61	741.44	873.5	1120.5	18128.	0.000740
48.5	712.91	742.33	879.4	1123.0	17771.	0.000751
49.0	712.97	741.99	880.9	1122.0	18014.	0.000758
49.5	713.03	740.10	882.4	1116.7	19303.	0.000780
50.0	713.08	740.64	883.9	1118.3	18961.	0.000780
50.5	713.14	745.23	885.3	1130.9	16305.	0.000745
51.0	713.18	747.11	886.8	1136.0	15428.	0.000735
51.5	713.22	748.45	888.3	1139.7	14868.	0.000729
52.0	713.26	688.38	889.8	754.5	-20689.	0.001331

LOCAL SURFACE AND BULK TEMPERATURE, AND ENTHALPY DIFFERENCES

Z, IN	TS, F	TB, F	HS, B/LB	HB, B/LB	DTSB, F	DESB, B/LB
0.	670.41	680.38	716.1	736.1	-9.97	-20.02
0.5	726.91	681.05	1065.3	737.6	45.86	327.70
1.0	728.89	681.73	1074.8	739.1	47.16	335.67
1.5	725.90	682.40	1060.3	740.6	43.49	319.72
2.0	722.95	683.08	1043.7	742.0	39.87	301.69
2.5	726.28	683.76	1062.2	743.5	42.52	318.70
4.5	728.13	686.29	1071.4	749.4	41.84	321.99
6.5	729.35	688.77	1077.0	755.3	40.59	321.68
8.5	730.58	691.24	1082.4	761.2	39.34	321.18
10.5	731.21	693.65	1085.0	767.1	37.55	317.95
12.5	731.67	695.83	1087.0	773.0	35.84	314.01
16.5	735.84	699.57	1102.8	784.8	36.27	318.03
20.5	736.52	702.82	1105.2	796.6	33.69	308.59
24.5	735.63	705.70	1102.2	808.4	29.94	293.79
28.5	736.43	707.98	1105.1	820.3	28.45	284.79
32.5	736.56	709.56	1105.5	832.1	27.00	273.46
36.5	740.69	710.67	1118.2	843.9	30.02	274.28
40.5	739.05	711.54	1113.5	855.8	27.51	257.71
42.5	740.52	711.93	1117.8	861.7	28.59	256.14
44.5	739.42	712.27	1114.7	867.6	27.16	247.08
46.5	741.44	712.61	1120.5	873.5	28.83	246.99
48.5	742.33	712.91	1123.0	879.4	29.42	243.55
49.0	741.99	712.97	1122.0	880.9	29.02	241.15
49.5	740.10	713.03	1116.7	882.4	27.07	234.32
50.0	740.64	713.08	1118.3	883.9	27.56	234.40
50.5	745.23	713.14	1130.9	885.3	32.09	245.57
51.0	747.11	713.18	1136.0	886.8	33.94	249.19
51.5	748.45	713.22	1139.7	888.3	35.23	251.41
52.0	688.38	713.26	754.5	889.8	-24.88	-135.28

LOCAL PARAMETERS VERSUS NORMALIZED DISTANCE AND TEMPERATURE

Z/DI	TB/TP	TS/TP	PB/DT	FT/TP	HTC,B/HRSQFTF	STE
0.	0.9528	0.9389	-3.3806	0.9458	-51380.	0.008946
2.0	0.9538	1.0180	0.7200	0.9859	11350.	0.000556
4.1	0.9547	1.0208	0.6857	0.9877	11043.	0.000543
6.1	0.9557	1.0166	0.7279	0.9861	11964.	0.000569
8.2	0.9566	1.0124	0.7771	0.9845	13042.	0.000603
10.2	0.9576	1.0171	0.7127	0.9873	12240.	0.000571
18.4	0.9611	1.0197	0.6635	0.9904	12445.	0.000566
26.6	0.9646	1.0214	0.6229	0.9930	12834.	0.000566
34.8	0.9681	1.0232	0.5796	0.9956	13247.	0.000567
42.9	0.9715	1.0241	0.5426	0.9978	13878.	0.000573
51.1	0.9745	1.0247	0.5076	0.9996	14544.	0.000581
67.5	0.9793	1.0306	0.3980	1.0052	14387.	0.000574
83.8	0.9844	1.0315	0.3315	1.0080	15492.	0.000592
100.2	0.9884	1.0303	0.2766	1.0094	17432.	0.000621
116.5	0.9916	1.0315	0.2102	1.0115	18345.	0.000641
132.9	0.9939	1.0317	0.1625	1.0128	19331.	0.000668
149.2	0.9954	1.0375	0.1085	1.0165	17411.	0.000666
165.6	0.9967	1.0352	0.0863	1.0159	18991.	0.000709
173.8	0.9972	1.0373	0.0693	1.0173	18276.	0.000713
181.9	0.9977	1.0358	0.0601	1.0167	19237.	0.000739
190.1	0.9982	1.0386	0.0445	1.0184	18128.	0.000740
198.3	0.9986	1.0398	0.0332	1.0192	17771.	0.000751
200.3	0.9987	1.0394	0.0314	1.0190	19014.	0.000758
202.4	0.9988	1.0367	0.0316	1.0178	19303.	0.000780
204.4	0.9989	1.0375	0.0289	1.0182	19961.	0.000780
206.5	0.9990	1.0439	0.0231	1.0214	16305.	0.000745
208.5	0.9990	1.0466	0.0206	1.0228	15428.	0.000735
210.5	0.9991	1.0484	0.0186	1.0238	14868.	0.000729
212.6	0.9991	0.9643	-0.0246	0.9817	-20689.	0.001331

LOCAL AVERAGE TEMPERATURES, ENTHALPIES, HEAT TRANSFER COEFFICIENTS, AND STANTON NUMBERS

TSA, F	TBA, F	HSA, B/LB	HBA, B/LB	HTCA, B/HR SQ FT	STTBA	STE A
698.66	680.71	890.72	736.88	28769.	0.004625	0.001174
727.90	681.39	1070.03	738.34	11194.	0.001800	0.000549
727.39	682.07	1067.52	739.82	11485.	0.001846	0.000556
724.42	682.74	1052.00	741.29	12479.	0.002006	0.000585
724.61	683.42	1052.96	742.76	12628.	0.002030	0.000586
727.21	685.02	1066.79	746.45	12342.	0.001855	0.000568
728.74	687.53	1074.18	752.34	12637.	0.001854	0.000566
729.97	690.00	1079.67	758.24	13037.	0.001913	0.000567
730.89	692.45	1083.71	764.14	13555.	0.001938	0.000570
731.44	694.74	1086.02	770.05	14203.	0.001835	0.000577
733.76	697.70	1094.92	778.90	14465.	0.001600	0.000577
736.19	701.20	1104.02	790.72	14919.	0.001436	0.000583
736.07	704.26	1103.73	802.54	16404.	0.001394	0.000606
736.03	706.84	1103.65	814.36	17877.	0.001209	0.000631
736.50	708.77	1105.30	826.18	18825.	0.000878	0.000654
738.62	710.11	1111.88	838.00	18320.	0.000604	0.000667
739.87	711.11	1115.83	849.84	18166.	0.000466	0.000687
739.79	711.73	1115.63	858.71	18626.	0.000425	0.000711
739.97	712.10	1116.23	864.63	18744.	0.000379	0.000726
740.43	712.44	1117.58	870.54	18666.	0.000377	0.000740
741.89	712.76	1121.73	876.46	17948.	0.000316	0.000745
742.16	712.94	1122.51	880.16	17892.	0.000262	0.000754
741.04	713.00	1119.37	881.64	18636.	0.000248	0.000769
740.37	713.05	1117.48	883.12	19130.	0.000254	0.000780
742.94	713.11	1124.59	884.60	17532.	0.000219	0.000762
746.17	713.16	1133.46	886.08	15854.	0.000155	0.000740
747.78	713.20	1137.86	887.56	15143.	0.000148	0.000732
718.41	713.24	947.10	889.04	100390.	0.000980	0.003128

AVERAGE LOCAL PARAMETERS VERSUS NORMALIZED DISTANCE AND TEMPERATURE

ZA/DI	TBA/TP	TSA/TP	APB/DT	AFT/TP	HTCA, B/HR SQ FT	STTBA	STE A
1.0	0.9533	0.9784	1.8581	0.9659	28769.	0.004625	0.001174
3.1	0.9542	1.0194	0.7024	0.9868	11194.	0.001800	0.000549
5.1	0.9552	1.0187	0.7058	0.9869	11485.	0.001846	0.000556
7.2	0.9561	1.0145	0.7512	0.9853	12479.	0.002006	0.000585
9.2	0.9571	1.0148	0.7436	0.9859	12628.	0.002030	0.000586
14.3	0.9594	1.0184	0.6881	0.9889	12342.	0.001855	0.000568
22.5	0.9629	1.0206	0.6433	0.9917	12637.	0.001854	0.000566
30.7	0.9663	1.0223	0.6013	0.9943	13037.	0.001913	0.000567
38.8	0.9698	1.0236	0.5613	0.9967	13555.	0.001938	0.000570
47.0	0.9730	1.0244	0.5252	0.9987	14203.	0.001835	0.000577
59.3	0.9772	1.0277	0.4522	1.0024	14465.	0.001600	0.000577
75.6	0.9821	1.0311	0.3657	1.0066	14919.	0.001436	0.000583
92.0	0.9864	1.0310	0.3054	1.0087	16404.	0.001394	0.000606
108.3	0.9900	1.0309	0.2439	1.0105	17877.	0.001209	0.000631
124.7	0.9928	1.0316	0.1866	1.0122	18825.	0.000878	0.000654
141.0	0.9947	1.0346	0.1338	1.0146	18320.	0.000604	0.000667
157.4	0.9961	1.0364	0.0976	1.0162	18166.	0.000466	0.000687
169.7	0.9970	1.0363	0.0773	1.0166	18626.	0.000425	0.000711
177.8	0.9975	1.0365	0.0644	1.0170	18744.	0.000379	0.000726
186.0	0.9980	1.0372	0.0517	1.0176	18666.	0.000377	0.000740
194.2	0.9984	1.0392	0.0384	1.0188	17948.	0.000316	0.000745
199.3	0.9987	1.0396	0.0320	1.0192	17892.	0.000262	0.000754
201.3	0.9988	1.0381	0.0311	1.0184	18636.	0.000248	0.000769
203.4	0.9989	1.0371	0.0298	1.0180	19130.	0.000254	0.000780
205.4	0.9989	1.0407	0.0254	1.0198	17532.	0.000219	0.000762
207.5	0.9990	1.0453	0.0215	1.0221	15854.	0.000155	0.000740
209.5	0.9991	1.0475	0.0193	1.0233	15143.	0.000148	0.000732
211.6	0.9991	1.0064	0.1204	1.0028	100390.	0.000980	0.003128

APPENDIX I

EXPERIMENTAL DATA

Nine sets of experimental data are presented in the following.
The data appears in the following sequence.

1. Run number
2. 01 = Venturi pressure drop in inches of mercury
00 = Venturi pressure drop in psi
3. Test section outside wall thermocouple potentials as indicated. 29 for XE and 27 for XF
4. Test section current and voltage
5. Large venturi pressure drop, test section pressure drop in psi, venturi wall temperature, test section inlet enthalpy, fluid specific volume at the venturi, fluid viscosity at the venturi, test section inlet pressure, and small venturi pressure drop.

XE 1

01 03

16.660	17.489	17.410	17.446	17.465	17.245	17.897
17.680	17.673	17.694	17.693	17.767	17.814	17.794
17.830	17.859	17.870	17.980	17.963	17.939	17.865
17.998	17.988	17.909	17.872	17.960	17.900	17.920
16.988	1128.00	36.07				
8.44000	9.00000	653.54848	679.22292	0.02560	0.192253300	0.00000
						1.01

XE 14

01 03

16.476	16.549	16.556	16.474	16.520	16.604	16.657
16.724	16.803	16.800	16.884	17.086	17.116	17.162
17.260	17.335	17.320	17.350	17.600	17.594	17.612
17.672	17.638	17.591	17.590	17.666	17.720	17.742
16.703	1123.20	32.1				
9.70000	11.00000	616.20833	616.06166	0.02344	0.210763300	0.00000
						0.95

XE 21

01 03

16.800	17.178	17.164	17.160	17.103	17.224	17.266
17.298	17.330	17.353	17.376	17.459	17.490	17.494
17.618	17.621	17.630	17.614	17.640	17.618	17.630
17.688	17.607	17.581	17.585	17.678	17.719	17.738
16.810	1155.40	33.0				
8.81000	9.00000	662.16888	664.67240	0.02530	0.190693400	0.00000
						1.06

XE 27

01 03

16.800	17.370	17.066	17.049	16.970	17.089	17.080
17.047	17.031	17.043	17.048	17.092	17.104	17.111
17.102	17.107	17.125	17.127	17.164	17.131	17.146
17.150	17.145	17.030	17.098	17.158	17.258	17.240
16.710	970.80	30.328				
7.40000	8.50000	676.13635	730.99605	0.02791	0.176634220	0.00000
						0.99

XE 31

01 03

16.730	17.723	17.719	17.710	17.690	17.748	17.890
17.728	17.704	17.744	17.761	17.786	17.808	17.831
17.840	17.863	18.000	17.969	18.010	18.058	18.110
18.133	18.089	18.093	18.134	18.258	18.324	18.320
17.848	892.52	31.05				
4.87000	6.00000	700.88182	767.90204	0.03315	0.148883000	0.00000
						0.73

XF 9

00 03

15.734	15.815	15.815	15.803	15.723	15.721	15.745
15.740	15.762	15.785	15.829	15.860	15.824	15.903
15.982	16.046	16.070	16.088	16.106	16.129	16.130
16.134	16.133	16.165	16.200	16.210	16.131	1002.0
24.75						
11.70000	4.50000	668.20833	707.71928	0.02643	0.186223400	0.00000
						1.50

XF 11

00 03

15.770	15.852	15.837	15.839	15.776	15.757	15.762
15.770	15.783	15.796	15.830	15.657	15.845	15.906
15.967	16.041	16.056	16.071	16.092	16.113	16.105
16.105	16.110	16.145	16.182	16.184	16.135	1002.00
24.75						
11.45000	4.50000	674.33334	712.42136	0.02695	0.182403260	0.00000
						1.49

XF 39

01 03

16.828	17.591	17.619	17.692	17.730	17.716	17.800
17.865	17.849	18.035	18.185	18.257	18.324	18.526
18.778	18.949	19.120	19.124	19.201	19.180	19.225
19.270	19.301	19.314	19.270	19.200	18.004	1236.0
31.25						
5.46000	0.	695.79166	760.16664	0.03207	0.153373300	0.00000
						0.

XF 40

01 03

16.990	17.720	17.724	17.783	17.805	17.800	17.945
17.898	17.902	18.025	18.169	18.292	18.409	18.616
18.789	19.011	19.232	19.284	19.350	19.371	19.446
19.470	19.505	19.534	19.690	19.612	18.465	1224.0
31.1						
9.02000	0.25000	708.91666	814.34026	0.03673	0.135283400	0.00000
						0.

APPENDIX J

POSSIBLE UNCERTAINTIES IN THE COMPUTED PARAMETERS

The possible extent of uncertainties in the thermodynamic and transport properties, and in the computation of the inside surface temperature have been already discussed in the previous sections. The object here is to relate the uncertainties in various parameters of interest to the above mentioned basic values. The maximum possible uncertainty in a parameter is obtained by forcing the uncertainties in its various terms to accumulate. This gives an upper limit to the possible uncertainty in a parameter, but it is improbable that all error would always add up.

The flow rate uncertainty was estimated at around one per cent, based on the scatter in the venturi flow coefficient. The error due to the venturi pressure drop measurements should not have exceeded 0.02 inch of mercury which for a pressure drop of 5.0 inches of mercury amounted to only 0.2 per cent. Since usually higher pressure drops were measured, this error was considered insignificant. At near critical temperatures the uncertainty may have increased another two or three per cent due to the steeper specific volume-temperature gradient.

The major source of error in the heat transfer coefficient was the wall-to-fluid temperature difference. It has been already estimated that a 4-5 per cent uncertainty may enter the inside wall tempera-

ture computations. Therefore, for 80 F outside wall-to-inside wall temperature drop, and a reliability of about 0.5 F in the readings of the outside wall thermocouple, a 4.5 F temperature uncertainty may exist for the inside wall temperature. This is the reason why sufficiently large wall to fluid temperature differences must be maintained to alleviate the effect of this uncertainty upon the heat transfer coefficient and the related parameters. Hence, for $t_w - t_b = 20$ F, a 22.5 per cent maximum uncertainty may be expected in the heat transfer coefficient. The heat flux error could not exceed one per cent since the volt-ammeter was guaranteed at one half of one per cent.

The uncertainty for the Stanton number was expected to be around 25 per cent under the above conditions. This assumed a two per cent uncertainty in the enthalpy values. However, in the critical region $St_e = \frac{q_w''}{(H_w - H_b)G}$ may have been subject to higher uncertainties because of the non-linearity of the isobaric enthalpy-temperature curves. The estimation of such an uncertainty becomes very difficult. The same comment holds for other parameters such as the Reynolds, Prandtl, and Nusselt numbers.

Because of the careful calibration of the thermocouples, the uncertainty in the measured temperatures should not have exceeded 0.5 F in general.

The good agreement with the available well established correlations in the appropriate regions indicated that the extent of uncertainties was less than the above predicted values. The heat transfer coefficient for temperatures up to about 650 F was well within ten per cent of the value computed from the Dittus-Boelter equation.

APPENDIX K

A DISCUSSION OF FLUID VIBRATIONS

The object here is to outline the various tests which were performed to determine the origin of the perturbations in the differential pressure measurements across the test section and the venturi meters. Needless to say, efforts also included as their objective the elimination of the fluctuations whether their course was understood or not. The scope of this appendix unfortunately does not permit full description of the many trials which lasted for over four months.

It was thought that the perturbations indicated by the differential pressure transducer could be due to three basic sources:

1. Fluid vibrations.
2. Mechanical vibrations affecting the transducer.
3. Transducer noise and instability caused by (1) or (2) above.

1. Experimental Methods

Preliminary examinations indicated that the perturbations would cease as soon as the system circulation pump was shut down. On restarting the pump the vibrations would start again. When, with the pump off, the pump casing was struck with a wrench or hammer the noise would appear on the transducer recorder. This effect could also be produced by striking any part of the system piping, although at some locations harder blows were necessary for the noise to be seen by the recorder.

The test section was the most susceptible piece. A slight touch or plucking of the test section could produce appreciable indications. For a moment then it seemed that mechanical vibrations were responsible for the perturbations.

It is very difficult to mechanically isolate sections of a system which are normally connected together by heavy welds or large bolted flanges. Special wooden clamps were made to help tie down the test section and the transducer lines at several locations in order to impede their mechanical vibrations. Also a special spring mounted base was installed under the pump and was bolted to the floor. However no progress was made.

During the next phase, for trial purposes low pressure flexible hose was installed in the pressure lines leading to the transducer, completely isolating it from other metal pieces. The hose would allow operations at pressures up to 200 psia. Appreciable reduction in the amplitude of the fluctuations was achieved at 50 psia system pressure. However, as the pressure was raised the amplitude increased too, probably due to the stiffening of the flexible lines. The hose could have also provided damping against fluid vibrations. For further study high pressure flexible rubber hose of 1/4 inch diameter was installed at the test section inlet and outlet, and at the transducer. The test section and the transducer were then completely isolated from the system. Again low pressure operations showed reductions in the amplitude of the vibrations, but no improvements were achieved at higher pressures.

A series of vibration and dynamic tests were then carried out using a vibration pick-up. The object was to determine if it were

possible for the pump vibration to carry through to the test section and the transducer. The pick-up was first installed on the pump to detect its vibrational modes. The known pump frequency together with a beat of four to five cycles per second could be observed. Loading the pump with 250 pounds decreased its effective frequency to one half the motor frequency. The effect of pump coolant flow was to add higher frequencies to the normal pump frequency.

The pick-up was then moved to the test section, the frequency of which was compared to the pump vibration output. It was found that the test section main frequency was around 10 cps. Pump cooling did not affect the test section frequency. When the pick-up was attached to other locations in the system and the framework, a frequency about one half the pump frequency was recorded. It must be mentioned that the pick-up was very sensitive and could detect minute perturbations.

The natural frequency of the transducer was obtained from transient tests clamping the pick-up to the transducer, and striking it. The recorded frequency was around 18.7 cps, and it demonstrated good damping characteristics. Comparison was also made between the oscilloscope traces of the pick-up mounted on the transducer and the transducer record of pressure drop across the test section. The pick-up reading gave the half pump frequency, while the transducer recorded other frequencies.

As a matter of interest and record Figures K-1 and K-2 show some of the findings of the above investigation. In Figure K-2 lower traces in the top two pictures show differential pressure transducer output. Figure K-3 shows the various instruments employed in the dynamic analysis. From left to right Consolidated Electrodynamics

PUMP FREQUENCY = 57.5 CPS

50 MSEC/CM

50 MSEC/CM
PUMP LOADED
250 LBS. WEIGHT

0.2 SEC/CM

SWEEP
TIME

PUMP BEAT FREQUENCY $\approx$ 43 CPS

FIGURE K-1. PUMP FREQUENCY

FIGURE K-2. VELOCITY PICK UP AND TRANSDUCER RECORDS

SWEEP
TIME

0.2 SEC/CM

50 MSEC/CM

50 MSEC/CM

PUMP COOLED

PUMP COOLED

PUMP NOT COOLED

PICK-UP
LOCATION

TEST SECTION

TRANSDUCER

PUMP

LOWER TRACE SHOWS DIFFERENTIAL PRESSURE TRANSDUCER OUTPUT

FIGURE K-3. EQUIPMENT USED IN DYNAMIC ANALYSIS

Corporation's Type 1-117 Vibration Meter, Brüel and Kjoer 2-20,000 cps Random Noise Voltmeter, CEC Velocity pick up, Statham differential pressure transducer, Ellis Bridge Amplifier Meter, Offner Type RS Dynograph, and Tektronix 502 dual-beam oscilloscope. The random noise voltmeter was used to obtain rms values of the pressure drop measurements on the transducer. The output signal of this instrument appeared steady when the input signal was integrated for 100 seconds.

In general, dynamic analysis ruled out the possibility of mechanical vibrations creeping into the transducer output. It was found that the natural frequency of the transducer was not close to the frequencies of the mechanical vibrations to which it was being subjected. Also, a strike on the transducer body did not produce any noise output. However, when lines containing fluid were manipulated, appreciable signals were observed on the transducer output. This furthered the belief that fluid vibrations were the cause of the perturbations. In the following, various tests carried out in conjunction with the problem of fluid vibrations in the system are discussed.

Fluid vibrations could be of three types: 1) Spring-mass type vibration; 2) wave type vibration where a pressure front travels through the system; and 3) vibration due to travelling eddies or vortices. To start with, restrictions consisting of five hypodermic needles were installed in the transducer lines to damp out vibrations. No results beyond slowing the response of the transducer were obtained. These were then augmented by needle valves installed in the lines to make possible variations in the length of the transducer circuit. Finally the transducer was moved and installed very close to the pressure

taps. There were no improvements, and it was concluded that no spring-mass type vibration was possible in the transducer circuit.

To investigate the possibility of eddies being responsible for the perturbations, the entrance of each tube was packed with several one inch long, 0.109 inch I.D. thin wall Monel tubes to act as flow straightners. The XE test section accommodated three and the XF seven tubes. No noticeable gain was made. Either there were no such eddies present or the tubes were not small enough to influence them.

The next series of tests were conducted to determine whether an optimum accumulator size, which would damp out the vibrations, could be determined. The two accumulators available with the system were ineffective at all pressure and temperature levels.

For this purpose an eight-gallon cylinder capable of handling pressures up to 400 psi was mounted directly above the test section where vacuum pump connections were ordinarily made. Pressurization was provided from a commercial nitrogen cylinder. The tests were run at 200 psia starting with the tank full of water. The effect of an accumulator full of water located above the test section and connected to it through a 1/4 inch tube, or a No. 18 hypodermic needle serving as an orifice, was first studied. No improvements were noticeable. The tests were continued by successive bleedings of 1/4 or 1/2 gallon of water from the tank which was replaced by nitrogen. Thus, the tests were ended with the tank (accumulator) fully charged with nitrogen. No significant improvements were accomplished. These tests also ruled out the hypothesis that during the filling process some air cavity was left that worked like a loose spring. Figure K-4

FIGURE K-4. EXPERIMENTAL SET UP FOR ACCUMULATOR TESTS

shows the experimental set-up for the above tests. In Figure K-5 some recorder traces of the transducer output are shown. It is observed that the gas to water ratio in the accumulator did not affect the amplitude of the vibrations to any appreciable extent. The frequency seems to have changed somewhat; however, such records must be examined over a longer period of time one second as is shown in Figure K-5. The mean value of the pressure drop was around 7.8 psi with about 0.6 psi perturbation amplitude. Percentage-wise this amounted to about 7.7 per cent.

The amount of air dissolved in the water was always a source of concern since it could perhaps be the cause of some voids in the fluid, and influence the smooth running of the pump. It was safe to assume that the water being admitted into the system was properly degasified, but the effect of the small amount of air left in the system due to incomplete evacuation could not be determined. In the section on Experimental Apparatus it was pointed out that vacuums better than 29.3 inches of mercury were rarely attained. In conjunction with this it should be said that it is doubtful if other investigations of this nature ever used a more air-free water.

In closing this part it must be mentioned that the test section tube diameter did not affect the perturbations noticeably. While amplitude of the fluctuations could be compared easily, it was difficult to pinpoint the frequency with any accuracy. In the following part a description is given of certain analytical and statistical methods employed concurrently with the experimental work.

FIGURE K-5. SOME RECORDS OF THE ACCUMULATOR EXPERIMENTS

2. Analytical and Statistical Methods

As a first approximation the ideal problem of non-damped vibration of a system consisting of two gas volumes and a leg of water (liquid) was considered. It was hoped that some estimate could be made of the frequency of such a system. The magnitude of the frequency would then help in deciding whether a vibration similar to a spring-mass system was at all possible. In this conjunction consider Figure K-6 which describes the system being considered. The system consists of two gas volumes V_{g1} , V_{g2} and a liquid volume of mass M . Originally a pressure p_0 prevails in the system. If the gas volume V_{g1} is given a volume displacement of V_1 , a displacement V_2 manifests itself in V_{g2} . The displacements are not necessarily equal since the liquid M may itself experience a volume change. A force balance gives:

$$F = A(p_{g1} - p_{g2}) = M \frac{d^2x}{dt^2} \quad K-1$$

where p_{g1} and p_{g2} are the pressures of V_{g1} and V_{g2} after the displacement. It is assumed that the gas volumes do not contribute appreciably to the system mass to be accelerated. The pressures p_{g1} and p_{g2} may be computed from an equation of state. Thus

$$p_{g1} = \frac{C_1 M_{g1} R_1 T_1}{V_{g1} + V_1} \quad K-2$$

$$p_{g2} = \frac{C_2 M_{g2} R_2 T_2}{V_{g2} - V_2} \quad K-3$$

FIGURE K-6. LIQUID MASS ENTRAPPED BETWEEN TWO GAS VOLUMES

However, originally

$$p_o = \frac{C_1 M_{g1} R_1 T_1}{V_{g1}} = \frac{C_2 M_{g2} R_2 T_2}{V_{g2}} \quad K-4$$

Assuming that $C_1 R_1 T_1 = C_2 R_2 T_2 = CRT$, then

$$\frac{M_{g1}}{V_{g1}} = \frac{M_{g2}}{V_{g2}} \quad K-5$$

and

$$\frac{A p_o V_{g1}}{M_{g1}} \left(\frac{M_{g1}}{V_{g1} - V_1} - \frac{M_{g2}}{V_{g2} - V_2} \right) = M \frac{dx^2}{dt^2} \quad K-6$$

Furthermore, on the assumption that $V_1 = V_2 = V$ the following equation is obtained.

$$M \frac{dV^2}{dt^2} = \frac{A^2 p_o V_{g1}}{M_{g1}} \frac{-V(M_{g1} + M_{g2})}{(V_{g1} - V)(V_{g2} - V)} \quad K-7$$

Where $V = AX$. Equation K-7 is non-linear and it is futile to attempt to solve it when none of the terms M_{g1} , M_{g2} , V_{g1} , V_{g2} are known.

Therefore, to give it a more manageable form let it be assumed that $V_{g1} = V_{g2} = V_g$ and $V^2 \ll V_g^2$. Equation K-7 then reduces to:

$$-\frac{dV^2}{dt^2} = \frac{2A^2 p_o}{V_g M} V \quad K-8$$

which has the solution

$$V = B \sin \sqrt{\frac{2A^2 p_o}{MV_g}} \tau \quad K-9$$

Frequency is given by

$$f = \frac{1}{2\pi} \sqrt{\frac{2A^2 p_o}{MV_g}} \quad K-10$$

where the term $\frac{A^2 p_o}{V_g}$ may be looked upon as the system equivalent spring constant. It is interesting to note that equation K-10 shows a pressure dependence for frequency, which has been observed.

For a sample case consider:

$$p_o = 200 \text{ psia}$$

$$M = 83.3 \text{ lb.} \quad 10 \text{ gallons}$$

$$A = 0.442 \text{ in}^2 \quad \text{based on system piping diameter}$$

$$V_g = 1.0 \text{ in}^3 \quad \text{assumed}$$

Then,

$$f = 3.0 \text{ cycles per second.}$$

Such low frequencies were not observed. Another fact that throws doubt into the creditability of equation K-10 is that the frequency is expected to decrease with increasing gas volumes, a point which was repudiated by the experiments. Admittedly such equations are too simple to predict a complex phenomenon.

A statistical method was used to obtain a better estimate of the frequencies that were predominant. Three hundred data points, spaced every 0.01 second, were obtained from a transducer record of

the test section pressure drop fluctuations at 200 psia and room temperature. Unfortunately the spacing of data only permitted evaluation of the frequencies up to 50 cycles per second, thus denying observations close to the circulating pump speed.

The autocovariance and power spectral density functions were evaluated. These functions are basically defined by the following relations:

Autocovariance Function:

$$C(\tau) = \lim_{T \rightarrow \infty} \frac{1}{2T} \int_{-T}^T x(\theta) \cdot x(\theta - \tau) d\theta \quad K-11$$

Power Spectral Density Function:

$$P(\omega) = \frac{1}{\pi} \int_{-\infty}^{+\infty} c(\tau) e^{-i\omega\tau} d\tau \quad K-12$$

or

$$P(\omega) = \frac{2}{\pi} \int_0^{\infty} c(\tau) \cos(\omega\tau) d\tau \quad K-13$$

where T is the time, $x(\theta)$ is the variable, ω is the frequency, and τ is the so called lag value.

Without sophistication, the autocovariance function will give:

1) a measure of randomness, 2) the mean square value of a random function, and 3) an indication of any periodicity in the record. The power spectral density function shows to what extent various frequencies are important. As defined in equation (K-12) it is the Fourier trans-

formation of the autocovariance function. Investigation and computation procedure followed the methods given in reference (41).

The autocovariance function was determined from Tukey's computational form, viz.

$$C(\tau_r) = \frac{1}{n-r} \sum_{i=1}^{n-r} x_i \cdot x_{i+r} \quad K-14$$

where n is the number of data points. The power spectral density function was then computed from

$$P(f_h) = 2\Delta\theta \left[C(0) + 2 \sum_{r=1}^{r=m-1} C(\tau_r) \cos\left(\frac{\pi hr}{m}\right) + C(\tau_m)(-1)^h \right] \quad K-15$$

where $m = \frac{\tau_{\max}}{\Delta\tau}$.

Figure K-7 shows three records illustrating pressure dependence of the frequency. Figure K-8 gives the final result of the statistical analysis for the 200 psia run. The average value of the pressure drop was around 8.0 psi. It should be mentioned that the average value was subtracted from the data which were analyzed. Figure K-8 shows that at 200 psia the predominant frequencies were around 15 cycles per second. The autocovariance function gave no indication of periodicity. A word of caution is in order. Although statistical methods are accurate, **to be significant** they require a large number of data. The **amount** of data used for these analysis were rather modest and per se the results should not be taken for granted.

Some work was carried out considering wave propagation in the system. However, these methods are quite complex and beyond easy

FIGURE K-7. PRESSURE DEPENDENCE OF THE
FREQUENCY

FIGURE K-8. POWER SPECTRAL DENSITY FUNCTION

reach. Time did not permit any further investigations. In general the work has led to the following conclusions:

1. It is quite improbable that mechanical vibrations per se influence the differential pressure transducer readings.
2. The possibility of spring-mass type vibrations could be ruled out.
3. Pressure wave propagation within the system or the presence of eddies and vortices seem to be the likely cause of the perturbations.

The latter point is substantiated by the fact that the amplitude of the fluctuations is larger across the test section than the venturi where the pressure taps are located closely. A wave or eddies traveling within the tube would first affect one pressure tap and some time later the other, the time and thus the differential amplitude being a function of the distance, and wave and/or fluid velocity. Furthermore, it has been observed that the displacement pump action influences the differential amplitude across the test section severely (see Figure K-9), while the venturi record is affected little. Undoubtedly, such pump action could be classed as wave propagation.

There seem to be two possible ways in which pressure waves could be generated: 1) pulses from the circulation pump action, and 2) mechanical vibration of loose components of the system such as long small diameter tubes. The latter has been experimentally proved. A small displacement in such long tubes causes a volume change in nearly incompressible water and therefore a pressure wave. This may also explain the increase in the frequency of the perturbations with in-

FIGURE K-9. RECORDED EFFECT OF THE DISPLACEMENT PUMP ACTION

creasing pressure. At high pressures these components could become taut, and vibrate at higher frequencies. It is doubtful that the small increase in the modulus of elasticity of water with pressure could cause such changes in frequency.

The presence of eddies or vortices was not detected, but could be assumed. These could be shed by the sharp bends in the flow path or by certain boundary layer separation effects. The author has recently learned that the venturi walls were not smooth, but more like steps. This could be a cause, though it must be proved.

By the farthest stretch of imagination, if the frequencies with which vortices are shed in a Kármán vortex street behind a circular cylinder are applied to this problem, it is found that a frequency range of 30-70 cps may be expected. Since, at high Reynolds number

$$S \text{ (Strouhal Number)} = \frac{f D_o}{\bar{U}} = 0.21$$

$$\text{or frequency} = \frac{0.21 \times \bar{U}}{D_o}$$

$$= \frac{0.21}{\frac{0.750}{12}} \times \bar{U} = 3.36 \times \bar{U}$$

The calibration curves for the venturis (Figure 19) show uncommon scatter which probably indicates unsmooth, high turbulence flow. Eddies and Kármán vortices may, therefore, result from the venturis.