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CALIFORNIA INSTITUTE OF TECHNOLOGY
PASADENA, CALIFORNIA

DEPARTMENT OF THE ARMY
HEADQUARTERS UNITED STATES ARMY MATERIEL COMMAND
WASHINGTON, D.C. 20315

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2 February 1967

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INVESTIGATION OF HEAT TRANSFER AND HIGH HEAT FLUX.
Contract W 04 200 ord 455, Jet Propulsion Lab, C I T,
by F. Kreit and M. Summerfield.
JPL progress report 4-95 ATI-63 763

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ORDCIT Project
Contract No. W-04-200-ORD-455
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Progress Report No. 4-95

INVESTIGATION OF HEAT TRANSFER AT HIGH HEAT FLUX: EXPERIMENTAL STUDY OF HEAT TRANSFER AND FRICTION DROP WITH n-BUTYL ALCOHOL IN AN ELECTRICALLY HEATED TUBE

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JET PROPULSION LABORATORY
California Institute of Technology
Pasadena, California

May 17, 1949

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I. INTRODUCTION AND SUMMARY

Heat-transfer and friction coefficients have been obtained experimentally for n-butyl alcohol flowing upward in an electrically heated stainless-steel tube ~~(approx 0.5-in. ID and 17.5-in. long)~~. The data extend over a heat-flux range from 0.2 to 2.6 Btu/sq in. sec in the pressure range from 30 to 250 psia at velocities from 20 to 40 ft/sec.

The results of the experiments indicate that the heat-transfer characteristics of the n-butyl alcohol in the surface boiling regime are similar to those observed with water ~~(Cf. Ref. 1)~~. When the heat-transfer surface temperature exceeds the boiling point of the liquid, the heat flux can be increased, and the velocity as well as the bulk temperature varied, without changing the surface temperature appreciably. The temperature of the surface-to-liquid interface in the surface boiling regime is determined primarily by the pressure on the liquid.

In the pure forced convection regime, the data agree closely with the results obtained from aniline tests (Cf. Ref. 2). In this regime the heat-transfer coefficient* can be predicted by

$$Nu = 0.034\, Re^{0.80}\, Pr^{0.33}\, (\mu_f/\mu_g)^{0.10} \tag{1}$$

Because the vapor pressure curve for n-butyl alcohol closely resembles that of water, and its viscosity is similar to that of aniline, the results presented in this report are in agreement with conclusions drawn by the authors as a result of previous experiments on heat transfer to liquids at high heat fluxes (Cf. Refs. 1 and 2).

In a majority of the tests the effect of heat transfer upon the frictional pressure drop was studied. It was found that the frictional pressure loss decreases (for a given flow rate of coolant) with an increase in heat flux. This decrease continues until boiling begins adjacent to the heat-transfer surface. After surface boiling has begun, the pressure loss increases with any increase in heat flux.

No burnouts (such as had been encountered when water was used as the coolant) occurred in the tests with n-butyl alcohol. However, in several tests at heat fluxes above 2 Btu/sq in. sec, the stainless-steel tubular test sections developed almost invisible longitudinal cracks. These cracks are believed to have been caused by metal fatique (possibly because of vibrations induced by the growth and collapse of bubbles within the test section). An analysis of the pressure fluctuation in the test section has shown that vibrations at high heat fluxes occurred at a frequency of about 2500 cyc/sec. The magnitude of the fluctuations could not be ascertained because of limitations in the response characteristics of the pressure pickup element.

II. EXPERIMENTAL TECHNIQUE

The over-all setup which was used in the experiments has been described in detail in a previous report (Cf. Ref. 1). The test section consists of a stainless-steel tube 0.528 inch ID. A heavy copper flange is silver-soldered to each end of this tube.

*The nomenclature used in this report is given in Table I.

Heating was accomplished by means of electric current which passed longitudinally through the walls of the tube, whose heated length was 17.5 inches. The test section was mounted in a vertical position, and the coolant entered at the lower end of the tube. By means of thermocouples and thermometers immersed in the liquid, the bulk temperature of the liquid was measured as it entered the tube and as it left the tube. The temperature rise of the liquid as it passed through the tube was measured independently by a differential thermocouple. The pressure existing in the tube during a test was determined by means of a Bourdon-type gage. The frictional pressure drop of the liquid flowing through the test section was measured by means of a differential manometer (in addition to the Barton meters described in Ref. 1). The flow rate of the liquid was determined by a double-orifice flowmeter. The total electric power consumed and the voltage and current were measured during each test. The temperature of the outer tube wall was determined by means of thermocouples. The temperature at the inner wall of an electrically heated tube of given configuration can be calculated from the measurements of the voltage drop and of the outer wall temperature. The equations are derived in Reference 1, which also gives a detailed description of this technique.

During tests with *n*-butyl alcohol, it was found that the thermocouple elements were loosened from the outer surface of the test section (to which they had been butt-welded) by severe vibration which was present. This difficulty was eliminated by bending the ends of the thermocouple elements (wires) and welding this bent-over portion (about 1/16 in. long) to the tube wall.

The temperatures recorded by thermocouples of both types (butt-welded and flat-welded) were compared under similar conditions of flow and heat flux, and found to agree within the accuracy of the recording potentiometer.

An attempt was made to determine the frequency and amplitude of pressure fluctuations which were observed in tests at high heat fluxes. For this analysis, two Wiancko pressure pickups were installed as shown schematically in Figure 1. One of the gages measured the liquid pressure in the tube, and the other gage was connected as a differential pressure meter. The Wiancko systems have a linear response up to 500 cyc/sec, and the carrier is able to handle frequencies up to 3000 cyc/sec. The pressures were recorded on the tape of a Miller oscillograph. Inaddition, a sound analyzer was placed in the vicinity of the test section, and the amplitude of the noise level was determined at frequencies up to 5000 cyc/sec.

The technique used in the reduction of the data has been described in detail in Reference 1. The *n*-butyl alcohol did not form a deposit on the heating surface, and the measurements of the wall temperature were reproducible within ±10°F under similar conditions of flow, heat flux, and pressure. The heat balances between the electrical power input and the thermal power output agreed within 2 per cent in the majority of the tests; in no case did the deviation exceed 3 per cent. It is estimated that the accuracy of the experimental heat-transfer coefficients is within ±5 per cent.

The friction coefficients under isothermal conditions agreed within 2 per cent with literature data for smooth tubes (Cf. Refs. 3 and 4). The measurements of frictional pressure loss with heat transfer were reproducible within ±3 per cent for similar conditions of heat flux, bulk temperature, flow rate, and pressure.

III. EXPERIMENTAL RESULTS

The investigation of heat transfer at high heat flux was continued after the completion of the aniline tests (Cf. Ref. 2) with *n*-butyl alcohol as the coolant. This alcohol was selected because its viscosity characteristics are similar to those of aniline, whereas its vapor pressure curve resembles that of water (Cf. Ref. 1). The *n*-butyl alcohol used in these tests was specially purchased for the heat-transfer tests and was 99.5 per cent pure. The results which were obtained from the study of heat transfer and frictional pressure loss to *n*-butyl alcohol are discussed in this section.

A. Heat Transfer With Surface Boiling

A summary of the results of the heat-transfer tests with *n*-butyl alcohol is presented in Figure 2.* The curves show (for an average fluid bulk temperature of 95°F) the temperature potential necessary for the removal of heat fluxes up to 3 Btu/sq in. sec, at pressures of 50, 100, and 200 psia, and for entrance velocities of about 20, 30, and 40 ft/sec. An inspection of the curves in Figure 2 shows that, when the surface temperature exceeds the boiling point of the alcohol, (a) substantial increases in heat flux result in only minor increases in the temperature of the heat transfer-to-liquid surface interface, and (b) the temperature of the heat-transfer surface is insensitive to variations of the coolant velocity in the surface boiling regime.

During one test in the surface boiling regime, the influence of bulk temperature upon the temperature of the heat-transfer surface was investigated (Cf. Fig. 3). The *n*-butyl alcohol was circulated at a constant flow rate and at constant pressure, but without being passed through the heat exchanger. This procedure resulted in the bulk temperature of the liquid increasing from a value of 100°F to a value which was higher than 200°F. The wall-to-liquid interface temperature was unaffected by this increase in the bulk temperature of the liquid and remained constant at 386°F, thus proving that in the surface boiling regime the temperature of the interface between the liquid and the heat-transfer surface is insensitive to changes in bulk temperature.

In Reference 1, the results of the experiments on heat transfer to water, with surface boiling, were correlated by plotting the excess temperature against pressure for constant heat fluxes. The excess temperature was defined as the temperature difference between the wall-to-liquid interface and the boiling point at the pressure of the liquid during the test. The results of the tests with *n*-butyl alcohol are presented in the same form in Figure 4, where the excess temperature is plotted against pressure for various heat fluxes. During these tests, the flow rate of the *n*-butyl alcohol was held constant at 3.85 lb/sec. It can be seen that the excess temperature required to remove a given heat flux at a constant bulk temperature decreases as the pressure increases. A corresponding relationship between excess temperature and pressure exists for water; however, the curves for water have a steeper negative slope than do those for the *n*-butyl alcohol. A further comparison of the results obtained from tests using *n*-butyl alcohol as a coolant with results obtained from tests using water as a coolant is presented in Figure 5. In this Figure the excess temperatures (from Fig. 4) are plotted against heat flux for constant pressures (100 and 25 psia); the results obtained in tests with water at corresponding pressures are superimposed on the same graph. An inspection of Figure 5 shows that the general trends of the curves for the *n*-butyl alcohol agree with trends of the curves for water. However, for the removal of the same heat flux, the *n*-butyl alcohol requires an excess temperature which is about 25°F higher than that required when water is used as the coolant. The test sections used for both liquids were of similar dimensions, and the Reynolds number (evaluated at the bulk temperature) was about 75,000 for both liquids. A bulk temperature of about 100°F was used during the tests with both of the liquids.

The qualitative remarks presented next are pertinent to the proper use and interpretation of the data on heat transfer with surface boiling. Because the bulk

*The basic data from the tests are given in Tables II and III with important calculated results.

temperature was nearly constant during the tests, the degree of subcooling of the bulk of the liquid was not the same in tests at different pressures. Even though the bulk temperature or degree of subcooling has no effect upon the wall temperature in the fully developed surface boiling regime, the point of transition from the pure forced convection regime to the surface boiling regime is dependent upon the bulk temperature and pressure of the liquid. Thus, at a higher bulk temperature, the transition will occur at a lower heat flux, and vice versa. The heat flux at transition can be calculated for a given bulk temperature and pressure by the following equations:

$$(q/A)_{trans.} = 0.034 \frac{K}{D} Re^{0.80} Pr^{0.33} \left(\frac{\mu_f}{\mu_w}\right)^{0.10} (T_S - T_F) \text{ for } n\text{-butyl alcohol} \qquad (2)$$

$$(q/A)_{trans.} = 0.027 \frac{K}{D} Re^{0.80} Pr^{0.33} \left(\frac{\mu_f}{\mu_w}\right)^{0.14} (T_S - T_F) \text{ for water} \qquad (3)$$

Before using the curves applicable only to heat transfer with surface boiling, it is necessary to determine first whether or not the surface temperature (for the pressure, flow conditions, and heat flux under consideration) exceeds the boiling temperature T_S of the coolant.

It has been shown (Cf. Refs. 2 and 5) that the exact point of transition from forced convection heat transfer to surface boiling heat transfer is influenced by the amount of dissolved gases or impurities in the liquid. When a degassed liquid is used as the coolant, the surface temperature may exceed the boiling point by as much as 20°F before bubbles begin to form. On the other hand, if a large amount of gas is dissolved in the coolant, bubble formation may occur at a surface temperature below the saturation temperature. For this reason, the curves of Figure 5 are not extended to excess temperatures of less than 10°F.

In previous tests with water, the heat flux that could be removed by forced convection with surface boiling was limited by burnouts of the tube. It is believed that these burnouts were caused by flow instabilities due to growth and collapse of vapor bubbles. No burnouts (such as had been encountered when water was used as the coolant) occurred in the tests with *n*-butyl alcohol. However, during several tests at heat fluxes above 2 Btu/sq in. sec, the stainless-steel tubular test section developed almost invisible longitudinal cracks. These cracks are believed to have been caused by metal fatigue, possibly because of vibrations induced by the growth and collapse of bubbles within the test section. A detailed discussion of the appearance of these cracks is presented in the appendix.

During the test at high heat fluxes, a loud whining noise was heard. It was believed possible that pressure fluctuations in the heater tube were the source of this noise. In order to analyze these fluctuations, the Wiancko pressure pickups (shown schematically in Fig. 1) were installed and the pressure fluctuations picked up by the Wiancko gages were recorded on the tape of a Miller oscillograph. An analysis of these records has shown that vibrations at high heat flux occurred at a frequency of about 2500 cyc/sec. The magnitude of the pressure fluctuations could not be ascertained because of limitations in the response characteristic of the pressure pickup element. Audibly, the highest noise level existed at about 2 Btu/sq in. sec and decreased with changes in heat flux in either direction. The noise level was measured at various frequencies by a noise primer which was located about 3 feet from the test section. The results obtained with the noise pr r were qualitatively in agreement with the pressure measurements and audible observations.

B. Heat Transfer by Forced Convection Without Surface Boiling

The heat-transfer data for the tests in which the liquid-to-surface temperature remained below the boiling point of the alcohol are shown in Figure 6 and Table III. The data which were obtained in the pure forced convection regime are plotted in Figure 6 as the dimensionless modulus $Nu/(Re^{0.80}\ Pr^{0.33})$ vs the ratio of viscosity at the bulk temperature to viscosity at the temperature of the heat-transfer surface. This type of presentation was chosen because, at the high heat fluxes used in the experiments, a steep temperature gradient exists adjacent to the heat-transfer surface. This temperature gradient causes a sharp decrease in the viscosity of the liquid near the wall, compared with the viscosity of the bulk of the liquid, and it is known from previous investigations (Cf. Refs. 6, 7, and 8) that the variation in viscosity has considerable influence on the heat-transfer process. The experimental data for the n-butyl alcohol may be correlated by Equation (1)

$$Nu = 0.034\ Re^{0.80}\ Pr^{0.33}\ (\mu_f/\mu_w)^{0.10}$$

The data for water and aniline previously obtained (Cf. Refs. 1 and 2) are also shown in Figure 6. It can be seen that the results from the tests with n-butyl alcohol agree closely with those obtained from earlier tests with aniline as the coolant. However, heat-transfer coefficients for both these liquids are from 15 to 25 per cent higher than would be predicted from the conventional Sieder and Tate equation (Cf. Ref. 6), which is

$$Nu = 0.027\ Re^{0.80}\ Pr^{0.33}\ (\mu_f/\mu_w)^{0.14} \qquad (4)$$

Equation (4) was found to be in good agreement with results obtained when water was used as the coolant.

The correlation of the forced convection data for n-butyl alcohol and aniline was subject to uncertainties because no precise data for the physical properties of these liquids were available for the full ranges of temperature and pressure covered in the experiments. In particular, there exists considerable uncertainty regarding the thermal conductivity of these liquids. Therefore, in reducing the data for the aniline and n-butyl alcohol, the thermal conductivities K were assumed to be independent of temperature. The values used are as follows:

For aniline, $K = 0.1$ Btu/sq ft hr °F/ft (Cf. Ref. 9)

For n-butyl alcohol, $K = 0.095$ Btu/sq ft hr °F/ft (Cf. vol 5, p. 228, of Ref. 10)

Values for the vapor pressure, specific heat, density, and viscosity of the n-butyl alcohol were taken from Reference 10 (Cf., respectively, p. 219 of vol 3, p. 108 of vol 5, pp. 27 to 33 of vol 3, and p. 215 of vol 7). The temperature range covered in the experiments exceeded the range for which data on the viscosity of the alcohol were available. The viscosity at higher temperatures was calculated by a technique described by Othmer (Cf. Ref. 11). The physical properties which were actually used in the reduction of data are plotted as a function of temperature in Figures 7, 8, 9, and 10. When more reliable data for the physical properties of aniline and n-butyl alcohol become available, the results of the heat-transfer tests can be re-evaluated. A single coefficient in Equations (1) and (4) may then correlate the results for all liquids.

It is interesting to note (Cf. Fig. 2) that the maximum heat flux that can be removed by pure forced convection with liquid n-butyl alcohol at a velocity of 24 ft/sec is about 1.15 Btu/sq in. sec. (More than twice this heat flux has been removed with surface boiling.) At this heat flux the surface temperature reaches the critical temperature. In general, the thermal conductivity of a fluid in the gaseous state is very much smaller (about one-tenth) than the conductivity of the same fluid in the liquid state. Therefore, at the same mass flow rate of fluid, less heat can be removed at the same temperature potential when the coolant is a fluid in the gaseous state. Hydrogen may be an exception because its thermal conductivity in the gaseous state is higher than the conductivity of most liquids (Cf. p. 391 of Ref. 12). At this time no information is available regarding the variation of physical properties of such fluids at or near the critical point. Therefore it is not safe to extrapolate the curves of Figure 2 to temperatures and pressures higher than critical. However, it is believed that the forced convection equation (Eq. 1) may be applied at pressures and temperatures above critical when correct values for the physical properties are used in the dimensionless moduli.

At the outset of the experiments, it was hoped that the experimental Nusselt moduli could be compared with those calculated from a semitheoretical analogy between heat transfer and momentum transfer by Boelter, Martinelli, and Jonassen (Cf. Ref. 13). These investigators arrived at a semitheoretical equation for the calculation of the Nusselt modulus at a severe temperature gradient. The equation (based on experimental data) assumes that the thickness of the laminar boundary layer decreases under high thermal gradients, and relates this variation in thickness of the laminar layer to the Reynolds number and the ratio of the viscosity at the bulk temperature to the viscosity at the edge of the laminar boundary layer. This comparison has bee postponed because it was believed that the possible error in the Nusselt modulus (due to the uncertainty in the value of the thermal conductivity) would be of such a magnitude that no positive conclusion could be drawn. Also, it is believed that, in order to obtain data for such a comparison, a test section of higher aspect ratio (L/D) should be used so that entrance and exit effects could be eliminated.

C. Frictional Pressure Drop With Heat Transfer

The results of previous experiments have shown that the frictional pressure loss (for a given flow rate of liquid) decreases when heat is transferred to the liquid. Data have been reported for friction coefficients with heat transfer by forced convection without surface boiling (Cf. Refs. 1, 6, and 11). In the surface boiling regime, on the other hand, the frictional pressure drop with heat transfer has not been studied extensively, and the test results which have been obtained do not even agree qualitatively. In order to clear up these uncertainties, the frictional pressure drop of n-butyl alcohol flowing in the stainless-steel heater tube was measured under isothermal and heat-transfer conditions.

The data on frictional pressure loss in isothermal flow are tabulated (Cf. Table IV) and plotted as C_f vs Re (Cf. Fig. 11). The accuracy of the pressure-drop measurements was checked by comparing the results with published data on friction coefficients in smooth tubes (Cf. Refs. 3 and 4). The experimental friction coefficients agreed with the accepted values within 2 per cent.

The influence of heat transfer upon the frictional pressure loss was observed in several series of tests. During each series the heat flux was increased stepwise from 0 to 2.6 Btu/sq in. sec, while the flow rate and liquid pressure were held constant. The heat-flux range spanned the pure forced convection regime and extended well into the regime of surface boiling. The effect of heat transfer upon the frictional pressure drop is illustrated in Figure 12. In this Figure the frictional pressure loss is plotted against heat flux for a constant mass flow rate of 3.85 lb/sec at four different pressures (30, 50, 100, and 200 psia). From an inspection of these curves it can be seen that the frictional pressure loss initially decreases (for a given flow rate of coolant) with an increase in heat flux. This decrease continues until boiling begins adjacent to the heat-transfer surface. After surface boiling has begun, the pressure loss increases with any increase in heat flux.

For given conditions of flow, heat flux, and bulk temperature, the liquid pressure has no influence on the frictional pressure drop in the pure forced convection regime. However, in the surface boiling regime the frictional pressure loss (at the same heat flux, bulk temperature, and flow rate) increases with a decrease in pressure. This result appears reasonable since (at the same flow rate, bulk temperature, and heat flux) boiling is more vigorous at lower pressure. It should be noted that, even at the lowest pressure and highest heat flux used during the tests, the frictional pressure loss did not exceed the isothermal value at the same liquid flow rate.

The discussion thus far in this section applies also to the phenomena which were observed at lower coolant flow rates. The curves of Figures 13 and 14 show the results of tests on the effect of heat flux upon the frictional pressure drop at mass flow rates of 3 and 2.2 lb/sec, respectively.

The results shown in Figures 12, 13, and 14 do not agree with measurements of the pressure loss in flow of coolant water through an annular test section under conditions of heat transfer with surface boiling (Cf. Refs. 14 and 15). In Reference 14 mention is made of increases in pressure loss under conditions of heat transfer with surface boiling as high as sixteen times the maximum value of the isothermal pressure loss at the same flow rate. Knowles (Cf. Ref. 15) reported negative pressure losses* in flow of water through an annulus at high heat fluxes with surface boiling. The reason for the discrepancy between the results obtained at this Laboratory and those reported by other investigators is not known.

In the regime of forced convection heat transfer without surface boiling, the friction coefficient is dependent upon the Reynolds number, the wall temperature, and the Prandtl number

$$C_F = \phi\,(Re,\ T_w,\ Pr) \tag{5}$$

Since the temperature of the liquid adjacent to the wall is above the temperature of the bulk of the liquid, the viscosity adjacent to the wall is less than in the center of the tube. Thus with heat transfer the tractive forces along the wall are decreased; consequently the value of the friction coefficient is lower than it would be under isothermal flow conditions at the same bulk Reynolds number. This fact is shown in Figure 15 where the ratio of isothermal to heat-transfer friction coefficients is plotted as a function of the ratio of viscosity at bulk temperature to viscosity at the wall temperature. The results of previous tests with water (Cf. Ref. 1) are also plotted on the same graph. An inspection of the curves shows that an increase in heat flux (which corresponds to a higher viscosity ratio) results in a decrease of the friction coefficient.

The results of previous work (Cf. Refs. 1, 6, and 16) have indicated that the friction coefficient with heat transfer can be related to the isothermal friction

*In some cases Knowles observed outlet pressures which were higher than the inlet pressures.

coefficient at the same bulk Reynolds number by

$$\frac{C_{F_{iso}}}{C_{F_{HT}}} = \left(\frac{\mu_F}{\mu_W}\right)^{0.14} \tag{6}$$

This relationship was found to be a satisfactory approximation up to viscosity ratios of about 3, regardless of the Reynolds number. However, at higher heat fluxes the ratio of the friction coefficients is larger than would be predicted by Equation (6). Furthermore, at the same viscosity ratio the ratio of the friction coefficients is larger for lower Reynolds numbers.

Attempts to correlate these experimental results by means of theoretical considerations have not been successful. Work in this direction should be continued because any attempts to find an analogy between heat transfer and momentum transfer must use a correct value of the friction coefficient as a stepping stone.

The results of the tests described have been applied to calculations of the frictional pressure loss in annular cooling channels of a rocket motor. Since only one side is heated during firing, the decrease in friction drop due to heat transfer should be only half the value observed in a circular-flow conduit with uniform heating. Based upon this assumption, the frictional pressure drop was calculated (from the experimental data presented in Fig. 15) for several rocket motor tests. The experimental and calculated frictional pressure drops were in agreement.

IV. CONCLUSIONS

Experimental data have been obtained on heat transfer from a stainless-steel tube to *n*-butyl alcohol in the heat-flux range from 0.2 to 2.6 Btu/sq in. sec over a pressure range from 30 to 250 psia for velocities from 20 to 40 ft/sec. The results of these tests (summarized in Fig. 1) permit the determination of the surface temperature necessary for the removal of a given heat flux in the forced convection and boiling regimes.

The temperature of the heat-transfer surface (when it exceeds the boiling temperature of the liquid) is insensitive to changes in velocity, heat flux, and bulk temperature. In the surface boiling regime the pressure of the liquid determines the surface temperature.

The excess temperature required for the removal of a given heat flux decreases with increase in pressure for the range of variables covered. The maximum excess temperature was less than 85°F during all tests.

In the pure forced convection regime the results for *n*-butyl alcohol are correlated within ±5 per cent by the following equation:

$$Nu = 0.034\, Re^{0.80}\, Pr^{0.33}\, (\mu_F/\mu_W)^{0.10}$$

No burnouts of the heater tube were encountered in the tests with *n*-butyl alcohol. However, at high heat fluxes the stainless-steel tube failed because of almost invisible cracks. It is believed that these cracks were caused by fatigue of the metal. Severe vibrations at a frequency of about 2500 cyc/sec were found to exist in tests at heat fluxes above 2 Btu/sq in. sec.

The frictional pressure loss decreases (for a given flow rate of coolant liquid) with an increase in heat flux. After surface boiling has begun, the pressure loss increases with any increase in heat flux. Even in the surface boiling regime the pressure loss with heat transfer was less than the isothermal pressure loss for the same mass flow rate. The friction coefficient for nonboiling heat-transfer conditions can be approximated from isothermal data by evaluating the friction coefficient at a Reynolds number based on a temperature equal to or slightly below the temperature of the heat-transfer surface. It would be desirable to predict the heat-transfer friction coefficient fromtheoretical considerations; work along this line should be encouraged.

TABLE I

NOMENCLATURE

A	area (sq in.).
C_F	friction coefficient $\left[\Delta H \cdot (D/L) \cdot (2\,g/v^2)\right]$.
c_p	specific heat (Btu/lb °F).
D	diameter (in.).
g	gravitational constant (ft/sec^2).
h	heat-transfer coefficient (Btu/sq in. sec °F).
ΔH	frictional head loss (ft of liquid).
K	thermal conductivity (Btu/sq ft hr °F/ft).
L	length (in.).
Nu	Nusselt number $(hD/K) \cdot 43{,}200$.
P	pressure (psia).
ΔP	frictional pressure loss $\left[(\text{in. Hg}) - (\text{in. liquid})\right]$.
Pr	Prandtl number $(c_p\mu/K) \cdot 3600\,g$.
q	heat flux (Btu/sec).
Re	Reynolds number $(vD\rho/\mu g)$.
T	temperature (°F).
ΔT	outlet-inlet temperature (°F).
v	velocity (ft/sec).
w	flow rate (lb/sec).
μ	viscosity (lb sec/sq ft).
ρ	density (lb/cu ft).

Subscripts:

e	entrance.
F	bulk of fluid.
HT	heat transfer.
in	inlet.
iso	isothermal
S	saturation.
W	wall.
X	excess above saturation temperature.

TABLE II

HEAT-TRANSFER AND FRICTIONAL PRESSURE-LOSS DATA FOR TESTS IN SURFACE BOILING REGIME

Test No.	w	T_{in}	ΔT	q/A		P	T_w	T_x	ΔP
				Electrical*	Thermal**				
1	3.84	93	--	1.50	--	68	371	35	
2	3.65	78	22.5	1.59	1.59	200	451	26	
3	3.85	80	16.4	1.17	1.15	50	348	31	
4	3.85	86	23.5	1.66	1.68	50	351	34	
5	3.86	75	23.4	1.63	1.68	200	450	25	
6	3.85	86	--	2.17	--	200	460	35	
7	3.84	89	29.7	2.17	2.13	250	482	38	
8	3.83	103	29.8	2.23	2.22	205	470	43	
9	3.84	91	23.1	1.66	1.67	250	470	26	
10	3.84	98	23.1	1.69	1.69	197	448	24	
11	3.83	103	23.5	1.66	1.74	150	432	33	
12	3.84	90	23.0	1.67	1.66	254	476	29	
13	3.89	95	23.7	1.72	1.73	102	407	42	
14	3.83	99	24.1	1.74	1.77	50	370	53	
15	3.83	101	24.1	1.74	1.77	29	340	56	
16	3.84	92	35.3	2.52	2.58	255	489	42	
17	3.84	96	34.9	2.59	2.57	200	473	48	
18	3.85	81	22.2	1.60	1.58	199	457	33	5.83
19	3.85	86	30.2	2.17	2.19	200	465	41	6.55
20	3.84	92	33.5	2.52	2.45	200	472	48	7.26
21	3.85	87	16.[illegible]	1.16	1.14	100	385	22	6.92
22	3.85	87	24.0	1.69	1.72	99	405	41	7.20
23	3.85	85	31.1	2.23	2.24	99	418	55	7.91
24	3.85	85	11.2	0.79	0.79	50	319	2	7.52
25	3.84	87	16.7	1.19	1.20	50	352	35	7.59
26	3.85	87	24.3	1.72	1.74	50	366	49	8.35
27	3.84	89	31.0	2.29	2.28	50	383	66	9.36
28	3.81	79	11.2	0.82	0.78	30	310	25	7.68
29	3.85	85	16.5	1.17	1.17	30	327	42	8.35
30	2.22	90	14.7	0.62	0.62	50	318	1	2.83

*q/A = (kilowatts)/(1.054 · A_{HT}).

**q/A = (c_p w ΔT)/(A_{HT}).

TABLE II (Cont'd)

Test No.	w	T_{in}	ΔT	q/A Electrical*	q/A Thermal**	P	T_W	T_X	ΔP
31	2.22	91	10.7	0.44	0.45	30	290	5	2.89
32	2.22	90	14.5	0.64	0.63	30	296	11	3.20
33	2.89	94	11.6	0.63	0.65	30	294	9	4.72
34	3.00	90	11.5	0.64	0.63	30	296	11	5.02
35	2.22	87	26.7	1.12	1.13	199	433	9	2.38
36	2.22	90	18.2	0.74	0.77	100	370	6	2.62
37	2.22	89	27.2	1.15	1.16	100	392	28	2.95
38	2.22	86	18.8	0.78	0.79	50	328	11	3.10
39	2.22	90	27.2	1.15	1.18	50	349	32	3.54
40	2.22	88	19.0	0.79	0.80	30	300	15	3.51
41	2.22	86	28.4	1.19	1.19	30	323	38	3.94
42	3.00	89	20.2	1.14	1.14	200	434	9	3.96
43	3.00	88	20.0	1.13	1.16	100	376	12	4.64
44	3.00	90	14.0	0.78	0.78	50	320	3	5.01
45	3.00	86	20.7	1.17	1.18	50	350	33	5.51
46	3.00	89	14.0	0.78	0.79	30	299	14	5.60
47	3.00	88	20.9	1.18	1.19	30	324	39	6.12
48	3.00	90	29.3	1.66	1.64	200	456	31	4.30
49	3.00	92	29.4	1.68	1.70	100	403	39	5.13
50	3.00	98	30.1	1.75	1.70	50	372	55	5.66
51	2.27	95	39.1	1.69	1.70	100	410	46	3.36
52	2.23	102	39.6	1.73	1.74	50	377	60	4.16
53	3.02	91	14.0	0.7[illegible]	0.80	30	321	36	5.40
54	3.84	99	22.2	1.67	1.63	100	404	40	7.59
55	3.84	92	--	1.79	--	100	408	44	7.74
56	3.84	94	--	1.19	--	100	389	25	6.98
57	3.84	97	--	1.72	--	100	[illegible]	48	7.64
58	3.84	93	--	1.22	--	100	384	20	7.00
59	3.84	99	--	1.70	--	100	409	45	7.90
60	3.00	95	--	2.20	--	200	470	45	4.88

*$q/A = (\text{kilowatts})/(1.054 \cdot A_{HT})$.

**$q/A = (c_p \, w \, \Delta T)/(A_{HT})$.

TABLE II (Cont'd)

Test No.	w	T_{in}	ΔT	q/A Electrical*	q/A Thermal**	P	T_W	T_X	ΔP
61	3.00	90	39.0	2.23	2.24	100	420	56	5.78
62	2.22	93	51.1	2.20	2.20	200	473	48	3.27
63	2.22	94	51.4	2.23	2.24	100	417	53	3.95
64	3.00	95	39.3	2.26	2.28	50	384	67	6.82
65	3.82	103	34.2	2.54	2.56	200	--	--	7.56
66	3.84	89	16.3	1.17	1.16	200	--	--	6.62
67	3.84	90	22.6	1.64	1.63	200	--	--	6.42
68	3.84	89	23.1	1.67	1.66	100	--	--	7.51
69	3.84	98	24.0	1.69	1.75	50	--	--	8.85
70	3.84	99	30.6	2.27	2.27	50	--	--	9.92
71	3.82	101	35.2	2.63	2.62	50	398	81	10.50
72	3.84	90	35.1	2.60	2.57	100	429	65	8.84
73	2.22	89	59.4	2.57	2.56	200	477	52	3.65

*$q/A = (\text{kilowatts})/(1.054 \cdot A_{HT})$.

**$q/A = (c_p\, w\, \Delta T)/(A_{HT})$.

TABLE III

HEAT TRANSFER AND FRICTION DATA FOR TESTS WITHOUT SURFACE BOILING

Test No.	w	T_{in}	ΔT	v	q/A Electrical	q/A Thermal	T_W	T_F	h	Re_F x 10^{-3}	Nu	$\frac{\mu_F}{\mu_W}$	$\left(\frac{Nu}{Re^{0.80}Pr^{0.33}}\right)$ x 10^{-2}	ΔP	$C_{F_{HT}}$	$\frac{C_{F_{iso}}}{C_{F_{HT}}}$
1	3.85	86	11.5	43.1	0.82		298	92	3.74	74.1	991	7.33	4.14			
2	3.85	85	11.4	40.9	0.80	0.77	303	88	3.59	66.1	952	8.10	4.30			
3	3.84	87	16.8	41.0	1.19	1.16	393	95	3.88	73.1	1028	11.81	4.39			
4	3.84	90	11.2	41.0	0.79	0.77	301	93	3.71	71.4	983	7.37	4.25			
5	3.84	97	11.3	40.8	0.79	0.79	314	87	3.48	65.8	922	8.66	4.17			
6	3.83	89	17.0	41.2	1.21	1.21	420	100	3.77	77.6	998	12.70	4.21			
7	3.84	97	6.2	41.2	0.44	0.46	225	100	3.66	77.6	970	4.00	4.02			
8	3.84	92	4.7	41.1	0.33	0.33	189	98	3.63	75.7	962	2.97	4.03			
9	2.79	93	6.6	29.9	0.34	0.33	224	98	2.59	55.0	686	4.10	3.71			
10	2.79	93	9.1	29.9	0.46	0.46	265	100	2.77	56.4	737	5.37	3.94			
11	2.79	94	14.9	30.0	0.78	0.79	375	104	2.91	59.4	769	9.50	4.10			
12	3.85	86	11.1	41.1	0.80	0.80	320	94	3.54	72.3	938	7.23	4.03	7.45	0.0119	1.67
13	3.85	87	16.5	41.2	1.18	1.16	405	98	3.78	75.9	1002	12.13	4.20	6.28	0.0100	1.96
14	3.85	88	11.1	41.1	0.79	0.79	320	96	3.47	74.0	920	7.06	3.91	7.58	0.0121	1.63
15	3.84	92	3.1	41.0	0.22	0.21	163	94.5	3.07	72.2	814	2.35	3.51	10.78	0.0173	1.15
16	3.84	89	4.8	41.0	0.34	0.34	192	92.5	3.39	70.6	899	3.28	3.91	10.02	0.0161	1.24
17	3.84	90	6.8	41.0	0.48	0.47	228	93.8	3.52	71.9	933	4.45	4.02	9.22	0.0149	1.34
18	3.84	94	3.0	41.1	0.21	0.21	164	96	3.09	73.9	819	2.27	3.48	10.84	0.0174	1.14
19	3.84	88	4.8	41.0	0.34	0.33	191	91.2	3.35	69.6	888	3.31	3.90	10.11	0.0162	1.23
20	3.84	91	6.6	41.1	0.47	0.46	226	94.8	3.53	72.2	936	4.34	4.03	9.28	0.0149	1.34
21	3.84	89	3.1	41.0	0.22	0.22	157	91.5	3.27	69.8	867	2.25	3.80	10.88	0.0175	1.15
22	3.84	96	4.7	41.1	0.33	0.33	196	99	3.50	76.6	928	3.12	3.87	10.01	0.0160	1.22
23	3.84	92	6.6	41.1	0.47	0.47	228	96.8	3.60	74.5	954	4.30	4.04	9.24	0.0148	1.33
24	3.04	90	3.0	41.0	0.21	0.21	158	91.8	3.24	70.0	859	2.25	3.76	10.84	0.0175	1.14
25	3.84	91	4.7	41.0	0.33	0.33	191	93.5	3.40	71.6	901	3.22	3.89	10.06	0.0161	1.24
26	3.84	93	6.4	41.1	0.46	0.46	225	97.5	3.59	75.4	952	4.12	4.00	9.30	0.0149	1.32
27	2.22	92	5.3	23.7	0.22	0.22	191	97	2.28	43.2	604	3.05	3.95	3.69	0.0177	1.26
28	2.22	92	7.9	23.8	0.33	0.33	245	95	2.19	42.2	580	5.01	3.85	3.22	0.0154	1.47
29	2.22	90	10.7	23.8	0.44	0.45	290	96	2.33	42.8	618	6.67	4.07	2.83	0.0136	1.65
30	2.22	90	14.4	23.8	0.60	0.62	350	100	2.48	44.9	657	9.09	4.14	2.40	0.0115	1.96

TABLE III (CONT'D)

Test No.	w	T_{in}	ΔT	v	q/A Electrical	q/A Thermal	T_W	T_F	h	Re_F x 10^{-3}	Nu	$\frac{\mu_F}{\mu_W}$	$\left(\frac{Nu}{Re^{0.80}Pr^{0.33}}\right)$ x 10^{-2}	ΔP	$C_{F_{HT}}$	$\frac{C_{F_{iso}}}{C_{F_{HT}}}$
31	2.22	89	5.1	23.7	0.21	0.21	190	93	2.21	41.3	586	3.18	3.93	3.72	0.0179	1.27
32	2.22	91	7.4	23.8	0.31	0.31	230	97	2.36	43.3	626	4.32	4.10	3.35	0.0161	1.40
33	2.22	87	10.6	23.8	0.43	0.45	298	96	2.24	42.8	594	7.00	3.90	2.82	0.0136	1.65
34	2.22	93	14.2	23.8	0.59	0.60	354	102	2.38	46.0	631	9.07	4.00	2.48	0.0119	1.86
35	2.22	92	5.1	23.7	0.21	0.21	190	95	2.23	42.2	571	3.11	3.78	3.72	0.0179	1.26
36	2.22	89	7.5	23.7	0.31	0.31	238	90	2.12	39.9	562	5.00	3.83	3.32	0.0160	1.43
37	2.22	94	10.5	23.8	0.44	0.47	294	98	2.42	43.8	640	6.72	4.16	2.88	0.0138	1.62
38	2.22	90	5.1	23.7	0.21	0.21	191	90	2.12	39.9	562	3.33	3.83	3.70	0.0178	1.29
39	2.22	90	7.8	23.7	0.32	0.33	242	92	2.19	40.6	580	5.08	3.92	3.25	0.0156	1.46
40	3.84	89	8.7	41.0	0.62	0.63	261	92	3.72	70.3	986	5.74	4.30	8.50	0.0136	1.47
41	3.85	86	9.1	41.0	0.64	0.64	265	90	3.68	69.1	975	6.04	4.29	8.36	0.0134	1.50
42	3.84	88	8.9	41.0	0.63	0.63	264	90	3.62	69.1	960	6.00	4.22	8.49	0.0136	1.48
43	3.84	95	8.7	41.1	0.62	0.63	261	98	3.86	75.7	1023	5.32	4.29	8.52	0.0136	1.45
44	3.85	86	3.2	40.9	0.21	0.21	149	86	3.40	64.8	901	2.19	4.10	10.88	0.0175	1.17
45	3.84	88	4.7	41.0	0.33	0.37	188	88	3.69	66.9	978	3.32	4.32	9.98	0.0161	1.25
46	3.84	92	6.6	41.0	0.47	0.47	221	92	3.61	70.2	957	4.25	4.18	9.25	0.0149	1.34
47	3.85	87	9.1	41.0	0.64	0.63	264	89	3.60	67.6	954	6.13	4.24	8.34	0.0134	1.51
48	2.89	95	3.1	30.9	0.21	0.21	172	95	2.75	54.8	729	2.55	3.81	6.05	0.0171	1.24
49	2.89	92	6.3	30.9	0.34	0.33	213	92	2.72	52.9	721	3.98	3.95	5.51	0.0156	1.37
50	2.89	93	8.5	30.9	0.45	0.46	257	95	2.82	54.9	747	5.43	4.01	4.96	0.0140	1.52
51	2.89	94	11.5	30.9	0.62	0.62	310	92	2.83	52.9	750	7.89	4.11	4.40	0.0125	1.71
52	3.00	92	3.8	32.0	0.21	0.21	170	91	2.72	53.9	721	2.65	3.87	6.72	0.0177	1.20
53	3.01	87	6.2	32.0	0.34	0.34	211	89	2.76	52.7	732	4.07	3.98	6.09	0.0161	1.33
54	3.00	88	8.2	32.0	0.45	0.46	253	90	2.81	53.9	745	5.56	3.99	5.48	0.0145	1.47
55	3.00	92	11.0	32.1	0.62	0.62	305	96	2.98	57.8	790	7.30	4.09	4.85	0.0130	1.61
56	3.00	91	10.9	32.1	0.61	0.61	302	94	2.96	[illegible].4	785	7.35	4.11	4.89	0.0129	1.64
57	2.22	94	17.6	23.9	0.75	0.76	410	108	2.52	49.9	668	10.91	4.07	2.25	0.0106	2.05
58	3.00	89	13.6	32.2	0.76	0.78	353	100	3.08	60.7	816	9.20	4.12	4.40	0.0114	1.81

TABLE IV

DATA FOR PRESSURE DROP IN ISOTHERMAL FLOW

Test No.	Manometer Reading (in. Hg)		w	v	Re*	C_f
	Left	Right				
1**	7.50	6.05	3.84	40.66	54,970	0.0222
2	4.95	4.0	3.07	32.51	43,950	0.0229
3	2.84	2.25	2.23	23.61	31,920	0.0247
4	1.90	1.50	1.78	19.59	25,600	0.0255
5	2.80	2.25	2.23	23.61	31,920	0.0245
6	4.25	3.45	2.85	30.18	40,800	0.0229
7	4.90	3.95	3.07	32.51	43,950	0.0226
8	7.25	5.98	3.84	40.66	54,970	0.0216
9***	7.1	5.95	3.84	40.66	54,970	0.0213
10	4.62	3.9	3.07	32.51	43,950	0.0218
11	2.63	2.3	2.22	23.51	31,790	0.0241
12	1.75	1.6	1.78	18.85	25,485	0.0255
13	4.15	3.52	2.85	30.18	40,800	0.0228
14	2.65	2.30	2.22	23.51	31,790	0.0242
15	1.75	1.6	1.78	18.85	25,485	0.0255
16	4.72	4.0	3.07	32.51	43,950	0.0223
17	7.1	5.95	3.84	40.66	54,970	0.0213

*Bulk temperature for all tests was 73°F.

**Tests 1 through 8 were made January 28, 1949.

***Tests 9 through 17 were made February 3, 1949, with a different tube of similar configuration.

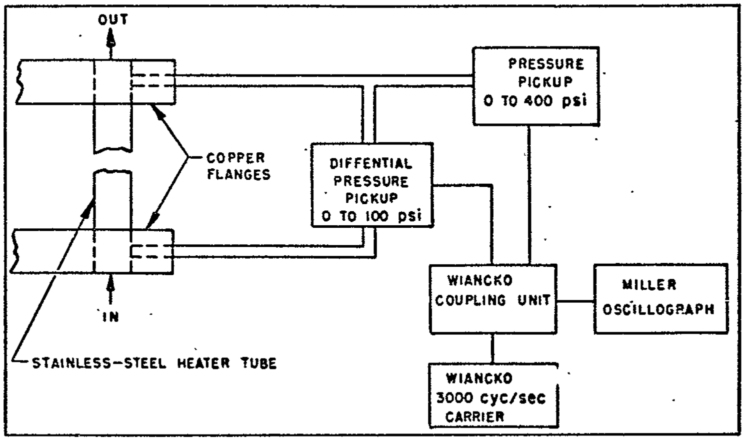

Figure 1. Schematic Line Diagram for Study of Pressure Fluctuations

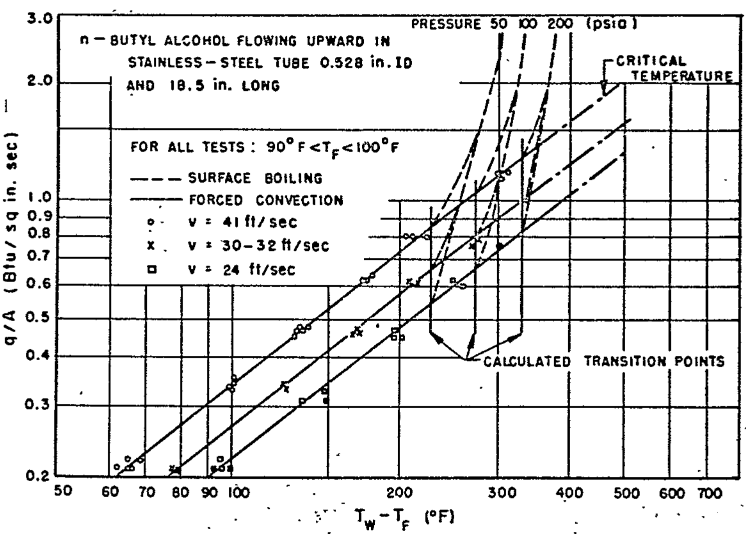

Figure 2. Heat Flux vs Temperature Potential for *n*-Butyl Alcohol at Various Flow Rates

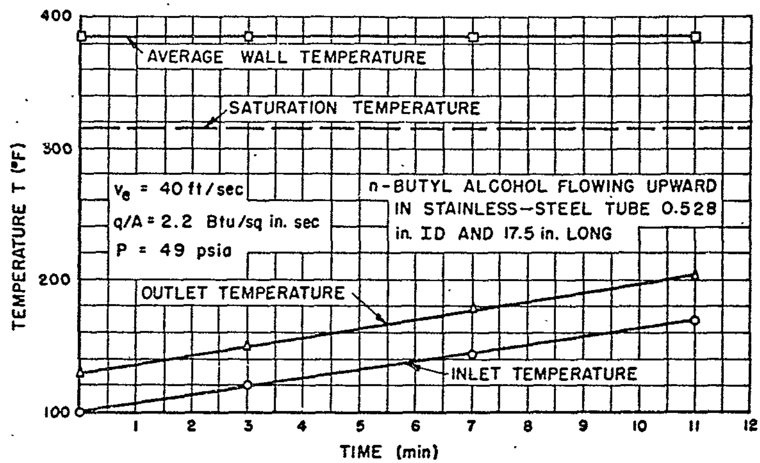

Figure 3. Constancy of Heat-Transfer Surface Temperature with Increasing Bulk Temperature

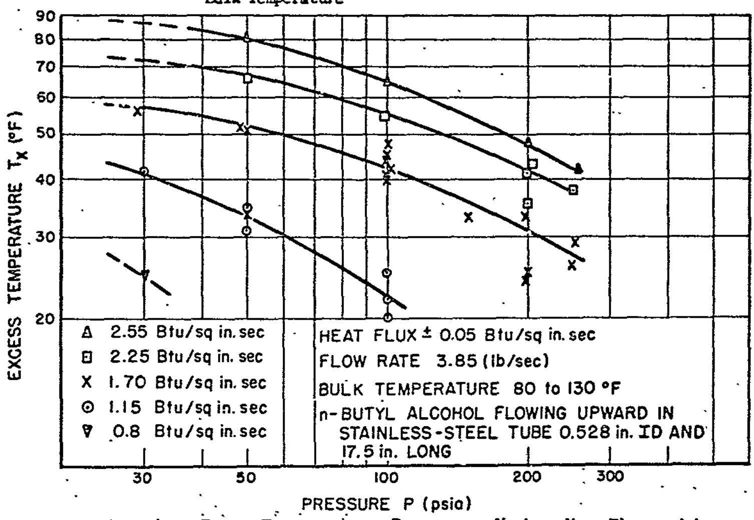

Figure 4. Excess Temperature vs Pressure at Various Heat Fluxes with Surface Boiling

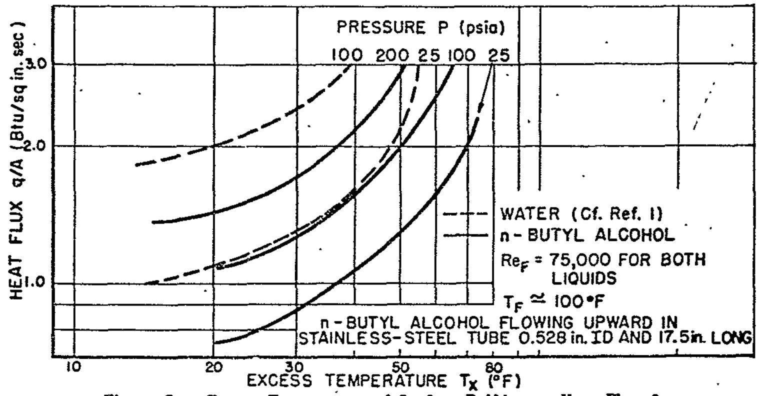

Figure 5. Excess Temperature of Surface Boiling vs Heat Flux for *n*-Butyl Alcohol and Water

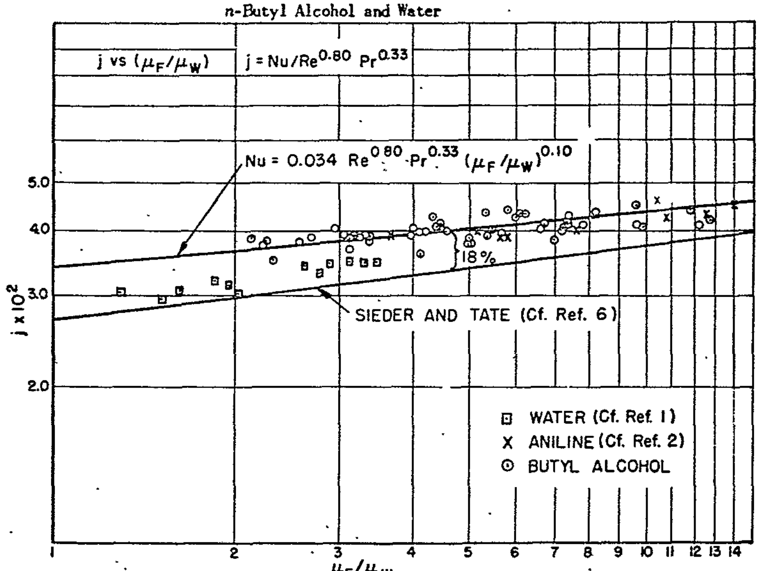

Figure 6. Correlation of Data for Heat Transfer by Forced Convection Without Surface Boiling

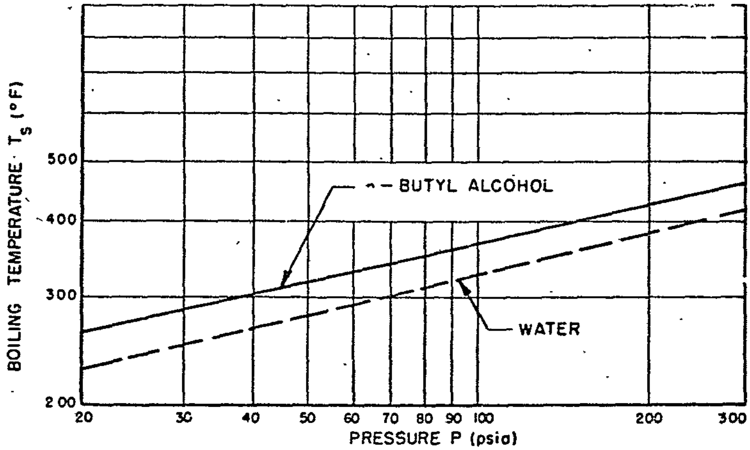

Figure 7. Boiling Temperature vs Pressure for *n*-Butyl Alcohol and Water

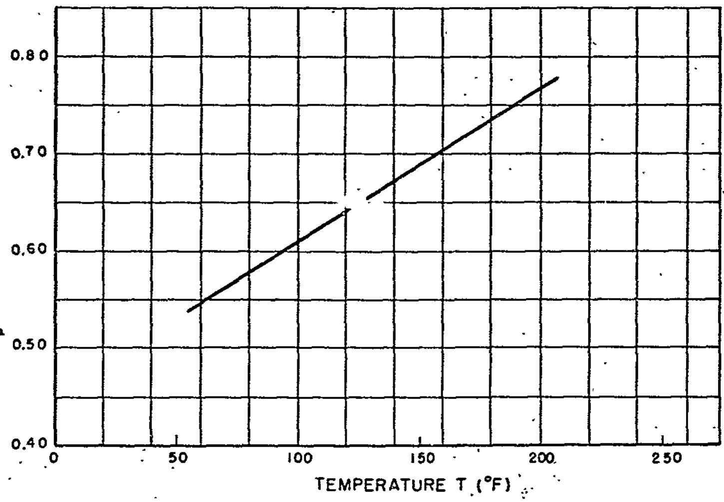

Figure 8. Specific Heat vs Temperature for *n*-Butyl Alcohol

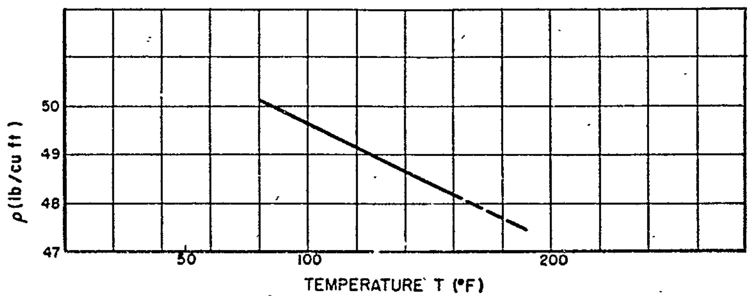

Figure 9. Density vs Temperature for *n*-Butyl Alcohol

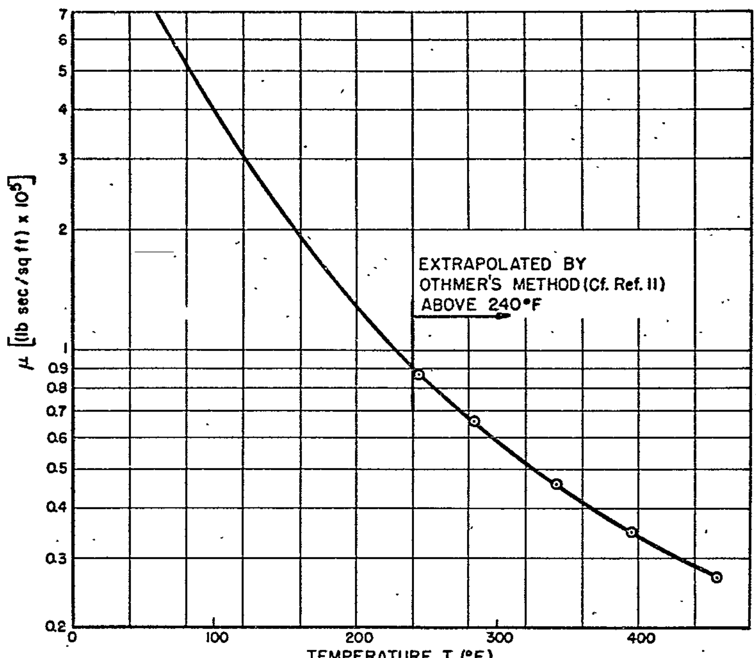

Figure 10. Absolute Viscosity vs Temperature for *n*-Butyl Alcohol

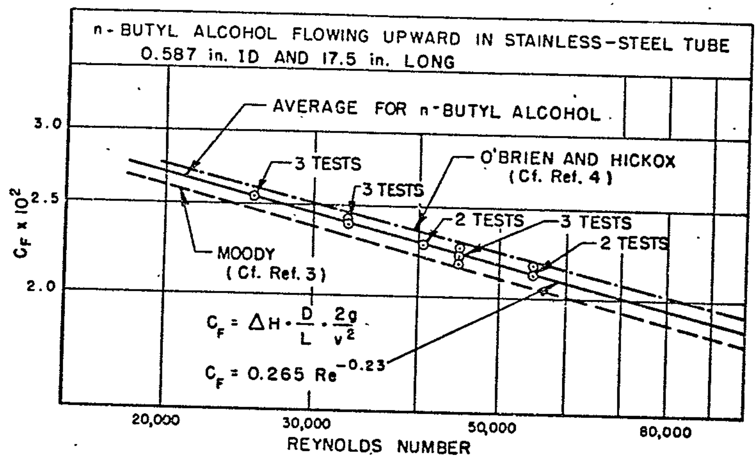

Figure 11. Isothermal Friction Coefficient vs Reynolds Number

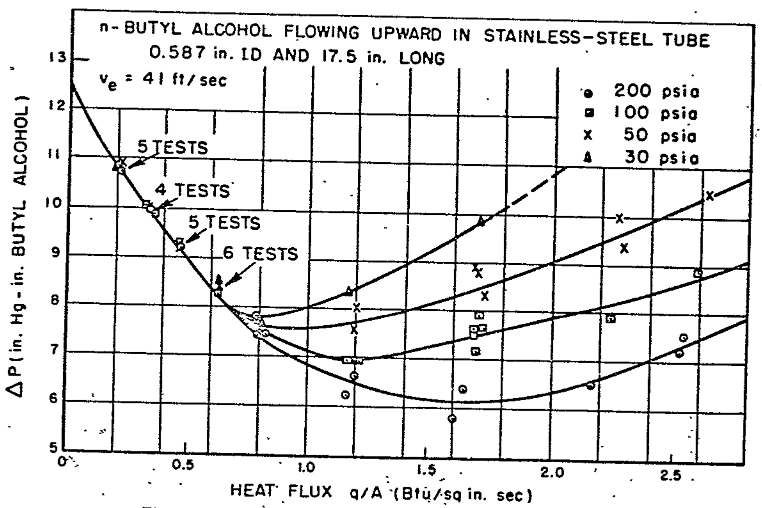

Figure 12. Effect of Heat Flux on Frictional Pressure Loss (v_e, 41 ft/sec)

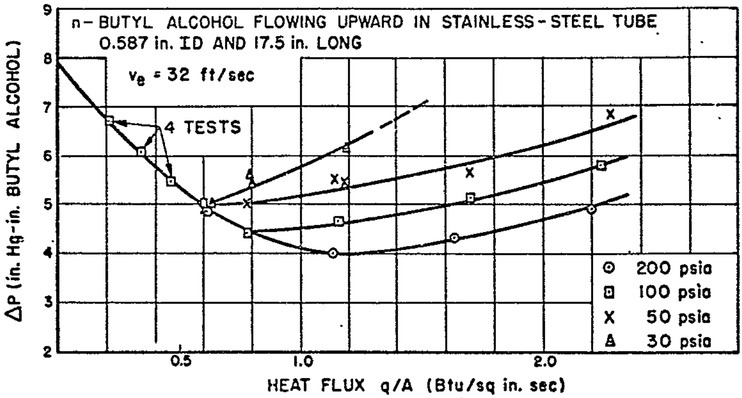

Figure 13. Effect of Heat Flux on Frictional Pressure Loss (v_e, 32 ft/sec)

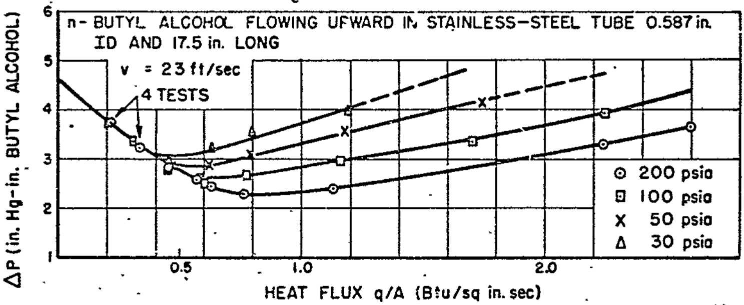

Figure 14. Effect of Heat Flux on Frictional Pressure Loss (v_e, 23 ft/sec)

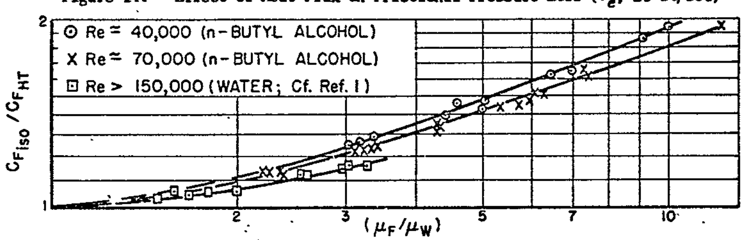

Figure 15. Ratio of Heat-Transfer (Nonboiling) Friction Coefficient to Isothermal Friction Coefficient vs Ratio of Viscosity at Bulk Temperature to Viscosity at Wall Temperature

REFERENCES

1. Kreith, Frank, and Summerfield, M., *Investigation of Heat Transfer at High Heat-Flux Densities: Experimental Study With Water of Friction Drop and Forced Convection With and Without Surface Boiling in Tubes,* Progress Report No. 4-68. Pasadena: Jet Propulsion Laboratory, April 2, 1948.

2. Kreith, Frank, and Summerfield, M., *Heat Transfer from an Electrically Heated Tube to Aniline at High Heat Flux,* Progress Report No. 4-88. Pasadena: Jet Propulsion Laboratory, February 7, 1949.

3. Moody, L. F., "Friction Factors for Pipe Flow," *Transactions of American Society for Mechanical Engineers,* 66:671-684, 1944.

4. O'Brien, M. P., and Hickox, G. H., *Applied Fluid Mechanics,* 1st ed. New York: McGraw-Hill Book Company, 1937.

5. McAdams, W. H., Addoms, J. N., and Kennel, W. E., *Heat Transfer at High Rates to Water with Surface Boiling,* Argonne National Laboratory, December, 1948.

6. Sieder, E. N., and Tate, G. E., "Heat Transfer and Pressure Drop of Liquids in Tubes," *Industrial and Engineering Chemistry,* 28:1429-1436, 1936.

7. Kreith, Frank, and Summerfield, M., *Investigation of Heat Transfer at High Heat-Flux Densities: Literature Survey and Experimental Study in Annulus,* Progress Report No. 4-65. Pasadena: Jet Propulsion Laboratory, February 20, 1948.

8. Morris, F. H., and Whitman, W. G., "Heat Transfer for Oil and Water in Pipes," *Industrial and Engineering Chemistry,* 20:234, 1928.

9. Kaye, G. W. C., and Higgins, W. F., "Thermal Conductivities of Certain Liquids," *Proceedings of Royal Society (London),* A117:459, 1928.

10. *International Critical Tables.* New York: McGraw-Hill Book Company, 1933.

11. Othmer, D. F., and Conwell, J. W., "Correlating Viscosity and Vapor Pressure of Liquids," *Industrial and Engineering Chemistry,* 37:1112-1115, 1945.

12. McAdams, W. H., *Heat Transmission,* 2nd ed. New York: McGraw-Hill Book Company, 1942.

13. Boelter L. M. K., Martinelli, R. C., and Jonassen, F., "Remarks on the Analogy Between Heat Transfer and Momentum Transfer," *Transactions of American Society of Mechanical Engineers,* 63:447-455, 1941.

14. *Eleventh Informal Monthly Report of MIT Project DIC 1-6489 for Period August 18 to September 18, 1947.* Cambridge: Massachusetts Institute of Technology, September 25, 1947.

REFERENCES (Cont'd)

15. Knowles, J. W., *Heat Transfer with Surface Boiling*. Montreal: Montreal Laboratories of the National Research Council, 1943.

16. Smith, J. F. Downie, "Heat Transfer and Pressure Drop Data for an Oil in a Copper Tube," *Transactions of American Institute of Chemical Engineers*, 31:83-111, 1934.

APPENDIX*

Several lengths of stainless-steel tubing which had cracked during heat-transfer tests were examined for possible causes of the failure. No obvious fault of the material was indicated as a cause of the failure.

The cracks in the wall of the stainless-steel tube (0.020-in. wall, 0.625 in. ID) were located in a longitudinal direction and were from 1/4 to 3/4 inch in length. The photomicrographs show a view of the cross section through a crack, unetched, at a magnification of 200 (Cf. Fig. A-1) and a longitudinal view at the same magnification after annealing and etching (Cf. Fig. A-2). Microhardness tests on polished surfaces showed hardness values corresponding to Rockwell C 35 to 40. The hardness of hard-drawn tubing falls within this range.

The literature cites instances of similar failures caused by applied stresses combined with high residual stresses and also of stress corrosion failures and fatigue failures. In the case of the stainless-steel tubes there is no evidence of stress corrosion failure. It is probable that the failure was caused by fatigue or an applied stress in addition to a high residual stress. Since this tube had a thin wall and was not annealed, high residual stresses from fabrication were unquestionably present. A high applied thermal stress could have been the result of cooling the inside wall of the heated tube. A fatigue failure could have been caused by cyclic stresses at a high frequency.

The microstructure shows inclusions elongated in the direction of drawing. These inclusions are not unusually severe for this type of tubing, but lower the strength and ductility in the hoop stress direction.

The most obvious remedy would be to anneal the tubing, thereby removing most of the residual stresses and increasing the ductility of the metal. This treatment would allow the tube to take up some of the thermal stresses without cracking.

*Lawrence J. Hull of this Laboratory helped in the preparation of the appendix.

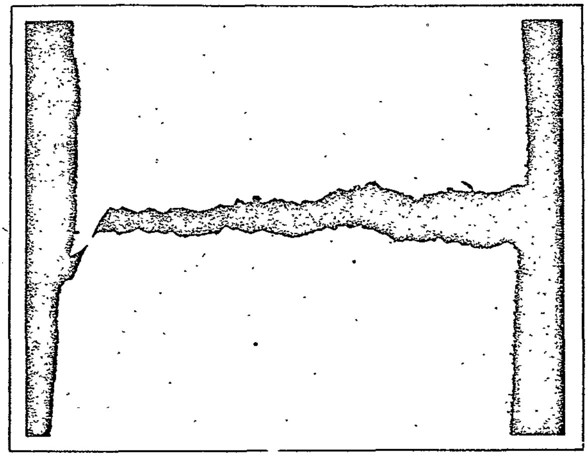

Figure A-1. Unetched Cross Section Through Crack of Stainless-Steel Tube

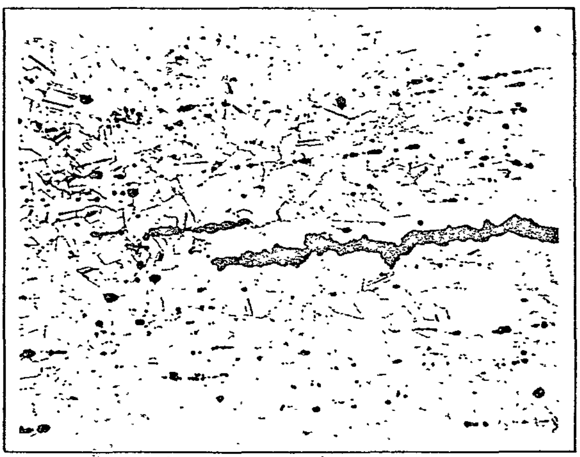

Figure A-2. Cross Section Through Crack of Stainless-Steel Tube After Annealing and Etching

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