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"EFFECT OF MIST FLOW ON
COOL-DOWN TEMPERATURES AND COOL-DOWN TIME.

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by

or J. W. H. Chi
Fluid Flow Laboratory

Approved By:

E. A. DeZubay

E. A. DeZubay, Manager
Fluid Flow Laboratory

H. S. McCreary

H. S. McCreary, Manager
Experimental Engineering

W. G. Roman

W. G. Roman, Manager
Reactor Engineering

MASTER

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E. A. DeZubay
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SUMMARY

An experimental study on the cool-down of aluminum test sections by liquid hydrogen has been carried out. The data have shown that film boiling occurs approximately 90% of the cool-down time. The dominant mechanism of two-phase flow observed is mist flow. A cool-down model was proposed and equations were derived for the prediction of cool-down wall and fluid temperatures at any time and location. The experimental wall temperature histories have been correlated well within experimental errors, and cool-down times were calculated with a standard deviation of $\pm 9.5\%$.

The equations developed permit estimations of cool-down time as well as wall and fluid temperatures at any time and distance along the channel provided that the two-phase fluid is in mist flow.

INTRODUCTION

In recent years, cryogenic fluids have found wide applications as coolants as well as the propellants in rocket propulsion. During transient operation of such systems, the transfer lines or flow passages must be cooled-down to the liquid boiling point before the maximum flowrate is obtained. In order to design coolant channels and transfer lines, the duration of the cool-down period must be known. Consequently, a number of cool-down studies have been carried out in an attempt to better understand the cool-down phenomena. Burke et. al.¹ studied the cool-down of a 175 foot, 2 inch outside diameter stainless steel line by liquid nitrogen, and Bronson et. al.² studied the cool-down of a 50 foot, 1-3/8 inch inside diameter test section by liquid hydrogen. The former investigators used a gross effect to calculate cool-down time. In this method, an average exit exhaust temperature was used for the duration of the cool-down period. Bronson et. al.² assumed a constant wall-to-coolant temperature difference along the entire length of the line. Although these techniques may provide sufficient accuracy when the gross effect over an extremely long transfer line is considered, it is doubtful that they can be applied over a short tube.

It was recognized that the mechanisms of two-phase flow and boiling heat transfer during cool-down is extremely complex; consequently, an experimental program was initiated to study the cool-down of metal test sections by liquid hydrogen. The data obtained is analyzed in an attempt to develop an equation for the prediction of cool-down time and temperatures.

EXPERIMENTAL RESULTS

EQUIPMENT AND TEST SECTIONS

The experimental equipment used in this study has been discussed in detail in a previous report.³ The test sections consisted of 26-inch aluminum tubes of 1/2-inch, 1-inch, and 2-inch outside diameters with 3/16-inch inside diameters. The 2-inch test section has 3-parallel channels.

INSTRUMENTATION

The test sections and test loop were instrumented with various pressure and temperature sensing devices and the outputs were recorded continuously on a high speed Minneapolis-Honeywell recording oscillograph (Visicorder^R). Fluid flow and pressure levels were controlled by remotely operated valves. Wall and fluid temperatures of the test sections were measured by copper-constantan thermocouples at five stations along the tubes. For wall temperature measurements, thermocouples were cemented into the wall at a distance of .067 inches from the inside wall. For the multiple passage test section, wall temperatures were measured at 3-radial positions .067-inch apart. Stream temperatures were measured by thermocouples positioned at the center of the test section. The thermocouple stations were located at equally spaced intervals. The positions of the thermocouple pressure taps for the 1/2-inch and 1-inch test sections are shown in Figure 1. Details on the location of instrumentation on the 3-parallel passage test section can be found in Reference 4. The first thermocouple plane is located 2-1/2-inches from the inlet and the last plane is located 1-1/2-inches from the exit of the test section. These planes shall be referred to as the "inlet" and the "exit" planes. In addition, bulk fluid temperatures at the orifice meter were measured to permit the calculation of mass flowrates and to assure single-phase gaseous flow through the orifice. The thermocouples were made from B & S 36 gauge wires. The beaded junctions had a maximum diameter of 12 mils. For wall temperature measurements, the thermocouples were cemented directly to the tube. For stream temperature measurements, Teflon^R insulated wires were inserted through sheath-type assemblies which were cemented to the test section; the junctions were positioned at the center of the tube. From the junction to the inside wall, the wires were uninsulated.

Pressures and pressure drops across the test section and the orifice meter were measured through a series of pressure taps connected to strain-gauge type pressure transducers. The same experimental procedure reported previously³ was used in this series of experiments.

EXPERIMENTAL RESULTS

Typical portions of bulk stream temperature traces are reproduced in Figure 2. Wall and time-smoothed vapor temperature histories from a typical run are shown in Figure 3. Sample data are presented in Tables I, II, and III. The tables represent data from the 1/2-inch, 1-inch, and 2-inch test sections, respectively.

For both the 1-inch and the 2-inch test sections, the radial wall temperature gradients were found to be within the experimental error; consequently, average wall temperature data are shown in Tables II and III. The fluid temperatures given for the multiple passage test section also represent average values for the three passages.

In a previous study on the cool-down of copper test section by liquid hydrogen,⁵ slug flow appeared soon after the initiation of cool-down and this mode of two-phase flow persisted throughout the cool-down period. In this study, slug flow was not observed until the test sections were partially cooled-down. The oscillating stream temperatures observed may therefore be attributed to mist flow. The occurrence of predominantly mist flow in these tests is due to the higher heat flux obtained from the thick-walled aluminum test sections and from the lower flowrates (see the Boiling number criterion for mist flow in Reference 6).

From the stream temperature traces, it was evident that at the entrance to the test section, the stream temperature dropped rapidly to the liquid boiling point.

DERIVATION OF EQUATIONS

In a previous paper,⁶ it was shown that film boiling prevails when the wall to liquid temperature difference exceeds 18.5°R. On this basis, the experimental data show that film boiling occurs approximately 90% of the cool-down time, the cool-down time being defined as the time required for the wall to reach the liquid boiling point. This fact suggests that a cool-down model should be based on film boiling.

By definition, a vapor film is in continuous contact with the wall during film boiling. Thus the heat transferred to the two-phase fluid must first be transferred to the vapor. This suggests that film boiling may be analogous to heat transfer to a single-phase gas. For all practical purposes, let the two-phase fluid be replaced by a fictitious fluid, having the properties of a gas. The heat flux from the wall to the fluid may then be given by the equation

$$\frac{Q}{A} = h (T_w - T_f) \quad (1)$$

Let it be assumed that the mass flowrate is constant and that the heat capacity and the density of the fictitious fluid is constant, then a heat balance on the fluid gives

$$h (T_w - T_f) - \frac{C_f w}{\pi D_i} \frac{\partial T_f}{\partial Z} = \frac{S \rho_f C_f}{\pi D_i} \frac{\partial T_f}{\partial t} \quad (2)$$

For an adiabatic system in which the thermal conductivity of the tube wall is relatively high, the local heat flux can be obtained from a simple energy balance, yielding

$$\frac{Q}{A} = - \rho C_p \delta \frac{dT_w}{dt} \quad (3)$$

Eliminating $\frac{Q}{A}$ from Equations (1) and (3) gives

$$-\rho C_p \delta \frac{dT_w}{dt} = h (T_w - T_f) \quad (4)$$

Assuming that the lumped parameter $\frac{h}{\rho C_p \delta}$ is a constant, and introducing the dimensionless variables

$$\xi = \frac{\pi D_i h z}{W C_f} \quad (5)$$

and

$$\tau = \frac{h}{\rho C_p \delta} \left(t - \frac{S \rho_f z}{W} \right) \quad (6)$$

Equations (2) and (4) can be simplified to

$$-\left(\frac{\partial T_f}{\partial \xi} \right)_{\tau} = T_f - T_w \quad (7)$$

and

$$\left(\frac{\partial T_w}{\partial \tau} \right)_{\xi} = T_f - T_w \quad (8)$$

If the temperature of the fluid entering the test section is at the liquid boiling point, then the boundary conditions are:

$$\begin{aligned} & \text{At } t = 0, & T_w = T_{wo}, \text{ all } z \\ \text{or} & \text{At } t = 0, & T_w = T_{wo}, \text{ all } \zeta \end{aligned} \quad (9)$$

$$\begin{aligned} & \text{At } z = 0, & T_f = T_{fo} = T_L, \text{ all } t \\ \text{or} & \text{At } \zeta = 0, & T_f = T_L, \text{ all } \tau \end{aligned} \quad (10)$$

Equations (7) and (8), together with the initial and boundary conditions (9 and 10) can be readily solved on an electronic analog computer. Analytical solutions can also be obtained by utilizing Laplace transforms.⁷ The solutions are:

$$\frac{T_w - T_{wo}}{T_L - T_{wo}} = \int_0^{\tau} e^{-(\tau + \zeta)} J_0(i\sqrt{4\tau\zeta}) d\tau \quad (11)$$

and

$$\frac{T_f - T_L}{T_{wo} - T_L} = \int_0^{\zeta} e^{-(\tau + \zeta)} J_0(i\sqrt{4\tau\zeta}) d\tau \quad (12)$$

where $J_0(x)$ is a zero-order Bessel function of the first kind. Equations (11) and (12) can be written in the alternate forms

$$\frac{T_w - T_L}{T_{wo} - T_L} = 1 - \int_0^{\tau} e^{-(\tau + \zeta)} J_0(i\sqrt{4\tau\zeta}) d\tau \quad (13)$$

and

$$\frac{T_{wo} - T_f}{T_{wo} - T_L} = 1 - \int_0^{\xi} e^{-(\tau + \xi)} J_0(i\sqrt{4\tau\xi}) d\tau \quad (14)$$

Note that Equations (13 and (14) permit the calculation of the wall and the fluid temperatures at any location. Curves representing numerical values of the integrals in Equations (13) and (14) have been presented by Hougen and Watson.⁸

ANALYSIS OF DATA

It can be shown that for a short test section, the second term in Equation (6) is negligible compared to t , hence the equation can be written

$$\tau = \psi t \quad (15)$$

where

$$\psi = \frac{h}{\rho C_p \delta} \quad (16)$$

According to the boundary conditions of the problem (10), the entering fluid is at the liquid boiling point. Thus at the entrance of the test section, Equation (4) becomes

$$\frac{dT_w}{T_w - T_L} = - \psi dt \quad (17)$$

which can be readily integrated to give

$$\int_{T_{wo}}^{T_w} \frac{dT_w}{T_w - T_L} = - \psi \int_0^t dt$$

or

$$\ln \frac{T_w - T_L}{T_{wo} - T_L} = -\psi t \quad (18)$$

Equation (18) shows that a plot of the logarithm of $\frac{T_w - T_L}{T_{wo} - T_L}$ versus the cool-down time at the entrance of the test section should yield a straight line with a slope $-\psi$. A correlation of the entrance wall temperature history by this equation should serve to determine the compatibility of the data with the assumptions used in the derivation of Equations (13) and (14).

It was evident from the fluid temperature traces at the entrance of the test section that the liquid boiling point was quickly attained. Thus the boundary condition (10) applies. Typical plots of the wall temperatures on semi-logarithmic paper is shown in Figures 4 and 5. Figure 4 gives the cool-down wall temperature histories of the one inch outside diameter aluminum test section at various flowrates while Figure 5 shows the cool-down inlet wall temperatures of the three different test sections. The excellent correlations shown in these figures indicate that the assumption of a constant lumped parameter is valid. However, similar plots for data taken at stations other than the inlet showed that there is no significant correlation by this equation. This is expected since the fluid temperature changes with increased heating. The parameter, ψ , was determined by "least-square" fit of Equation (18) to the data from the entrance plane, and the results were used to determine τ in Equation (6). The wall and fluid temperature histories at any location of the test section were then calculated from Equations (13) and (14).

Since it was assumed that ψ and C_f are constants, C_f and C_p were evaluated at the initial (ambient) temperature. The heat transfer coefficients were calculated from Equation (16). The results were then used to calculate $\mathcal{L}$, from Equation (5). For the three passage test section, the total flowrate was used to calculate average local wall and fluid temperature histories. Typical wall temperature and fluid temperature histories calculated by Equations (13) and (14) are compared with the experimental data in Figures 6 and 7. Figure 6 shows the calculated and observed wall and time-smoothed fluid temperature histories at the exit of the one inch aluminum test section.

Figure 7 is a comparison of calculated results with the data at the exit plane of the two inch, three parallel passage test section. The observed wall temperature history represents the average of values measured at three radial positions of the wall, while the fluid temperature history represents the average of the three flow passages. It is evident from these figures that excellent agreement on the wall temperature histories were obtained. As for the fluid temperatures, the calculated values were initially in good agreement but became consistently higher than the observed values as cool-down progressed. This discrepancy can be attributed to the finite response of the thermocouples. It can be shown that if the frequency of liquid droplets or slugs is too high, the thermocouples would not indicate the true temperatures. Instead, temperatures lower than the true values will be indicated. A rigorous analysis of this phenomena is not within the scope of this report.

It was observed that if there are significant deviations between observed and calculated wall temperatures, the deviations tend to increase with decreasing wall temperatures. It is doubtless that this is due to the increased errors in measured wall temperatures as the hydrogen boiling point is approached. It can be shown that near the hydrogen boiling point, a two percent error in oscillograph readings can introduce a temperature error as high as 40°R . This results from the fact that for copper-constantan thermocouples, a small error in emf in this range causes a large error in the temperature reading. In addition, galvanometer drift may introduce additional errors. The maximum possible error in wall temperature measurements was estimated to be 60°R . The deviations between observed and calculated wall temperatures are well within this maximum.

COOL-DOWN TIME

Since Equations (13) and (14) were developed on the basis of film boiling, the calculations on the wall temperatures were terminated when the wall to liquid temperature difference reached 18.5°R . In view of the fact that film boiling occurs 90% of the cool-down time, the cool-down time was therefore estimated by dividing the time corresponding to $T_w - T_L$ of 18.5°R by .90. The cool-down times thus calculated are plotted versus the observed values in Figure 8. The

cool-down time for the three different test sections were calculated for several locations. It is evident that the calculated cool-down times are in good agreement with the observed values. The maximum deviation between observed and calculated cool-down times is $\pm 21\%$. The standard deviation is $\pm 9.5\%$.

HEAT TRANSFER COEFFICIENT

The derivation of Equations (13) and (14) assumes that the lumped parameter ψ is a constant. The validity of this assumption was verified by the successful correlation of the data by Equations (13) and (14). For a given test section, ρ and δ are constants. Thus it appears that the ratio h/C_p remains a constant during cool-down. Therefore in order to facilitate the evaluation of the heat transfer coefficient, values were calculated from the experimentally determined lumped parameter with C_p evaluated at the initial, or ambient temperature. In this limit, the heat transfer coefficient is expected to conform to the Dittus-Boelter equation of the form:⁶

$$\frac{h D}{K_f} = A \left(\frac{D W}{\mu_f S} \right)^{.8} \left(\frac{C_p \mu}{K} \right)_f^{.4} \left(\frac{T_w}{T_b} \right)^{-.55} \quad (19)$$

Since the cool-down experiments were carried out at the same initial wall temperature (ambient, $435 \pm 5^\circ\text{R}$), the heat transfer coefficients evaluated at the initial condition should be a function of the mass flowrate only. Note that at the entrance of the test section, $T_b = T_f$, therefore, Equation (19) reduces to

$$h = A' W^{.8} \quad (20)$$

where

$$A' = \frac{A K_f}{D^{.2} (\mu_f S)^{.8}} \left(\frac{C_p \mu}{K} \right)_f^{.4} \left(\frac{T_w}{T_L} \right)^{-.55} \quad (21)$$

is a constant and the physical properties are to be evaluated at the film temperature, $\frac{T_w + T_L}{2}$. T_L is used in Equation (21) because the heat transfer coefficients were determined from wall temperature histories at the inlet, where the fluid temperature is T_L . Equation (20) indicates that a plot of the logarithm of the heat transfer coefficient versus the logarithm of the mass flowrate should yield a straight line with a slope of 0.8, i. e.,

$$\log h = \log A' + 0.8 \log W \quad (22)$$

A test of the data by Equation (22) is shown in Figure 9, where a straight line with a slope of 0.8 is fitted to the data. The constant A' evaluated by "least squares", gives .0418 as the value of A' . From this A was determined to be .0310.

It is not surprising that this coefficient is higher than that proposed previously⁶ for heat transfer to gaseous hydrogen (0.0205), because the "effective" heat transfer coefficient given here includes the influence of boiling.

DISCUSSIONS AND CONCLUSIONS

The derivation of Equations (13) and (14) was carried out with the following assumptions:

1. Negligible radial wall temperature gradients.
2. Negligible axial conduction.
3. The fluid is homogeneous.
4. The system is adiabatic.
5. The inlet fluid is at its saturation temperature.
6. The flowrate is constant.

Although these assumptions apply specifically to the system studied here, more general equations may be derived for other initial and boundary conditions. However, it can be shown that these will lead to unwieldy equations that are not amenable to closed-form analytical solutions.

In previous reports^{4,6} cool-down of a copper test section by liquid hydrogen was studied. The experimental data obtained clearly indicated that slug flow is the dominant mechanism of two-phase flow. Moreover, the non-isothermal two-phase fluid entering the test section consisted of liquid slugs existing simultaneously with a superheated vapor. These phenomena suggest that Equations (13) and (14) will not apply in this case. A test on the data first by Equation (18) and then by Equations (13) and (14) showed significant deviations in all cases, thus verifying the fact that Equations (13) and (14) cannot be generally applied to the prediction of cool-down.

The successful correlation of the data shown here suggests that for the case of mist flow, a homogeneous fluid can be assumed, and for simple systems conforming to the conditions listed above, Equations (13) and (14) may be used to estimate cool-down time and temperatures. The lumped parameter, ψ , may be evaluated at the initial or ambient temperature with the heat transfer coefficient given by the equation

$$\frac{h D}{K_f} = .0310 \left(\frac{D G}{\mu_f} \right)^{.8} \left(\frac{C_p \mu}{K} \right)_f^{.4} \left(\frac{T_w}{T_L} \right)^{-.55}$$

where

$$T_f = \frac{T_w - T_L}{2}$$

It is obvious that the simple treatment presented here cannot be applied to transient reactor startup where a variety of two-phase flow phenomena is expected to occur. Moreover, the flowrate is not expected to remain constant. However, the method does permit a rapid estimation of cool-down temperatures wherever it is applicable. It should be particularly useful when designing test equipment or the experimental tests. An example where the equations are particularly pertinent is the cool-down by a cold gas.

REFERENCES

1. Burke, J. C. , et. al. , "Pressurized Cool-Down of Cryogenic Transfer Lines," *Advances in Cryogenic Engineering*, Vol. 4, Plenum Press, Inc. , New York, 1960, p. 378.
2. Bronson, J. C. , et. al. , "Problems in Cool-Down of Cryogenic Systems," *Advances in Cryogenic Engineering*, Vol. 4, Plenum Press, Inc. , New York, 1960, p. 198.
3. Campi, F. A. , et. al. , "Transient Two-Phase Heat Transfer and Flow Characteristics of Liquid Hydrogen," WANL-TNR-102, April, 1963.
4. Chi, J. W. H. and Campi, F. A. , "Horizontal Multiple Passage Transient Boiling Flow With No Heating," WANL-TME-379, May 30, 1963.
5. Chi, J. W. H. and Vetere, A. M. , "Two-Phase Flow During Transient Boiling of Hydrogen and Determination of Non-Equilibrium Vapor Fractions," *Advances in Cryogenic Engineering*, Vol. 9, Plenum Press, 1964.
6. Chi, J. W. H. , "Forced Convection Boiling Heat Transfer to Hydrogen," WANL-TNR-154, 1964.
7. Mickeley, H. S. , Sherwood, T. K. , and Reed, C. E. , "Applied Mathematics for Chemical Engineers," 2nd Edition, McGraw-Hill Book Company, Inc. , 304 (1957).
8. Hougen, O. A. and Watson, K. M. , "Chemical Process Principles, Part III," John Wiley and Sons, pp. 1086-1088 (1947).

NOMENCLATURE

A	= a constant
B	= a constant
C_f	= heat capacity of the fluid, BTU/lb.
C_p	= heat capacity of the test section, BTU/lb.
D	= diameter of the test section
h	= heat transfer coefficient, BTU/in ² -sec-°R
k	= thermal conductivity, BTU/in-sec-°R
$\frac{Q}{A}$	= local heat flux, BTU/in ² -sec
t	= cool-down time, seconds
T	= temperature, °R
w	= mass flowrate, lb/sec
z	= distance from the inlet of the test section, inches

Subscripts

f	= fluid
i	= inside
L	= liquid
o	= initial
w	= wall

Greek Letters

δ	= test section geometrical factor, volume/heat transfer surface, in.
ξ	= dimensionless distance parameter
μ	= viscosity, lb/in-sec
ρ	= density, lb/in ³
τ	= dimensionless time parameter
ψ	= lumped parameter, $\frac{h}{\rho C_p \delta}$, sec ⁻¹

TABLE I
COOL-DOWN WALL AND STREAM TEMPERATURES
ONE-HALF INCH ALUMINUM TEST SECTION
(TEMPERATURE, °R)

t, sec.	Plane I		Plane II		Plane III		Plane IV		Plane V	
	Wall	Fluid	Wall	Fluid	Wall	Fluid	Wall	Fluid	Wall	Fluid
Run 2-AL-01 $W = 1.68 \times 10^{-3}$ $P_{in} = 20$ psig										
0	551				552	547	553	552	557	552
5	490				525	220	519	335	529	287
10	409				450	200	463	257	466	250
20	290				308	155	336	200	350	175
30	215				210	60	246	140	256	75
40	175				165	65	212	95	211	75
50	140				130	65	150	80	182	58
60	120				93	50	115	50	157	50
70	90				93	40	93	50	135	45
80	80				65	40	93	40	135	40
90	40				40	40	80	40	100	40
100	40				40	40	40	40	90	40
110	40				40	40			80	
120	40				40	40			75	

Run 2-AL-02 $W = 1.12 \times 10^{-3}$ lb/sec $P_{in} = 15$ psig

0	540	540	540	540	540	540	540	540	540	540
1		174		215		333		410		375
2		112		152		244		344		282
3		112		149		229		321		263
4		112		138		210		292		246
5	483	85	488	145	500	201	509	268	500	236
10	405	70	393	120	435	155	655	226	429	186
15	330	59	311	100	357	125	386	178	341	140
20	280	56	247	85	290	93	298	145	299	125
30	197	50	141	52	175	59	235	100	203	80
40	128	45	110	45	130	45	172	80	180	50
50	148	40	95	40	120	40	136	65	155	40
60	130		72	40	99	40	112	40	140	
70	105				80		90		111	
80	90				60		79		102	
90	70				60		60		90	
100	59				40		50		70	
110	40				40		40		40	
120										
130										

TABLE II
COOL-DOWN WALL AND STREAM TEMPERATURES
ONE-INCH ALUMINUM TEST SECTION
(TEMPERATURE, °R)

<u>t, sec.</u>	<u>Plane I</u>		<u>Plane II</u>		<u>Plane III</u>		<u>Plane IV</u>		<u>Plane V</u>	
	<u>Wall</u>	<u>Fluid</u>	<u>Wall</u>	<u>Fluid</u>	<u>Wall</u>	<u>Fluid</u>	<u>Wall</u>	<u>Fluid</u>	<u>Wall</u>	<u>Fluid</u>
Run 1-AL-05 $W = 2.72 \times 10^{-3}$ lb/sec $P_{in} = 21$ psig										
0	534	530	532	530	531	530			534	530
10	476	174	493	200	486	260			497	353
20	423	145	450	172	443	223			462	312
30	370	86	398	142	404	178			429	262
40	328	94	358	150	305	191			396	272
60	262	60	312	95	275	105			334	162
80	210	82	305	100	208	123			274	172
100	167	62	152	45	155	60			218	102
120	131	40	125	40	115	60			168	95
140	100	40	100	40	79	40			132	75
160	76	40	70	40	54	40			103	40
180	60	40	40		50				80	
Run 1-AL-08 $W = 4.80 \times 10^{-3}$ lb/sec $P_{in} = 27$ psig										
0	532	532	532	532	532	532			532	532
10	568	122	480	130	475	165			468	240
20	380	97	402	100	390	125			393	214
30	305	62	318	90	310	65			320	120
40	245	52	250	85	240	82			262	80
50	210	45	200	40	190	40			210	60
60	165	40	148	40	135	40			165	40
70	135		108		100				140	40
80	110		72		65				118	
90	100		40		58				112	
100	85		40		50				92	
110	78		40		40				50	
120	70				40				40	
130	58									
140	40									

TABLE III
COOL-DOWN WALL AND STREAM TEMPERATURES
TWO-INCH ALUMINUM TEST SECTION
(TEMPERATURE, °R)

<u>t, sec.</u>	<u>Plane I</u>		<u>Plane III</u>	
	<u>Wall</u>	<u>Fluid</u>	<u>Wall</u>	<u>Fluid</u>
0	512	510	510	510
10	450	122	472	358
20	387	118	477	358
40	298	82	427	285
60	236	65	389	232
80	188	59	327	187
100	157	55	276	152
120	133	55	228	131
140	109	40	189	105
160	92		155	80
180	44		134	68
200	40		117	55
220			109	40
240			90	40
260			40

FIGURE 1

SCHEMATIC DIAGRAM OF A SINGLE PASS TEST SECTION

TYPICAL COOL-DOWN STREAM TEMPERATURE TRACES, SECOND PLANE

FIGURE 3

TYPICAL WALL AND AVERAGE FLUID TEMPERATURE HISTORIES
DURING COOL-DOWN OF THE 1/2 INCH O.D. ALUMINUM TUBE

FIGURE 4
CORRELATION OF EXPERIMENTAL COOL-DOWN
WALL TEMPERATURES BY EQUATION 18
(INLET OF 1-INCH TEST SECTION)

FIGURE 5
CORRELATION OF EXPERIMENTAL COOL-DOWN
WALL TEMPERATURES BY EQUATION 18
(INLET OF THE TEST SECTIONS)

FIGURE 6

TYPICAL COMPARISON OF EXPERIMENTAL
AND CALCULATED COOL-DOWN TEMPERATURES
(1-INCH TEST SECTION)

FIGURE 7

TYPICAL COMPARISON OF EXPERIMENTAL
AND CALCULATED COOL-DOWN TEMPERATURES
(2-INCH TEST SECTION)

FIGURE 8

COMPARISON OF OBSERVED AND CALCULATED COOL-DOWN TIMES

FIGURE 9

EFFECT OF FLOWRATE ON THE HEAT TRANSFER COEFFICIENT