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TID/SNA--164

T. C. CORE
J. F. HARKEE
B. MISRA
K. SATO

MAIL

SEPTEMBER 1959

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HEAT-TRANSFER STUDIES

T. C. CORE
J. F. HARKEE
B. MISRA
K. SATO
Aerojet-General Corporation

September 1959

Power Plant Laboratory
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United States Air Force
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FOREWORD

This final report (Aerojet No. 1671) was prepared by T. C. Core, J. F. Harkee, B. Misra, and K. Sato of the Aerojet-General Corporation, Azusa, California. It is based on work conducted under Air Force Contract AF 33(616)-5289, "Heat-Transfer Studies." The work was directed by the WADC Power Plant Laboratory - WCLP, and administered by the Los Angeles Air Procurement District, San Bernardino Air Materiel Area, Los Angeles, California.

ABSTRACT

Heat transfer to hydrogen in the regions of forced convection in the liquid and gaseous phases, nucleate boiling, and stable film boiling has been investigated with a blow-down type apparatus utilizing an electrically heated stainless-steel tubular test section through the following range of conditions:

Velocity	25 to 55 ft/sec
Inlet bulk temperature	40 to 57°R <i>4-151 psi Sat. p</i>
Outlet bulk temperature	40 to 147°R
Inlet pressure	28 to 213 psia
Heat flux	0.01 to 6.0 Btu/in. ² sec
Wall temperatures	51 to 1505°R

All experiments were terminated prior to tube burnout. Based on the experimental heat-flux data on hydrogen under conditions of stable film boiling and forced convection in the gaseous phase, the heat-transfer analysis in a low-pressure thrust chamber was carried out. The results indicate the feasibility of employing hydrogen as a regenerative coolant.

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I. INTRODUCTION

With rapid development of cryogenic engineering, liquefied gases, such as hydrogen and oxygen, have become indispensable as propellants for modern high-performance rockets. Because of their high heat-sink capabilities, the liquefied gases are very promising as potential regenerative coolants. However, experimental studies* with liquefied gases indicate that a large reduction in the heat-transfer coefficient takes place when the fluid is near its critical point. Since the cryogenic fluids possess very low critical temperatures, it is impossible to avoid the critical point in their application as regenerative coolants. Therefore, the objective of this investigation is to (a) determine the heat-transfer characteristics of hydrogen under forced-flow conditions between the normal boiling point and the critical point and (b) to ascertain the feasibility of using hydrogen as a regenerative coolant.

II. EXPERIMENTAL PROCEDURE

The experimental system used was basically a single-pass blow-down loop employing an electrically heated, stainless-steel test section located in a vacuum chamber to minimize errors caused by environmental heat sources. A technical discussion of system components is presented in Appendix I. A photograph of the experimental apparatus is shown in Figure 1. The following parameters were measured: (a) fluid-flow rate, (b) inlet pressure of fluid, (c) inlet temperature of fluid, (d) bulk-temperature rise of the fluid, (e) outside-wall temperature of test section at two locations, (f) voltage drop between the electrodes, and (g) electric current flowing through test section.

A typical test was conducted as follows:

A pre-operational check of the system was conducted to ascertain that all valves and instruments were in operational condition. Also, a check was made to see that a sufficient vacuum existed in the tank and test section enclosures. The fluid tank and flow system were then purged with hydrogen gas to remove any condensable gases.

Fluid-loading was accomplished by attaching a vacuum-jacketed fill line between the transport trailer and the fluid tank. The fill line was evacuated and purged with helium to remove all air. The entire test system fill line and trailer were electrically connected to a common ground line. The fluid tank was then vented to a lower pressure than that in the trailer. Fluid flow was

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*W. B. Powell, "Heat Transfer to Fluids in the Region of the Critical Temperature," JPL Progress Report No. 20-285 (1956).

initiated by opening the liquid valve on the trailer and the fill valve on the fluid tank. Filling was terminated when a thermocouple, located on the fluid tank vent line, indicated a temperature below 142°R (the normal bubble point of air). The fill line was purged with nitrogen gas before its removal from the supply trailer.

The test was conducted by pressurizing the fluid tank with gaseous hydrogen to a pre-calculated pressure in order to attain the proper fluid-flow rate. The back-pressure regulator was set to the required test-section pressure. The test-section line valve was opened several times momentarily to permit small amounts of test fluid to flow and pre-chill the test section to a low initial temperature. The test-section valve was opened, and the desired flow rate was established. Electrical power was applied to the test section, and all parameters were recorded after stable readings were attained. The procedure was repeated by increasing the electrical power in small increments. The test was terminated when wall temperatures of the order of 1500°R were reached or test fluid was exhausted. The inlet fluid temperature was varied by bubbling and condensing dry gaseous hydrogen into the fluid tank. The test-section pressure was varied by changing the fluid tank pressure and back-pressure valve settings. The fluid-flow rate for each test condition was varied by changing the cavitating venturi or by changing the fluid tank pressure. For each test the inlet bulk temperature, fluid flow rate, and pressure of the fluid were kept constant during the experiment.

III. TECHNICAL DISCUSSION

A. EXPERIMENTAL RESULTS

Twenty-seven heat-transfer tests with liquid hydrogen were carried out under the following range of conditions:

Velocity	20 to 55 ft/sec
Inlet bulk temperature	40 to 57°R
Outlet bulk temperature	40 to 147°R
Inlet pressure	28 to 213 psia
Heat flux	0.01 to $6.0 \text{ Btu/in.}^2 \text{ sec}$
Wall temperature	51 to 1505°R

It was stated previously* that the bulk temperatures of hydrogen, as measured by the platinum resistance thermometer, were inconsistent with the bulk temperatures calculated from the pressure measured at the throat of the cavitating venturi (assuming the measured throat pressure is equal to the saturation pressure of the fluid, which determines the bulk temperature). This discrepancy was caused by the use of simplified extrapolation procedures in obtaining the temperature-resistance relationship for the platinum resistance thermometers. However, when the extrapolation procedure of Hoge and Brickwedde (Appendix II) was followed, the temperature readings of the resistance thermometers agreed well with the temperatures

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*Heat Transfer Studies, Aerojet Report No. 0109-01-8, 23 July 1959.

calculated from the pressure readings of the cavitating venturi. A brief discussion of the techniques used to measure the temperature and the temperature differences is presented in Appendix II.

For simplicity, the throat pressure of the cavitating venturi was used in all cases to determine the inlet bulk temperature of the liquid hydrogen. The outer-wall temperature was measured by means of the Chromel-Alumel thermocouples. The flow rate was measured with a cavitating venturi, which had been calibrated with water. The electrical power dissipated in the test section was obtained as the product of the measured voltage drop across the test section and the current through the test section.

The four experimental values (inlet bulk temperature, outer-wall temperature, flow rate, and electrical-power dissipation), together with the static-pressure upstream of the test section, constitute the measured data used to calculate the values presented in Table 1. This table gives the heat flux at various power levels, inlet pressure, mass velocity, temperature difference between bulk fluid and inside-wall, bulk temperature at the thermocouple location, weight percent of vapor in the bulk fluid, and the generalized liquid-film heat-transfer coefficient.

B. CALCULATIONS

Assuming an adiabatic outer wall and uniform power dissipation throughout the tube wall, the heat flux (Btu/in.² sec) was calculated as

$$\frac{Q}{A} = 0.948 \frac{P}{A} \quad (1)$$

where

P = power (kw)

A = heat-transfer area (in.²)

The enthalpy change of fluid across the test section is a function of the power and flow rate and is given by

$$\Delta H = 0.948 \frac{P}{W} \quad (2)$$

where

ΔH = enthalpy change (Btu/lb)

W = flow rate (lb/sec)

P = power (kw)

The inlet bulk temperature is assumed to be equal to the saturation temperature corresponding to the vapor pressure measured at the throat of the cavitating venturi.

The position of the Chromel-Alumel thermocouple on the heated tube is a known fraction of the distance between the upstream and the downstream electrodes. Therefore, the enthalpy increase of the bulk fluid up to the thermocouple is equal to this fraction of the total enthalpy change as calculated above. The enthalpy change, together with the inlet bulk temperature and measured inlet pressure,

determines the state (temperature, pressure, and percentage evaporated, if any) of the bulk fluid at the thermocouple position. The temperature-entropy diagram (NACA File No. 4842) was used to obtain the state of the bulk liquid at the thermocouple location.

Assuming constant power dissipation in the wall material and an adiabatic outer wall, the temperature drop through the wall ($^{\circ}\text{R}$) is given by

$$\Delta T_{\text{wall}} = \frac{Q/A}{k_m} (r_1) \left[\frac{1}{1 - (r_1/r_o)^2} \ln \frac{r_o}{r_1} - 1/2 \right] = 0.00489 \frac{Q/A}{k_m} \quad (3)$$

where

k_m = mean thermal conductivity (Btu/in. sec $^{\circ}\text{R}$)

r_1 = inner radius (0.08375 in.)

r_o = outer radius (0.09375 in.)

The liquid-side wall temperature (T_{wl}) is equal to the measured outer-wall temperature minus the temperature drop through the wall. From these calculations the parameters h and h' can be obtained:

$$h = \frac{Q/A}{\Delta T_f}$$

$$h' = h D^{0.2} / G^{0.8}$$

where

h = heat transfer coefficient defined as the proportionality constant in Newton's Law of Cooling, $Q/A = h \cdot \Delta T_f$ (Btu/in.² sec $^{\circ}\text{R}$)

ΔT_f = film temperature difference, $T_{wl} - T_b$ ($^{\circ}\text{R}$)

T_{wl} = liquid-side wall temperature ($^{\circ}\text{R}$)

T_b = bulk-liquid temperature ($^{\circ}\text{R}$)

h' = generalized heat-transfer coefficient

$$\text{Btu/in.}^2 \text{ sec } ^{\circ}\text{R} \times \frac{(\text{in.})^{0.2}}{(\text{lb/in.}^2 \text{ sec})^{0.8}}$$

D = diameter (in.)

G = mass velocity (lb/in.² sec)

C. ANALYSIS AND INTERPRETATION OF DATA

For general application of the data, it is desirable to normalize the heat-transfer coefficient by taking the different fluid-flow conditions into account. Therefore, the generalized heat-transfer coefficient (h') rather than the conventional heat-transfer coefficient (h) was calculated for analysis. The calculated values of Q/A and h' at each power level and at the two thermocouple locations are summarized in Table 1.

Heat balance, based on the electrical energy dissipation in the test section and the enthalpy rise of the fluid, could not be made in the majority of the experiments because of partial or complete vaporization of the fluid (see Table 1) and the lack of instrumentation to measure the quality of the fluid. However, heat balance based on the remainder of the data indicates that the experimental data are accurate within $\pm 10\%$.

The variation of Q/A and h' with ΔT_f is presented in Figures 2, 3, 4 and 5. A few of the data points which deviated extremely from the norm have been excluded from these figures. When plots of Q/A vs ΔT_f and h' vs ΔT_f were made for each inlet pressure condition below and above the critical pressure of hydrogen, no effect of pressure could be noticed. Therefore, it was concluded that the heat-transfer data under the present experimental conditions in the subcritical and supercritical pressure regions are independent of pressure, and the data for all pressures are presented in singular plots.

Figures 2, 3, 4 and 5 show some scatter in data, especially below a ΔT_f of about 100°F . It is not possible to attribute the scatter to any single factor. The scatter observed is attributed to

1. Uncertainties in measuring low temperatures (e.g., -400 to -420°F) and computation of small film temperature differences (e.g., 2 to 15°R)*
2. Flow instability due to entrance effects; i.e., by the lack of an upstream calming section and low length to diameter ratios
3. Flow instability due to phase change and two-phase flow
4. Oscillation, in the mode of heat-transfer phenomenon; i.e., change from forced convection to nucleate and film boiling caused by microscopic change in fluid-flow conditions.

It was not practical to measure the fluid bulk-temperature and the wall temperature at a number of points on the short test section, to establish clearly where the mode of heat-transfer phenomenon changed from non-boiling forced convection to nucleate and stable film boiling. Therefore, the "best fit" lines and the heat-transfer regions, as indicated in Figures 2 and 4, should be considered as approximate.

The rapid increase of heat flux with film temperature drop in the nucleate and the stable film boiling regions for liquid hydrogen, as indicated by Figure 2, is similar to the behavior of the common liquids. The heat flux curve for hydrogen increases slowly in the transition boiling region, indicating an orderly transition from nucleate to film boiling, whereas instability and decrease in heat flux have been observed with the conventional liquids.

One of the significant effects that can be observed from Figure 2 is that the heat flux (Q/A) increases continuously with increasing film temperature drop (ΔT_f) even though the fluid experiences a phase change and subsequently is heated in the gaseous state. This is a very important observation from the standpoint of

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*For example, if the uncertainties in the measurement of bulk temperature (at $T_b = 55^\circ\text{R}$) and wall temperature (at $T_{w1} = 70^\circ\text{R}$) are ± 2 and $\pm 10^\circ\text{R}$ respectively, then the uncertainty in the film temperature computation is $\pm 80\%$.

using hydrogen as a regenerative coolant, since liquid hydrogen possesses very small sub-cooling temperatures for transfer of heat by nucleate boiling. The superior heat-transfer characteristics of gaseous hydrogen observed in these experiments may be attributed to its favorable transport properties and the large increase in the linear velocity of fluid when the phase change occurs.

Figure 4 indicates the existence of a minimum h'^* in the transition boiling region near $\Delta T_f = 250^\circ R$.

Calculations show that an appreciable pressure drop across the test section occurs especially at higher mass-flow and heat-flux rates. Therefore, in the absence of pressure-drop measurements for tests conducted with inlet pressure slightly above the critical pressure (i.e., tests between 190 and 200 psia), distinction between data points which are subcritical or supercritical in pressure should not be made. Therefore, the data above 190 psia are presented separately in Figures 4 and 5 without indicating the different boiling regions.

The heat flux vs film temperature drop curve for hydrogen above its critical pressure, as shown in Figure 3, is similar to the heat flux curve in the subcritical pressure region. However, no minimum h' at $\Delta T_f = 250^\circ R$ is indicated by Figure 5. The existence of a minimum h' in this region could not be established owing to lack of data at larger film temperature drops.

From the study of heat-transfer characteristics of liquids in the subcritical region, most of the investigators have empirically correlated their data in terms of $\Delta T_{sub} = (T_{saturation} - T_{bulk})$, and linear velocity of the liquid. However, no such correlation could be made in this study, due to the rapidly diminishing value of ΔT_{sub} and the change of phase leading to complete vaporization of the liquid. Also, no velocity-profile stabilization could be assured because of the short length of the test section. It is the opinion of the authors of this report that an empirical correlation based on the present data alone would not have a very wide applicability. Therefore, all of the data are presented in tabular and graphical forms.

D. CONCLUSIONS AND RECOMMENDATIONS

The study of the heat-transfer characteristics of hydrogen below the critical state shows that these characteristics are independent of pressure and change of phase. Based on the experimental results, heat-transfer analysis (Appendix III) in a low-pressure thrust chamber indicates the feasibility of employing hydrogen as a regenerative coolant. The experimental data below the critical pressure indicate the existence of a minimum h' .

The outlet temperature of hydrogen encountered in this investigation was below $150^\circ R$, due to the short length of the test section. Although reasonable extrapolation of the data could be made beyond the experimental range, it would be desirable to extend the experimental data to higher bulk temperatures since higher coolant bulk temperature are likely to occur in regeneratively cooled thrust chambers (Appendix III). Also, neither experimental measurement nor analytical calculation of the pressure drop through the short test section could be made with reliable degree of accuracy. Therefore, it is recommended that the experimental data be extended to higher bulk temperatures employing longer test sections, and the pressure drop through the test section be measured and compared with the analytical values.

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* Isothermal h' (i.e. when $\Delta T_f = 0$) at $T_b = 40^\circ R$ is $2 \text{ Btu/in.}^2 \text{ sec} \times \frac{(1\text{in.})^{0.2}}{1\text{b/sec ft}^2}$

SYMBOLS

A	Heat-transfer area (in. ²)
D	Diameter (in.)
G	Mass velocity (lb/in. ² sec)
h	Film coefficient (Btu/in. ² sec °R)
h'	Generalized film coefficient $\text{Btu/in.}^2 \text{sec}^{\circ\text{R}} \times \frac{(\text{in.})^{0.2}}{(\text{lb/in.}^2 \text{sec})^{0.8}}$
ΔH	Enthalpy change (Btu/lb)
K _m	Mean thermal conductivity (Btu/in. sec °R)
P	Power (kw)
Q	Heat flow (Btu/sec)
Q/A	Heat flux (Btu/in. ² sec)
r _i	Inner radius of tube (0.08375 in.)
r _o	Outer radius of tube (0.09375 in.)
T _{wl}	Liquid-side wall temperature (°R)
T _b	Liquid bulk temperature (°R)
ΔT _{wall}	Wall temperature drop (°R)
W	Fluid flow rate (lb/sec)
ΔT _f	Film temperature drop (T _{wl} - T _L) °F

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Experimental Apparatus

HEAT FLUX VS FILM TEMPERATURE DROP (INLET PRESSURE 28-158 PSIA)

HEAT FLUX VS FILM TEMPERATURE DROP (INLET PRESSURE 191-213 PSIA)

GENERALIZED HEAT TRANSFER COEFFICIENT VS FILM
TEMPERATURE DROP (INLET PRESSURE 28-158 PSIA)

GENERALIZED HEAT TRANSFER COEFFICIENT VS FILM
TEMPERATURE DROP (INLET PRESSURE 191-213 PSIA)

TABLE 1

HEAT TRANSFER DATA FOR HYDROGEN

INLET PRESSURE (PSIA)	HEAT FLOW (BTU/IN ² SEC)	MASS FLOW RATE (#/IN ² SEC)	BULK TEMP. STATION #1 (°R)	WEIGHT % VAPOR STATION #1	ΔT _{FILM} STATION #1 (°R)	$h' \times 10^6$ STATION #1	BULK TEMP. STATION #2 (°R)	WEIGHT % VAPOR STATION #2	ΔT _{FILM} STATION #2 (°R)	$h' \times 10^6$ STATION #2
31	0.02	0.722	38.7	0	39	440	38.8	0	53	296
31	0.08	0.722	39.1	0	35	2080	39.3	0	54	1350
32	0.29	0.740	40.3	0	250	746	41.6	0	240	778
34	0.39	0.727	42.4	2	338	1050	42.4	7	326	1090
36	0.65	0.735	42.8	5	438	1319	42.8	16	415	1393
38	0.95	0.703	43.3	9	555	1582	43.3	28	527	1665
28	0.19	0.722	40.8	0	231	745	40.8	5	229	751
31	0.60	0.771	41.7	7	442	1167	-	-	-	-
35	1.37	0.590	42.6	27	739	1977	42.6	57	766	1908
37	1.81	0.545	43.0	38	832	2465	43.0	83	1011	1419
39	2.63	0.545	43.5	54	1291	2317	62.5	100	1442	1451
54	0.10	0.723	46.3	2	41	2184	46.3	4	170	711
55	0.16	0.714	46.5	3	197	754	46.5	7	219	711
56	0.28	0.695	46.6	8	264	989	46.6	14	286	917
47	0.04	0.814	44.8	0	43	703	45.0	0	41	740
45	0.07	0.822	44.7	0	36	1565	44.7	0	41	1360
58	0.12	0.800	46.9	0	21	4700	46.9	2	40	2390
58	0.21	0.800	46.9	2	149	1170	46.9	5	390	448
58	0.30	0.800	46.9	3	300	824	46.9	9	437	565
59	0.41	0.800	47.0	6	362	946	47.0	14	481	712
61	0.48	0.795	47.3	8	380	1070	47.3	17	505	806
62	0.58	0.795	47.4	10	413	1190	47.4	22	530	926
63	0.65	0.791	47.5	12	436	1260	47.5	25	547	1000
64	0.75	0.750	47.6	14	465	1420	47.6	32	571	1160
64	0.02	0.712	44.5	0	41	425	44.7	0	50	341
64	0.08	0.712	44.9	0	35	1990	45.2	0	48	1465
64	0.20	0.703	45.8	0	276	670	47.7	1	294	628
65	0.38	0.672	47.7	3	385	950	47.7	9	396	922
66	0.67	0.612	47.9	9	519	1342	47.9	22	491	1420
56	1.05	0.699	46.5	14	590	1665	46.5	37	535	1832
56	1.05	0.948	46.5	4	589	1306	46.5	13	535	1437
64	1.30	0.744	47.7	18	695	1650	47.7	41	639	1805
65	1.59	0.699	47.8	23	799	1904	47.8	59	733	2023
66	1.95	0.663	47.9	37	679	2790	47.9	79	721	2628
67	2.36	0.608	48.2	42	776	3160	53.4	100	852	2878
76	2.83	0.626	48.0	52	839	3427	62.3	100	937	3068
66	3.42	0.608	48.0	77	947	3755	81.7	100	1164	3550
68	4.13	0.567	50.2	100	1127	4032	113.3	100	1367	3325
72	0.06	1.434	44.8	0	-	-	45.3	0	52	606
72	0.17	1.425	45.4	0	57	1572	47.3	0	115	779
73	0.37	1.416	47.5	0	165	1188	49.0	0	289	678
74	0.63	1.411	49.2	0	341	982	49.3	8	446	751
83	0.01	1.380	45.0	0	28	193	45.3	0	57	95
83	0.04	1.366	45.3	0	29	752	45.5	0	54	404
92	0.07	0.889	47.0	0	35	1536	48.0	0	59	911

TABLE 1 (cont.)

HEAT TRANSFER DATA FOR HYDROGEN

INLET PRESSURE (PSIA)	HEAT FLUX (BTU/IN ² SEC)	MASS FLOW RATE (#/IN ² SEC)	BULK TEMP. STATION #1 (°R)	WEIGHT % VAPOR STATION #1	ΔT_{FILM} STATION #1 (°R)	$h_1 \times 10^6$ STATION #1	BULK TEMP. STATION #2 (°R)	WEIGHT % VAPOR STATION #2	ΔT_{FILM} STATION #2 (°R)	$h_2 \times 10^6$ STATION #2
95	0.02	0.739	49.5	0	4	5800	49.9	0	5	3960
92	0.08	0.739	50.4	0	3	26900	51.2	1	8	9000
96	0.19	0.739	50.7	0	233	0744	51.5	2	248	0700
98	0.55	0.740	51.7	5	288	998	51.7	16	487	999
98	0.74	0.740	51.7	9	557	1184	51.7	27	540	1221
98	0.82	0.740	51.8	10	594	1228	51.8	29	568	1285
98	0.88	0.740	51.8	12	612	1284	51.8	32	580	1354
99	0.08	0.754	49.7	0	1	52800	50.4	0	14	5050
96	0.20	0.754	50.2	0	245	724	51.6	0	269	657
95	0.67	0.749	51.4	7	526	1115	51.4	22	519	1130
96	0.72	0.749	51.5	15	678	936	51.5	38	629	1009
96	1.36	0.753	51.5	22	748	1601	51.5	52	571	2097
96	1.62	0.749	51.5	28	814	1750	51.5	64	709	2010
97	1.92	0.749	51.6	35	747	2337	51.6	78	803	2174
98	2.64	0.749	51.8	49	857	2709	55.4	100	795	2920
91	2.99	0.749	51.1	57	902	2920	61.1	100	875	3009
92	3.25	0.749	51.2	64	930	3080	67.0	100	933	3069
91	3.57	0.749	51.1	70	990	3180	72.5	100	1006	3130
94	4.28	0.749	51.4	88	1077	3507	87.8	100	639	5910
96	0.14	1.451	48.4	0	44	1655	49.4	0	51	1435
93	0.57	1.458	50.4	0	253	1170	51.3	4	267	1110
96	1.14	1.439	51.6	0	617	962	51.6	14	583	1018
95	1.42	1.439	51.5	3	702	1060	51.5	21	651	1143
95	1.91	1.443	51.5	10	800	1245	51.5	32	708	1406
96	2.63	1.443	51.6	18	750	1831	51.6	48	720	1908
95	2.96	1.443	51.5	22	768	2014	51.5	55	729	2121
99	0.20	1.332	49.1	0	379	2710	50.2	0	38	2720
100	1.00	1.307	52.0	2	558	1008	52.0	32	556	1011
95	1.93	1.321	51.5	19	755	1432	51.5	41	671	1612
96	3.32	1.316	51.6	35	838	2224	51.6	76	911	2045
98	3.96	1.316	51.7	43	916	2430	51.7	95	1001	2223
100	5.09	1.302	52.0	55	1110	2595	60.0	100	1161	2481
111	6.02	1.289	53.0	70	1238	2779	70.0	100	1178	2920
93	0.02	1.257	50.7	0	25	466	50.8	0	48	243
92	0.09	1.357	50.0	0	-	-	50.5	0	49	1009
94	0.20	1.366	50.5	0	41	2660	51.4	1	119	916
95	0.42	1.357	51.4	1	192	1200	51.4	8	270	854
96	0.74	1.357	51.6	5	392	1036	51.6	17	471	861
98	0.94	1.357	51.8	7	501	1029	51.8	22	550	938
103	0.05	0.585	52.0	0	35	1156	52.3	0	42	538
100	0.28	0.458	52.0	12	321	1139	52.0	26	362	1011
102	0.54	0.458	52.2	25	-	-	52.2	52	471	1497
101	0.63	0.649	52.1	16	-	-	52.1	39	488	1269
102	0.80	0.499	52.2	37	-	-	52.2	70	523	1869

TABLE 1 (cont.)

HEAT TRANSFER DATA FOR HYDROGEN

INLET PRESSURE (PSIA)	HEAT FLUX (BTU/IN ² SEC)	MASS FLOW RATE (#/IN ² SEC)	BULK TEMP. STATION #1 (°R)	WEIGHT % VAPOR STATION #1	ΔT_{FILM} STATION #1 (°R)	$h_1' \times 10^6$ STATION #1	BULK TEMP. STATION #2 (°R)	WEIGHT % VAPOR STATION #2	ΔT_{FILM} STATION #2 (°R)	$h_2' \times 10^6$ STATION #2
127	0.01	1.003	48.6	0	37	189	48.8	0	44	159
126	0.08	0.994	49.0	0	24	2337	49.3	0	43	1304
125	0.14	1.053	49.3	0	20	4702	49.5	0	42	2237
125	0.41	1.066	50.3	0	338	806	53.8	0	409	666
127	0.63	0.962	53.5	0	-	-	54.5	14	508	895
139	0.03	0.640	45.0	0	9	3333	46.2	0	130	231
140	0.12	0.640	46.7	0	44	2727	47.5	0	104	1154
138	0.26	0.644	50.5	0	176	1468	53.7	0	270	958
138	0.51	0.536	55.8	10	375	1565	55.8	32	407	1442
142	0.02	1.384	50.0	0	-	-	50.2	0	40	270
143	0.08	1.400	50.5	0	-	-	52.0	0	39	1097
145	0.19	1.389	51.0	0	-	-	52.1	0	136	753
147	0.36	1.375	52.2	0	-	-	54.0	0	194	1007
143	0.01	0.740	48.7	0	8	1113	48.8	0	46	193
144	0.08	0.744	49.2	0	17	4175	50.3	0	-	-
144	0.19	0.735	49.2	0	125	1360	52.0	0	273	623
148	0.37	0.731	52.0	0	287	1159	55.1	0	410	812
145	0.66	0.731	53.8	0	513	1157	56.5	13	582	1020
145	0.85	0.731	55.1	0	560	1365	56.5	31	591	1293
146	0.20	0.916	-	-	-	-	56.3	0	11	13400
147	0.31	0.916	56.0	0	74	3190	57.0	0	90	2620
145	0.08	0.630	49.9	0	27	2860	50.8	0	44	1770
145	0.20	0.635	51.4	0	174	1140	53.4	0	224	882
151	0.43	0.630	-	-	-	-	55.9	0	487	894
148	0.73	0.635	55.2	0	634	1154	56.8	15	634	1154
149	1.04	0.635	56.9	0	744	1411	56.9	43	687	1528
148	1.27	0.635	56.8	9	705	1819	56.8	62	751	1707
149	1.51	0.635	56.9	20	758	2004	56.9	80	809	1877
149	1.87	0.635	56.9	33	828	2275	57.5	100	746	2540
145	2.27	0.635	56.5	50	893	2552	61.5	100	806	2828
145	2.72	0.635	56.4	67	964	2840	70.5	100	872	3139
144	3.05	0.635	56.4	77	1017	3036	77.0	100	914	3357
144	3.19	0.635	56.4	84	1043	3078	-	-	-	-
145	3.41	0.635	56.4	92	1084	3169	-	-	-	-
142	3.74	0.635	57.1	100	1133	3324	-	-	-	-
143	4.14	0.635	59.0	100	1200	3470	-	-	-	-
149	1.01	1.243	55.4	0	648	920	-	-	-	-
150	1.97	1.203	57.0	9	704	1690	-	-	-	-
151	2.77	1.189	52.1	27	742	2276	-	-	-	-
150	3.42	1.184	57.0	43	855	2441	-	-	-	-
151	3.52	1.162	57.1	49	949	2303	-	-	-	-
152	5.00	1.153	57.2	81	1030	3025	-	-	-	-
158	0.02	1.325	42.0	0	40	280	42.2	0	46	243
158	0.09	0.871	42.2	0	37	1900	43.5	0	45	1562
158	0.21	0.853	45.0	0	44	3790	47.2	0	73	2284
158	0.41	0.699	49.3	0	223	1711	53.8	0	294	1298

TABLE 1 (cont.)

HEAT TRANSFER DATA FOR HYDROGEN

INLET PRESSURE (PSIA)	HEAT FLOW (BTU/IN ² SEC)	MASS FLOW RATE (#/IN ² SEC)	BULK TEMP. STATION #1 (°R)	WEIGHT % VAPOR STATION #1	ΔT_{FILM} STATION #1 (°R)	$h_f \times 10^8$ STATION #1	BULK TEMP. STATION #2 (°R)	WEIGHT % VAPOR STATION #2	ΔT_{FILM} STATION #2 (°R)	$h_2 \times 10^8$ STATION #2
191	0.13	1.044	57.0	-	23	3650	57.2	-	32	2630
192	0.63	1.090	57.6	-	79	5180	58.8	-	15	26800
184	1.29	1.180	58.4	0	465	1681	59.5	20	629	1254
199	0.29	0.907	-	-	-	-	58.2	-	40	5480
199	0.40	0.907	57.8	-	25	11950	58.7	-	37	8100
199	0.46	0.907	57.9	-	56	6250	58.9	-	71	4880
196	0.01	0.982	-	-	-	-	56.3	-	37	252 ?
196	0.05	0.977	-	-	-	-	56.7	-	39	842 ?
196	0.09	0.982	-	-	-	-	56.9	-	41	1610
196	0.16	0.959	-	-	-	-	57.1	-	40	2810
196	0.24	0.950	-	-	-	-	58.4	-	42	4280
196	0.33	0.977	-	-	-	-	58.6	-	46	5100
196	0.39	0.977	-	-	-	-	58.8	-	50	5580
196	0.44	0.959	-	-	-	-	58.9	-	63	5030
196	0.55	0.959	58.3	-	41	9800 ?	59.2	-	93	4300
196	0.78	0.959	58.8	-	180	4320 ?	59.5	-	242	2320
196	0.92	0.950	59.9	-	274	2460 ?	59.8	-	319	2110
205	0.02	0.959	56.4	-	22	592 ?	56.5	-	38	354 ?
208	0.04	0.950	-	-	-	-	56.7	-	41	795 ?
210	0.08	0.950	-	-	-	-	56.9	-	45	1290 ?
211	0.35	0.967	-	-	-	-	58.7	-	52	4760
213	0.48	0.973	58.2	-	45	7600	59.0	-	80	4240
213	0.58	0.967	58.5	-	54	7730	59.4	-	113	3680
200	0.01	1.026	54.8	-	7	977 ?	54.9	-	43	159 ?
199	0.03	1.007	54.9	-	19	1601 ?	55.0	-	48	434 ?
203	0.06	0.835	58.4	-	8	6301 ?	59.0	-	43	1122 ?
202	0.01	0.989	55.4	-	-	-	55.4	-	47	16 ?
202	0.01	0.989	55.5	-	-	-	55.6	-	43	164 ?
208	0.05	0.985	55.8	-	20	1778	56.1	-	46	773 ?

APPENDIX I

EXPERIMENTAL APPARATUS

1. Cryogenic Tank

The portion of the cryogenic tank, as shown in Figures 1 and 2 of this appendix, consists of a stainless steel tank (to contain the fluid under test), and a vacuum enclosure around the tank to minimize heat flow from the surroundings. The fluid tank (Figure 2, Item 1), fabricated by welding three stainless steel G-1 oxygen tanks end to end, gives a volume of approximately 3 cu ft, and a working pressure of 400 psig. The tank was wrapped with a smooth covering of aluminum foil to reduce heat gain by radiation.

The tank enclosure (Figure 2) consists of two cylindrical shells, one located inside the other. The original plan was to fill the outer annular volume with an insulating powder, but in practice, it was found impractical to attain a good vacuum in the presence of the powder because of excessive moisture and adsorbed gasses. The powder was therefore eliminated from the system. The inner cylinder (Figure 2, Item 2) which is made of copper, is attached to a liquid nitrogen reservoir on the upper surface of the enclosure, enabling it to function as a thermal radiation shield. In practice, the tests were conducted within a few minutes after completion of the fluid loading operations, thereby not requiring the use of the liquid-nitrogen shield. Thus, in practice, these cylinders were used effectively as radiation shields.

For successful operation of the vacuum enclosure, it was found necessary to use a mass spectrometer type of leak detector. Another item worth mentioning is the use of epoxy resin as a sealant for small leaks and porous welds. This resin adheres to most materials, withstands considerable temperature cycling, and, when "set," has a negligible vapor pressure. The leak rate for this cryogenic tank was found to be approximately 60 micron-ft³/day. The vacuum within the insulation shell was controlled between 15 to 30 microns at the room temperature, which resulted in a vacuum of less than 1 micron at the operating temperature. The fluid loss rate was estimated to be less than 4% per day. This was satisfactory in view of the short-term storage required of the system. After initial checkout of the system, no maintenance problems were encountered.

2. Test Unit Enclosure

A view of this section of the system is shown in Figure 2, Item 3, and Figure 3. The test unit is close-coupled to the top of the cryogenic tank, and is within a vacuum enclosure which is separated from that of the fluid tank. This enclosure functions to reduce convective heat gains to the test element which would be significant if exposed to ambient conditions. Components located within the enclosure are: (a) the test section valve (Figure 2, Item 4) which isolates, by remote actuation, the test fluid supply from the test section; and (b) the test section (Figure 2, Item 5 and 6) which comprises the test element, the fluid approach section, the downstream mixing section, the power leads, and transducers that measure the inlet, and outlet temperature of the fluid and wall surface temperatures. Checkout tests, utilizing liquid nitrogen as a working fluid, showed that heat flux adjacent to the contact area of the copper electrodes and heater tube, caused by inherent heat capacity of the copper electrodes and power leads, resulted in film boiling in these localized areas during the initial period of the test. To minimize this effect, the electrodes were redesigned, using a conductor with a low thermal conductivity (stainless steel). In addition, the test procedure was modified so that small amounts of test fluid were flushed through the system prior to the test to pre-chill the entire test unit. A sketch and photo of the test section and the 3/16-in. OD and 0.010 in.-thick wall, test element is shown in Figures 4 and 5, respectively.

3. Valves and Regulators

The operating temperature required for this program was considerably below those warranted by manufacturers of commercially available valves and regulators. Since the use of commercially available components would be the most expedient and economical method of providing the necessary flow control, it was decided that, with the exception of the test section valve, the commercially available components were suitable. To prevent extensive damage to the system in the event of a failure in the test enclosure, it was necessary to fabricate the test section valve to assure such design qualities as reliability, low thermal capacity, and high impedance to conductive heat flow. A sketch and a photo of this valve are shown in Figures 6 and 7, respectively. In operation, this valve exceeded all requirements, no significant maintenance or adjustments were necessary after numerous actuations.

A regulator relief valve, Grove Mity Mite Model 94 RR (Figure 2, Item 7), was used to control pressure in the cryogenic tank. This valve has the capability of pressurizing the tank to a controlled magnitude, and also to vent off the excess volume of gas if the vapor pressure of the test fluid exceeds the control pressure. A Hydromatics cryogenic ball-type valve (Figure 2, Item 8) was used as an emergency pressure relief valve on the tank, for any situation when the desired vent rate might exceed the vent rate of the Mity Mite regulator. A Grove PB-300 with a 1/4-in. orifice (Figure 2, Item 9), was used as a back-pressure regulator to control the test-section pressure. This regulator consists of a steel housing which contains the major working parts - the disc and seat, the diaphragm, and the lever arm. Movement of the diaphragm is

transmitted through a push rod to the lever arm, which, in turn, controls the movement of the disc in relation to the seat. The push rod is sealed by a combination of Teflon V-rings and O-rings, so that the inner diaphragm chamber is a closed unit. The push-rod side of the diaphragm is loaded to the desired back pressure with nitrogen through a hand regulator located in the control room, while the other side of the diaphragm is exposed to the test-fluid pressure. The valve disc seeks a position such as to balance the pressures on each side of the diaphragm. The pressure sensing line was connected directly to the inlet of the test section, thereby automatically allowing the regulator to compensate for the pressure loss between the test section and the back-pressure regulator. System checkout runs revealed unstable pressure oscillations within the test section, which resulted from surges of vaporization of test fluid in the pressure sensing line to the back-pressure regulator. Pressure stability was restored by placing an orifice in the pressure sensing connection.

Since the Mity Mite vent valve and the back pressure valve are components of the vent system, heat gain to these components was inconsequential to the conduct of the tests. These components were provided with water sprays in order to maintain their operating temperatures closer to rated values and improve operational reliability. These components functioned throughout the program without failure.

The initial choice of the cryogenic tank fill valve (Figure 2, Item 10) was a packless diaphragm type, encased in a vacuum jacket. Considerable difficulty was encountered with this component because the least amount of moisture freezing on the inner surface of the valve bound the moving surfaces so that valve operation was impossible. This valve was replaced with a 1-in., Powell, globe-type valve with a long stem (built for liquid oxygen service), and 3000 psi working pressure. This valve was encased in foam plastic insulation. Although the thermal capacity of this component was more than desired, it rendered trouble-free service for the rest of the program.

4. Venturi

A cavitating venturi (Figure 2, Item 11) was used to measure and control the flow. The venturi was located in the lower portion of the cryogenic tank to provide a known isothermal environment. The design parameters considered were: flow rate, downstream recovery of upstream pressure, vapor pressure of test fluid, and thermal expansion of venturi material. Four venturis were chosen to cover 20 different flow conditions; two of the venturis were so designed as to cover velocities of 50 ft/sec at the low test pressures, as well as velocities of 25 ft/sec at higher test pressures. All venturis were equipped with throat pressure taps to ascertain whether cavitation was occurring. The throat pressure was used as a backup indication of fluid temperature. Flow characteristics of the venturis were mathematically corrected to 27°K, which amounted to approximately 1-1/2% difference from ambient conditions.

5. Hydrogen Detection System

A thermal conductivity cell, Gow-Mac Model R00K, was used to detect any hydrogen leakage. The cell was calibrated with nitrogen-helium mixtures, simulating air-hydrogen concentrations up to 10% of hydrogen. The test area was monitored by aspirating a sample of gas adjacent to the apparatus through the cell. Clean fresh air was used through the standard leg of the cell. The bridge output was placed on a recording potentiometer equipped with an alarm switch which actuated an audio alarm when a 2% hydrogen concentration was detected. The system operated reliably, and no material leakage problems were encountered.

STORAGE AND TEST
SECTION ASSEMBLY

ITEM NO	COMPONENT
1	FLUID TANK
2	RADIATION SHIELD
3	TEST ENCLOSURE
4	TEST SECTION VALVE TEFLON PLUS TYPE, REMOTE
5	TEST SECTION
6	INSTRUMENTATION & POWER LEADS
7	GROVE MIGHTYMITE PRESSURE CONTROL & RELIEF REGULATOR MODEL 94RR
8	HYDROMATIC CRYOGENIC VENT VALVE, REMOTE
9	GROVE PRESSURE BUSTER BACK PRESSURE REGULATOR MODEL PB300
10	POWELL GLOBE VALVE HAND TYPE
11	CAVITATING VENTURI
12	SUPERIOR PACKLESS VALVE HAND TYPE

SCHEMATIC OF TEST APPARATUS

558-1154

Test Section and Related Parts

SECTIONAL VIEW OF TUBULAR TEST SECTION

958-1136

Test Element

25

Figure 5
Appendix I

458-1216

Cryogenic Valve

Figure 7
Appendix I

APPENDIX II

MEASUREMENT OF TEMPERATURE

1. Thermometers and Thermocouples

The problems associated with accurate measurement of temperature become more severe when one attempts to measure small differences of temperature in the region of liquid hydrogen. These problems are mainly due to inadequate development of techniques of measuring low temperatures, since the development of low-temperature thermometry did not keep pace with the rapid growth of space technology, which requires the use of cryogenic fluids. Therefore, the choice of adapting techniques of measuring low temperature is limited. Both the platinum resistance thermometer and copper-constantan thermocouples have been used successfully with cryogenic fluids.

Although it was recognized that platinum resistance thermometers, with a hand-balanced Mueller bridge or copper-constantan thermocouples with a Leeds & Northrup Type K potentiometer and a sensitive galvanometer, would provide the highest degree of precision of measuring temperature, these systems could not be used because of the short duration of experiments necessitating continuous recording of temperature.

Since the sensitivity of the thermocouples decreases as the temperature is lowered (e.g., the sensitivity of the copper-constantan thermocouple is only $5 \mu\text{v}/^\circ$ at 20°K , compared with its value of $40 \mu\text{v}/^\circ$ at "room temperature"), it becomes increasingly difficult to record small differences of temperature. Experience with d-c amplifiers shows that the commercial units are suitable only under ideal conditions (e.g., no noise and stray fields, etc.). Therefore, it was decided to record the temperatures with platinum resistance thermometers.

The measurement of the absolute temperature of the fluid did not seem to present any special problems. But the recording of small differences in temperature demanded greater accuracy than the bulk-temperature measurement. To improve the accuracy of measurement, it was decided to use two resistance thermometers differentially rather than to use two thermometers to measure the inlet- and the outlet-temperature of the fluid. Also a visual observation of temperature rise from a single recording for controlling the power input to the test section during the course of the experiment was desirable.

For simplicity, only two resistance thermometers were used to measure the inlet bulk temperature and the temperature rise of the fluid. Figure 1(a) of this appendix shows the conventional bridge circuit of recording the temperature and the temperature differences. T_{r1} measures the inlet bulk temperature of the fluid, and T_{r1} in combination with T_{r2} measures the temperature rise of the fluid.

Unlike conventional thermocouples, it is not practical to make two resistance thermometers having the same temperature-resistance characteristics. Therefore, special calibration techniques with modification of the bridge circuits had to be developed to record small temperature differences. Figure 1(b) shows the modified bridge circuit of measuring differential temperature. The function of r_1 is to equalize the resistance of two thermometers at a chosen temperature (assuming $T_{r1} < T_{r2}$) and r_2 serves to equalize the slopes of the resistance-temperature relationship curves of the two thermometers, assuming the resistance-temperature relationship to be linear.

2. Calibration

The calibration of the recording potentiometer (thermometer) which measures the absolute temperature of the fluid was carried out in a routine manner using a standard bridge circuit and substituting a decade box in place of the thermometer. From the calibration data of the thermometer, the decade box could be adjusted to correspond to any temperature. To calibrate the differential thermometer, T_{r1} and T_{r2} were substituted by decade boxes (R_1 and R_2). At any temperature (e.g., at -400°F), the decade boxes were adjusted to correspond to the resistance of the thermometers. The compensating resistors r_1 and r_2 were so chosen that the output of the bridge circuit was zero. Keeping R_1 fixed to correspond to -400°F , R_2 was adjusted to correspond to -375°F , and the output of the circuit was noted. Similarly, calibration was carried out for different temperature intervals and at different temperatures. A typical calibration curve is presented in Figure 2. From the resistance reading of T_{r1} (R_1), the inlet temperature of the fluid (t_1) is obtained from the temperature-resistance relationship of the transducer. From the readings of the differential bridge and T_{r1} , the differential resistance ΔR , between T_{r1} and T_{r2} was obtained from Figure 2. Therefore, the resistance of T_{r2} is equal to $R_1 + \Delta R$. The outlet temperature (t_2) of the fluid was obtained from the temperature-resistance relationship of T_{r2} . Thus $t_2 - t_1$ gives the temperature rise of the fluid.

Two types of temperature transducers (manufactured by Transonics, Inc., Burlington, Mass.) were used. The tape-on or cement-on types, Figure 3, were used to record the wall temperature of the test section, and the open-grid type (having low thermal-time lag), Figure 4, was used to record the bulk temperature and the temperature rise of the fluid.

The calibration data supplied by the manufacturer did not cover the temperature range of application. Therefore, the resistance-temperature relationship had to be extrapolated. The following equation was used to extrapolate the data to lower temperature.*

$$R_T = R_{T_1} - \frac{R_{T_1} - R_{T_2}}{R_{L6_{T_1}} - R_{L6_{T_2}}} (R_{L6_{T_1}} - R_{L6_{T_2}})$$

where

R_T, R_{T_1}, R_{T_2} = Resistance of the transducer at temperatures T, T_1 and T_2 , respectively

$R_{L6}, R_{L6_{T_1}}, R_{L6_{T_2}}$ = Resistance of the NBS thermometer No. L6 at temperatures T, T_1 and T_2 , respectively.

From the calibration data of any transducer at temperatures T_1 and T_2 and the values of L6 at T, T_1 and T_2 , the resistance at any other temperature, T , can be predicted from the above equation.

However, the extrapolated temperature-resistance relationship, based on the above equation, gave too high temperatures when compared with the values determined from the pressure readings of the cavitating venturi. Therefore, the extrapolation procedure of Hoge and Brickwedde** was followed to extrapolate the calibration data of the transducers to temperatures up to 20°R as explained below:

The extrapolation procedure is the same as that used in extrapolating and interpolating the limited calibration data of conventional thermocouples*** when a standard temperature-emf table is available except that

$$\left(\frac{R}{R_0} \right)_{L6} - \left(\frac{R}{R_0} \right)_{T_r}$$

was taken as the deviation, and the data for L6 was taken as the standard.

* R. J. Corruccine, NBS Engineering Laboratory Notes, Project 8131, File 57-25, 11 December 1951.

** H. J. Hoge and F. G. Brickwedde, "Establishment of a Temperature Scale for the Calibration of Thermometers Between 14° and 83°K." J. Research Nat. Bur. Standards, 22, 351-373 (March 1939).

*** Temperature, Its Measurement and Control in Science and Industry, Reinhold Publishing Corp., N.Y., Vol. 1, 1941.

The temperature-emf data for the Chromel-Alumel thermocouples were available only up to -300°F . Therefore, the Chromel-Alumel thermocouples were calibrated at dry ice acetone temperature, and at the normal boiling points of nitrogen and hydrogen to extend the data to low temperatures.

Figure 1a.

WHEATSTONE BRIDGE CIRCUIT FOR
MEASURING TEMPERATURE AND TEMPERATURE DIFFERENCES

Figure 1b.

Calibration Certificate

TRANS-SONICS*

Temperature Transducer

Type 1371A Serial No. 5Range -300 to -110°F

Temperature Degrees F	Resistance Ohms
-300	71.4
-250	100.3
-200	128.6
-150	156.4
-110	178.4
32° (ice point)	255.1

Calibrated by L.M. ChetaniDate SEPTEMBER 20, 1957

TRANS-SONICS, INC. HAS EXPENDED THE
UTMOST IN ENGINEERING DESIGN AND MANUFACTURING SKILL
IN THE PRODUCTION AND TESTING OF THIS TRANSDUCER. IF
THIS INSTRUMENT DOES NOT PERFORM PRECISELY AS IT SHOULD
OR IF YOU HAVE QUESTIONS ABOUT ITS PROPER USE, YOU WILL
FIND OUR CUSTOMER SERVICE DEPARTMENT READY TO GIVE
YOU COMPLETE AND IMMEDIATE ASSISTANCE.

TRANS-SONICS
INCORPORATED
BURLINGTON, MASSACHUSETTS

*Trade-mark

FORM 109926

958-1134

Tape-on-Type Temperature Transducer

TRANS-SONICS, INC. BURLINGTON, MASS.
TEMPERATURE TRANSDUCER
CALIBRATION

TYPE 1351 SERIAL 11 S.O. X-2997

RANGE -400 TO -110 °F
TEMPERATURE DEGREES F RESISTANCE OHMS

-400	8.7
-375	10.2
-350	30.5
-325	44.1
-300	58.0
-275	71.3
-250	85.5
-225	99.1
-200	112.5
-175	125.9
-150	139.2
-125	152.3
-110	160.2
32° ICE POINT	233.5

FORM 47 PREPARED BY L.M.C. DATE MARCH 3, 1958

Open-Grid-Type Temperature Transducer

APPENDIX III

HEAT TRANSFER ANALYSIS IN A THRUST CHAMBER

To illustrate the usefulness of the experimental data for design of regeneratively cooled, low-pressure thrust chambers, heat-transfer analysis in a thrust chamber was carried out based on the following specifications:

Thrust   10,000 lb

Propellant $\text{LiqO}_2 - \text{LiqH}_2$

M. R. = 5

$P_e = P_a$

$A_c/A_t = 3$

$A_e/A_t = 10$

$L^* = 20''$

$\alpha = 15^\circ$ = half angle of divergent nozzle section

$\beta = 30$ = half angle of convergent nozzle section

Based on the above conditions, the typical dimensions of the thrust chambers and the nozzle are shown in Figure 1 of this appendix.

The following assumptions were made in calculating the wall temperature and heat fluxes at various locations of the rocket:

1. The actual characteristic velocity, C^* , for the $\text{H}_2\text{-O}_2$ system is equal to 95% of the theoretical C^* .

2. The actual flame temperature, T_c , is equal to $(0.95)^2$ of the theoretical T_c .

3. The heat flux through the coolant tubes is one dimensional.

4. The radiant heat flux is negligible.

5. There is no fouling of the heat transfer surface from impurities.

6. Coolant flows in single pass, entering at the nozzle end at 40°R and at 100 psia.

Wt flow rate of propellants = 28.29 lb/sec

Wt flow rate of hydrogen = 4.715 lb/sec

Mass velocity of propellant at throat = $\frac{0.226 \text{ lb}}{\text{sec in.}^2}$

Assume, wall temperature, T_w , at throat = 1180°R

gas film heat transfer coefficient:

$$\begin{aligned}(h_g)_t &= 0.0295 \frac{c_p}{Pr^{2/3}} \left(\frac{\mu}{D}\right)^{0.2} \left[\frac{2T_c G}{(1 + \frac{\gamma-1}{2} M^2) (T_r + T_w)} \right]^{0.8} \\&= \frac{0.0295 \times (0.861)}{0.8205} \left[\frac{0.1388}{3600 \times 12 \times 12.76} \right]^{0.2} \left[\frac{2 \times 4942 \times 0.226}{(4897 + 1180) (1 + \frac{1.24-1}{2})} \right]^{0.8} \\&= 0.0006059 \text{ Btu/in.}^2 \text{ sec}^\circ\text{R}\end{aligned}$$

$$\begin{aligned}(Q/A)_t &= h_{g_t} (T_r - T_w)_t = 0.0006059 \times (4897 - 1200) \\&= 2.25 \text{ Btu/in.}^2 \text{ sec}\end{aligned}$$

To check the validity of assuming $T_w = 1180^\circ\text{R}$:

From Figure 2 of text the experimental $\Delta T_{\text{film}} (T_{w_i} - T_b)$, corresponding to $Q/A = 2.25 \text{ Btu/in.}^2 \text{ sec} = 875^\circ\text{F}$

From heat balance, T_b at throat = 200°R

Temperature drop through the 0.01-in. stainless-steel tube wall = 97°F

Therefore $T_w = T_b + \Delta T_{\text{film}} + \Delta T_w = 1172^\circ\text{R}$

Thus, the calculated T_w agrees well with the assumed T_w . When the calculated T_w did not agree with the assumed T_w , the calculation was repeated as follows:

Assume a new T_w ; calculate Q/A based on the old gas film coefficient

Read ΔT_f from Figure 2 of text, corresponding to the new Q/A

Calculate T_w as before and repeat the iteration process till the calculated T_w agrees with the assumed T_w . When the calculated T_w and the assumed T_w are too far apart, make necessary corrections in computing the gas film coefficient.

Proceeding in the same manner as for the throat, the gas-film heat-transfer coefficient, chamber wall temperature, and heat flux at each station on the nozzle (Figure 1) were calculated. A summary of the calculated data is presented in Table 1. The total heat transferred to the chamber walls was calculated by graphically integrating the heat flux at the different axial locations of the nozzle.

Total heat rejected = 3300 Btu/sec

Enthalpy rise of fuel = 700 Btu/lb

From the enthalpy-temperature diagram, the outlet temperature of hydrogen was found to be equal to 234° R. Therefore, the coolant is completely vaporized.

Calculations on the pressure drop along the coolant passage shows that the total pressure drop due to friction and acceleration of the fluid is less than 35 psi. Assuming the pressure drop through the injector to be equal to 15 psia, the inlet pressure of fuel of 100 psia would be adequate to operate the chamber at 50 psia.

By comparing the experimental heat fluxes and the wall temperatures with the calculated values in a low-pressure, 10,000-lb thrust chamber, it can be seen that regenerative cooling by hydrogen is completely adequate.

SYMBOLS (APPENDIX III)

A	Cross sectional area (in. ²)
A_c/A_t	Nozzle area contraction ratio
A_e/A_t	Nozzle area expansion ratio
C^*	Characteristic velocity (ft/sec)
C_p	Specific heat (Btu/lb ^o R)
D	Diameter (in.)
G	Mass Velocity (lb/in. ² sec)
h_g	Gas film coefficient (Btu/in. ² sec ^o R)
L^*	Characteristic length (in.)
M	Mach number
M.R.	Mixture ratio
P	Pressure (psia)
P_r	Prandtl number
Q/A	Heat flux (Btu/in. ² sec)
T	Temperature (^o R)
α	Half angle of divergent nozzle section
β	Half angle of convergent nozzle section
γ	Ratio of the specific heats
μ	Viscosity of the combustion products (lb/hr ft)
Subscripts	
a	Ambient conditions
b	Conditions in the bulk fluid
c	Conditions inside chamber

SYMBOLS (APPENDIX III) (cont.)

Subscripts

e	Exit conditions
i	Conditions inside coolant passage
r	Conditions at the outer edge of boundary layer
t	Conditions at the throat
w	Conditions at the wall

TABLE III-1
HEAT TRANSFER DATA FOR A 10,000 LB
LOW-PRESSURE THRUST CHAMBER

<u>Station No.</u>	<u>Bulk Temperature of Fluid, °R</u>	<u>Hot-Side Wall Temperature, °R</u>	<u>Heat Flux Btu/in.² sec</u>
1	234	855	1.05
2	227	1005	1.50
3	200	1180	2.25
4	167	895	1.34
5	130	700	0.94
6	97	515	0.51
7	51	375	0.32
8	49	290	0.22

WELDED-TUBE CHAMBER AND NOZZLE