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AEC-NASA SPACE NUCLEAR PROPULSION OFFICE

PRESSURE DROP IN
A SINGLE U-TUBE TEST SECTION

NERVA PROGRAM

CONTRACT SNP-1

JANUARY 1967

MASTER

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ROCKET ENGINE OPERATIONS - NUCLEAR

JANUARY 1967

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REPORT NO. RN-S-0334

PRESSURE DROP IN
A SINGLE U-TUBE TEST SECTION

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ABSTRACT

An experimental pressure drop study of a single Technology Engine U-tube test section was conducted. The static pressure profile was determined along the modelled nozzle flow passage while flowing cryogenic hydrogen without heat addition, and gaseous hydrogen both with and without asymmetric heat addition. Those tests with heat addition were conducted by resistance-heating a copper cladding on the crown of the stainless steel test section. Defects in the cladding resulted in two of the test section burnouts; after the second repair the program was restricted to adiabatic flow tests with liquid hydrogen.

Experimental profiles were compared with those generated analytically in an attempt to better define the factors controlling pressure drop, with the ultimate objective of upgrading current computational procedures. Although the experimental static pressure profiles were not reproduced analytically, the test results indicate that pressure loss prediction can be improved by considering tube curvature and local surface roughnesses of both the tube and jacket.

PRESSURE DROP IN A SINGLE U-TUBE TEST SECTION

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I. INTRODUCTION

In the design of a convectively-cooled rocket nozzle, the prediction of the total coolant pressure drop and the bulk temperature rise is based on balancing the heat transferred from the hot gases to the gas-side of the nozzle wall with that absorbed by the coolant from the inner surface of the nozzle wall. Such calculations thus require an accurate estimate of the heat transfer coefficients on both the hot-gas and the liquid side of the nozzle wall. Of equal importance is the ability to predict the coolant pressure drop, which is a function of both the flow rate and the coolant passage geometry.

Prediction of the heat transfer coefficients has been accomplished by performing extensive scale-model testing, empirically correlating the test results, and judiciously applying scale factors and/or extrapolating these results to the full-scale hardware. The proof of design adequacy is then determined by how well the measured values for pressure drop and coolant-temperature rise from nozzle inlet outlet plena match the predicted values.

It was the purpose of this study to experimentally determine the flow characteristics of a single U-tube coolant passage, under conditions approaching (within the limitations of test equipment) the Technology Engine design parameters; this was accomplished by measurements of the static pressure profile both with and without heat addition to a fluid (either gaseous or cryogenic hydrogen). These experimental results were compared with various analytically-determined pressure drops in an attempt to assess current pressure drop computational procedures. Direct measurement of local coolant passage geometrical factors (roughness, flow area) coupled with known local heat transfer rates for tests with heat addition permitted partial isolation of these factors for evaluation of their effect on pressure drop.

This study therefore served to evaluate current analytical techniques used to characterize pressure drop in a U-tube nozzle flow channel by experimentally establishing the static pressure profiles for gaseous and liquid hydrogen flowing under isothermal conditions and with asymmetric heat addition.

II. SUMMARY

A series of tests was conducted using a single U-tube test section of the NERVA Technology Engine configuration designed to flow gaseous and cryogenic hydrogen both with and without heat addition. Asymmetric heat addition to the flow passage, representative of the heat transfer in a nozzle coolant channel, was attempted by resistance-heating a copper-cladding on the tube crown. However, deficiencies in this cladding resulted in two test section burnouts. After the first repair the section was restricted to heat transfer tests with gaseous hydrogen only; after the second burnout only adiabatic tests could be made.

The steady-state static pressure profile along the length of the U-tube was determined by measurements taken in 19 places using a high-speed sampling valve and a single pressure transducer. The test data were then compared with analytically-derived pressure profiles.

It was concluded that the pressure drop profile in a U-tube can be modeled mathematically from a knowledge of the geometry of the flow passage and the surface roughnesses of the tube walls and the jacket. For the specific test section employed, the pressure profile was significantly affected by the surface finish (i.e., roughness) of the jacket side of the coolant passage. The $It\bar{o}$ correlation for the effect of curvature on pressure drop appears to adequately predict the shape of the static pressure profile in the throat region. Test section curvature did not significantly affect heat transfer during the gaseous hydrogen flow tests.

III. TECHNICAL DISCUSSION

The problem of adequately predicting the heat transfer and pressure drop characteristics of a propellant flowing turbulently in the coolant passages of a convectively-cooled rocket nozzle cannot be solved rigorously by analytical techniques. Reliance must be placed on mathematical-models incorporating empirically-determined factors. The general design procedures and uncertainties included in these computations are discussed in Reference (1) while problems specific to the nozzle of the nuclear Technology Engine are discussed in Reference (2).

Evaluation of the considerable experimental data obtained from scale-model studies conducted to characterize the liquid and gas-side heat transfer coefficients over a wide range of operating characteristics have resulted in an adequate understanding of the nozzle heat transfer problem. However, changes in nozzle coolant passage geometries from the simple shapes studied in conventional single-tube coolant-side tests to more complex shapes plus the inability to control and/or know accurately the rate of local heat addition to the coolant in a nozzle test made it desirable to study a full-scale nozzle coolant passage under controlled laboratory conditions where flow and shape characteristics were separated from the effects induced by the heat addition.

Typically, the test section used to determine the heat transfer characteristics of hydrogen has been a single tube, generally straight, with heat generated uniformly both in the circumferential and longitudinal directions. Differences between such straight tubes and the actual coolant passage are generally geometric. In an effort to better simulate NERVA nozzle geometry, tests have been conducted in both straight and curved tubes of both circular and non-circular cross-section, but of constant flow area. These studies are reported in References (3) and (4). The principal differences remaining between the specimens used in these tests and full-scale nozzle tubes (in terms of heat transfer) is basically the non-uniform flow area and the variation in local heat input in the longitudinal direction typical in nozzle tubes.

The modeling for studies of pressure drop has not been so extensive. An experimental study was conducted at Rocketdyne, Reference (5), to determine the applicability of theoretical calculating procedures to prediction of pressure drop in curved coolant tubes. Isothermal tests of a coolant tube were made by measuring the pressure drop across the coolant passage, flowing helium at ambient temperature and at different flow rates. After the test the tube was sectioned, the surface roughness was measured, and the resulting data used in the analytical computation of pressure drop. Tests were conducted without heat addition, and the static pressures at the inlet and outlet of the test section were matched analytically.

In a convectively-cooled nozzle the friction factor for flowing hydrogen at supercritical pressures is subject to uncertainties in the temperature range between 40 and 100°R because of property changes alone. Further, in a rocket nozzle the heat gain comes primarily from the hot-gas side through the nozzle wall to the coolant. Since the other wetted surfaces are adiabatic, a large temperature drop occurs across the boundary layer on only one side of the coolant passage. This results in a large transport and physical properties variation which significantly influences the temperature and velocity profiles in the coolant passage. Since the friction factor is dependent upon the velocity profile, it is apparent that the friction factor for the turbulent flow of cryogenic hydrogen at super-critical pressures, with heat addition, can differ markedly from the factor typical of fluids flowing isothermally. This conclusion is supported by the work of Maurer and LeTourneau in Reference (6) where, in tests with water, it was found that the friction factor decreased with heat addition. Since the heat addition in these tests was uniform in the circumferential direction the effect of asymmetric heat addition - as in a nozzle coolant tube - cannot be deduced. It was concluded, however, that the concept of hydraulic equivalent diameter can be used with the standard friction factor curve for circular tubes to predict the frictional pressure drop in smooth, rectangular, tubes for Reynolds numbers ranging

from 4×10^3 to 5×10^5 . The equation for predicting the change in friction factor, with heat addition for both water and air flowing turbulently, is:

$$\frac{f_h}{f_{iso}} = \left[\frac{\mu_b}{\mu_w} \right]^{(-10 f_{iso})} \left[\frac{\rho_b}{\rho_w} \right]^{-0.5} \quad (1)$$

In the Rocketdyne study, Reference (5), it was found that passage curvature or change in cross-section contributed no appreciable increase in pressure drop. However, the work of H. Itō, Reference (7), shows that curvature influences the resistance for turbulent flow. Itō found that the influence of geometry could be coupled with the Reynolds number and, when compared with the corresponding resistance to flow in a straight tube, result in the following equation:

$$\frac{f_{\text{curved}}}{f_{\text{straight}}} = [\text{Re } (r/R)^2]^{1/20} \quad (2)$$

for values of $[\text{Re } (r/R)^2] > 6$

where:

Re = Reynolds number

r = tube radius

R = radius of curvature

In the scale-model tests reported in Reference (4), the effect of curvature on resistance-to-flow was noted experimentally. The heat transfer results of Miller in Reference (8) show an increase in heat transfer and would be indicative of an increase in flow resistance.

Despite the approaches noted, prediction of the pressure drop, as discussed in Reference (2), continues to be accomplished by either (1) assuming that the tube is smooth and using an empirically-determined multiplier to account for differences between the measured and calculated pressure drop,

or (2) using a form of the Colebrook equation* and the average roughness of the nozzle construction materials to account separately for one or more of the parameters otherwise lumped in the friction factor multiplier. With these techniques it is assumed that either the effects of curvature and heat addition upon the smooth tube friction factor can be included in a common factor, or that they are essentially negligible when the effect of surface roughness is included directly in the calculation of the friction factor. A further complication to pressure drop prediction in a U-tube flow channel is the variation in roughness between tube material and jacket material. The tube and jacket are fabricated by different methods, which may result in a wide variation in the surface roughness. In addition, local irregularities which may affect flow exist at the point of juncture between the jacket and tube.

It was thus the purpose of the test program to evaluate these factors and attempt to deduce their significance in contributing to local pressure drop.

*See Equation (7)

IV. DESIGN OF EXPERIMENT

A. SCOPE OF INVESTIGATION

The test program consisted of a total of twenty tests flowing either cryogenic hydrogen (without heat) or gaseous hydrogen (with heat), as shown in Table I.

The test section was fabricated using a stock Technology Engine nozzle U-tube as the base with a welded back-plate to close the channel. It included pressure taps on the back (or ambient temperature side), copper cladding on the crown (or heated side) and associated pressure tap and wall temperature thermocouples for the analysis of the heat transfer results. It was the intent of the program to simulate test conditions, insofar as possible, which would be typical for the nuclear Technology Engine. However, because of limitations in the experimental equipment, the tests were conducted at conditions below the design level, particularly in terms of heat addition.

The flow tests with cryogenic hydrogen were conducted at inlet static pressures ranging from 212 up to 440 psia, with flow rates ranging from 0.18 to 0.48 lb/sec. The flow tests with gaseous hydrogen were conducted at pressures ranging from 226 to 286 psia, at flow rates from 0.016 to 0.031 lb/sec. Heat addition rates as high as 17.4 Btu/sec in the gaseous flow tests resulted in the acquisition of test data under non-isothermal test conditions.

B. TEST SECTION DESIGN AND FABRICATION

A stock U-tube channel for the Technology Engine nozzle was selected as the basis for the test section. In this nozzle the U-tubes are brazed into a steel jacket; therefore, to form the U-tube test section a replacement for the steel jacket, or back plate, was required. This back plate was fashioned from thin stainless steel sheet to keep the electrical resistance high. The tube (P/N 1114610) was fabricated by first scribing and cutting a stainless steel

TABLE I
SUMMARY OF TEST PARAMETERS

<u>Parameter</u>	<u>Liquid Hydrogen</u>	<u>Gaseous Hydrogen</u>
Pressure Inlet, psia	212 - 441	226 - 286
Flow Rate, lb/sec	0.018 - 0.48	0.016 - 0.031
Inlet Temperature, °R	48.4 - 52.6	469 - 521
Heat Input, Btu/sec	-----	5.6 - 17.4
Outlet Temperature, °R	52.3 - 55.5	523 - 665

strip 0.014- to 0.016-in. thick to fit the open side of the U-tube channel (P/N 707217-9). This channel closure (or back plate) was positioned into the tube channel and spot welded at 0.06-in. intervals along the full length of the "U" section. After spot welding, the edges of the tube channel were fusion-welded to the channel closure using a heliarc torch. Special closures on each end of the tube completed the flow channel. The pressure tap holes were located, drilled and, after each hole was carefully deburred, a one-inch length of 0.125-in. OD tubing was welded over the pressure tap hole.

Copper was then electroformed over the crown portion of the tube to within 1.50 in. of the end of the channel on the inlet end and 1.00 in. on the outlet end. The electro-formed copper was between 0.030 and 0.045 in. thick. The copper coating was designed to serve as the principal electrical conductor, carrying approximately 90% of the current while the stainless steel carried the remaining 10% in the tests conducted with heat addition. The test section prior to plating is shown in Figure 1A and the tube with electrodes attached, as it appeared after completion of the pressure tests, is shown in Figure 1B.

After this basic tube assembly was fabricated into a test section, the copper cladding along the crown of the tube in the chamber region and half way down the convergence section was found to be porous. The back side of the tube developed leaks, apparently from local corrosion which developed after the copper plating process. The test section was modified by removing the last 16 in. of the test section (that is, the chamber end of the tube). The outlet of the test section was therefore moved from the position shown in Figure 1B to a point just downstream of the outlet electrode, as shown in Figure 2, which is a photograph of the modified test section with thermocouples and voltage taps installed.

A stiffening section added to the back side of the tube prevented the tube from deforming from internal pressure without affecting the heat transfer capabilities of the test section. The stiffener was a resin-impregnated glass strip cemented to the tube with a polyurethane resin and

Figure 1

Welded Test Section Before and After Electroforming with Copper

Figure 2

Instrumented Test Section

wrapped to the assembly with 0.015-in. diam. glass string. The first 21.5 in. and the last 4.5 in. of the tube were wrapped as shown in Figure 2. The geometry of the flow passage at two different locations, with and without the wrapping and stiffener, is shown in Figure 3. The wall temperature thermocouples were located on the middle of the crown surface of the copper and were insulated electrically by a 0.0005 in. thick layer of mica. These were held in place mechanically by a circumferential wrapping of glass string. The thermocouples were made from 40-gauge chromel-alumel thermocouple wire, with the hot junction formed by discharge welding. The voltage taps were discharge-welded to the tube and wrapped with glass string for mechanical support. The positions of the pressure taps, voltage taps and wall thermocouples are shown schematically in Figure 4. Dimensions are as tabulated in Table II.

C. EXPERIMENTAL APPARATUS

The experimental flow system, shown schematically in Figure 5, is a gaseous-hydrogen pressurized single-pass, blow-down loop, consisting of a 150 gal foam-insulated run vessel and a low-pressure test section enclosure containing a helium atmosphere at approximately 5 psi. A burst diaphragm with a design failure rating of 30 psi prevented this housing from overpressure in case of test section failure. Figure 6 is a view of the test stand with the test section enclosure raised. The fluid discharge from the run vessel passes through the flange on the top of the tank, through the test section, back through the flange, through a flow-control valve, and finally through a vent line to the atmosphere. The fluid is expelled by high pressure gas stored in the high pressure reservoir. The pressure at the test section inlet is maintained by a Grove dome-loaded regulator, and the flow rate by a motor-operated flow-control valve in the vent line. Flow rate of the coolant is measured by a sharp-edged orifice located upstream of the test-section inlet mixing station. Coolant temperatures and pressures are sensed in mixing stations at the inlet and outlet of the test section (which include plenums to permit measurement of static pressure and temperature). Direct-current heating of the test section was supplied by four 15-volt, 3300 ampere dc rectifiers.

A FLOW PASSAGE 29.4 INCHES FROM INLET (5X)

B FLOW PASSAGE AT 9.8 INCHES FROM INLET (5X)

Figure 3

Cross-Sectional Views of Test Section After Testing

- ▮ PRESSURE TAP P-1 THROUGH P-21
- ┘ THERMOCOUPLES T_w -1 THROUGH T_w -7
- × VOLTAGE TAPS V-1 THROUGH V-5

Figure 4

Instrumentation Schematic

TABLE II
TEST SECTION
INSTRUMENTATION LOCATIONS

Pressure		Wall Temperature		Voltage	
Pressure Tap No.	Distance from Inlet in.	Wall Thermocouple No.	Distance from Inlet in.	Voltage Taps	Spacing in.
1	5.21	1	9.0	V-1 - V-2	14.00
2	21.406	2	19.0	V-2 - V-3	10.9
3	22.972	3	26.8	V-3 - V-4	4.85
4	24.972	4	29.9	V-4 - V-5	4.82
5	25.890	5	31.2	V-5 - V-6	6.47
6	26.940	6	36.1		
7	27.884	7	38.6		
8	28.401				
9	28.888				
10	29.495				
11	(Removed After Test HT-7-103)				
12	30.133				
13	30.602				
14	31.165				
15	31.671				
16	32.193				
17	33.168				
18	34.150				
19	35.162				
20	36.154

Figure 5

Simplified Flow Schematic

Figure 6

Cryogenic Heat Transfer Loop

The static pressures along the test section were monitored by a strain gauge pressure transducer, mounted in a scanning-type pressure sampling valve designed for measuring multiple pressures with one pressure transducer. The particular valve used, Scanivalve Model 48J, was designed for sampling up to 48 pressure taps at working pressures to 500 psi. The Scanivalve was connected to the test section by approximately five ft of 0.063-in. diam. nylon tubing. The sampling switch is close-coupled to the pressure transducer with a free volume of approximately 0.003 in.³. The scanning is achieved by a rotor driven by a 28 volt dc motor, resulting in a variable speed of rotation and sampling rate. In general, the sampling rate was approximately 0.2 sec per tap or 4 sec accumulated time-per-scan. The scanning rate was determined by monitoring, by means of an oscillograph, the position transmitter on the valve.

The measured test parameters (current, voltage, temperatures and pressures) were all digitized and recorded on tape at a sampling rate of 156 data bits per second. During the steady-state holds, data were summarized and averaged over a period of 10 to 12 seconds. At this sampling rate the pressure at each pressure tap was read approximately 12 times, and the pressure at each tap determined by averaging the middle 6 to 8 bits of data. One scan at any steady-state flow or flow and power condition constituted a test.

V. EXPERIMENTAL PROGRAM

A. TEST PROCEDURE

The test procedure consisted of passing gaseous or liquid coolant through the test section, either with or without heat addition. Flow rate was varied by opening the flow control valve and increasing the tank pressure. Once the system reached equilibrium, the conditions were maintained while data were recorded.

B. CALCULATION PROCEDURE

1. Gas Flow Tests Without Heat Addition

The tube cross-sectional flow area was measured after the completion of the tests by sectioning the tube at eighteen axial locations corresponding to the points selected as stations for analysis. The tube roughness on both the crown of the tube and on the base was measured in three axial positions: the straight inlet, the curved throat, and the straight exit region. These, along with the dimensions of the as-measured tube are tabulated in Table III. The computed flow area of the tube is plotted on Figure 7.

The analytical pressure profile along the tube was calculated by subtracting the cumulative computed station-to-station pressure drops from the test section inlet pressure. The pressure drop along the coolant passage is the sum of the velocity head and the frictional pressure drop.

$$\Delta P_t = \Delta P_v + \Delta P_f \quad (3)$$

where

$$\Delta P_v = \frac{\bar{\rho} (v_{n+1}^2 - v_n^2)}{288 g_c} \quad (4)$$

TABLE III
TEST SECTION GEOMETRY

<u>Station</u>	<u>X, in.</u>	<u>Y, in.</u>	<u>Developed Length, in.</u>	<u>U-Tube Wetted Perimeter P_t, in.</u>	<u>Total Wetted Perimeter P_{tot}, in.</u>
1	0.405	0.551	-0-	1.312	1.717
2	0.388	0.503	5.091	1.228	1.616
3	0.351	0.498	9.907	1.196	1.547
4	0.299	0.461	13.889	1.083	1.382
5	0.252	0.431	17.892	1.005	1.257
6	0.234	0.397	22.838	0.927	1.161
7	0.181	0.366	24.815	0.835	1.016
8	0.142	0.351	26.825	0.783	0.925
9	0.137	0.358	28.304	0.794	0.931
10	0.132	0.346	28.810	0.767	0.899
11	0.130	0.332	29.416	0.738	0.868
12	0.140	0.325	30.061	0.730	0.870
13	0.161	0.323	31.091	0.738	0.899
14	0.169	0.321	32.122	0.739	0.908
15	0.201	0.322	34.079	0.759	0.960
16	0.227	0.332	35.035	0.793	1.020
17	0.247	0.331	36.096	0.803	1.050
18	0.361	0.347	40.670	0.900	1.261

Figure 7

Measured Tube Flow Area

$$\Delta P_f = \frac{2 \bar{f} \bar{\rho} \bar{V}^2 (L_{n+1} - L_n)}{144 g_c D_e} \quad (5)$$

ΔP_t = static pressure drop, psi

ΔP_v = momentum pressure drop, psi

ΔP_f = frictional pressure drop, psi

$\bar{\rho}$ = average density, lbm/ft³

V = bulk velocity, ft/sec

g_c = conversion constant - 32.17 ft/lbm/lbf-sec²

L = axial length, in.

D_e = equivalent diameter, in.

Errors caused by averaging density or velocity are minimized by selecting stations sufficiently close together to minimize geometry changes, since the isothermal nature of certain of the tests produced negligible longitudinal density variations.

The friction factor "f", is necessary to the computation of the frictional pressure drop. For turbulent flow in a smooth tube the friction factor may be expressed (Reference 9) as a simple function of Reynolds number by:

$$\frac{1}{\sqrt{2f}} = 2.54 \log_{10} (Re) - 2.17 \quad (6)$$

where:

f = isothermal friction factor

Re = Reynolds number, $\left(\frac{D_e \rho V}{\mu} \right)$, where $Re > 10^5$
(in consistent units)

Since the frictional pressure drop calculated using this friction factor is representative of turbulent flow in a smooth tube, the resultant pressure drop is generally unrealistically low. It has generally been necessary to increase the magnitude of the predicted pressure drop in nozzle tests. This has been accomplished by the use of a multiplier factor, the magnitude of which was empirically determined for each test. This factor is a coefficient that accommodates departures of the tube from a smooth surface, and accounts for other losses such as those from flow around a curve and the possible effects of heat addition.

The friction factor can also be calculated by the Colebrook equation to include the effect of surface roughness. In this program, the friction factor was calculated using a simplified form of the Colebrook equation:

$$f = 0.001375 \left[1 + \left(2 \times 10^4 \frac{e}{D_e} + \frac{10^6}{Re} \right)^{1/3} \right] \quad (7)$$

where:

e = absolute surface roughness, in.

D_e = tube equivalent diameter, in.

This form of the equation agrees with the Colebrook equation within $\pm 5\%$ for a turbulent flow over a Reynolds number range from 4000 to 10^7 and surface roughness-to-diameter ratios below 0.05.

2. Gas Flow Tests with Heat Addition

The method of calculating the static pressure profile for the tests without heat addition was the same as that used for the tests with heat addition; however, the bulk temperature computation was altered by the

addition of heat to the fluid. The electrical heat added to the fluid between each station was computed based on the measured voltage gradient and the total current. In this calculation a linear variation in voltage between taps was assumed although in actuality the resistance variances are a result of both geometry and temperature effects. The thermal gradient in the radial and longitudinal direction was assumed to be zero; and the temperature drop through the wall was calculated using the equation:

$$T_{n+1} = T_n - (T_{n-1} - T_n) \left[\frac{\Delta r}{r_n} + \frac{(k_m)_{n-1}}{(k_m)_n} \right] - \frac{C (\Delta r)^2 (\Delta E/L)^2}{(\rho_m)_n (k_m)_n} \quad (8)$$

where:

$$\begin{aligned} k_m &= \text{thermal conductivity of the metal, Btu/in.-sec-}^\circ\text{R} \\ \rho_n &= \text{resistivity of the metal, } \Omega \cdot \text{in.} \times 10^{-6} \\ r &= \text{tube radius, in.} \\ \Delta E/L &= \text{voltage gradient, volts/in.} \\ C &= 0.9486 \times 10^{-3}, \frac{\text{Btu/sec}}{\text{watt}} \end{aligned}$$

The heat flux at the inner face of the tube was based on the computed temperature gradient in the tube, or:

$$q/A = (k_m)_n \left(\frac{T_{n-1} - T_n}{\Delta r} \right) \quad (9)$$

3. Cryogenic Flow Without Heat Addition

The test data for the isothermal flow tests with cryogenic hydrogen were reduced in the same manner as the gaseous flow tests. The isothermal pressure drop was computed using both methods of calculating the frictional pressure drop; and the computed and measured pressure drops were compared by plotting as a function of axial locations.

VI. EXPERIMENTAL RESULTS

A. SUMMARY OF TEST PARAMETERS

In the first tests, cryogenic (i.e., $T \approx T_{\text{crit}}$) hydrogen was flowed at supercritical pressures in the single-tube test section fabricated from a Technology Engine U-tube. These tests were isothermal, with no attempt made to obtain data with heat addition. Twelve steady-state runs were conducted and an analytical comparison was made of six of these tests. This evaluation consisted of comparing, as a function of developed length, the experimental measurements of pressure along the flow passage with computed pressure profiles.

A second series of tests was conducted in which gaseous hydrogen was flowed at ambient temperatures through the same test section both with and without heat addition. Eight tests were conducted; however, because of similarity in flow rate and inlet temperatures and pressure, it was considered sufficient to select only three of these tests for critical analysis. Again, the method of analysis was to compare the experimental measurements with the results of the analytical computations as performed on the IBM 7094 computer using the cryogenic heat transfer program 8047 (as discussed in Section IV, B). The reduced data for these tests are included in Appendix A.

B. PRESSURE DROP

Two different computational techniques are currently in use for calculating the pressure drop for a convectively-cooled rocket nozzle employing the U-tube method of construction: (1) a friction factor multiplier, or " P_f " factor, which lumps together the non-ideal qualities of the flow passage, is estimated, or (2) analytical accommodation of departures of the coolant passage from a smooth, straight, unheated tube (i.e., roughness, curvature) is attempted. In practice, the P_f multiplier is determined by matching the calculated pressure drop from inlet plenum to outlet plenum with measurements from actual nozzle tests. Each nozzle fabricated could then be characterized

by an individually-determined P_f factor. The alternative procedure is to attempt to identify the significant departures of the coolant passage from a straight, smooth, circular, uniformly-heated tube and to accommodate these effects analytically. Again, a comparison of the measured pressure drop in flow tests with the calculated value is a measure of the adequacy of the analytical model.

In this study, one objective was to eliminate or minimize uncertainties in the flow geometry, surface roughness, and local heat addition rates by design or through measurement. Pressure-profiles therefore were established for a single U-tube test section (1) flowing cryogenic hydrogen without heat addition, and (2) flowing gaseous hydrogen both with and without heat addition. Inherent electrical limitations of the test section prevented tests with heat addition to a flow of cryogenic hydrogen and restricted the magnitude of the heat addition in the gaseous hydrogen tests.

The measured pressure profiles for the nine representative tests analyzed have been compared graphically with the analytically-determined pressure profiles, as shown by Figures 8 through 16*. The friction factors used for the calculated pressure profiles on these figures were based on the following approaches:

1. Using Equation (6) and a P_f factor of 1.0.
2. Using Equation (7) and measured tube wall roughnesses and assuming a smooth jacket. A curvature correction was calculated using Equation (2).
3. Computed as in the preceding discussion, including curvature effects but assuming a jacket roughness different from the tube.

*Note that the direction of the coolant flow as indicated is countercurrent to the hot gas flow in the nozzle.

TEST NO. HT-7-104-3	
W = 0.21 LB/SEC TEMP = 50.5 °R	REYNOLDS NO.
	INLET 8.8 x 10 ⁵
	THROAT 18.6 x 10 ⁵
	OUTLET 13.0 x 10 ⁵

Figure 8

Isothermal Pressure Profile for Cryogenic Flow,

Inlet Pressure 222 psi

TEST NO. HT-7-104-5		
FLOW RATE, 0.21 LB/SEC INLET TEMP. 51.7°R	REYNOLDS NO.	
	INLET	8.0 x 10 ⁵
	THROAT	16.8 x 10 ⁵
	OUTLET	11.8 x 10 ⁵

Figure 9

Isothermal Pressure Profile for Cryogenic Flow,

Inlet Pressure 242 psi

TEST NO. HT-7-104-6		
FLOW RATE, 0.20 LB/SEC		REYNOLDS NO.
INLET TEMPERATURE, 53.6°R		INLET 8.9×10^5
		THROAT 18.9×10^5
		OUTLET 13.2×10^5

Figure 10

Isothermal Pressure Profile for Cryogenic Flow,

Inlet Pressure 297 psi

HT-7-108-1		
INLET TEMP., 50.2°R FLOW RATE, 0.34 LB/SEC	REYNOLDS NO	
	INLET	12.5×10^5
	THROAT	25.6×10^5
	OUTLET	17.7×10^5

Figure 11

Isothermal Pressure Profile for Cryogenic Flow,

Inlet Pressure 315 psi

HT-7-108-4	
INLET TEMP., 49.1°R FLOW RATE, 0.31 LB/SEC	REYNOLDS NO
	INLET 11.6 x 10 ⁵
	THROAT 23.6 x 10 ⁵
	OUTLET 14.0 x 10 ⁵

Figure 12
Isothermal Pressure Profile for Cryogenic Flow,
Inlet Pressure 422 psi

HT-7-108-7		REYNOLDS NO.	
INLET TEMP. 53.7 °R		INLET	21.0×10^5
FLOW RATE 0.48 LB/SEC		THROAT	44.8×10^5
		OUTLET	30.7×10^5

Figure 13

Isothermal Pressure Profile for Cryogenic Flow,

Inlet Pressure 419 psi

HT-7-105-7	
INLET TEMP. 521°R FLOW RATE 0.031 LB/SEC	REYNOLDS NO.
	INLET 1.46×10^5
	THROAT 2.96×10^5
	OUTLET 2.01×10^5

Figure 14

Isothermal Pressure Profile for Gaseous Hydrogen

HT-7-105-4	
INLET TEMP., 493.3°R FLOW RATE, 0.018 LB/SEC	REYNOLDS NO.
	INLET 0.88×10^5
	THROAT 1.48×10^5
	OUTLET 0.96×10^5

Figure 15

Pressure Profile for Flow of Gaseous Hydrogen
with Heat Addition

HT-7-105-9	
INLET TEMP., 511°R FLOW RATE 0.03 LB/SEC	REYNOLDS NO.
	INLET 1.45×10^5
	THROAT 2.60×10^5
	OUTLET 1.70×10^5

Figure 16

Pressure Profile for Flow of Gaseous Hydrogen
with Heat Addition

These figures show that the calculated profiles deviate from the measured profiles in nearly all cases. The measured static pressure is always less than that predicted using a P_f factor of 1.0. Increasing the P_f factor would never reconcile the differences between the measured and the predicted profiles but would decrease the difference between the end values for the experimental and predicted curves at the outlet of the test section; the magnitude of the P_f factor would be approximately 2.0. It is likewise apparent that the assumptions in approaches 2 and 3 do not bring the analytical profiles into convergence with the measured profiles. For the cryogenic hydrogen flow tests some roughness intermediate between the smooth and rough jacket values would appear to give the best prediction of pressure drop.

In each of the tests the measured profiles are different from those calculated. Before analyzing these variations it is pertinent to consider possible errors in the experimental profiles. These exist for flow area, flow rate, and pressure and temperature measurements. Flow areas were measured directly after the test by sectioning the tube, giving a high degree of confidence in these values. Heat balances were subject to errors of less than 4% and since electrical measurements are low in error, one may reasonably infer that flow rates were relatively accurate. For example, assuming that the pressure measurements are valid up to a developed distance of 28 in., one must assume an increase in flow rate of approximately 10% to give a reasonably good comparison between the measured and the calculated pressure profile curve; based on the measured tube roughness and assuming a smooth jacket. On this basis, then, the absolute magnitude of the pressure measurements would appear most subject to question. Since there is no way to directly evaluate any such error, its presence somewhat reduces confidence in the values shown in the measured profiles but does not obscure the conclusion that no reasonable error could create the difference shown between the measured and calculated profiles. It must be concluded then, that prediction techniques can only grossly delineate the actual result of the combination of conditions encountered.

1. Surface Roughness Effects

The surface roughness of the test section was measured in three places, using a Model 3 Talysurf. The as-measured centerline average roughness has been tabulated in Table IV. The roughness of the tube wall was measured on each side and on the crown for the straight sections of the tube between Stations 3-4 and 16-17. The crown of the tube was curved between Sections 12-13 and it can only be assumed that the sides are representative of the crown. The waviness of the back-side (jacket-side) of the tube precluded measurement using this device.

Since the surface of the tube wall and the jacket are different, the selection of the surface roughness to be used for computing the friction factor was calculated as follows, using a weighted average of the roughness based on the wetted perimeter of the tube and jacket:

$$e_{avg} = \frac{z_{tube}}{z_{total}} e_{tube} + \frac{z_{jacket}}{z_{total}} e_{jacket} \quad (10)$$

where z = wetted perimeter

In one case the jacket roughness was assumed to be the same as that of the tube; this is the "smooth-jacket" curve. However, for this test section, the penetration of the weld into the flow passage in the region of the pressure taps and at the corners between the tube wall and jacket, and a certain unassessed waviness of the back plate, lead to the assumption of a rough back-side. It was assumed that the roughness of the jacket was equivalent to a galvanized steel pipe. Using Equation (10), the roughness e_{avg} was calculated as tabulated in Table V. It is apparent that the assumption of a rough jacket contributes to the calculation of too large a pressure drop for the tests with cryogenic hydrogen where the Reynolds number is above 10^6 . It is not as significant for the gaseous hydrogen tests where $Re \leq 3 \times 10^5$.

TABLE IV
MEASURED TUBE ROUGHNESS

<u>Developed Distance, in.</u>	<u>Arithmetic Average Value, micro inches</u>		
	<u>Sides</u>	<u>Crown</u>	<u>Jacket</u>
14.0	6 - 62	32	*
30.0	10 - 10	--	*
36.0	10 - 38	10	*

*Back-side (jacket-side) too uneven to measure

TABLE V

AVERAGE FLOW PASSAGE SURFACE ROUGHNESS

<u>Developed Distance, in.</u>	<u>e_{avg.} μ-in.</u>
0	380
17.9	350
24.8	300
26.8	280
28.3	270
30.0	300
32.1	360
36.1	400

$$e_{avg} = \frac{Z_{tube}}{Z_{total}} e_{tube} + \frac{Z_{jacket}}{Z_{total}} e_{jacket}$$

where:

$$e_{jacket} = 1500 \mu\text{-in.}$$

$$e_{tube} = 60 \mu\text{-in.}$$

$$Z = \text{wetted perimeter}$$

2. Geometry Effects

Included in the friction factor multiplier is the effect of axial curvature of the coolant passage on the static pressure drop. In the calculations of the frictional pressure drop (which included surface roughness, as discussed previously) test section curvature effects were also included. The frictional drop as computed using Equation (5) was increased by the ratio $f_{\text{curved}}/f_{\text{straight}}$ as determined by Equation (2). For this test section, the value of the increase in resistance caused by curvature ranged from 1.25 to 1.53 for the gaseous and liquid hydrogen tests, depending on the Reynolds number.

The static pressure profiles (including the tube roughness effects) also include the effect of curvature between developed distances of 28.4 and 31.6 in., the DRAD, and URAD point of tangencies in the coolant passage (where DRAD and URAD are the downstream and upstream throat radii - referred to the gas flow direction).

3. Heat Addition Effects

The work of Maurer and LeTourneau (Reference (6)) indicates that the effect of heat addition to a flowing turbulent liquid is to reduce the pressure drop. In the conventional heated-tube tests, the heat is supplied uniformly in the circumferential direction and the effect is therefore present around the entire wetted perimeter. In the case of the U-tube tested in this program, the heat addition was restricted primarily to the crown of the tube, constituting only a fraction of the wetted perimeter. Considering that heat addition effectively reduces the isothermal friction factor, an analytical equivalent is a reduction in the surface roughness over that portion of the tube that is heated. For this particular test section, since the roughness of the back-side contributes most significantly to the pressure drop, the influence on pressure drop of heat addition through the

crown portion alone is believed to be insignificant. To use the non-isothermal friction factor multiplier in a rocket nozzle coolant passage calculation, it is suggested that a weighted perimeter approach such as the following:

$$f_{avg} = \frac{\text{heated perimeter}}{\text{total perimeter}} (f_h) + \frac{\text{unheated perimeter}}{\text{perimeter}} (f_{iso}) \quad (11)$$

be used to calculate the effective friction factor, where f_h is evaluated using Equation (1). Since the wall-to-bulk temperature ratio was relatively low for the gaseous hydrogen tests, the contribution of the non-isothermal effect to the pressure drop was small, and was not included in the computation of the static pressure profiles.

C. HEAT TRANSFER

The flow tests with gaseous hydrogen included tests with heat addition. The test data for tests HT-7-105-4 and -9 were reduced and the local heat transfer coefficients calculated at the points where the wall temperatures were measured. The calculated wall temperatures and local heat fluxes for these tests have been plotted as a function of developed length and are presented as Figures 17 and 18. In each of the tests both the local wall temperatures and heat fluxes increased as the throat section (curved portion of the tube) was approached and then decreased beyond the throat. Heat fluxes varying by a factor of 4:1 were calculated for these tests, with wall-to-bulk temperature ratios ranging from 1.2 to 2.3. The heat transfer coefficients for these data points have been compared with the equation of McCarthy and Wolf (Reference 10), and the results are plotted in Figure 19. The results show that the heat transfer in this test section can be correlated within $\pm 20\%$ by the equation:

$$Nu_b = 0.025 Re_b^{0.8} Pr_b^{0.4} (T_w/T_b)^{-0.55} \quad (12)$$

- INNER-WALL TEMPERATURE, °R
- △ LOCAL HEAT FLUX, BTU/IN² SEC

Figure 17

Wall Temperature and Heat Flux Profiles for

Test HT-7-105-4

Figure 18

Wall Temperature and Heat Flux Profiles for
Test HT-7-105-9

Figure 19

Heat Transfer to Gaseous Hydrogen in a U-Tube Test Section

The effect of curvature cannot be fully evaluated in these tests; however, it appears that the enhancement is not significant. This is somewhat inconsistent with the results reported by Hendricks and Simon in Reference (11) where an apparent enhancement was reported for gaseous hydrogen tests in curved tubes. If Reynolds analogy holds, the increase in resistance-to-flow of 25% based on Equation (2) should be accompanied by an increase in heat transfer of 25%. This effect was not apparent in these tests.

D. TEST SECTION DEFICIENCIES

A bimetallic test section design was selected for this test program. Previous experience with copper-clad stainless steel had indicated that copper could be electro-deposited on stainless steel tubes which could then be used as test sections, with the copper carrying in excess of 95% of the electric current. Due to the complexities of the coolant-passage geometry, the only feasible approach seemed to be to start with a stock formed U-tube, weld in a back plate to close the flow perimeter, and electroform copper on the crown of the tube. A thin back plate (jacket side) and unclad side walls were required to keep the electrical resistance reasonably high and therefore result in a low electrical current density in that section. The copper thickness was chosen so that approximately 90% of the current flowed in the copper and 10% in the steel.

The use of a thin jacket-side closure resulted in two immediate problems: (1) keeping the closure surface flat during welding and when internally pressurized, and (2) obtaining good static pressure taps. While inability to completely control these problems no doubt contributed to perturbations in the measured pressure profiles, the primary difficulty with the test section lay in the copper cladding. Two incidents of test section failure with heat addition were encountered in the test program. The first of these occurred during a checkout test while flowing cryogenic hydrogen; the test section burned out almost instantaneously, severing the tube at a developed distance of 29.5 in. from the inlet electrode. Figure 20 is a view

Figure 20

View of Test Section After Burnout

of the installation following burnout. No other damage to the system was incurred by the failure which occurred at a current flow of approximately 2000 amps and no measurable (with test console instrumentation) voltage drop, indicating a very low coefficient of resistance in the test section. Post-test examination of the cross-section at the point of failure revealed a separation in the copper cladding in the region of the crown. This is evident in Figure 21, which is a microphotograph of the tube presented in Figure 3A. It is surmised that this type of separation in the copper resulted in poor heat transfer, local overheating, and ultimate burnout.

The second incident of test-section failure occurred during Test HT-7-105, a gaseous hydrogen test. In this test, failure occurred away from the minimum-cross-section maximum-flux region of the tube at a developed distance of 22 in. from the inlet electrode. In this, as in the previous case, it appeared that a separation in the copper resulted in local overheating of the copper and ultimate melting. Once the copper is destroyed the stainless steel cannot accommodate the high current and it likewise fails by overheating.

The test section was repaired in each case by forming a copper block to fit over the severed section and torch brazing the two ends of the break into the copper block. View B of Figure 21 is a microphotograph of a section of the tube adjacent to the tube crown. In this view the separation within the copper has not developed, and apparently the thermal resistance was not as high as in the crown region.

The extremely low electrical resistance of the test section at cryogenic temperatures is predictable for pure copper and can only be overcome in a test such as this by making certain that the copper is in good thermal contact with the steel at all points, starting the test at low flow rates, and increasing power and flow rate simultaneously so that the electrical

A TUBE CROWN SHOWING SEPARATION IN COPPER

B TUBE SIDE

Figure 21

Micro Photograph of Tube Cross-Section

resistance never approaches the superconductivity range. The deficiencies found in the copper cladding on this test section precluded successful tests with cryogenic hydrogen even under the best of conditions. This was demonstrated by the second failure of the test section during a gaseous-hydrogen test which occurred at a relatively low flux and a wall temperature well above the cryogenic region and well below the melting point of copper.

VII. CONCLUSIONS AND RECOMMENDATIONS

The experiments described in this report lead to the following general conclusions:

1. A good point-to-point check between measured and predicted static pressures was not achieved in tests flowing hydrogen under isothermal conditions. While some of the difference may be ascribed to instrumentation errors (e.g., static pressure, flow rate, etc.), over-simplification in the mathematical models is considered the basic cause. Since computed profiles were similar to the experimentally-determined profiles, it appears that proper assessment of local surface roughnesses and fluid momentum conservation in the converging-diverging curved flow passage should result in an adequate prediction of the static pressure profile.

2. The effect of test section curvature on heat transfer to gaseous hydrogen was not found to be significant. The data from two tests were correlated within $\pm 20\%$ by the McCarthy-Wolf equation, Reference (9).

$$Nu_b = 0.025 Re_b^{0.8} Pr_b^{0.4} (T_w/T_b)^{-0.55} \quad (13)$$

3. Inherent test-section deficiencies prevented the acquisition of heat transfer test data for flowing cryogenic hydrogen. The limitations of this bimetallic test section precluded its use with cryogenic hydrogen, and substantiates the use of chemically-milled stainless steel sections to achieve asymmetric heating.

4. The analytical prediction of static pressure profiles in the U-tube test section appeared to be significantly affected by the surface finish on the back-side (or jacket side) of the tube. It is therefore emphasized that the surface roughness of the jacket should be minimized to reduce the drag; particularly where the Reynolds number is $>10^6$.

In conclusion, the primary objective of this test program was to generate measured static pressure profiles under controlled conditions for comparison with equivalent pressure data as generated by current prediction techniques. The results indicate that through empirical factors a plenum-to-plenum pressure match can be accommodated; however, the profiles may differ significantly, particularly in the critical throat region. It is apparent that further effort is needed to properly assess the effect of local resistance contributions to pressure drop.

It is therefore suggested that further analytical effort include the following, using the data reported herein as the basis for evaluating the validity of the assumptions employed:

1. Assess the contribution of jacket roughness to pressure drop by
 - (a) assuming local values until agreement with test data is reached, and
 - (b) analyzing available flow data for various NERVA nozzles with reference to jacket land finish (these have varied from a 16 finish to as rough as a 125 finish).

2. Develop the effect of tube axial curvature beyond the empirical I_{to} approach to better assess loss due to change in fluid momentum. It should be noted that the I_{to} correlation does not accommodate the changing flow area case, which is probably the single factor contributing most to the difference between measured and computed pressure profiles.

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NOMENCLATURE

A	heated area, in. ²
C	conversion factor, 0.948 Btu/sec-kilowatt
C _p	specific heat, Btu/lbm°R
D	inside diameter of tube, in.
D _e	equivalent diameter of tube, in.
e	surface roughness, microinches
E	electrical potential, volts
f	flow resistance coefficient
g _c	conversion coefficient, 32.17 ft-lbm/lbf sec ²
h	heat transfer coefficient, Btu/in. ² sec°R
k	thermal conductivity, Btu/in. sec°R
L	length of tube, in.
Nu	Nusselt number, h D/k (consistent units)
P _f	Empirical multiplier for flow resistance coefficient
Pr	Prandtl number, C _p μ/k (consistent units)
ΔP	pressure drop, lbf/in. ² , (psia)
q/A	heat flux, Btu/in. ² sec
r	tube radius, in.
Δr	change in tube radius, in.
R	radius of curvature, in.
Re	Reynolds number, D _p V/μ (consistent units)
T	temperature, °R
V	velocity, ft/sec
Z	wetted perimeter, in.
ρ	density, lbm/ft ³
μ	viscosity, lbm/in-sec

Subscripts

b	properties evaluated at coolant temperature, T_b
f	refers to frictional pressure drop
h	indicates heat addition
iso	indicates no heat addition
t	refers to static pressure drop
v	refers to momentum pressure drop
w	properties evaluated at wall temperature, T_w

APPENDIX A
TEST DATA SUMMARY

<u>Table</u>		<u>Page</u>
I.	Adiabatic Flow of Cryogenic Hydrogen (Test HT-7-104)	57
II.	Flow of Gaseous Hydrogen with Heat Transfer (Test HT-7-105)	58
III.	Adiabatic Flow of Cryogenic Hydrogen (Test HT-7-108)	60

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TABLE I

ADIABATIC FLOW OF CRYOGENIC HYDROGEN
(TEST HT-7-104)

Parameter	Pass Number							
	1	2	3	4	5	6	7	8
Inlet Mixer								
Temperature, °R	47.0	46.8	48.2	48.4	49.7	51.9	52.6	54.4
Pressure, psia	169.9	166.9	221.6	215.9	236.9	287.7	289.2	230.5
Outlet Mixer								
Temperature, °R	52.7	52.0	52.5	52.3	53.4	54.9	55.4	55.7
Pressure, psia	138.6	134.0	187.5	180.0	210.9	256.2	258.3	171.4
Flow Rate, lbm/sec	0.207	0.213	0.214	0.221	0.186	0.201	0.197	0.267
Static Pressure, psia								
Tap No. 1	167.3	164.0	222.5	216.1	242.1	297.3	298.1	230.6
2	162.2	158.0	216.6	209.4	236.8	292.1	293.6	220.6
3	158.7	155.6	214.2	206.4	235.2	288.4	291.2	215.4
4	155.5	151.7	210.7	201.3	231.4	286.0	287.2	208.0
5	152.0	148.0	207.6	199.2	230.4	281.5	283.6	200.9
6	147.8	143.3	200.8	194.3	225.6	278.5	279.7	193.6
7	143.4	138.6	198.2	189.9	222.7	274.3	276.3	185.6
8	140.8	135.9	195.5	186.3	220.5	272.6	274.1	179.1
9	140.3	134.1	192.4	182.6	219.4	269.5	272.4	177.3
10	139.0	133.8	191.9	184.9	216.2	268.2	270.2	176.6
11	-	-	-	-	-	-	-	-
12	142.0	137.0	195.2	186.4	220.4	270.2	272.9	177.8
13	138.9	133.6	192.9	185.0	218.3	268.2	270.6	173.5
14	142.0	137.2	198.0	189.2	220.2	271.4	274.1	179.8
15	141.9	136.6	197.8	190.2	220.8	271.9	273.7	179.5
16	143.8	139.2	198.6	190.4	221.1	273.0	275.4	182.4
17	144.3	140.9	200.0	191.6	222.8	274.2	276.1	186.9
18	145.8	140.0	201.6	191.7	223.0	273.4	277.2	184.4
19	147.8	142.8	203.6	194.6	225.5	276.2	278.9	188.7
20	149.8	144.6	205.4	196.6	226.4	277.4	279.9	190.8
21	147.8	142.8	202.0	195.4	224.8	274.9	276.0	186.2

TABLE II

FLOW OF GASEOUS HYDROGEN WITH HEAT TRANSFER
(TEST HT-7-105)

Parameter	Pass Numbers								
	1	2	3	4	5	6	7	8	9
Inlet Mixer									
Temperature, °R	491.0	469.7	489.6	493.2	499.8	506.3	521.3	510.8	510.9
Pressure, psia	286.4	224.5	266.8	262.4	265.2	288.2	267.8	272.6	266.7
Outlet Mixer									
Temperature, °R	496.8	564.0	634.7	730.1	613.9	663.8	523.6	616.6	665.8
Pressure, psia	272.4	213.7	250.6	246.8	234.1	252.6	230.3	234.0	221.3
Flow Rate, lb/sec	0.0220	0.0164	0.0197	0.0182	0.0271	0.0296	0.0313	0.0300	0.0307
Heat Input, Btu/sec	-	5.57	9.93	15.14	10.89	16.35	-	11.52	17.46
Heat Balance, %	-	+1.8	-1.06	+0.10	+0.15	+0.2	-	+2.6	+3.9
Scanivalve Pressure, psia									
Tap No. 1	295.6	227.1	275.0	271.0	271.2	295.6	271.0	277.9	268.0
2	293.1	225.4	272.5	268.7	266.5	290.4	265.2	272.2	261.6
3	291.6	224.3	271.0	267.7	264.8	288.5	263.4	270.3	259.8
4	290.1	223.2	269.4	266.0	261.3	284.7	259.2	266.2	255.1
5	288.3	221.8	267.5	264.6	258.6	281.4	256.0	262.9	251.6
6	286.6	220.6	265.6	262.5	253.9	275.7	250.1	257.0	244.6
7	284.4	219.0	263.5	260.9	251.6	273.2	247.8	254.6	242.0
8	283.2	217.9	261.8	259.7	248.4	269.8	244.0	250.8	237.9
9	282.8	217.6	261.3	258.2	245.8	266.4	240.5	246.9	233.0
10	283.4	216.4	259.1	256.1	246.9	263.6	236.8	245.0	227.7
11	-	-	-	-	-	-	-	-	-
12	282.8	218.0	261.2	257.9	245.3	265.5	240.0	246.4	232.2
13	282.4	217.4	260.3	256.9	243.4	263.1	237.5	243.5	229.0
14	283.8	218.4	261.9	258.6	246.2	266.7	241.2	247.2	233.5
15	283.8	218.4	261.7	258.3	246.0	266.2	241.0	247.0	233.2
16	284.4	219.0	262.4	258.9	247.4	268.2	242.9	248.7	235.6
17	284.7	219.4	262.7	259.3	248.0	268.8	243.8	249.5	236.3
18	284.9	219.4	262.8	259.2	248.3	269.5	244.2	249.8	237.1
19	285.9	220.4	264.1	260.4	250.6	272.0	246.9	252.7	240.5
20	285.7	220.8	263.9	260.4	251.5	272.8	248.4	254.2	242.5
21	284.7	220.2	263.9	260.2	248.5	270.2	244.1	253.0	243.5

TABLE II (cont.)

<u>Parameter</u>	<u>Pass Numbers</u>								
	<u>1</u>	<u>2</u>	<u>3</u>	<u>4</u>	<u>5</u>	<u>6</u>	<u>7</u>	<u>8</u>	<u>9</u>
Heat Generated, Btu/sec									
Between Voltage									
Tap No. V-1, -2	-	1.48	2.60	3.69	2.84	4.12	-	2.99	4.35
V-2, -3	-	1.65	3.06	4.89	3.25	5.08	-	3.45	5.48
V-3, -4	-	0.98	1.79	3.01	2.01	3.18	-	2.12	3.43
V-4, -5	-	0.75	1.28	1.85	1.46	2.09	-	1.55	2.20
V-5, -6	-	0.69	1.18	1.68	1.32	1.86	-	1.39	1.97
Wall Temperature, °R									
Thermocouple No. 1		705.0	849.1	1032.2	808.6	923.8	-	810.4	932.3
2		754.8	948.9	1270.7	867.7	1035.2	-	865.0	1045.5
3		805.7	1035.1	1453.3	978.0	1218.8	-	979.2	1253.2
4		761.0	950.9	1294.2	921.0	1119.1	-	920.8	1143.0
5		707.0	847.0	1077.1	838.7	976.4	-	843.8	991.9
6		650.7	757.7	902.0	727.4	810.7	-	731.3	818.9
7		651.6	759.3	909.7	724.4	807.3	-	726.6	814.4

TABLE III

ADIABATIC FLOW OF CRYOGENIC HYDROGEN
(TEST HT-7-108)

Parameter	Pass Number						
	1	2	3	4	5	6	7
Inlet Mixer, Temperature, °R	47.0	50.0	52.9	49.1	50.1	51.5	53.6
Pressure, psia	334.5	402.6	436.1	431.9	467.5	454.4	446.5
Outlet Mixer, Temperature °R	52.8	53.4	55.5	54.7	53.2	53.9	55.5
Pressure, psia	251.4	307.8	333.9	362.4	308.7	301.0	277.8
Flow Rate, lb/sec	0.339	0.355	0.370	0.313	0.477	0.465	0.479
Scanivalve Pressure, psia							
Tap No. 1	315.2	383.8	416.4	416.0	435.9	424.0	413.8
2	303.4	370.5	402.0	405.8	413.5	402.4	389.1
3	305.8	372.9	404.2	408.6	417.2	405.7	393.9
4	291.8	356.2	387.2	395.8	398.8	379.1	363.9
5	286.9	349.7	379.6	391.4	378.2	368.2	351.2
6	274.2	335.2	365.2	381.3	354.8	345.2	326.7
7	270.0	331.7	359.6	377.2	346.7	337.6	318.7
8	263.6	323.0	352.5	371.9	334.0	325.2	303.2
9	259.8	317.2	345.0	368.0	321.7	315.2	290.4
10	246.0	300.2	317.9	337.9	286.8	287.9	268.2
11	-	-	-	-	-	-	-
12	252.6	309.5	288.2	363.4	311.2	303.5	280.6
13	249.9	306.8	334.0	361.2	307.6	299.2	274.9
14	255.0	315.1	341.8	367.0	319.2	310.8	288.2
15	256.0	314.2	341.6	366.7	319.4	310.9	288.1
16	260.0	318.0	345.6	369.8	325.2	317.5	295.3
17	262.5	320.6	348.2	372.2	329.5	320.6	298.7
18	262.2	321.4	348.9	372.4	331.8	323.5	301.8
19	269.5	328.5	356.2	376.9	342.8	334.0	313.6
20	273.0	332.4	360.9	379.8	351.2	340.9	320.9
21	274.9	330.5	355.6	377.4	334.9	337.8	314.3