REFERENCE MASTER

FLUID FLOW AND TRANSIENT HEAT TRANSFER OF GASEOUS HYDROGEN AT LOW TEMPERATURES

by

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SUMMARY

As part of a program to study liquid hydrogen cool-down and forced convection transient boiling heat transfer to hydrogen in a single tube, cold gaseous hydrogen runs were made to study the characteristics of the test apparatus. The results have shown that:

1. The orifice meter can be used with confidence to measure the flow-rates of single phase, gaseous hydrogen , and
2. The experimental local heat transfer coefficients for single phase heat transfer can be correlated by well-known equations.

INTRODUCTION

As part of a program to study liquid hydrogen cool-down and forced convection transient boiling heat transfer to hydrogen in a single tube, cold gaseous hydrogen runs were made in an attempt to:

1. Check the reliability of the orifice meter for the measurement of flow-rates,
2. Check the equation recommended by Thomas for correlating single phase local heat transfer, and
3. Determine the accuracy of the pressure drop measurements across the test section.

EXPERIMENTAL APPARATUS AND PROCEDURE

A detailed description of the test apparatus and procedures will not be given here. Only those pertinent to these tests are discussed. The hydrogen gas used in these tests was obtained from six cylinders connected to a manifold. For each run, the initial cylinder pressure as well as the pressures at fixed time intervals were recorded.

The test section consists of two feet of copper tubing of 3/16 inch inside diameter and 1/4 inch outside diameter. The outside wall temperature and the stream temperature at the inlet as well as the outlet of the test section were measured by copper-constantan thermocouples and recorded continuously on a Visicorder. The inlet static pressure and pressure drop across the test section were also recorded.

The orifice meter, with standard flange taps, is located well downstream from the test section. The pressure drop across the orifice, static pressure at the upstream pressure tap, and the downstream fluid temperature were measured and recorded continuously.

Hydrogen gas was passed through a liquid nitrogen bath prior to entering the test section. All the piping up to the entrance of the test section was first cooled-down to liquid nitrogen temperature before the actual test began. Thus in the test section, the cold hydrogen gas came into contact with walls that were initially at ambient temperature.

METHODS OF CALCULATIONS

Flow-Rates

The mass flow-rates of the hydrogen can be calculated by three methods.

From the known pressure drop in the gas cylinders in a given time period, the mass flow-rate can be calculated by assuming isothermal ideal gas behavior or by assuming a reversible adiabatic process. It can also be calculated from the pressure drop across the orifice by the well-known equation:

$$W = C_o F_{Re} Y S_o \sqrt{2 g_c (p_a - p_b) \rho_a} \quad (1)$$

where W is the mass flow-rate, C_o is a coefficient which is a function of β , the relative diameter of the orifice to the tube. A chart giving C_o as a function of β is given by McCabe and Smith³. For the orifice used here, C_o was taken to be 0.608. F_{Re} , a function of the Reynold's number, corrects for the viscosity of the fluid. It can be calculated from the equation:

$$F_{Re} = 1 + \left(\frac{\beta D_a}{N_{Re,o}} \right) \left(A' + \sqrt{\frac{530}{D_a}} \right) \quad (2)$$

The factor A' is a function of β and is given on page 106 of reference 3. The expansion factor can be calculated from the following equation:

$$Y = 1 - \left(\frac{0.41 + 0.35 \beta^4}{K} \right) \left(\frac{p_a - p_b}{p_a} \right) \quad (3)$$

where K is the specific heat ratio of the gas. For the high Reynold's numbers encountered in these tests, F_{Re} can be taken as 1 and Equation (1) reduces to:

$$W = .608 S_o Y \sqrt{2 g_c (p_a - p_b) \rho_a} \quad (4)$$

Local Heat Transfer Coefficients

Neglecting axial conduction and radial temperature differences in the tube, a shell energy balance on a differential section of the tube of length dL gives

$$-h \pi D_i dL (T_w - T_f) = \rho C_p \frac{\pi}{4} (D_o^2 - D_i^2) dL \frac{dT_w}{dt}$$

or

$$h = -\rho \frac{C_p (D_o^2 - D_i^2)}{4 (T_w - T_f) D_i} \frac{dT_w}{dt} \quad (5)$$

Note that Equation (5) is independent of position; hence it defines the local heat transfer coefficient at any point along the tube.

Based on a literature survey, Thomas' recommended the following equation for correlating single-phase local heat transfer coefficients for hydrogen:

$$N_{Nu} = A (N_{Re})^{.8} (N_{Pr})^{.4} \left(\frac{T_w}{T_f} \right)^{-B} \left[1 + C \left(\frac{X}{D_i} \right)^{-D} \right] \quad (6)$$

Where X is the distance from the entrance to the test section, and A, B, C and D are constants. The values of the constants recommended by Thomas are:

$$A = .025$$

$$B = .55$$

$$C = .3$$

$$D = .07$$

Pressure Drop

The pressure drop across the test section can be calculated from the standard equation

$$\Delta p = 4 f \frac{L}{D_i} \frac{\rho_f V^2}{2 g_c} \quad (7)$$

where f is the friction factor, L is the length of the tube over which the pressure drop is measured, and V is the average fluid velocity.

DISCUSSION

Experimental Results

The inlet and outlet wall and fluid temperature histories for Run 1-GH-1 are shown in Figures 1 and 2. The static pressure and pressure drop histories for the same run are plotted in Figure 3. From Figure 2, note that at the exit of the test section, the fluid temperature became higher than the wall temperature after twelve seconds. From a comparison of the fluid temperatures in Figures 1 and 2, it is apparent that heat had been transferred continuously from the wall to the fluid in the first 30 seconds, yet the wall at the exit became cooler than that at the inlet after 22 seconds. Since there is no heat source between the two positions and no heat sink in the vicinity of the exit, it is concluded that the observed phenomenon is physically impossible, and that the wall temperature at the exit must be in error.

Recalibration of the exit thermocouple at the steady state operating conditions resulted in the corrected temperature histories shown on Figures 4 and 5 for the two runs made. The corrections were well within the expected 2% error from Visicorder readings alone.

Mass Flow-Rates

As seen from Figure 3, the pressure drop across the orifice increased from zero to a steady state value within a few seconds. Since the average flow-rates were estimated from cylinder pressure drop over six minutes of test time, for all practical purposes, the pressure drop at steady state can be used to calculate the flow-rates from Equation (4).

The temperature at the orifice decreased with time as shown in Figure 6; consequently, a time-averaged temperature was computed so that an average density can be determined. Since the conversion of ortho to parahydrogen in the absence of a catalyst is extremely slow, normal hydrogen was assumed to be the gas at all times. The physical properties were obtained from reference 2. The mass flow-rates estimated and calculated by the three methods are given below:

Run	Δp , psi	$\bar{T}$, °R	Mass Flow-Rates, Lb/sec.		
			Isothermal Process, Ideal Gas	Isentropic Process	Equation 4 (Orifice)
1-GH-1	2.1	371	.00276	.00239	.00208
1-GH-2	3.1	364	.00388	.00262	.00298

The above comparison shows that the assumption of an isentropic process gave better estimates of the mass flow-rates. This was expected, since the gas should deviate drastically from ideal gas behavior under high pressures. Although the assumption of a reversible adiabatic process for the discharge of pressurized hydrogen from cylinders is questionable, it is conceivable that such a process can be devised under idealized conditions.

It is estimated that flow-rates calculated on the basis of an isentropic process can be in error by as much as $\pm 10\%$. In view of the fact that the orifice meter was designed for a coefficient of 0.60 ($C_o Y$), the close agreements between the flow-rates

so estimated and those calculated by Equation (4) suggest that the equation can be used with confidence for the determination of single phase gaseous hydrogen flow-rates.

Heat Transfer Coefficients

The pressure drop across the orifice reached steady values in approximately five seconds; consequently, a comparison of heat transfer coefficients is significant only for those values computed for times in excess of five seconds. Strictly speaking, the calculation of the local heat transfer coefficient by Equation (5) should be based on the inside wall temperature; however, the combination of the high thermal conductivity for copper and the thin-wall of the tubing resulted in a negligible temperature difference across the wall so that the outside wall temperatures can be used. An estimate of the temperature difference across the tube under experimental conditions indicated that it was of the order of 0.5°R . At any given time the slope $\frac{dT_w}{dt}$ was determined by graphical differentiation of the time-wall temperature curve. Cryogenic properties of copper were obtained from reference 4.

A 5-3/4 inch copper tubing precedes the point at which inlet measurements were made, and the position at which exit measurements were made is 24 inches downstream from the inlet position; therefore, $\frac{X}{D}$ is very large and the term $\left[1 + C\left(\frac{X}{D}\right)^{-D}\right]$ becomes 1. Under the experimental conditions, the ratio $\frac{T_w}{T_f}$ was always less than 2. For $\frac{T_w}{T_f}$ less than 2, McCarthy and Wolf⁶ recommended the coefficient A to be .023. Accordingly, the equation used for predicting local heat transfer coefficients is

$$\frac{h D_i}{K} = .023 \left(\frac{4 W}{\pi D_i \mu} \right)^{.8} \left(\frac{C_p \mu}{K} \right)^{.4} \left(\frac{T_w}{T_f} \right)^{-.55} \quad (9)$$

The local heat transfer coefficients calculated and predicted by Equations (6) and (9), respectively, are compared in Table I.

Within the range of temperatures in a given run, the predicted local heat transfer coefficients were practically constant. These values were consistently higher than those determined experimentally. The deviations were less than 16%. For a better correlation of the experimental data, the constant A in Equation (7) should be .0205. Values of A as low as .0196 have been reported in the literature. A variety of exponents to the temperature ratio, Reynold's and Prandtl's numbers have been used by different investigators'. In view of the divergencies, the agreement between observed and predicted heat transfer coefficients are considered excellent. This fact provides an additional check on the reliability of the orifice meter.

Pressure Drop Across The Test Section

The friction factors used in Equation (7) were obtained from reference 5. The approximate steady state fluid temperature of 280°R for Run 1-GH-1 and 250°R for Run 1-GH-2 were used to determine the fluid properties. The observed and predicted pressure drops are as follows:

TABLE I

COMPARISON OF EXPERIMENTAL AND PREDICTED
LOCAL HEAT TRANSFER COEFFICIENTS

$(h \times 10^3 \text{ BTU/in.}^2\text{-sec. } ^\circ\text{R})$					
t, Seconds	$T_f \text{ } ^\circ\text{R}$	INLET		EXIT	
		Experimental Equation (6)	Predicted Equation (7)	Experimental Equation (6)	Predicted Equation (7)
<u>Run 1-GH-1 - $\bar{w} = .00208 \text{ lb/sec.}$</u>					
5	358	.66	.80	.89	.84
6	352	.61	.80	.78	.83
7	343	.62	.79	.71	.83
8	342	.57	.79	.81	.82
9	340	.57	.79	.72	.82
10	339	.51	.78	.82	.81
<u>Run 1-GH-2 - $\bar{w} = .00298 \text{ lb/sec.}$</u>					
6	322	.545	1.01	1.26	1.06
8	310	.422	1.00	1.38	1.02
10	305	.372	.98	1.58	1.01
12	296	.367	.96	1.37	.99
14	287	.325	.96	1.53	.99
$h \times 10^3 \text{ BTU/in.}^2\text{-sec. } ^\circ\text{R}$					

Run	Average Observed	Predicted	% Deviation
1-GH-1	.69	.80	16
1-GH-2	.92	1.00	8.7

Run	Pressure Drop, PSI		Per Cent Deviation
	Observed	Predicted	
1-GH-1	4.6	3.7	19.5
1-GH-2	6.4	5.8	9.4

The agreement between predicted and observed pressure drops are extremely good, considering that flow disturbances due to the presence of thermocouples in the stream and other unknown factors have not been taken into account. The relatively small deviations can be readily accounted for by the errors in measuring the flow-rates.

CONCLUSION

From this work, it has been shown that in single-phase flow and heat transfer to gaseous hydrogen at cryogenic temperatures, the pressure drops and heat transfer coefficients can be readily predicted from well established equations and generalized correlations.

NOMENCLATURE

A	-	A constant
A'	-	A coefficient, a function of β
B	-	A constant
C	-	A constant
C_o	-	Orifice coefficient, a function of β
C_p	-	Heat capacity of the tube
D	-	A constant
D_a	-	Upstream inside diameter for the orifice
D_i	-	Inside diameter of the test section
D_o	-	Outside diameter of the test section
F_{Re}	-	A viscosity correction factor
f	-	Friction factor
g_c	-	A conversion factor, $32.2, \text{ft. lb}_m/\text{lb}_f \text{ sec.}^2$
h	-	Local heat transfer coefficient
K	-	Specific heat ratio of the fluid
L	-	Length of the test section
N_{Re}	-	Reynold's number
N_{Pr}	-	Prandtl number
p_a	-	Orifice upstream static pressure

NOMENCLATURE (con't.)

p_b	-	Orifice flange tap static pressure
S_o	-	Orifice cross-sectional area
t	-	Time
T_w	-	Inside wall temperature
T_f	-	Fluid temperature
V	-	Average fluid velocity
W	-	Mass flow-rate
X	-	Distance from inlet to the test section
Y	-	Expansion factor
β	-	Ratio of orifice diameter to upstream tube diameter
ρ	-	Density of tube
ρ_a	-	Orifice upstream fluid density
ρ_f	-	Average fluid density in the test section

FIGURE 1. INLET WALL AND FLUID TEMPERATURE HISTORIES
 RUN 1-GH-1, $\bar{W} = .00208$ LB/SEC

FIGURE 2. EXIT WALL AND FLUID TEMPERATURE HISTORIES
RUN 1-GH-1

FIGURE 3. PRESSURE AND PRESSURE DROP HISTORIES
RUN 1-GH-1

FIGURE 4. EXIT WALL (CORRECTED) AND FLUID TEMPERATURE HISTORIES

RUN 1-GH-1
 $\bar{W} = .00208 \text{ LB/SEC}$

FIGURE 5. EXIT WALL (CORRECTED) AND FLUID TEMPERATURE HISTORIES

RUN 1-GH-2
 $\bar{W} = .00298$ LB/SEC

FIGURE 6. TYPICAL ORIFICE TEMPERATURE HISTORY
RUN 1-GH-2

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