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Experimental Study of H_2O-LH_2 and H_2O-LN_2 Heat Exchangers*

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by

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ABSTRACT

A study was made of heat transfer coefficients and ice formation during the steady-state operation of countercurrent H_2O-LH_2 and H_2O-LN_2 heat exchangers. The cryogen flowed through a 1" O.D. (0.064-in. wall), 347 stainless steel tube while the water flowed through the annulus between the inner tube and the 1½-in. I.D. Plexiglas outer tube. Flow rates were 3250-10,400 lb/hr for water, 227-778 lb/hr for hydrogen, and 1820-7000 lb/hr for nitrogen. Pressures ranged from 50 to 150 psia. The inlet water temperature ranged from 565-594 °R while the bulk outlet water temperature approached the ice point. Hydrogen outlet temperatures ranged from 300 to 507°R while the nitrogen temperatures never exceeded the saturation temperature at the pressures involved.

The experimental hydrogen heat transfer coefficients were 1/2 to 1/9 of those predicted by the Dittus-Boelter equation. The experimental two-phase nitrogen coefficients were 1/4 to 1/26 of the theoretical single-phase gas coefficients.

A stable layer of ice formed during the hydrogen runs while ice was essentially absent during the nitrogen runs. Substantial pressure oscillations were observed during the hydrogen runs but none were seen during the nitrogen runs. It was concluded that the use of water as a heat exchange medium to cryogenic fluids presents no new operating difficulties.

* Work performed under the auspices of the U.S. Atomic Energy Commission.

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Introduction

At the Nuclear Rocket Development Station in Nevada, modifications are currently being planned for the Rover test facility at Test Cell C. These plans incorporate heat exchangers to provide ambient temperature hydrogen and nitrogen gas. Since the required quantities of these gases is large, gas storage was not economically feasible. The proposed heat exchangers utilize hot water as the energy source. Water was chosen because of its availability and inherent safety. Since this type of heat exchanger, with ice formation, had not been used previously, a model was constructed and tested at the Los Alamos Scientific Laboratory. The results of this investigation are discussed herein.

* Work performed under the auspices of the U.S. Atomic Energy Commission.

Equipment

The full-scale hydrogen heat exchanger is to consist of four parallel heat exchangers each consisting of six passes, each pass having a length of 80 ft. The water and the hydrogen are to flow countercurrently: the hydrogen in the inner, 5-in. schedule XX aluminum tube and the water in the annulus between the inner and the outer, 10-in., schedule 40 carbon steel tube.

The experimental model was a single-pass countercurrent unit 47 ft long. The inner flow passage was a 1-in., schedule 5S, 347 stainless steel tube while the outer was a 1½-in. I.D. Plexiglas tube. Steam-heated water was supplied from a 1300-gallon storage tank through a positive displacement pump (see Figure 1). The hydrogen (or nitrogen) was supplied from a 300ℓ dewar capable of being pressurized to 300 psia. Table 1 summarizes the full-scale design conditions and the average values obtained on the experimental model. Specific data and computational methods will be discussed later.

Instrumentation

Figure 2 is a schematic drawing of the test section showing the location and type of instrumentation used. All thermocouples were copper-constantan. A nitrogen reference junction was used for the nitrogen runs and a hydrogen reference junction for the hydrogen runs. All thermocouples measuring bulk fluid temperatures were bare and located on the centerline of the flow path with the exception of four thermocouples located 10 and 14 feet from the inlet. These latter were located halfway between the centerline and tube wall and were used to detect radial temperature gradients within the cryogen stream. The surface thermocouples were soft soldered to the outside surface of the inner tube. The experimentally-observed temperatures had an accuracy of $\pm 2^{\circ}\text{R}$.

Both inlet and outlet, water and cryogen, pressures were observed with an accuracy of ± 2 psi. The cryogen flow rate was determined both from a turbine flowmeter and from the supply dewar load cell. Such flow rates had an accuracy of $\pm 5\%$. Determination of the water flow rate was made by weighing water which passed through the apparatus. No measurements of ice thickness were made, though the extent and character of ice formation was visually observed.

Procedure

In all runs the water flow was established at the desired level prior to the initiation of cryogen flow. In the hydrogen runs the cryogen flow was controlled by means of the hand-operated valve downstream of the test section; in the nitrogen runs, the upstream control valve was used with the downstream valve being completely open. The raw temperature data were recorded on a Visicorder and Multiplex unit. The Multiplex unit recorded data from each thermocouple 20 times per second. These data were then reduced by computer giving plots of temperature vs. time.

Methods of Calculation

Average heat transfer coefficients were calculated using the average temperatures between axial locations on the test section. Calculations were made for three sections of the experimental equipment: 14 to 22 feet from the cryogen inlet, 22 to 24 feet, and 24 to 34 feet. It was necessary to calculate these averages rather than a local value in order to obtain a ΔT in the water large enough to be accurately measurable.

The heat load, $\dot{Q}$, was taken as $WC_p \Delta T$ of the water, ΔT being taken between any two axial locations where temperature data were available. The overall heat transfer coefficient, U , was then obtained by dividing the flux, $\dot{Q}/A$ (based on the outside area of the inner pipe), by the arithmetic average temperature driving force. This average was used because the difference between it and the log mean ΔT was negligible. This average ΔT was equal to $\frac{1}{2}[(T_{x_1}^{H_2O} - T_{x_1}^{H_2}) + (T_{x_2}^{H_2O} - T_{x_2}^{H_2})]$. The thermal resistance, $1/h_c$, of the cryogen film was computed by subtracting the resistance of the water film, the ice, and the pipe wall from the total resistance, $1/U$.

A few words may be said about the calculation of these thermal resistances. For runs 1-5, the hydrogen runs, the water-side resistance was computed by dividing the heat flux by ΔT between the water and the wall. If the wall temperature was lower than 32°F, indicating the presence of an ice layer, the ΔT across the water film was taken as the water temperature minus 32°F and the ice

resistance was computed using the ΔT from the ice surface (32°F) to the tube wall. The ΔT across the tube wall was calculated from the known heat flux, tube wall thickness, and thermal conductivity of stainless steel.

For the nitrogen runs (runs 6-20) the same techniques were used with one exception. The water-side heat transfer coefficients were calculated theoretically by means of the Dittus-Boelter equation. This was done because the temperature differences between the water and wall were in general less than twice the accuracy of the measured temperatures. Since the nitrogen-side resistance controlled the heat transfer rate (see Table 3), it was not necessary to know the water-film coefficient very accurately.

Results and Discussion

A. Hydrogen Runs 1-5

The internal consistency of the data is established in Figure 3 which is a plot of Q vs. $W_{H_2}^8$. It can be seen that all points fall within $\pm 20\%$ of the straight line. Further, it can be seen that the heat gained by the hydrogen checks with the heat lost by the water to within 18% in all but one case, that case being within 28%. Heat lost to air was estimated as less than 0.1×10^5 Btu/hr (less than 5% of the total heat transferred).

The test results may be discussed under three categories - heat transfer coefficients, ice formation, and pressure oscillations. Heat transfer coefficients for the hydrogen runs are summarized in Table 2. The theoretical values were calculated by means of the Dittus-Boelter equation ($Nu = 0.023 Re^{.8} Pr^{.4}$) with fluid properties evaluated at the average bulk temperature. In all of these runs, the water-side coefficients were between 1000-1700 Btu/hr-ft²-°R. The heat transfer rate was determined primarily by the hydrogen film resistance, although the resistance of the ice layer was significant when ice was present. The experimental hydrogen-film coefficients were all low by a factor of 2 to 3.8 at the downstream Section c (see Figure 2) and low by a factor of 5.8 to 9 at the upstream Section a when compared to the Dittus-Boelter values. It is postulated that the very low coefficients at Section a are the

result of a central core of high density hydrogen in the pipe which does not mix with the warmer fluid along the hot wall. Radial temperature gradients of 30°R in $1/8$ inch were measured in the hydrogen stream at a point 193 pipe diameters (14 feet) from the inlet even though Reynolds numbers were of the order of a million.

Ice was formed in the hydrogen runs for a length of about 24 feet from the hydrogen inlet in most cases. The ice, which adhered strongly to the pipe wall, appeared to craze as it was first formed, but the cracks were immediately filled with water which froze, providing a smooth surface. The thickness of the ice layer achieved its steady state value in less than a minute. Ice tended to form more heavily on the bottom of the inner tube than on the top. The effect of the ice layer on the axial temperature gradient in the water may be seen in Figures 4 and 5. With no ice (Figure 4), the temperature of the water is a linear function of length. When ice covered part of the length, as in run 5 (Figure 5), the temperature gradient in the water was more closely represented by two straight lines intersecting at the break point between ice and no ice.

Pressure oscillations were observed in all of the runs using liquid hydrogen. The amplitude of these oscillations was as large as ± 50 psi at a pressure level of 200 psi.

The existence of the cold fluid core, asymmetrical ice formation, and large pressure oscillations suggested that a swirl inducer might

improve the operation of the heat exchanger. Accordingly, a copper strip about 0.85-inch wide was twisted into a helix 14-feet long with a twist of 180° per three inches of length and inserted into the first 14 feet of the inner tube. The use of the swirl inducer resulted in uniform ice formation on the circumference of the pipe, but had no effect on the pressure oscillations.

B. Nitrogen Runs 6-20

A plot of Q vs. $W_{N_2}^8$ for the nitrogen runs (Figure 6) shows all data points but one to be within 10% of the straight line.

Both the measured nitrogen temperatures and the heat balances indicated that the nitrogen was in the two-phase region throughout the test section in all runs. The heat transfer coefficients for nitrogen (summarized in Table 3) were lower than those predicted from the Dittus-Boelter equation by factors of 4-26. The Dittus-Boelter calculations were made assuming the flow to be saturated gas. As with hydrogen, the coefficients obtained in Section a were lower than those obtained in the downstream Section c, although the difference was less significant. It is of interest to note the relatively high overall coefficients obtained in Section b. Section b was a 2-foot long section which contained a soldered, sleeve pipe joint. The increased coefficients in this section were due to

increased turbulence in the nitrogen stream. Temperature measurements indicated the presence of a subcooled core of liquid similar to the observed high density hydrogen core.

Ice formed for less than two feet from the nitrogen inlet. This was primarily an end effect due to the reduced water velocity at the outlet water plenum. No pressure oscillations were observed.

Summary and Conclusions

A. Hydrogen Experiments

Experimental hydrogen heat transfer coefficients were found to be considerably lower than those predicted from theory. The presence of a high density fluid core surrounded by less dense fluid is believed to be the cause of the low coefficients. Large pressure oscillations were evident in all runs. Ice formed in a crazed fashion with a tendency to build preferentially on the outside bottom surface of the inner tube. In the iced section of the heat exchanger, the resistance to heat transfer was approximately the same in the ice film and hydrogen film. The presence of a swirl inducer in the initial 14 feet of the heat exchanger did not reduce oscillations; however it did produce uniform circumferential ice formation. It was concluded that the use of H_2O as a heat exchange medium for LH_2 heat exchangers presents no special operational problems from the viewpoint of ice formation and ice stability.

B. Nitrogen Experiments

Experimental two-phase nitrogen heat transfer coefficients were lower than those predicted from single-phase theory. Ice formed for very short distances from the nitrogen inlet due to the very low nitrogen heat transfer coefficients. No pressure oscillations of consequence were detected and no operational difficulties were encountered.

NOTATION

- A - heat transfer area based on O.D. of inner tube, ft^2
- a - section of heat exchanger 14-22 feet from cryogen inlet
- b - section of heat exchanger 22-24 feet from cryogen inlet
- c - section of heat exchanger 24-34 feet from cryogen inlet
- C_p - specific heat, $\text{Btu/lb-}^\circ\text{R}$
- G_c - mass velocity of cryogen - lb/hr-ft^2
- h_c - heat transfer coefficient for cryogen, $\text{Btu/hr-ft}^2\text{-}^\circ\text{R}$
- l - liter
- Nu - Nusselt Number, dimensionless
- Pr - Prandtl Number, dimensionless
- Q - total heat transferred in full length of heat exchanger (45 feet), Btu/hr
- Re - Reynolds Number, dimensionless
- T - temperature, $^\circ\text{R}$
- ΔT - temperature difference, $^\circ\text{R}$
- U - overall heat transfer coefficient, $\text{Btu/hr-ft}^2\text{-}^\circ\text{R}$
- W - mass flow rate, lb/hr
- x_1, x_2 - axial location where data are obtained

Subscript

- c - cryogen

Table 1

Heat Exchanger Full-scale Design Data and Model Experimental Data

(A Summary)

	Full-scale Design Data	Average Experimental Data	
	Hydrogen	Hydrogen	Nitrogen
G_c , lb/hr-ft ²	5.5×10^5	1.3×10^5	11.0×10^5
G_{H_2O} , lb/hr-ft ²	1.8×10^6	1.2×10^6	0.8×10^6
Re_c	10.7×10^6	1.1×10^6	(Two Phase)
Re_{H_2O}	5.6×10^5	0.22×10^5	0.15×10^5
T , °R Cryogen in	55	40	136
T , °R Cryogen out	520	400	168
T , °R Water in	660	590	570
T , °R Water out	540	500	518
Q/A , BTU/hr-ft ²	1.2×10^5	0.53×10^5	0.23×10^5
P , psia Cryogen in	1000	100	70
h_c , BTU/hr-ft ² -°R	550	220	38
h_{H_2O} , BTU/hr-ft ² -°R	1220	1380	920

Table 2

Experimental Hydrogen Data

Run	Section*	H ₂ inlet press. psia	W _{H₂O} , lb/hr	W _{H₂} , lb/hr	Re _{H₂O} x 10 ⁻⁴	Re _{H₂} x 10 ⁻⁵	U, BTU/hr-ft ² -°R	Expt. h _{H₂}	Theo. h _{H₂}
1	c	50	7500	227	2.9	2.3	163	213	485
2	c	50	7500	227	2.6	2.5	183	250	485
3	a	150	10000	729	1.9	18.	58	136	792
3	c	150	10000	729	2.5	12.	192	388	963
4	a	150	6850	637	1.3	17.	40	90	795
4	c	150	6850	637	1.7	14.	131	231	883
5	c	125	9100	778	2.4	14.	207	395	983

* See Figure 2

Table 3

Experimental Nitrogen Data

Run	Section*	N ₂ , inlet press. psia	W _{H₂O} , lb/hr	W _{N₂} , lb/hr	Re _{H₂O} × 10 ⁻⁴	U, BTU/hr-ft ² -°R	Expt. h _{N₂}	Theo. h _{N₂} Gas
6	c	50	8000	1820	2.7	31	32	189
7	a	50	8000	1820	2.5	20	20	189
8	c	50	3250	1820	0.82	43	48	189
9	a	100	8260	5950	2.5	41	43	450
9	b	100	8260	5950	---	164	---	---
10	a	100	6500	5950	2.0	16	17	450
11	a	80	6010	6650	1.9	53	58	515
11	b	80	6010	6650	---	201	---	---
11	c	80	6010	6650	2.0	52	57	515
12	c	80	4505	7000	1.5	27	28	542
13	a	75	5500	3200	1.4	36	38	290
13	b	75	5500	3200	---	164	---	---
13	c	75	5500	3200	1.7	44	48	290
14	a	65	5750	4130	1.5	30	32	350
15	c	65	4500	4210	1.3	31	34	350
16	a	70	4250	4410	1.1	33	36	400
17	c	75	5100	5550	1.5	41	44	463
18	a	70	4500	5180	1.2	23	23	430
19	b	70	4250	4780	---	210	---	---
19	c	70	4250	4780	1.3	42	45	400
20	c	70	4130	5060	1.3	40	44	400

* See Figure 2

Figure 1

Schematic diagram of the heat exchanger flow system

Figure 2

Schematic drawing of test section showing instrumentation

Figure 3

Heat load vs (hydrogen flow)^{0.8} for runs 1-5

Figure 4

Axial temperature gradient without ice present

Water temp.,
°R

Figure 5

Axial temperature gradient for water --- ice present