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APOLLO FUEL-CELL CONDENSER HEAT-TRANSFER TESTS

by Michael B. Weinstein

Lewis Research Center

Cleveland, Ohio

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SUMMARY

An experimental evaluation of the Apollo fuel-cell condenser showed that, in the expected region of condenser operation, the coolant temperature needed to condense the correct amount of water decreases linearly with increasing fuel-cell waste heat production. It was also found that, in this operating region, the overall condenser heat-transfer coefficient is approximately constant at 42.5 Btu per square foot per hour per $^{\circ}F$.

INTRODUCTION

The fuel-cell system currently being developed to generate electrical power for the Apollo spacecraft produces both waste heat and water in the electrochemical reaction between hydrogen and oxygen: $H_2 + \frac{1}{2}O_2 \rightarrow H_2O$ + power + heat. To maintain the optimum operating temperature and electrolyte concentration in the fuel-cells, both the heat and the water produced are continuously removed by transfer to a recirculating hydrogen reactant stream. This gas stream is passed through a small, counterflow, heat-exchanger condenser in which the waste heat is transferred to the coolant fluid, while the product water is condensed, to be removed by a downstream water separator.

This condenser, a typical plate-fin heat exchanger (figs. 1 to 5), consists of small trapezoidal flow passages in which both the gas and the coolant are flowing laminarly with Reynolds numbers of about 170 and 20, respectively. The large heat-transfer area thus provides good hot-to-cold stream thermal contact in a small volume.

The operating condenser can be thought of as divided into two roughly equal sections along the flow path. The hot gas stream is rapidly cooled to the condensation temperatures in the first section and then is condensed in the second section. The condenser operates under a total gas pressure of 60 pounds per square inch absolute, with a gas pressure drop of 0.03 to 0.05 pound per square inch absolute (ref. 1), approximately that needed to clear the small flow tubes of condensate.

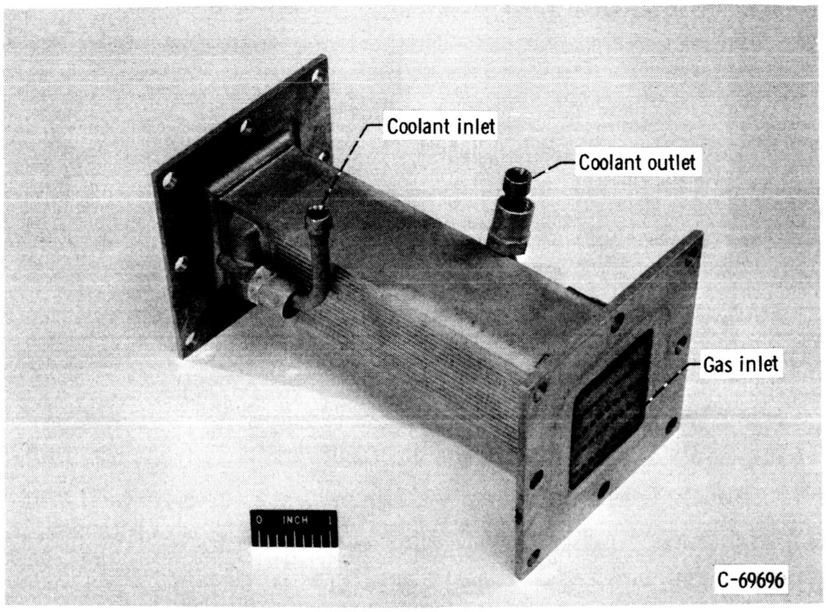


Figure 1. - Apollo fuel-cell condenser.

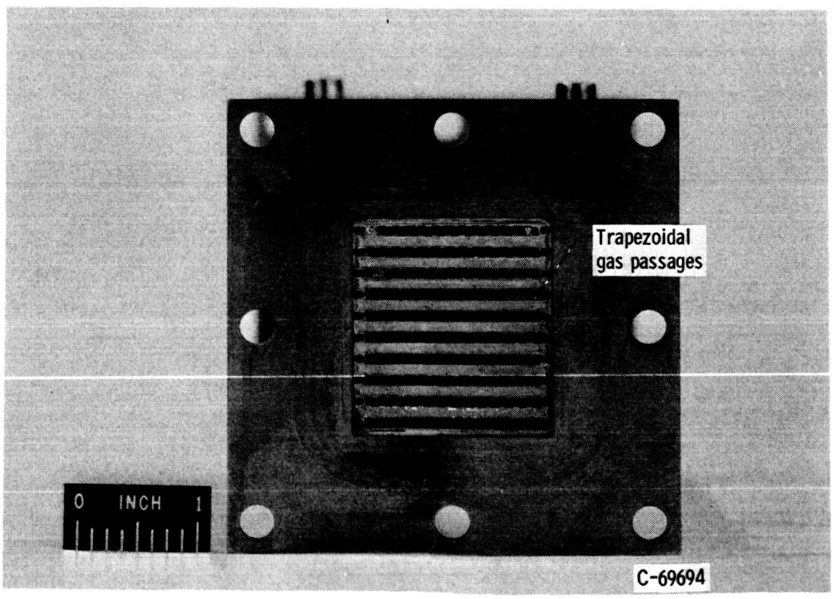


Figure 2. - Condenser exit.

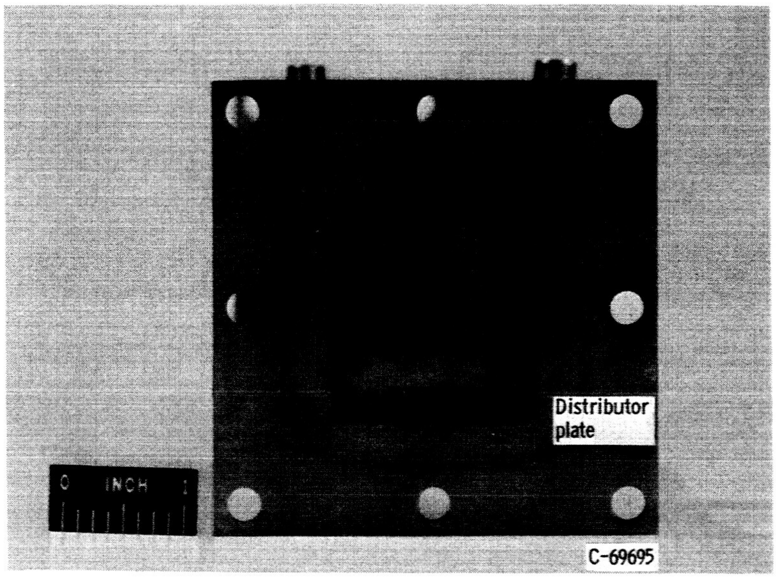


Figure 3. - Condenser gas inlet.

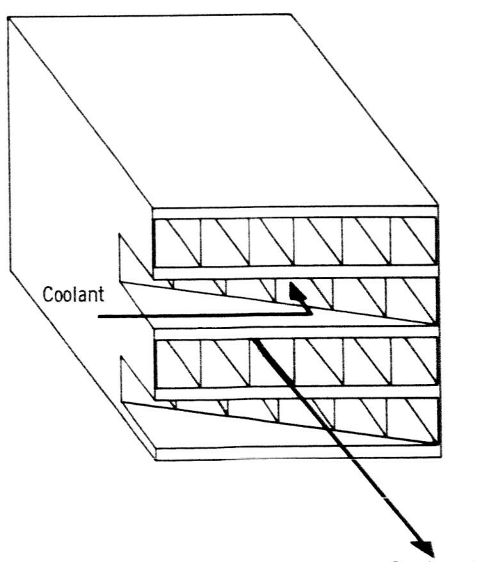
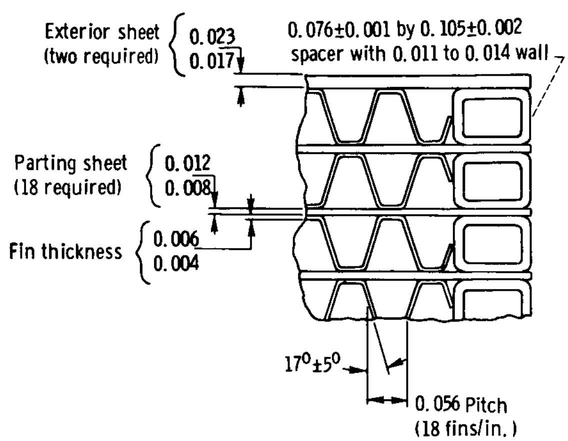
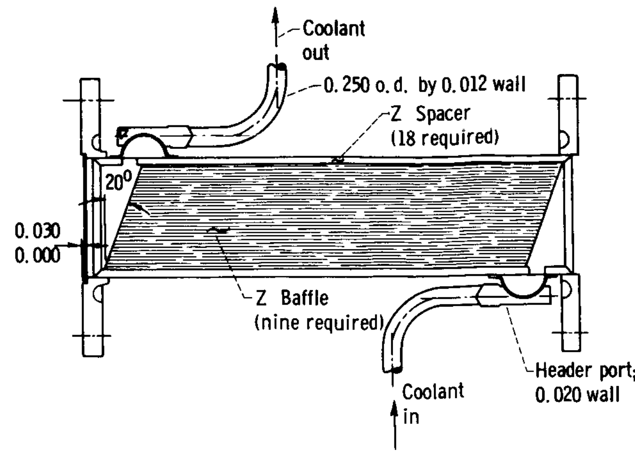


Figure 4. - Condensing and coolant passages.



(a) Section of core (x10).



(b) Coolant passage; nine places alternating with reactant passages.

Figure 5. - Condenser. (All dimensions in inches.)

TABLE I. - CONDENSER DESIGN POINT OPERATING CONDITIONS

Fuel-cell gross power, W	Gas inlet temperature, °F	Hydrogen inlet flow, lb/hr	Water inlet flow, lb/hr	Water vapor outlet flow, lb/hr	Condensing rate, lb/hr	Gas outlet temperature, °F
450	191	3.42	3.07	2.69	0.36	161
500	197	3.42	3.09	2.69	.40	161
600	204	3.40	3.23	2.74	.49	162
700	214	3.39	3.37	2.81	.56	163
800	220	3.39	3.44	2.81	.63	163
900	230	3.38	3.58	2.87	.69	164
1000	243	3.37	3.74	2.94	.80	165
1100	258	3.35	3.89	3.00	.89	166
1200	270	3.35	3.98	3.00	.98	166
1300	284	3.34	4.14	3.06	1.08	167
1400	305	3.33	4.30	3.13	1.17	168
1500	323	3.33	4.40	3.13	1.27	168

Previous tests on this condenser to determine its internal flow stability were used as a basis for tests to determine experimentally the coolant temperatures needed to remove the correct amount of heat and water for hot gas conditions within the expected fuel-cell system operating range (table I). An apparatus (fig. 6) similar to that used to determine the internal flow stability was constructed with the condenser held in a horizontal position to negate the effect of a gravity field along its axis. With this configuration all the pertinent parameters, such as flow rates and temperatures, could be controlled over a wide range of values. The information obtained in this experiment can be used in a system model to predict this condenser's operation under other possible conditions.

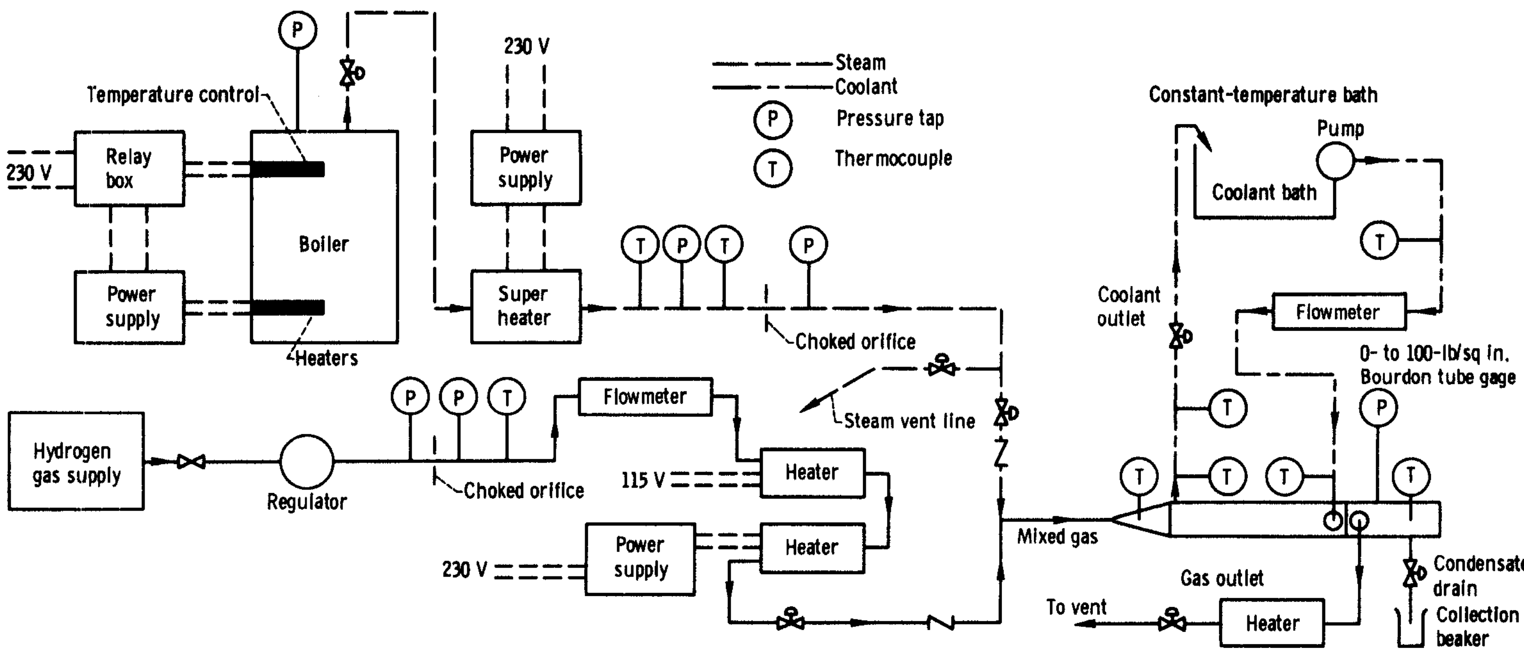


Figure 6. - Simplified schematic drawing of test rig.

SYMBOLS

A orifice area, sq in. (1 sq in. = 6.4516×10^{-4} sq m)

a condensing area, sq ft (1 sq ft = 9.290×10^{-2} sq m)

C_p heat capacity, Btu/(lb)(^{o}F) for glycol ((1 Btu/(lb)(^{o}F) = 4.184×10^{3} J/(kg)(^{o}C)) and Btu/(lb)(mole)(^{o}F) for gas (1 Btu/(lb)(mole)(^{o}F) = 4.184×10^{3} J/(kg)(mole)(^{o}C))

$\mathscr{C}$ constant

ΔH_v heat of vaporization, Btu/lb (1 Btu/lb = 2.32×10^{3} J/kg)

$\mathscr{H}$ humidity, lb water/lb hydrogen (1 lb water/1 lb hydrogen = 2.2 kg water/2.2 kg hydrogen)

P pressure, lb/sq in. abs (1 lb/sq in. abs = 6.894×10^{3} N/sq m)

$\dot{Q}$ heat transfer, Btu/hr (1 Btu/hr = 0.293 J/sec)

T temperature, ^{o}F (1^{o} F = $\frac{9^{o}}{5}$ K - 459.67)

U_0 overall heat transfer coefficient, $\dot{Q}/(aD/\log E)$, Btu/(hr)(sq ft)(^{o}F) (1 Btu/(hr)(sq ft)(^{o}F) = 5.67 W/(sq m)(^{o}K))

$\dot{w}$ weight flow rate, lb/hr (1 lb/hr = 1.26×10^{-4} kg/sec)

Subscripts:

c condensing

g gas

gly glycol

H_2 hydrogen

H_2O water

i inlet

ℓm log mean

o outlet

s steam

MEASUREMENT AND CONTROL

All the equipment used to measure and/or control the parameters of interest (table II) was carefully calibrated prior to running the tests. This careful calibration was necessary to determine reliability limits of data obtained during the experiment. The methods of calibration are presented in this section.

TABLE II. - CONDENSER OPERATING PARAMETERS

Parameter	Measurement	Direct control
Inlet hydrogen flow rate	x	x
Inlet water vapor flow rate	x	x
Inlet gas temperature	x	x
Condenser pressure	x	x
Condenser gas pressure drop	x	
Outlet gas temperature	x	
Condensing rate	x	
Coolant flow rate	x	x
Coolant inlet temperature	x	x
Coolant outlet temperature	x	

Coolant Flow Rate

The coolant, a solution of 62.5 percent ethylene glycol and 37.5 percent water, was pumped from a constant-temperature bath through the condenser. A rotameter calibrated at two temperatures measured the flow rate (fig. 7), while a valve placed downstream of the condenser was used to make fine flow adjustments.

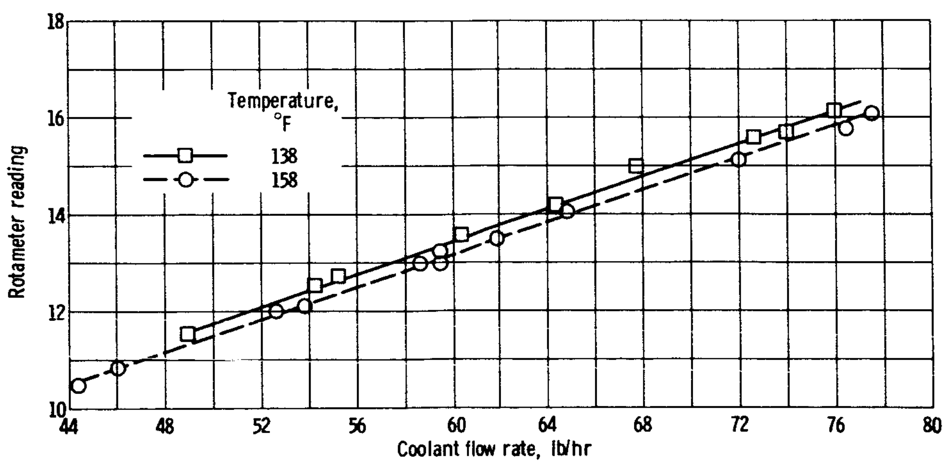


Figure 7. - Coolant flowmeter calibration.

Coolant Temperatures

All temperatures (coolant included) were measured by calibrated Chromel-Alumel or copper-Constantan thermocouples. For calibration, each thermocouple was immersed in a constant-temperature bath, and the bath temperature was read with a previously calibrated precision thermometer. A potentiometer or a digital voltmeter was then used to measure the thermocouple outputs. A typical calibration is shown in figure 8. Note that this figure can be read only to $\pm\frac{1}{2}^{\circ}$ F, whereas the calibrations were more precise.

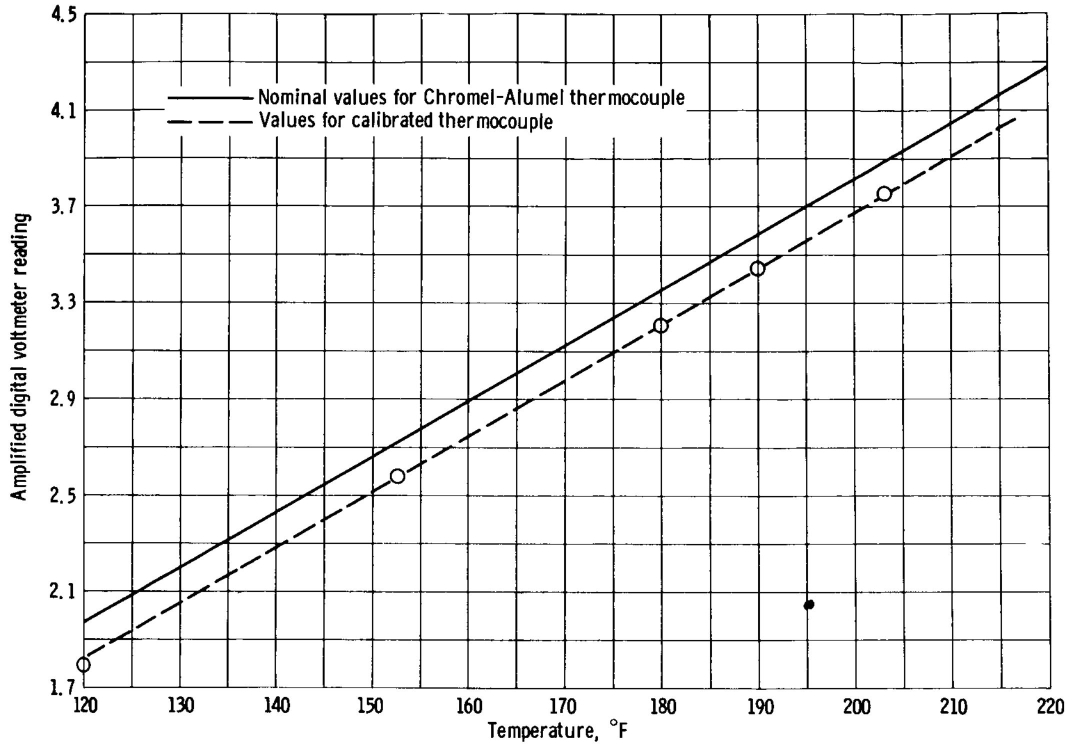


Figure 8. - Typical thermocouple calibration. Ice bath reference, 32° F.

TABLE III. - REPRESENTATIVE THERMOCOUPLE CALIBRATIONS

[Reference temperature, 32° F.]

Thermocouple	Temperature, °F		Ratio of voltage change to temperature change, mV/°F
	122	160	
	Thermocouple reading		
1	1.892	2.774	0.0233
2	1.899	2.782	.0232
3	1.873	2.753	.0233
4	1.873	2.757	.0232
7	1.882	2.760	.0231
8	1.864	2.739	.0230
9	1.863	2.774	.0232

The thermocouples used to measure coolant temperatures were then inserted into insulated lines carrying the coolant about 6 inches from the inlet and outlet fittings of the condenser (fig. 1, p. 2). The condenser was also insulated, and tests with both gas and coolant entering at equal temperatures showed no observable temperature drop in either stream. Finally, the difference in the coolant inlet-outlet thermocouple readings, taken directly from a digital voltmeter, was divided by a slope of 0.0232 millivolt per $^{\circ}$F to yield the coolant temperature change (table III).

Condensate Rate

The water condensing from the mixed hydrogen and water vapor stream was gravity-separated from the gas stream in an outlet header, collected in a beaker, and weighed on a torsion balance. The condensate was collected continuously during the 1/2-hour runs, and the samples weighed about 150 grams. The torsion balance is readable to 0.1 gram, and therefore, the condensate rate measurement error is less than 0.5 percent.

Gas Flow Rates

The flow rate of steam was controlled by a calibrated choked orifice. The flow rate through such an orifice depends only on the gas used, the orifice area, the gas temperature, and the pressure upstream of the orifice. In simplified form this relation can be written as

$$\dot{w}(\mathrm{lb/hr}) = \frac{P(\mathrm{lb/sq\ in.\ abs})\ A(\mathrm{sq\ in.})\ \mathscr{C}}{\sqrt{T}\,(^{\circ}\mathrm{R})} \tag{1}$$

where the constant $\mathscr{C}$ combines the orifice coefficient, a conversion factor, and a function of gas specific heats.

It was necessary to calibrate the steam-choked orifices by flowing superheated steam through them while condensing and weighing of condensate proceeded downstream. These calibration results were used to calculate A. Equation (1) was then used to calculate $\dot{w}$ at various upstream pressures and temperatures. In all the tests, the steam entering the orifice was superheated by at least 50° F.

The hydrogen flow rate was also controlled by a choked orifice, but was measured with a rotameter.

TABLE IV. - HEAT-TRANSFER

[Average heat loss,

Run	Power, W	Hydrogen flow rate, lb/hr	Water flow rate, lb/hr	Gas inlet temperature, ^{o}F	Gas outlet temperature, ^{o}F	Average gas temperature, ^{o}F	Heat capacity of hydrogen, Btu/lb	Heat capacity of water vapor, Btu/lb	Gas temperature change, $T_i - T_o$, ^{o}F	Condensate rate, lb/hr	Heat of vaporization, Btu/lb	Heat lost by hydrogen, Btu/hr	Heat lost by water vapor, Btu/hr
1	800	3. 41	3. 41	224	164	194	3. 43	0. 449	60	0. 440	998	702	90
2		3. 40	3. 40	223. 5	163	193			60. 5	. 538	998	705	90
3		3. 42	3. 40	223	162. 5				60. 5	. 585	999	710	90
4			3. 41	224	162				62	. 622		725	95
5			3. 41	223	162				61	. 630		715	95
6			3. 40	223	161. 5	192			61. 5	. 653		715	95
7	1000	3. 36	3. 72	244. 5	165. 5	205	3. 43	0. 450	79	. 74	996	910	130
8		3. 37	3. 72	244. 5	165				79. 5	. 76		915	135
9			3. 73	244					79	. 78		910	130
10			3. 73						79	. 80		910	130
11			3. 74						79	. 87		910	130
12			3. 73		164. 5	204			79. 5	. 83		915	135
13		3. 36	3. 73	244. 5	164	204			80. 5	. 78	997	930	135
14		3. 34	3. 74	243. 5	163	203			80. 5	. 79	997	920	135
15	1300	3. 34	4. 13	284. 5	166	225	3. 44	0. 451	118. 5	1. 04	994	1360	220
16		3. 34	4. 12	284	166. 5				117. 5	1. 06	994	1350	220
17		3. 34	4. 12	284. 5	165. 5				119	1. 12	995	1365	220
18		3. 36	4. 12	283	166				117	1. 07	994	1350	215
19		3. 33	4. 11	284	166. 5				117. 5	1. 17		1350	220
20		3. 34	4. 11	282. 5	167. 5				115	1. 12		1320	215
21		3. 33	4. 12	283	168	226			115	1. 03		1320	215
22		3. 33	4. 13	285	166. 5	226			118. 5	1. 12		1360	220
23		3. 33	4. 13	285	166	226			119	1. 07		1365	220
24	450	3. 38	3. 06	191. 5	163	177	3. 43	0. 448	28. 5	0. 37	999	330	40
25				191	163				28	. 34	999	325	
26				192. 5	162. 5				30	. 31	1000	348	
27				192	162				30	. 31	1000	348	
28	600	3. 42	3. 24	204	163. 5	194	3. 43	0. 449	40. 5	. 450	999	475	60
29		3. 42		203. 5	163	193			40. 5	. 465		475	
30		3. 40		205	162. 5	194			42. 5	. 480		495	
31		3. 40		204	162	193			42	. 470		490	
32		3. 41		203. 5	161. 5	193			42	. 495		490	
33	1000	3. 46	3. 12	236. 5	158	197	3. 43	0. 449	78. 5	. 62	1001	930	110
34		3. 48	3. 12	235. 5	157. 5	197			78	. 665	1001	930	
35		3. 49	3. 13	234. 5	157	196			77. 5	. 735	1001	930	
36		3. 50		234. 5	156	195			78. 5	. 785	1002	940	
37		3. 48		235. 5	155				80. 5	. 785		960	115
38		3. 49		235	154. 5				80. 5	. 83		960	115
39		3. 47	3. 12	235. 5	154				81. 5	. 78		970	115
40	1000	3. 24	4. 44	257. 5	174	216	3. 43	0. 450	83. 5	. 820	990	925	165
41		3. 21	4. 44	256. 5	174	216			82. 5	. 790		910	
42		3. 25	4. 45	256. 5	173. 5	215			83	. 785		925	
43		3. 23	4. 44	256	173	215			83	. 815		925	

40 Btu/hr.]

Heat of condensation, Btu/hr	Total gas heat loss	Coolant flow rate, lb/hr	Coolant inlet temperature, °F	Coolant outlet temperature, °F	Coolant temperature change (digital voltmeter readings), °F	Mean coolant temperature, °F	Heat capacity of glycol, Btu/lb	Total coolant heat gain	Gas inlet humidity, $\frac{\text{lb water}}{\text{lb hydrogen}}$	Temperature at start of condensation, °F	Average temperature for heat of vaporization, $\frac{(T_c + T_o)}{2}$
440	1230	50.0	155	184.5	28.7	170	0.809	1160	1.000	170.5	167
540	1335	50.0	153	184	30.8	169	.808	1245	1.00	170.5	167
585	1385	50.0	151	183.5	32.3	167	.807	1305	0.995	170	166
620	1440	49.9	150	183.5	33.4	167	.807	1345	.997		
630	1440	60.1	151	179	28.1	165	.806	1360	.997		
650	1460	74.9	153	176.5	23.0	165	.806	1390	.995		
740	1780	49.7	148.5	191.5	42.5	170	.809	1710	1.107	174.5	170
760	1810	49.6	147	191.5	43.4	169	.808	1740	1.104	174	
780	1820	49.5	146	191	44.4	169	.808	1775	1.107	174.5	
800	1840	49.4	144.5	190	45.0	167	.807	1775	1.107		
870	1910	59.0	147.5	187	38.7			1840	1.110		
830	1880	60.0	148	186	38.1			1845	1.107		
780	1845	59.5	148.5	185.5	37.4			1795	1.107		169
790	1845	75.2	149	179	29.6	164	.805	1795	1.120	175	169
1035	2615	49.7	139.5	204	63.5	172	.811	2560	1.236	~179	173
1055	2625	49.7	138.5	204.5	65.2	172	.811	2630	1.233	179	173
1115	2700	59.5	140	197.5	56.0	169	.808	2690	1.233		172
1065	2630	60.0	142	197.5	54.2	170	.809	2630	1.226		173
1165	2735	60.0	142	199.5	55.8	171	.810	2705	1.234		173
1115	2650	59.8	144	199	54.0	172	.811	2620	1.231		173
1025	2560	60.0	146	199	52.2	173	.811	2535	1.237		174
1115	2695	74.9	148	192.5	43.9	170	.809	2660	1.240		173
1065	2650	74.8	149	192.5	43.0	171	.810	2610	1.240		173
370	740	49.8	156	172.5	17.0	164	.805	682	0.905	166.5	165
340	715	59.6	157.5	170	13.2			634			165
310	700	75.2	158.5	169.5	10.5			636			164
310	700	74.5	158	169	10.5			630			164
450	935	50	153.5	176.5	23.0	165	.805	926	.947	168	166
465	990	49.5	152.5	176	23.6	164		940	.947	168	166
480	1035	50	151.5	176	24.5	164		986	.953	168.5	166
470	1020	60	152.5	173.5	20.6	163		995	.953	168.5	165
495	1045	76	153.5	170.5	16.9	162	.804	1035	.950	168.5	165
620	1660	50.5	141	182	40.3	162	.804	1635	.901	166	162
665	1705	49.8	139	181.5	42.3	160	.803	1690	.897		162
735	1775	49.6	136	181	44.3	159	.802	1775	.897		162
785	1835	49.6	134	180	45.7	157	.801	1815	.895		161
785	1860	59.6	136.5	176	39.6	156	.800	1890	.899		161
830	1905	74.0	138.5	171	32.5	155	.799	1920	.897		160
780	1875	74.0	139.5	171	31.3	155	.799	1855	.899		160
810	1900	50.4	154.5	201	45.9	178	.815	1885	1.370	~187	181
780	1855	50.0	155.5	201	45.7	178	.815	1860	1.382	187	181
775	1865	60.0	158.5	196	37.2	177	.814	1815	1.369	187	180
805	1895	75.2	160.5	191.5	30.7	176	.814	1880	1.374	187	180

TEST PROCEDURE

Tests to determine the coolant temperatures were run for five of the fuel-cell system power levels of table I (p. 4) (450, 600, 800, 1000, and 1300 W). Two off-design tests to determine the effect of changing the gas conditions by $\pm 10^\circ$ F at the 1000-watt level were also run. At each power level coolant flow rates of 50, 60, and 75 pounds per hour were used to determine the effect of this variable on performance. Each test was run in the following manner:

First the gas and coolant flow rates and temperatures entering the condenser were set and controlled and the water condensing rate was measured over a 1/2-hour period. During this 1/2-hour period measurements of temperatures and flow rates were taken at 5-minute intervals. If it was then determined that the condensing rate was not as required (table I(p. 4)), the coolant inlet temperature was changed accordingly. The results of these tests are presented in table IV (pp. 10 and 11).

CALCULATIONS

Heat Balance

To check the accuracy of the data obtained in this program (table IV), a heat balance for each run was calculated. The heat lost by the gas stream should equal the heat picked up by the coolant. From figure 9 and the heat capacity data of table V (from ref. 2), which in a temperature functional form are

$$C_{p,gly} = 6.67\times10^{-3}T + 0.696$$

$$C_{p,H_2O} = 1.18\times10^{-3}T + 7.89$$

$$C_{p,H_2} = 0.24\times10^{-3}T + 6.89$$

the heat picked up by the coolant stream is

$$\dot{Q}_{gly} = \dot{w}_{gly}\int_{T_{g,i}}^{T_{g,o}} C_{p,gly}\,dT$$

By integration

$$\dot{Q}_{gly} = \dot{w}_{gly}\left[0.696(T_{g,o} - T_{g,i}) + 3.33\times10^{-3}\left(T_{g,o}^2 - T_{g,i}^2\right)\right]$$

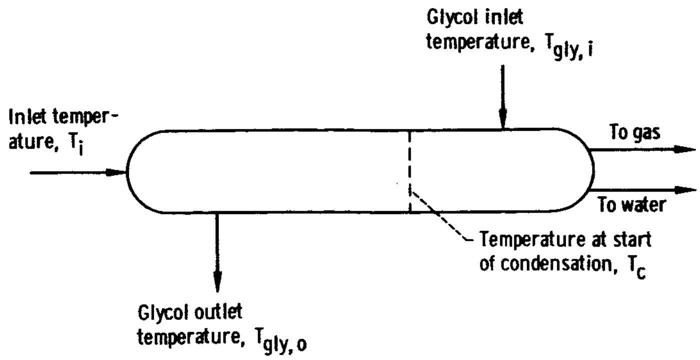


Figure 9. - Schematic diagram of condenser.

TABLE V. - HEAT CAPACITY DATA FOR HYDROGEN, WATER VAPOR, AND 62.5 PERCENT ETHYLENE GLYCOL

[Data from ref. 2.]

Temperature, °F	Glycol	Hydrogen	Water vapor
	Heat capacity		
	Btu/lb	Btu/(lb)(mole)(°F)	
50	0.730	----	----
100	.763	6.89	8.01
150	.795	----	----
200	.830	6.92	8.10
250	.863	----	----
300	.893	6.96	8.19

There are five terms for the heat lost by the gas:

(1) Condensation

$$\dot{w}_{H_2O,c}\,\Delta H_v$$

(2) Heat lost from T_i to T_c (where it is assumed that all condensation occurs at $(T_c + T_o)/2$)

$$\frac{\dot{w}_{H_2}}{2.016}\int_{\frac{T_o+T_c}{2}}^{T_i} C_{p,H_2}\,dT + \frac{\dot{w}_{H_2O}}{18.0}\int_{\frac{T_o+T_c}{2}}^{T_i} C_{p,H_2O}\,dT$$

(3) Liquid water

$$\dot{w}_{H_2O,c}(1.00)\,\frac{T_c + T_o}{2} - T_o$$

(4) Hydrogen after condensation

$$\frac{\dot{w}_{H_2}}{2.016}\int_{T_o}^{\frac{T_c+T_o}{2}} C_{p,H_2}\,dT$$

(5) Water vapor after condensation

$$\frac{\dot{w}_{H_2O} - \dot{w}_{H_2O,c}}{18.0}\int_{T_o}^{\frac{T_c+T_o}{2}} C_{p,H_2O}\,dT$$

Combining these heat-transfer terms with the heat capacities as functions of T gives

$$\dot{w}_{gly}\left[0.696(T_{g,o} - T_{g,i}) + 3.33\times10^{-3}\left(T_{g,o}^2 - T_{g,i}^2\right)\right]$$

$$= \frac{\dot{w}H_2}{2.016}\left[6.89(T_i - T_o) + 1.2\times10^{-4}\left(T_i^2 - T_o^2\right)\right] + \dot{w}_{H_2O,c}\ \Delta H_v$$

$$+ \dot{w}_{H_2O,c}\frac{1}{2}(T_c - T_o)$$

$$+ \frac{\dot{w}_{H_2O}}{18}\left[7.89(T_i - T_o) + 0.59\times10^{-3}\left(T_i^2 - T_o^2\right)\right]$$

$$- \frac{\dot{w}_{H_2O,c}}{18.0}\left[\frac{1}{2}(7.89)(T_c - T_o) + 0.59\times10^{-3}\left(\frac{T_c + T_o}{2}\right)^2 - T_o^2\right]$$

To determine whether any of these terms can be neglected, the detailed calculation for the run at 1000 watts and a coolant flow rate of 49.4 pounds per hour is carried out. For this run

$T_i = 244^\circ$ F	$\dot{w}_{H_2} = 3.37$ lb/hr
$T_o = 165^\circ$ F	$\dot{w}_{H_2O} = 3.73$ lb/hr
$T_{g,i} = 144.5^\circ$ F	$\dot{w}_{H_2O,c} = 0.80$ lb/hr
$T_{g,o} = 190^\circ$ F	$\dot{w}_{gly} = 49.4$ lb/hr
$T_{g,o} - T_{g,i} = 45.0^\circ$ F	$\Delta H_v = 996$ Btu/lb
$T_c = 174^\circ$ F	$\dfrac{T_c + T_o}{2} = 169.5^\circ$ F

which gives

$$49.4\left[0.696(45.0) + 3.33\times10^{-3}(190^2 - 144.5^2)\right]$$

$$\approx 1.67\left[6.89(244 - 165) + 1.2\times10^{-4}(244^2 - 165^2) + 0.80(996)\right]$$

$$+ 0.80\left(\frac{1}{2}\right)(174 - 165) + 0.207\left[7.89\,(244 - 165) + 0.59\times10^{-3}(244^2 - 165^2)\right]$$

$$- 0.0445\left[\left(\frac{1}{2}\right)(7.89)(174 - 165) + 0.59\times10^{-3}(169.5^2 - 166^2)\right]$$

Carrying out the calculation results in

$$1782 \approx 912 + 797 + 3.6 + 133 - 0.2 = 1844 \text{ Btu/hr}$$

This same result could be obtained by using a simplified formula

$$\dot{w}_{gly} C_{p,gly}\, \Delta T_{gly} = \dot{w}_{H_2O,c}\, \Delta H_v + \frac{\dot{w}_{H_2}}{2.016} C_{p,H_2}(T_i - T_o) + \frac{\dot{w}_{H_2O}}{18.0} C_{p,H_2O}(T_i - T_o)$$

where $C_{p,gly}$, C_{p,H_2O}, and C_{p,H_2} are taken at the mean coolant and gas temperatures, respectively.

This equation used with the following heat capacity values

$$C_{p,gly} = 0.696 + 0.105 = 0.801 \text{ Btu/lb}$$

$$C_{p,H_2O} = 7.87 + 0.24 = 8.11 \text{ Btu/(lb)(mole)}$$

$$C_{p,H_2} = 6.87 + 0.04 = 6.91 \text{ Btu/(lb)(mole)}$$

gives

$$49.4\ (0.801)(45.0) \approx 0.80(996) + 1.67(6.91)(79) + 0.187(8.11)79$$

$$1780 \approx 797 + 912 + 120 = 1829 \text{ Btu/hr}$$

Since this result approaches that previously obtained, this latter method for showing the heat balances was used in all the calculations. The final calculations show a consistent difference of about 40 Btu per hour, which can be taken as the average heat loss (table IV, pp. 10 and 11).

Heat-Transfer Coefficients

With the data in table IV, calculations of the overall heat-transfer coefficient were carried out in which

$$U_0 \frac{\text{Btu}}{(\text{hr})(\text{sq ft})(^{\circ}\text{F})} \triangleq \frac{\dot{Q}(\text{Btu/hr})}{a(\text{sq ft})\ \Delta T_{\ell m}}$$

$$\Delta T_{\ell m} \triangleq \frac{T_{g,i} - T_{gly,o} - (T_{g,o} - T_{gly,i})}{\ln \dfrac{T_{g,i} - T_{gly,o}}{T_{g,o} - T_{gly,i}}}$$

For these calculations $\dot{Q}$ was assumed to be the mean value of heat lost by gas and heat picked up by coolant while a is 1.32 square feet (fig. 5, p. 4). The results are presented in table VI and figures 10 to 12.

For all these runs, U_0 lies between 38.5 and 47.5 Btu per hour per square foot per $^{\circ}$F, with no observable trend with changing power level or gas flow rate. The average value of U_0 is 42.5±1.6 Btu per hour per square foot, with a slight increase in U_0 with increasing coolant flow rate.

TABLE VI. - CALCULATED HEAT-TRANSFER COEFFICIENTS FOR SELECTED RUNS

Run	Heat transfer, $\dot{Q}$, Btu/hr	Temperature difference, $T_{g,i} - T_{gly,o}$, b, ^oF	Temperature difference, $T_{g,o} - T_{gly,i}$, c, ^oF	D, b - c, ^oF	E, b/c	ln E	D/ln E	Overall heat-transfer coefficient, U_0
4	1392	40.5	12	28.5	3.37	1.215	23.5	43.2
5	1400	44	11	33	4.00	1.388	23.7	43.5
6	1415	46.5	8.5	38	5.47	1.700	22.4	47.5
10	1810	54	21.5	32.5	2.51	.920	35.3	38.9
13	1825	59	15.5	43.5	3.80	1.335	32.5	42.6
14	1820	64.5	14	50.5	4.60	1.525	33.1	41.7
16	2695	79.5	28	51.5	2.83	1.040	49.6	41.3
21	2550	84	22	62	3.81	1.339	46.3	41.7
23	2630	92.5	17	75.5	5.44	1.692	44.6	44.7
24	710	19	7	12	2.71	.997	12.0	44.7
25	670	21	5.5	15.5	3.82	1.340	11.6	43.7
27	665	23	4	19	5.76	1.751	10.8	46.6
30	1005	29	11	18	2.63	.967	18.6	41.0
31	1010	30.5	9.5	21	3.21	1.168	18.0	42.5
32	1040	33	8.0	25	4.13	1.42	17.6	44.7
36	1825	54.5	22	32.5	2.47	.905	35.9	38.5
37	1875	59.5	18.5	41	3.22	1.170	34.7	41.0
39	1865	64.5	17	47.5	3.79	1.344	35.3	40.1
41	1860	55.5	18.5	37.5	3.00	1.100	34.1	41.4
42	1840	60.5	15	45.5	3.90	1.360	33.5	41.6
43	1890	64.5	12.5	52.5	5.16	1.642	32.0	44.7

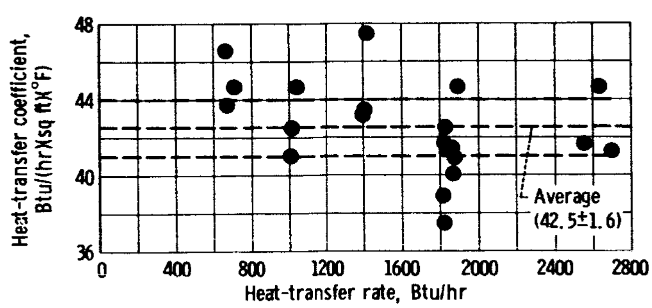


Figure 10. - Overall heat-transfer coefficient as function of heat-transfer rate.

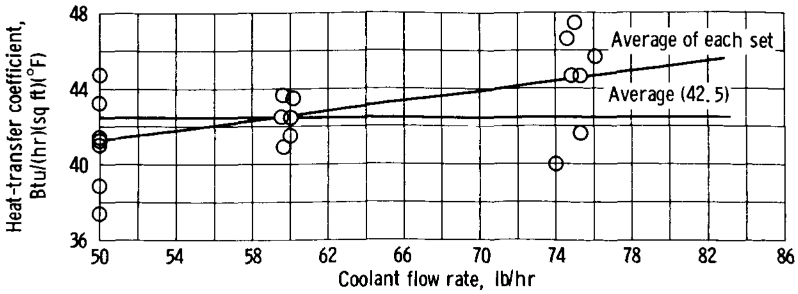


Figure 11. - Overall heat-transfer coefficient as function of coolant flow rate.

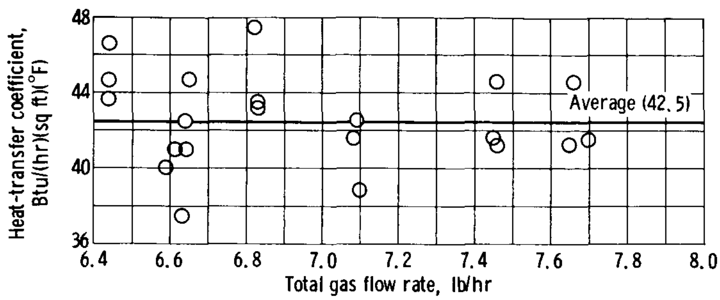


Figure 12. - Overall heat-transfer coefficient as function of total gas flow rate.

ERROR ANALYSIS

Heat Balance

To show the magnitudes of the errors present in the heat balance calculations, it is necessary to know the measurement errors in all the measured variables.

The coolant $\Delta T(T_{gly,o} - T_{gly,i})$ is known to $\pm 0.2^{\circ}$ F since these thermocouples were calibrated to 0.1° F. The thermocouples used to measure the gas temperatures (inlet and outlet) were read to an accuracy of $\pm 0.5^{\circ}$ F. The rotameter used to measure the coolant flow rate can be read to ± 0.05 scale reading; the error in $\dot{w}_{gly}$ is therefore ± 0.3 pound per hour. Since the steam-choked orifice was calibrated directly and the only possible error is the scatter of the calibration data, $\dot{w}_s$ is correct to within a maximum of ± 1 percent. Now the only unknown is the error in $\dot{w}_{H_2}$, the hydrogen flow rate. This error can be estimated from the measured condenser gas exit temperatures.

Since there is only one saturation humidity value (lb water vapor/lb hydrogen) for a measured outlet temperature (fig. 13), and if it is assumed that the gas and condensate flow rates are known and the mass balance can be used to calculate an exit humidity, a simple comparison of the two values can be made for each run (table VII). Then, since

$$\mathscr{H} = \frac{\dot{w}_{H_2O}}{\dot{w}_{H_2}}$$

and by differentiation

$$d\mathscr{H} = \frac{\dot{w}_{H_2}\left(d\dot{w}_{H_2}\right)\left(d\dot{w}_{H_2O}\right) + \dot{w}_{H_2O}\left(d\dot{w}_{H_2}\right)}{\dot{w}_{H_2}^2}$$

$d\mathscr{H}$ could be estimated.

From table VII the average value of $\mathscr{H}$ calculated by a mass balance minus $\mathscr{H}$ read from figure 4 (p. 3) is 0.016, which is $d\mathscr{H}$. The average values for $\dot{w}_{H_2}$, $\dot{w}_{H_2O}$, and 1 percent of $\dot{w}_{H_2O}$ are 3.37, 3.74, and 0.037, respectively. Therefore,

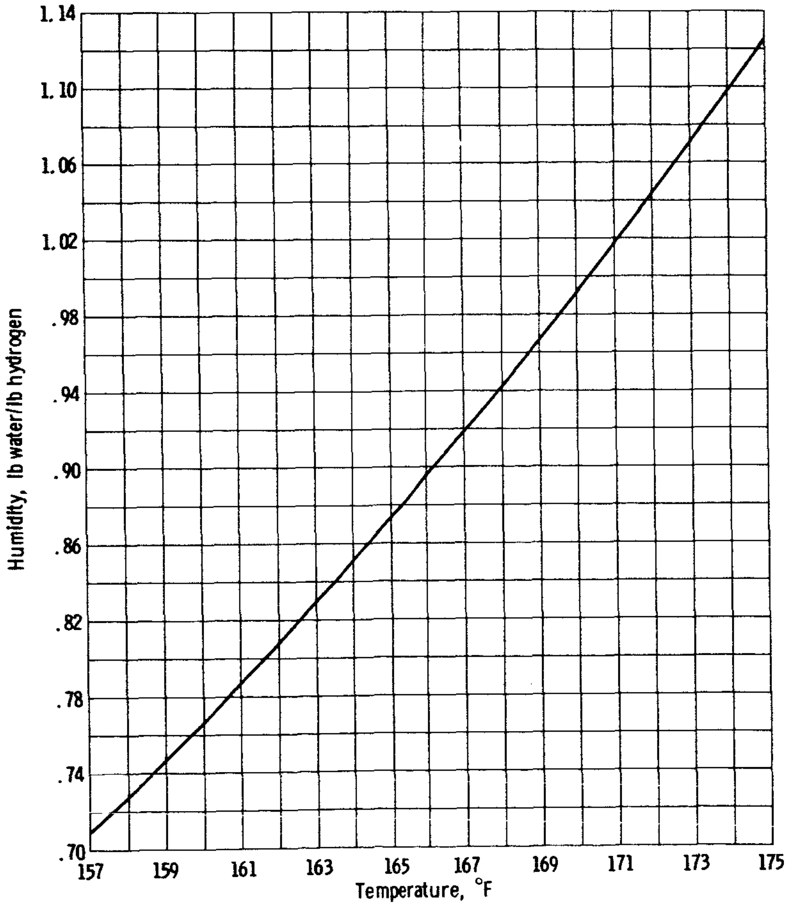


Figure 13. - Saturation curve; humidity (lb water/lb hydrogen) at saturation as function of temperature. Pressure, 60 pounds per square inch absolute.

TABLE VII. - CALCULATION OF EXIT HUMIDITIES AND

ESTIMATION OF HYDROGEN FLOW FOR RUNS 1 TO 20

Flow rate of water, lb/hr	Flow rate of hydrogen, lb/hr	Condensing rate, lb/hr	Humidity calculated from mass balance	Outlet temperature, ^{o}F	Humidity for outlet temperature	Difference between calculated and measured humidity
3.41	3.41	0.44	0.871	164	0.849	0.032
3.40	3.40	.54	.841	163	.828	.013
3.40	3.42	.59	.822	162.5	.817	.005
3.41	3.42	.62	.817	162	.806	.011
3.41	3.42	.63	.813	162	.806	.007
3.40	3.42	.65	.805	161.5	.796	.009
3.72	3.36	.74	.887	165.5	.883	.004
3.72	3.37	.76	.878	165	.872	.006
3.73	3.37	.78	.875	165	.872	.003
3.73	3.37	.80	.870	165	.872	.002
3.74	3.37	.87	.852	165	.872	.020
3.73	3.37	.83	.861	164.5	.860	.001
3.73	3.36	.78	.878	164	.850	.028
3.74	3.34	.79	.883	163	.828	.055
4.13	3.34	1.04	.925	166	.894	.031
4.12	3.34	1.06	.916	166.5	.906	.010
4.12	3.34	1.12	.898	165.5	.883	.015
4.12	3.36	1.07	.908	166	.894	.014
4.11	3.33	1.17	.882	166.5	.906	.024
4.11	3.34	1.12	.896	167.5	.930	.036

$$d\dot{w}_{H_2} \sim \frac{0.016(11.3) + 3.37(0.037)}{3.74}$$

$$d\dot{w}_{H_2} \sim \frac{0.18 + 0.12}{3.74} \sim 0.08 \text{ lb/hr}$$

which is approximately 2 percent of 3.37 pounds per hour.

An error of ±2 percent in the hydrogen flow rate can be assumed. The heat picked up by the glycol is $\dot{w}_{gly} C_{p,gly}(T_{g,o} - T_{g,i})$; call this $\dot{Q}_{gly}$(Btu/hr). Then

$$d\dot{Q}_{gly} = \dot{w}_{gly} C_{p,gly}\, d(\Delta T) + (d\dot{w}_{gly}) C_{p,gly}\, \Delta T + \dot{w}_{gly}\, \Delta T (dC_{p,gly})$$

which for the sample calculation carried out previously (p. 15) is

$$d\dot{Q}_{gly} = (49.4)(0.801)(0.2) + (0.3)(0.801)(50) + (49.4)(50)(0.000)$$

where no error in $C_{p,gly}$ is assumed. The value of $d\dot{Q}_{gly}$ is 20 Btu per hour, or 1 percent of 1782 Btu per hour. For the gas stream,

$$\dot{Q}_g = \dot{w}_{H_2O,c}\,\Delta H_v + \frac{\dot{w}_{H_2}}{2.016} C_{p,H_2}(T_i - T_o) + \frac{\dot{w}_{H_2O}}{18.0} C_{p,H_2O}(T_i - T_o)$$

Again,

$$d\dot{Q}_g = \Delta H_v\left(d\dot{w}_{H_2O}\right) + \frac{d\dot{w}_{H_2}}{2.016} C_{p,H_2}(T_i - T_o) + \frac{\dot{w}_{H_2}}{2.016} C_{p,H_2}(dT_i + dT_o)$$

$$+ \frac{d\dot{w}_{H_2O}}{18} C_{p,H_2O}(T_i + T_o) + \frac{\dot{w}_{H_2O}}{18} C_{p,H_2O}(dT_i + dT_o)$$

where C_p and ΔH_v are known constants. When $d\dot{w}_c$ is 0.5 percent of $\dot{w}_c$, $d\dot{Q}_g$ becomes, for the sample,

$$d\dot{Q}_g = 0.996(0.004) + \frac{0.063}{2.016}(6.91)(79) + \frac{3.37}{2.016}(6.91)(1) + \frac{0.037}{18.0}(8.11)(79) + \frac{3.73}{18}(8.11)(1)$$

or,

$$d\dot{Q}_g = 4 + 17 + 11 + 1 + 1 = 34 \text{ Btu/hr}$$

which is about 2 percent of 1828.

It then can be assumed that the calculated coolant heat-transfer rate is correct to about 1 percent; for the gas it is about 2 percent in error.

Overall Heat-Transfer Coefficients

An estimate of the error inherent in the calculation of U_o, the overall heat-transfer coefficient, can be obtained in a simplified manner by changing each term according to its maximum deviation and then recalculation of U_0:

$$U_0 = \dot{Q}/(a\,\Delta T_{\ell m})$$

$$U_0 = \frac{\frac{\dot{Q}}{a} \ln \frac{T_{g,i} - T_{gly,o}}{T_{g,o} - T_{gly,i}}}{T_{g,i} - T_{gly,o} - (T_{g,o} - T_{gly,i})}$$

If the values used in the previous calculation (p. 15) are used, the results are

$$\dot{Q} = 1805 + 2 \text{ percent of } 1805 => 1841 \text{ Btu/hr}$$

$$a = 1.32 - 1 \text{ percent of } 1.32 => 1.31 \text{ sq ft}$$

$$T_{g,i} = 244 + 0.5 = 244.5^{\circ} \text{ F}$$

$$T_{gly,o} = 190 - 0.5 = 189.5^{\circ} \text{ F}$$

$$T_{g,o} = 165 - 0.5 = 164.5^{\circ} \text{ F}$$

$$T_{gly,i} = 144.5 + 0.5 = 145^{\circ} \text{ F}$$

With these values U_0 becomes equal to

$$\frac{\frac{1841}{1.31} \ln \frac{55}{19.5}}{35.5} = \frac{(1841)(1.04)}{(1.31)(35.5)} = 41.1 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F})$$

which is 2.2 Btu per hour per square foot per $^{\circ}$F higher than the previously calculated value of 38.9 (run 10, table VI, p. 17)). Under the conditions of this simple approximation, U_0 is thus correct to within ±5 percent.

RESULTS AND DISCUSSION

Analysis of the data presented in table IV (pp. 10 and 11) shows that, in the expected region of condenser operation, lines of coolant inlet and outlet temperature are approximately linear with respect to heat-transfer rate (fig. 14). A plot such as figure 14 can be used to determine coolant temperatures at any heat-transfer rate within the tested range, and could be used to some extent for extrapolation to higher heat loads. It should be noted that the gas inlet temperature and the gas component flow rates are based on nominal fuel-cell conditions and thus are functions of power output (fig. 15). If other, off-

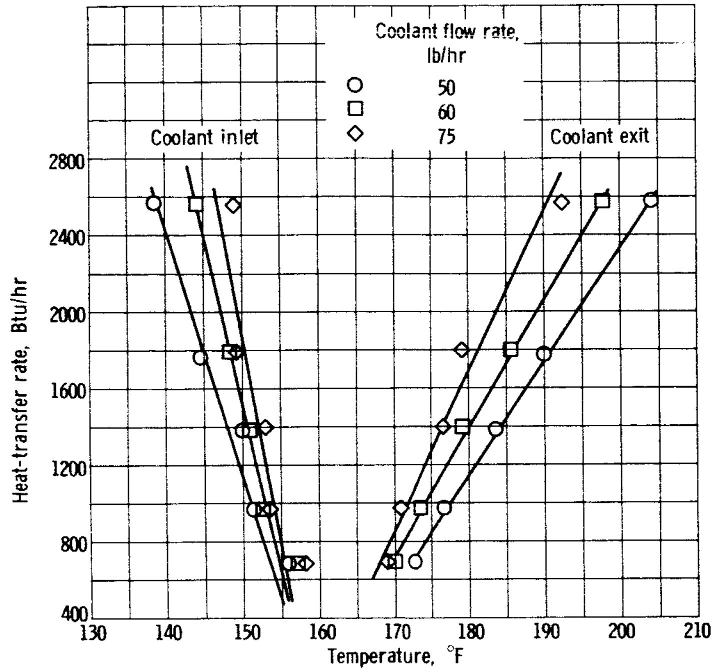


Figure 14. - Apollo condenser inlet and outlet coolant temperatures at suggested heat-transfer rates.

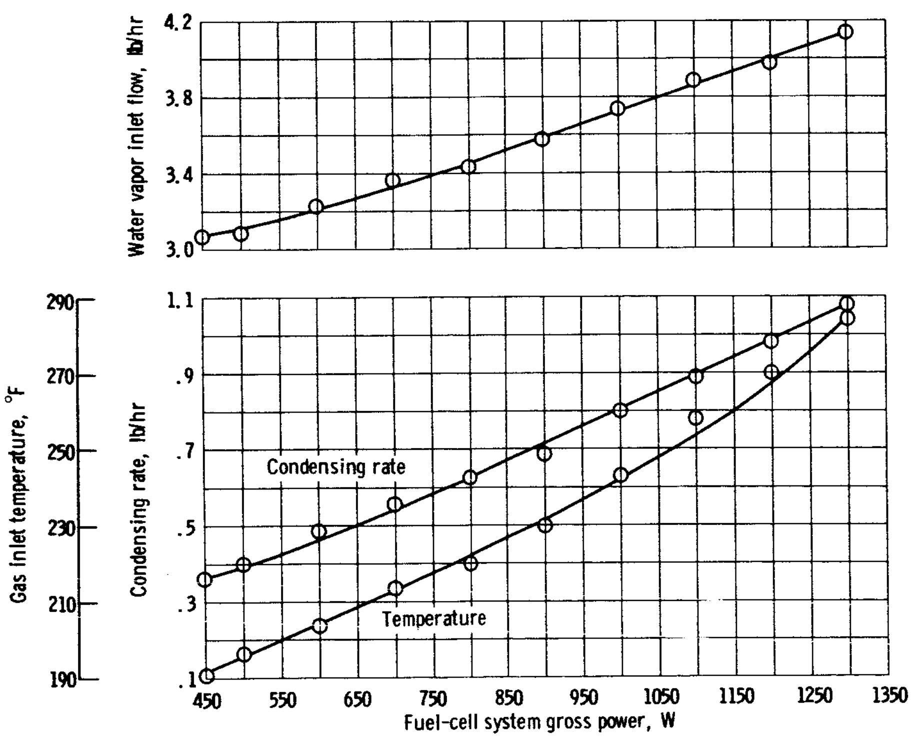


Figure 15. - Nominal condenser gas conditions as function of power level.

nominal, conditions are experienced, some other means of determining the coolant temperature is needed.

For these off-design conditions the average value of the overall heat-transfer coefficient can be used. The data obtained for the three sets of runs at the 1000-watt power level (table IV; runs 7 to 14, 33 to 43 (pp. 10 and 11)) show that for a 10^{o} F change in the average condenser temperature, this overall heat-transfer coefficient remains constant. These runs were set up so that the inlet and outlet gas temperatures changed approximately 10^{o} F before the coolant temperatures were measured (table VIII). When the gas temperatures change by $\pm 10^{o}$ F, the coolant temperatures also change by $\pm 10^{o}$ F, respectively (table IX).

The calculated values of U_0 were remarkably constant under all the test conditions (table VI, p. 17); the average value was about 42.5 Btu per hour per square foot per ^{o}F. There was some slight increase in U_0 with increasing coolant flow rate (fig. 10, p. 17), but no trend was observed with either increasing gas flow rate or total heat-transfer rate. These results are consistent with the data presented in reference 3 for compact heat exchangers operating in the low Reynolds number flow region.

TABLE VIII. - RESULT SUMMARY FOR RUNS AT 1000 WATTS

Run	Approximate coolant flow rate, lb/hr	Overall heat-transfer coefficient, Btu/(hr)(sq ft)(^{o}F)
36	50	38.5
10	50	38.9
41	50	41.4
37	60	41.0
13	60	42.6
43	60	44.7
39	75	40.1
14	75	41.7
43	75	44.7

TABLE IX. - CHANGES IN COOLANT TEMPERATURE CORRESPONDING TO GAS TEMPERATURE CHANGES

Run	Gas inlet temperature, ^{o}F	Gas outlet temperature, ^{o}F	Coolant inlet temperature, ^{o}F	Coolant outlet temperature, ^{o}F
36	234.5	156	134	180
10	244	165	144.5	190
41	256.5	174	155.5	201
37	235.5	155	136.5	176
13	244.5	164	148.5	185.5
42	256.5	173.5	158.5	196
39	235.5	154	139.5	171
14	243.5	163	149	179
43	256	173	160.5	191.5

CONCLUSIONS

Heat-transfer tests on an Apollo fuel-cell condenser at design conditions showed that for a gross power output from 450 to 1300 watts, there is a decreasing linear trend in the needed coolant inlet temperatures with increasing fuel-cell waste heat production. The coolant temperatures within this range can therefore be predicted.

The needed coolant temperatures and the overall condenser performances at off-nominal conditions can be predicted by noting that the overall heat-transfer coefficient remains essentially constant and that coolant temperature changes follow gas temperature changes exactly, at least at the 1000-watt power level.

Lewis Research Center
National Aeronautics and Space Administration,
Cleveland, Ohio, June 28, 1966,
123-34-02-01-22.

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