N86-14094

An Analysis of Cryotrap Heat Exchanger Performance Test Data (400 Area) and Recommendations for a System to Handle Apollo RCS Engines

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Contents

A. INTRODUCTION

The attached photograph shows the current arrangement of a Platecoil heat exchanger which uses LN_2 on the inside of parallel tubes, in counter flow to the test cell engine exhaust gases which are drawn through a box surrounding the plates by the existing vacuum blowers. As a result of inadequate performance and special test data it was decided to redesign the system to accommodate an Apollo RCS engine.

B. CHARACTERIZATION OF EXISTING SYSTEM

Test LN_2 - 3 (TR-329-002) was used to determine tube and shell side heat transfer coefficients for the system (Table I). At the 68 second mark it should be noted that only sensible heat is transferred, i.e., at an exit temperature of 35°F no ice formation would be occurring since the H_2O vapor pressure in the bulk gas phase is higher than in the inlet vapor stream. During this test the LN_2 rate was 2.5 gpm on the average. The Platecoil catalogue 5-63 indicates that each tube is about 1" in I.D. with a wall thickness of .083." There are 84 tubes, each 47" in length.

At a cell pressure of .2079 psia (68 second mark) Altitude Prediction methods (TR-329-001) yield a rate of 91.5 scfm through the exchanger. Since there are 84 tubes this would be about 1 scfm per tube.

The composition of the gases at the heat exchanger inlet was taken to be:

		Volume %
H_2	-	8.3
N_2	-	33.3
CO_2	-	6.8
CO	-	9.2
H_2O	-	40.1
CH_4	-	2.1
		99.8

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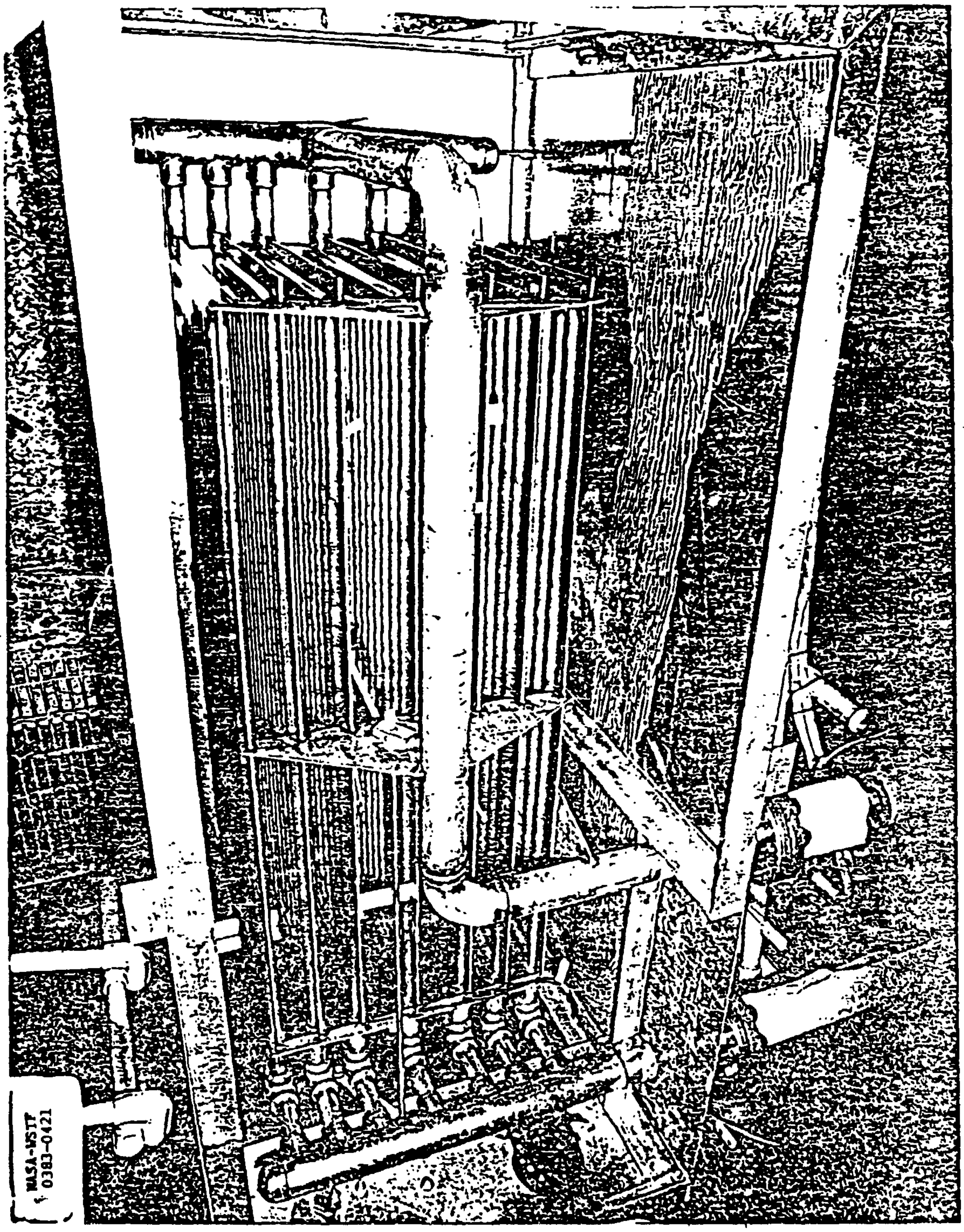

TABLE I.

TEST NO.: LN_2 - 3 DATE: 4-1-83

FIRING PROFILE: Steady State (68 sec) (0.37 lbs mass/sec)

BLOWERS IN OPERATION: 4 Sets (West and South)

		TIME (SECONDS)				
		-10	20	40	60	68
4000R	Heat Exchanger Inlet Flowrate (gpm)	-----	-----	-----	-----	-----
4001T	Heat Exchanger Inlet Temp (°F)	-316	-317	-314	-312	-312
4002T	Heat Exchanger Outlet Temp (°F)	-313	-316	-314	-312	-311
4003T	Cold Plate A - Inlet Skin Temp (°F)	-288	-261	-213	-166	-148
4004T	Cold Plate A - Middle Skin Temp (°F)	-230	-209	-179	-150	-140
4006T	Cold Plate A - Outlet Skin Temp (°F)	-273	-251	-225	-198	-189
4011T	Cold Plate C - Inlet Skin Temp (°F)	-293	-268	-223	-182	-166
4012T	Cold Plate C - Middle Skin Temp (°F)	-255	-242	-213	-183	-172
4014T	Cold Plate C - Outlet Skin Temp (°F)	-295	-272	-243	-214	-203
4015T	Heat Exchanger Gas Inlet Temp (°F)	97	357	433	430	424
4017T	Heat Exchanger Gas Inlet Temp (°F)	91	362	438	439	433
4018T	Heat Exchanger Gas Outlet Temp (°F)	- 92	- 11	9	30	35
4020T	Heat Exchanger Gas Outlet Temp (°F)	-141	- 8	6	23	36
4021P	Heat Exchanger Outlet Pressure (Baratron) (psia)	.0037	.083	.142	.189	.1996
5032P	Chamber (Heat Exchanger Inlet) Pressure (Baratron) (psia)	.0043	.086	.146	.192	.2074
	Cell Altitude (ft x 1000)	192.5	114.5	101.8	95.6	93.8
5035T	North Chamber Outlet Temp (°F)	108	135	169	178	183
5048T	Right North Blower Inlet Temp (°F)	------	------	------	------	-----
5049T	Left North Blower Inlet Temp (°F)	------	------	------	------	-----
5036T	South Chamber Outlet Temp (°F)	73	130	162	181	185
5050T	Right South Blower Inlet Temp (°F)	79	110	128	138	140
5051T	Left South Blower Inlet Temp (°F)	80	110	129	141	143
5037T	West Chamber Outlet Temp (°F)	- 51	3	33	61	73
5052T	Right West Blower Inlet Temp (°F)	67	69	75	80	82
5053T	Left West Blower Inlet Temp (°F)	59	65	73	80	82
	Predicted Cell Pressure (psia)	------	.0662	.1187	.1675	.1888
	Predicted Cell Altitude (ft x 1000)	------	120.9	106.7	98.7	96.0

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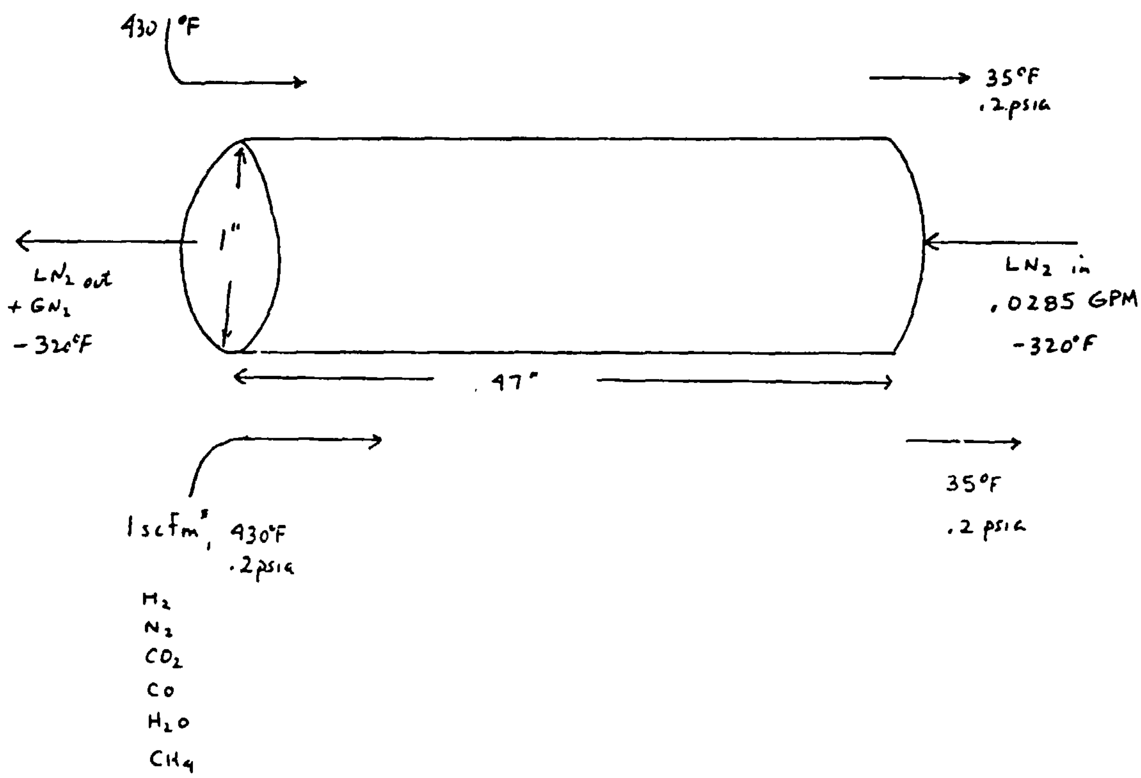
It should be noted that at 1 scfm per tube, with 40% H_2O by volume in the inlet gas, the heat load to freeze all the water would be 22.4 BTU/min. At an LN_2 rate of 2.5 gpm (.0285 gpm/tube) there is only 17 BTU/min available on the N_2 side if a constant T of -320°F is to be maintained, i.e., there is inadequate LN_2 to do the job of removing water and sensible heat.

The conditions for characterizing the system are summarized in the following sketch:



* unreacted N_2O_4 + MMH
not considered in analysis

The rate of heat transfer Q (BTU/hr) for this case, neglecting wall and ice resistances, is given by:

$$Q = \frac{2 \quad L \quad (T_i - T_o)}{\frac{1}{Rh_i} + \frac{1}{Rh_o}}$$

Where L = length of tube

T_i = inside bulk temperature

T_o = outside bulk temperature

R = Radius of tube

h_i = inside coefficient of heat transfer

h_o = outside coefficient of heat transfer

The rate of heat transfer is also given by:

$$Q = \dot{m}\, C_p\, \Delta T = \text{(gas flowrate) (heat capacity) } (430\text{-}35^{o}F)$$

since only sensible heat is transferred from the gas side.

So $$Q = 1 \frac{\text{scfm}\ (ft^3)}{\text{MIN}} \times \frac{1 \text{ lb mole}}{359 \ ft^3} \times 7.5 \frac{\text{BTU}}{\text{lb mole } ^{o}F} (395^{o}F)$$

$$Q = 8.25 \frac{\text{BTU}}{\text{min}} = 495 \frac{\text{BTU}}{\text{hr}}$$

On the outside we have forced convection heat transfer, therefore, as an approximation, we can employ a correlation for flow normal to a cylinder (see attached Figure from McAdams). There are eight channels in the heat exchanger, therefore, dividing the flow by eight gives us a velocity of 1820 ft/min past the cylinders. Correcting for temperature and pressure (we're at .2 psi and 230°F on the average as opposed to 14.7 psi and 32°F for standard conditions) gives a value of .00054 lb/ft^3 for the gas density resulting in a Reynold's number of approx. 100. This gives a Nusselt number, Nu, of 5 and an outside heat transfer coefficient of 1.2 BTU/hr ft^2 °F.

Using a mean outside temperature of 230°F we get:

$$h_i = 3.36 \text{ BTU/hr ft}^2 \text{ °F}$$

As a check we can compute the inside coefficient from a forced convection correlation for flow inside a tube:

$$h_i = \frac{k}{D} (.023) (Re)^{.8} (Pr)^{.4}$$

At a Reynold's number of 1317 we get:

$$h_i = 11.5 \text{ BTU/hr ft}^2 \text{ °F}$$

Remember, however, that this correlation is for turbulent flow, therefore, it's not unusual to get a higher number for h_i since our Reynold's number is only 1317. Therefore, an inside coefficient of 3.36 is realistic.

Before we use this information to redesign the system it is worth mentioning that in forced convection boiling (see attached notes) the heat transfer coefficient will rise significantly and then fall off again as one goes down the tube. This might explain the fact (see LN_2 -3 test data) that the skin T on the outside of the tube is highest at the middle for most of the run.

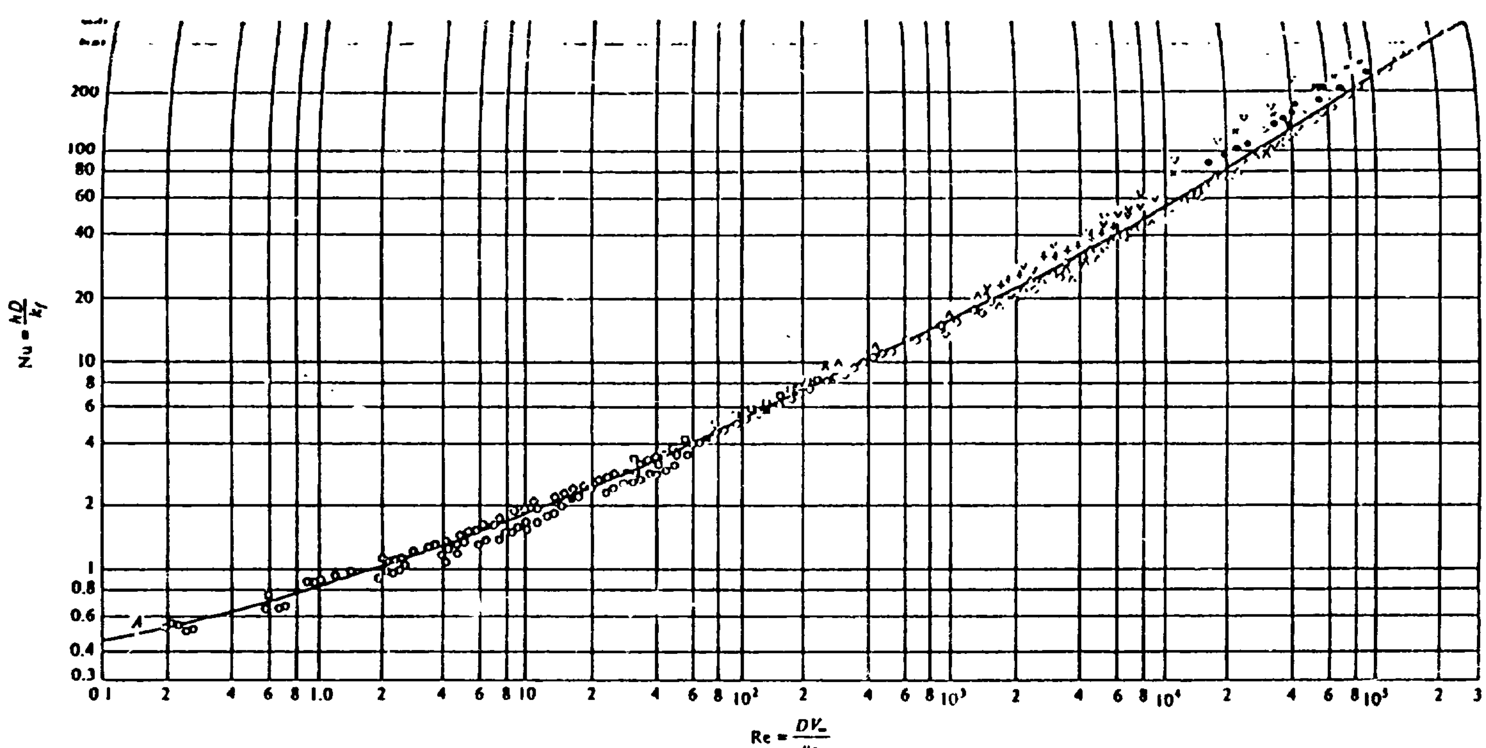


Figure 4-11 Average Nusselt number versus Reynolds number for a circular cylinder in air, placed normal to the flow. (SOURCE: W. H. McAdams, *Heat Transmission*, 3rd ed. McGraw-Hill, New York, 1954) D = diameter of cylinder, V_∞ = fluid full stream velocity, ν_f = fluid kinematic viscosity, h = heat transfer film coefficient, k_f = fluid thermal conductivity, Re = Reynolds number, Nu = Nusselt number

Notes

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10.2.4 Forced-Convection Boiling

Forced-convection boiling is usually associated with boiling from the inner surface of a heated tube through which a liquid is flowing. Bubble growth and separation are strongly influenced by the flow velocity, and hydrodynamic effects are significantly different than those corresponding to pool boiling. The process is complicated by the existence of different two-phase flow patterns that preclude the development of generalized theories.

Consider flow development in the heated tube of Figure 10.5. Heat transfer to the subcooled liquid which enters the tube is initially by forced convection and may be predicted from the correlations of Chapter 8. However, boiling is soon initiated, with bubbles appearing at the surface growing and being carried into the mainstream of the liquid. There is a sharp increase in the convection heat transfer coefficient associated with this *bubbly flow regime*. As the volume fraction of the vapor increases, individual bubbles coalesce to form plugs or slugs of vapor. This *slug-flow regime* is followed by an *annular-flow* regime in which the liquid forms a film. This film moves along the inner surface, while vapor moves at a faster velocity through the core of the tube. The heat transfer coefficient continues to increase through the *bubbly flow* and much of the *annular-flow* regimes. However, dry spots eventually appear on the inner surface,

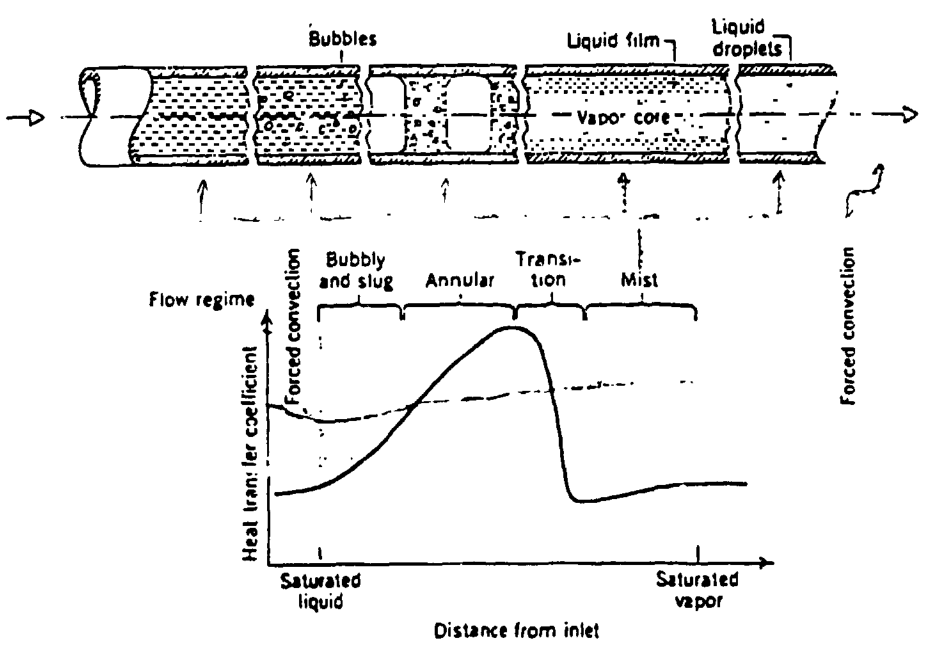


Figure 10.5 Flow regimes for forced-convection boiling inside a tube.

Cryo Panels 47"X 22"
7 panels in Assy

LEMSCO 3/9/83

GOLD TRAP BOX

FRAME : 2 ALUM L, WELDED

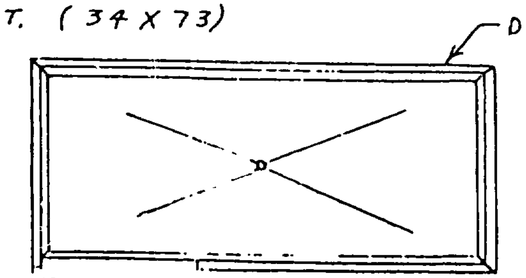
SIDES : SHEET ALUM/SS POP RIVETED TO FRAME

- A - 30 X 13 APPROX.
- B - 69 X 36 WITH 25 1/4 X 8 1/4 CORNER CUTOUT
- C - 30 X 36 WITH 15 1/4 DIA HOLE (DETAIL →)
- D - 69 X 36 APPROX.

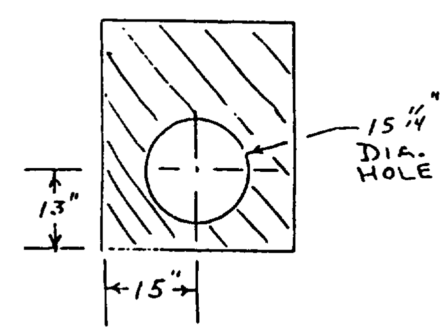
TOP : PLEXIGLASS SHT. (34 X 73)

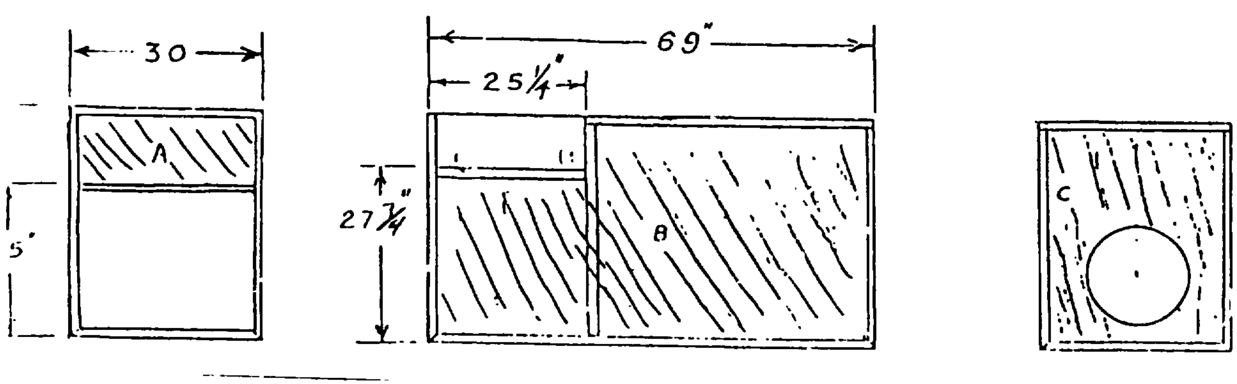
BOTTOM : S.S. DRIP PAN WITH AN424-16 WELDED IN CTR ON OUTSIDE SURFACE

Note: Do Not USE ANY GALVINIZED COMPONENTS!

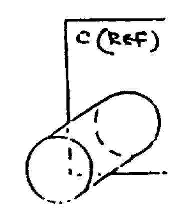


C





FAB A 23 X 15 1/4 O.D. SHEET ALUM DUCT STICK THRU "C" AND SECURE TO PAN



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C. DESIGN OF A HIGHER CAPACITY SYSTEM

The configuration chosen for the future modified system according to I. D. Smith will be three heat exchangers per test cell each in line with two blowers. At any time two heat exchangers will be utilized while the third is being regenerated due to ice buildup. Therefore, half the total gas flow from an Apollo RCS will go through each exchanger. As a result, we now have to design for 2 scfm per tube. Also we'll set the outlet gas temperature at 32°F and provide enough LN_2 to theoretically freeze out 1/2 the H_2O entering the exchanger.

The amount of H_2O to be theoretically removed is:

$$\frac{1}{2}\ (.4)\ (2\ \text{scfm})\ \left(1\ \frac{\text{lb mole}}{359}\right) = .00111\ \frac{\text{lb moles}}{\text{min}}$$

$$.00111\ \frac{\text{lb mole}}{\text{min}}\ H_2O \times 1117\ \frac{\text{BTU}}{\text{lb}} \times \frac{18\ \text{lb}}{1\ \text{lb mole}} = 22.3\ \frac{\text{BTU}}{\text{min}}$$

The sensible heat load is ~ 9.4 BTU/min. With a total heat load of 32 BTU/min we need about twice the previous LN_2 rate, i.e., we now need .0570 gpm of LN_2 per tube or 5 gpm for the entire exchanger.

Based on the previous discussion of forced convection boiling and a higher LN_2 rate it's probable that we'll have a higher h_i this time, however, we'll design conservatively and use the same h_i and h_o.

With a load of 32 BTU/min (4 times as great as the previous case) we need 4 times the original length.

In summary, our conservative estimate yields, for the same configuration:

a) 4 x the existing length

b) 5 gpm LN_2

Lastly, it is worth looking at whether a finned tube arrangement will provide any advantages. According to Ludwig (Applied Process Design for the Chemical Industries, Vol. 3) "Economically, the outside coefficient should be about 1/5 or less than the inside to make a finned unit look attractive." They have a general chart which gives the reduction in number of tubes on the ordinate and the following expression on the abscissa:

$$\frac{1}{\frac{1}{h_i} + r_i} \Big\backslash \frac{1}{\frac{1}{h_o} + r_o}$$

An abscissa value of 2.38 shows a 20% reduction in the number of tubes required with fins employed. Therefore, it is recommended that the current non-fin configuration be utilized.

D. RECOMMENDATIONS

1) Relocate exchanger outside test stand with a stronger box.

2) Provide 5 gpm of LN_2 per exchanger

3) Use the identical units now in use only add more units to the system in series if necessary. In this regard it would be best to start with one exchanger at the higher LN_2 flow and observe the performance of the system. If insufficient keep adding modules in series until the required performance is met.

4) As previously indicated, one exchanger will be regenerated due to ice buildup while the other two are in operation. Data taken by A. Rakow on 5-27-83 at Test Stand 400 indicates a linear buildup of ice at the inlet section of the exchanger. In fact, toward the end of the run (Vernier RCS) the off center left and right side plate buildup was close to occluding the flow, i.e., regeneration is important.

ADDENDUM

In addition to determining the heat exchanger configuration required for testing an Apollo RCS it was decided to determine the necessary design for a Vernier RCS as well as a procedure for regenerating the exchangers through ice melt off.

Vernier

Test VRCS 2-118 (5-27-83) was used to analyze the vernier case. In this test the burn rate was .093 lbs/sec and at a cell pressure of .07 psia the inlet and outlet gas temperatures were 250°F and 32°F, respectively.

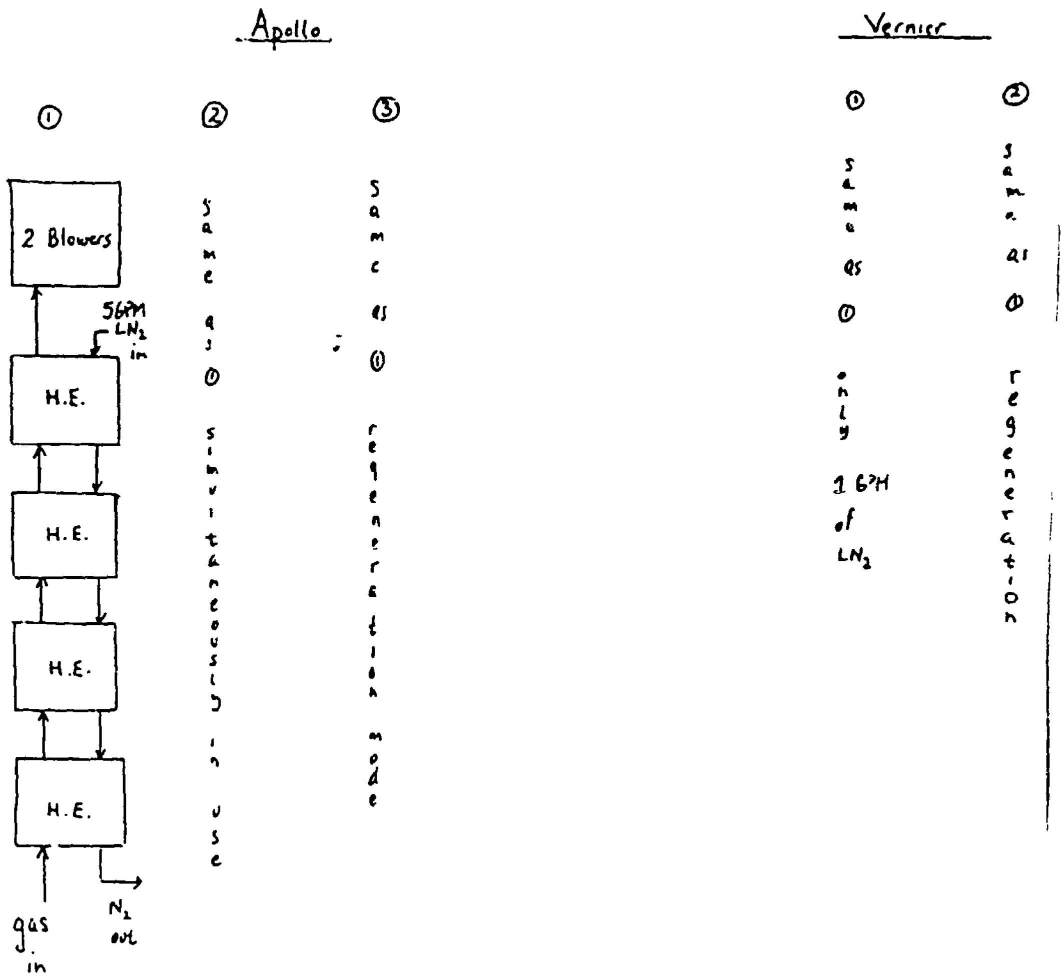
Use of the same analytical approach as the Apollo yields the data given in Table I. Most notable is the fact that in the vernier case only 1 gpm of LN_2 is needed and only 1/2 to 1 module is necessary. Therefore, if one set of two blowers can maintain altitude only one series of heat exchangers will do the job. The recommended scheme for an Apollo vs. a vernier is summarized in Figure 1.

Regeneration

A melting procedure could be the use of hot air or GN_2 on the inside of the tubes with gravity collection of a maximum of 2.0 gallons of liquid.

TABLE I. APOLLO -VS- VERNIER RCS

	APOLLO	VERNIER
Test Data		
Gas Rate/Tube, scfm	1 scfm	.43
Cell Pressure, psia	.2	.07
Anticipated Rate Based on Stiochiometry per Tube	2 scfm	.48
Inlet Gas Temperature	430°F	250°F
Outlet Gas Temperature	35°F	32°F
Heat Load per Tube if 1/2 Incoming H_2O is Removed	32 BTU/min	7 BTU/min
Area Required	3-4 Modules	1/2 - 1
LN_2 Required	5 gpm	1 gpm
LN_2 Actually Used in Test	2.5 gpm (average)	1 gpm (not reliable)



*NOTE: It is conceivable that four modules are not required, therefore, to start it is recommended that only two modules be built and tested. If more are needed they can be added later.

FIGURE 1.*