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AuthorizedAbstract

High heat flux electrical cartridge heaters were tested with direct air cooling under simulated ORR Loop conditions. The cartridges and the heater design were found to be satisfactory.

A gas cooler of concentric pipe design utilizing air, water, and air-water mixtures as the coolant was also evaluated and found to be satisfactory.

Heat transfer calculations are compared with the experimental data and found to correlate within 10%.

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TEST OF HEATER AND COOLER CONCEPTS FOR GCR-ORR LOOP
DESIGN 4

I. Introduction

The test apparatus is pictured in Figure 1. The heater test was designed to evaluate the heater design concept of a cluster of closely spaced electrical cartridge heaters with direct air cooling, the coolant moving axially along the heater bundle. The calculated heat transfer performance was compared to the experimental data as a guide to the confidence which might be placed in the calculations for the ORR helium loop components. The cartridges themselves, of a new design permitting a high heat flux (4.7 KW for a 5/8" O.D., one foot heated length), were tested for endurance.

Results indicate that the design concept is satisfactory, that heat transfer and pressure drop can be calculated within 10%, and that the cartridges are adequate for several reactor cycles at the operating conditions of 80% of rated power and 1600°F maximum sheath temperature. (The maximum expected operating conditions are 60% of rated power and 1400°F sheath temperature.)

The conceptual GCR-ORR Loop layout and operating requirements dictated that the cooler and main loop supply pipe fit into an 8 inch I.D. sleeve in the ORR pool wall, that the coolant supply and exit lines be outside the pool, and that the cooling load be continuously variable from 0 to 50 KW. The cooler test was run in conjunction with the heater test to utilize the available heat source. The cooler is of compact concentric pipe design, and must handle a wide range of heat loads by using air, water, and mixtures of the two as coolants. The purposes of the test were to evaluate the stability of cooler operation at all load levels, to determine the proper air-water ratios for stable operation, and to check validity of calculations.

Results indicate that the design is satisfactory; that cooling with air only will handle a small range of low heat loads; that cooling with a constant small amount of air and varying amounts of water will handle a

THERMOCOUPLE SCHEDULE

TYPE	PT. NO.	TEMPERATURE
V	1	SUPPLY AIR
V	2	HEATER OUTLET AIR (COOLER INLET)
V	3	COOLER ANNULUS CENTER
V	4	COOLER ANNULUS BOTTOM
L	5	CENTER HEATER CARTRIDGE
L	6	CENTER HEATER CARTRIDGE
L	7	COOLER 1/2" PIPE, 90° FROM TOP PRESSURE TAP
L	8	COOLER 1/2" PIPE, CENTER
L	9	COOLER 1/2" PIPE, 90° FROM BOTTOM PRESSURE TAP
L	10	WATER INLET
V	11	COOLER VENT
L	12	SELECTOR SWITCH FOR HEATER CARTRIDGES 1-6

PARTS LIST

PART NO.	Q'TY.	
M	1	AIR PRESSURE REGULATOR, 1", 200 CFM AT 80°F AND ATM. PRESS., 60 PSIG REGULATED PRESSURE. 1¢C
N	2	AIR ROTAMETER, 100 CFM AT 80°F AND ATM. PRESS., CALIBRATED FOR 60 PSIG. 1¢C
O	2	3/4" AIR THROTTLING VALVE. 1¢C
P	1	PRESSURE GAGE, 0-15 PSI, 1% ACCURACY. 1¢C
Q	1	WATER PRESSURE REGULATOR, 0-5 GPM, 50 PSIG REGULATED PRESSURE. 1¢C
R	2	PRESSURE GAGE, 0-100 PSI, 1% ACCURACY. 1¢C
S	1	WATER ROTAMETER, 0-5 GPM. 1¢C
T	1	1/2" TUBING OR 3/8" PIPE WATER THROTTLING VALVE. 1¢C
U	1	MANOMETER, MAX. DIFFERENTIAL 2 PSI. 1¢C
V	5	1/16" O.D. SHEATHED THERMOCOUPLES, JUNCTION EXPOSED. 1¢C
L	12	SURFACE THERMOCOUPLE WITH 1/16" CERAMIC INSULATION. 1¢C
W	1	PRESSURE GAGE, 0-200 PSI, 1% ACCURACY. 1¢C
X	1	AIR FILTER. 1¢C

NOTES:

1. SEE SK-WHK-2001 FOR HEATER DETAILS. SEE SK-WHK-2005 FOR COOLER DETAILS.
2. PROVIDE UNIONS ON AIR AND VENT LINES NEAR TEST PIECES TO PERMIT EASY REMOVAL.
3. UNLETTERED VALVES ARE GLOBE TYPE.

GCR-ORR LOOP HEATER AND
 COOLER TESTS ASSEMBLY

SK-WHK-2007

4-10-59

Figure 1

105 004

wide intermediate range of heat loads (utilizing the latent heat of vaporization of the water), and that cooling with water only, below the boiling point, will cover a narrow range of high heat loads. Cooling with water only, boiling, results in unstable operation (variation in outlet gas temperature). Heat transfer and pressure drop can be calculated within 10%.

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II. Heater Test Design

The heater design is shown in Figure 2. The heater cartridge cluster is a mockup of the seven central rods of the thirteen-cartridge heater designed for the in-pile loop. The prototype loop heater typical operating conditions of 1100°F maximum cartridge sheath temperature at 50% of rated power (4.7 KW per cartridge) were used as the design point criteria in this test.

The design conditions of air temperature, pressure, and flow were determined by assuming several values for one variable and calculating the corresponding values of the others, and plotting curves of the results to establish the design point values. The following calculations are presented to prove the compatibility of the design point conditions; they are approximate to the extent that gas properties are evaluated at the average temperature with no correction for non-linear variation over the temperature range used.

Open cross-sectional area for flow, $A = 0.00401 \text{ ft.}^2$

Equivalent diameter of open area, $D_e = 0.00958 \text{ ft.}$

Surface for heat transfer, $S = 1.145 \text{ ft.}^2$

Air properties at (350°F, 3.0 psig):

Density, $\rho = 0.058 \text{ lb./ft.}^3$

Viscosity, $\mu = 0.60 \text{ lb./ft. hr.}$

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PARTS LIST

PART	QTY.	DESCRIPTION
A	1	2 1/4" SCH. 40 ST'L. PIPE, BORED TO 2 1/8" I.D. FOR 5" AS SHOWN.
B	1	ADAPTER FOR PRESSURE GAGE, 1/2" PIPE THREAD.
C	1	3/4" O.D. COPPER TUBING, INSTALLED IN FIELD, LENGTH TO SUIT, HARD SOLDER TO PART A.
D	1	2" SCH. 40 PIPE, ANY 300 S.S., BORED TO 2 5/8" I.D. LENGTH 16".
E	1	1/4" ST'L. PLATE 2 7/8" O.D., 1 1/4" DRILL THROUGH 2 HOLES 180° APART AS SHOWN.
F	2	KOVAR SEAL CONTACT TASK ENGINEER.
G	16	#12 BARE SOLID COPPER WIRE 3" LONG CENTERED IN KOVAR SEAL SLEEVES AND HARD SOLDERED.
H	7	WATLOW HEATER CARTRIDGES. CUT LEADS AND HARD SOLDER TO PARTS G. USE INSULATING SLEEVES. RUN ONE LEAD FROM EACH HEATER TO EACH KOVAR SEAL. IN 9201-3 B/M STORES.
I	6	3/16" X 1/16" STRIP 13 3/4" LONG, ANY 300 S.S.
J	12	3/16" ROD, ANY 300 S.S.
K	3	STOP BLOCK, 3/32" X 1/4" X 1/4", ANY 300 S.S. TACK WELDED TO CUT END OF PART D.
L	8	THERMOCOUPLES. CONTACT HUGH METZ, 14C, 9201-3.
M	1	PRESSURE REGULATOR. 14C.
N	1	ROTAMETER. 14C.
O	1	MANUAL THROTTLING VALVE. 14C.
P	1	PRESSURE GAGE, 0-15 PSI, 1% ACCURACY, 14C.

PART	QTY.	DESCRIPTION
Q	1	TEMPERATURE RECORDER. 14C.
R		VARIACS AND POWER SUPPLY. CONTACT E. LEES, RPD ENGINEERING SERVICES, 9201-3.

ASSEMBLY PROCEDURE

1. PRELIMINARY WORK (ORDER NOT IMPORTANT): FABRICATE PARTS. HARD SOLDER WIRES G IN KOVAR SEALS F, HARD SOLDER SEALS F TO PLATE E, HARD SOLDER PARTS B AND C TO PART A, AND STOPS K IN PART D.
2. ASSEMBLE CARTRIDGE BUNDLE, PARTS H, I AND J, AND CHECK FOR FIT IN PART D. GRIND CORNERS OF PARTS I AT POINT OF CONTACT WITH PART D IF NECESSARY.
3. TAPE BUNDLE TOGETHER, CUT LEADS AND HARD SOLDER TO WIRES G. NOTE NUMBER CODE ABOVE. COVER EXPOSED CONDUCTORS WITH SLEEVING. MARK LOCATIONS FOR THERMOCOUPLES.
4. SPREAD HEATERS APART AND ATTACH THERMOCOUPLES. ASSEMBLE BUNDLE IN PART D. TACK WELD ALL CARTRIDGES AND RODS J TOGETHER AT THE OPEN END. WELD STRIPS I TO PART D AS INDICATED BY SHADING IN END VIEW ABOVE. USE CARE NOT TO BLOCK OTHER AIR PASSAGES.
5. WELD PART A TO PART E USING VERY LIGHT BEAD. WELD PART A TO PART D (CARTRIDGES MUST CONTACT STOPS).
6. ASSEMBLE TEST PER SCHEMATIC LAYOUT ABOVE ON SUITABLE UNISTRUT FRAME. USE 3/4" O.D. TUBING FOR AIR LINES. USE HEATER NUMBERS FOR THERMOCOUPLES (HEATER #7 1/2'S 7 #8).

GCR-ORR LOOP HEATER TEST
SK-WHK-2001
3-26-59

Figure 2

105 006

Thermal Conductivity, $k = 0.0214 \text{ Btu/hr. ft.}^2 \text{ }^\circ\text{F/ft.}$

Specific Heat, $c_p = 0.244 \text{ Btu/lb.}$

(Prandtl No.) $^{0.4}$, $N_{pr}^{0.4} = 0.860$

Assume $T_{\text{air avg.}} = 350^\circ\text{F}$, ΔP for one foot length = 4.7 psi, and friction factor = 0.0064, and solve for the required (average) air velocity in:

$$\Delta P = \frac{2fLV^2}{gD} \times \frac{\rho}{144} \text{ psi}$$

$$4.7 = \frac{2(0.0064) 1 (V^2) 0.058}{32.2 (0.00958) 144}$$

$$V^2 = 281,000$$

$$V = 530 \text{ fps} = 1,910,000 \text{ fph}$$

$$N_R = \frac{\rho DV}{\mu} = \frac{0.058 (0.00958) 1,910,000}{0.060} = 17,711$$

$f = 0.0064$ (checks assumed value) (See "Heat Transmission", McAdams, 3rd Ed., p. 156)

$$\text{Flow rate } w = VAp = 1,910,000 (0.00401) 0.058 = 444 \text{ lb./hr.}$$

$$Q_{50\% \text{ of rating}} = \frac{4.7 \text{ KW/cart.} \times 7 \text{ cartridges} \times 3413 \text{ Btu/hr. KW}}{2} = 56,150 \text{ Btu/hr.}$$

$$\Delta T_{\text{air}} = \frac{Q}{wc_p} = \frac{56,150}{(444) 0.244} = 520^\circ\text{F}$$

$$T_{\text{air exit}} = 520 + 80 = 600^\circ\text{F}$$

$$T_{\text{air avg.}} = \frac{80 + 600}{2} = 340^\circ\text{F (checks assumed value of } 350^\circ\text{F)}$$

$$\text{Film coefficient } h = 0.023 \frac{k}{D} (N_R)^{0.8} (N_{PR})^{0.4}$$

$$= 0.023 \frac{0.0214}{0.00958} (17,700)^{0.8} (0.860) = 108 \text{ Btu/hr. ft.}^2 \text{ } ^\circ\text{F}$$

$$\Delta T_{\text{metal to air avg.}} = \frac{Q}{hS} = \frac{56,150}{(108) 1.145} = 454^\circ\text{F}$$

$$T_{\text{sheath max.}} = 454 + 600 = 1054^\circ\text{F}$$

Entrance and Exit Pressure Drop: (See McAdams p. 159)

	$v \text{ ft.}^3/\text{lb.}$	$D \text{ or } D_e \text{ ft.}$	$V \text{ fps}$	$V^2 \text{ (fps)}^2$	K	$^\circ\text{F}$
1	9.2	0.18	44.5	1,980	0.45	80
2	9.2	0.00958	282	79,400	0.45	80
3	26.7	0.00958	820	672,000	----	600
4	26.7	0.18	129	16,600	----	600

Entrance

$$v_{1-2} (p_1 - p_2) = \frac{v_2^2 - v_1^2}{2g} + \frac{KV_1^2}{2g}$$

$$\frac{S_2}{S_1} = \frac{0.00401}{0.0254} = 0.158$$

$$K = 0.45$$

$$9.2 (p_1 - p_2) = \frac{77,420}{64.4} + \frac{0.45 (1,980)}{64.4}$$

(McAdams p. 159)

$$p_1 - p_2 = 146 \text{ lb/ft.}^2 = 1.0 \text{ psi}$$

Exit

$$v_{3-4} (p_3 - p_4) = \frac{v_4^2 - v_3^2}{2g} + \frac{(v_3 - v_4)^2}{2g}$$

$$26.7 (p_3 - p_4) = \frac{-655,400}{64.4} + \frac{477,000}{64.4}$$

$$p_3 - p_4 = -103 \text{ lb./ft.}^2 = -0.7 \text{ psi (pressure gain)}$$

Allowance for unheated length (0.146 ft.) of heater cartridges:

$$\Delta P = \frac{0.146}{1} \times 4.7 = 0.7 \text{ psi}$$

Total heater ΔP :

$$\Delta P = (4.7 + 0.7) + 1.0 - 0.7 = 5.7 \text{ psi}$$

(Does not include drops due to thermocouples
or deposits from oil in air).

III. Heater Performance Test Results

The calculations were checked experimentally by setting the heater input power at 50% of the rating for 7 heaters (16.45 KW), and adjusting the air flow to obtain 600° outlet temperature. Calculated and experimental data are compared on the following page, together with calculated performance for the ORR Loop heater at helium loop conditions. Correlation of the calculated and experimental performance data is adequate within the intent and purpose of the facility and calculations. It is probable that the heater thermocouple readings are low due to heat conduction through the lead wires, a probability which is heightened by the air flow being lower than predicted and by examination of certain cooler test data from similar thermocouples (See "Cooler Test", Section V). It was necessary to run the center heater at 10% less power than the others in order to balance the maximum sheath temperatures, which generally ran highest at the center of the heater cluster, and progressively cooler to the outside.

TABLE I
HEATER PERFORMANCE

	AIR TEST CALCULATED	AIR TEST EXPERIMENTAL	HELIUM LOOP CALCULATED
NUMBER OF HEATERS	7	7	13
50% OF RATED POWER KW	16.45	16.45 (Controlled)	30.5
GAS FLOW LB./HR.	444 (Air)	410 (Air)	348 (Helium)
AVERAGE PRESSURE PSIG	3	3	400
INLET GAS TEMP. °F	80	80	630
OUTLET GAS TEMP. °F	600	600 (Controlled)	870
MAX. CARTRIDGE SHEATH TEMP. °F	1054	960 1060 1070 980 1034 avg. 995 1110 1060 FOR SEVEN CARTRIDGES	1114
AT MAX. GAS TO MAX. SHEATH	454	330 460 470 380 395 510 460 Average 434	244
TOTAL ΔP PSI	5.7 (Does not include effect of T/C's or oil deposits from air)	7.0	0.3

IV. Heater Endurance Test

The 1000 hour endurance test was run at the stringent conditions of 80% of rated power and 1600°F maximum sheath temperature. (No more than 60% or rated power, 1400°F is anticipated for the GCR-ORR Loop) No failures occurred until the 550 hour point (one reactor cycle is 500 hours), when one heater lost one of its four parallel resistance windings. Similar failures occurred on two cartridges at 700 hours, and one at 750 hours; one failed completely at 830 hours.

Examination of the heaters after test indicated that all failures were due to hot spots caused by oil and water in the plant air used as the coolant. Four cartridges had a total of five holes melted in the sheaths, four of the holes being in two pairs opposite each other in a common air passage. In every case, evidence of plugging of the passage surrounded the point of failure. Three of these failures occurred in the first inch of heated length, where heavy deposits were evident on all heaters. The remaining two occurred four inches from the hot end, opposite each other in a common passage where a mass of contaminants had lodged after breaking loose further upstream. No failure occurred in the hottest third of the heated length where failures due to materials breakdown would be expected. It is reasonable to assume that these heaters will operate at 80% of rating and 1600°F maximum sheath temperature for several thousand hours in a clean system.

Photograph 5 shows the individual heater cartridges after disassembly of the bundle. Originally deposition of contaminants occurred at the beginning of the heated length, causing burnout of the first element in three heaters. The new ring of deposits can be seen one-fourth further along the heated portion on these three. Heater No. 7 failed completely, the burnout at the entrance region causing failure of the central conductor supplying power to all elements. The third element of one cartridge failed due to displaced deposits, and two cartridges were intact at the finish of the test.

V. Cooler Test

The test cooler construction is shown in Figure 3. It incorporates two concentric annuli and a central pipe, the process air flowing downward through the outside annulus, coolant air entering the one inch tube at the top and entraining water from the 1/4 inch O.D. line, and the coolant mixture moving down the central pipe and up the inner annulus.

Pertinent cooler operating temperatures traced directly from recorder charts are shown in Figure 4. The bottom half of the chart is from a continuous run; the top half is from a series of separate runs. The shift in heater temperatures is due to plugging of the air passages by oil-water deposits from the plant air. The depression of the cooler inlet temperature with increased cooling is due to the thermocouple location on the side of the coolant inlet pipe away from the cooler process air inlet and slightly below it. Thus considerable cooling occurred before the process stream reached the inlet thermocouple. This temperature was actually nearly constant as shown by the dashed line, since the heater temperatures, power input, and air flow were all constant. The process stream ΔT with no cooling is due to losses to ambient. The quantities of air and water refer to the coolant stream.

Stable operation was easily attained over the entire range of load levels obtainable, from 0 to 8.2 KW, with considerable overlap of the ranges handled by air-only cooling, air-water mixtures and water only. Very little air was needed in air-water mixtures, its function being to maintain good coolant distribution for a wide range of water flow rates. Operation with no air and small quantities of water resulted in local boiling and very unstable cooler process outlet temperatures.

All calculations checked well with the experimental data with one exceptions; heat pickup by the coolant could not be calculated except in the case of water-only cooling, since the combination of air, steam, and entrained water in the coolant vent line prevented an easy determination of what portion of the water had been vaporized. While this difficulty

PARTS LIST

PART	DESCRIPTION
1	1 1/2" SCH. 40 PIPE, ANY 300 S.S. 3'-5" LONG, THREADED ONE END
2	SAME AS ITEM 1, 0'-2" LONG. THREAD ONE END BEFORE CUTTING TO LENGTH.
3	2 1/2" SCH. 40 PIPE, ANY 300 S.S., 3'-7" LONG, MACHINE 2 1/4" D. HOLE AS SHOWN
4	2" SCH. 40 PIPE, ANY 300 S.S., 0'-4" LONG.
5	2 1/2" SLIP ON FLANGE, 150 LB. STEEL
6	1 1/2" SCH. 40 BUTT WELD TEE, STEEL
7	1 1/2" SCH. 40 BUTT WELD PIPE CAP, ANY 300 S.S.
8	1 1/2" SCH. 40 PLAIN STEEL COUPLING
9	1" O.D. TUBING, 0.049 OR GREATER WALL, ANY 300 S.S., 4'-4" LONG.
10	1/4" PLATE, STEEL, 7" DIA.
11	1/4" PLATE, STEEL, 2" DIA.

ASSEMBLY PROCEDURE

1. PRELIMINARY WORK: WELD 1/16" S.S. WELDING ROD SPACERS TO PARTS 1 AND 9, 4 SPACERS 90° APART ON EACH PART AS SHOWN. CUT SPACERS TO LENGTH AFTER WELDING SO THAT PIPING C'S COINCIDE. MAKE WELDS 5-1, 5-2, 5-3, AND MAKE UP PARTS 1 AND 2 INTO COUPLING 8 USING ANTI-SEIZE COMPOUND.
2. SLIP PLATE 10 OVER 1 1/2" PIPE ASSEMBLY BUT DO NOT WELD, MAKE WELD 5-4.
3. INSERT TUBING 9 IN 1 1/2" PIPE ASSEMBLY, SLIP ON PLATE 11, MAKE WELD 5-5.
4. ADJUST TUBE 9 FOR 1/2" CLEARANCE WITH CAP 7 AS SHOWN, MAKE WELD 5-6.
5. BOLT PLATE 10 TO FLANGE 5, ADJUST HEIGHT OF ASSEMBLIES. MAKE WELD 5-7.

INSTALLATION DETAIL

GCR-ORR LOOP COOLER TEST
SK-WHK-2005

4-6-89

Figure 3

COOLER TEST TEMPERATURES

Figure 4

could have been overcome (by bubbling the vent mixture through water in an insulated tank, and taking a temperature rise over a period of time) it was not considered worthwhile since the conditions for maximum cooling could be calculated, and the data indicated that any cooling level from maximum to zero could be attained. The case of air-only cooling could not be calculated because of heat transfer from the coolant annulus to the coolant inlet pipe stream; the film coefficients for the three parallel air streams were of the same order, and no thermocouples were provided to show the coolant temperature gradient between inlet and outlet. While a similar situation existed in the case of water-only cooling, the high water side film coefficient ($\sim 850 \text{ Btu/hr.ft.}^2 \text{ }^\circ\text{F}$) as compared to the air side (~ 33) resulted in the coolant-process wall running at a nearly constant temperature. Hence, the coolant in the annulus was assumed to run at constant temperature (the vent temperature); it transferred heat to the coolant inlet pipe at the same rate that it picked up heat from the process air. This assumption is especially valid because the coolant ΔT from inlet to outlet was only 25°F ; any temperature selected in this range would not introduce too much error into the calculations, the coolant to process ΔT being over 500°F .

The coolant pipe wall temperatures (not shown in Figure 4) were measured by thermocouples exposed to the process gas stream, and indicated some 90° high as compared to values calculated from both the process and coolant sides which checked within 7°F . This leads to the suspicion that similar thermocouples indicating heater maximum sheath temperatures were reading low, the gas being colder than the metal in that instance.

The calculations which follow illustrate the methods used in analyzing cooler performance. They pertain to the water-only run, because this condition can be analyzed completely and it represents maximum cooler performance. Results are tabulated following the calculations.

Process Stream Losses to Ambient: (at no cooling load)

$$Q = w_c \Delta T_p = 450 \text{ lb./hr.} (0.257 \text{ Btu/lb.}^\circ\text{F}) 55^\circ\text{F} \quad (825 - 770) = 55^\circ\text{F} \\ = 6,360 \text{ Btu/hr.}$$

This loss rate applies to a $T_{\text{process average}}$ of 800°F , or a ΔT of

720° between process and ambient. Lower process temperatures will result in proportionately lower losses.

Total Process Stream Heat Loss: ($T_{\text{process avg.}} = 640^{\circ}\text{F}$)

$$Q = w_c \Delta T_p = 450 (0.252) 300 = 34,000 \text{ Btu/hr.} \quad (825 - 525) = 300^{\circ}\text{F}$$

Loss to ambient at $T_{\text{process avg.}} = 640^{\circ}\text{F}$

$$Q = 6,360 \times \frac{560}{720} = 4,950 \text{ Btu/hr.}$$

Process Stream Heat Transferred to Coolant:

$$\begin{array}{r} 34,000 \\ - 4,950 \\ \hline Q = 29,050 \text{ Btu/hr.} \end{array}$$

Heat Gain by Coolant:

$$Q = w_c \Delta T_p = 1,085(1)25 = 27,100 \text{ Btu/hr.} \quad (90 - 65) = 25^{\circ}\text{F}$$

This value checks the process side value of 29,050 within 7%.

An average value of 28,000 (8.2 KW) is used in further calculations.

Process Side Film Coefficient:

$$V = \frac{w}{\rho A} = \frac{450}{0.048 \times 0.0136} = 690,000 \text{ fph}$$

$$h = 0.023 \frac{0.0273}{0.0473} \left(\frac{0.048(0.0473)690,000}{0.074} \right)^{0.8} (0.857) \quad N_R = 21,180$$

$$= 33 \text{ Btu/hr.ft.}^2\text{.}^{\circ}\text{F} \quad N_R^{0.8} = 2,900$$

Coolant Wall Process Side Temperature Calculated from Process Side:

$$\Delta T_{\text{process to wall}} = \frac{Q}{h_a} = \frac{28,000}{33 \times 1.74} = 488^{\circ}$$

This is the log mean ΔT for the conditions pictured below:

It can be seen that in $\Delta T_{LM} = \frac{(T_1 - T_3) - (T_2 - T_4)}{\ln \frac{T_1 - T_3}{T_2 - T_4}}$ the value of

the numerator must be 300° because $T_1 - T_2 = 300^\circ$ and $T_3 = T_4$. Therefore since ΔT_{LM} must equal 488° to satisfy the heat transfer conditions,

$$\ln \frac{T_1 - T_3}{T_2 - T_4} = \frac{300}{488} = 0.615, \text{ and } \frac{T_1 - T_3}{T_2 - T_4} = 1.85. \text{ Thus } \frac{T_1 - T_3}{T_2 - T_4} = 1.85,$$

$$\frac{790 - T_3}{490 - T_3} = 1.85, \text{ and } T_3 = 137^\circ\text{F} = T_4.$$

Coolant Side Film Coefficient:

$$V = \frac{w}{A_D} = \frac{1,084 \text{ lb/hr}}{62.1 \text{ lb/ft}^3 \times 0.00213 \text{ ft}^2} = 8,200 \text{ fph} \quad N_R = 2,860$$

$$= 2.28 \text{ fps} \quad N_R^{0.8} = 582$$

(For equation on following page.)

$$h_{I.D.} = 0.023 \frac{0.36}{0.0104} \left(\frac{62.1 (0.0104) 8,200}{1.85} \right)^{0.8} \left(\frac{1 \times 1.85}{0.360} \right)^{0.4} N_{PR} = 5.14$$

$$= 953 \text{ Btu/hr.ft.}^2 \text{ } ^\circ\text{F} \quad N_{PR}^{0.4} = 2.057$$

$$h_{O.D.} = 953 \times \frac{1.61}{1.9} = 848 \text{ (in terms of outside surface)}$$

Coolant Wall Process Side Temperature Calculated from Coolant Side:

$$\Delta T_{\text{coolant to wall (inside)}} = \frac{Q}{hA} = \frac{28,000}{953 \times 1.47} = 20^\circ$$

$$\Delta T_{\text{through metal}} = \frac{QL}{KA} = \frac{28,000 (0.00858)}{9.1 (\pi \times 0.115 \times 3.5)} = 21^\circ$$

$$T_{\text{coolant wall (outside)}} = T_{\text{coolant}} + \Delta T_{\text{film}} + \Delta T_{\text{metal}}$$

$$= 90 + 20 + 21 = 131^\circ\text{F} \text{ (compares with } 137^\circ\text{F calculated from process side.)}$$

Overall Film Coefficient (process to coolant):

$$U = \frac{1}{\frac{1}{h_o} + \frac{x}{k} + \frac{1}{h_i}} = \frac{1}{\frac{1}{33} + \frac{0.0086}{9.1} + \frac{1}{848}}$$

$$= \frac{1}{0.0324} = 30.9 \text{ Btu/hr.ft.}^2 \text{ } ^\circ\text{F}$$

Heat Transfer from $Q = UA\Delta T$:

$$Q = 30.9 (1.74) 529 = 28,500 \text{ Btu/hr.}$$

(Or 28,800 if the 6° discrepancy in wall temperature is added to the 529° stream-to-stream ΔT .)

TABLE II
COOLER PERFORMANCE

HEAT TRANSFERRED (Single Stream)	29,050 Btu/hr.	27,100 Btu/hr.
REMARKS	from $Q = w c_p \Delta T$ Process Stream	from $Q = w c_p \Delta T$ Coolant Stream
ΔT PROCESS TO COOLANT BASED ON 28,000 Btu/hr.	529°	535°
REMARKS	Calculated from $Q = h A \Delta T$	From Experimental Data
HEAT TRANSFERRED, FROM OVERALL COEFFICIENT	28,500 Btu/hr.	28,800 Btu/hr.
REMARKS	From $Q = U A \Delta T$ using calculated $\Delta T_{\text{process-coolant}}$	From $Q = U A \Delta T$ using experimental $\Delta T_{\text{process-coolant}}$

VI. CONCLUSION

Predicted heater performance for the helium loop is shown in Table I in Section III of this report. Endurance of the cartridges is adequate; actual life will depend on loads and temperature levels in the ORR loop. No unexpected results were evolved by the test other than the fact that the central cartridges must be run at slightly less power than the outer ones.

Photograph 3 in the final section of this report is indicative of the rugged construction of the cartridges as evidenced by their ability to perform well under the scaling conditions caused by oil and water in the plant air.

Calculations similar to those in Section II indicate that the helium loop cooler will reduce the temperature of the gas at the design flow rate to 300°F at a 50 kw load level, the most stringent condition anticipated. Pressure drop for the test cooler including the entrance loss and 90° turn was 1 psi, which was checked by rough calculations. No detailed comparison was made since this loss is only a small fraction of the heater pressure drop which will be only 0.3 psi for the helium loop. The nozzle shown on the water inlet line in the cooler construction sketch was found to be unnecessary, its removal to permit more water flow having no effect on stability. A larger water line or several small ones will be required on the helium loop to permit adequate flow.

Photograph 6 shows a scale deposit in the coolant annulus, due primarily to minerals in the plant water. The annulus configuration was changed from that shown in Figure 3 to a 1/16 inch annulus with spiral wires to improve the film coefficient. Demineralized water will be used in the full scale cooler.

The heater and cooler designs are considered adequate as evaluated by this test, and the design and construction of full scale prototypes is proceeding. These units will be utilized in a combination thermocouple calibration-compressor performance facility to permit further testing of

the heater and cooler, and to save the cost of special components for the facility. After proving, the prototypes will be used in the in-pile loop.

The heater and cooler test facility was designed and constructed at a cost of \$7,000 in an elapsed time of one month from start of drawings to operational startup.

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Photograph 1. Test Facility

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Photograph 2. Test Section Cut Open

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PHOTO 34472

Photograph 3. Heater Assembly Cut Open

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PHOTO 34471

Photograph 4. Heater Assembly End View

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PHOTO 34470

Photograph 5. Heater Cartridges and Spacers

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PHOTO 34473

Photograph 6. Cooling Annulus Detail

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