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R-9958

SOLAR-RECEIVER HEAT-FLUX
CAPABILITY AND STRUCTURAL INTEGRITY

FINAL REPORT

May 1976

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TEST IN PROGRESS
KEEP OUT
HI-PRESS. STEAM

1259-95-18H 4-16-76

Frontispiece. Horizontal Radiant Heating Test Facility in Operation.

ACKNOWLEDGMENT

The author gratefully acknowledges the cooperation and technical capability of the B-1 Division Thermal Laboratory personnel under the supervision of Roger S. Watson. Special acknowledgment is accorded to Walter Moen for the basic design and development of the radiant heating facility and to Robert Scherer for overall facility activation and test operation. The efforts of Al Mason and Howard Nelson also contributed greatly toward the successful completion of the test program.

FOREWORD

The work upon which this publication is based was performed pursuant to Contract No. E(04-3)-1103 with the Energy Research and Development Administration. The period of performance extended from July 1975 to May 1976.

ABSTRACT

An experimental program was conducted to determine the operating characteristics of full length (65 feet) single and multi-tube once-through steam generator test sections subjected to radiant heat flux levels commensurate with commercial solar tower receiver application.

Absorbed heat flux levels ranging from 0.15 to 0.71 Btu/in.²-sec (0.25 to 1.16 MW/m²) were achieved in a horizontal facility utilizing graphite radiant heater arrays. Steam exit temperatures ranged from 625 F (two-phase) to 1380 F at pressures of 1000 to 2300 psia. Wall temperature profiles and fluid pressure losses were obtained and compared with an existing computer model.

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INTRODUCTION

Current interest in alternate energy sources has led to increased effort in the utilization of solar energy. A promising concept is the use of a large heliostat field focused on a tower-mounted solar receiver steam generator. One version of the solar tower receiver concept utilizes a once-through flow circuit with subcooled water entering at the bottom and superheated steam exiting at the top. In order to maximize the receiver efficiency, it is desirable to operate at as high a heat flux level as is feasible.

The purpose of this contractual effort was to experimentally investigate the thermal and structural characteristics of a once-through steam generator panel when subjected to high radiant heat flux levels ($\approx 0.75 \text{ Btu/in.}^2\text{-sec}$) commensurate with commercial solar tower receiver application.

The program was of ten months' duration divided into two phases. The first phase involved testing of a single 65-foot long tube up to the maximum heat flux limit of the radiant heating facility ($\approx 0.7 \text{ Btu/in.}^2\text{-sec}$). These tests provided facility checkout and evaluation as well as basic heat transfer data on the cooling characteristics of the once-through water/steam concept. The second phase was directed toward testing of a 65-foot long five-tube panel assembly structurally simulating a commercial type solar receiver design.

Rocketdyne is also currently involved in a study of a proof of concept experimental device (POCE) at the 10 MW level. That program involves radiant heating of a full scale panel assembly. The primary difference is that the POCE is concerned with relatively low heat flux design levels of about $0.13 \text{ Btu/in.}^2\text{-sec}$, up to five times lower than the heat fluxes achieved in this program. These lower heat flux levels, however, permit testing of a full size (≈ 3 feet wide) panel.

The contractual effort discussed herein provides basic information on the thermal characteristics and structural integrity of a once-through steam generator panel subjected to radiant heat flux levels consistent with full scale commercial solar tower receiver operation.

SUMMARY

Parametric analyses were performed to determine the basic design and operating characteristics of single tube and multi-tube test sections for evaluation in a horizontal radiant heating facility.

Single tube test sections 65-feet long were fabricated of 1/4 and 3/8-inch outer diameter (x 0.035 wall thickness) stainless steel tubing. A 65-foot long panel was fabricated of five 5/16-inch O.D. x 0.040 inch wall Incoloy 800 tubes longitudinally welded together to achieve a light-tight seal.

A series of 23 tests was conducted with the 3/8-inch O.D. single tube test section in the horizontal radiant heating facility. Heat flux levels ranged from 0.15 to 0.71 Btu/in.²-sec (absorbed) at steam outlet pressures of 1000 to 2250 psia. Steam exit temperatures up to 1380 F were achieved. The test section was undamaged at the conclusion of the test series.

Eleven tests were conducted with the five-tube panel at heat flux levels up to 0.65 Btu/in.²-sec (absorbed), pressures up to 2300 psia and steam temperatures up to 1000 F. The test section was undamaged except for three pinhole leaks resulting apparently from electrical arcing between the panel and graphite heaters.

Two types of high solar absorptance paints were evaluated on the single tube and panel test sections. Both paints blistered and flaked off over large areas of the test sections. Inadequate curing and/or a hot steam environment are the probable causes of the paint failure.

The flow was stable in all tests. Minor flow and pressure oscillations (< 5 percent) were noted during the start of four of the single tube tests but damped out after only two to three cycles.

The single tube and five-tube test sections operated satisfactorily over the entire range of test conditions. Maximum wall temperatures were, in general, less than 1200 F except for extreme off-design operation. The tube and panel sliding supports, designed to permit unrestrained longitudinal thermal expansion, performed satisfactorily. Maximum measured test section growth was 6.75 inches.

Measured wall temperature profiles were compared with computer model predictions. Deletion of a two-phase region pressure correction factor from the model resulted in satisfactory agreement between measured and predicted wall temperature profiles.

TECHNICAL DISCUSSION

OBJECTIVE

The primary objective of this program was to demonstrate the feasibility of the once-through boiler concept, shown pictorially in Fig. 1, in a radiant heating environment simulating a commercial solar tower receiver application.

The once-through boiler concept provides a lightweight, low-cost, compact, simple manifold approach to a solar tower receiver design. Primary areas of investigation are wall temperature prediction accuracy, flow stability, and pressure drop.

The commercial (250 MW_E) solar tower receiver application is characterized by significantly higher heat flux levels as compared to Proof of Concept Experiment (POCE) studies (10 MW_E) currently being conducted by various contractors. An increased heat flux level results in a reduction in the thermal loss fraction as shown in Fig. 2. The increased system thermal efficiency permits a reduction in the size of the required heliostat field with a resultant significant cost saving.

APPROACH

The technical approach utilized in this program to achieve the desired objective was to design, fabricate, and test full length (65-foot) single tube and five-tube panel test sections in a horizontal radiant heating facility at heat flux levels commensurate with commercial type solar receiver application.

PRELIMINARY ANALYSIS

Test Section Configuration

Upon initiation of this contract, the overall program and facility requirements were reviewed in terms of best simulation of the full scale commercial (≈ 250 MW) system design within the scope and funding of this program. The originally proposed wrap-around test section was modified (with contract

ONCE-THROUGH BOILER CONCEPT

● CONCEPT ADVANTAGES

- SIMPLE MANIFOLDING
- LIGHT WEIGHT
- LOW-COST
- COMPACT

● AREAS OF INVESTIGATION

- WALL TEMPERATURE PREDICTION
- PRESSURE DROP
- FLOW STABILITY

Figure 1

EFFECT OF DESIGN HEAT FLUX ON THERMAL LOSSES

Figure 2

monitor approval) to a straight horizontal test configuration to facilitate obtaining the desired longitudinal heat flux profile. Panel and radiant heater fabrication was also simplified by use of a straight, rather than wrap-around configuration. Flow stability was not expected to be a problem in the horizontal configuration due to the relatively high fluid velocities (high ratio of inertia to density forces) required for cooling at high heat fluxes. The test section was placed below the graphite heater array to minimize natural convection effects.

Panel Tube Geometry

A preliminary thermal and structural evaluation of panel tube geometry was conducted early in the program so that the relatively long lead time tubing could be ordered. The evaluation was based on the straight horizontal configuration in conjunction with the desired heat flux profile and facility power limitations.

The baseline heat flux profile utilized in the evaluation is shown in Fig. 3 as taken from Ref. 1 for a 250 MW commercial plant. The heat flux profile is for the north side of a cylindrical receiver using a single point aim mirror system. This profile is the most severe since the peak power input occurs near the mid-point of the tube wherein two-phase flow exists. The maximum heat flux is seen to be about $0.75 \text{ Btu/in.}^2\text{-sec.}$

The panel tests were planned to be conducted at peak heat flux levels up to $1.0 \text{ Btu/in.}^2\text{-sec.}$ The panel tube geometry was analyzed at this maximum test condition assuming the previously discussed heat flux profile adjusted upward proportionately.

The tube material selected in the parametric evaluation was Incoloy 800 based on current studies for the solar tower receiver design. Similarly, a nominal wall thickness of 0.040 inch was selected as being representative of tower designs for commercial application.

CYLINDRICAL RECEIVER
NORTH SIDE
SINGLE POINT AIM
NOON-SUMMER SOLSTICE

Figure 3. Typical Commercial Solar Receiver Axial Heat Flux Profile

A brief structural analysis was conducted based on preliminary two-dimensional tube temperature distributions as shown in Fig. 4 . The results of this analysis indicated a maximum wall temperature of 1100 F would meet the ASME code structural requirements and provide for a fatigue life greater than 50,000 cycles. This operating temperature, in conjunction with the maximum imposed heat flux level of $1.0 \text{ Btu/in.}^2\text{-sec}$ determined the required coolant film coefficient (h_c) of approximately $0.0030 \text{ Btu/in.}^2\text{-sec-F}$. It was further assumed that the maximum heat flux occurred at the point wherein the coolant capability was a minimum. Based on experimental data taken last year in a company-funded project, the required coolant mass flux (G_c) at a nominal pressure of 2000 psia was determined to be $4.0 \text{ lb/in.}^2\text{-sec}$ (see Appendix I).

The required coolant mass flux was used to determine panel tube length as a function of average heat flux level for various candidate tube sizes assuming a water inlet temperature of 500 F and a nominal steam outlet temperature of 950 F. The results of this analysis are shown in Fig. 5 . The heat flux profile discussed previously (Fig. 2) has an average-to-peak ratio = 0.47. This would require tube lengths of about 61, 92, and 123 feet for corresponding tube outer diameters of 1/4, 5/16, and 3/8-inch in order to simulate the full heat flux profile.

In terms of a commercial system design, it is, of course, desirable to utilize as large a tube as is feasible to minimize the possibility of plugging. Preliminary commercial tower analysis has indicated maximum tube outer diameters of about 3/8 inch (with a 0.040-inch-thick wall) dependent somewhat on receiver size and heat input distribution. From the facility standpoint, however, tube lengths much in excess of 60-70 feet become unwieldy in terms of space, power, and instrumentation considerations.

The tube diameter can be increased and the length decreased by truncating the heat flux profile at either end. If the profile is truncated at a point wherein the minimum value is about one-sixth the peak, the resulting ratio of average-to-peak heat flux is increased to 0.66. Deletions of the low heat

CROWN HEAT FLUX = 0.9769

INCOLOY 800 TUBE
 $d_o = 5/16$ IN.
 $t_w = 0.040$ IN.

$T_{COOL} = 600$ F

$h_c = 0.0030$ BTU/IN²-SEC

Figure 4. Panel Tube 2-D Temperature Distribution

$(Q/A)_{\text{PEAK}} = 1.0 \text{ BTU/IN}^2\text{-SEC}$
 WATER $T_{\text{IN}} = 500 \text{ F}$
 STEAM $T_{\text{OUT}} = 950 \text{ F}$
 $G_c = 4.0 \text{ lb/in.}^2\text{-sec}$

Figure 5. Panel Heated Length Requirements

flux regions at either end of the profile will not significantly affect test results. The corresponding tube lengths are 44, 65, and 88 feet for corresponding tube diameters of 1/4, 5/16, and 3/8 inch.

The 5/16-inch diameter tube with a heated length of 65 feet was selected as the baseline for the panel design. This appears to be a reasonable compromise between a desired large diameter tube and excessive test section length while closely maintaining the desired heat flux profile. The resulting truncated normalized heat flux profile is shown in Fig. 6 for the 65-foot test section.

The number of tubes in the panel are limited by total facility power capability. The input power requirements for various numbers of tubes are shown in Fig. 7. Based on a current facility power capability of 1200 KW, a maximum of six tubes of 5/16-inch diameter can be utilized in the test panel. The preliminary baseline panel design selected, therefore, consisted of six 65-foot long Incoloy (800) tubes of 5/16-inch outer diameter with a nominal wall thickness of 0.040 inch. (This design was later changed to five tubes based on actual facility limitation as determined in single tube tests.)

The maximum water flowrate requirements for various panel designs are shown in Fig. 8. For the baseline panel design, a maximum flowrate of about 8 gpm is required, based on a coolant mass flux of $4.0 \text{ lb/in.}^2\text{-sec.}$ This is within the facility capability and with a useful tank volume of 200 gallons (facility tank $\approx$ 250 gallons maximum) would permit maximum accumulated run durations of 40 minutes before retanking.

Figure 6, Solar Receiver Test Panel Normalized Heat Flux Profile

Figure 7, Panel Heat Input Requirements

PEAK $Q/A = 1.0$ BTU/IN-SEC

$G_c = 4.0$ LB/IN²-SEC

$t_{wall} = 0.040$ IN.

$T_{IN} = 500$ F

$T_{OUT} = 950$ F

Figure 8. Panel Water Flowrate Requirements

Single Tube Geometry

The primary purpose of this program was to evaluate a tube panel configuration comparable to a commercial tower design. In addition, however, tests were to be conducted on single tube configurations to provide for facility checkout as well as obtain basic heat transfer data at high heat flux conditions. The basic heat transfer data obtained from the relatively inexpensive single tube test sections would permit more accurate estimation of the panel operating conditions and limitations, thereby assuring safe operation of the more expensive panel test section.

A primary consideration in the selection of the single tube geometry was ready availability. Tubing in the required length of 65 - 70 feet requires a special mill run with a lead time of 8 - 12 weeks. The facility activation and single tube test schedule did not permit consideration of a mill run. The single tube test section was, therefore, composed of several tube lengths butt-welded together to form a single tube of the required length.

Available in-house tubing consisted of 1/4-inch and 3/8-inch stainless steel with nominal wall thicknesses of 0.035 inch. Both of these tube sizes were analyzed to determine their range of applicability.

Single Tube Operating Conditions

A preliminary evaluation of the operating temperature profiles of the 3/8-inch and 1/4-inch O.D. CRES tubes was made using the Rocketdyne Solar Tower Tube Design computer program (Appendix A). A range of peak heat flux levels of 0.25 to 1.4 Btu/in.²-sec and inlet pressures of 1500 to 2500 psia were investigated. The heated tube length was taken as 65 feet since the single tube tests were to be conducted in the same facility as the panel tests.

An estimate was made of the circumferential heat input distribution using the tube and heater arrangement shown in the inset of Fig. 9. The resulting normalized distribution (neglecting reflection from the structure side walls) is compared with the usual cosine distribution in Fig. 9. The resulting integrated heat input around the circumference is about 8 percent lower than for an assumed cosine distribution due to the finite width of the heater.

Typical tube wall temperature profiles based on the normalized heat flux profile discussed previously are presented in Fig. 10 for the 3/8-inch O.D. tube at a peak heat flux level of 0.75 Btu/in.²-sec and various water inlet pressures. It is apparent that the coolant pressure has a significant effect on the wall temperature peak in the two-phase regime. At an inlet pressure of 2500 psia, the peak temperature (in the two-phase region) is less than 900 F whereas at a pressure of 1500 psia the wall temperature increases to about 1500 F. These pressure effects on two-phase cooling capability were determined experimentally in a Rocketdyne in-house program last year and the results incorporated into the computer program utilized in the current analyses.

A cross-plot of maximum two-phase region wall temperature as a function of peak heat flux level and coolant inlet pressure is shown in Fig. 11 for the 3/8-inch O.D. CRES tube. Corresponding total coolant pressure drops are

Figure 9. Single Tube Circumferential Heat Load Distribution

Figure 10. Single Tube 1-D Surface Temperature Profiles

Figure 11. Maximum 2-Phase Region Wall Temperature for 3/8 in. Single Tube

shown in Fig. 12 . These temperatures (as well as those in Fig. 10) are based on a modified one-dimensional conduction analysis (modified such that the program approximates the correct 2-D temperatures at nominal design heat flux levels).

The foregoing results are based on using the same normalized heat flux profile as developed for the panel in conjunction with a design outlet steam temperature of 950 F (water inlet temperature = 500 F). These two factors essentially determine the water flowrate and resultant coolant mass flux. Increasing the water flowrate will reduce the wall temperature but will, of course, reduce the steam outlet temperature unless the Q/A profile is flattened (i.e., the end point heat flux levels are raised) or the inlet temperature increased. There are, therefore, various modifications to the test conditions that can be utilized to extend the operating range of the 3/8-inch O.D. tube.

The 1/4-inch O.D. x 0.035-inch wall stainless steel tube was also evaluated in a manner similar to that for the 3/8-inch tube. The non-truncated heat flux profile (Fig. 3) was utilized for the 1/4-inch O.D. tube to minimize coolant flowrate and pressure drop while maintaining a steam outlet temperature of 950 F. The resultant tube operating temperatures are presented in Fig. 13 . The lower wall temperatures, as compared to the 3/8-inch O.D. tube, result from increased coolant mass flux levels for the smaller tube. The coolant pressure drop for the 1/4-inch O.D. tube is, however, about 4.5 times that required for the 3/8-inch O.D. tube.

Panel Operating Conditions

A parametric evaluation of the six-tube panel operating conditions was conducted in a similar manner to the single tube evaluation. The relative circumferential heat flux distribution was determined for each tube using the heater-tube configuration shown in Fig. 14 and neglecting reflections from the structure side-wall (not shown). These results normalized with respect to the tube crown value are shown in Fig. 14 for inner and outer tubes. The innermost tube closely approximates a cosine distribution such that the integrated heat input equals the tube crown heat flux times the tube projected

Figure 12. 3/8 in. O.D. Single Tube Coolant Pressure Drop

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Figure 13. Maximum 2-Phase Region Wall Temperature for 1/4 in. O.D. Single Tube

Figure 14. Relative Circumferential Heat Flux Distribution for the Radiant Heated Panel

area. The outermost tube has an integrated heat input approximately 7 percent lower than the inner tube. The use of side reflectors will tend to reduce this difference. (In the actual test phase, the heater width was slightly less than the five-tube panel utilized. Based on the coolant temperature rise differential, the variation in heat input between center and outer tube was about 7 percent.)

The computer predicted panel tube wall temperature profiles for peak heat flux level of 0.75 are shown in Fig. 15 for coolant inlet pressures of 2500, 2000, and 1500 psia. The maximum two-phase region wall temperatures are cross-plotted as a function of peak heat flux and inlet pressure in Fig. 16.

The maximum 1-D temperatures again occur at the lower inlet pressure conditions. The worst case appears to be a temperature of slightly over 1300 F at a pressure of 1500 F. A detailed 2-D computer analysis of this condition at the maximum heat flux condition was conducted. The resulting computer output temperature distribution is shown in Fig. 17. The maximum 2-D temperature is seen to be 1397 F as compared to a 1-D value of 1314 F.

A preliminary structural analysis at this worst case temperature condition indicated a membrane stress level less than the material yield stress. The minimal number of tests (≈ 2 or 3) at the extreme off-design operating conditions in conjunction with the short hold-times ($\approx 2 - 3$ minutes) indicated no creep problem. The panel is, therefore, capable of safe operation over the range of interest.

The predicted coolant pressure drops for the panel over the operating range of interest is presented in Fig. 18. The maximum pressure drop of about 430 psi occurs during maximum heat flux operation ($1.0 \text{ Btu/in.}^2\text{-sec}$) at minimum inlet pressure ($\approx 1500 \text{ psia}$).

Figure 15. Panel 1-D Surface Temperature Profiles

Figure 16 , Panel Maximum Two-Phase Region Tube Wall Temperature

CROWN HEAT FLUX = 0.9766 BTU/IN²-SEC

$P_{cool} = 1500$ psia
 $T_{cool} = 593$ F

Incoloy 800 Tube

$d_o = 5/16$ inch

$T_w = 0.040$ inch

$T_{max\ 1-D} = 1315$ F

Figure 17. Panel Tube 2-D Temperature Distribution - Worst Case Conditions

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Figure 18. Panel Coolant Pressure Drop

Stress Analysis

A parametric stress analysis of the panel tube geometry was conducted to ascertain operating limits in terms of allowable wall temperature levels. The analysis technique is presented in Appendix B. The cyclic fatigue life is affected primarily by the thermal strains induced by local temperature differentials across the hot-wall and between the hot-wall and cooler back wall.

The maximum allowable heated surface temperature to achieve 10,000 cycles as a function of back wall temperature for various hot-wall temperature differentials is shown in Fig. 19. At a maximum heat flux of $1.0 \text{ Btu/in.}^2\text{-sec}$, the wall $\Delta T \approx 150 \text{ F}$. Assuming a nominal back wall temperature of 670 F (based on saturation temperature at 2500 psia), the maximum allowable hot surface temperature is seen to be in excess of 1100 F .

The foregoing preliminary analysis was performed to provide an estimate of tube and panel test section operating characteristics for use as a guide during the test phase of the program. Actual test results, to be discussed later, indicated wall temperatures lower than predicted at reduced coolant mass flux levels. In particular, the predicted significant effect of reduced coolant pressure on increased wall temperature was not noted.

MAXIMUM TUBE SECTION TEMPERATURE DISTRIBUTION
FOR $\epsilon_{\text{EFF.}} = 0.440\%$ AT 10,000 CYCLES

Figure 19. Design Temperature Limits for Solar Tower Receiver Tube

DESIGN

The primary goal of both the single tube and multi-tube test section designs was to permit unrestrained longitudinal thermal expansion while minimizing movement toward the radiant heater assembly. In addition, the multi-tube panel assembly design was to simulate, insofar as possible, a commercial type receiver design.

Single Tube Test Section

No formal design effort was performed for the single tube test section. Rather, a simplified support structure was formulated based on the basic design concept utilized for the panel assembly. A simple inverted T-section support composed of welded stainless steel sheet (1/16-inch thick) was considered for the 1/4 inch and 3/8-inch O.D. single tube test section as shown in Fig. 20. The bottom of the inverted T is free to slide along the bottom on the reflector housing assembly and is restrained from excess vertical movement through the use of loosely fitting ($\approx 1/16$ -inch clearance) guides. The inverted T supports are located at 3-foot intervals along the 65-foot long test section length.

Panel Test Section

The basic panel design as originally envisioned consisted of six 5/16-inch O.D. x 0.040-inch wall x 65 foot long Incoloy 800 tubes. This was later reduced to five tubes based on facility power limitations. The original panel assembly shown in Fig. 21 and 22 incorporated, insofar as possible, the structural design features then envisioned at Rocketdyne for a commercial type solar tower receiver design. In particular, the design permits unrestrained longitudinal expansion while maintaining a relatively flat panel surface profile along the length. This latter constraint is important in the radiant heating facility since variations in distance between the radiant heaters and tube surface can affect the incident heat flux profile. The tubes are longitudinally welded together to provide a light-tight seal.

Figure 20. Sketch of Single Tube Support Structure

Figure 21. Six Tube Panel and Structural Support Assembly

FACILITY: BOB SCHERER 8-259-2065-4240 WALT MOEN
TUBE WELDING: GENE WALKER - NYS DOUG McFARLANE M-P

UNLESS OTHERWISE SPECIFIED, DIMENSIONS ARE IN INCHES AND APPLY TO ALL PARTS.		DESIGN: NOZZI		Rockwell International Corporation Rockledge Division Covington, California	
TOLERANCES ON ANGLES IN ° 30		DATE: 12/21/61		LAYOUT: 12/21/61	
TOLERANCES ON ANGLES IN ° 30		DRAWN: [blank]		HEAT FLUX AND STRUCTURAL TEST	
TOLERANCES ON ANGLES IN ° 30		CHECKED: [blank]		E 02602 AP75-144	
TOLERANCES ON ANGLES IN ° 30		DESIGN ACTIVITY: [blank]		SCALE: [blank]	
TOLERANCES ON ANGLES IN ° 30		DATE: [blank]		SHEET: [blank]	
TOLERANCES ON ANGLES IN ° 30		DO NOT SCALE PRINT		[blank]

-019	INCO 800 ROD	.500 O.D. $\times$ 1.25	
-017	321 CRES PLATE	.56 $\times$ 1.56 $\times$ 1.62	
-015	INCO 800 TUBE	.312 O.D. $\times$.040W $\times$ 70FT	
-013			
-009			
-007			
-005			
-003	INCO 800 TUBE	.312 O.D. $\times$.040W $\times$ 70FT	
DASH NO.	MATERIAL	DESCRIPTION	SPECIFICATION

8. PRESSURE TEST WITH WATER AT
6000 PSI FOR 2 MINUTES
7. GRIT BLAST OXIDIZE AND ABSORPTIVITY
COAT THIS SIDE ONLY AT TEST FACILITY
6. MACHINE PER RAO 103-002
5. WELD PER RAO107-022 CLASS I
4. WELD PER RAO107-027 CLASS II
3. AUTOMATIC TIG WELD PER SPEC
2. TUBING STOCK IS DELIVERED TO ROCKETDYNE IN 10 FT DIA COILS
1. TUBE BEND RADII 1.00

-001	FINAL
PART NO	ASSY

NOTED

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES AND
APPLY PRIOR TO FINISH.

SU/ MACH. SURF. ROUGHENED

TOLERANCES ON ANGLES: ± 9°
DECIMALS: .01 ± .05 .0001 ± .005

HOLES NOTED "DRILL"

OVER THRU	TOLERANCE
.0009 .0009	± .0015 ± .00
.0010 .0010	± .0020 ± .00
.0015 .0015	± .0025 ± .00
.0020 .0020	± .0030 ± .00
.0025 .0025	± .0035 ± .00
.0030 .0030	± .0040 ± .00
.0035 .0035	± .0045 ± .00
.0040 .0040	± .0050 ± .00
.0045 .0045	± .0055 ± .00
.0050 .0050	± .0060 ± .00
.0055 .0055	± .0065 ± .00
.0060 .0060	± .0070 ± .00
.0065 .0065	± .0075 ± .00
.0070 .0070	± .0080 ± .00
.0075 .0075	± .0085 ± .00
.0080 .0080	± .0090 ± .00
.0085 .0085	± .0095 ± .00
.0090 .0090	± .0100 ± .00
.0095 .0095	± .0105 ± .00
.0100 .0100	± .0110 ± .00

DO NOT SCALE DRAWING

OWN	NOZZI		DATE	7-9
CHK			DATE	
DSGN				
MATL				
STRUCT				
DESIGN ACTIVITY APVD			DATE	

Rockwell International Corporation Rockaway Division Camarillo Park, California		
PANEL, ASSY OF HEAT FLUX & STRUCTURAL TEST		
BULK E	CODE IDENT NO. 02602	DISPATCH NO. AP75-135

SIZE E	CODE IDENT NO. 02602	DRAWING NO. AP75-135
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Subsequent detailed structural analysis under Rocketdyne's 10 MW Solar Receiver subcontract (L/C 075008)^{*} upon which the foregoing design is based, indicated required modifications to increase panel fatigue life capability. In the original design, a solid steel band (for attachment of the support structure) was welded across the entire width of the tube panel and tended to restrict thermal growth in this direction with resultant fatigue life degradation. The current pilot plant panel design utilizes attachment tabs welded to several individual tubes across the width of the panel.

In order to maintain the basic solar panel support structure concept as currently envisioned and to minimize fabrication costs, the simplified support design shown in Fig. 23 was utilized in this program. The box-type supports are attached at 3-foot intervals. The design permits panel width thermal expansion through the flexibility of the thin vertical support webs. In addition, unrestrained longitudinal expansion is permitted as the supports are free to slide along the base of the reflector assembly housing. Vertical movement (toward the heaters) is limited to about 1/16 inch by the loosely fitting slide guides.

Individual fittings are installed on each tube with provisions for orifices at the inlet end. This latter feature is to provide maximum testing flexibility in terms of flow stability control, particularly at low flowrate operation. Manifolds are provided at each end, as shown in Fig. 24 and 25 with fittings for adapting to the test facility.

^{*}Prime Contract E(04-3)-1108

Figure 23. Sketch of Revised Panel Support Structure

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED
1.	MAY BE REWORKED		
2.	CANNOT BE REWORKED		
3.	RECORD CHANGE		
4.	NOV SHOP PRACTICE		
5.	PARTS MADE OK

Figure 24. Inlet Manifold Assembly for Six-Tube Panel

NO	MATERIAL	SIZE	SPECIFICATION
-005	321 CR STEEL	500x238x1.75	
-003	321 CR STEEL	1.00 DIA x 1.00 LONG	

-001	FINAL
NO	ASSY

HEAT TREAT		UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES AND APPLY PRIOR TO FINISH. 125/ MACH. SURF. ROUGHNESS		DATE 9-9-75		Rockwell International Corporation Rocketdyne Division Canoga Park, California	
FINISH		TOLERANCES ON: ANGLES $\pm$ 0° 30'		DATE 9-9-75		INLET MANIFOLD, ASSY OF HEAT FLUX TEST PANEL.	
MATERIAL		DECIMALS .XX $\pm$.03 .XXX $\pm$.010		DRAWN 10221		SIZE CODE IDENT NO. DRAWING NO.	
NOTED		HOLES NOTED "DRILL"		MATERIAL 10221		D 02602 AP75-131	
		OVER THRU TOLERANCE		DATE		SHEET	
		.0000 .0400 + .0015 - .0010		R-9958		39	
		.0400 .1300 + .0030 - .0010					
		.1300 .2290 + .0045 - .0010					
		.2290 .5000 + .0060 - .0010					
		.5000 .7500 + .0070 - .0010					
		.7500 1.0000 + .0090 - .0010					
		1.0000 2.0000 + .0120 - .0010					
		DO NOT SCALE PRINT					

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED
	1. MAY BE REWORKED		3. RECORD CHANGE
	2. CANNOT BE REWORKED		4. NOW SHOP PRACTICE
	5. PARTS MADE OK

Figure 25. Discharge Manifold Assembly for Six-Tube Panel

-005	2.00 GRES BAR	9.70 x 2.00 x .300	
-003	2.00 GRES BAR	1.25 DIA x 1.00 LONG	
NO	MATERIAL	SIZE	SPECIFICATION

-001	FINAL
NO	ASSY

2 WELD PER RAO 107-027
 1 MACHINE PER RAO 103-002

NOTE: UNLESS OTHERWISE SPECIFIED

HEAT TREAT		UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES AND APPLY PRIOR TO FINISH. 125/ MACH. SURF. ROUGHNESS		Rockwell International Corporation Rocketdyne Division Canoga Park, California	
FINISH		TOLERANCES ON: ANGLES $\pm$ 0° 30' DECIMALS $\pm$.03 .001 $\pm$.010 HOLES NOTED "DRILL" OVER THRU TOLERANCE .0000 .0400 + .0015 - .0010 .0400 .1300 + .0030 - .0010 .1300 .2250 + .0045 - .0010 .2250 .5000 + .0060 - .0010 .5000 .7500 + .0070 - .0010 .7500 1.0000 + .0090 - .0010 1.0000 2.0000 + .0120 - .0010		DISCHARGE MANIFOLD, ASSY OF HEAT FLUX TEST PANEL	
MATERIAL		DO NOT SCALE PRINT		DESIGN ACTIVITY APVD	
NOTED		R-9958		DATE	
		SCALE 1/1		SHEET	

FABRICATION

Single Tube Test Section

Two single tube test sections were fabricated. The primary test section, utilized in the single tube test series, was fabricated from 3/8-inch O.D. x 0.035-inch wall 304 stainless steel tubing. A backup test section was fabricated of 1/4-inch O.D. x 0.035-inch wall 321 stainless steel tube. In both cases the full 65-foot length was achieved by end-to-end butt welding of available shorter length tubing. This approach was taken to facilitate the single tube test program due to the long lead times required to obtain the required full length tubing.

The single tube test sections were grit-blasted to provide the necessary surface finish for application of a high solar absorptance ($\alpha_s = 0.93 - 0.95$) paint.

The inverted T-section supports were welded to the 3/8-inch tube test section as shown in Fig. 26. These supports are free to slide in the facility so as to permit unrestrained longitudinal growth while restricting transverse movement.

The single tube test section was instrumented with 33 chromel-alumel thermocouples as shown schematically in Fig. 27. These thermocouples were silver-brazed to the tube as shown in Fig. 28. The thermocouples were then masked and the B-1 Division S-31 high solar absorptance paint applied to the tube surface and cured, using a hot-air flow through the tube.

Panel Test Section

The panel test section was fabricated of five 3/16-inch O.D. x 0.040-inch wall Incoloy 800 tubes. A special mill run (Hardy & Harmon Tube Company, Pennsylvania) was required to obtain the 70-foot long tubes. The 14 tubes ordered were shipped in a 10-foot diameter coil. The tubing was satisfactorily straightened by a local vendor.

Figure 26. Single Tube Test Section With Supports

Figure 27. Thermocouple Installation for Single Tube Tests

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Figure 28. Silver Brazed Thermocouple
Attachment

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The major fabrication step in the panel assembly was the continuous longitudinal welds required for a light-tight seal. These welds were accomplished using the automatic welding table, shown in Fig. 29 , constructed for this purpose under the 10MW Pilot Plant program.

Extensive characterization of the various weld parameters was undertaken to assure satisfactory weldment of the full length panel. Numerous short panels ($\sim 3 - 4$ feet long) were fabricated, sectioned, and subjected to metallurgical examination. Four of the original 14 tubes were utilized in the automatic weld facility characterization.

It was determined that a simple clamping together of the tubes was unsatisfactory due to the sensitivity of the tracking wheel to variations in tube height. The tubing was, therefore, tack-welded at short (~ 6 -inch) intervals while clamped in position as shown in Fig. 30 and 31 . The tubing surface was cleaned with alcohol prior to welding.

The five-tube panel was then longitudinally welded along one side only as shown in Fig. 32 . Each of the four longitudinal welds took only about 15 minutes based on the facility weld rate of 54 inches per minute. A 0.030-inch diameter Incoloy 82 wire was utilized for the weld material. No internal gas purge was utilized during the welding process since the earlier test specimens had indicated the weld did not penetrate through the tube wall.

The welded panel was grit-blasted and then shipped to the B-1 Division for final assembly to the support structure and manifolds. Once the manifolds were installed, a leak check was performed with the panel pressurized to 200 psig with gaseous nitrogen. Twelve small leaks were discovered using the standard soap solution leak-check technique. The leaks appeared to originate between the tubes and were randomly located in terms of tube and axial position.

The weld material was ground out in one of the leak locations in order to view the tube surface. The leak was located using a 10-power glass in conjunction with the soap solution. There appeared to be a small pin hole leak

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Rockwell International
Rockeladyne Division

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Figure 29. Automatic Welding Table

Fig. 30. Preparation of Five-Tube Panel for Automatic Longitudinal Welding

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Figure 31. Tack-Welded Panel

Figure 32. Longitudinal Welding of Five-Tube Panel

near the root of the weld. The actual hole could not be seen, only the fuzz formed by the soap solution.

A detailed investigation was conducted as to the nature of the leaks discovered in the panel tubes. A section of tubing containing a leak was removed from the panel as shown in Fig. 33. A new section was welded in place (Fig. 34) using techniques Rocketdyne has developed for thrust chamber repair. The removed piece of tubing was sectioned and micrographed. The conclusion of Rocketdyne Materials and Processes personnel, after reviewing the micrograph, was that the leaks were caused by weld burn-through and subsequent localized oxidation. Future longitudinal welding procedures will incorporate an inert gas purge to eliminate oxidation and resultant porosity of the weld.

The results of the leak investigation indicated the panel was repairable since no cracks due to thermal stresses were noted. The remaining panel tube leaks were repaired using high temperature silver braze to minimize cost. The panel was subsequently leak-checked with a soap solution while pressurized to 1000 psia with nitrogen gas. No further leaks were discovered.

The repaired panel was instrumented with 38 chromel-alumel thermocouples utilizing a high temperature silver braze attachment technique. The thermocouple locations are noted schematically in Fig. 35. Pressure taps were welded to each of the tubes at the inlet end to allow for pressure drop measurements across the inlet orifices. Taps were also welded to each tube at the outlet end to measure individual exit coolant temperatures.

Finally the panel was sprayed with Pyromark Black high-temperature paint to provide a high absorptance surface. The thermocouples were masked during this step so as to provide a low absorptance surface and minimize the thermocouple error due to radiant heat input.

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Figure 33. Panel with Tube Section Removed
for Metallurgical Examination

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Figure 34. Panel with Tube Section Replaced

THERMOCOUPLES INSTALLED ON CENTER
TUBE EXCEPT AS NOTED

Figure 35. Thermocouple Installation for Five Tube Panel

The final panel assembly, prior to installation in the test facility, is shown in Fig. 36 . The inlet manifold end is shown in Fig. 37 where the installed tube pressure taps can be seen. The outlet end is similar except fluid temperature taps replace the pressure taps.

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Figure 36. Completed Panel Assembly

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Figure 37. Panel Inlet Manifold Assembly

TEST FACILITY

The single tube and five-tube panel were tested at the Rockwell B-1 Division Thermal Laboratory radiant heating facility. A view of the 65-foot long horizontal test position is shown in Fig. 38.

Flow System

A simplified facility flow schematic is shown in Fig. 39 . The system consists of a 250-gallon, high pressure (3000 psia) tank with a GN_2 pressurant system, a flow-control valve, a filter (5 micron), a calibrated orifice for flow measurement, in-line heaters for water inlet temperature control, the radiantly heated test section, and an outlet orifice for back pressure control. The superheated steam is vented to atmosphere.

Radiant Heaters

The heart of the facility is the horizontal radiant heating test bed. This rig is composed of 20 premium grade ATJ graphite heater strips lined up end-to-end. The graphite heaters are 1-3/8 inch wide x 3/8 inch thick and of varying lengths, ranging from 26 inches to 52 inches, dependent on the desired heat flux distribution. The heater installation technique is shown in Fig. 40.

The required heater power is furnished by a 1200 KW 3-phase 480 VAC power source which feeds saturable reactors and 480 to 50 volt stepdown transformers. Each heater circuit is energized using single phase power and separately controlled with individual variacs. The system power supply is shown in Fig. 41.

During operation, the heater array is maintained in an atmosphere of argon supplied through a manifold assembly.

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Figure 38. Horizontal Radiant Heating Test Facility

Figure 39. Flow Schematic for Radiant Heated Solar Receiver Tube and Panel Tests

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Figure 41. Radiant Heating Facility
Power Supply

Reflector Assembly

Each graphite heater assembly is contained in a 6061T6 aluminum reflector assembly with high reflective surfaces to minimize heat loss. A cross-section of the reflector assembly is shown in Fig. 42 . The top of the assembly contains molding sand saturated with water to provide top reflector cooling. The side and bottom reflectors are of sufficient mass (1 in. thick) to permit adequate test durations (~3 minutes) based on heat sink cooling. Typical reflector assemblies are shown in Fig. 43.

Data Acquisition

During tests, data were recorded using various gauges, strip charts, multi-point recorders and a computer data acquisition system. The facility control panel containing the various recorders is shown in Fig. 44 . The test section fluid inlet and outlet pressure, flow measurement orifice pressure drop, and a power signal were monitored on a Dynagraph pen recorder. The coolant outlet pressure was also monitored on an easily read 0-3000 psi gauge. The coolant inlet-to-outlet pressure differential was monitored separately on a 0-400 psi gauge. The coolant inlet and outlet temperature were recorded on separate single point strip chart type devices. A multi-point strip chart recorder was utilized to monitor six selected test section wall surface thermocouples.

The 48-channel computer acquisition system was utilized to record all of the test section surface thermocouples as well as fluid inlet and outlet temperature. In addition, for the panel tests, the inlet orifice pressure differentials were recorded. A power signal was also recorded to provide indication of radiant power cutoff and to time-relate the computer data to the Dynagraph instrument. The computer system samples data every .01 seconds and provides a tabular printout and/or graphical representation of the results.

Figure 42. Aluminum Reflector Assembly

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Figure 43. Reflector Assemblies

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Figure 44. Facility Control Panel

WATER PURITY

High purity water was utilized during all single tube and panel tests conducted under the current contractual effort.

The water purity specifications utilized are the same as imposed for the 10-MW pilot plant Subscale Research Experiment (SRE) under contract E(04-3)-1108. These specifications are:

Dissolved solids		0.050 parts per million
Dissolved oxygen	=	0 (trace)
pH	=	9.5 $\pm$ 0.1

Deionized water was supplied by the Industrial Water Division of Arrowhead Puritas Water. The typical analyses of the deionized water are given in Table 1. In order to insure that the dissolved solids remain less than the specified value, the delivered water was cycled through an on-site deionizer bank to further purify or polish the water. The actual dissolved solids content was determined by an electrical resistance measurement using a Balsbaugh 900M-18M Monitor.

The dissolved oxygen was controlled by addition of hydrazine (N_2H_4). A Hydrazine Comparator Block was utilized to measure the water hydrazine content. A nominal hydrazine content of about 0.5 ppm is indicative of zero dissolved oxygen.

The addition of hydrazine tended to increase the pH level of the water. Ammonia was added, as needed, to bring the pH level of the water to the specified value. The pH of the water was monitored with a Beckman pH meter.

The foregoing specified parameters were monitored often during the test program to insure that the required water purity was maintained. In addition, the facility plumbing was composed entirely of stainless steel tubing to minimize water contamination. A 5-micron filter was utilized upstream of the test section as a further safeguard against tube or orifice fouling.

TABLE 1. WATER PURITY ANALYSIS AS DELIVERED

TYPICAL ANALYSIS ARROWHEAD PURITAS DEIONIZED WATER	
	Parts per Million
Calcium	0.00
Magnesium	0.00
Sodium	Less than 0.01
Iron	0.00
Copper	0.00
Sulfates	0.00
Chlorides	0.00
Silica	0.01
Carbon Dioxide	0.10
Total Dissolved Solids	0.04
Resistance—Ohms	10-15,000,000
pH	7.0

SUBSCALE CALIBRATION TESTS

Heater Evaluation

A single section of the graphite heater assembly (≈ 52 inches long) was installed in the subscale test rig (Fig. 45) for purposes of heat flux calibration and thermocouple instrumentation evaluation. Two five-foot long 3/8-inch O.D. x 0.035-inch wall CRES (304) tubes were prepared for these calibration tests. The tubes were grit-blasted and coated with the high absorptance Pyromark paint. External and internal thermocouples were installed in various manners as shown in Fig. 46 to measure surface wall temperature and coolant bulk temperature rise. The thermocouples include both closed and open tip and grounded and ungrounded junctions with welded, silver brazed and strapped attachments. A photo of one of the tube assemblies is shown in Fig. 47.

Calibration tests were conducted with the aforementioned assembly using low pressure subcooled water which permits accurate analytical prediction of wall temperature as a function of imposed heat flux. The tests indicated the silver brazed thermocouple attachment technique is most accurate as discussed later. The water temperature rise data were utilized to determine integrated heat load and heat flux for comparison to the calorimeter calibration device. These calibration tests were utilized to optimize heater length, reflector placement, and to determine resultant total facility power requirements.

The maximum heat flux measured in these tests was slightly over $1.0 \text{ Btu/in.}^2\text{-sec}$ utilizing a reduced heater length of 26 inches. This heat flux level is satisfactory for the panel test requirements but is somewhat lower than the maximum proposed for the single tube tests ($\approx 1.4 \text{ Btu/in.}^2\text{-sec}$). It appears that the above heat flux level is a practical facility limit without resorting to excessive heater power inputs and/or impractically short heater length. Based on these preliminary facility power calibration results, the multi-tube panel assembly was modified to a five rather than six tube configuration to insure sufficient fluid heating.

Figure 45. Radiant Heater Characterization Test Rig

1. CALIBRATE HEAT FLUX RATE USING ASYMPTOTIC WATER COOLED CALORIMETER MODEL C-130C AT 200 BTU/FT²-SECOND. (DISTANCE BETWEEN HEATER AND THER. SENSED IFC VOLTAGE & POWER.
2. RECORD OUTPUT FROM 5 TEST THERMOCOUPLES ON HYDRO-MECHANICAL LABORATORY COMPUTER DATA ACQUISITION SYSTEM. PLOT ALL OUTPUTS (SCALE SET) USING INDIVIDUAL CHANNELS FOR EACH THERMOCOUPLE ON EXPANDED SCALE. STYLE "A" THERMOCOUPLE SHALL USE DYNAMICS MODE 32/HZ/HMB AMPLIFIER. SHIELDING OF TMS THERMOCOUPLE TO BE CARRIED FROM METAL SKIN-THRU TO DATA ACQUISITION UNIT.
3. CONDUCT A SECOND TEST RUN RECORDING OUTPUTS FROM 5 THERMOCOUPLES ON SINGLE PRINT BALANCED INSTRUMENTS.
4. CONDUCT A THIRD TEST RUN TO DISTRIBUTION SEEKING A MINIMUM OF 1000F ON THERM. VIAL. USE DATA ACQUISITION CALLED OUT IN 2. ABOVE.

USE ISOLATED UNGROUND THERMOCOUPLES ONLY

FOR PARTS LIST, GENERAL NOTES, & APPLICATION DATA SEE SEPARATE PARTS LIST									
1000 1001 1002 1003 1004 1005 1006 1007 1008 1009 1010 1011 1012 1013 1014 1015 1016 1017 1018 1019 1020 1021 1022 1023 1024 1025 1026 1027 1028 1029 1030 1031 1032 1033 1034 1035 1036 1037 1038 1039 1040 1041 1042 1043 1044 1045 1046 1047 1048 1049 1050 1051 1052 1053 1054 1055 1056 1057 1058 1059 1060 1061 1062 1063 1064 1065 1066 1067 1068 1069 1070 1071 1072 1073 1074 1075 1076 1077 1078 1079 1080 1081 1082 1083 1084 1085 1086 1087 1088 1089 1090 1091 1092 1093 1094 1095 1096 1097 1098 1099 1100 1101 1102 1103 1104 1105 1106 1107 1108 1109 1110 1111 1112 1113 1114 1115 1116 1117 1118 1119 1120 1121 1122 1123 1124 1125 1126 1127 1128 1129 1130 1131 1132 1133 1134 1135 1136 1137 1138 1139 1140 1141 1142 1143 1144 1145 1146 1147 1148 1149 1150 1151 1152 1153 1154 1155 1156 1157 1158 1159 1160 1161 1162 1163 1164 1165 1166 1167 1168 1169 1170 1171 1172 1173 1174 1175 1176 1177 1178 1179 1180 1181 1182 1183 1184 1185 1186 1187 1188 1189 1190 1191 1192 1193 1194 1195 1196 1197 1198 1199 1200 1201 1202 1203 1204 1205 1206 1207 1208 1209 1210 1211 1212 1213 1214 1215 1216 1217 1218 1219 1220 1221 1222 1223 1224 1225 1226 1227 1228 1229 1230 1231 1232 1233 1234 1235 1236 1237 1238 1239 1240 1241 1242 1243 1244 1245 1246 1247 1248 1249 1250 1251 1252 1253 1254 1255 1256 1257 1258 1259 1260 1261 1262 1263 1264 1265 1266 1267 1268 1269 1270 1271 1272 1273 1274 1275 1276 1277 1278 1279 1280 1281 1282 1283 1284 1285 1286 1287 1288 1289 1290 1291 1292 1293 1294 1295 1296 1297 1298 1299 1300 1301 1302 1303 1304 1305 1306 1307 1308 1309 1310 1311 1312 1313 1314 1315 1316 1317 1318 1319 1320 1321 1322 1323 1324 1325 1326 1327 1328 1329 1330 1331 1332 1333 1334 1335 1336 1337 1338 1339 1340 1341 1342 1343 1344 1345 1346 1347 1348 1349 1350 1351 1352 1353 1354 1355 1356 1357 1358 1359 1360 1361 1362 1363 1364 1365 1366 1367 1368 1369 1370 1371 1372 1373 1374 1375 1376 1377 1378 1379 1380 1381 1382 1383 1384 1385 1386 1387 1388 1389 1390 1391 1392 1393 1394 1395 1396 1397 1398 1399 1400 1401 1402 1403 1404 1405 1406 1407 1408 1409 1410 1411 1412 1413 1414 1415 1416 1417 1418 1419 1420 1421 1422 1423 1424 1425 1426 1427 1428 1429 1430 1431 1432 1433 1434 1435 1436 1437 1438 1439 1440 1441 1442 1443 1444 1445 1446 1447 1448 1449 1450 1451 1452 1453 1454 1455 1456 1457 1458 1459 1460 1461 1462 1463 1464 1465 1466 1467 1468 1469 1470 1471 1472 1473 1474 1475 1476 1477 1478 1479 1480 1481 1482 1483 1484 1485 1486 1487 1488 1489 1490 1491 1492 1493 1494 1495 1496 1497 1498 1499 1500 1501 1502 1503 1504 1505 1506 1507 1508 1509 1510 1511 1512 1513 1514 1515 1516 1517 1518 1519 1520 1521 1522 1523 1524 1525 1526 1527 1528 1529 1530 1531 1532 1533 1534 1535 1536 1537 1538 1539 1540 1541 1542 1543 1544 1545 1546 1547 1548 1549 1550 1551 1552 1553 1554 1555 1556 1557 1558 1559 1560 1561 1562 1563 1564 1565 1566 1567 1568 1569 1570 1571 1572 1573 1574 1575 1576 1577 1578 1579 1580 1581 1582 1583 1584 1585 1586 1587 1588 1589 1590 1591 1592 1593 1594 1595 1596 1597 1598 1599 1600 1601 1602 1603 1604 1605 1606 1607 1608 1609 1610 1611 1612 1613 1614 1615 1616 1617 1618 1619 1620 1621 1622 1623 1624 1625 1626 1627 1628 1629 1630 1631 1632 1633 1634 1635 1636 1637 1638 1639 1640 1641 1642 1643 1644 1645 1646 1647 1648 1649 1650 1651 1652 1653 1654 1655 1656 1657 1658 1659 1660 1661 1662 1663 1664 1665 1666 1667 1668 1669 1670 1671 1672 1673 1674 1675 1676 1677 1678 1679 1680 1681 1682 1683 1684 1685 1686 1687 1688 1689 1690 1691 1692 1693 1694 1695 1696 1697 1698 1699 1700 1701 1702 1703 1704 1705 1706 1707 1708 1709 1710 1711 1712 1713 1714 1715 1716 1717 1718 1719 1720 1721 1722 1723 1724 1725 1726 1727 1728 1729 1730 1731 1732 1733 1734 1735 1736 1737 1738 1739 1740 1741 1742 1743 1744 1745 1746 1747 1748 1749 1750 1751 1752 1753 1754 1755 1756 1757 1758 1759 1760 1761 1762 1763 1764 1765 1766 1767 1768 1769 1770 1771 1772 1773 1774 1775 1776 1777 1778 1779 1780 1781 1782 1783 1784 1785 1786 1787 1788 1789 1790 1791 1792 1793 1794 1795 1796 1797 1798 1799 1800 1801 1802 1803 1804 1805 1806 1807 1808 1809 1810 1811 1812 1813 1814 1815 1816 1817 1818 1819 1820 1821 1822 1823 1824 1825 1826 1827 1828 1829 1830 1831 1832 1833 1834 1835 1836 1837 1838 1839 1840 1841 1842 1843 1844 1845 1846 1847 1848 1849 1850 1851 1852 1853 1854 1855 1856 1857 1858 1859 1860 1861 1862 1863 1864 1865 1866 1867 1868 1869 1870 1871 1872 1873 1874 1875 1876 1877 1878 1879 1880 1881 1882 1883 1884 1885 1886 1887 1888 1889 1890 1891 1892 1893 1894 1895 1896 1897 1898 1899 1900 1901 1902 1903 1904 1905 1906 1907 1908 1909 1910 1911 1912 1913 1914 1915 1916 1917 1918 1919 1920 1921 1922 1923 1924 1925 1926 1927 1928 1929 1930 1931 1932 1933 1934 1935 1936 1937 1938 1939 1940 1941 1942 1943 1944 1945 1946 1947 1948 1949 1950 1951 1952 1953 1954 1955 1956 1957 1958 1959 1960 1961 1962 1963 1964 1965 1966 1967 1968 1969 1970 1971 1972 1973 1974 1975 1976 1977 1978 1979 1980 1981 1982 1983 1984 1985 1986 1987 1988 1989 1990 1991 1992 1993 1994 1995 1996 1997 1998 1999 2000 2001 2002 2003 2004 2005 2006 2007 2008 2009 2010 2011 2012 2013 2014 2015 2016 2017 2018 2019 2020 2021 2022 2023 2024 2025 2026 2027 2028 2029 2030 2031 2032 2033 2034 2035 2036 2037 2038 2039 2040 2041 2042 2043 2044 2045 2046 2047 2048 2049 2050 2051 2052 2053 2054 2055 2056 2057 2058 2059 2060 2061 2062 2063 2064 2065 2066 2067 2068 2069 2070 2071 2072 2073 2074 2075 2076 2077 2078 2079 2080 2081 2082 2083 2084 2085 2086 2087 2088 2089 2090 2091 2092 2093 2094 2095 2096 2097 2098 2099 2100 2101 2102 2103 2104 2105 2106 2107 2108 2109 2110 2111 2112 2113 2114 2115 2116 2117 2118 2119 2120 2121 2122 2123 2124 2125 2126 2127 2128 2129 2130 2131 2132 2133 2134 2135 2136 2137 2138 2139 2140 2141 2142 2143 2144 2145 2146 2147 2148 2149 2150 2151 2152 2153 2154 2155 2156 2157 2158 2159 2160 2161 2162 2163 2164 2165 2166 2167 2168 2169 2170 2171 2172 2173 2174 2175 2176 2177 2178 2179 2180 2181 2182 2183 2184 2185 2186 2187 2188 2189 2190 2191 2192 2193 2194 2195 2196 2197 2198 2199 2200 2201 2202 2203 2204 2205 2206 2207 2208 2209 2210 2211 2212 2213 2214 2215 2216 2217 2218 2219 2220 2221 2222 2223 2224 2225 2226 2227 2228 2229 2230 2231 2232 2233 2234 2235 2236 2237 2238 2239 2240 2241 2242 2243 2244 2245 2246 2247 2248 2249 2250 2251 2252 2253 2254 2255 2256 2257 2258 2259 2260 2261 2262 2263 2264 2265 2266 2267 2268 2269 2270 2271 2272 2273 2274 2275 2276 2277 2278 2279 2280 2281 2282 2283 2284 2285 2286 2287 2288 2289 2290 2291 2292 2293 2294 2295 2296 2297 2298 2299 2300 2301 2302 2303 2304 2305 2306 2307 2308 2309 2310 2311 2312 2313 2314 2315 2316 2317 2318 2319 2320 2321 2322 2323 2324 2325 2326 2327 2328 2329 2330 2331 2332 2333 2334 2335 2336 2337 2338 2339 2340 2341 2342 2343 2344 2345 2346 2347 2348 2349 2350 2351 2352 2353 2354 2355 2356 2357 2358 2359 2360 2361 2362 2363 2364 2365 2366 2367 2368 2369 237									

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Figure 47. Thermal Instrumentation Calibration Tube

Thermocouple Evaluation

One of the most important aspects of the current experimental program was the measurement of the single tube and panel surface temperatures when exposed to a high radiant heat flux environment. Knowledge of the tube heated surface temperature over a range of operating conditions is vital for verifying the basic feasibility of the once-through boiler concept for advanced commercial solar tower receiver application. The measured heated surface temperatures in conjunction with the corresponding measured back surface temperatures can also be utilized to more accurately predict the fatigue life capability of the receiver tubes.

The primary difficulty in the measurement of the heated surface is that the thermocouple must be interposed between the tube surface and the radiant heater. A significant amount of heat is, therefore, absorbed by the thermocouple device itself causing its temperature to increase above that of the surface to which it is attached. The actual temperature differential between the surface and the thermocouple is dependent on various factors such as thermocouple type and geometry, attachment technique, and thermocouple reflectance.

The most common thermocouple type, particularly for high temperature application, utilizes a metallic sheath to contain the thermocouple wires. The thermocouple wires are isolated from the sheath and each other (except at the junction) by means of a suitable insulator such as MgO. If the sheath is enclosed at the measurement end, it is commonly referred to as a closed tip thermocouple. In this case, the thermocouple junction can either be attached to the sheath tip (grounded) or not (ungrounded). If the thermocouple wires are exposed at the tip of the sheath, the device is usually referred to as an open tip thermocouple. If the exposed thermocouple wires are then attached directly to the surface to be measured, a grounded junction is formed.

Based on previous experience, and verified in the calibration tests, the grounded type thermocouple junction (either open or closed tip) is unsatisfactory in a high energy electrical field due to high noise levels. The use of ungrounded thermocouples appear to be the only feasible approach for use in the radiant heating facility.

The attachment technique can also have a significant effect on the measured temperature, particularly for a sheathed ungrounded thermocouple. It is common practice to simply weld the thermocouple tip to the surface to be measured. In the proposed application, this would mean welding the thermocouple tip to the tube crown only. The remainder of the thermocouple sheath exposed to the radiant heat input is essentially uncooled and the absorbed energy must be transmitted through the thermocouple tip and weld joint. The result is an increased measurement error (i.e., higher tip temperature) as well as a possible excessive sheath temperature at some point around the circumference of the tube or panel. If the attachment surface area is increased, both the measurement error and maximum sheath temperature should be diminished.

Analysis. In order to evaluate the foregoing parameters, an analysis was conducted to ascertain the effects of radiant energy input on both temperature measurement error and maximum thermocouple sheath temperature. The analysis was conducted using the Rocketdyne Differential Equation Analyzer Program (DEAP) to predict the thermocouple temperature distribution. A sketch of the nodal model utilized in the analysis is shown in Fig. 48. A 1/16-inch O.D. sheathed closed tip thermocouple was taken as a baseline configuration. The sheath material was CRES with a nominal wall thickness of 0.010 inch. The surface reflectance was taken as 0.9 based on utilizing a high temperature gold tape to protect the thermocouple sheath.

Figure 48. Nodal Representation of Thermocouple

The analysis was conducted for a crown heat flux level of $1.0 \text{ Btu/in.}^2\text{-sec.}$ Various amounts of circumferential weld were investigated. The circumferential tube wall temperature as calculated from a different DEAP model analysis was taken as a reference for thermocouple evaluation.

Preliminary results assuming only the thermocouple tip was welded to the tube crown indicated a temperature error of about $+100 \text{ F}$ resulting from radiant heat input and conduction through the tip to the tube. Based on a tube crown temperature of 1400 F , the maximum thermocouple sheath temperature was about 1775 F about 45 degrees (angular) from the crown.

The use of a full weld (or braze) from the crown to the 90 -degree (angular) point reduced the tip temperature error to 50 F with a maximum sheath temperature of 1500 F .

These analytical results indicate the approximate minimum temperature measurement error likely to occur in the radiant heating tests at the maximum heat flux level of $1.0 \text{ Btu/in.}^2\text{-sec.}$ At reduced heat flux levels the error would be proportionately less. Reduction in thermocouple reflectance would tend to increase the error due to higher heat absorption.

Experimental Results. The heat flux calibration tests discussed previously were also utilized to evaluate thermocouple installation techniques in conjunction with the computer data acquisition system. Two instrumented five-foot long O.D. CRES tubes were utilized in these tests. The calibration tests were conducted using low pressure subcooled water which permits accurate analytical prediction of wall temperature as a function of imposed heat flux. The water flowrate was reduced gradually at a fixed heat flux level until the wall temperature no longer increased thereby insuring operation in the nucleate boiling regime.

A typical temperature response of an ungrounded silver brazed thermocouple is shown in Fig. 49 at a nominal heat flux of $1.0 \text{ Btu/in.}^2\text{-sec.}$ At the initial nominal water flowrate the surface wall temperature is seen to be about 550 F .

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Figure 49. Measured Surface Temperature of 3/8-inch O.D. Calibration Tube

The flow is decreased at about 90 seconds into the run with a subsequent increase in wall temperature to about 590 F. Further reductions in coolant flowrate do not result in increased wall temperature indicating that nucleate boiling has been achieved.

An analytical wall temperature prediction was made using the Rocketdyne Differential Equation Analyzer Program in conjunction with test operating conditions. The resulting temperature distribution is shown in Fig. 50. The maximum predicted surface temperature is seen to be 500 F. This compares with a measured value of 590 F indicating a discrepancy of 90 F. The estimated error in the calculated wall temperature is about ± 25 F at these nucleate boiling conditions. The thermocouple error due to radiant heat input would, therefore, be estimated at $90 \text{ F} \pm 25 \text{ F}$ at a nominal heat flux of $1.0 \text{ Btu/in.}^2\text{-sec.}$

The tip-welded thermocouples recorded temperatures several hundred degrees higher than the silver brazed thermocouple indicating that the former attachment technique is unsatisfactory in a high radiant heat flux environment. The grounded thermocouples were unsatisfactory in terms of system noise at the high power levels utilized in the calibration tests.

The higher than predicted temperature discrepancies may be attributed to a lower surface reflectance than utilized in the analysis. The high temperature gold reflective tape applied to the thermocouple surfaces proved unsatisfactory. The tip-welded thermocouples would be expected to be more sensitive to increased heat loads resulting from lower surface reflectance.

The thermocouple installed through the tube such that its tip was flush with the tube crown also proved unsatisfactory due to excessive cooling by the water flowing through the tube and around the exposed thermocouple sheath. This thermocouple tended to measure temperatures only slightly above the coolant bulk temperature.

Based on the foregoing results, the silver brazed thermocouple attachment technique was selected for use in the single tube and panel tests.

ROUND TUBE ANALYSIS - RADIANT HEAT INPUT
NUCLEATE BOIL AT P=65 PSIA

DEEP-11
12/01/75

#033508226
120175 0001
CASE NO. 1

INCIDENT HEAT FLUX = 1.0 BTU/IN²-SEC

ISOTHERM
LEGEND

A = 100
B = 125
C = 150
D = 175
E = 200
F = 225
G = 250
H = 275
K = 300
L = 325
M = 350
N = 375
O = 400
P = 425
R = 450
S = 475
T = 500
U = 525
X = 550
Z = 600

Figure 50. Analytical Prediction of Calibration Tube Temperature Distribution

TEST PROCEDURES

The test procedures discussed in this section apply to both the single tube and multi-tube test sections.

Prior to testing, three different flow measurement orifices were calibrated on-site by determining the required time to fill a known volume container. The three orifices were selected to provide measurement over a wide range of flowrates at sufficient orifice pressure drop so as to insure reasonable accuracy. The orifice calibrations were conducted at various back pressures, ranging from atmospheric to 2000 psia, to ascertain possible pressure effects on the transducer. No pressure effects were noted. The orifices were also calibrated periodically during the test effort to assure that no zero-shift or possible plugging occurred.

The various graphite heater/reflector configurations were calibrated in the subscaled test rig discussed previously. These calibration data were used to determine preliminary power settings. Final heater calibration was determined on-site based on actual heat absorbed by a known water (single-phase) flowrate. The various types of heaters (short or long) were calibrated in groups rather than individually. The actual power input into each short (high heat flux) heater was recorded individually and utilized in conjunction with total heat input to estimate individual heater heat flux level.

Desired system back pressure was achieved by the use of various selected downstream (outlet) orifices. This technique proved more satisfactory than the use of the available back pressure regulator which was too large for the flow range of interest.

The actual run procedure was relatively simple. The water tank was pressurized to the necessary level, usually 2000 to 3000 psia, with gaseous nitrogen. The flowrate was established at the desired level with the flow control valve. The power to the in-line heaters was increased until the desired water inlet temperature ($\approx 400 - 500$ F) was achieved. The water pre-heat normally resulted in an increase in back pressure to about 300-500 psia as the liquid flashed in the downstream orifice.

Once the inlet temperature was stabilized (≈ 3 to 5 minutes), the radiant heater power was increased to the desired level. The short high heat flux heaters were, in general, activated about 30 to 60 seconds after the low flux heaters such that all the reflectors achieved their limiting temperatures at approximately the same time.

Various reflector temperatures were monitored during the test. The test was terminated when the aluminum reflectors achieved a maximum temperature of about 500 - 600 F.

During the test, all pressures, flow, temperatures, and short heater power input were continuously recorded on strip charts or by the computerized data acquisition system.

TEST RESULTS

SINGLE TUBE

A series of tests were conducted with the 3/8-inch O.D. stainless steel tube test section in the horizontal radiant heating facility. Figure 51 shows the single tube test section in the test facility prior to installation of the heaters and aluminum reflector assemblies. A total of 23 tests were conducted including calibration runs to determine average heat flux levels of various heater sections. A summary of nominal test conditions is given in Table 2. Essentially 16 of the tests resulted in either two-phase or superheated steam exit conditions at pressures ranging from 1100 to 2250 psia.

The initial four tests were conducted with 15 long (≈ 52 -inch) graphite heater assemblies using only top and bottom reflectors (the final reflector assemblies were not yet completed). Nominal uniform heat flux profiles ranging from 0.11 to 0.18 Btu/in.²-sec were achieved. Tests 5 through 11 were conducted with the desired reflector assemblies (Fig. 43) installed. Heat flux levels of about 0.24 Btu/in.²-sec resulted from the improved reflectors with the system operating at about 90 percent of maximum power. Several attempts to achieve maximum rated transformer power output (≈ 50 kw) resulted in blown variac fuses due to excessive currents.

Tests 12 to 23 were conducted with a revised heater configuration to achieve increased heat flux levels. The three central 52-inch long graphite heaters were replaced with six half-length (26-inch) heaters. Each of these heaters was connected to 100 kw power sources. Based on earlier calibration tests, this configuration should have provided a maximum heat flux capability of about 1.0 Btu/in.²-sec. The two long heaters on either side of the short central heaters were replaced with three thinner (1/4 vs 3/8-inch thick), intermediate length (≈ 34 inches) heaters.

Figure 51. Single Tube Test Section
Installed in Facility

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Table 2. Summary of Single Tube Tests

TEST NO.	HEAT FLUX, BTU/IN ² -SEC	MASS FLUX, LB/IN ² -SEC	COOLANT PRESSURE, PSIA	COOLANT PRESSURE DROP, PSI	INLET TEMPERATURE, F	OUTLET TEMPERATURE, F	COMMENTS
1	0.11	2.29	250	-	73	263	CALIBRATION
2	0.15	0.79	1300	~10-15	580	735	
3	0.15]	0.70	1590	10	602	650	TWO-PHASE
4	0.18	0.68	1250	~10-15	484	1010	
5	0.20	2.41	240	15	63	385	CALIBRATION, IMPROVED REFLECTORS
6	0.25+	1.09	-	-	500	-	HEATER FUSE BLOWN
7	0.25+	1.09	-	-	500	-	HEATER FUSE BLOWN
8	0.25+	1.09	-	-	500	-	HEATER FUSE BLOWN
9	0.22	1.09	1800	15	527	741	
10	0.25+	1.09	-	-	500	-	HEATER FUSE BLOWN
11	0.25	1.14	1900	15	508	780	
12	0.27	1.08	1515	17	540	869	CHANGE TO SHORT CENTER HEATER
13	0.33	1.44	1590	15	479	857	
14	0.30	1.42	1800	15	620	1380	IN-LINE HEATER POWER INCREASED INADVERTENTLY, TWO-PHASE INLET CONDITIONS
15	0.30	1.44	1965	11	469	627	TWO-PHASE OUTLET - REFLECTORS DAMAGED
16	0.40	1.42	1890	11-15	449	628	TWO-PHASE OUTLET CONDITIONS
17	0.40	1.44	1950	11	456	632	TWO-PHASE OUTLET CONDITIONS
18	0.40	1.44	2190	12	417	645	
19	0.40	-	-	-	-	-	COMPUTER SHUTDOWN, TEST TERM.
20	0.40	1.14	1965	12	491	678	
21	0.50	1.09	2250	14	560	918	
22	0.50	1.05	1100	31	494	785	
23	0.71	1.20	1775	18	490	700	MAXIMUM FACILITY POWER

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Several top reflectors were damaged due to overheating while trying to extend the test duration to about 4 minutes. (Subsequent tests were limited to about three minutes.) It was determined that insufficient contact area between the reflector and the cooled (with water-saturated molding sand) aluminum housing resulted in the melting of the reflectors. The top reflectors over the short and intermediate heaters were removed and the aluminum housing surface was buffed and utilized directly as the top reflector.

The last test of the series (No. 23) was conducted at maximum facility power in order to determine the maximum heat flux capability of the facility. This was determined to be about $0.71 \text{ Btu/in.}^2\text{-sec}$ based on a subsequent calibration run. This heat flux level is somewhat below that achieved in the subscale calibration rig and is believed due to the wider side reflector spacing required for uncooled operation. (The subscale test utilized a water convective cooled reflector assembly deemed too costly for full scale application.)

Heat Flux Profiles

The various heat flux profiles utilized in the single tube test series are shown in Fig. 52. The profiles are numbered for cross-reference to the resulting wall temperature profiles presented later. At the maximum transformer setting of 50 volts, considerable differences in power output to the six short central heaters were noted, resulting in the uneven maximum heat flux profiles. The reason for the power variation may have been due to contact resistance effects between the heater and supporting buss bars.

The heat flux profiles actually achieved differ from the nominal design profile noted previously (Fig. 6). In the low heat flux tests ($< 0.25 \text{ Btu/in.}^2\text{-sec}$) a uniform axial profile was utilized to simulate 5-point aim systems currently under study. Such a multi-point aim system results in a nearly uniform axial heat load on the solar receiver and appears more desirable in terms of receiver thermal design.

Figure 52. Heat Flux Profiles - Single Tube Tests

Increased heat flux levels (0.30 Btu/in.²-sec) could only be achieved in this facility in the 12-foot center section (i.e., six short heaters) and resulted in a much steeper and narrower heat flux profile than originally planned. Such a profile represents a most severe condition in terms of cooling since the resulting reduced average-to-peak heat flux level results in reduced coolant flowrate (and mass flux) to achieve the desired exit temperature. Although the higher heat flux tests were conducted at coolant mass flux levels less than the predicted requirement, the tube operating temperatures remained acceptable as will be seen shortly.

Temperature Response

The test durations appeared to be adequate in most cases to achieve near equilibrium temperatures. Typical computer plots of the fluid outlet temperature and a selected wall temperature transient response are shown in Fig. 53 and 54. The coolant pressure responses were relatively smooth as shown in Fig. 55. During heat-up, the flow tended to decrease somewhat requiring manual adjustment of the flow control valve to maintain the desired flowrate.

In some cases, as will be seen in the next section, the fluid outlet temperature tended to peak and then start downward as the test was terminated. The temperature peak may be due to a flow compression during start as the back pressure builds up across the outlet orifice. Such a compression was not reflected upstream to the flow measurement orifice or inlet pressure.

Figure 53. Single Tube Outlet Temperature Response - Test No. 21

LEGEND: Thermocouple No. 27
Location: 42 ft from Inlet

Figure 54. Single Tube Wall Temperature Response - Test No. 21

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Figure 55. Single Tube Pressure Responses - Test No. 21

Flow Stability

A primary concern in a once-through steam generator is flow stability. The majority of the tests were completely stable in terms of flow and pressure as noted in the typical traces of Fig. 55 previously. Four of the tests (No. 11, 12, 14 and 22), however, encountered flow and pressure oscillations during the radiant heating transient. These oscillations were of small magnitude, as shown in Fig. 56, and were damped out after about three cycles. Maximum flow variation was about ± 3 percent.

The flow oscillation was reflected in outlet temperature and the wall temperature response as shown in Fig. 57 and 58. The wall temperature is seen to fluctuate about ± 75 F but not to exceed the final steady state value. The outlet temperature variation was much less.

The reason for the flow oscillation during heat-up on these particular tests is not known. The absolute flows and heat flux levels were neither the minimum or maximum tested and the radiant heater heat-up rate was similar to other non-oscillatory tests.

The important factor is that the oscillations were rapidly damped indicating inherent stability over the operating range of interest. It should be pointed out that no orifice was utilized at the test section inlet which is the usual technique for stabilizing flow. The flow measurement orifice was located approximately 20 feet upstream of the test section inlet and, as such, would not be expected to influence the test section stability characteristics.

FIGURE 56. Single Tube Pressure Responses - Test No. 11

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Figure 57. Single Tube Outlet Temperature Response - Test No. 11

LEGEND: Thermocouple No. 27
Location: 42 ft from Inlet

Figure 58. Single Tube Wall Temperature Response - Test No. 11

Temperature Profiles

The measured temperature profiles for all of the tests which achieved two-phase or super-heat outlet conditions are presented in Fig. 59 to 72. The temperatures are uncorrected for thermocouple error and are, therefore, somewhat higher than the actual tube temperature. For the lower heat flux regions ($Q/A = 0.2 \text{ Btu/in.}^2\text{-sec}$), the heated surface temperature error is estimated to be less than 20 F based on earlier subscale calibration tests. In the test section central region at increased heat flux levels, the maximum measured heated surface temperature error is about 60 F at a heat flux of $0.7 \text{ Btu/in.}^2\text{-sec}$.

The back wall temperature profiles are also shown in Fig. 59 - 72. These temperatures do not require correction since they are not exposed to the radiant heat flux. The back wall temperatures are good indicators of the fluid temperature profile since there is minimal circumferential thermal conduction with the thin wall tubing utilized. The end of the two-phase region can, therefore, be estimated from the back wall temperature measurements.

The temperature difference between hot-wall and back wall temperature is a prime factor affecting cyclic fatigue life as noted previously in Fig. 19. The maximum temperature differential noted was about 350 F which occurred in Test 21 (Fig. 70). Based on a measured back wall temperature of 650 F, the allowable heated surface temperature for 10,000 cycles is about 1250 F (Fig. 19), resulting in an allowable temperature differential of 600 F. Based on the measured temperature differential, it is apparent that the solar panel tube is capable of well in excess of the required 10,000 thermal cycles.

In actuality, the estimated tube fatigue life is in excess of 50,000 cycles such that any attempt to induce a fatigue life failure through thermal cycling was well beyond the capability of the current contractual effort.

Figure 59. Single Tube Temperature Profile

Figure 60. Single Tube Temperature Profile

Figure 61. Single Tube Temperature Profile

Figure 62. Single Tube Temperature Profile

Figure 63. Single Tube Temperature Profile

Figure 64. Single Tube Temperature Profile

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Figure 65. Single Tube Temperature Profile

Figure 66. Single Tube Temperature Profile

Figure 67. Single Tube Temperature Profile

Figure 68. Single Tube Temperature Profile

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Figure 69. Single Tube Temperature Profile

Figure 70. Single Tube Temperature Profile

Figure 71. Single Tube Temperature Profile

Figure 72. Single Tube Temperature Profile

The most interesting aspect of the wall temperature profiles is the lack of a significant temperature peak in the two-phase region as was predicted with the computer analysis (see Fig. 10). Maximum two-phase region wall temperatures were less than 1000 F in all tests. These wall temperatures appeared relatively unaffected by coolant pressure level. More detailed comparison of measured temperature profiles with the analytical model is presented in a later section.

One of the most severe tests (No. 14) performed on the single tube resulted when the in-line preheater power level was increased inadvertently along with the radiant heaters. This resulted in a two-phase inlet condition and a maximum steam outlet temperature of 1380 F before the in-line heater power could be reduced. The resulting maximum temperature profile is shown in Fig. 66. The two outlet end hot-wall thermocouples recorded a maximum reading of about 1500 F near the end of the test before the silver braze melted and the thermocouples became detached from the surface. Based on the temperature rise rate, it is estimated that the maximum surface temperature was about 1600 F. No damage was done to the test section and it performed satisfactorily in subsequent tests.

Posttest Condition

Upon completion of the single tube test phase the test section was inspected and found to be in satisfactory condition with no leaks. The S-31 high absorptance point, however, was found to have flaked off in numerous locations as shown in Fig. 73. The loss of this paint may have resulted from inadequate curing of the paint prior to test. Based on the difficulty of curing this paint on-site, an alternate high absorptance paint (Pyromark) was selected for evaluation on the panel. This alternate paint has equal or better solar

Figure 73. Condition of S-31 High Solar Absorptance Paint after Test

absorptance and is more readily applied and cured on-site. In particular, the Pyromark paint air-dries and can then be handled without loss of paint while being installed in the facility.

FIVE-TUBE PANEL

Upon completion of the single tube test series, the five-tube panel was installed in the radiant heating facility (Fig. 74). A series of eleven tests were conducted, including heater calibrations, and the pertinent test conditions are summarized in Table 3. The same multiple length heater array and reflector assemblies as completed the single tube tests were utilized for panel testing.

The panel incorporated individual orifices (0.052 in.) at the inlet of each tube for flow measurement purposes. Only 3 tube flowrates could be measured during any one test due to availability of only three pressure transducers (in addition to the transducer used for total flow measurement). Inlet and outlet pressure taps were located in the respective manifold assemblies. Taps were located near the end of each tube (upstream of the exit manifold) to monitor individual outlet temperatures. An additional temperature tap was added downstream of the exit manifold to determine overall steam exit temperature. Due to limited computer channels, one of the individual outlet temperature measurements had to be sacrificed to obtain the bulk average outlet temperature record.

Heat Flux Profiles

The various resulting heat flux profiles are shown in Fig. 75 and are numbered for cross-reference with the measured temperature profiles. In general, the average heat flux levels for the panel tests were slightly lower than for the tube since a portion of the lower reflector area is replaced by the high absorptance panel.

Figure 74. Five-Tube Panel Installed in Test Facility

Table 3. Summary of Five-Tube Panel Tests

TEST NO.	HEAT FLUX, BTU/IN ² -SEC	MASS FLUX, L3/IN ² -SEC	COOLANT PRESSURE, PSIA	*COOLANT PRESSURE DROP, PSI	INLET TEMPERATURE, F	AVERAGE OUTLET TEMPERATURE, F	CENTER TUBE OUTLET TEMPERATURE, F	COMMENTS
24	0.23	0.72	1265	28	321	1045	1096	CALIBRATION
25	0.23	1.07	1740	45	412	793	833	
26	0.23	1.07	1865	45	451	950	1019	
27	0.23	1.06	2300	42	450	890	971	
28	0.40	1.14	500	35	65	359	371	CALIBRATION
29	0.40	1.27	2155	54	382	734	784	
30	0.40	1.30	2200	55	423	764	879	
31	0.40	1.07	1100	76	500	979	1071	
32	0.65	1.31	600	35	73	389	432	CALIBRATION
33	0.65	1.32	2000	52	426	643	800	
34	0.65	1.05	2000	45	472	745	826	

* INCLUDES INLET ORIFICE PRESSURE DROP

Figure 75. Five-Tube Panel Heat Flux Profiles

Again it is noted that at the maximum facility voltage setting the varying power output capability of the six center heater transformers resulted in the uneven heat flux profile shown. A maximum heat flux level of about 0.65 was achieved for the panel tests.

Temperature Response

Total test durations for all panel tests were approximately three minutes. For the high heat flux tests, the central six heaters were activated about 1 to 1½ minutes after the rest of the heater assemblies.

A typical outlet temperature response curve is shown in Fig. 76 . This temperature measurement was located at the outlet of the exit collection manifold and represents an average of the five tubes. A comparison of the outlet temperature response of the individual tubes is shown in Fig. 77. (Only 5 outlet temperature measurements were possible on a single test so that one of the tube temperatures was sacrificed after Test No. 25 to obtain a total outlet temperature measurement.)

The difference in heat input across the panel is due primarily to the fact that the heater width of 1.38 inches is somewhat narrower than the total panel width of 1.56 inches. The side reflectors tend to compensate somewhat for the narrower heater. The heat flux differential across the panel represents an extra severity to the test conditions in terms of differential thermal stress. Such a differential could actually occur in a solar tower receiver due to partial cloud cover of the heliostat field. The comparative wall temperature response profiles of three adjacent tubes is shown in Fig. 78. The maximum tube surface differential is seen to be about 150 F.

Flow Stability

None of the tests conducted with the panel exhibited any instabilities in terms of flow or inlet or outlet pressure. All starts were smooth as can be seen from the typical pressure traces shown in Fig. 79 . No oscillations, such as occurred in the single tube tests, were noted with the panel. This

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Figure 76. Panel Inlet and Average Fluid Outlet Temperatures - Test No. 26

Figure 77. Tube Outlet Temperatures - Test 26

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Figure 78. Tube Wall Temperatures near Outlet - Test 26

Figure 79. Panel Pressure Response - Test 26

relative stability may have resulted from the higher coolant mass fluxes (≈ 40 percent) inherent in the smaller panel tubes in conjunction with the individual tube inlet orifices ($d = .052$ inch).

A comparison of the tube inlet orifice pressure traces during the tests, however, indicate small flow oscillations between adjacent tubes as the system settled into equilibrium conditions. Typical inlet orifice pressure differential responses are shown in Fig. 80 . Only three of the tubes could be monitored on any given test due to limitation of available pressure transducers. The individual flow oscillations did not appear to have a significant effect on wall temperatures or outlet temperatures or pressures as noted previously.

Temperature Profiles

The measured temperature profiles for all of the tests which achieved two-phase or super-heated outlet conditions are presented in Fig. 81 to 88 . Fewer axial temperature measurements were made on the panel in order to obtain greater information on tube-to-tube differentials within the total instrumentation limitation of 48 computer channels.

Again, it is noted that the two-phase region wall temperatures (uncorrected) do not exhibit a peak and are less than 900 F in all cases.

In general, maximum tube-to-tube temperature variations occur near the tube exit due primarily to differences in tube coolant exit temperatures as previously discussed.

More detailed data analysis is presented in a later section.

Posttest Inspection

During the last planned panel test (no. 34), it was noted that the expected exit temperature (≈ 950 F) was not achieved and that at least two heaters were inoperative. The reflector assemblies were removed and the heater

Figure 80. Inlet Orifices Pressure Drop - Test 26

TEST NO. 25	SYMBOL	TUBE NO.
Q/A PROFILE NO. 8	○	1
$G_c = 1.07 \text{ LB/IN}^2\text{-SEC}$	▽	2
$\Delta P = 45 \text{ PSI}$	○	3
$P = 1740 \text{ PSIA}$	□	4
$T_{IN} = 412 \text{ F}$	△	5
$T_{OUT\text{AVG}} = 793 \text{ F}$	OPEN SYMBOLS - HEATED SURFACE	
	CLOSED SYMBOLS - BACK SURFACE

Figure 81. Five-Tube Panel Temperature Profiles

TEST NO. 26	SYMBOL	TUBE NO.
Q/A PROFILE NO. 8	□	1
$G_C = 1.07 \text{ LB/IN}^2\text{-SEC}$	▽	2
$\Delta P = 45 \text{ PSI}$	○	3
$P = 1865 \text{ PSIA}$	□	4
$T_{IN} = 451 \text{ F}$	△	5
$T_{OUT\text{AVG}} = 950 \text{ F}$	OPEN SYMBOLS - HEATED SURFACE	
	CLOSED SYMBOLS - BACK SURFACE

Figure 82. Five-Tube Panel Temperature Profiles

TEST NO. 27	SYMBOL	TUBE NO.
Q/A PROFILE NO. 8	○	1
$G_C = 1.06 \text{ LB/IN}^2\text{-SEC}$	▽	2
$\Delta P = 42 \text{ PSI}$	○	3
$P = 2300 \text{ PSIA}$	□	4
$T_{IN} = 450 \text{ F}$	△	5
$T_{OUT\text{AVG}} = 890 \text{ F}$	OPEN SYMBOLS - HEATED SURFACE	
	CLOSED SYMBOLS - BACK SURFACE

Figure 83. Five-Tube Panel Temperature Profiles

TEST NO. 29	<u>SYMBOL</u>	<u>TUBE NO.</u>
Q/A PROFILE NO. 9	○	1
$G_C = 1.27 \text{ LB/IN}^2\text{-SEC}$	▽	2
$\Delta P = 54 \text{ PSI}$	○	3
$P = 2155 \text{ PSIA}$	□	4
$T_{IN} = 382 \text{ F}$	△	5
$T_{OUT\text{AVG}} = 734 \text{ F}$	OPEN SYMBOLS - HEATED SURFACE	
	CLOSED SYMBOLS - BACK SURFACE

Figure 84. Five-Tube Panel Temperature Profiles

TEST NO. 30	SYMBOL	TUBE NO.
Q/A PROFILE NO. 9	○	1
$G_C = 1.30 \text{ LB/IN}^2\text{-SEC}$	▽	2
$\Delta P = 55 \text{ PSI}$	○	3
$P = 2200 \text{ PSIA}$	□	4
$T_{IN} = 423 \text{ F}$	△	5
$T_{OUT_AVG} = 764 \text{ F}$	OPEN SYMBOLS - HEATED SURFACE	
	CLOSED SYMBOLS - BACK SURFACE

Figure 85. Five-Tube Panel Temperature Profiles

TEST NO. 31	SYMBOL	TUBE NO.
Q/A PROFILE NO. 9	○	1
$G_c = 1.07 \text{ LB/IN}^2\text{-SEC}$	▽	2
$\Delta P = 76 \text{ PSI}$	○	3
$P = 1100 \text{ PSIA}$	□	4
$T_{IN} = 500 \text{ F}$	△	5
$T_{OUT\text{AVG}} = 979 \text{ F}$	OPEN SYMBOLS - HEATED SURFACE	
	CLOSED SYMBOLS - BACK SURFACE

Figure 86. Five-Tube Panel Temperature Profiles

TEST NO. 33	SYMBOL	TUBE NO.
Q/A PROFILE NO. 10	○	1
$G_C = 1.32 \text{ LB/IN}^2\text{-SEC}$	▽	2
$\Delta P = 52 \text{ PSI}$	○	3
$P = 2000 \text{ PSIA}$	□	4
$T_{IN} = 426 \text{ F}$	△	5
$T_{OUT_AVG} = 643 \text{ F}$	OPEN SYMBOLS - HEATED SURFACE	
	CLOSED SYMBOLS - BACK SURFACE

Figure 87. Five-Tube Panel Temperature Profiles

TEST NO. 34	SYMBOL	TUBE NO.
Q/A PROFILE NO. 10	○	1
$G_c = 1.05 \text{ LB/IN}^2\text{-SEC}$	▽	2
$\Delta P = 45 \text{ PSI}$	○	3
$P = 2000 \text{ PSIA}$	□	4
$T_{IN} = 472 \text{ F}$	△	5
$T_{OUT \text{ AVG}} = 745 \text{ F}$	OPEN SYMBOLS - HEATED SURFACE	
	CLOSED SYMBOLS - BACK SURFACE

Figure 88. Five-Tube Panel Temperature Profiles

assemblies inspected. Three of the heaters were broken, one in the center and two at the ends. Numerous heaters had also eroded visibly during the test series. This erosion, in general, appeared to occur at the heater ends where they were joined to the graphite buss bars.

The two heaters broken at the ends were short, high-heat flux heaters (No. 11 and 17). The center break heater (No. 27) was of intermediate length (≈ 34 inches) and was thinned to virtually zero thickness in the center where the break occurred. Such severe erosion could result from a water/steam spray due to a tube leak.

The panel was hydrostatically pressure-checked and, indeed, three leaks were determined. Two of the leaks were in the center tube crown directly under the badly eroded and broken heater. The two leaks were about 1/4-inch apart and had the appearance of electrical discharge type damage. They are shown in Fig. 89 . Apparently either the heater sagged or the panel bowed upward during a test and the heater shorted to the tube, resulting in the electrical burn-through.

The third panel leak was in an outer tube approximately 90 degrees from the tube crown at the outer edge of the panel. This hole also had the appearance of an electrical discharge burn-through. This leak occurred under one of the long heaters and apparently resulted when the panel touched a graphite support utilized to prevent sagging of these long heaters. (No supports were utilized for the short or intermediate length heaters.) More discussion on the effect of the leaks is presented in the Data Analysis section.

None of the twelve repaired leaks resulting from the longitudinal welding reappeared, indicating the availability of satisfactory repair techniques. The three leaks that resulted during testing are readily repairable using similar weld or silver braze techniques. No repair was undertaken since the test program was essentially completed satisfactorily.

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Figure 89. Posttest Condition of Five-Tube Panel

Inspection of the longitudinal welds along the back of the panel indicated no visible cracking. The light-tight seal remained intact.

The Pyromark paint blistered and flaked badly in numerous areas (Fig. 90) while in other regions it remained in excellent condition. Loss of paint occurred in low as well as high heat flux regions. The poor performance of the Pyromark paint was somewhat surprising since it had survived intact on the subscale (5-foot) calibration tube at heat fluxes up to $1.0 \text{ Btu/in.}^2\text{-sec.}$

It is possible that water spill-over from the reflectors (less sand and more water was utilized in the panel top reflector) resulted in a steam atmosphere adjacent to the hot tube which may have deteriorated the paint. More detailed investigation of the high solar absorptive paint problem is currently under way in Rocketdyne's 10 MW Solar Receiver program.

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Figure 90. Posttest Condition of Pyromark Paint

DATA ANALYSIS

SINGLE TUBE TESTS

The single tube tests provided more detailed axial temperature profiles than the panel test series due to a limited number of available thermocouple recording channels. The single tube data were, therefore, compared with predicted temperature profiles as determined from an existing computer model (Appendix A).

As mentioned previously, the measured wall temperatures did not appear to exhibit a peak in the two-phase region as predicted in the Preliminary Analysis section of this report. A comparison of the data taken in Test 11 with predicted wall temperatures for a uniform axial heat flux of 0.25 Btu/in.²-sec is shown in Fig. 91. The original computer model, with a two-phase film boiling region pressure correction factor, indicates a maximum temperature of about 1025 F as compared to a measured value of 815 F. Deletion of the pressure correction factor provides a much closer approximation of the actual data although about 50 F low. In the nucleate boiling and super-heat regimes, wherein both models are identical, agreement between predicted and measured temperatures is excellent.

Only one test (No. 21) in the single tube test series appeared to exhibit a temperature peak as expected. This test was also compared with analytical predictions using the input heat flux profile noted. The resulting comparison is shown in Fig. 92. In this case, deletion of the pressure correction factor underestimates the measured peak temperature by about 150 F whereas the original model predicts about 75 F high. Again, agreement in the nucleate boiling and super-heat regions is good.

Test 22 utilized the same heat flux profile as Test 21 but was conducted at a pressure of about 1100 psia (vs 2250 psia for Test 21) to ascertain possible coolant pressure effects. Previous company-sponsored experimental

Figure 91. Comparison of Analytical and Experimental Tube Wall Temperature Profiles

TEST NO. 21

Q/A PROFILE NO. 6

$G_C = 1.09 \text{ LB/IN}^2\text{-SEC}$

$\Delta P = 14 \text{ PSI}$

$P = 2250 \text{ PSIA}$

$T_{IN} = 560 \text{ F}$

$T_{OUT} = 918 \text{ F}$

○ HEATED SURFACE TEMP.

□ BACK SURFACE TEMP.

Figure 92. Comparison of Measured and Predicted Temperature Profiles

work with an electrical heated tube indicated this pressure reduction would result in a significant increase in wall temperature. In actuality, the peak wall temperature was considerably less at the reduced pressure.

A comparison of the data with the predicted temperature profiles for Test 22 is shown in Fig. 93. Inclusion of the pressure correction factor grossly overpredicts the temperature by nearly 1000 F, whereas the modified model overpredicts by about 200 F.

The predicted and measured pressure drops are in good agreement as noted on Fig. 91 to 93.

It appears at this time that the modified computer model, without the two-phase film boiling region pressure correction factor, is best suited for analytical design of solar tower receiver tubes. Prediction of peak two-phase region wall temperature generally within about 100 to 150 F is possible with this model. Further experimental work in the two-phase regime is required to better define the controlling cooling parameters over a wide range of conditions.

It should be pointed out that one of the primary reasons for conducting full scale tube and panel tests in this program was to determine actual wall operating temperatures under simulated solar heating conditions rather than rely on analytical predictions.

FIVE-TUBE PANEL TESTS

The five-tube panel temperature profiles were, in general, similar to the results obtained with the single tube test series. Comparison with analytical predictions would, therefore, be similar to that discussed previously.

One of the most interesting aspects of the panel tests concerns the effect of the leaks that developed in the center tube due to contact with the heater. The inlet orifice pressure drop traces were reviewed in an attempt

Figure 93. Comparison of Predicted and Measured Temperature Profiles

to ascertain when the leaks occurred. Based on this review, it was concluded that the leaks occurred near the end of Test 30.

The inlet orifice pressure drop traces for Test 30 are shown in Fig. 94. The center tube trace (Channel 48) is denoted by a plus (+) symbol. It can be seen that at about 150 to 160 seconds the center tube orifice pressure drop suddenly increases and settles on a value about 25 percent higher than earlier in the run. This would imply an increase in total center tube flow of about 12 percent.

Reviewing the individual tube outlet temperature traces shown in Fig. 95 indicates a jump in center tube fluid outlet temperature (denoted by a small square) at about the same time period. Similarly, a typical center tube wall temperature in the super-heat region registers a sharp increase as shown in Fig. 96. (The extremely sharp spikes shown in Fig. 95 appear to result from a computer malfunction and/or electrical noise since these events were not observed on the control center strip charts.)

Based on the increased temperature differential between center and outer tubes after the leak occurred, it appears that the center tube flow reduction downstream of the leak amounted to about 7 to 10 percent as compared to nominal operation. This value is consistent with loss calculations based on treating the two holes in the tube as orifices of 0.020 and 0.0135 inch diameter.

As a result of this decreased flow in the center tube, the wall temperature in the super-heat region increases due both to decreased coolant mass flux and increased local coolant temperature. The most pronounced temperature difference between center and outer tube would, therefore, be expected near the panel exit. This effect is readily apparent in Fig. 97 wherein the center tube wall temperature at cutoff is about 200 F higher than the outer tube.

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Figure 94. Inlet Orifices Pressure Drop - Test 30

Figure 95. Outlet Temperatures - Test 30

Figure 96. Center Tube Wall Temperatures - Test 30

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Figure 97. Tube Wall Temperatures Near Outlet End - Test No. 30

The important factor to note is that in spite of leaks and a partially starved tube the resultant wall temperatures were not excessive and the panel operated satisfactorily. The average fluid outlet temperature was virtually unaffected as can be seen in Fig. 98. This implies a higher degree of reliability to the once-through steam generator concept than might have been previously believed. Minor tube leaks can be tolerated without serious consequences to solar tower receiver panels or overall system operation.

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Figure 98. Panel Inlet and Average Outlet Fluid Temperatures

APPLICATION TO COMMERCIAL SOLAR TOWER DESIGN

The experimental results obtained during this program were reviewed in terms of influencing the design of a commercial solar tower receiver, as reported in Ref. 1. The thermal analysis relating to that commercial design effort utilized an earlier version of the computer model discussed herein (Appendix A) that did not contain a pressure correction factor in the two-phase film boiling regime. The earlier analysis is, therefore, generally supported by the test results obtained under the current contractual effort.

The test data did indicate that a reduced coolant mass flux and larger tube diameter is feasible for a commercial design compared to original estimates. The 3/8-inch O.D. tube test results were, therefore, utilized to evaluate a modified tube design for solar tower receiver application.

The baseline design selected utilized a 3/8-inch O.D. x 0.040 in. wall Incoloy 800 tube. The worst case single-point aim heat flux profile (from Ref. 1), shown in Fig. 3 previously, was compressed (not truncated) to provide the total heat input consistent with the desired coolant mass flux and end point fluid temperatures ($T_{in} = 400$ F, $T_{out} = 950$ F). The resulting tube length was 64 feet.

The resulting temperature profiles for the solar tower receiver tube are shown in Fig. 99. A maximum wall temperature of about 1060 F occurs about 2/3 of the length from the inlet. This wall temperature, in conjunction with previous stress analyses, indicates a feasible design for commercial solar tower receiver application. Coolant pressure drop is 20 psi plus sufficient inlet orifice pressure drop (≈ 40 psi) to insure stable flow.

Figure 99. Typical Temperature Profiles for a Commercial Solar Tower Receiver

CONCLUSIONS

The once-through steam generator concept is feasible for commercial solar tower receiver application in terms of thermal, structural, and flow stability characteristics. Tube wall operating temperatures of less than 1200 F can be maintained over a wide range of operating conditions with coolant pressure drops of 12 - 30 psi and a reasonable tube diameter (outer) of 3/8 inch, or perhaps larger.

The candidate high solar absorptance paints needed to improve receiver efficiency require further evaluation in terms of durability. The unsatisfactory results (flaking and peeling) obtained herein for two different paints are probably due to improper curing and/or a possible steam environment resulting from reflector cooling water entering the heater/test section confinement.

The existing computer program, minus the two-phase pressure correction, appears capable of adequately predicting the wall temperature profile of a once-through steam generator tube.

The use of graphite radiant heaters to simulate the heat flux levels associated with concentrated solar energy is feasible. Actively (water) cooled reflectors are desirable at higher heat flux levels ($> 0.25 \text{ Btu/in.}^2\text{-sec}$) to permit greater test durations and/or slower starts to minimize heater thermal shock.

REFERENCE

1. R-9645, Final Report - Solar Tower Receiver Study, MDAC Contract 774023-H, Rocketdyne Division, Rockwell International, Canoga Park, California, January 1975.

APPENDIX A

THERMAL ANALYSIS PROGRAM FOR A ONCE-THROUGH STEAM GENERATOR

The thermal analysis of the single tube and multi-tube test sections was accomplished using an existing computer program developed for solar tower receiver design. The basic cooling correlations utilized in the various flow regimes are presented in Table A-1.

The pressure correction factor, $f(P)$, was derived from experimental data taken under a company-sponsored program using an electrically heated stainless steel tube. Typical measured temperature profiles from that experimental effort are depicted in Fig. A-1 for various pressure levels. The increase in temperature with decreasing pressure in the film boiling regime is apparent. The data were utilized to relate the effective minimum film coefficient to pressure and coolant mass flux as shown in Fig. A-2. The pressure correction factor was developed from these data in terms of a saturated steam cooling condition.

The location of the minimum film coefficient point was determined in terms of local quality and heat flux level as shown in Fig. A-3. A similar plot was obtained for determination of initiation of film boiling. A polynomial curve fit of the transition, minimum, and single phase end points completes the model.

A comparison of the computer predicted and measured tube temperature profiles for an electrically heated tube test is shown in Fig. A-4.

TABLE A-1
HEAT TRANSFER CORRELATIONS

SINGLE PHASE (SUBCOOLED LIQUID OR SUPERHEATED VAPOR):

$$h_c = 0.023 \frac{G_c C_p}{R_e^{0.2} P_r^{2/3}}$$

DITTUS-BOELTER
EQUATION

LOW QUALITY REGION (NUCLEATE BOILING):

$$h_c = 0.0196 \left(\frac{Q}{A} \right)^{3/4} e^{P/900}$$

MODIFIED JENS & LOTTES
EQUATION

FILM BOILING REGION:

$$\text{MINIMUM } h_c = 0.023 \frac{G_c C_p}{R_e^{0.2} P_r^{2/3}} \cdot f(P)$$

SATURATED STEAM
PROPERTIES

WHERE $f(P)$ = PRESSURE CORRECTION FACTOR FROM EXPERIMENTAL DATA

POLYNOMIAL CURVE FIT OF TRANSITION, MINIMUM AND END POINTS AS A
FUNCTION OF HEAT FLUX AND QUALITY FROM EXPERIMENTAL DATA

COOLANT PRESSURE EFFECT ON TUBE WALL TEMPERATURE

Figure A-1

COOLANT PRESSURE AND MASS FLUX EFFECT ON MINIMUM HEAT TRANSFER COEFFICIENT

Figure A-2

HEAT FLUX AND QUALITY RELATION AT
MINIMUM HEAT TRANSFER CONDITION

Figure A-3

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A-5

COMPARISON OF PREDICTED AND MEASURED OUTER WALL TEMPERATURE PROFILE

Figure A-4

APPENDIX B

TUBE STRESS ANALYSIS

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REQUIRED: GENERATE A METHOD TO
 DETERMINE A ΔT_{max} FOR
 VARIOUS HOT SURFACE AND
 COLD WALL TEMPERATURES
 WITH THE CONSTRAINT OF THE
 MAXIMUM EFFECTIVE STRAIN,
 $\epsilon_{eff} \leq .44\%$, BASED ON THE
 DESIGN LOW CYCLE FATIGUE PLOT FOR
 INCOLOY 800, THE MAXIMUM
 ALLOWABLE STRAIN RANGE
 FOR 10,000 CYCLES IS
 $\epsilon_{eff} = .44\%$ (SEE FIGURE B1)

NOTE: THE DESIGN CURVE WAS
 DETERMINED IN ACCORDANCE
 WITH ASME BOILER CODE PRACTICES.

Figure B-1

FATIGUE DATA & DESIGN CURVE - RECEIVER TUBE

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ANALYSIS : ASSUMPTIONS

- 1) ASSUME LINEAR TEMPERATURE DISTRIBUTION THROUGH THE HOT WALL.
- 2) ASSUME LINEAR TEMPERATURE DISTRIBUTION AROUND CIRCUMFERENCE OF TUBE.
- 3) ASSUME PLANE STRAIN CONDITION FOR TUBE CROSS SECTION.

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ANALYSIS: METHOD

A.) "BASIC" EQUATION

$$\epsilon_{\text{apparent}} = \epsilon_{\Delta T} - \epsilon_{\text{MECHANICAL UNRESTRAINED GROWTH}}$$

B) AXIAL STRAIN : ϵ_{axial}

1) SECTION TUBE CROSS SECTION INTO N REGIONS AND CALCULATE REGION'S AVERAGE TEMPERATURE

2) ASSUME VALUE FOR $\epsilon_{\text{apparent}}$

3) CALCULATE $\epsilon_{\text{mechanical}}$ USING $\epsilon_{\text{apparent}}$ OF STEP 2) AND $\epsilon_{\Delta T}$ FOR EACH OF THE N AREAS

NOTE: $\epsilon_{\Delta T} = \alpha_{T_i} (T_i - 70^\circ)$

T_i = AVE TEMP OF AREA i

α_{T_i} = COEF OF THERMAL EXPANSION AT TEMP T_i

4) CALCULATE $\epsilon_{\text{MECHANICAL}}$ USING $\epsilon_{\text{mechanical}}$ OF STEP 3) AND APPROPRIATE STRESS-STRAIN CURVES (ASME SECT III FOR INCOLOY 800)

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5) VERIFY

$$\sum_{i=1}^N F_i = 0$$

WHERE $F_i = \sigma_{i, \text{MECH}} A_i$

$A_i = \text{AREA}_i$

6) CALCULATE PEAK $\epsilon_{\text{AXIAL MECHANICAL}}$

$$\epsilon_{\text{AXIAL MECHANICAL}} = \epsilon_{\Delta T \text{ PEAK}} - \epsilon_{\text{apparent (STEP 2)}}$$

WHERE $\epsilon_{\Delta T \text{ PEAK}} = \alpha_H (T_H' - 70^\circ)$

7) HOOP STRAIN : ϵ_{HOOP}

$R = \text{AVE RADIUS OF TUBE}$

8) CALCULATE δ

$$\delta \approx R \left[\alpha_{T_{\text{HOT WALL}}} (T_{\text{HOT WALL}} - 70^\circ) - \alpha_{T_{\text{COLD WALL}}} (T_{\text{COLD WALL}} - 70^\circ) \right]$$

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REF
SPACE DIV.
STRUCT.
MANUALS.

2) DETERMINE LOAD "P" THAT PRODUCES S.

3) DETERMINE MOMENT AT HOT CROWN OF TUBE

4) DETERMINE ELASTIC $\sigma_{MECH \textcircled{1}}$

$$\sigma_{MECH} = \frac{Mc}{I}$$

5) DETERMINE ELASTIC $\epsilon_{MECH \textcircled{1} \text{ HOOP}}$

$$\epsilon_{MECH \textcircled{1} \text{ HOOP}} = \frac{\sigma_{MECH \textcircled{1}}}{E}$$

6) DETERMINE $\epsilon_{MECH \textcircled{2} \text{ HOOP}}$ DUE ΔT ACROSS HOT WALL

$$\epsilon_{MECH \textcircled{2} \text{ HOOP}} = \frac{[\alpha_{TH'} (T_H' - 70^\circ) - \alpha_{TH} (T_H - 70^\circ)]}{2}$$

7) SUM: $\epsilon_{MECH \textcircled{1} \text{ HOOP}} + \epsilon_{MECH \textcircled{2} \text{ HOOP}} = \epsilon_{HOOP}$

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D) EFFECTIVE STRAIN : ϵ_{eff}

USING EFFECTIVE STRAIN CALCULATION
FOR AN ELEMENT IN THE PLASTIC
STATE UNDER BIAXIAL LOADING:

$$\epsilon_{eff} = \frac{2}{\sqrt{3}} \left[\epsilon_{axial}^2 + \epsilon_{ax} \epsilon_{hoop} + \epsilon_{hoop}^2 \right]^{1/2}$$

solve for ϵ_{eff} .

NOTE: STRAINS DUE TO INTERNAL
PRESSURE LOADING WAS NOT CONSIDERED
SINCE THEY WERE SMALL RELATIVE TO
THE THERMAL EFFECTS. BECAUSE THE
PRESSURE STRAINS ARE TENSILE, THEIR
OMISSION MAKES THE ANALYSIS CONSERVATIVE

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ANALYSIS (CONTINUED) :

RESULTS : AN ITERATIVE METHOD WAS SET UP TO CALCULATE THE VARIOUS TEMPERATURES TO PRODUCE AN EFFECTIVE STRAIN RANGE. ($\Delta \epsilon_{\text{EFF}} = \epsilon_{\text{EFF}}$) OF .44% STRAIN (FIG. 1)

1) ASSUME $T_{\text{HOT WALL}}$, $T_{\text{COOL WALL}}$, AND $\Delta T'$

2) CALCULATE ϵ_{EFF}

3) IF $\epsilon_{\text{EFF}} > .44\%$ REDUCE $T_{\text{HOT WALL}}$

IF $\epsilon_{\text{EFF}} < .44\%$ INCREASE $T_{\text{HOT WALL}}$

WITH THE METHOD DESCRIBED ABOVE, $T_{\text{HOT WALL}}$, $T_{\text{COOL WALL}}$, AND $\Delta T'$ CAN BE DETERMINE SUCH THAT LOW CYCLE FATIGUE FAILURE IS PREVENTED

IN FIGURE 19, LINES OF CONSTANT $\Delta T'$ ARE PLOTTED FOR THE MAXIMUM HOT WALL TEMPERATURES VS. MINIMUM COOL WALL TEMPERATURES

A LINE OF ALLOWABLE HOOP STRESS IS PLOTTED CALCULATED FROM INTERNAL PRESSURE AND BASED ON ALLOWABLE STRESSES OF CASE 1325-7, ASME BOILER & PRESSURE VESSEL CODE