DOE/NASA CONTRACTOR REPORT

DOE/NASA CR-150569

AIR-LIQUID SOLAR COLLECTOR FOR SOLAR HEATING, COMBINED HEATING AND COOLING, AND HOT WATER SUBSYSTEMS

Prepared by

Owens Illinois
P. O. Box 1035
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Under Contract NAS8-32259 with

National Aeronautics and Space Administration
George C. Marshall Space Flight Center, Alabama 35812

for the U. S. Department of Energy

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Solar Energy

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This work was accomplished under the technical management of Mr. John Caudle, Marshall Space Flight Center, Alabama.

16. ABSTRACT

This report contains a collection of quarterly reports prepared by Owens Illinois in the development of an air-liquid solar collector for solar heating, combined heating and cooling, and/or hot water subsystems.

These reports have been reformatted, pages renumbered, and cost information removed.

17. KEY WORDS	18. DISTRIBUTION STATEMENT Unclassified-Unlimited William A. Brooksbank, Jr. WILLIAM A. BROOKSBANK, JR. Manager, Solar Heating & Cooling Proj Ofc

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TABLE OF CONTENTS

First Quarterly Report
Contract No. NAS8-32259
November 1, 1976 through February 1, 1977

Summary. The principal activities during this reporting period consisted of the installation and layout design of the air collector system for commercial applications, completion of the Preliminary Design Review, detail design effort, preparation of the Verification Test Plan and the detail test procedures necessary to verify the subsystem to the Interim Performance Criteria, preparation of an updated Subsystem Performance Specification and the performance testing of a prototype model of a two manifold, one hundred and forty-four tube air collector array.

Schedules. The schedule, established to meet contract requirements, is attached as Figure 1. The Preliminary Design Review was completed on January 5, 1977, as scheduled by the Technical Program Manager, MSFC. The specific schedules for the Quarterly Review and the Prototype Design Review have not been designated, as yet, by the Technical Program Manager.

Technical Performance.

A. General description of work accomplished during reporting period.

1. Installation and Layout Design.

The installation and layout drawings of the air collector, Model SEC-601, were compiled and formed the basis of the Preliminary Design Review on January 5, 1977.

2. Preliminary Design Review.

The Preliminary Design Review of the air collector, Model SEC-601, was conducted on January 5, 1977. Nine RID's were generated and responded to on January 13, 1977. Further contractor action is required on five of the nine RID's.

3. Detail drawings are proceeding on schedule towards vendor coordination and material selections during February, 1977. Simple test fixtures are being fabricated for the verification of the design approach for seals.

4. The Verification Test Plan was submitted to MSFC on January 18 for review and comment.

5. A draft of the detail test procedures to be used in the evaluation of the Model SEC-601 air collector relative to the requirements of the Interim Performance Criteria. The verification and detail test plans have been submitted to Smith, Hinchman and Grylls for review and comment as the designated Independent Test Agency.

6. An updated Subsystem Performance Specification was submitted to MSFC on December 9, 1976 for review and comment.

7. Air collector performance and operational testing.

The testing of the collector since the installation was completed in December 1976 has been devoted to the investigation of general operating characteristics. Air flow rate vs. system pressure drop, collector thermal gain over a range of air flow rates and at ambient air temperature, the effect of snow cover on collector thermal performance, air flow distribution between tube pairs and duct thermal gradients and impact on air flow distribution between manifolds were among the characteristics investigated. Investigation of one of the most critical factors was not initiated until mid-January 1977. Air flow leakage rates were determined to be of the order of 20% to 30% of the total system flow. Corrective action was taken and the leakage flow was reduced to the order of 1%. However, as a result all experimental data accumulated prior to the correction of the leakage problem can be used only for trend information. Testing following the corrective action was initiated on January 27, 1977. Thus, sufficient data has not been accumulated during this reporting period to allow the development of quantitative performance data.

Certain qualitative test results may be indicated, however. The air flow distribution pattern from tube pair to tube pair without regard to location in the manifold and from manifold to manifold appears to be very satisfactory. As reported in the December progress report, the air distribution appears to be within 25% of the mean value which is significantly less than the 2:1 flow variation determined to be acceptable on the basis of computer studies by Grunes.(1)

Snow cover of the collector reduces the thermal performance of the system but good thermal gain is still realized. Heavy snowfall caused the space between the collector tube elements and the backing screen to be completely snow filled and the tube elements to be covered by a thin (approximately one inch) layer of snow. This caused a decrease in the level of light enhancement from reflections from the backing screen. On a qualitative basis, the reduction in thermal performance was only in the order of 25%. Wind action swept the snow from the cover tubes and initiated some removal of the snow between the tubes and the backing screen. Adjacent liquid collector test systems were under test during this same period. The spacing between the tubes and the backing screen was greater than that used in the air collector installation. In this case, the snow did not fill the volume between the tubes and the backing screen. The loss of thermal performance was not significant within the normal scatter of the test data. It may be desirable to increase the spacing between the collector tubes and backing screen for installations in the snow belt region to enhance the removal of snow in this volume due to natural wind action.

Very cold ambient temperatures and high winds causing wind chill factors of -40°F to -65°F do not cause a significant decrease in the thermal performance of the collector. The qualitative difference in performance was within the normal scatter of the test data. It should be noted that the degree of scatter of test points should be reduced with the automatic recording of data in five minute increments and data reduction using the computer program.

(1) "Vitalization and Operation Characteristics of Evacuated Tubular Collectors Using Air as the Working Fluid" by Howard Grunes, Mechanical Engineering Department, University of Wisconsin, Madison, Wisconsin.

Based on a single day of operation under no flow conditions and reasonably good levels of insolation, collector tube stagnation temperatures reached in excess of 500°F while temperatures in the manifold stabilized at temperatures in the order of 150°F or less. Thus, severe temperature levels during extended periods of no air flow are not expected to represent a critical materials problem. On start up of air flow, however, manifold temperatures will rise towards the stagnation temperatures reached by the collector tubes. Qualitative tests indicate that these temperature excursions can effect manifold temperature levels in excess of 300°F. Operating procedures might be required in the appropriate manuals to exclude system start up after extended periods of exposure to high levels of insolation under no flow conditions.

B. Forecast of activities to complete tasks.

The completion of the detail design of the air collector Model SEC-601 and the Prototype Design Review are considered to be high priority items. The design review meeting should be scheduled by late February or early March to avoid delays in the fabrication of test hardware and the first article.

To date, only the testing of the air collector, installed under the companion ERDA contract, has been initiated towards completion of the Verification Test Plan. Copies of the Verification Test Plan and a preliminary draft of the detail test procedures have been forwarded to the proposed Independent Test Agency, Smith, Hinchman and Grylls, for review, comment and estimate of cost. The formalization of any agreement with S,H & G and the initiation of detail tests will be delayed until approval to proceed is received from MSFC. The involvement of MSFC in the O-I liquid evacuated tubular collector has increased greatly since the submission of the Verification Test Plan for the air collector. Therefore, a preferred approach at this time would be for NASA-MSFC to act as the evaluation agency for the air system.

Other actions and activities required to complete the tasks are identified in the detail schedules contained in the Monthly Progress Reports and the Verification Test Plan.

C. Identification of major problem areas or difficulties.

No unforeseen problem areas or difficulties have been encountered to date in the course of design and testing of the air system.

D. Data submittals.

1. Test data to evaluate air flow distribution is attached. The information is for only two days of testing and is submitted to indicate the method of testing and analysis rather than as conclusive quantitative data.

2. The thermal performance data for two days of testing is summarized in the attachment. Hand calculations were used to derive the qualitative information. The principal item of interest is the useful energy collected in terms of BTU/ft.^{2} day versus the total insolation in the plane of the collector for the same operating period.

3. A curve of system static pressure drop vs. air flow rate is attached. At the design flow rate of 4 SCFM per tube pair, the pressure drop is .48 inches water gage. Using the expression:

$$HP_A = 157.5 \times 10^{-6} pQ \quad (1)$$

p = pressure, in. water gage
Q = volume flow CFM
$HP_A = 157.5 \times 10^{-6} \times .48 \times 288 = .022$ H.P.

(1) Marks' Mechanical Engineers' Handbook, Sixth Edition, page. 14-67.

Data for Jan. 27, 1977

Temperature Distribution by Tube Pair

Tube Pair	ΔT	Weighted Tube Perf. 11:30	Corrected ΔT	ΔT	Weighted Tube Perf. 1:30	Corrected ΔT	ΔT	Weighted Tube Perf.	Corrected ΔT
1A-2A	59	.99	59.6	70	.99	70.7	68	.99	68.69
3A-4A	31	1	31	36	1	36	30	1	30
5A-6A	66	1.036	63.46	71	1.036	68.53	60	1.036	57.92
7A-8A	60	1.057	56.76	65	1.057	61.49	55	1.057	52.03
9A-10A	64	1.052	60.83	69	1.052	65.59	59	1.052	56.08
1B-2B	44	.976	45.08	45	.976	46.11	38	.976	38.93
3B-4B	51	.984	51.83	60	.984	60.98	58	.984	58.94
5B-6B	46	.96	47.92	52	.96	54.17	44	.96	45.83
7B-8B	45	.945	47.62	50	.945	52.91	39	.945	41.27
9B-10B	55	.945	58.2	54	.945	57.14	49	.945	51.85
1C-2C	60	.968	61.98	65	.968	67.15	57	.968	58.88
3C-4C	48	1	48	54	1	54	49	1	49
5C-6C	50	.9925	50.38	56	.9925	56.42	51	.9925	51.39
7C-8C	45	.975	46.15	45	.975	46.15	43	.975	44.10
Upper An.	96			100			85		
Lower An.	102			111			97		
Upper Man ΔT	70			66			55		
Lower Man ΔT	74			65			56		
Total ΔT	67			70			58		
$S_{\perp}$	292 BTU/ft.^{2} hr.			304 BTU/ft.^{2} hr.			220 BTU/ft.^{2} hr.		

Data for Jan. 27, 1977

	$\overline{\Delta T}$	11:30 $\Delta T/T.P.$	% Dev.	$\overline{\Delta T}$	1:30 $\Delta T/T.P.$	% Dev.	$\overline{\Delta T}$	3:30 $\Delta T/T.P.$	% Dev.
Top Manifold	52.79			58.07			50.61		
1A-2A		59.6	113%		70.7	122		68.69	136
3A-4A		31	58.7		36	62		30	59
5A-6A		63.46	120		68.53	118		57.92	114
7A-8A		56.76	108		61.49	106		52.03	103
9A-10A		60.83	115		65.59	113		56.08	111
1B-2B		45.08	85		46.11	79		38.93	77
		317.73			348.42			303.65	
Bottom Manifold	51.51			56.12			50.16		
3B-4B		51.88	101		60.98	109		58.94	118
5B-6B		47.92	93		54.17	97		45.83	91
7B-8B		47.62	92		52.91	94		41.27	82
9B-10B		58.2	113		57.14	102		51.85	103
1C-2C		61.98	120		67.15	120		58.88	117
3C-4C		48	93		54	96		49	98
5C-6C		50.38	98		56.42	101		51.39	102
7C-8C		46.15	90		46.15	82		44.10	
		412.08			448.92			401.26	
$S_{\perp}$	292			304			220		
Upper ΔT	70			66			55		
Lower ΔT	74			65			56		
Total ΔT	67			70			58		

Data from 2:00 P.M. - Jan. 27, 1977
Lower-East Quadrant Air Flow Distribution

T.P./Temp.	In	Out	ΔT	Tube Pair Rating	Weighted T/R Rating	Weighted ΔT	%	Dev.
	87	30	57	1023	.971	58.7	.96	-.04
	90	27	63	1035	.982	64.15	1.05	+.05
	90	27	63	1035	.982	64.15	1.05	+.05
	91	27	64	1070	1.015	63.05	1.03	+.03
	92	27	65	--	--	--	--	--
	94	31	63	1022	.97	64.95	1.06	+.06
	84	32	52	1051	1.0	52	85	-.15
	100	30	70	1090	1.034	67.7	1.11	+.11
	98	32	66	1061	1.007	65.54	1.07	+.07
	95	33	62	1074	1.019	60.84	1.00	--
	106	34	72	1058	1.004	71.71	1.17	+.17
	91	38	53	1064	1.009	52.53	.86	-.14
	105	35	70	1078	1.023	68.43	1.12	+.12
	86	38	48	1084	1.028	46.69	.76	-.24
	95	34	61	1077	1.022	59.69	.98	-.02
	96	34	62	1004	.953	65.06	1.07	+.07
	97	33	64	1054	1.0	64	1.05	+.05
	86	38	48	1042	.989	48.55	.80	-.20
				17,922		1037.74		
			÷ 17 =	1054	÷ 17 =	61.04		

The distribution of air flow between manifolds and individual tube pairs is implied from relative values of temperature rise per tube pair modified by individual tube ratings of stagnation temperature. The lower manifold east quadrant has been instrumented with thermocouples for every tube pair. Thus, temperature rise data is available for one quadrant of 18 tube pairs in the lower manifold and for 3 tube pairs in each of the remaining three quadrants of the collector array. An analysis of the test data for January 27, 1977, indicates no trends for air flow maldistribution relative to tube pair location in the manifold or from manifold to manifold. Further testing is anticipated to investigate the reasons for specific tube pairs being well out of the expected error band. The work sheets are attached for information.

Data for 2/1/77

Temperature Distribution by Tube Pair

Tube Pair	ΔT	Weighted Tube Perf. 11:00	Corrected ΔT Dev.	ΔT	Weighted Tube Perf. 11:15	Corrected ΔT Dev.	ΔT	Weighted Tube Perf. 3:00	Corrected ΔT Dev.
1A-2A	44	.99	44 +.20	55		56 + .30	75		76 +.37
3A-4A	22	1.04	21 -.43	27		26 -.39	36		35 -.37
5A-6A	46	1.04	44 +.20	54		52 +.21	69		66 +.19
7A-8A	42	1.05	40 +.09	49		47 -.09	62		59 +.06
9A-10A	45	1.03	44 +.20	53		52 +.21	67		65 +.17
1B-2B	29	.976	30 -.18	30		31 -.28	37		38 -.32
3B-4B	36	.984	37 -	46		47 +.09	62		63 +.14
5B-6B	32	.98	33 -.10	35		36 -.16	47		48 -.14
7B-8B	30	1.01	30 -.18	31		31 -.18	47		47 -.15
9B-10B	37	.957	39 +.06	45		47 +.09	56		59 +.06
1C-2C	43	.956	45 +.23	50		52 +.21	63		66 +.19
3C-4C	35	.983	36 -.02	41		42 -.02	54		55 -.01
5C-6C	35	.995	35 -.04	41		41 -.04	54		54 -.03
7C-8C	34	.975	35 -.04	40		41 -.04	45		46 -.17
		$\overline{\Delta T}$ = 36.64				42.93			55.5

East quadrant - lower manifold - 1:30 p.m.

	ΔT	Corrected ΔT	Dev.	ΔT	Corrected ΔT	Dev.	ΔT	Corrected ΔT	Dev.
1	56	58	+.09	58	56	+.05	50	49	-.08
2	55	56	+.05	53	53	-.01	53	57	+.07
3	58	59	+.11	62	61	+.14	45	45	-.16
4	56	55	+.03	45	45	-.16			
5.	66	--	--	58	58	+.09	$\overline{\Delta T}$ =	53.3	
6	52	54	+.01	42	41	-.23			
7	59	59	+.11	48	47	-.12			

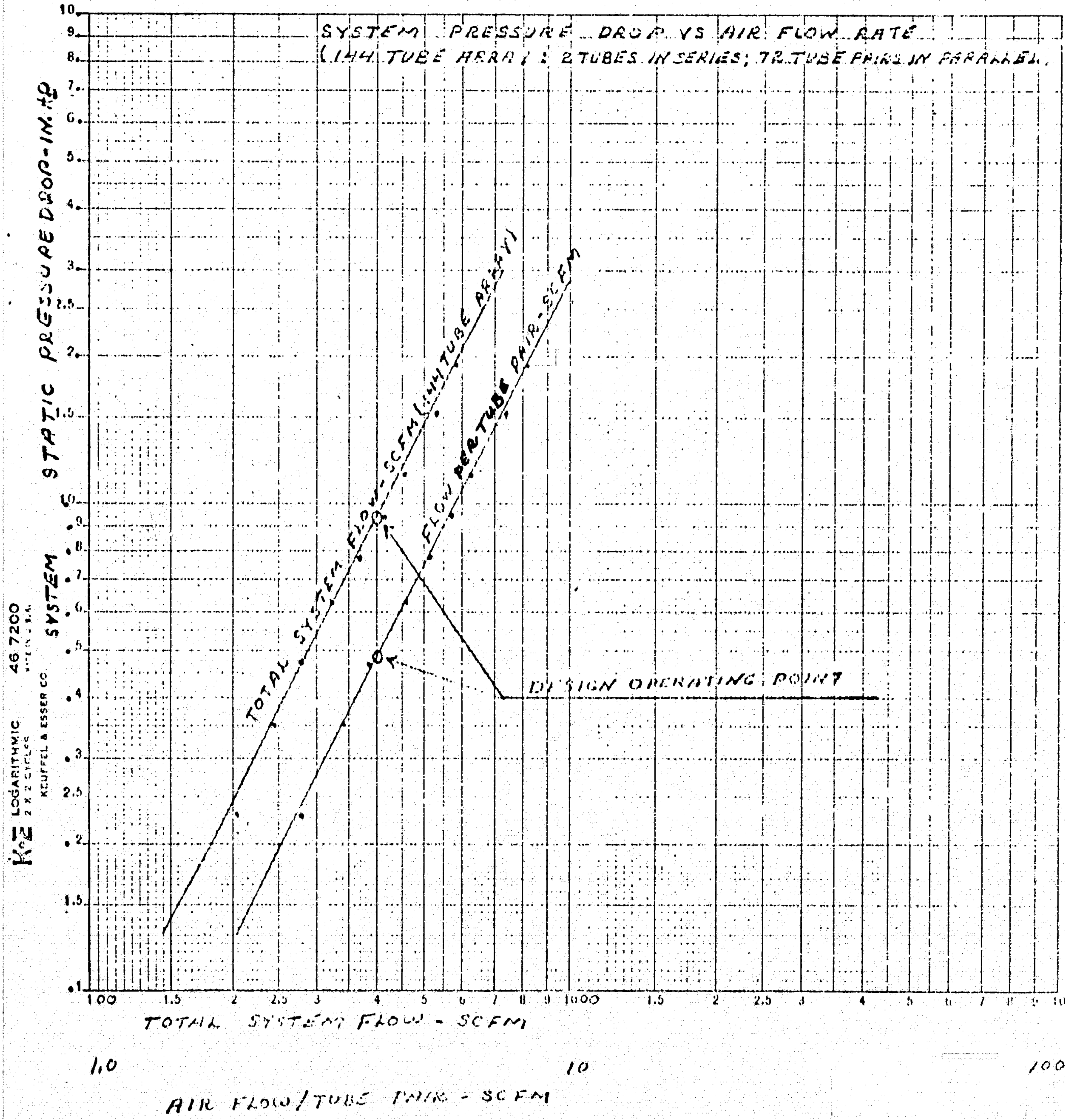
Summary of Thermal Performance Data

January 31, 1977

Time	9:30	10:30	11:30	12:30	13:30	14:30	15:30	16:15	$\frac{BTU}{ft.^2\ Day}$
Bottom Array									
T out °F	58	64	64	86	96	102	92	73	
ΔT°F	38	43	40	58	65	69	58	40	
$\overline{qu}$ $\frac{BTU}{hr.\ ft.^2}$	74	86	80	118	132	140	122	81	833
$\overline{S_\perp}$ $\frac{BTU}{hr.\ ft.^2}$	152	168	240	240	260	264	200	140	1664
Top Array									
T out °F	60	64	63	85	95	102	92	73	
ΔT °F	40	43	39	57	64	67	54	38	
$\overline{qu}$ $\frac{BTU}{hr.\ ft.^2}$	78	86	79	118	130	136	113	77	817
$\overline{S_\perp}$ $\frac{BTU}{hr.\ ft.^2}$	152	168	240	240	260	264	200	140	1664
T amb. °F	6	8	11	13	16	19	20	19	

February 1, 1977

Bottom Array	9:00	10:00	11:00	12:00	13:00	14:00	15:00	16:00	
T out °F	44	77	78	85	83	105	102	97	
ΔT °F	17	45	43	48	46	64	61	52	
$\overline{qu}$ $\frac{BTU}{hr.\ ft.^2}$	32	84	80	90	85	120	114	97	702
$\overline{S_\perp}$ $\frac{BTU}{hr.\ ft.^2}$	78	145	147	155	160	277	202	199	1363
Top Array									
T out °F	44	75	77	83	82	105	101	97	
ΔT °F	14	45	43	49	47	67	62	57	
$\overline{qu}$ $\frac{BTU}{hr.\ ft.^2}$	26	84	80	92	87	125	116	107	717
$\overline{S_\perp}$ $\frac{BTU}{hr.\ ft.^2}$	78	145	147	155	160	277	202	199	1363
T amb. °F	16	21	21	22	24	27	27	26	



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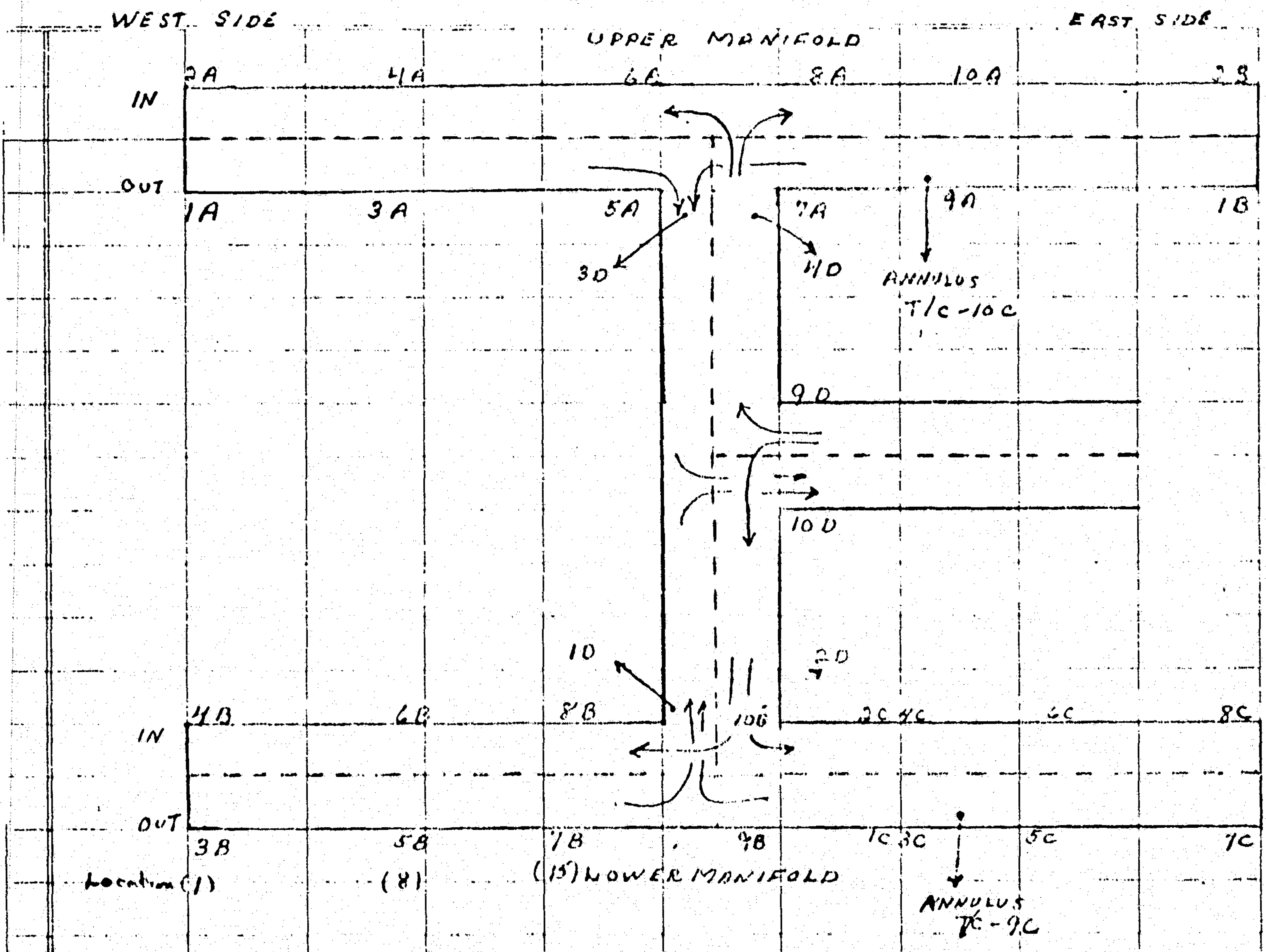

Note: Four switching units of ten elements each are used for the read out of temperatures at various locations in the test installation. Thermocouples 1A through 8C are located within one inch of the inlet and exit air flow paths to and from collector tube pairs. Thermocouples 9C and 10C are located approximately two feet up in the annulus between the absorber and feeder tubes. Thermocouples 1D and 2D are located in the transition section between the air ducts and the lower manifold. Thermocouples 3D and 4D are similarly located in the upper feeder duct/ manifold. Thermocouples 9D and 10D are located in the main header ducts connected to the distribution ducts.

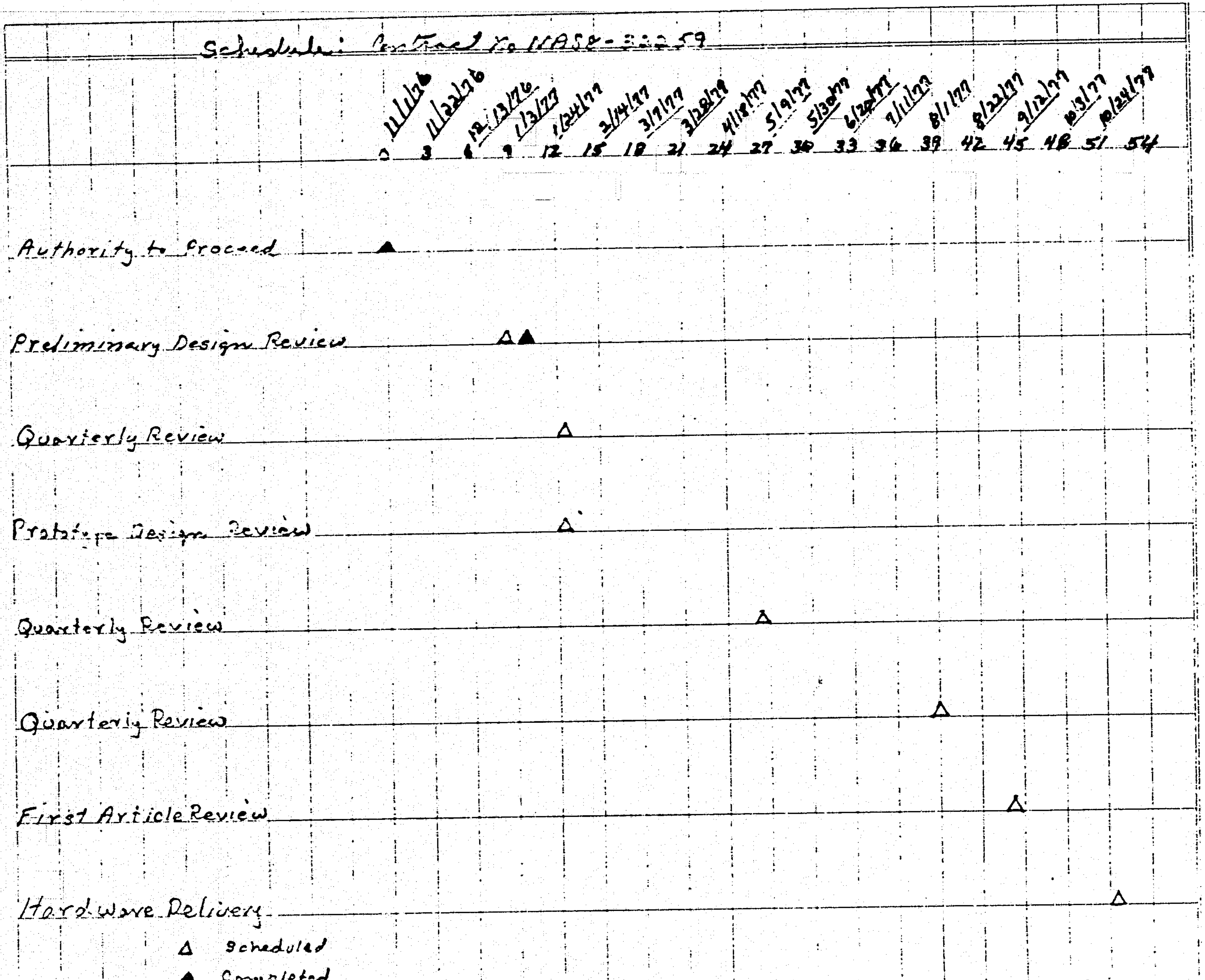

Figure 1

A-15

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Summary. The principal activities during this reporting period consisted of completion of the detail design and material specification for the base-line collector Model SEC-601, completion of the Prototype Design Review, completion and approval of the Verification Test Plan, submission and approval of the Detail Test Procedures for the Verification of the Collector to the Interim Performance Criteria, and completion of the performance testing and evaluation of the 144 ERDA air collector array.

Technical Performance

A. General description of work accomplished during the reporting period.

1. Installation and layout design. The installation layout and detail design and material specification have been completed and formed the basis of the Prototype Design Review on March 15, 1977.

2. Preliminary Design Review. All RID's, generated at the Preliminary Design Review, have been formally cleared and closed.

3. The Prototype Design Review was conducted on March 15, 1977. Additional information and documentation were requested relative to detail material specifications and the technical basis used in establishing the design criteria for the Model SEC-601 base-line collector. The material specifications were upgraded. An engineering design specification relating the design of the Model SEC-601 to the 144 tube ERDA test array was prepared. The documentation was submitted, forming the basis of a telephonic design review conference on May 3, 1977. Approval of the base-line design of the Model SEC-601 collector was indicated and formally issued in mid-May 1977. A set of reproducible drawings and specifications with authorizing signatures for release for fabrication are being prepared for forwarding to MSFC. The information requested for all RID's generated at the Prototype Design Review was supplied and the RID's have all been formally closed.

4. Formal approval of the Verification Test and Plan was received and acceptance of the Detail Test Procedure was acknowledged. A subcontract to Smith, Hinchman and Grylls to act as the Independent Certification Agency was formalized. Several technical meetings have been held with William Louis and David Miller (the principals of SH&G) for the certifications activity. Major emphasis has been on the test procedures and data analysis of the 144 tube ERDA test array. The detail test procedure to be used in satisfying the requirements of the Interim Performance Criteria was also reviewed and an agreeable monitoring process agreed upon. Physical testing of the Model SEC-601 collector and components will be delayed from schedule pending fabrication of units in accordance with the approved base-line collector design.

5. Air collector performance and operational testing. A paper prepared for presentation at the Orlando, Florida 1977 meeting of ISES summarizes the data and analysis of the 144 tube ERDA test array. A preprint of the paper is attached as documentation of the testing and analysis performed to date.

6. Installation, operation and maintenance manual. A preliminary draft of the document was forwarded to MSFC for informal review and comment. A telephone conference call on May 16, 1977 with MSFC personnel provided many valuable inputs. The document will be revised and updated for formal submission during the next reporting period.

B. Forecast of activities to complete tasks.

1. The fabrication of units of the Model SEC-601 baseline collector will receive major emphasis. The manufacturing group will be responsible for this phase of the activity supported by the design engineer. Fabrication forecast schedules are being prepared to monitor progress. The development of a Quality Assurance Plan will be a part of the fabrication process to ensure all components meet design specification requirements.

2. Test fixtures for the physical evaluation of components are in design and scheduled for early fabrication awaiting delivery of the necessary collector components. Negotiations are underway with Desert Sunshine Exposure Testing, Inc. and NBS to undertake their designated test tasks.

3. The 144 tube test array will be modified to incorporate an air-liquid heat exchanger and simulated load elements to complete this phase of the scheduled test program.

C. Identification of major problem areas or difficulties.

1. No technical problems adversely affecting thermal performance or qualification of the Model SEC-601 base-line collector are foreseen at this time.

2. The completion of the contract on schedule is now considered to be in jeopardy. The earliest date for fabrication of components is estimated as late August. The detail fabrication plan, based on firm vendor quotations, will identify the date more exactly. A three to four month slippage in the completion of the long-term exposure testing should be anticipated.

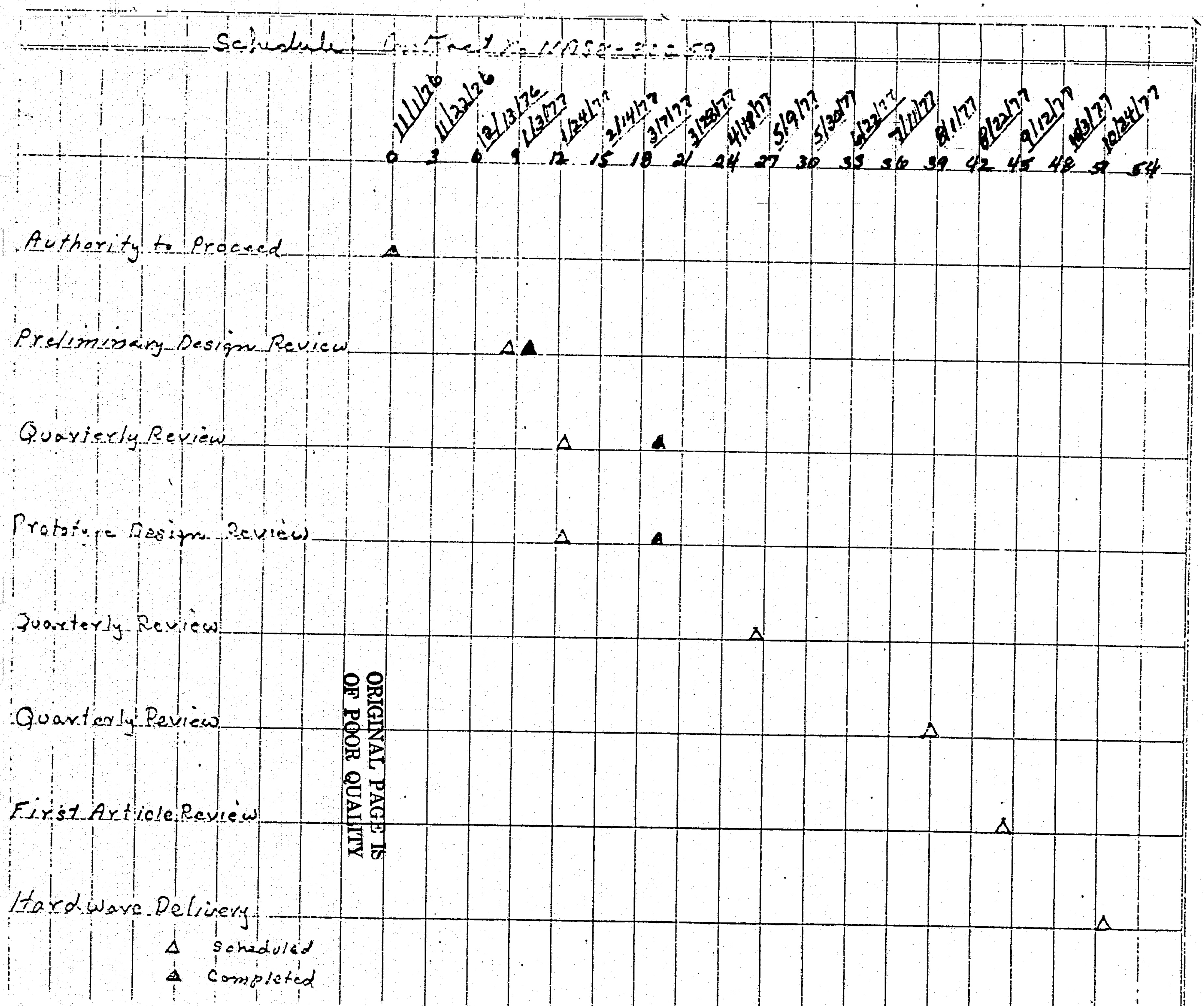
D. Data submittals.

1. The 144 tube collector performance analysis is attached.

2. Other documentation submitted during the reporting period include:

Reference	Description
259-18	Additional comments on RID's 70I-7 and 70I-8
259-19	Prototype design review data
259-20	Technical directive No. 2

Reference	Description
259-21	February status report
259-22	Technical directive No. 3
259-23	Draft of SH&G subcontract
259-24	Partial response to RID's 70I-16 and 70I-17
259-25	Proposed base-line design
259-26	Prototype design review RID's
259-27	March monthly status report
259-28	Drawing SK-3558 and draft installation, operation and maintenance manual
259-29	Data in response to 4/26/77 conference call
259-30	Update of interim performance criteria summary


Schedule
11/1/76
11/22/76
12/13/76
1/3/77
1/24/77
2/14/77
3/7/77
3/28/77
4/18/77
5/9/77
5/30/77
6/22/77
7/11/77
8/1/77
8/22/77
9/12/77
10/3/77
10/24/77
0 3 6 9 12 15 18 21 24 27 30 33 36 39 42 45 48 51 54
Authority to Proceed
Preliminary Design Review
Quarterly Review
Prototype Design Review
Quarterly Review
Quarterly Review
First Article Review
Hardware Delivery
Scheduled
Completed

-1 DEVELOPMENT PLAN

FIGU   A.

4.0 CONTRACTOR TASKS

4.1. DEVELOPMENT PLAN
TASK 1 - DESIGN 72 TUBE MODULE
DESIGN REVIEW
TASK 2 - BUILD 4 MODULE ARRAY
INSTALLATION DESIGN
INSTRUMENTATION SPECIFICATION

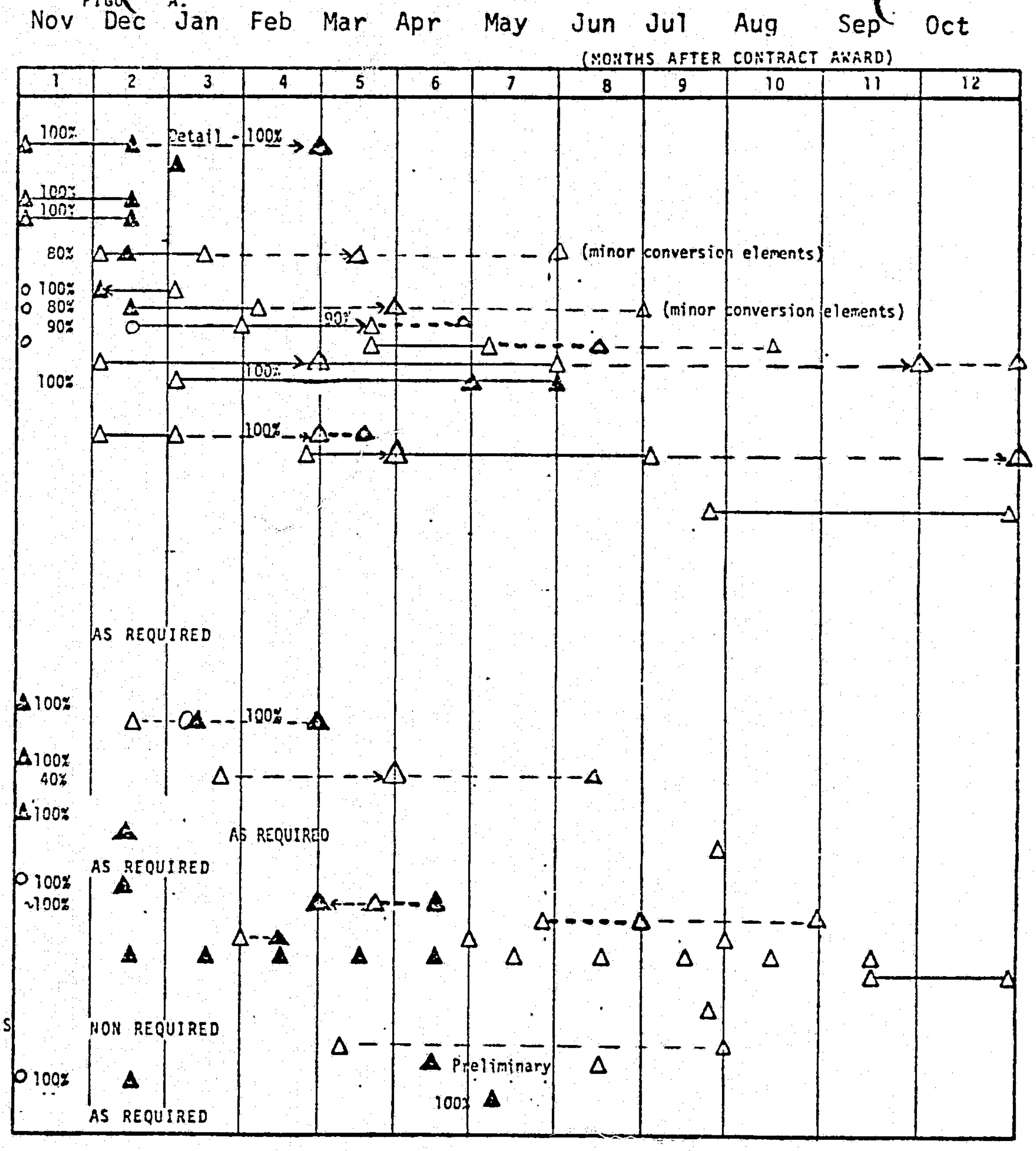
PROCUREMENT
INSTALLATION
SITE PREPARATION
INSTALLATION
TASK 3 - THERMAL PERFORMANCE TESTS
TASK 4 - OPERATION TESTS
4.1.2. INTERIM PERFORMANCE CRITERIA TESTS
4.1.3. PACKAGING

4.2. CERTIFICATION
REVIEW OF TEST PROCEDURES
REVIEW OF TEST DATA, ANALYSIS & DOCUMENTS

4.3. MANUFACTURING
4.3.1. FABRICATION

4.4. DOCUMENTATION EVALUATION DATA

4.5. MANAGEMENT AND DOCUMENTATION
505-2 VERIFICATION PLAN
PRELIMINARY
COMPLETE
505-3 QUALITY ASSURANCE PLAN
PRELIMINARY
FINAL
505-4 SYSTEM PERFORMANCE SPECIFICATION
PRELIMINARY
UPDATE
505-5 SOURCE CONTROL DRAWINGS & SPECIFICATIONS
505-6 CHANGE PROPOSAL
505-7 PRELIMINARY DESIGN REVIEW DATA
505-8 PROTOTYPE DESIGN REVIEW DATA
505-9 FIRST ARTICLE REVIEW DATA
505-10 QUARTERLY REPORT
505-11 MONTHLY STATUS REPORT
505-12 ACCEPTANCE DATA PACKAGE
505-13 QUALIFICATION AND ACCEPTANCE TEST PROCEDURES
505-14 QUALIFICATION/ANALYSIS REPORT
505-15 SPECIAL HANDLING/INSTALLATION/MAINTENANCE TOOLS
505-16 SPARE PARTS LIST
505-17 INSTALLATION/OPERATION/MAINTENANCE MANUALS
505-18 SYSTEM HAZARD ANALYSIS
505-19 DESIGN DATA BROCHURE
505-20 NON CONFORMANCE REPORT



NOTE: △ indicates phase completion
△--→△ Schedule extended
△←--△ Schedule retracted

Figure 3

An Analysis of the Low Loss Evacuated Tubular Collector Using Air as the Heat Transfer Fluid

By

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Toledo, Ohio

TABLE OF CONTENTS

Acknowledgements

The support of the Solar Division of the Energy Research and Development Administration through Contract No. EY-76-C-02-2919 contributed significantly to the construction, testing and data analysis of the air driven solar energy collector subsystems.

Introduction:

The economics and cost effectiveness of solar thermal collectors may be enhanced significantly if they are able to drive cooling cycles effectively as well as provide for space and domestic hot water heating applications. Higher temperatures introduce the potential for wider usage of solar thermal energy in providing heat for industrial processes, desalination, agriculture and food processing and similar frequently discussed energy dependent enterprises. The use of air as the energy transport fluid is becoming a preferred solution to many solar energy system practicioners.

A solar thermal energy collector, using the spaced evacuated tubular absorber configuration, has demonstrated the ability to provide high grade solar energy. An understanding of the differences from and similarities to the traditional flat surface collector family is important to the user. The use of vacuum to essentially eliminate the thermal losses due to conduction and convection, in conjunction with a highly selective absorber surface results in a low loss coefficient collector. The exploitation of the symmetry of the absorber surface through the spacing of the elements in conjunction with a simple diffuse light reflecting background enhances the effectiveness of each of the cost sensitive units. The light enhancement factor combined with the low loss coefficient characteristic improves the effectiveness in the use of air as the heat transfer or collector cooling fluid.

A one hundred and forty-four (144) evacuated, spaced tubular collector array was designed, installed and tested to evaluate and demonstrate the effectiveness of an air driven collector. The effort was partially funded by the Solar Division of the Energy Research and Development Administration. The thermal performance and operating characteristics were explored for a range of air mass flow rates. The testing was conducted using data collection and evaluation on an all day basis. This allowed a considerable degree of flexibility in the design of the test, in the accuracy of the control required for air mass flow rate, collector inlet temperature and the fluctuation of solar insolation throughout the daily test period. The latter is a very important consideration, at least in Toledo, Ohio, where clear day conditions are indeed a rarity.

The air driven collector demonstrated a high level of thermal performance throughout an operating temperature regime from ambient to collector fluid discharge temperatures in the range of 300°F. The performance was essentially unaffected by ambient conditions where temperatures of 15°F below zero and chill factors of 50°F below zero were experienced. Heavy and extended periods of snow fall in addition to high winds and extended cold were common during the 1976-1977 winter season in Toledo; the collector was operated with no significant adverse affects due to such climatic conditions.

Discussion:

The design of a multi-tubular collector array, using air as the heat transfer fluid, must provide simplicity in the attachment of the many discrete elements to an air manifold. The ease of the insertion and support of the tubes, the avoidance of air leakage at each interface, the uniformity of air flow distribution to each of the elements, a low thermal loss, aesthetics and cost are among the factors requiring consideration. The thermal performance of evacuated tubular collector elements using a variety of heat transfer fluid characteristics is discussed by Beekley, et. al.(1). Based on the Beekley analysis, it was decided to investigate a range of collector air flow rates from 4 SCFM to 8 SCFM 8 to 16 pounds per hour per square foot. The resulting data would offer a flexibility in system design considering collector performance, manifold and ducting size and air pumping power.

Preliminary testing and analysis at Owens-Illinois had characterized the air flow-pressure drop properties of the tubular collector element. While affected by dimensional values such as the diameter and location of the distributor tube relative to the absorber tube, a relationship of the form:

$$\Delta P = \frac{CFM^x}{C} \quad \text{in, W. G.} \tag{1}$$

was established and for the particular system chosen a relation of:

$$\Delta P = \frac{CFM^{1.76}}{23} \quad \text{in. W.G.} \tag{2}$$

was determined to be valid for a flow path where two collector tubes were operated in a series flow arrangement. The air flow path for the two tube series flow arrangement is indicated schematically in Figure 1. Each tube pair is operated in parallel flow relative to the manifold arrangement. Some researchers such as Grunes [2] and Hughes, et. al.[3] report that a two to one ratio of flow distribution can be accepted without serious impact on the thermal performance of an air driven evacuated tubular collector array. An objective of the one hundred and forty-four (144) tubular collector design was to establish a test system (manifold arrangement) which would minimize the number of penetrations required through a roof/facade and which would maintain better than a two to one ratio of flow maldistribution; the latter towards the potential of increasing the size of a total array without the necessity of adding to the number of roof penetrations required.

The manifold design chosen is shown schematically in Figure 2. The side by side air flow channels in the manifold will be noted. The unobstructed flow cross sectional area in the inlet and exit air flow ducts is about forty (40) square inches. The manifold transition flow areas to and from the two tube flow path are in the extensions at the top of the manifold. This arrangement was chosen to result in a minimum of the manifold extending above the false background which acts as the diffuse reflector and hides the larger portion of the manifold simply for aesthetic purposes. A secondary advantage is that the distributor tubes are

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disassociated from the main air flow path. This provides a somewhat improved transition region between the main flow sections of the manifold and the tubes towards attaining some reduction in overall system pressure drop.

The distributor tube is about 1 inch in diameter. Aluminum thin wall tubing was chosen for expediency; glass tubing is planned for volume production. The tubing is held in place and sealed to prevent air leakage to the exit duct in the manifold by a gasket mounted in holes in the thin metal plate dividing the inlet and exit air flow paths. The two foamed manifold halves utilize gasket material to hold the metal plate in place and seal against air leakage. A metallic dividing plate was found to provide adequate thermal insulation between the two flow paths.

The manifold is center fed and in the test arrangement eighteen tube pairs extend to either side of the transition feed connecting the manifolds to the main air ducting located within the building enclosure. A two tier manifold installation was selected towards maximum flexibility in the arrangement of future commercial installations of the air collector array. The manifold construction utilized an inner and outer skin formed by hand layup of a glass reinforced high temperature resin. The tubular connector, as a matter of convenience, utilized components developed for the liquid tubular collector in prototype volume production. The manifolds were assembled in halves which included the connectors for the individual tube elements. The cavity was filled with a low density-high temperature polyurethane compound. The liquid mixture was pumped into the cavity and allowed to foam in place to form a relatively rigid structure. The manifold fabrication process was chosen as one having the future potential for high volume-low cost production following the teachings of the automotive field. A similar procedure is currently employed in the fabrication of the manifolds for the liquid tubular collector.

The collector tubes elements were taken from finished goods inventory for components of the liquid evacuated tubular collectors in pilot production. A collector tube element is shown schematically in Figure 3. The absorber tube is about

45 inches long and extends at its open end beyond the cover tube to fit into the tube socket and "O" ring seal. A selective coating is applied to the external surface of the absorber tube prior to assembly into the cover tube. Stability between the cover and absorber tubes is provided by a spring arrangement at the closed ends of the tubes. The spring provides radial support while allowing freedom in the axial direction to accomodate differential expansion. Attached to the support spring is a barium getter which contributes to the extended life of the vacuum which is lower than 10^{-4} torr applied to the volume between the two tubes. Photographs, Figures 4 through 6, show the completed array with and without snow cover.

The test loop which was installed is shown schematically in Figure 7. Ducting provides for full ingestion and overboard dumping from and to ambient air. A bypass and suitable dampers allow for up to full recirculation of the collector discharge air. An electric air preheater provides for approximate control of the collector inlet air temperature. A variable speed drive is used to control the air fan to provide air flow rates up to approximately 10 SCFM per tube pair.

The test system was extensively instrumented with temperature sensors. Four element thermopiles are located in the inlet and exit ducts of each of the banks of the array to sense the collector temperature rise. Thermocouples are also mounted in the same vicinity to indicate the air temperatures in each of the ducts. The air mass flow is derived by an array of pitot tubes manifolded together behind an air straightener in a manner to derive average velocity pressure sensed by a pressure transducer. Four thermocouples disposed 90 degrees apart sense the air temperature exiting from the velocity pressure sensor for density compensation. A pyronometer senses total insolation in the plane of the collector. This information is fed into a signal conditioning unit for analog to digital conversion and integration of the insolation data. The information is recorded on magnetic tape in five minute increments throughout a twenty-four hour day.

Additional thermocouples were added to sense temperatures in the locations indicated. The thermocouple pair mounted in the main inlet and exit ducts, respectively, provide a check on the thermal performance of the overall 144 tube array. The thermocouples mounted in the manifolds in the inlet and exit air flow paths of the respective tube pairs sense the temperature rise in individual tube pairs. The temperature rise per tube pair is an excellent indication of air mass flow rate distribution. The thermocouples mounted in the annulus of the tubes are useful in sensing the annulus air temperature under both flow and no-flow conditions. The latter sensed temperature may be useful in the logic used to control collector operation; the former sensed temperature provides a useful indication of the absorber surface temperature at its highest temperature axial position. Thirty of these temperatures require manual read out and recording; the remaining are available on a strip chart recorder. Static pressure taps sense the pressure levels in the inlet and exit air flow ducts.

Upon completion of the installation, air flow vs. collector pressure drop and leakage flow tests were performed. The variable speed fan was adjusted to provide a static pressure relative to ambient equal to that existing under rated flow conditions. The exit duct was blocked during the tests. A leakage flow of less than 2% of rated flow was indicated. The air flow rate-pressure drop relationship was then measured. A best fit line on a log-log plot of the data indicates that the relationship for the manifold and collector tube assembly can be expressed as:

$$\Delta P = \frac{CFM^{1.84}}{27} \text{ in. W.G.} \quad (3)$$

At a flow rate of 4 SCFM/tube pair, the pressure drops are found to be equal within the limits of experimental error for the two tube pair and the one hundred and forty-four tube array and verify that the pressure drop due to air flow in the manifold is negligible. An excellent flow distribution between tube pairs can also be inferred from the similarity in temperature rise measured for tube pairs. A significant increase in the number of tube pairs per manifold/roof penetration can therefore be anticipated without an adverse affect on air flow distribution. The agreement in the flow-pressure drop relationships between the 144 tube array and the 2 tube experimental testing also implies that the leakage and mass flow rates are within 5% of the measured values for the two systems.

The large pressure drop, often expressed as a potential problem when using air as the heat transfer fluid, is avoided by the flow path and manifold arrangement which is employed. At a 0.5 in. W.G. pressure drop and flow rate of 4 SCFM per tube pair, the theoretical air horsepower required to drive a 144 tube array is only 0.023 HP. Even at a fan efficiency of 20%, a 0.1 horsepower drive would accomodate the pumping power requirements of the collector or require about one percent of the useful solar energy collected. Of course, system pressure drop factors for ducting, dampers, storage and similar elements must also be considered. On the average, the system air pumping power should be less than 5 percent of the total useful energy collected. The energy loss associated with the efficiency of the air fan is captured in the air stream as a temperature rise and, in fact, does not represent an energy penalty to the system.

The thermal performance of the 144 tube air collector array is shown in Figure 8. The ordinate represents the percent of useful energy gain available from the collector per day. The abscissa is in a form similar to that used for the common "instantaneous" efficiency presentation but represents integrated daily values rather than the common "instantaneous" values. Thus, $\sum_{1}^{N} T_{i,in}$ is the integrated value of the collector fluid inlet temperature over the day summed for time increments of "i" for N increments of time interval. The value $\sum_{1}^{N} T_{i,a}$ and $\sum_{1}^{N} I_{i,TP}$ represent similar integrated values for ambient temeprature and total insolation in the plane of the collector, respectively. Daily data points are indicated for three different air flow values of approximately 4 SCFM, 6 SCFM and 8 SCFM. A linear least squares best fit line has been drawn for these air flow values. The excellent efficiency of the spaced tubular collector will be noted by the relatively high value of the intercepts and the low loss coefficient characteristic emphasized by the shallow slope of the efficiency curve. A factor not apparent from Figure 8, but of significance to the systems engineer/user, is the relatively high temperature rise in the air as it passes through the collector. At a flow rate of 4 SCFM per tube pair, a typical value for the average temperature rise in the air as it passes through the collector on a reasonably good day would be in the order of 80°F. Thus, the air collector provides a high grade of energy to the storage/load systems.

Figure 8 is represented in Figure 9 without the data points. Instead, the expressions for the intercept, η_o, and the slope of the efficiency curve have been inserted. The intercept, η_o, is defined by:

$$\eta_o = \frac{A_c}{A_p} F_R \phi \tau \alpha (I_{eff}/I_{TP}) \tag{4}$$

The ratio A_c/A_p represents the total cross sectional area of the absorber tubes divided by the total effective installed area of the collector. This term accounts for the gaps between the spaced absorber tubes. For a seventy-two (72) tube module with the centerline-to-centerline tube spacing equal to twice the cover tube diameter,

$$A_c/A_p = .42$$

The flow factor, F_R, is defined by Beekley(5) as:

$$F_R = \frac{1}{1 + \frac{\Pi U_L A_c}{2\dot{m} C_p}} \tag{5}$$

$$\text{Slope} = F_R \frac{A_c}{A_p} \Pi U_L \tag{6}$$

$$F_R = \text{slope}\,(A_p/A_c)(1/\Pi U_L)$$

$$\text{Let } K = \text{slope}(A_p/A_c)$$

$$U_L = \frac{K \times 2\dot{m}Cp}{\Pi \times 2\dot{m}Cp - \Pi KA_c} \quad (7)$$

$$A_c = 35 Ft.^2$$

$$A_p = 82.8 Ft.^2$$

The term, I_{eff}, represents the increase in the level of solar radiation received by the absorber surface as a result of tube spacing. At solar noon, the light enhancement due to direct and diffuse light reflections from the backing screen has been determined to be 1.63 times the total incident radiation falling directly on the absorber tube cross sectional area. The term Φ represents the effect of the solar noon light enhancement factor integrated on a weighted basis over an entire day. Given the collector constants derived from other experiments, τ = .92, α = .85 and (I_{eff}/I_{TP}) = 1.63, the air collector coefficients may be determined as a function of flow rate.

Flow CFM/Tube Pair	$\dot{m}$ #/Hr.	Slope BTU/Ft.²Hr.°F	K	U_c BTU/Ft.²Hr.°F	F_R	η_o	Φ
4	324	-.225	.53	.191	.92	.616	1.25
6	486	-.200	.47	.161	.95	.644	1.27
8	648	-.213	.50	.169	.94	.659	1.31

It is evident that the collector coefficient, F_R, remains at a remarkably high value down to a flow rate of 4 CFM per tube pair. The percentage loss in collector performance is offset by the reduction in air pumping power when operating at 4 CFM vs. 6 or 8 CFM per tube pair flow rates. The daily energy enhancement factor, Φ, would be expected to be a constant independent of flow rate. Since the numbers are in agreement within the standard deviation factors for slope and intercept values which result in determining the linear least squares fit, it would be expected that additional test data would verify this fact.

The fact that the term, Φ, had to be added to the relationship commonly applied to η_o is important to the understanding of the optical properties which obtain for the spaced tubular configuration. The factor, Φ is related to the Incident Angle Modifier term now being introduced by some researchers in the field such as Simon [4]. The value of Φ will be greater than unity for the spaced tubular elements and diffuse light backing screen collector configuration; the value of Φ will be less than unity for most, if not all, flat absorber surface collector configurations.

This factor presents a strong argument for the development of collector performance on a daily basis rather than an "instantaneous" basis as is now being proposed by some advisory bodies. The use of an indoor simulator light source does not introduce the effects of diffuse light in an acceptable fashion. Outdoor testing over selected time increments to develop a value for Φ lacks in the stability of the light source especially at the larger values of the hour angle, ω. The selected time increment is dependent on the time constant of the collector which for the tubular air collector is quite long at the order of 15 minutes. This implies the requirement for a stable test condition for a period of 45 to 60 minutes following any upset. Several attempts have been made to correlate the available test data for the air system on the basis of discrete time steps throughout the day. The results have not been useful due to the extreme scatter in the reduced data.

On the other hand, the data reduced on a daily basis, as indicated in Figure 8, shows a remarkably low scatter band. The types of days include relatively clear sunny conditions, overcast, intermittant clouds and even intermittant snow and rain conditions. At times, air inlet temperatures to the collector were held to the order of $\pm$ 5°F and at others allowed to float as they would under normal system operating conditions. This suggests that the characteristics of a collector can best be characterized through the use of all day test data which acknowledges that the energy source is randomly time dependent and unpredictable load factors dictate the operating boundry conditions for the solar collector.

Since the surface area of the air manifold is relatively large, its loss term and impact on detail design was considered to be significant. The value of the loss coefficient, U_L, for flow rates of 4 CFM per tube pair is:

$$U_L = .191 \text{ BTU/Hr.Ft.}^2\text{°F}$$

The radiation loss term, U_{LC}, may be determined from prior knowledge of the typical emittance value of the selective absorber surface. This has been the subject of extensive investigation as a part of the development of the liquid system. A typical value for the emittance, ε, is 0.070. The radiation loss is determined by:

$$Q_L = \sigma\varepsilon(T_C{}^4 - T_a{}^4) \qquad (8)$$

$$U_{LC} = \sigma\varepsilon\, \frac{(T_C{}^2 + T_a{}^2)(T_C + T_a)(T_C - T_a)}{(T_C - T_a)} \qquad (9)$$

The conventions of solar collector analysis linearize the loss term as indicated in equation (5) and equate T_c to T_{in} to a good approximation for the analysis of the loss coefficient, U_L. Typical operating temperature conditions for an air driven collector of T_{in} = 200°F and T_a = 70°F have been selected for the following linearized analysis.

$$U_{LC} = \sigma\varepsilon(T_{in}{}^2 + T_a{}^2)(T_{in} + T_a) \qquad (10)$$

$$U_{LC} = .1713 \times 10^{-8} \times .070(660^2 + 530^2)(660 + 530)$$

$$U_{LC} = .10 \text{ BTU/Hr.Ft.}^2\text{°F}$$

The total loss coefficient U_L is related to the losses associated with the collector tube element radiation losses and manifold conduction/convection losses by:

$$U_{LM} \times A_M \Delta T + \pi A_C U_{LC} \Delta T = \pi A_C U_L \Delta T$$

$$U_{LM} = \frac{\pi A_C}{A_M}(U_L - U_{LC}) \qquad (11)$$

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The values for A_C and A_M are 35 Ft.^{2} and 82.8 Ft.^{2} respectively for the collector array under test.

$$U_{LM} = \frac{\pi \times 35}{82.8} (.175 - .10)$$

$$U_{LM} = .12 \text{ BTU/Hr.Ft.}^2 {}^\circ F$$

An analysis of the dimensional and insulation values of the manifold implies a value for $U_{LM} = 0.10$ BTU/Hr.Ft.^{2}°F. The remainder of the manifold losses are attributed to tube attachment points.

Using the imperically determined constants, a model describing the thermal performance of an air driven evacuated tubular collector may be developed:

$$\eta = \frac{A_C}{A_P} F_R \left[\frac{\phi\, \tau\alpha \bar{I}_{eff}}{\bar{I}_{TP}} - \frac{\pi U_L (T_{in} - T_a)}{\bar{I}_{TP}} \right] \qquad (12)$$

For the specific collector configuration tested at a flow rate of 4 SCFM per tube pair, on a daily efficiency basis this expression may be reduced to:

$$\bar{\eta} = .62 - \frac{.23 (\bar{T}_{in} - \bar{T}_a)}{\bar{I}_{TP}}$$

The collector was facing south and mounted at a tilt angle of 45°. The latitude of the test site is 42°. The background was a diffuse white flat surface constructed of plywood and painted to provide a flat white surface. Shaped background developments and tested on liquid systems have demonstrated a significant improvement in performance over the plain white diffuse backing surface. A simple conical shaped reflector would be expected to improve the daily performance of the collector by about 20 percent at the intercept. Such shapes may be applied without alteration to either the air or liquid collector tube elements. Field testing on liquid systems is now underway prior to the release of the shaped background reflectors for general use.

The three terms required to evaluate the thermal performance of an air driven, spaced, evacuated tubular collector south facing and mounted at a tilt angle equal to the lattitude may be derived by manipulation of data or taken directly from the Climatic Atlas. The user must determine his average load requirements and select a suitable thermal storage capacity. From this data, an average value $\bar{T}_{in}$ during the daily collection period may be applied. Because of the low loss coefficient of the collector, a relatively large error in the determination of $\bar{T}_{in}$ may be accepted without a serious impace on the final results. The average value of $\bar{T}_a$ for the daily collection period and the average daily value of $\bar{I}_{TP}$ are available from the Climatic Atlas.

Conclusions:

The use of air as the energy transport medium in the spaced, evacuated tubular collector provides a high temperature - high performance subsystem. The selection of air mass flow rates in the range of eight to sixteen pounds per hour per effective square foot of installed collector area restrains the collector pressure drop to acceptable levels. The relatively high temperature rise in the air as it passes through the collector is accounted for in the thermal performance equation. The rather large increase in collector exit fluid temperature provides the high grade of energy required for cooling cycles and many industrial heat applications. The inherent problems associated with liquid transport fluids (freezing, boiling, corrosion, etc.) are avoided. The high stagnation temperature problems of the low loss coefficient evacuated tubular collectors experienced when using liquids are eliminated.

The excellent performance reported for the evacuated tubular collector is realized because of the spacial arrangement of the collector elements and the inherently low loss characteristic resulting from the use of vacuum as insulation in combination with a highly selective absorber surface. The low emittance of the surface controls its losses as the temperature increases. The spaced arrangement of the cylindrical tubes contributes significantly to the light enhancement factor and, in addition, minimizes the ratio of loss surface area to effective collection surface area. Thus, the collector flow factor remains very high for the evacuated tubular collector configuration when air is used as the heat transfer fluid; the lower heat transfer film coefficient experienced when air is used seriously degrades the collector flow factor for flat surface collectors with their inherently higher loss coefficient and resulting sensitivity to absorber surface temperature.

The correlation of performance based on an incremental time analysis of data has been unsuccessful because of the relatively long time constant of the air driven collector and the random time dependence of the energy source. Five days of data were investigated based on 20 minute and 40 minute time increments on either side of solar noon. No meaningful correllation of results were obtained. It is expected that a minimum time increment of 60 to 75 minutes and a massive amount of experimental data would have to be evaluated to obtain acceptable data. On the other hand, when all day data is used, the degree of scatter in the data is acceptable even for moderate numbers of data points. The self integrating effect of the energy source, since it starts and stops always at a value of zero, and the low thermal mass of the collector, are believed to be the principal factors contributing to excellent test analysis correllation when the data is derived on a daily basis.

With the use of pebble bed storage, the upper limit on acceptable operating temperature is extended significantly. Fluid pressure considerations and inherent implications are eliminated. The high temperature capability of the collector significantly reduces the storage problem in terms of size, weight and cost. Off peak power utilization becomes feasible with the acceptance of higher excursions of operating temperature.

The use of all day data in describing the collector characteristics greatly reduces the demands on the collector test loop and the need for "ideal day" test conditions. Collectors featuring the enhancement of diffuse light and which have relatively long time constant characteristics require extended test periods. The limitations of the choice of test site locations imposed by the stringent requirement for continuity in the level of solar insolation during the prescribed "instantaneous" test periods are avoided.

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(2) Howard Grunes, "Utilization and Operational Characteristics of Evacuated Tubular Solar Collectors Using Air as the Working Fluid", University of Wisconsin - Madison, 1976.

(3) Patrick J. Hughes, "The Design and Predicted Performance of Arlington House", University of Wisconsin - Madison, 1975.

(4) F. F. Simon, "Flat-Plate Solar Collector Performance Evaluation With a Solar Simulator as a Basis for Collector Selection and Performance Prediction." Solar Energy, Vol. 18, pp. 451 - 466. Pergamon Press 1976.

List of Test Equipment

Measurement	Instrumentation
I_{TP} : Solar total radiation in plane of collector	Epply Precision Pyranometer with integrator.
T_a : Ambient Temperature	Copper-Constantan radiation shielded thermocouple
$T_{out} - T_{in}$: Collector bank temperature rise	Four element copper-constantan thermopiles
T_{in} : Inlet temperature to each collector bank	copper-constanan thermocouples
m : Air flow rate to collector	Air Monitor Corp. Duct Air Monitor Device
ΔP : Collector Pressure Drop	Magnehelic, Model No. 2002C, Dwyer Instruments, Inc.
Data Logger :	John Fluke Co. (Model 2240 Data Logger)

Nomenclature

The symbols and nomenclature used in the body of the paper are listed generally in the order in which they first appear. Specific values for areas and constants are included as appropriate.

$$\Delta P = \frac{CFM^x}{C} \tag{1}$$

ΔP = collector pressure drop, inches, W.G.

CFM = Air flow rate, cubic feet per minute

x = exponent

C = constant

$$\eta_o = \frac{A_c}{A_p} F_R \Phi \tau \alpha \left(\frac{I_{eff}}{I_{TP}} \right) \tag{4}$$

η_o = Y intercept value of the daily efficiency

A_c = Collector loss area, 35Ft.^{2}

= Nd_aL_a

N = number of collector elements, 72

d_a = diameter of absorber tube, 1.69 in.

L_a = effective length of absorber tube, 41.375 in.

A_p = Collector effective installed area, 82.8Ft.^{2}

= Nd_sL_a

d_s = tube spacing, 4 in.

F_R = Collector flow factor, ratio of collector performance at actual absorber surface operating temperature to performance if operated at the fluid inlet temperature.

$$= \frac{1}{1 + \frac{\pi U_L A_c}{2 \dot{m} C_p}} \tag{5}$$

U_L = collector loss coefficient, BTU/Hr.Ft.^{2}°F

$\dot{m}$ = air mass flow rage, Lbs/Hr.

C_p = specific heat of air = .241 BTU/Lb.°F

Φ = collector performance factor; the ratio of the daily light enhancement vs. the solar noon light enhancement.

$$\Phi = \frac{\int^{day} I_{BP}\left\{\operatorname{Sec}\Omega + (1+\rho\Lambda)\left[1 + \frac{1-\rho\Lambda}{1+\rho\Lambda}(\operatorname{Sec}\Omega - 1)\right]\right\} dt}{I_{BP}\left\{\operatorname{Sec}\Omega + (1+\rho\Lambda)\left[1 + \frac{1-\rho\Lambda}{1+\rho\Lambda}(\operatorname{Sec}\Omega - 1)\right]\right\}\Big/_{\omega = 0}}$$

I_{BP} = Beam radiation in the plane of the collector.

τ = reflectance of the backing screen.

Λ = constant derived from the integration of optical effects related to collector geometry such as tube diameter - tube spacing, background screen spacing, etc.

Ω = true sun angle related to yearly declination, hour angle, collector tilt, lattitude of location, and angularity of collector plane relative to true south.

ω = hour angle.

The above applies strictly only to the beam portion of the solar radiation. Computor analysis indicates that in the case of the specific geometry and simpligying assumptions made for the 144 tube test array, the beam and diffuse radiation functions result in equal constants.

$$I_{eff} = I_{BP}\left\{\operatorname{Sec}\Omega + (1+\tau\Lambda)\left[1 + \frac{1-\tau\Lambda}{1+\tau\tau\Lambda}(\operatorname{Sec}\Omega - 1)\right]\right\}\Bigg|_{\omega = 0}$$

τ = Cover glass transmittance = .92

α = absorber surface absorptance = .85

ε = absorber surface emittance = .070

I_{eff}/I_{TP} = light enhancement factor - 1.63

TWO BANK AIR COLLECTOR ARRAY
SERIES/PARALLEL AIR FLOW PATH
(36 TUBE PAIRS PER BANK)

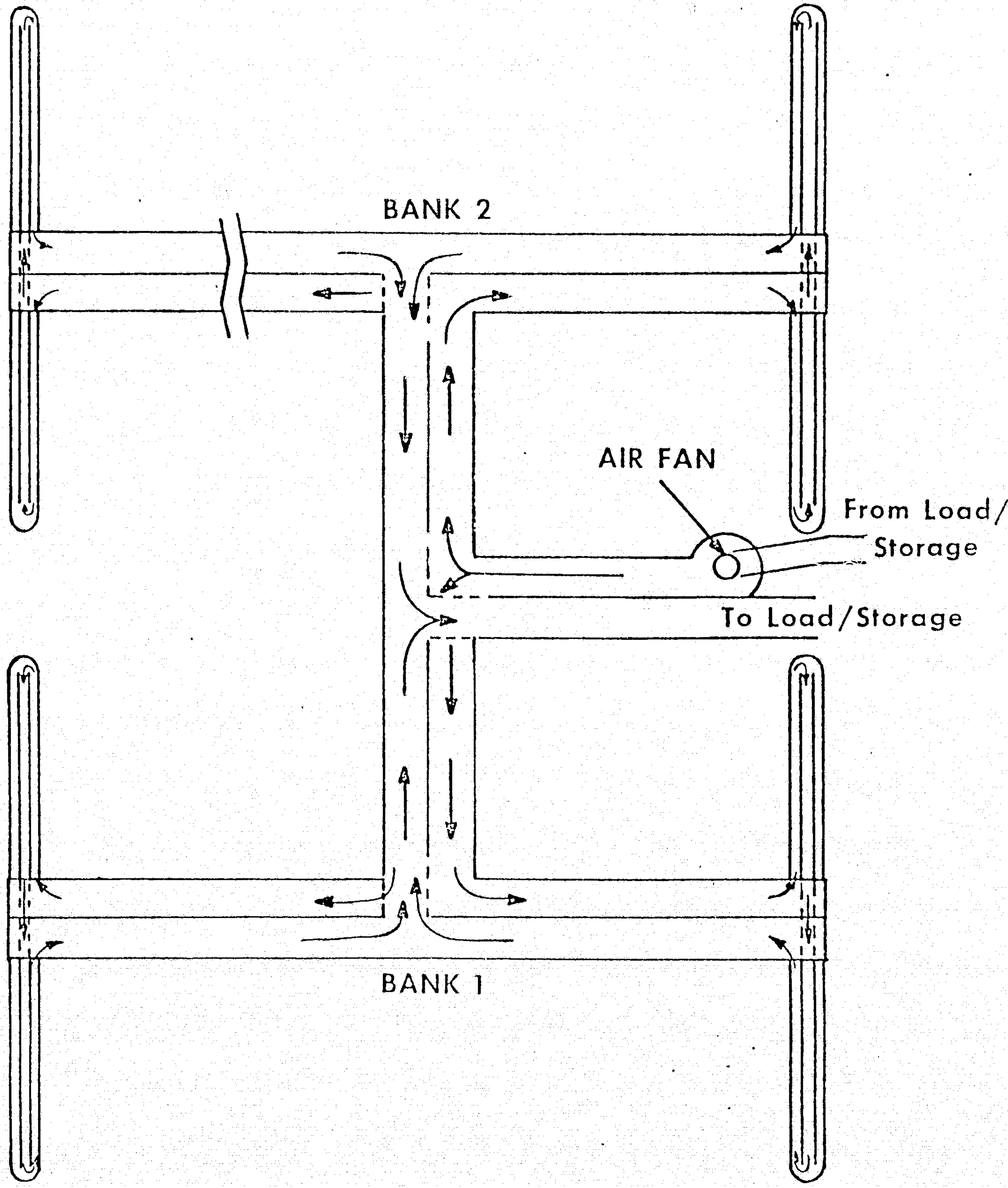
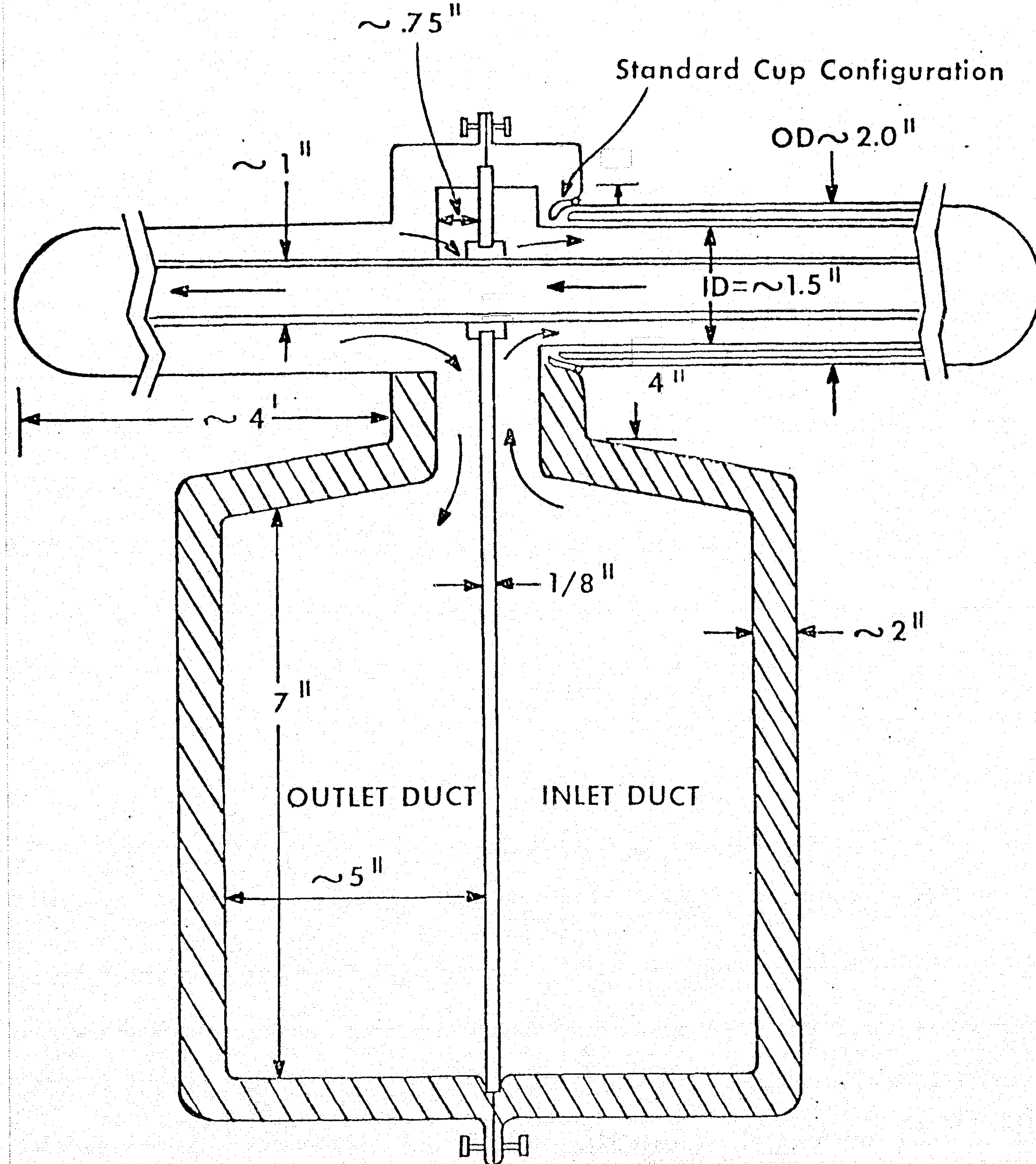


FIGURE 1.



~ .75"
Standard Cup Configuration
OD~2.0"
~ 1"
ID=~1.5"
4"
~ 4'
1/8"
~2"
7"
OUTLET DUCT
INLET DUCT
~5"

Figure 2

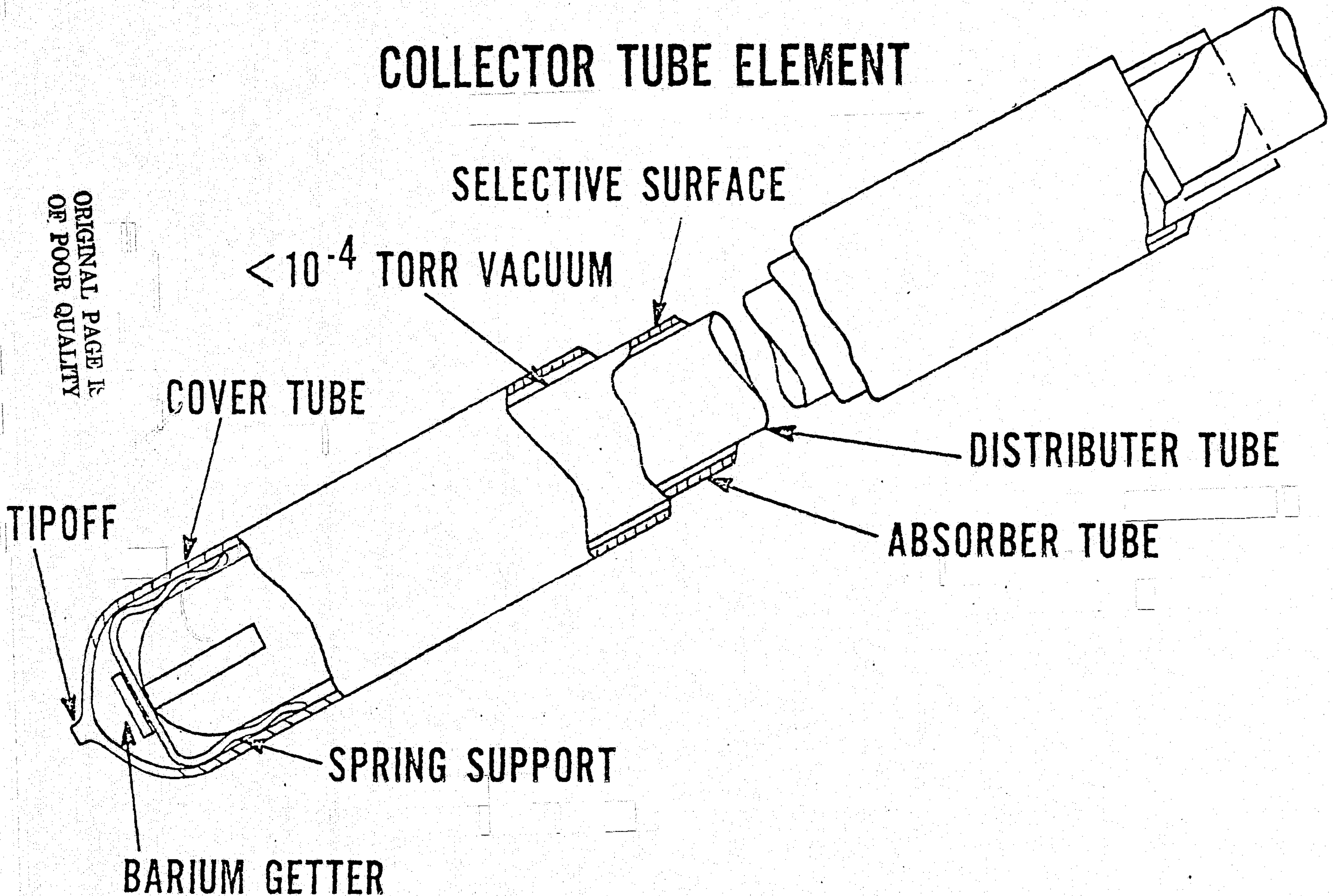


Figure 3

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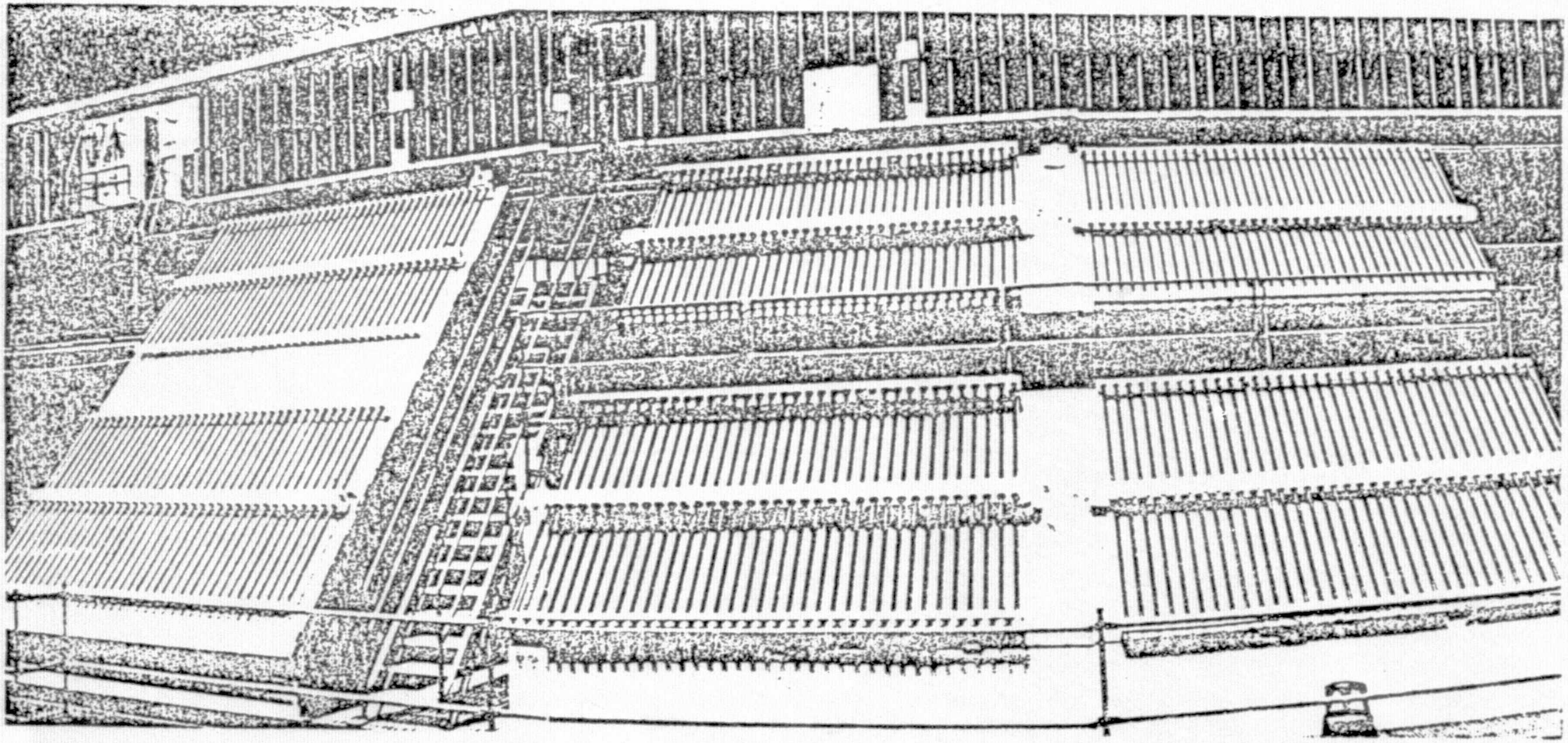


FIGURE 4

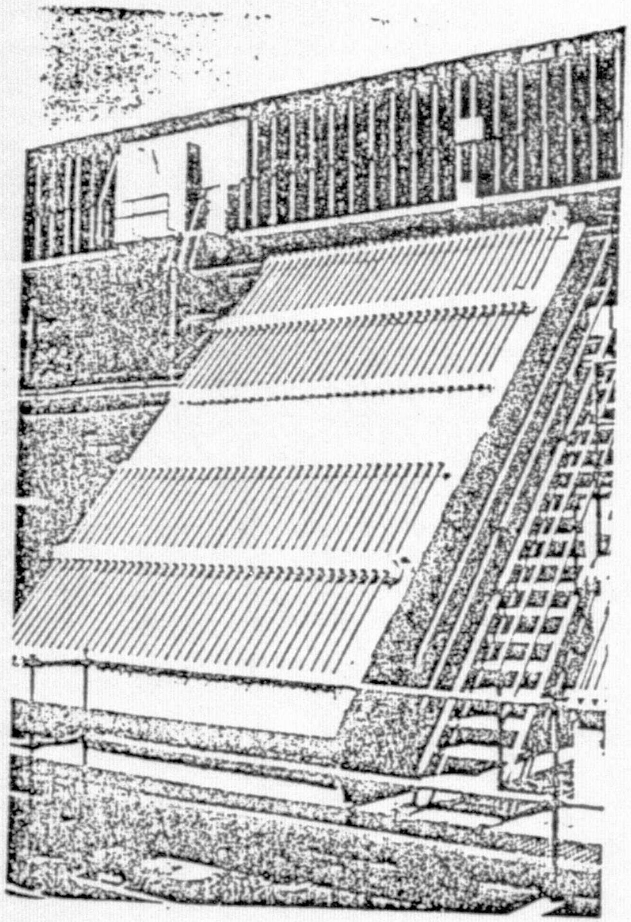


FIGURE 5

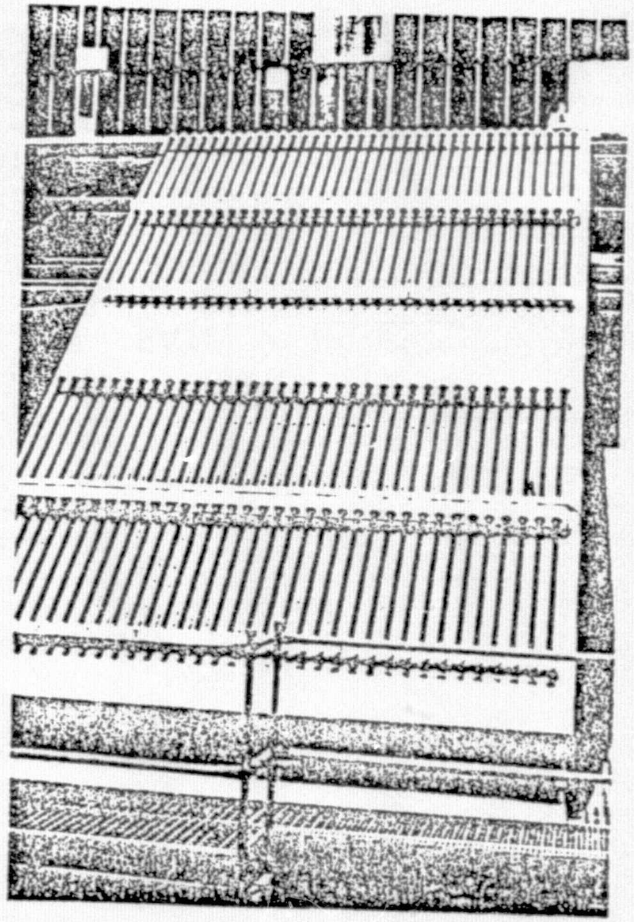


FIGURE 6

ERDA Air Collector Test Loop

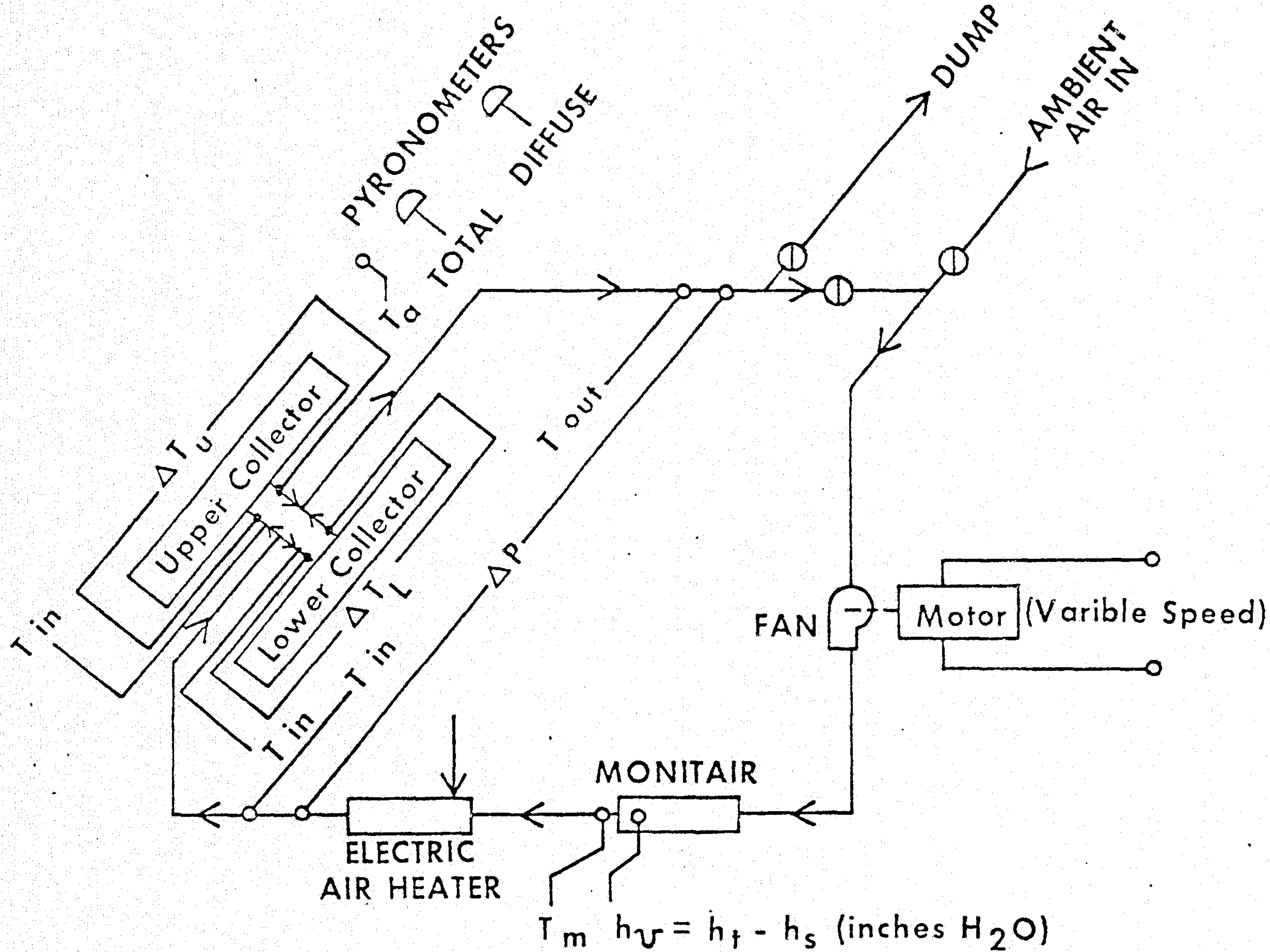


Figure 7

Air Driven Spaced Evacuated Tubular Collector

EXPERIMENTAL DATA POINTS
● 4 SCFM/TUBE PAIR
○ 6 SCFM/TUBE PAIR
□ 8 SCFM/TUBE PAIR

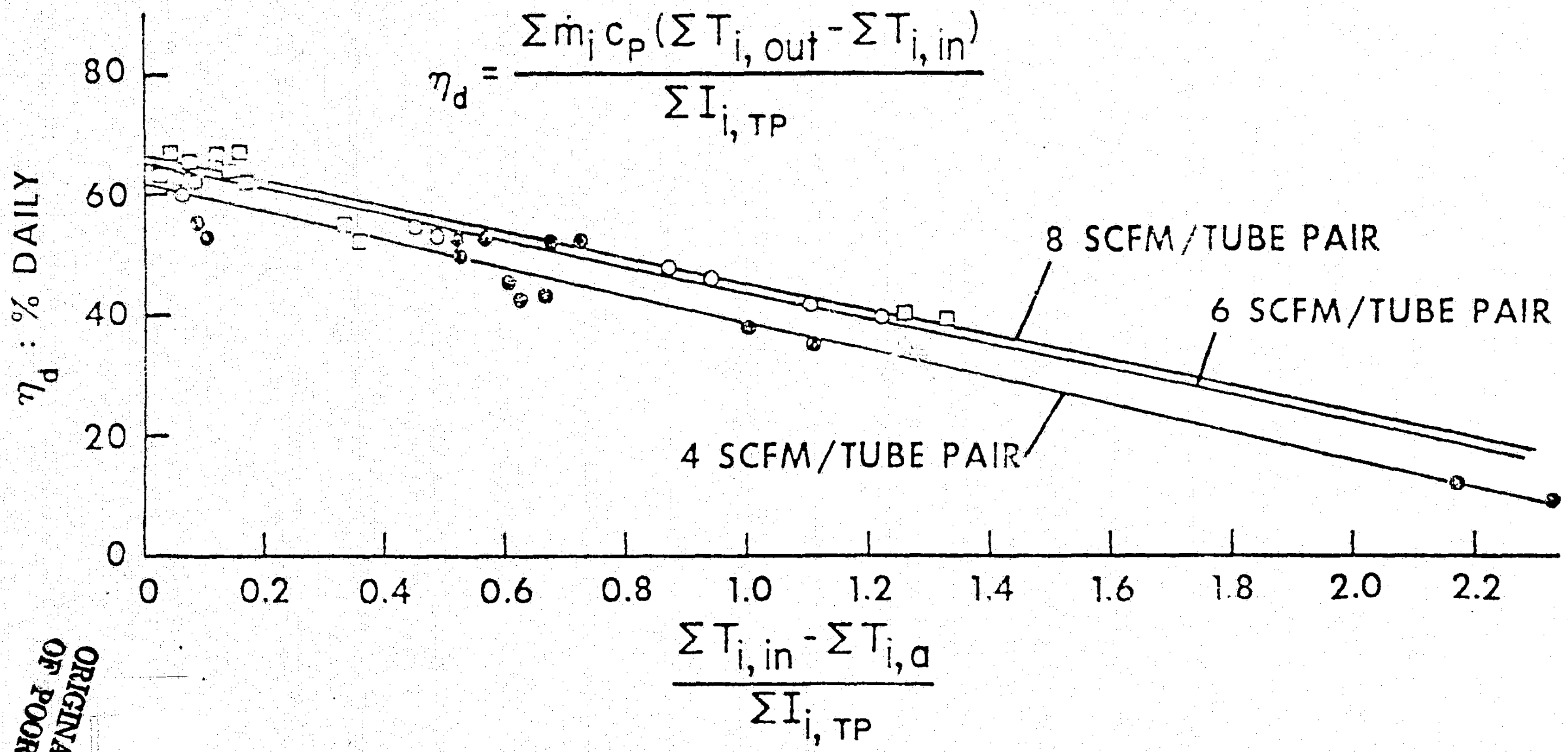


Figure 8

Air Driven Spaced Evacuated Tubular Collector

$$\eta = \frac{A_c}{A_p} F_R \left[\phi \tau \alpha \frac{I_{eff}}{I_{TP}} - \frac{\Pi U_L (T_{in} - T_a)}{I_{TP}} \right]$$

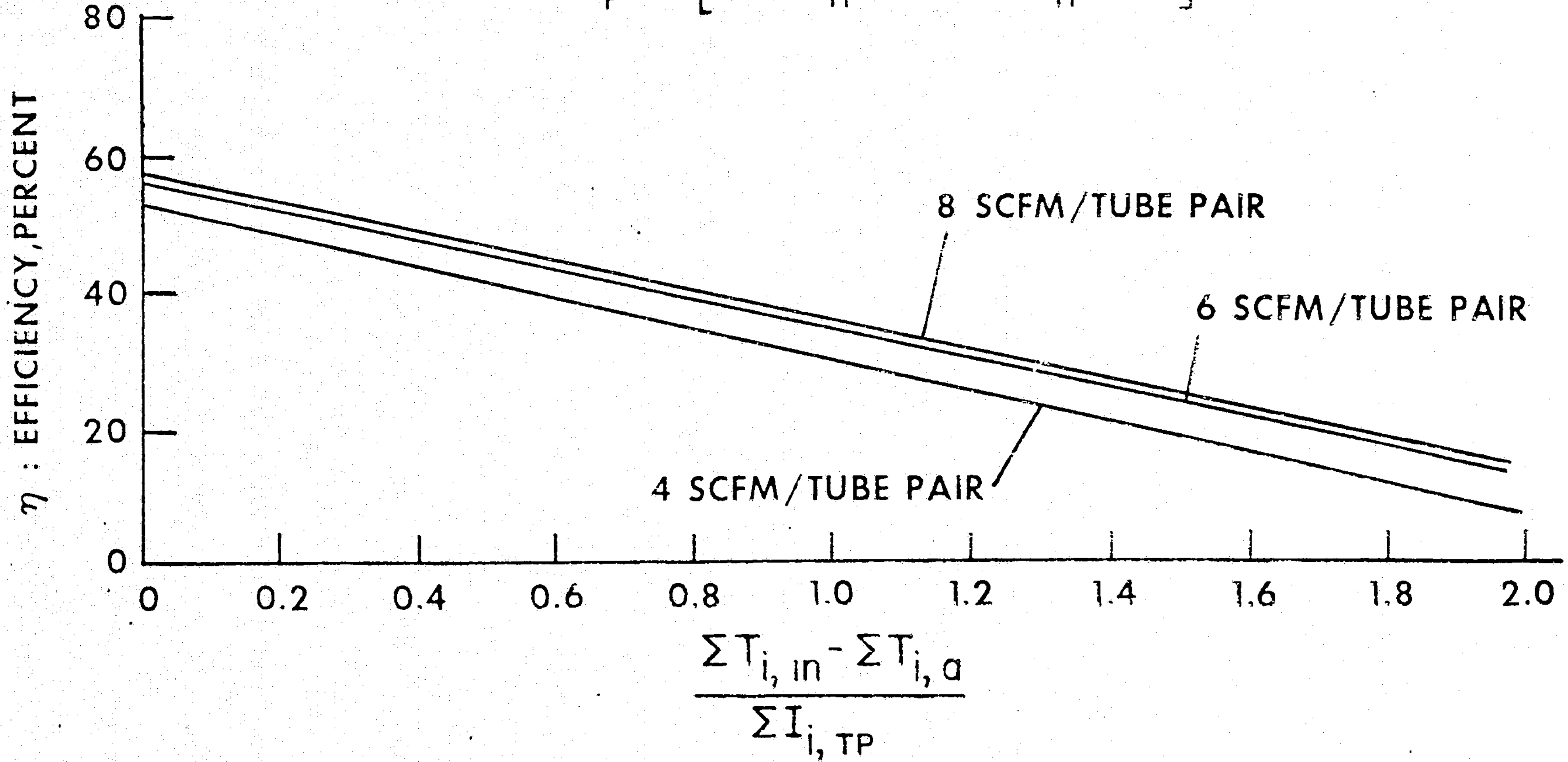


Figure 9

Contract No. NAS8-32259

Third Quarterly
May 1 - July 31, 1977

Summary. The principal activities during this reporting period consisted of the detail analysis of the 144 tube air collector performance. The results predict that the collector Model SEC-601 will meet or exceed the performance requirements of the subject contract. Firm vendor quotations revealed unexpectedly high costs for the sheet metal components making up the air manifold. In an effort to conserve project funds, essentially all other activities were curtailed until the problem can be resolved. A preliminary quality assurance plan was completed and submitted for review.

Technical Performance

A. General description of work accomplished during the reporting period.

1. Installation and layout design. The product design drawings for the manifold were submitted to four vendors for formal quotation and reviewed with additional sources. The formal quotations from two local HVAC sheet metal contractors proved to be exceedingly high. Detail formed costs of $125.00 per pound were quoted for several of the parts. Volume sheet metal fabricators were then contacted. A preferred vendor was selected and awarded a design

contract to produce shop drawings from the product drawings. Considerable leeway was granted so long as the thermal, structural or air containment properties of the part were not affected. Results from this effort are not expected until late August. As a result, some regression in percent completion of the design task will be experienced.

2. Prototype Design Review. Product design changes resulting from the sheet metal vendor activity will be prepared for formal approval.

3. Verification Tests. Design and construction of the test fixtures and the surveillance by Smith, Hinchman and Grylls have been suspended pending release for fabrication of the prototype baseline collectors.

4. Air Collector Performance and Operational Testing. The results of testing contained in the technical paper transmitted with the second quarterly report were updated with data taken during the intervening period. The updated results were presented at the annual contractors' meeting of the R&D Branch, Solar Division, ERDA, in early August. A copy of the presentation is attached. Performance exceeding contract requirements continue to be evidenced.

5. Installation, Operation and Maintenance Manual. Activities on these manuals have been suspended pending potential revisions per 1. above.

6. Quality Assurance Plan. A preliminary plan has been prepared and submitted for review under separate cover.

B. Forecast of activities to complete tasks.

1. The release for fabrication has been held pending the resolution of the cost of fabrication. Alternate design approaches are being developed through the sketch stage toward the attainment of a significant reduction in cost of fabrication. A cost target of less than $20.00 per sq. ft. has been established for intermediate volume production.

2. The work on test fixtures will be reinstated following release for fabrication of components. Lead time on the test fixtures is less than for the fabrication of components and are therefore not on the critical path.

3. All components needed for the modification of the test loop to incorporate the air liquid heat exchanger and simulated load elements are on order. The 350 gallon storage tank is currently on the critical path. The promised delivery is August 25. Based on such a delivery date, the test system is expected to be up and operating by mid-September. The heat exchanger is instrumented with six element thermopiles to obtain an accurate measure of temperature drop on the air side. The liquid side is also instrumented with a liquid flow meter and thermocouples to measure the thermal gain of the water as it passes through the exchanger. The latter set of experimental data will be used as a measure of the useful energy gain of the collector-heat exchanger combination. An energy balance will provide a check on the critical air mass flow measurement used in the analysis of the important collector parameters.

C. Identification of major problem areas or difficulties.

1. The high cost of fabrication of the sheet metal components of the manifold is the major problem area at this time. If an alternate design is required

to meet target costs, a further delay will be experienced. Alternate design approaches are being explored and rough costed in order to avoid the kinds of delays currently being experienced. A conversion cost of the order of 125:1 based on the base material cost was simply not anticipated.

2. The revised task schedule based on the use of the present product design was forwarded to the MSFC under separate cover. Its effect on contract schedule is reflected in the attached chart.

D. Data submittals.

PREFACE

This presentation covers the results of testing and analysis of a 144 element evacuated tubular array (168 ft.^{2} effective area) using air as the heat transfer fluid. The installation and test program was funded by ERDA. The period of performance is July 1976 through August 1977. A technical paper, prepared for the ISES meeting in Orlando, Florida has been updated to incorporate many more days of test data than were available at the time the original paper was prepared. Copies of the paper are available by request to our Toledo office. I will concentrate only on the performance highlights in the interest of time today.

Discussion

The first slide shows what the collector looks like. The test site is at the O-I Development Center, Toledo, Ohio. All of the ancillary equipment such as the air fan, ducting, instrumentation, etc. are located in the shed under the roof line.

The next slide presents a composite of the performance of the evacuated tubular collector on an average daily basis versus the target performance contained in the contract. The actual performance is well in excess of the target values.

Also shown is the performance of an air driven collector using instantaneous performance values and the incident angle modifier appropriate to a single glazed flat plate collector. The data for this plot was taken from ASHRAE 93-P dated January 15, 1977. Note where the air driven flat plate collector intersects the evacuated tubular collector performance.

The next series of slides present a plot of collector efficiency versus the common $\Delta T/I$ parameter for air flow rates from 8 pounds per hour - foot squared to 16 pounds per hour - foot squared. The dependence of the collector thermal performance on air flow rate is evident. The relatively high values of the intercept, η_o and the shallow slope reflecting a very low loss coefficient are worthy of comment. It should also be noted that the data is analyzed on a daily rather than an instantaneous basis. More on the reasons for this later.

In this slide we have concentrated on an air flow rate of 8 pounds per hour per foot squared. Our interest was centered on the effect of diffuse light on thermal performance. More data is needed to quantify the results but the trend for diffuse light days to be better than for clear days can be noted. Our diffuse day at 1300 BTU per foot squared day is much lower in total insolation than most people consider reasonable. The high population density parts of the country do not enjoy 365 days per year with insolation levels of 1800 BTU per foot squared day and above in the tilt plane of the collector. We believe performance under diffuse light conditions to be important.

The next slide presents a composite of the collector parameters of interest in comparing collectors. At .58 to .66, the η_o value is not quite as high as reported for many collectors. This results not because of a poor absorber selective surface, but rather because of the definition of the effective surface area of the collector. This includes the spacing between the tubes as well as the absorber surfaces. Thus, the intercept value is derated by the ratio of these areas which, for the collector data being reported, has a ratio of 0.42. The intercept, on the other hand, is also higher than the $F_R\tau\alpha$ Ac/Ap product because of the light enhancement factor and the greater than unity value of the incident angle modifier integrated over the day.

Note that values of the negative slope increase from .18 to .20 as the flow rates increase. One would at first expect the opposite to be true. However, when using air, the radiation term between the absorber tube and the distributor tube plays an important role in the heat transfer function. The decrease in the film coefficient due to a reduction in air flow rate appears to be offset by the increase in the radiation term and the result is essentially a push. The manifold heat losses also impact the slope values reported.

The high values for F_R in the order of .98 to .99 result from both the basic low loss coefficient of the evacuated tubular collector and the effect of tube spacing. For example, a close packed tubular array would have an F_R of about 0.92 at a nominal flow rate of 8 pounds per hour - foot squared.

The incident angle modifier on a daily basis is greater than 1. One would expect a constant value independent of air flow rate. Further testing and analysis would be desirable. The characteristic equation for the collector is shown at the bottom of the slide.

The next slide presents clearly one reason for our analysis of the test data on a daily rather than an instantaneous basis. The collector time constant, as measured experimentally, is approximately 25 minutes. The slide represents the analysis of data for the 6 clearest days available during the 6 month test period using increments of one time constant and numbered in order from 1 to 10 starting on either side of solar noon. The scatter in the data from time increment to time increment is as expected. No attempt is made to hold either the air flow rate or collector inlet temperature within close limits. It is unnecessary when testing on a daily basis. The change in energy state of the collector is both minimal and is accounted for in the analysis of the data. This makes the test procedures much simpler, cheaper to implement and very low in testing cost once the installation is completed.

This slide presents a second reason for the use of all day rather than instantaneous, clear day solar noon efficiency data. The fraction of diffuse light has an important role in the determination of the incident angle modifier. Secondary reflections play an important role in the hour angle dependence of the collector efficiency. Clear day testing (greater than 200 BTU/hr.ft.^{2}) will not identify collector performance characteristics under low level diffuse light conditions where the spaced evacuated tubular collector still operates quite effectively. The number of data points for diffuse type days with a total insolation of 1300 BTU per day foot squared is relatively small.

Air can be used as the heat transfer fluid in the collector and still be very effective in driving a load element requiring a liquid medium. Using the Duffie-Beckman approach to evaluating a heat exchanger in the loop, the F_R term may be modified to an F_R' term according to the equation shown on this slide. An air liquid heat exchanger having an effectiveness of only 50 percent will reduce the collector efficiency by only 4 percent or the order of only 2 percentage points in the operating range of interest.

Conclusions

Air cooled collectors have many inherent advantages. To be effective, such a collector must combine the characteristics of a very low loss coefficient with light enhancement features. The space evacuated tubular collector incorporating a highly selective absorber surface and spacing of the elements in conjunction with a diffuse light backing screen satisfy these requirements. Such a collector can drive air or liquid load loops with equal effectiveness. The air pumping power is low in comparison to the energy gain.

The space evacuated tubular collectors require testing on an all day basis and under both clear day and high fractions of diffuse light conditions. The long time constant of the collector and its sensitivity to diffuse light impose these requirements. The test procedures under ASHRAE 93-77 are not applicable or feasible for spaced evacuated tubular collectors.

The evacuated tubular collector configuration has received little attention compared to the flat plate collector, both experimentally and analytically. The collector configuration appears to have several advantages. The recommendations are:

1. Extend the period for testing and analysis. Even with the presently available data, time has not allowed an adequate evaluation of all of the factors affecting performance of the collector. Additional data points for bright and high fraction diffuse type days are needed to verify the trends found thus far. An air-liquid heat exchanger is being added to the test loop for brief testing under another ERDA contract. This element will increase the reliability of the air

mass flow measurement which has a direct impact on the experimental data and analysis.

2. Add at least one more test site to provide independent test data and under predominantly clear day test conditions. Such a test program would validate the collector performance in all parts of the country since the two extemes in weather conditions existing in the country would be investigated.

3. The extension of the program in conjunction with the testing at the Arlington House by the University of Wisconsin would provide a thorough evaluation of the air driven evacuated tubular collector.

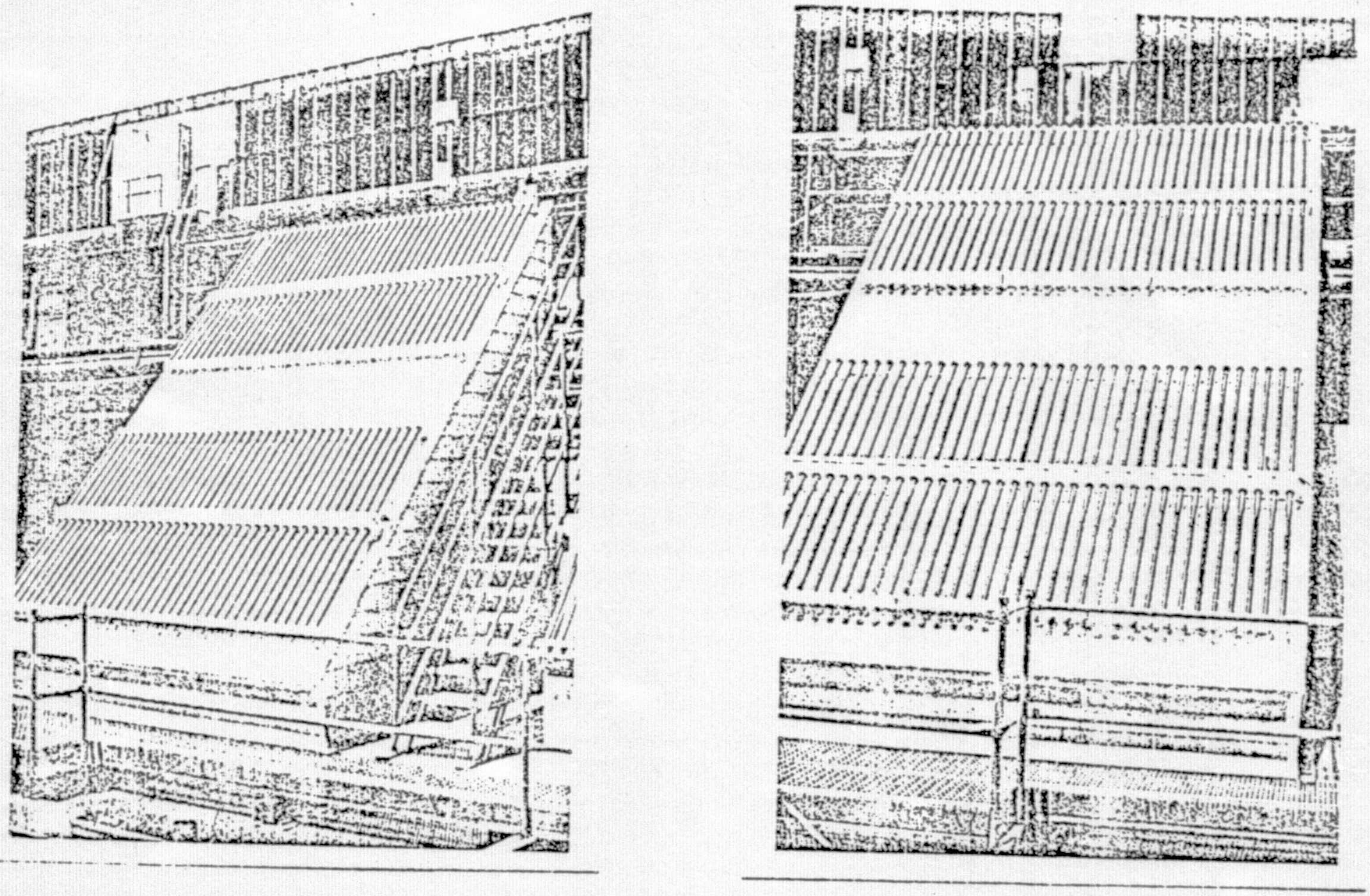
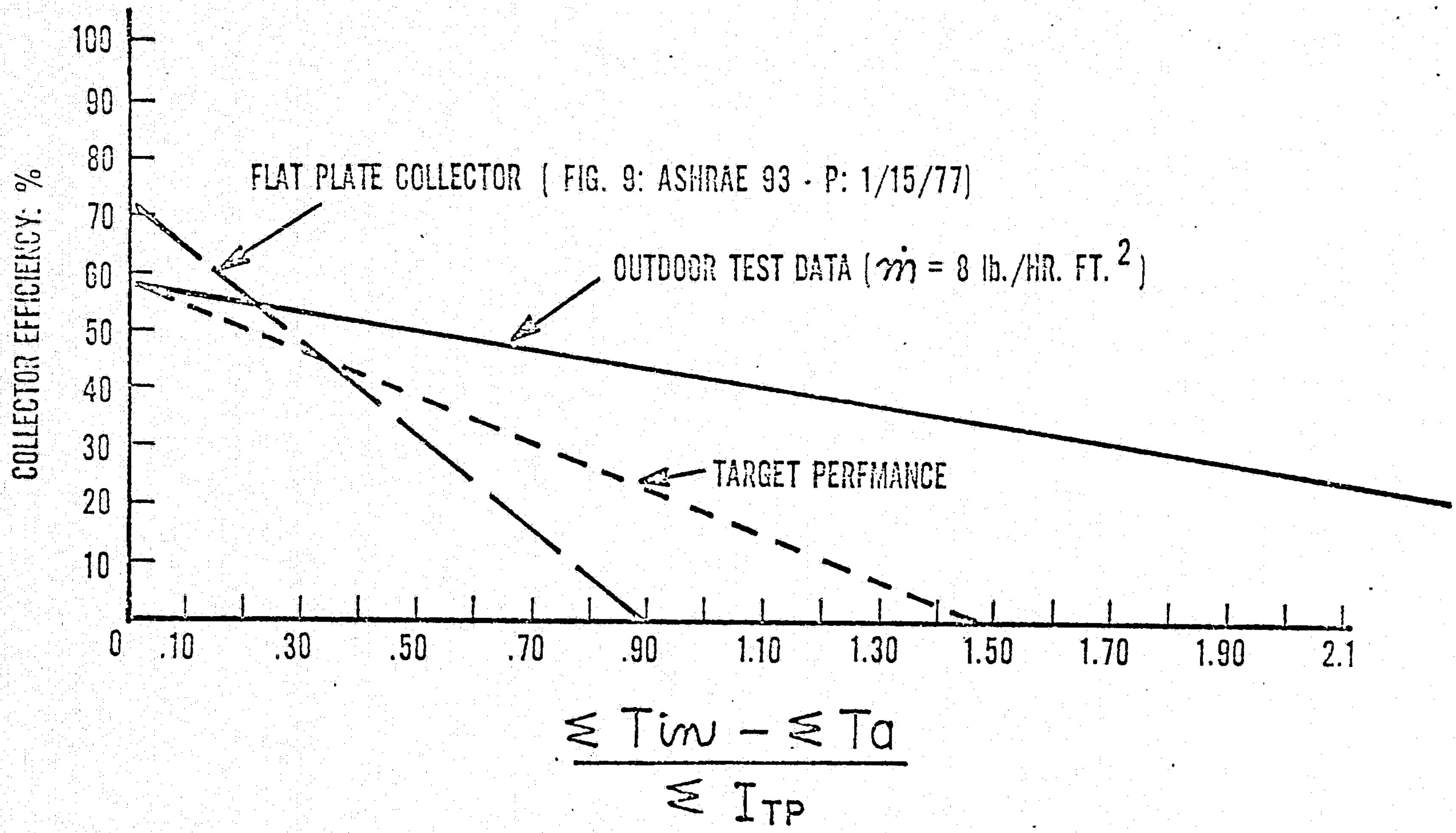


FIGURE 1

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR



C-11

FIGURE 2

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

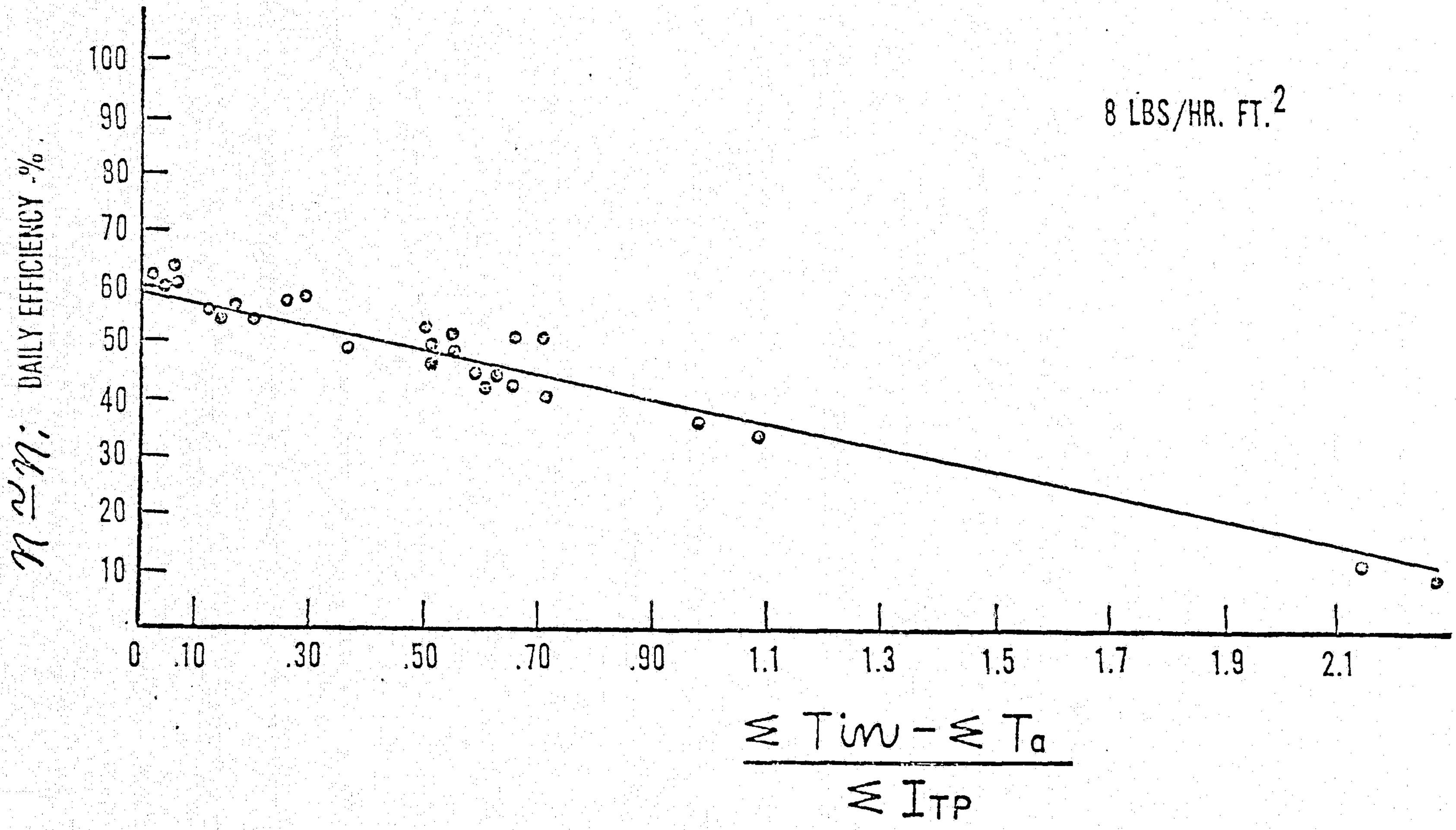


FIGURE 3A

C-12

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AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

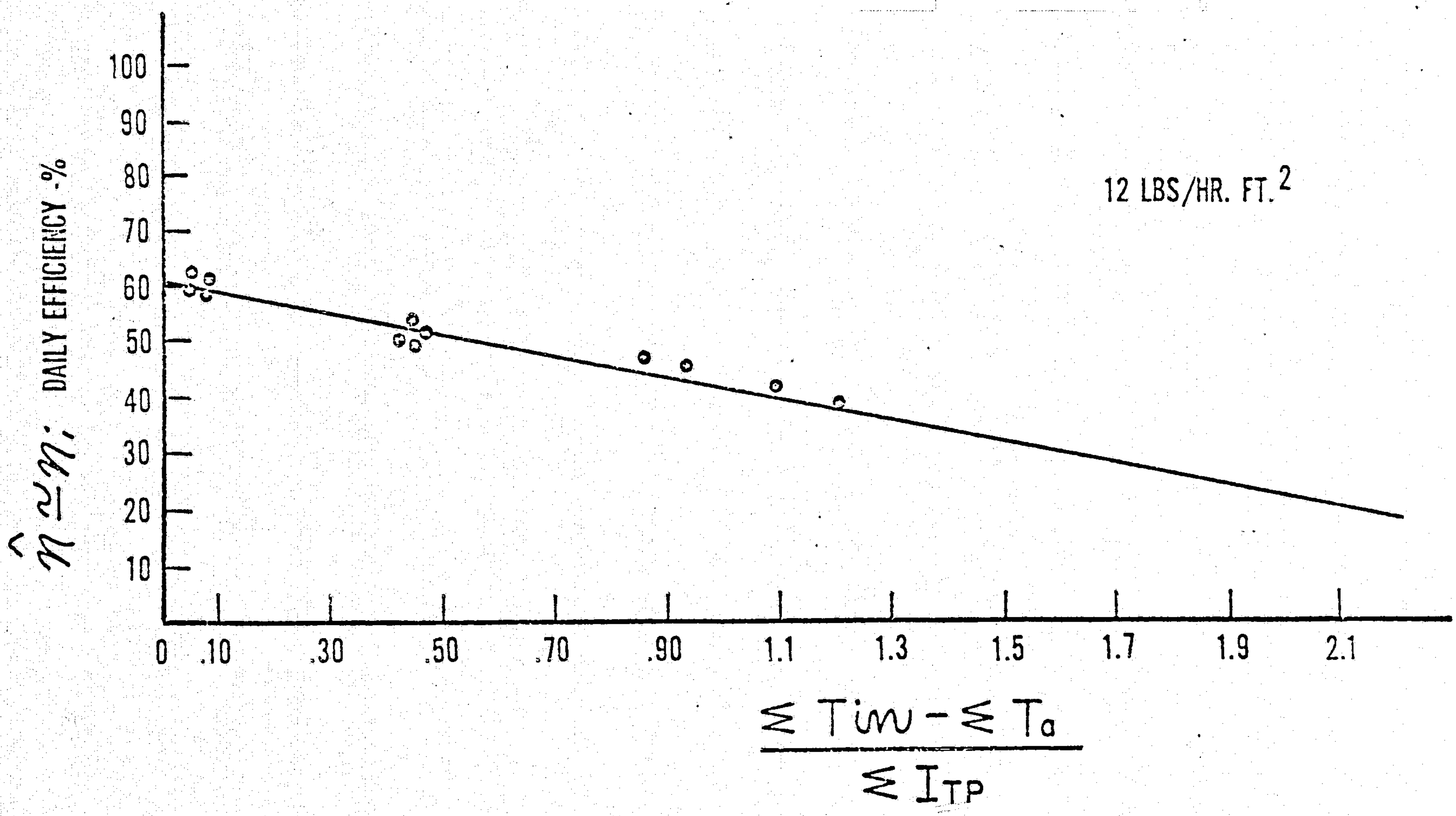
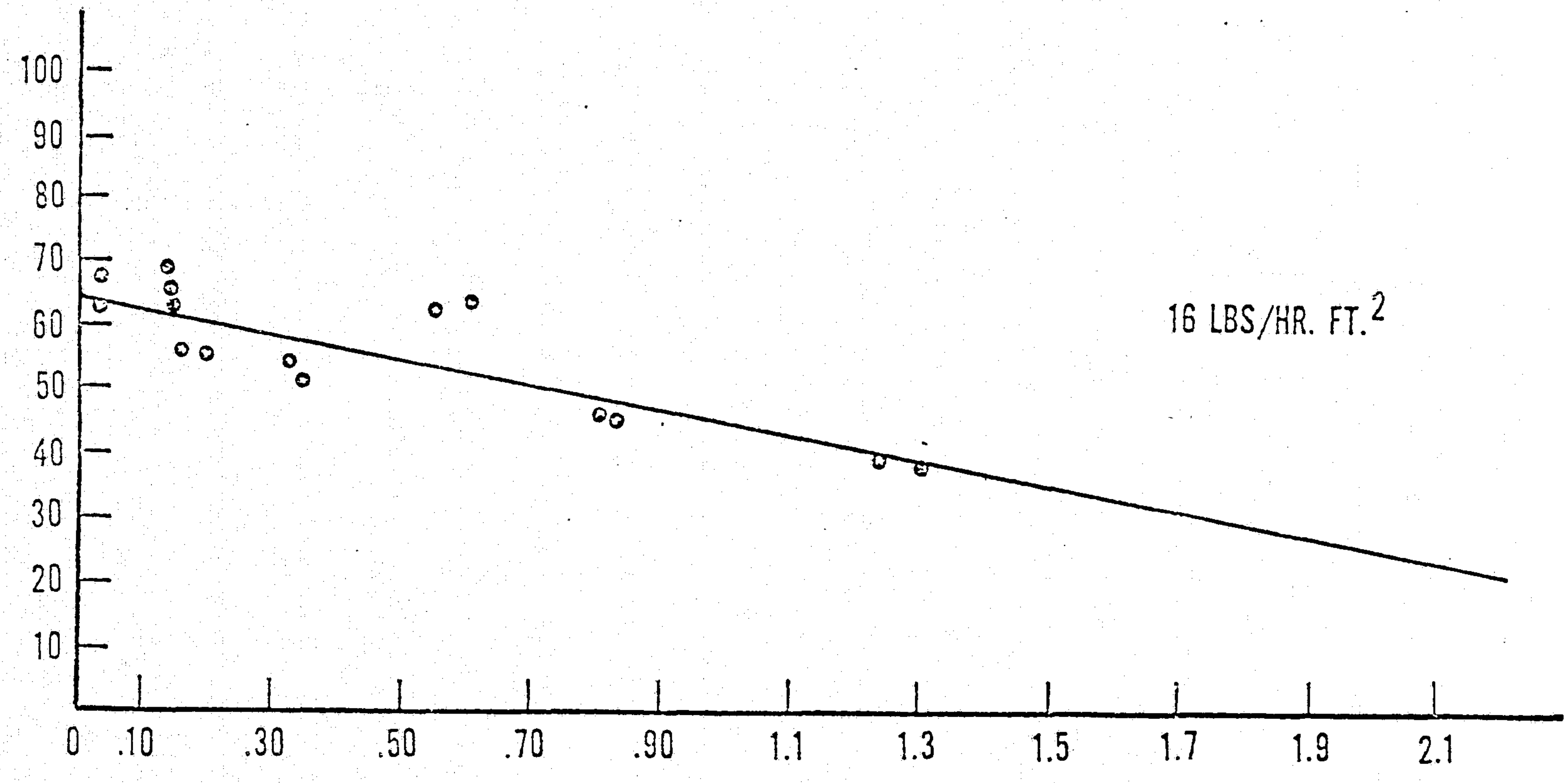


FIGURE 3B

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR



$$\frac{\Sigma T_{in} - \Sigma T_a}{\Sigma I_{TP}}$$

C-14

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FIGURE 3C

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

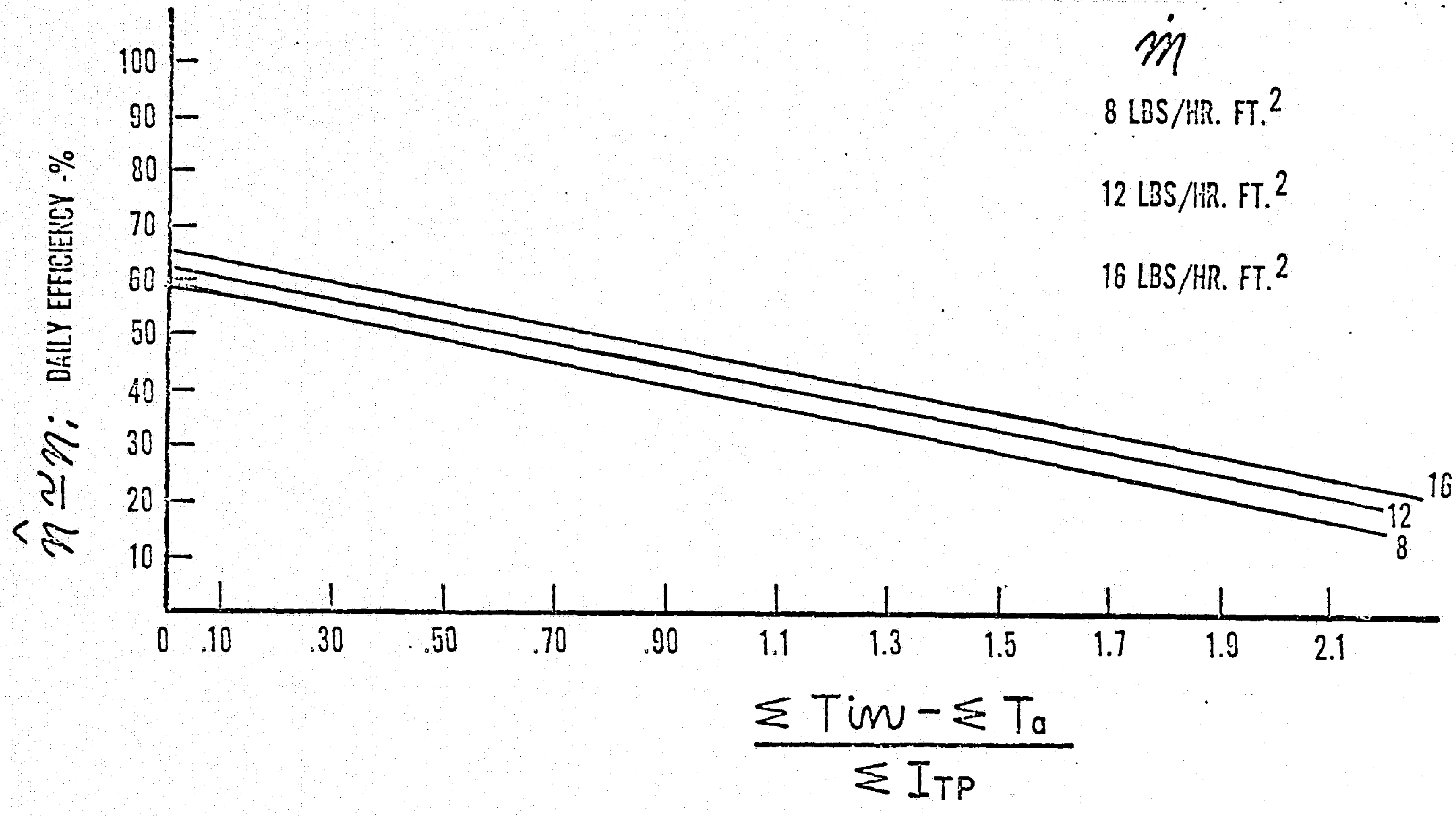

FIGURE 4

Air Driven Spaced Evacuated Tubular Collector

Design $\dot{m}$ = 8 LBS/HR. FT.^{2}

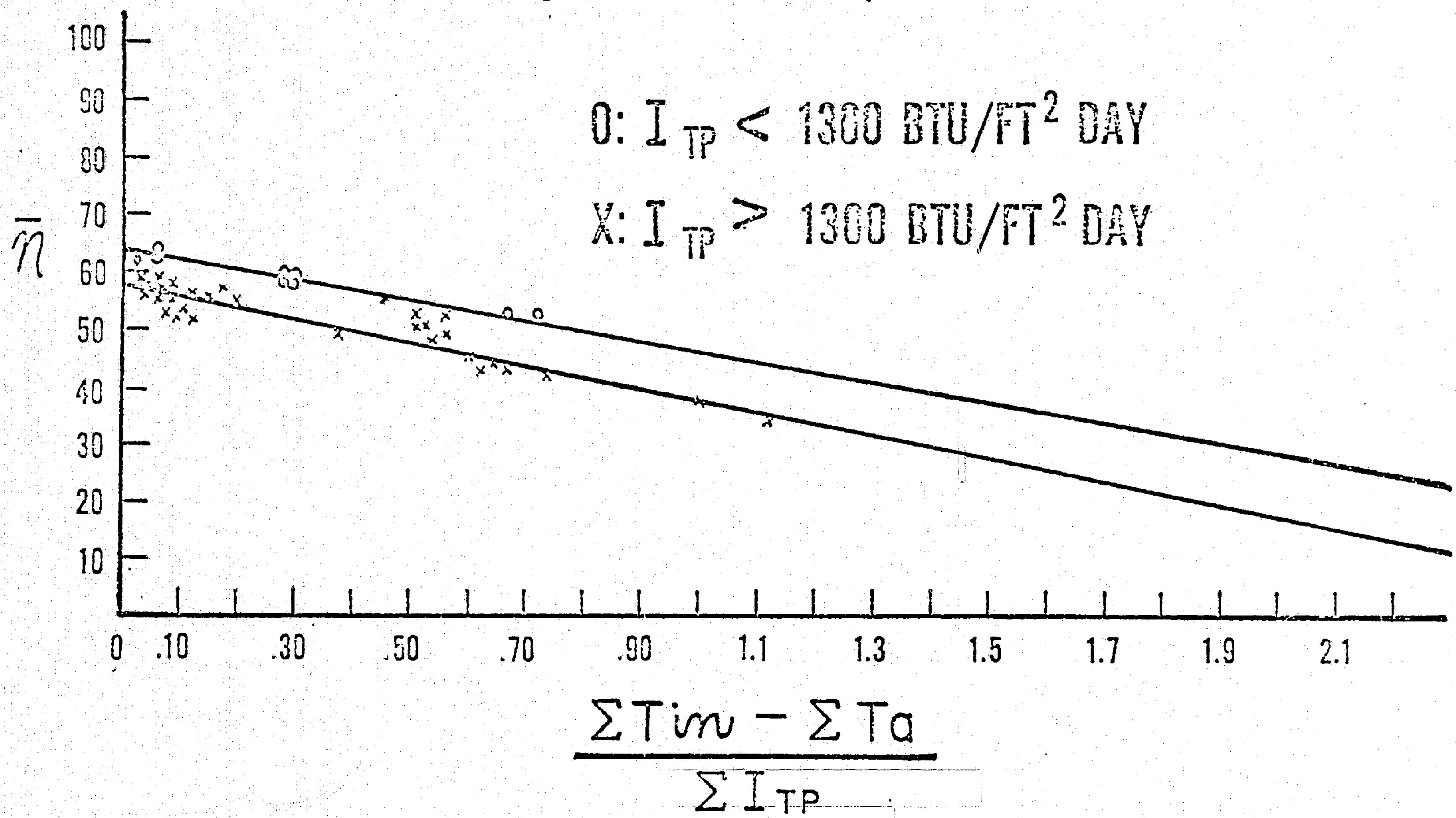

FIGURE 5

$\dot{m}$: LBS/HR.FT.^{2}	η_o	SLOPE	U_L	F_R	ϕ = I.A.M.
8	0.581	-0.178	0.137	0.976	1.10
12	0.624	-0.188	0.144	0.983	1.18
18	0.655	-0.196	0.150	0.987	1.23

$$\bar{\eta} = \frac{A_C}{A_P} F_R \left[\phi \tau \alpha \frac{I_{eff}}{I_{TP}} - \pi U_L \left(\frac{\sum T_{in} - \sum T_A}{\sum I_{TP}} \right) \right]$$

FIGURE 6

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

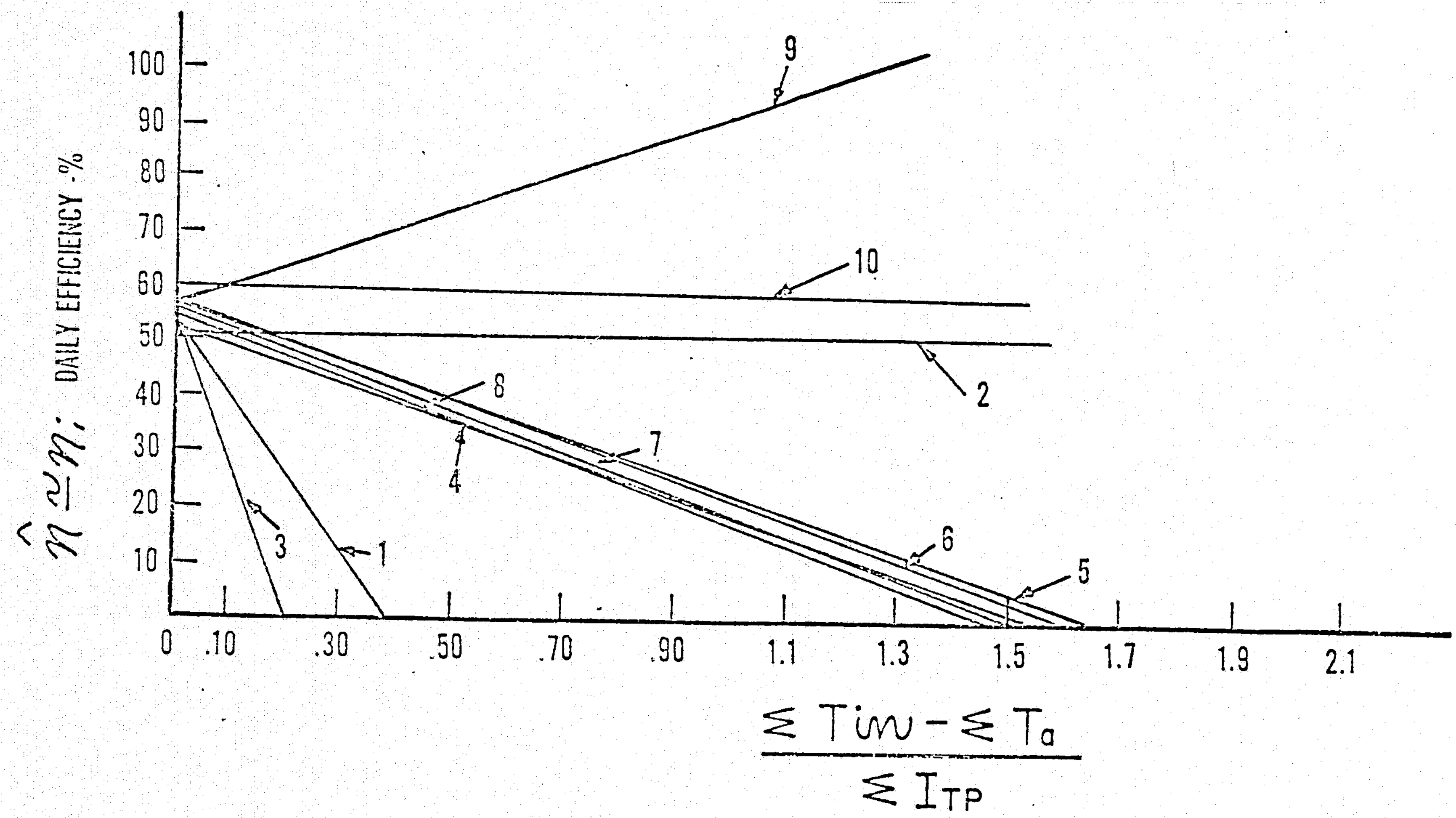

FIGURE 7

Diffuse VS. Clear Day Performance
Flow Rate Equal 8 LBS/HR. FT.^{2}

I_{TP}	η	SLOPE	U_L	F_R	ϕ
< 1300	0.638	-0.175	0.135	0.977	1.21
> 1300	0.576	-0.193	0.149	0.974	1.10
ALL DAYS	0.581	-0.178	0.137	0.976	1.10

FIGURE 8

EFFECT OF INTRODUCING AIR LIQUID HEAT EXCHANGE

$$F_R' = F_R \left[\frac{1}{1 + \frac{A_C F_R U_L}{C_C} \left(\frac{C_C}{E_{HX} \; C_{min}} - 1 \right)} \right]$$

$$F_R' = 0.976 \left[\frac{1}{1 + \frac{70 \times .976 \times .42\pi \times .137}{312} \left(\frac{1}{E_{HX}} - 1 \right)} \right]$$

$$F_R' = .939$$

FIGURE 9