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RESEARCH ON THE APPLICATION OF SOLAR ENERGY TO
INDUSTRIAL DRYING OR DEHYDRATION PROCESSES

Final Phase Report

March 1977

Work Performed Under Contract No. EY-76-C-05-5123

California Polytechnic State University
San Luis Obispo, California
TRW, Systems and Energy
Redondo Beach, California

MASTER

U.S. Department of Energy

Solar Energy

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Research on the Application of Solar Energy
to Industrial Drying or Dehydration Processes

Final Phase Report

March 1977

by

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I. INTRODUCTION

The California Polytechnic State University, San Luis Obispo, in conjunction with TRW and assisted by the Pacific Gas and Electric Company (P.G. & E.), selected food dehydration as an appropriate industrial process for demonstrating the utilization of solar energy.

It is the purpose of this project to establish the viability of solar energy as a reliable and practical source of heat in industrial dehydration processes. The project is divided into three phases; namely, the analysis and design, construction, and evaluation of a complete solar dehydration facility operating under normal commercial conditions.

Phase I of the investigation, dealing with the analysis and design of a solar collector and heat storage system attached to a single tunnel of the Lamanuzzi and Pantaleo (L&P) dehydration facility in Fresno, California, is based partly on the results of a previous study undertaken by the same research team in 1975. This study was funded by a NSF RANN grant (RANN-GI-42944) and established the feasibility of solar-assisted industrial food dehydration facilities. The solar collector unit proposed for construction in Phase II of the present project, is directly based on the design and evaluation of a full-size prototype collector developed

under the auspices of the previous NSF study.

The principal reasons for the selection of solar energy as an alternative and supplementary source of heat for the food dehydration industry are related to the projected termination of natural gas supplies by 1980; the natural affinity between the sun and plant growth; the favorable climatic conditions prevalent in California, where the largest section of the industry is concentrated; and the availability of low cost land in proximity of commercial dehydration facilities. In particular:

1. It has been announced by the California Public Utilities Commission (April, 1975) that the demand for energy by the food dehydration industry cannot be satisfied in terms of natural gas beyond 1979. At that time, it is planned to terminate all commercial and industrial natural gas services as a means of satisfying the increasing demand of residential customers. Energy is the life-line of the food dehydration industry, and if this essential industry is to survive beyond 1980, a means of satisfying its energy demands with non-fossil fuel will have to be found on a high priority basis.
2. The peak energy needs of the dehydration industry generally coincide with periods of high insolation. The food drying

season typically extends from July to November.

3. Commercial dehydration facilities are normally located in rural areas and are typically surrounded by ample land for the accommodation of large solar collector arrays. Typically, a twelve tunnel dehydration facility will require approximately ten acres of solar collection area.
4. A life cycle cost analysis prepared by this research team in 1975 (i.e. NSF RANN-GI-42944) indicated that a solar food dehydration system with a capital cost of \$5.00 per square foot of collector area, amortized over ten years and operating for 200 days per year, could become competitive with an oil converted system if the price of oil rose to \$17.00 per barrel. Investigations conducted under Phase I of the present project appear to generally support these findings, although the conclusions are slightly less optimistic.
5. The solar collector and heat storage system proposed for construction during Phase II of this study is of simple design, not subject to patent restrictions, and could be readily manufactured by existing commercial sheet metal fabricators. With the widespread acceptance of this design, it is to be expected that competitive influences will reduce

the estimated total system costs.

Based on analysis and design studies undertaken in Phase I of this project, a solar dehydration system comprising 22,000 square feet of collector area and 20,000 cubic feet of rock heat storage was submitted for bid to selected contractors in the Fresno locality. Following the receipt of bids, which reflected the general unfamiliarity of the construction industry with solar projects, minor modifications were incorporated in the design. These changes will not, in any way, effect the operation of the solar dehydration system, but are related to a reduction of the solar collector area to 19,500 square feet and the heat storage volume to 12,500 cubic feet.

FIRST PART

- PHASE 1: II. Evaluation of Dehydration Operation
 III. System Design
 IV. Economic Analysis

II. EVALUATION OF DEHYDRATION OPERATION

The Lamanuzzi and Pantaleo (L&P) dehydration company in Fresno, California, utilizes twelve (54 ft. by 7 ft. by 7 ft.) truck and tray or tunnel dehydrators for the drying of raisins.

The tunnel dehydrator is one of the more efficient types of mechanical dehydrators and is best suited for the drying of fruits and vegetables. The raw material is placed on shallow trays (6 ft. by 3 ft.) which are stacked 26 high on cars (Figure II-1). The trays are vertically spaced to permit the circulation of hot air between and over the raw material. A typical tunnel will hold 14 cars, which are pulled through the tunnel on a continuous basis by a drag cable and ratchet. The L&P tunnels are operated in a counterflow manner to facilitate the rate of removal of moisture from the raw material (Figure II-2). Typically, wet raisins will enter the tunnel with a moisture content of 83% and emerge after 24 hours with the moisture content reduced to 14% or less. A tunnel dehydrator will normally dry 6 to 7 tons of raw material per day.

Major fruits dried in tunnel dehydrators include apricots, peaches, nectarines, pears, raisins (as well as, reprocessed sun dried raisins), figs and prunes. Less frequently other products, such as lumber, coffee beans, potatoes, rice, nuts, bell peppers,

garlic, onions and textiles, are dried in tunnel dehydrators. For obvious reasons, garlic and onions are, as a rule, not dried in the same facility as more "delicate" crops. Cotton is perhaps another candidate for solar dehydration facilities, where the absence of open burners would significantly reduce the danger of cotton dust explosions.

L&P dries primarily a golden-bleached raisin. This raisin is obtained from a Thompson seedless grape which has been bleached for about two hours in sulphur dioxide gas prior to entry into the tunnel. The quality of golden-bleached raisins is measured by the USDA, Agricultural Marketing Service, Fruit and Vegetable Division, Processed Products Standardization and Inspection Branch. The raisins are inspected in samples and graded according to established standards of moisture, color, damage, mold, size, and foreign material present. Each sample is evaluated and a grade assigned, such as "U.S. Fancy", "U.S. Choice" or "U.S. Standard".

Air flows were measured in the #12 tunnel of the L&P facility. The fan in this tunnel is a Westinghouse AXIFLO, size 360 AX. The data obtained for static pressure and flow rate varied a great deal from the manufacturer's literature because the fan did not operate at constant speed under varying loads. Under normal

Figure II-1
Discharge End of Tunnel Dehydrator

Figure II-2

Dry (Fd) and Wet (Fw) Temperatures Recorded in the L&P
No. 12 Tunnel on August 28, 1976 (11:00 a.m.-13:00 p.m.)

operating conditions the tunnel was found to require about 22,000 cfm with a static pressure of 1.8 inches of water. Observations appeared to indicate that only approximately one-third of the heated air passes directly over the wet material.

In the normal operation of tunnel dehydrators a portion of the discharged air is returned to the inlet by a process called recirculation (Figure II-2). Based on tests it was found that about 40% of the discharged heat is recirculated to the entrance of the dehydrator.

The heat load was calculated with the aid of a psychrometric chart from data taken at noon and was found to be about 900,000 Btu/hr., (i.e. 260 therms per day) at a recirculation rate of 40%. The ambient temperature and insolation values assumed for the NOAA Climatological Summary, are given below:

Month	Temperature		Horizontal Insolation Mean Daily
	Daily Maximum	Daily Minimum	
June	90.3°F	57.5°F	2606 Btu/sq. ft.
July	98.2	62.9	2565
August	96.0	60.6	2307
September	91.0	56.5	1909
October	79.8	48.6	1404
November	66.1	40.8	947
December	54.6	36.9	593

The energy flow for a tunnel shown in Figure III-7a, indicates that only about 40% of the heated air comes into contact with the produce to be dried. In order to save some of this energy, recirculation is employed (Figure III-7b) to utilize an additional 30% of the total energy provided.

The temperature required for drying varies through the season and will start in late June at about 140°F and increase to a maximum of 150°F in August and September. From November onward, if the L&P facility is not shut down for cleaning, it processes sun dried raisins and uses a drying temperature of about 120°F.

Tests and observations conducted during the majority of the 1976 drying season form the basis of the establishment of the following solar system design criteria:

Design Day	September 21
Design Temperature	150°F to 120°F
Design Load (without air recirculation)	360 x 10 ⁶ Btu/day
Design Flow Rate	10,000 cfm

Figure II-3 shows the operational cash flow distribution of the L&P dehydration facility for the 1975 drying season. The percentages are dependent on the price of the dried product,

which is normally around \$300/ton but may rise above \$1,000/ton at times. The total operating cost for a typical tunnel in 1976 was approximately \$12,500 per year.

Figure II-3

Cash Distribution for L & P Dehydration Facility for 1975

III. SYSTEM DESIGN

A. DESCRIPTION AND OPERATING MODES:

The solar system shown schematically in Figure III-1 is capable of providing approximately 70% of the total heat requirements of one tunnel of the L&P dehydration facility. Temperatures indicated represent average design values corresponding to September operation and weather conditions. The system was designed to supply a fixed design flow rate of fresh air at 150°F to the dryer for 24 hours a day. In all modes of operation the ambient air is first preheated by heat obtained through a rotary wheel type heat recovery unit from the moist dryer exhaust air. This heat recovery unit in effect replaces the normal fossil fuel-fired dehydrator heat recirculation system in which part of the exhaust air is readmitted back into the dryer entrance. Besides increasing the amount of heat recovered, the heat recovery unit was substituted for recirculation in order to avoid contamination of the collector and storage system with moist residue-laden air from the dryer exhaust.

After preheating, the air is further heated in one of the following modes; by passing through the solar collectors, by passing upwards through the previously solar heated rock bed heat storage unit, by standard gas fired burners (or oil fired steam coils) in the dehydration tunnel, or by a combination of these means,

Figure III-1
Schematic of Proposed System

determined by the availability of heat from the collector and rock storage unit at that particular point in time. When there is insufficient solar radiation to heat the air preheated by the heat recovery unit and the storage unit is bypassed, then all of the air is heated to the dryer inlet temperature (i.e. normally 150°F, but less toward the end of the season) by the auxiliary fuel supplied to the burners. However, if any heat is available in the storage unit, all of the air will flow up through the storage unit before entering the tunnel. If the storage is not hot enough to heat the air to the required drying temperature, then the thermostatically controlled burners are automatically activated. The collectors are designed to supply all of the hot air to the dryer whenever they can produce air at a higher temperature than the storage unit. If the storage unit can supply hotter air than the collectors, at the full dryer flow rate, then the flow is divided between the collectors and storage in a ratio to ensure that the two air flows are at equal temperatures. These latter modes of operation ensure that the collectors operate whenever solar energy is available at a higher temperature than the heat available from the storage unit.

During the middle of an average day, the collectors may overheat the air supplied to the dehydrator. If this is the case, the storage fan is operated to increase the flow through the collectors above that required by the dryer, so that the collector exit

temperature is reduced to that required by the dehydrator. The extra flow through the collectors is bypassed downward through the storage and returned to the collectors. In this manner, the heat storage unit is "loaded" for night operation. If the storage unit is filled to capacity it is possible that the storage fan, operating at maximum flow, will be unable to keep the collector exit air temperature down to the required dehydrator inlet temperature. Under these conditions the overheated air flowing into storage at an elevated temperature, together with that portion going to the dehydrator, is diluted with bypass air preheated by the heat recovery unit, so as to ensure that the mixture is delivered to the dehydrator at the required temperature. During a night following this type of daily operation, the overheated air obtained from the overheated storage unit will again be diluted by the preheated bypass air obtained from the heat recovery wheel.

During all of the operating modes, both the fan in the dehydration tunnel and the inlet fan operate continuously at a constant flow rate. These fans only turn off when the tunnel doors are opened to load, unload, or check the produce. In view of the varying system pressure drops which occur during different modes of system operation, the inlet fan flow is kept at a constant flow rate by means of automatically controlled inlet guide vanes.

The amount of flow through the collectors and storage unit in the

above modes of operation is determined by the storage fan damper (i.e. the storage fan is shut off when this damper is closed), and the other system dampers. All of these dampers are automatically activated by motors under the control of a programmed system controller, which senses temperatures at the collector exit, the inlet fan exit, the storage unit exit, and the dehydrator inlet. Detailed descriptions of the system control and operating modes are provided in Section V.

Figure III-2 shows a perspective view of the proposed system layout. There are 32 rows (i.e. since reduced to 30) of single-glazed solar air-collectors. Each row of collectors is 100 ft. long by 4 ft. wide, is tilted at an angle of 36 degrees above the horizontal and faces south. The total net collector area is 20,800 sq. ft. (i.e. since reduced to 19,500 sq. ft.) and occupies approximately 37,000 sq. ft. of ground area. A cross-sectional view of the solar collectors is shown in Figure V-A-1. Section V discusses the collector design and optimization details. The rock storage unit is shown as a 25,000 cubic feet concrete block structure (i.e. since modified to a 12,500 cubic feet silo structure as shown in Figure III-3) containing $1/2$ in. $\pm$ $1/8$ in. (i.e. since increased to $1-1/4$ in.) graded river gravel, 12 ft. deep (i.e. since reduced to 9 ft. in the silo structure). The gravel is supported above a concrete floor by a steel screen to allow vertical air movement (i.e. downward for filling, and upward for emptying).

Figure III-2
Perspective View of Proposed Solar System

INDICATED SOLAR ENERGY FOR INDUSTRIAL DECONTAMINATION PROJECT		TRW TECHNICAL RESEARCH CORPORATION	
TITLE SHEET		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT	
PERSPECTIVE GENERAL CONSTRUCTION NOTES LOCATION MAP		SYSTEM INTEGRATION LABORATORY	
DATE	SPEC. NO.	REV. NO.	6821.24-00
DRAWN BY	DESIGNED BY	APPROVED BY	PROGRAM OFFICE

Figure III-3: Revised System Layout

The storage unit design details are described in Sections V and VI. The duct, which connects the tunnel dehydrator and heat recovery unit to the collector and storage unit is placed underground and has internal insulation. Spiral sheet metal ducting with external sprayed and sealed polyurethane insulation is used above ground.

The inlet fan located near the dehydrator, and the storage fans located at the storage discharge (i.e. far right) are of the backward-curved centrifugal fan type.

B. SYSTEM SIMULATION:

The prediction of system performance and associated operating costs, based on given climatic conditions under normal 24 hour commercial operation, required the development of an elaborate computer model capable of simulating the complex interactions among the various system components. Each component, with the exception of the heat storage unit, was modeled algebraically with time independent equations. Mathematical modeling of the heat storage unit required the simultaneous solution of two time dependent differential equations, utilizing a standard finite difference technique. As shown in Figure III-4, the simulation program determined the system operating conditions during a typical 24 hour cycle, based on the input of environmental conditions at 15 minute intervals.

Figure III-4
System Simulation Computer Program
Block Diagram

The simulation model was used to evaluate the influence on system performance of numerous design and environmental parameters, including the percentage of heat supplied by the solar system.

Life-cycle costs were determined independently of the simulation model, as described in Section IV.

C. PREDICTED PERFORMANCE:

Figure III-5 indicates typical performance characteristics predicted by the simulation program for the nominal "design" day, September 21, at the Fresno installation. A number of system operating characteristics are illustrated. Commencing at about 2 a.m., the storage unit exit temperature begins to drop below the 150°F required by the dehydrator. The burners therefore supply part of the heat requirements from 2 a.m. to about 9 a.m., at which time the collectors are able to supply all of the dehydrator heat and in addition deliver heat to the storage unit. At 11 a.m., the storage fan has reached its maximum capacity, and the temperature of the air leaving the collector exceeds 150°F. The mean storage temperature (i.e. determined by averaging the storage temperature profile at any time) reaches its minimum at 8 a.m., and its maximum at 3:30 p.m. Since the storage unit approaches its maximum capacity at 3:30 p.m., the exit storage temperature (not shown) begins to rise and this accounts for the sharp rise in collector inlet temperature between 2 p.m. and 3:30 p.m.

The difference between the collector and fan inlet temperatures during the night hours is due to the temperature rise of the air passing through the fan. The storage fan yields a similar temperature rise. The outlet temperature of the storage unit rises slightly after sundown due to the regurgitation of heat from the rock that was overheated during the midday period. A plot of storage temperature profiles (i.e. temperature versus depth) is shown in Figure III-6 as a function of time of day, for the same day represented in Figure III-5. The daily average collector efficiency is 34.8% and the peak efficiency of 48% occurs at midday for the day plotted.

Considering the tunnel dehydrator and heat recovery wheel as one and the same "dehydrator unit", the total heat energy that must be supplied by some external means to this unit on a given day is provided by a combination of solar energy, electrical energy introduced by the operation of fans, and energy supplied to the auxiliary burners. This is illustrated in Figure III-7c, which indicates that 47% of the energy input to the system (at x) is supplied by the auxiliary burner and the solar system with its associated electrical requirement. Of this, 78% is solar energy, 12% is electric power and 10% is supplied by the burner. Figure III-7c illustrates the manner in which "percent solar energy" is supplied as defined in this report. Using this definition, Figure III-8 indicates the proportions of energy supplied to the tunnel dehydrator by the sun, electric power, and auxiliary fuel,

Figure III-5:

Hourly System Operating Conditions, on September 21
Predicted for Fresno Operation

19500 ft² Collector
12500 ft³ Rock Storage

during each month of the operating season at the Fresno site. Solar energy alone is expected to contribute about 69% of the total energy needs of the system over a 214 day season which forms the basis of Figure III-8. Similar results for the Albuquerque and Omaha sites are given in Section IV. The same monthly drying temperature schedule was assumed for all locations and the same results were used to evaluate the system life-cycle costs for the three potential system locations. In addition, this type of simulation result provided a basis for the evaluation of the influence on life cycle costs of numerous system parameters, such as insulation thickness, duct size, collector configuration, rock storage volume, etc., as described in Section V.

Figure III-7c corresponds to the system considered under the Plan B economic analysis. Figure III-7b, with air recirculation, corresponds to the system considered under Plan A of the economic analysis, and is the same as that operated by L&P during the past drying season. This system needs to supply only about 70% of the energy that is required for a system with no air recirculation, such as that shown in Figure III-7a. In September, the energy input to the system (b) of the Plan A economic analysis, is approximately 270 MBtu/day or 70% of the 386 MBtu/day required by system (a). This 270 MBtu/day is the rate of energy input that the proposed system (c) must supplant. From an economic viewpoint, to be

Comparison of Energy Flow in Typical Dehydration Tunnel (a) to that Used in Current Operation (b) and the Proposed System (c)

Seasonal Totals; Energy Supplied to System = 3358 MBTU (100%)
 Energy Supplied by Solar = 2314 MBTU (69%)
 Energy Supplied by Electricity = 420 MBTU (12.5%)
 Energy Supplied by Fuel = 625 MBTU (18.5%)

Figure III-8.

Fresno Energy Distribution Schedule

competitive, the cost of system (c) must be less than the cost to supply the 270 MBtu/day by the burner in system (b). Thus the equivalent daily cost of the solar energy system and heat recovery unit (i.e. the only additional purchased items in system (c) not existing in system (b)) and divided by the 270 MBtu/day yields the effective energy cost of system (c). This explains the energy related basis of the cost analyses presented in Sections IV and V.

IV. ECONOMIC ANALYSIS

The purpose of this analysis is to evaluate the economic viability of the solar system proposed for installation and operation in Phases II and III, respectively.

The analysis is divided into two main parts; namely, Part A and Part B. Part A contains the required analysis under the standard conditions prescribed by ERDA, and Part B comprises additional analyses for extended assumptions.

A. LIFE-CYCLE COST ANALYSIS OF PROPOSED SOLAR SYSTEM:

The life-cycle cost calculations are based on: (1). the standard conditions as outlined by ERDA in "Method of Economic Analysis for Comparison of Solar Process Heat Systems" by Dickinson and Shearer; (2). the estimated construction costs provided by the program "ASEID 1" (ref.: Section VI); (3). the calculated heat required from a system to replace the existing system, provided by the system simulation computer program (ref.: Section V); and (4). the method of "Equivalent Uniform Annual Net Disbursements" referenced in Engineering Economy by Grant and Ireson.

The locations used for comparison are Albuquerque, Omaha and Fresno. The format of Plan A, Plan B and Plan C is used; where Plan A is defined as the present system using fossil fuel only; Plan B is

defined as the combined solar and fossil fuel system; and Plan C is defined as the cost of solar energy only. Calculation of the payback period is also included in Part A.

Three computer programs were used to simplify the economic analysis. The first program (ASEID2) was written to calculate the cost of energy for Plans A, B, and C, and to calculate the payback period and the cost of the installed system in terms of dollars per square foot. ASEID2 was implemented on a Hewlett-Packard (H. P.) 9825 programmable calculator and was written in BASIC. The second program (ASEID3) was designed to provide a tabular cash flow for the systems under consideration and verification of the algebraic method of calculating costs as outlined in the ERDA guidelines and implemented in ASEID2. The equipment used for this second program comprised the 9825 H. P. calculator interfaced with a Digital Decwriter II printer. The third program (ASEID4) provided plotting capabilities for the sensitivity analysis. An H. P. 9862A flat-bed plotter was linked to the 9825 H. P. calculator for this analysis. A summary of "Assumed Conditions", as outlined in the ERDA guidelines, is shown in Table IV-A-1.

	Albuquerque	Omaha	Fresno
1. Annual requirement for process hot air, in MBTU	5.21×10^3	5.42×10^3	5.3×10^3
2. Annual contribution from solar facility in MBTU	4.79×10^3	3.82×10^3	4.68×10^3
3. Annual contribution from back up fuel in MBTU	0.42×10^3	1.60×10^3	0.62×10^3
4. Annual Operation			
a) Electrical-Actual Exposure:	\$4,265	\$3,335	\$4,099
b) Maintenance Replacement Insurance @ 1% of first cost:	\$3,444	\$3,444	\$3,444
5. Area of solar collectors: 19,500 sq. ft.			
6. Installed solar system cost: \$17.66/sq.ft.			
(\$17.66/sq. ft. x 19,500 sq. ft. = \$344,435)			

TABLE IV-A-1

Part A - SUMMARY OF ASSUMED CONDITIONS

Cost of fuel oil at site	\$ 15/bbl No Inflation \$ 21/bbl 5% Inflation
Electrical costs	0.03/kw. hr.
Salvage value	Ø
Useful life	20 years
Tax rate	50%
Tax credit	10% of total project cost
After tax rate of return on investment	10%
Annual costs of boiler	\$0.02 per MBtu
Boiler conversion efficiency	70%
TOTAL PROJECT COSTS	\$ 344,435
214 day season (see Table IV-A-4)	
Net collector size	19,500 sq. ft.
Storage size	12,500 cu. ft.

TABLE IV-A-2

ASSUMED CONDITIONS FOR PLANS A, B, AND C

Table IV-A-3 shows the estimated construction cost, based on the interactive computer program ASEID1 developed for this project. The program is described in detail in Section VI. Adjustments for assumed labor charges of \$15 per hour are shown as a reduction in the total estimated cost of the project. This cost includes only the cost of actually building the project, without Phase II or III monitoring equipment that might be required additionally by ERDA.

Component	Description	Cost
1	Site Preparation	\$ 7,500
2	Collector	129,600
3	Collector-support and installation	38,000
4	Storage System	33,800
5	Modification of Facility	31,325
6	Ducting	78,175
7	Fans and Dampers	22,000
8	Electrical	12,000
9	Controls	11,200
		<u>\$363,000</u>
Adjustments for labor charges to \$15/hr.		<u>- 18,565</u>
		344,435

TABLE IV-A-3

ESTIMATED CONSTRUCTION COSTS OF PROJECT BY COMPONENT

The heat required to be replaced by the combined solar and fossil fuel system was calculated using a season length of 214 days. A weighted average was taken of the monthly needs as provided by the simulation program, taking into account a 43% recirculation of heat. Averages were derived for Albuquerque, Omaha, and Fresno. The summary is shown in Table IV-A-4.

MONTH	Number of Days/Month	Heat Load (MBtu per day)		
		Albuquerque	Omaha	Fresno
June	30	20.95	22.01	23.48
July	31	23.12	23.17	24.64
August	31	23.87	23.99	25.52
September	30	26.29	26.99	26.99
October	16	26.80	27.55	26.51
October	15	19.92	20.70	19.86
November	30	24.83	26.43	23.50
December	31	27.73	30.35	25.94
AVERAGE DAILY HEAT LOAD (weighted average) BTU/day		24.33x10 ⁶	25.31x10 ⁶	24.78x10 ⁶

TABLE IV-A-4

HEAT LOAD FOR ALBUQUERQUE, OMAHA AND FRESNO BY MONTH OF OPERATION
(Derived from System Simulation Program)

The results of the economic analysis for Part A are shown in Table IV-A-5. The figures are given for oil assumed to be \$15/bbl and for oil at \$21/bbl (i.e. levelled for 5% inflation net of general inflation).

	\$15/bbl			\$21/bbl		
	Albuquerque	Omaha	Fresno	Albuquerque	Omaha	Fresno
Plan A	\$ 3.895	3.895	3.895	\$ 5.372	5.372	5.372
Plan B	12.659	12.807	12.543	12.778	13.244	12.717
Plan C	15.99	15.06	13.50	15.99	15.06	13.50
B Payback in years	22.026	24.494	22.995	17.603	20.030	18.302

TABLE IV-A-5

RESULTS OF ECONOMIC ANALYSIS - PART A
(in \$/MBtu)

Details for these calculations are included in Table IV-A-6, and follow the format in the guideline.

	\$15/fuel			\$21/fuel		
Plan A	Albuquerque	Omaha	Fresno	Albuquerque	Omaha	Fresno
Total Fuel required:	5207 MBtu	5416 MBtu	5303 MBtu	5207 MBtu	5416 MBtu	5303 MBtu
Cost of fuel:	19,236.28	20,011.11	19,592.07	26,930.79	28,015.55	27,428.90
Cost of Boilers	1,041.32	1,083.27	1,060.58	1,041.32	1,083.27	1,060.58
Plan B						
Repayment to Equity for Solar Facility	40,457.21	40,457.21	40,457.21	40,457.21	40,457.21	40,457.21
10% Investment	- 4,045.72	- 4,045.72	- 4,045.72	- 4,045.72	- 4,045.72	- 4,045.72
Cost of Back Up Fuel	1,557.56	5,906.08	2,308.67	2,180.59	8,268.59	3,232.14
Cost Boilers	1,041.32	1,083.27	1,060.58	1,041.32	1,083.27	1,060.58
Electrical Costs	4,264.93	3,444.55	4,098.72	4,264.93	3,444.55	4,098.72
Solar Maintenance, Replacement & Insurance	3,444.35	3,444.35	3,444.35	3,444.35	3,444.35	3,444.35
Deductable Expenses:						
Fuel Oil	1,557.56	5,906.08	2,308.67	2,180.59	8,268.52	3,232.14
Electricity	4,264.93	3,344.55	4,098.72	4,264.93	3,344.55	4,098.72
Boilers	1,041.32	1,083.27	1,060.58	1,041.32	1,083.27	1,060.58
OMRI	3,444.35	3,444.35	3,444.35	3,444.35	3,444.35	3,444.35
Depreciation	17,221.75	17,221.75	17,221.75	17,221.75	17,221.75	17,221.75
Total	27,529.91	30,990.00	28,134.07	28,152.94	33,352.43	29,057.54
Effect of expenditures for Plan B process heat system on reduction of income taxes	-13,764.96	-15,495.00	-14,067.04	-14,076.47	-16,676.22	-14,528.77
after-tax energy cost	32,954.70	34,684.74	33,256.77	33,266.21	35,865.95	33,718.51
before-tax energy cost						
Per MBtu before tax	12.659	12.807	12.543	12.778	13.244	12.717
Cost per sq. ft.	17.663	17.663	17.663	17.663	17.663	17.663
Payback in years	22.026	24.494	22.995	17.603	20.030	18,302
Plan C: Solar Alone Costs						
Repayment to Equity for Solar Facility	40,457.026	40,457.026	40,457.026	40,457.026	40,457.026	40,457.026
10% Tax Credit	- 4,045.72	- 4,045.72	- 4,045.72	- 4,045.72	- 4,045.72	- 4,045.72
Electrical Costs	4,264.93	2,264.93	4,264.93	4,264.93	4,264.93	4,264.93
Solar Maintenance, Replacement Insurance	3,444.35	3,444.35	3,444.35	3,444.35	3,444.35	3,444.35
Deductable Expenses:						
OMRI	7,709.28	6,778.90	3,444.35	7,709.28	6,778.90	3,444.35
Depreciation	17,221.75	17,221.75	17,221.75	17,221.75	17,221.75	17,221.75
Total:	24,931.03	24,000.65	28,134.07	24,931.03	24,000.65	29,057.54
Effect of expenditures for Plan C on reduction of income tax	-12,465.52	-12,000.32	-14,067.04	-12,465.52	-12,000.32	-14,067.04
after-tax cost	41,627.67	40,790.32	31,572.15	41,627.67	40,790.32	31,572.15
before-tax cost						
Solar Cost Alone per MBtu	15.99	15.06	13.50	15.99	15.06	13.50

TABLE IV-A-6

ANNUAL ENERGY COST CALCULATION

	Albuquerque	Omaha	Fresno
1. Percentage of heat supplied by solar in MBtu/day	92	70	88
2. Percentage of heat supplied by boiler in MBtu/day	8	30	12
3. KWHR of electricity required per day to run the system	664.32	519.4	638.4
4. Length of season in days	214	214	214
5. Collector size in sq. ft.	19,500	19,500	19,500
6. Costs for the total project	\$ 344,435	344,435	344,435
7. Depreciable costs	\$ 344,435	344,435	344,435
8. Depreciation method	Straight Line		
9. Salvage Value of system	\$ 0.00		
10. Useful life of system-years	20 years		
11. Maintenance required per year as percent of first cost	1%		
12. Tax rate	50%		
13. Tax credit rate	10%		
14. Minimum required rate of return on investment	10%		
15. Annual costs of boiler - \$/MBtu	0.20		
16. Boiler conversion efficiency	70%		

TABLE IV-A-7

ADDITIONAL CONDITIONS ASSUMED FOR PART A ANALYSIS

CONCLUSIONS:

It is indicated that solar energy cannot, under the assumed conditions, compete with conventional fossil fuels at their present cost levels. However, as is shown in Part B of this Section, based on a combination of assumptions, solar energy can become competitive with fossil fuels.

B. ANALYSIS OF PROPOSED SOLAR SYSTEM UNDER EXTENDED ASSUMPTIONS:

In addition to the economic analysis for the proposed solar and combined solar-fossil fuel systems, an analysis was performed for a second set of parameters. The major change in assumptions was the existence of a heat recovery unit already installed on site prior to retrofit. This would increase the recovery of heat from approximately 43% to 60%.

A sensitivity analysis was undertaken to gain some understanding of the effects of different variables on the cost of energy. The graphs included in Figures IV-B-3 to IV-B-11 show several variables on a one-at-time basis. For this analysis, the standard set of assumptions was used (Table IV-A-2) and a single variable was allowed to range over an extended interval. In each case, dollars per MBtu is the dependent variable for the curves labeled Plan A, Plan B, and Plan C. Payback period is the dependent variable in the graph accompanying each energy cost calculation.

The following variables were chosen:

<u>FIGURE</u>	<u>VARIABLE</u>	<u>RANGE</u>
IV-B-3	Fuel cost with inflation effects	\$5/BBL to \$75/BBL
IV-B-4	Life of project	5 years to 80 years
IV-B-5	Maintenance of system	0% to 10% of first cost
IV-B-6	Net tax rate	40% to 60%
IV-B-7	Tax credit rate	0% to 30%
IV-B-8	Minimum "attractive" rate of return on investment	0% to 30%
IV-B-9	Electricity costs for operation	\$.01/kwhr to \$.10/kwhr
IV-B-10	Project cost	\$100,000 to \$500,000
IV-B-11	Season for operation of dehydrator	20 days to 360 days

Finally, five scenarios were devised to represent various future projections, from "Very Unattractive" to "Very Attractive" for solar energy . These scenarios were based on the project as it might be implemented in Fresno, under traditional commercial conditions. It is assumed that the proposed solar system can be considered as an incremental investment that provides for fuel savings compared with the present system and allows for the continuation of business should natural gas or fuel oil become unavailable in the local area. The assumed "savings", therefore, appear as positive cash flows in the scenarios. Additional assumptions include the possibility of a varying percentage of debt and equity capital and an advanced method of depreciation as currently used by the dehydrator owner for a

major portion of tunnel equipment, and this is considered to be a reasonable assumption. Construction or mortgage funds might also be available for commercial operators. At the same time, while it is admitted that percentage of debt is a much-discussed topic, debt ratio and leverage are clearly not the main concern here. These scenarios are introduced to illustrate several combinations of variables as a means of providing an insight into costs based on different possible future projections.

1. HEAT RECOVERY INSTEAD OF RECIRCULATION: For this analysis, the Plan A format was followed with the following additional stipulations:

Plan AA: Current operation, with heat recovery unit installed instead of recirculation.

Plan BB: Solar system installed, with cost of project reduced to take account of existing heat recovery unit - 30% of fan cost also deducted.

Plan CC: Solar system cost alone.

The assumed cost of the project for this secondary analysis was \$310,885, a reduction in cost of \$33,550 (i.e. 11%). Assumed conditions and a summary of costs are presented in Tables IV-B-1 and IV-B-2, respectively. The Energy Distribution figures are shown in Figure IV-B-1 and IV-B-2.

2. SENSITIVITY ANALYSIS: The purpose of the sensitivity curves is to provide a graphical representation of single variables and their individual effects as shown in Figures IV-B-3 to IV-B-11.

ASSUMED CONDITIONS:	Albuquerque	Omaha	Fresno
1. Percentage of heat supplied by solar in MBtu/day	74	43	69
2. Percentage of heat supplied by boiler in MBtu/day	13	47	19
3. KWHR of electricity required per day to run the system	465.02	363.58	446.9
4. Length of season in days	214		
5. Collector size in sq. ft.	19,500		
6. Cost for total project	310,885		
7. Depreciable costs	310,885		
8. Depreciation method	Straight Line		
9. Salvage value of system	\$ 0.00		
10. Useful life of system	20 years		
11. Maintenance required per year as percent of first cost	1		
12. Tax rate	50%		
13. Tax credit rate	10%		
14. Minimum required rate of return on investment	10%		
15. Annual cost of boiler - \$/MBtu	0.20		
16. Boiler conversion efficiency	70%		

TABLE IV-B-1
ASSUMED CONDITIONS FOR ALBUQUERQUE, OMAHA, AND FRESNO

	\$15/BBL			\$21/BBL		
	Albuquerque	Omaha	Fresno	Albuquerque	Omaha	Fresno
Plan AA	\$ 3.895/MBtu	\$ 3.895	\$ 3.895	\$ 5.372	\$ 5.372	\$ 5.372
Plan BB	17.525	18.399	17.603	17.711	19.098	17.878
Plan CC Solar Alone	17.525	38.51	24.23	22.65	38.51	24.23
B Payback in years	30.038	36.248	30.867	25.093	31.844	25.962

TABLE IV-B-2

Results of Part B Economic Analysis

Seasonal Totals: Energy Supplied to System = 3339 MBTU (100%)
 Energy Supplied by Solar = 2485 MBTU (74.4%)
 Energy Supplied by Electricity = 437 MBTU (13.1%)
 Energy Supplied by Fuel = 417 MBTU (12.5%)

Figure IV-B-1
 Albuquerque Energy Distribution

Seasonal Totals: Energy Supplied to System = 3381 MBTU (100%)
 Energy Supplied by Solar = 1440 MBTU (42.6%)
 Energy Supplied by Electricity = 342 MBTU (10.1%)
 Energy Supplied by Fuel = 1600 MBTU (47.3%)

Figure IV-B-2
 Omaha Energy Distribution

Figure IV-B-3

Sensitivity Analysis: Fuel Costs of \$5/BBL to \$75/BBL with Inflation Effects

Figure IV-3-4

Sensitivity Analysis: Life of Project 5 Years to 80 Years

MAINTENANCE AS % OF COST

MAINTENANCE AS % OF COST

Figure IV-B-5

Sensitivity Analysis: Maintenance Costs of 0% to 10% of First Cost

Figure IV-B-6

Sensitivity Analysis: Net Tax Rate of 40% to 60%

Figure IV-B-7
Sensitivity Analysis: Tax Credit Rate of 0% to 30%

Figure IV-B-8

Sensitivity Analysis: Minimum "Attractive" Rate of Return = 0% to 30%

ELEC. COST PER KWHR

ELEC. COST PER KWHR

Figure IV-B-9

Sensitivity Analysis: Electricity Costs of \$0.01/kwhr to \$0.10/kwhr

Figure IV-B-10
Sensitivity Analysis: Project Costs of \$100,000 to \$500,000

Figure IV-B-11

Sensitivity Analysis: Length of Operating Season 20 days to 360 days

3. SCENARIOS: The following five analyses were based on sets of parameters that vary from "Very Unattractive" to "Very Attractive" for solar energy. The assumption holds that the system design and the heat required by the process are approximately the same as for Part A. Additional assumptions follow, identified by number with the appropriate scenario:

SCENARIO	1	2	3	4	5
	Very Unattractive	Unattractive	Indifferent	Attractive	Very Attractive
Variable Set:					
1 - Fuel Inflation	0	2	4	6	8
2 - Depreciation Method	S/L	S/L	S/L	DDB	DDB
Tax Credit (%)	10	12	14	16	18
3 - Financing Available (%)	-	20	40	60	80
Interest Rate (%)	-	10	8	6	4
Summary - \$MBtu	9.300	7.561	5.380	2.253	-0.713

TABLE IV-B-3

Variable set (1) consists of the future hypothetical inflation rates for fuel. The range is from 0% to 8% inflation, net of general inflation. This is largely an unknown quantity, but one that affects the desirability of solar systems more than any other set of variables, with the possible exception of government-financed loans. The multiplying factors used to level fuel costs are not applied due to the tabular method of analysis, as shown in the following cash flow tables.

Variable set (2) consists of two factors which are under control of the federal government. Varying depreciation methods and tax credit percentages allowed are two variables which can be used to support solar installations or any other project currently in favor. Variable set (3) consists of two assumptions related to the availability of financing. The last two columns for Scenario 4 and 5 assume that government-guaranteed low-interest loans might be available. The percentages used are hypothetical only.

The Cash Flow Tables and a printout of the assumptions and costs are shown in Tables IV-B-4 through IV-B-8. It can be seen that Scenario 5 is so favorable that it actually pays the operator to own a solar system. Obviously, a combination of favorable actions by the government and financing sources would make solar energy more attractive to potential solar users, just as natural gas has been made attractive in the past.

A7U+B

END OF YEAR	C.F.B.T.	DEPRECIATION	INT.EXP.	TAXABLE INC.	TAXES	CREDIT&PMT.	A.T.C.F.	PRES.WORTH	SUM-P.W.
0.00	-350000.00	0.00	0.00	0.00	0.00	35000.00	-315000.00	-315000.00	-315000.00
1.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	12500.00	-302500.00
2.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	11363.64	-291136.36
3.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	10330.58	-280805.79
4.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	9391.44	-271414.35
5.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	8537.67	-262876.68
6.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	7761.52	-255115.17
7.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	7055.92	-248059.24
8.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	6414.48	-241644.76
9.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	5831.34	-235813.42
10.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	5301.22	-230512.20
11.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	4819.29	-225692.91
12.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	4381.17	-221311.74
13.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	3982.89	-217328.85
14.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	3620.80	-213708.05
15.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	3291.64	-210416.41
16.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	2992.40	-207424.01
17.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	2720.36	-204703.64
18.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	2473.06	-202230.58
19.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	2248.23	-199982.35
20.00	10000.00	-17500.00	0.00	-7500.00	3750.00	0.00	13750.00	2043.85	-197938.50

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 CASH FLOW TABLE
 FOR
 HSEID PROJECT

 Annual realt for
 heat-Mbtu?
 5000
 Annual contrib.
 from solar-Mbtu?
 4500
 Total Project Co
 sts?
 350000
 Depreciable cost
 s?
 350000
 Salvage value?
 0
 Amt. to be finan
 ced? (as %)
 0
 Project life?
 20
 Mortgage interes
 t rate?
 0
 Depreciation met
 hod? S/L=1;DOB=2
 ?
 1
 Min. attractive
 rate of return?
 10
 Net cash savings
 before taxes?
 10000
 Tax Rate?
 50
 Tax credit rate?
 10
 Inflation rate o
 n savings only?
 0
 % DEBT CAPITAL
 0
 CASH SAVINGS OF
 10000.00
 ANNUAL NORTH
 AFTER TAXES
 -23249.78
 PRESENT NORTH
 AFTER TAXES
 -197938.50
 ANNUAL NORTH
 BEFORE TAXES
 -46499.56
 \$/Mbtu
 9.300

Table IV-B-4

Scenario 1: Very Unattractive for Solar

ATU+B

END OF YEAR	C.F.B.T.	DEPRECIATION	INT.EXP.	TAXABLE INC.	TAXES	CREDIT&PMT.	A.T.C.F.	PRES.WORTH	SUM-P.W.
0.00	-280000.00	0.00	0.00	0.00	0.00	42000.00	-238000.00	-238000.00	-238000.00
1.00	10000.00	-17500.00	-7000.00	-14500.00	7250.00	-8222.17	9027.83	8207.11	-229792.89
2.00	10200.00	-17500.00	-6877.78	-14177.78	7088.89	-8222.17	9066.72	7493.16	-222299.73
3.00	10404.00	-17500.00	-6743.34	-13839.34	6919.67	-8222.17	9101.50	6838.09	-215461.64
4.00	10612.08	-17500.00	-6595.46	-13483.38	6741.69	-8222.17	9131.60	6237.00	-209224.64
5.00	10824.32	-17500.00	-6432.79	-13108.47	6554.23	-8222.17	9156.38	5685.39	-203539.24
6.00	11040.81	-17500.00	-6253.85	-12713.04	6356.52	-8222.17	9175.16	5179.14	-198360.11
7.00	11261.62	-17500.00	-6057.02	-12295.39	6147.70	-8222.17	9187.15	4714.46	-193645.65
8.00	11486.86	-17500.00	-5840.50	-11853.65	5926.82	-8222.17	9191.51	4287.91	-189357.74
9.00	11716.59	-17500.00	-5602.34	-11385.74	5692.87	-8222.17	9187.29	3896.31	-185461.43
10.00	11950.93	-17500.00	-5340.35	-10889.43	5444.71	-8222.17	9173.47	3536.77	-181924.67
11.00	12189.94	-17500.00	-5052.17	-10362.23	5181.11	-8222.17	9148.88	3206.63	-178718.04
12.00	12433.74	-17500.00	-4735.17	-9801.43	4900.71	-8222.17	9112.28	2903.45	-175814.59
13.00	12682.42	-17500.00	-4386.47	-9204.05	4602.03	-8222.17	9062.27	2625.02	-173189.57
14.00	12936.07	-17500.00	-4002.90	-8566.83	4283.42	-8222.17	8997.31	2369.27	-170820.30
15.00	13194.79	-17500.00	-3580.97	-7886.18	3943.09	-8222.17	8915.71	2134.35	-168685.95
16.00	13458.68	-17500.00	-3116.85	-7158.17	3579.08	-8222.17	8815.59	1918.53	-166767.42
17.00	13727.86	-17500.00	-2606.32	-6378.46	3189.23	-8222.17	8694.91	1720.24	-165047.17
18.00	14002.41	-17500.00	-2044.73	-5542.32	2771.16	-8222.17	8551.40	1538.04	-163509.13
19.00	14282.46	-17500.00	-1426.99	-4644.53	2322.26	-8222.17	8382.55	1370.61	-162138.52
20.00	14568.11	-17500.00	-747.47	-3679.36	1839.68	-8222.17	8185.62	1216.74	-160921.78

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 CASH FLOW TABLE
 FOR
 ASEID PROJECT

 Annual req't for
 heat-MBtu?
 5000
 Annual contrib.
 from solar MBtu?
 4500
 Total Project Costs?
 350000
 Depreciable cost
 \$?
 350000
 Salvage value?
 0
 Amt. to be financed?
 (as %)
 20
 Project life?
 20
 Mortgage interest
 rate?
 10
 Depreciation method?
 S/L=1:008=2
 1
 Min. attractive
 rate of return?
 10
 Net cash savings
 before taxes?
 10000
 Tax Rate?
 50
 Tax credit rate?
 12
 Inflation rate on
 savings only?
 2
 % DEBT CAPITAL
 20
 CASH SAVINGS OF
 10000.00
 ANNUAL WORTH
 AFTER TAXES
 -18901.81
 PRESENT WORTH
 AFTER TAXES
 -160921.78
 ANNUAL WORTH
 BEFORE TAXES
 -37803.62
 \$/MBtu
 7.561

Table IV-B-5

Scenario 2: Unattractive for Solar

ATU+B

END OF YEAR	C.F.B.T.	DEPRECIATION	INT.EXP.	TAXABLE INC.	TAXES	CREDIT&PMT.	A.T.C.F.	PRES.WORTH	SUM-P.W.
0.00	-210000.00	0.00	0.00	0.00	0.00	49000.00	-161000.00	-161000.00	-161000.00
1.00	10000.00	-17500.00	-11200.00	-18700.00	9350.00	-14259.31	5090.69	4627.90	-156372.10
2.00	10400.00	-17500.00	-10955.26	-18055.26	9027.63	-14259.31	5168.32	4271.34	-152100.76
3.00	10816.00	-17500.00	-10690.93	-17374.93	8687.47	-14259.31	5244.16	3940.01	-148160.75
4.00	11248.64	-17500.00	-10405.46	-16656.82	8328.41	-14259.31	5317.74	3632.09	-144528.66
5.00	11698.59	-17500.00	-10097.15	-15898.57	7949.28	-14259.31	5388.56	3345.87	-141182.79
6.00	12166.53	-17500.00	-9764.18	-15097.65	7548.83	-14259.31	5456.05	3079.80	-138102.99
7.00	12653.19	-17500.00	-9404.57	-14251.38	7125.69	-14259.31	5519.57	2832.41	-135270.58
8.00	13159.32	-17500.00	-9016.19	-13356.87	6678.44	-14259.31	5578.45	2602.39	-132668.20
9.00	13685.69	-17500.00	-8596.74	-12411.05	6205.53	-14259.31	5631.91	2388.48	-130279.72
10.00	14233.12	-17500.00	-8143.74	-11410.62	5705.31	-14259.31	5679.12	2189.55	-128090.17
11.00	14802.44	-17500.00	-7654.49	-10352.05	5176.02	-14259.31	5719.16	2004.53	-126085.64
12.00	15394.54	-17500.00	-7126.10	-9231.56	4615.78	-14259.31	5751.01	1832.45	-124253.19
13.00	16010.32	-17500.00	-6555.45	-8045.13	4022.56	-14259.31	5773.58	1672.40	-122580.79
14.00	16650.74	-17500.00	-5939.14	-6788.40	3394.20	-14259.31	5785.63	1523.54	-121057.26
15.00	17316.76	-17500.00	-5273.53	-5456.76	2728.38	-14259.31	5785.84	1385.08	-119672.17
16.00	18009.44	-17500.00	-4554.66	-4045.23	2022.61	-14259.31	5772.74	1256.32	-118415.86
17.00	18729.81	-17500.00	-3778.29	-2548.48	1274.24	-14259.31	5744.74	1136.57	-117279.29
18.00	19479.00	-17500.00	-2939.81	-960.80	480.40	-14259.31	5700.10	1025.21	-116254.08
19.00	20258.17	-17500.00	-2034.25	723.92	-361.96	-14259.31	5636.90	921.68	-115332.40
20.00	21068.49	-17500.00	-1056.25	2512.25	-1256.12	-14259.31	5553.06	825.43	-114506.97

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 CASH FLOW TABLE
 FOR
 ASIED PROJECT

 Annual req't for
 heat-Mbtu?
 5000
 Annual contrib.
 from solar-Mbtu?
 4500
 Total Project Co
 st?
 350000
 Depreciable cost
 350000
 Salvage value?
 0
 Hnt. to be finan
 ced? (as %)
 40
 Project life?
 20
 Mortgage interes
 t rate?
 8
 Depreciation met
 hod? S/L=1;DOB=2
 ?
 1
 Min. attractive
 rate of return?
 10
 Net cash savings
 before taxes?
 10000
 Tax Rate?
 30
 Tax credit rate?
 14
 Inflation rate o
 n savings only?
 4
 % DEBT CAPITAL
 40
 CASH SAVINGS OF
 10000.00
 ANNUAL NORTH
 AFTER TAXES
 -13449.95
 PRESENT NORTH
 AFTER TAXES
 -114506.97
 ANNUAL NORTH
 BEFORE TAXES
 -26399.89
 \$/Mbtu
 5.380

Table IV-B-6

Scenario 3: Indifferent

A7U+B

END OF YEAR	C.F.B.T.	DEPRECIATION	INT.EXP.	TAXABLE INC.	TAXES	CREDIT&PMT.	A.T.C.F.	PRES.WORTH	SUM-P.W.
0.00	-140000.00	0.00	0.00	0.00	0.00	56000.00	-84000.00	-84000.00	-84000.00
1.00	10000.00	-35000.00	-12600.00	-37600.00	18800.00	-18308.76	10491.24	9537.49	-74462.51
2.00	10600.00	-31500.00	-12257.47	-33157.47	16578.74	-18308.76	8869.98	7330.56	-67131.94
3.00	11236.00	-28350.00	-11894.40	-29008.40	14504.20	-18308.76	7431.44	5583.35	-61548.59
4.00	11910.16	-25515.00	-11509.54	-25114.38	12557.19	-18308.76	6158.59	4206.40	-57342.19
5.00	12624.77	-22963.50	-11101.58	-21440.31	10720.16	-18308.76	5036.17	3127.06	-54215.13
6.00	13382.26	-20667.15	-10669.15	-17954.05	8977.02	-18308.76	4050.52	2286.41	-51928.71
7.00	14185.19	-18600.44	-10210.78	-14626.02	7313.01	-18308.76	3199.44	1636.69	-50292.02
8.00	15036.30	-16740.39	-9724.90	-11428.99	5714.49	-18308.76	2442.04	1139.23	-49152.79
9.00	15938.48	-15066.35	-9209.87	-8337.74	4168.87	-18308.76	1798.59	762.78	-48390.02
10.00	16894.79	-13559.72	-8663.93	-5328.86	2664.43	-18308.76	1250.46	482.11	-47907.91
11.00	17908.48	-12203.75	-8085.24	-2380.51	1190.26	-18308.76	789.98	276.88	-47631.03
12.00	18982.99	-10983.37	-7471.83	527.78	-263.89	-18308.76	410.34	130.75	-47500.28
13.00	20121.96	-9885.03	-6821.62	3415.31	-1707.66	-18308.76	105.55	30.57	-47469.71
14.00	21329.28	-8896.53	-6132.39	6300.36	-3150.18	-18308.76	-129.66	-34.14	-47503.85
15.00	22609.04	-8006.88	-5401.81	9200.36	-4600.18	-18308.76	-299.90	-71.79	-47575.64
16.00	23965.58	-7206.19	-4627.39	12132.00	-6066.00	-18308.76	-409.18	-89.05	-47664.69
17.00	25403.52	-6485.57	-3806.51	15111.44	-7555.72	-18308.76	-460.96	-91.20	-47755.89
18.00	26927.73	-5837.01	-2936.37	18154.34	-9077.17	-18308.76	-458.20	-82.41	-47838.30
19.00	28543.39	-5253.31	-2014.03	21276.05	-10638.03	-18308.76	-403.39	-65.96	-47904.26
20.00	30256.00	-4727.98	-1036.34	24491.67	-12245.83	-18308.76	-298.60	-44.38	-47948.64

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 CASH FLOW TABLE
 FOR
 ASIED PROJECT

 Annual req't for
 heat-Mbtu?
 5000
 Annual contrib.
 from solar-Mbtu?
 4500
 Total Project Co
 sts?
 350000
 Depreciable cost
 350000
 Salvage value?
 0
 Amt. to be finan
 ced? (as %)
 60
 Project life?
 20
 Mortgage interes
 t rate?
 6
 Depreciation met
 hod? S/L=1;DOB=2
 2
 Min. attractive
 rate of return?
 10
 Net cash savings
 before taxes?
 10000
 Tax Rate?
 50
 Tax credit rate?
 16
 Inflation rate o
 n savings only?
 6
 % DEBT CAPITAL
 60
 CASH SAVINGS OF
 10000.00
 ANNUAL NORTH
 AFTER TAXES
 -5632.03
 PRESENT NORTH
 AFTER TAXES
 -47948.64
 ANNUAL NORTH
 BEFORE
 TAXES
 -11264.06
 \$/Mbtu
 2.253

Table IV-B-7

Scenario 4: Attractive

END OF YEAR	C.F.B.T.	DEPRECIATION	INT.EXP.	TAXABLE INC.	TAXES	CREDIT&PMT.	A.T.C.F.	PRES.WORTH	SUM-P.W.
0.00	-70000.00	0.00	0.00	0.00	0.00	63000.00	-7000.00	-7000.00	-7000.00
1.00	10000.00	-35000.00	-11200.00	-36200.00	18100.00	-20602.89	7497.11	6815.55	-184.45
2.00	10800.00	-31500.00	-10823.88	-31523.88	15761.94	-20602.89	5959.05	4924.84	4740.39
3.00	11664.00	-28350.00	-10432.72	-27118.72	13559.36	-20602.89	4620.47	3471.43	8211.82
4.00	12597.12	-25515.00	-10025.92	-22943.80	11471.90	-20602.89	3466.13	2367.41	10579.23
5.00	13604.89	-22963.50	-9602.84	-18961.45	9480.72	-20602.89	2482.72	1541.58	12120.81
6.00	14693.28	-20667.15	-9162.84	-15136.71	7568.35	-20602.89	1658.74	936.32	13057.13
7.00	15868.74	-18600.44	-8705.23	-11436.93	5718.46	-20602.89	984.32	505.11	13562.24
8.00	17138.24	-16740.39	-8229.33	-7831.48	3915.74	-20602.89	451.09	210.44	13772.67
9.00	18509.30	-15066.35	-7734.39	-4291.44	2145.72	-20602.89	52.13	22.11	13794.78
10.00	19990.05	-13559.72	-7219.65	-789.32	394.66	-20602.89	-218.19	-84.12	13710.66
11.00	21589.25	-12203.75	-6684.32	2701.19	-1350.59	-20602.89	-364.23	-127.66	13583.00
12.00	23316.39	-10983.37	-6127.57	6205.45	-3102.72	-20602.89	-389.22	-124.02	13458.98
13.00	25181.70	-9835.03	-5548.56	9748.11	-4874.05	-20602.89	-295.24	-85.52	13373.46
14.00	27196.24	-8896.53	-4946.39	13353.32	-6676.66	-20602.89	-83.31	-21.94	13351.52
15.00	29371.94	-8006.88	-4320.13	17044.93	-8522.47	-20602.89	246.58	59.03	13410.55
16.00	31721.69	-7206.19	-3668.82	20846.69	-10423.34	-20602.89	695.46	151.35	13561.90
17.00	34259.43	-6485.57	-2991.45	24782.40	-12391.20	-20602.89	1265.34	250.34	13812.24
18.00	37000.18	-5837.01	-2287.00	28876.17	-14438.09	-20602.89	1959.20	352.38	14164.62
19.00	39960.19	-5253.31	-1554.36	33152.52	-16576.26	-20602.89	2781.04	454.72	14619.35
20.00	43157.01	-4727.98	-792.42	37636.61	-18818.31	-20602.89	3735.82	555.31	15174.65

 CASH FLOW TABLE
 FOR
 ASSED PROJECT

 Annual req't for heat-MBtu? 5000
 Annual contrib. from solar-MBtu? 4500
 Total Project Costs? 350000
 Depreciable cost 350000
 Salvage value? 0
 Amt. to be financed? (as %)
 Project life? 20
 Mortgage interest rate? 8
 Depreciation method? S/L=1;DOB=2
 Min. attractive rate of return? 10
 Net cash savings before taxes? 10000
 Tax Rate? 50
 Tax credit rate? 18
 Inflation rate on savings only? 8
 % DEBT CAPITAL 80
 CASH SAVINGS OF 10000.00
 ANNUAL MORTGAGE AFTER TAXES 1782.41
 PRESENT WORTH AFTER TAXES 15174.65
 ANNUAL MORTGAGE BEFORE TAXES 3564.82
 \$/MBtu -0.713

Table IV-B-8

Scenario 5: Very Attractive

SECOND PART

PHASE 1: V. System Analysis

VI. Construction Costs

V. SYSTEM ANALYSIS

A. SOLAR COLLECTORS:

1. DESIGN BACKGROUND: Previous investigations made by the University under a 1974 NSF funded (i.e. grant ERT-7419063) Solar Energy Application to Dehydration (SEAD) program indicated that flat plate solar collectors would be more cost effective than low concentration ratio flat plate reflector augmented configurations (1). This was in concurrence with the state-of-the-art as perceived at that time. In addition, these studies concluded that solar systems utilizing air heat collectors would be more cost effective than those utilizing water heating collectors. The relative merits of water and air systems are discussed in detail in Attachment V-A-2.

Under the SEAD study a number of different types of air collectors, for which performance data were available (2), were investigated. The terms of the NSF grant restricted the SEAD investigation to relatively state-of-the-art collectors of proven life and performance. Under these conditions, it was concluded that flat-black painted metal-absorber collectors in which air passes under the absorber surface were most suitable. Although more exotic flat plate collectors, such as those employing selective surfaces, honeycombs, and porous absorbers, etc., were analysed and found to be potentially of higher performance, it was felt that they required considerably more research and development effort for immediate application to large scale optimized systems.

To verify the performance predictions for the type of flat plate collector investigated in the SEAD program, test collectors were fabricated and tested. The performance prediction methods used are described in Attachment V-A-1, where a cross-section of the test collectors is shown in Figure 2. The collectors were fabricated from commercial steel decking connected end-to-end to make 70 ft. and 140 ft. long collector units, in anticipation of application to large scale systems utilizing long collector modules with attendant reductions in manifold costs. These collectors used decking of fixed passage depth (i.e. 7.6 cm. or 3 in.). Two collector lengths were fabricated in order to be able to test this configuration at two different air passage Reynolds numbers for a fixed air temperature rise. In short, since the ratio of collector length to air channel depth determines the heat transfer rate between the solar absorbing surface and the air flowing in the air passages, an increase in the length to depth ratio yields higher heat transfer rates resulting in a relatively cooler absorbing surface and concomitant higher collector efficiency. However, pressure drop also increases and is accompanied by an increase in fan operating costs. Conversely, decreasing the length to depth ratio reduces fan operating costs, but results in inefficient collectors and increased capital costs for the supply of a given amount of heated air at the required temperature. Thus, the existence of an optimum channel depth can be expected. The results of testing the two collector lengths are shown in Figure 3

of Attachment V-A-1 and are in fair agreement with theoretically predicted curves. Likewise, the results verify the expected dependence of efficiency on the length to depth ratio. The predictions were based on standard collector analysis procedures as outlined in Attachment V-A-1, and were considered to provide a sound basis for the collector system optimization analysis of any systems utilizing this type of collector.

2. COLLECTOR DESIGN: The type of collector studied under the SEAD program was not expected to be competitive with predicted oil prices without substantial reductions in material and fabrication costs. The metal decking used for the solar absorbing surface and air passages constitutes a major cost item. Although the metal webbing between the top and bottom of the air passages provided by the decking acts like fins and augments the heat transfer between the absorber and air flow, life-cycle cost studies, indicated that the webbing and the bottom of the metal air passage were not economically justified. In other words, the cost of the metal was found to be more than savings due to the increased efficiency (and concomitant reduction in required collector area) of the collectors.

The proposed collector design cross-section is shown in Figure V-A-1. In these collectors, only the solar absorber side of the air passage is metal, and of reduced thickness compared to the SEAD

Figure V-A-1

Cross Section of Proposed Solar Collector Design

collectors. The bottom metal skin (i.e. below the insulation), and the small (i.e. 1 in.) steel angle glazing supports are designed to resist torsional forces on the sides of the collector due to the dishing of the absorber surface when the air passages are under pressure. The bottom metal skin is so thin that it is cheaper than a comparable plywood sheet. The middle plywood member takes the compression load resulting from the pressurization of the collector. The triple 2 in. by 4 in. redwood sides of the collector are placed under compression by bolts at 18 in. centers.

Similar to the SEAD collector, a single fiberglass reinforced polyester glazing (with Tedlar UV protection surface) supported by the 1 in. steel angles and collector sides, is used. Although of marginal cost advantage, relative to glass, this glazing is expected to be extremely simple to install, and is likely to be much more resistant to vandal action, a major concern in this project.

The 3 - 9/16 in. air channel depth, shown, has been determined to be near optimum with respect to the heat transfer versus fan cost trade-off, discussed in the previous section. Using the system simulation program, the annual system cost was determined as a function of the collector channel depth, and is plotted in Figure V-A-2. This result was determined for a system using 19,500 sq. ft. of collector area and 12,500 cu. ft. of rock storage,

Figure V-A-2
Collector Cost versus Collector Depth

supplying 150°F air for a 200 day season under conditions prevalent at the Fresno site. Oil at \$15 per barrel, zero inflation, a 10% tax credit, a 50% tax rate, a 10% interest over a twenty year life, and a 1% maintenance, replacement and insurance rate, were assumed in preparing Figure V-A-2.

The insulation thicknesses used on the collector were also determined through a similar life-cycle cost optimization procedure.

3. COLLECTOR PERFORMANCE PREDICTIONS AND TEST RESULTS: A 70 ft. long section of the collector (Figure V-A-3) was constructed for experimental verification of structural integrity, air tightness, performance and ease of fabrication. The efficiency performance results are shown in Figure V-A-4, along with predictions made using equations (1), (2), and (3), below. These prediction equations are similar to those used to predict the SEAD collector results described in Attachment V-A-1, except that they were derived for the case of preheated air and not ambient air entering the collector. It can be shown that the air temperature at the outlet of a collector of the type proposed is given by the equation:

$$T_0 = T_1 + [(\tau \alpha) I - U_F' (T_1 - T_a)] (E)/U_F' \quad (1)$$

where

$$E = 1 - \exp \left[- U_F' / (GC_p (1 + U_F' / (hk))) \right]$$

Figure V-A-3
The 70 ft. Test Collector

and T_1 is the air inlet temperature; and the other variables are as defined in Attachment V-A-1.

The collector efficiency is given by:

$$\eta = GC_p (T_0 - T_1) / I \quad (2)$$

The average absorber surface temperature, required for the determination of the heat loss coefficient U_F is given by:

$$T_{pa} = T_a + ((\tau\alpha) I (1 + U_B/(hk) - \eta I)/U_F \quad (3)$$

The various heat transfer coefficients were evaluated on the basis of the equations given in the attachment. The effective transmissivity-absorbtivity product, $(\tau\alpha)$, included allowance for mutual shading of collectors at low sun angles during the fall season of operation. These same equations were used to model the collector performance in the system simulation computer program.

Due to the shortness of the test collectors relative to those to be used in the Fresno application, efficiency results have so far only been obtained for flow rates corresponding to relatively low temperature rises, and for no flow conditions. Nevertheless, the experimental data are felt to substantiate the prediction procedure. The pressure drop measurements were found to be below those expected from the SEAD studies, allowing for some reduction in expected operating costs. The pressure drop measurements on

the design collector resulted in equivalent friction factors that are shown superimposed on those friction factors measured in the SEAD experiment in Figure 5 of Attachment V-A-1.

Also shown in Figure V-A-4, are the performance predictions for the full length (180 ft.) collectors of this type under operating conditions at the dehydrator site. (It should be noted that the peculiar behavior of the two prediction curves of Figure V-A-4, near $G=1$, is due to an approximation in the prediction model at this low flow rate for which laminar and free convection effects dominate the absorber-to-flowing-air heat transfer. This flow region is largely unused during system operation.)

The curves of Figure V-A-4 are for collectors heating ambient air only. If the inlet air is preheated by a heat recovery unit, the efficiency of the collectors will change. Thus, a more general plot of the predicted design collector performance was prepared, based on equations (1), (2), and (3), and plotted in Figure V-A-5.

The tests conducted on the 70 ft. collector also demonstrated that the design was structurally sound, while the air passages were cycled from atmospheric pressure to pressures of eight inches water column (i.e. more than twice the expected operating pressure). Air leakage appeared to be minimal (i.e. with internal caulking of the seams) and is expected to be additionally reduced by the application of sprayed polyurethane side insulation planned for the prototype

Figure V-A-4

Test results and predicted performance for collectors heating ambient air

Figure V-A-4

Test Results and Predicted Performance for Collectors Heating Ambient Air

Figure V-A-5

Predicted Performance with Preheated Inlet Air

collectors.

4. COLLECTOR AREA REQUIRED: The collector field layout (see Figure III-2, and construction drawings of Section VI) was largely dictated by available space. The collector banks are spaced at 6 ft. 6 in. centers to minimize mutual collector shading without the need for undue manifold duct extensions. With this spacing the collectors shade each other only during the late fall portion of the operating season. At the winter solstice, the collectors are shaded, at most, 20% in the early morning and late afternoon (i.e. at collector startup and shutdown).

The total collector area required was determined by using the system simulation model. Figure V-A-6 shows the equivalent annual system cost per MBtu supplied (i.e. as defined in Figure III-7c), as a function of collector area for various values of oil cost. This figure also indicates the proportion of system energy supplied by solar energy (i.e. as defined in Figure III-7) as a function of collector area. These curves are for a 200 day season at the Fresno facility, but assume the conditions of September 21, (i.e. the design day) to be common to all 200 days. All the curves assume a fixed storage volume of 12,500 cubic feet, with the same economic parameters assumed for Figure V-A-2.

As expected, the cost curves indicate that a zero area collector

Figure V-A-6

Energy Cost and Percent Solar Versus Collector Size Using the September Fresno Simulation

is the most cost effective if oil is available at \$15/barrel.

Accordingly, oil should be burned, totally, except for about 6% of the energy supplied by electricity to run the heat recovery fan, a value that could be reduced by sizing this fan appropriately to cater only for the heat recovery unit. At current oil prices the optimum size of the storage unit is about 7,000 cu. ft.

(Figure V-C-1). To increase the percentage contribution of solar energy, the storage volume was fixed at 12,500 cu. ft. In order to obtain an optimum solar collector area with a fixed heat storage volume, the oil price must be above \$15/barrel. For oil costs higher than this, the minimum cost appears to occur at collector areas of 19,500 sq. ft. and higher, regardless of the cost of oil.

The decision as to the required size of the collector field was based on the argument that the system should demonstrate the use of twenty-four hour storage, and correspondingly deliver a large part of the dehydration heat requirements by means of solar energy. If, and when, such a system becomes economical, it would appear from Figure V-A-6 that collector areas in the order of 20,000 sq. ft. would deliver a substantial fraction of solar energy at minimum cost. (Figure V-A-6 assumes a storage volume of 12,500 cu. ft. This figure was arrived at as discussed in Section V-C, where it is shown that when oil prices are in the \$50 to \$100 range, 12,500 cu. ft. of storage yields near minimum costs.)

A 19,500 sq. ft. collector area was chosen and this corresponds to the net useful collector area of 30 of the collector modules proposed for the Fresno project. As shown in Figure V-A-6, the corresponding proportion of solar energy supplied is 79%. The expected seasonal proportion of solar energy supplied (i.e. 69% according to Figure III-8) is less than 79% since September is not necessarily a typical month.

REFERENCES

- (1) Research on the Application of Solar Energy to the Food Drying Interim Report, TRW Report No. 27319-6001-TU-00 prepared for California Polytechnic State University, Oct. 1974.
- (2) Research on the Application of Solar Energy to the Food Drying Industry, Third Quarter Progress Report, California Polytechnic State University, Oct. 1974, Report NSF/RANN/SE/GI 42944/PR/74/3.

ATTACHMENT V-A-1

DESIGN AND PERFORMANCE OF AN AIR COLLECTOR
FOR INDUSTRIAL CROP DEHYDRATION

by

P. W. Niles, E. J. Carnegie, J. G. Pohl and J. M. Cherne

Paper presented at a Joint Conference of the American Section of International Solar Energy Society and the Solar Energy Society of Canada, held at Winnipeg, Manitoba, Canada in August, 1976.

Abstract

Test results are reported for the operation of unglazed and single-glazed solar collectors used to heat air to the 90°C (194°F) range. The collectors were constructed of standard black-painted metal decking and were tested in various lengths so that pressure drops and convective heat transfer rates could be varied independent of collector operation temperature.

It is shown that the experimental collector performance results with single pass operation are in substantial agreement with standard collector analysis procedures. These results give a firm basis for collector and system optimization.

Figure 1
(Attachment V-A-1)

Solar Dehydration System Configuration

Introduction

California Polytechnic State University in cooperation with TRW Corporation has completed a project under National Science Foundation RANN Grant #ERT 7419063 to investigate, design, construct and evaluate a solar collector to assist in the dehydration of food products. Solar energy represents an attractive alternate source of energy for this industry which, in California alone, consumes gas equivalent to that used by a city of nine million people. Since in the food dehydration industry most of the drying is accomplished with the use of hot air, this investigation concentrated on air collectors.

The type of collectors chosen for the investigation were constrained by the intentions of the grant to be relatively state-of-the-art in concept and configuration and to be applicable to state-of-the-art dehydration facilities.

Initial feasibility studies indicated that unglazed and single-glazed flat plate collectors should be used for this type of application where outlet temperatures from 40-90°C (104 to 194°F) are desired. Such collectors were expected to be suitable for the supply of all the air needed for dehydration, for preheating the air only, or for supplying topping heat to air preheated by a

heat recovery unit. A system configuration considered for typical application is shown in Figure 1.

Figure 2 shows a cross-section of the collector design chosen for testing. A single glazing of fiberglass-reinforced polyester sheet (Kalwall Corp.) was used as shown for the single-glazed tests. For unglazed tests the Kalwall and associated supports were removed. The air passages consist of galvanized steel decking (Ryerson Corp.) which was obtained in 0.6m x 10.7m (2 ft. x 35 ft.) sections. The collectors were run in single-pass fashion (air flow in air passages only) in the unglazed tests and single and double-pass (air first flowing under the glazing and returning through the air passages) in the single-glazed tests.

With this type of collector, it was expected that the solar absorber (top of decking) to cooling air heat transfer rates would have a substantial influence on system life-cycle costs. That is, high air velocities would be expected to increase collector efficiency (and thus reduce collector costs), but also increase required fan power (and thus increase operating cost). Thus, with future needs for solar system optimization in mind, the experiment was designed so that convective heat transfer rates and air pressure losses could be varied independently of collector operating temperatures. To accomplish this, the facility was constructed so that both 21m and 42m (70 ft. and 140 ft.) long collectors with the cross-section

All Metal Surfaces Painted with Black Nextel

Total Flow Area = .041 m²

Collector Horizontal with Air Flow North to South

Figure 2
(Attachment V-A-1)

Cross-Section of Solar Collector Tested

of Figure 2 could be tested. These lengths were expected to supply the required air temperatures at flow rates that would be near the optimum with respect to minimizing life-cycle costs. The actual value of the optimum length depends strongly on collector costs, system and electrical costs, temperature increase, operating days per year, and local environmental conditions.

This paper summarizes the main experimental results obtained along with results predicted on the basis of the standard Hottel-Whillier-Bliss (1,2) theoretical analysis procedure.

Performance Results

Figure 3 shows measured and calculated efficiencies as a function of the ratio of air temperature rise to insolation normal to the collector for single-pass flow operation. The data points shown are for tests made within 2 hours of solar noon, during which time the incidence angle of the sun on the collector was always less than 34° during the May 25 to August 9 period for which data is reported. Insolation during these tests was between 820 and 945 W/m^2 (260 and 301 Btu/hr. ft.^2), and wind velocities were generally light with gusts to 2.7 m/s (6 mph). The basis for the theoretical curves in Figure 3 is discussed in the next section.

Comparing the experimental results with the predictions in Figure 3

shows substantial agreement, with the theoretical predictions slightly optimistic in the unglazed case and perhaps slightly pessimistic in the single-glazed cases. A number of possible parameter adjustments in the theoretical model could be made to account for the differences, but without experimental verification with independently changed parameters, the precise nature of these adjustments could not be ascertained.

The predicted increase in performance due to the increase in collector length (and thus flow velocity for a given $\Delta T/I$) from 21.3m to 42.6m (70 ft. to 140 ft.) is seen to be comparable to that which occurred in the experiments, verifying the strong dependence of efficiency on flow velocity.

It may be noted that favorable agreement occurred between the predicted and experimental results at the low flow rates corresponding to transition region flow, although fully turbulent convection was assumed in the predictions. No experimental or theoretical results were obtained in the laminar flow region, although no-flow experimental results were obtained as shown on the abscissa of Figure 3.

Based on the theoretical equations, the heat losses through the insulation and through the sides of the space between the glazing and absorber had a substantial effect on the performance. It was

estimated that if the insulation were perfect, the efficiency curves for the single-glazed 21.3m (70 ft.) case would be raised 7 efficiency points at a flow rate of 20 kg/hr m^2 (4 lb/hr ft^2). If the sidewalls under the glazing were perfectly insulated, an additional 2 percentage points would be added.

For the 42.7m (140 ft.) collector, both insulation improvements would be expected to increase the efficiency by about 12 percentage points at a flow rate of 20 kg/hr m^2 .

Figure 4 compared the test data for a 21.3m (70 ft.) collector in double-pass operation with that for the single-pass 42.6 (140 ft.) collector from Figure 3. These two sets of data were compared because of equal pressure drops that occurred in each when operating at the same flow rate of air. Thus they represent collectors of identical cross-section and fan power, but different air flow routes. The comparison indicates that the double-pass configuration had a lower efficiency when operating with high temperature rises ($\Delta T/I > 0.01^\circ\text{C hr. m}^2/\text{KJ}$) and higher efficiency at low temperature rises ($\Delta T/I < 0.01^\circ\text{C hr. m}^2/\text{KJ}$). These results are qualitatively in agreement with Close (3) who predicted similar consequences in comparing a single-pass glazed collector to one with parallel flow under the glazing and in the air channel.

Pressure drop measurements taken during the collector tests were correlated by use of the Moody friction factor, f . Figure 5

Figure 3
(Attachment V-A-1)

Comparison of Theoretical and Experimental Collector Performance

Figure 4
(Attachment V-A-1)

Comparison of Single and Double Pass Performance Data

Figure 5
(Attachment V-A-1)

Experimental Friction Factors
Plotted on Moody Diagram (4)

shows experimental values of f versus Reynolds number, with properties evaluated at the average film temperature in the collector. This figure shows the f value to be generally higher than expected for commercial galvanized ducting, for which ASHRAE suggests using a relative roughness (see Figure 5) of 0.003. It should be noted in this respect, however, that the metal decking used had mechanically dimpled vertical surfaces, apparently for structural reasons.

The data for f was represented by the equation $f = 0.262 \text{ Re}^{-0.2}$ for use in the heat transfer analysis.

Basis for Performance Predictions

It can be shown by elementary energy balances (for example, see Threlkeld (5)) that the temperature rise of ambient air passing through solar collector having uniform flow beneath a flat-plate absorber is given by

$$T_o = T_a + (\tau\alpha) I\epsilon/U_F' \quad (1)$$

where:

$$\epsilon = 1 - \exp \left[-U_F' / (G C_p (1 + U_F'/hk)) \right]$$

T_a = ambient air temperature, °C

T_o = air temperature leaving collector, °C

$(\tau\alpha)$ = effective transmittance-absorbance product of cover

and absorber combination and essentially equal to the

fraction of incident solar radiation that is absorbed by the absorber plate.

I = total (direct plus diffuse) insolation normal to the collector, KJ/hr. m^2 .

U_F = combined convection and radiation heat loss coefficient between absorber and the outdoor environment, $\text{KJ/hr. m}^2 \text{ } ^\circ\text{C}$.

h = convection heat transfer coefficient from the inside wall of a circular duct (of equivalent hydraulic diameter to the collector air channel) to the air flowing through the channel, $\text{KJ/hr. m}^2 \text{ } ^\circ\text{C}$.

k = factor by which h is increased to account for heat transfer augmentation due to fins attached to the absorber plate, and due to radiation across the air channel.

$U_F' = U_F (1 + U_F/hk)$, is the effective heat loss coefficient including the losses through the insulation around the air channel.

U_B = overall conductance between the moving air and the outside environment via the insulation air channel walls. This was estimated to be $4.91 \text{ KJ/hr. m}^2 \text{ } ^\circ\text{C}$ (based on absorber area) for the configuration shown in Figure 2.

C_P = specific heat of air taken as $(1 + 1.84w) \text{ KJ/kg}^\circ\text{C}$ where w is the absolute humidity (kg water/ kg air).

G = mass flow rate of air per unit area of collector
surface exposed to the sun, kg/hr. m^2 .

Equation 1 is essentially that derived by Bliss (2) and Whillier (1) except for the method of including heat losses through the insulation. Since the useful heat output is $G C_p (T_o - T_a)$, then the instantaneous collector efficiency can be written as:

$$\eta = G C_p (T_o - T_a)/I \quad (2)$$

Using equation 1, this becomes

$$\eta = G C_p (\tau\alpha) \epsilon / U_F' \quad (3)$$

The average absorber plate temperature T_{pa} is required to determine U_F and can be obtained from a relationship derived similar to equation 1:

$$T_{pa} = T_a + (\tau\alpha)(I/U_F')(1 - G\epsilon C_p/U_F') \quad (4)$$

Except for the explicit dependence of T_{pa} on I , and thus U_F on I , the efficiency predicted by (3) is only dependent on the ratio $(T_o - T_a)/I$, and thus for a given flow rate, the values of both $\Delta T/I$ and η should be constant all day long for a given collector except for effects of large solar incidence angles and shading during early morning and late afternoon hours and thermal storage effects.

a. Determination of U_F : In the unglazed configuration U_F was taken as the sum of the convective loss (based on the Wind Coefficient of equation 3.9 of Whillier (6)), and the radiant loss to atmosphere. The effective sky temperature for radiation was assumed to be 11°C (20°F) below ambient, a value consistent with the "typical-atmosphere" correlation of Bliss (7) at the ambient dewpoint temperatures experienced during the test period. For the single glazed case (Kalwall) as shown in Figure 2, U_F was taken as the sum of the conductance through the glazing as determined by equation 3.9 of Whillier (6) and the overall conductance through the 5 cm. (2 in.) high wood glazing supports. The latter was calculated to be $1.43 \text{ KJ/hr. m}^2 \text{ }^{\circ}\text{C}$ ($0.07 \text{ Btu/hr. ft.}^2 \text{ }^{\circ}\text{F}$) based on the absorber area. It was assumed that the black painted absorber and glazing emissivities were 0.95 and 0.9 respectively. The Kalwall transmissivity to thermal radiation was assumed to be zero.

b. Determination of h and k : The convective heat transfer coefficient h was estimated using Reynolds analogy in the form $(St)(Pr)^{2/3} = f/8$, where the Moody friction factor f was determined from the experimental correlation shown in Figure 5. Substituting this into the above heat transfer equation yields

$$Nu = 0.033 Re^{0.8} Pr^{1/3} \quad (5)$$

This equation was used to evaluate the coefficient h with all properties evaluated at the film temperature.

A hydraulic diameter of 8.5 cm (3.4 in.) was calculated for the air channel and the average flow velocity for the total channel cross-sectional area was used in the calculations.

It should be noted that equation 5 was used in determining h for all predictions of Figure 3 including the transition flow region. Free convection effects were not calculated.

It was estimated through fin and radiation heat transfer calculations that the channel in Figure 2 would have 40% higher heat transfer from the absorber surface to the cooling air than would be expected if the fins and metal bottom did not conduct heat and were totally reflective to thermal radiation. Thus, a value of $k = 1.4$ was assumed. Although calculated for mid-range performance conditions, this value was used for all predictions.

c. Determination of $(\tau\alpha)$: A $(\tau\alpha)$ of 0.93 was assumed for the unglazed configuration with the flat-black painted absorber. In the glazed configuration this was modified as described by Whillier (6) to include the glazing transmissivity (measured with an Epply pyranometer as 0.88 for the 0.102 cm (0.040 in. Kalwall)) cover reflections and cover solar heating (together increasing $(\tau\alpha)$ by $1\frac{1}{2}\%$), and shading (primarily due to the center support rod) of about 1%. Together these factors reduce $(\tau\alpha)$ to 0.83 for the glazed collector cases.

All calculations were based on a wind velocity of 3.2 km/hr (2 mph), ambient temperature of 21.1°C (70°F) and an insolation of 863 W/m² (275 Btu/hr. ft.²).

Discussion

Because of the relatively close agreement between the experimental results and the predictions, it is felt that the theoretical methods used are sufficiently accurate for application in the advanced planning of facilities wherein these types of collectors are desired.

As a part of ERDA's program for Application of Solar Energy to Industrial Drying or Dehydration Processes, the University and TRW have, under contract, designed a complete solar heat source system, similar to Figure 1, to be applied to an existing prune and raisin dehydration facility located near Fresno, California. Collectors similar to those discussed herein are being considered for this application. This system will include a heat recovery unit not presently used in the facility and crushed rock heat storage unit necessary for 24 hour operation. Operating modes will allow collection, storage, and use functions to occur simultaneously as required.

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ATTACHMENT V-A-2

A COST COMPARISON OF WATER AND AIR SOLAR FOOD DRYING SYSTEMS

The following analysis indicates the basis upon which an air solar collector system was chosen in favor of a water system. The analysis is of a preliminary nature and should not preclude the future investigation of a water system in comparable detail to the air system study undertaken in this project.

Solar collector systems which use water as the working fluid have two major advantages compared to solar collector systems utilizing air. Because of the high heat capacity of water, the size of the heat storage component is significantly smaller. Furthermore, the initial cost of water pumps is much lower than the cost of fans for an air system. Major disadvantages of water systems are the higher cost of water solar collectors and a shorter life due to corrosion, erosion and fouling. An air-water heat exchanger is also required with a water system, and its inefficiency results in the requirement of a larger solar collector area. Moreover, although the size of the pipes used in a water system are significantly smaller and less costly per unit length compared to the large diameter ducts used in an air system, the total plumbing cost of a water system utilizing commercially available solar collectors is higher than

that of the air system proposed in this project. This is due to the larger number of manifolds needed to connect the relatively short length water collectors.

Using the proposed solar dehydration system as a basis, estimates of the effective cost savings of components in an equivalent water solar collector system were made. These effective cost savings were then added to the cost of the proposed air solar collectors to arrive at an allowable cost for equivalent water solar collectors, for the same cost per therm of solar energy. It was found that the allowable cost for water solar collectors could be as high as \$5.69/sq. ft. (i.e. cost of the insulated solar collectors without supports or manifolds), and may be much lower if plumbing costs are as high as reported in previous solar collector studies (1,4).

Since typical single-glazed solar water collectors appear to cost in excess of \$6.30 sq. ft. (see Table 1), they would apparently not be favorable for the proposed food dehydration application. The same conclusion was reached by TRW in a preliminary report (2). In this analysis the efficiency of the air and water solar collectors was assumed to be the same, while the cost of treatment equipment and glycol or similar solutions required in water systems for corrosion and frost protection were not included.

The operating costs of solar systems using water were not analysed but the operating costs of a solar system utilizing air would be higher because of the larger fan power required as compared to a water system; however, this energy is directly converted into a temperature rise of the air. In a water solar system, energy not provided by fans must be obtained by increasing the solar collector area. Since the electrical rate is about the same as the cost of energy supplied by the solar portion of the food dehydration system, no large variation is expected to occur in a cost comparison due to differences in fan power of air and water systems.

The analysis used to obtain the allowable cost for a water solar collector follows:

1. STORAGE: The designed air storage system uses rock as the heat storage medium. If water were passed through the rock, the volume of the storage unit could be 1.83 times smaller. If the present cost of the air storage system is reduced by a factor of 1.83, a cost saving with a water system of \$18,400 is realized. Although the designed air storage system could be reduced in volume by a factor of 2.7 if a total water storage system were used, sharp temperature gradients that occur with rock storage would be more difficult to obtain. The cost of obtaining a leak-proof water storage unit has not been included.

Present Cost of Air-Rock Storage	\$40,600
Proposed Cost of Water-Rock Storage	\$22,200
Effective Savings with a water system	\$18,400

Assumptions: $C_{\text{rock}} = 0.2 \text{ Btu/lb.}^{\circ}\text{F}$; volume fraction of rock = 0.3;
density of rock = 165 lb./cu. ft.

2. PLUMBING: Plumbing costs for a water solar collector system were obtained from Means (3), based on 3 in. and 4 in. diameter PVC schedule 90 pipes with hangers and fittings, and spray-on polyurethane foam insulation. It was assumed that 8 ft. long solar collectors (i.e. the length of the lowest priced solar water collector) would be used with manifolds at the top and bottom of each collector. These manifolds require an additional 5,500 ft. of piping as compared to the designed air system, which brings the cost of plumbing for the water system to \$81,480. The ratio of the plumbing costs of the water system to the cost of the solar collectors of the designed air system is 0.67 which is considered a conservative estimate. In other collector projects the cost of the pipes and fittings range from one (1) to almost three times (4) the cost of the solar collectors, in addition to which the cost of the water solar collectors was significantly higher than the cost of the collectors in the proposed air system. The cost of air traps and drain valves was not considered in this study.

Present Air Duct Costs

Manifolds to Solar Collector	\$ 30,320
Above Ground Duct @ \$24/sq. ft.	\$ 10,800
Above Ground Insulation	\$ 9,700
Below Ground Duct @ \$100	\$ 22,600
	<hr/>
	\$ 73,420

Water Pipe Costs (3)

Manifolds to Solar Collector	
\$12.00/ft.	\$ 70,800
Above Ground Pipes Insulated	
\$12.00/ft.	\$ 5,400
Below Ground Pipes Insulated	\$ 5,280
	<hr/>
	\$ 81,480

Effective Cost increase with water system	\$ 8,060
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3. AUXILIARY EQUIPMENT: The estimated cost of low-volume-flow-rate water pumps is considerably lower than the cost of the high-volume-flow-rate fans used in the air solar collector system.

Including the cost of an air-water heat exchanger, the effective cost saving estimated for auxiliary equipment is \$5,890 when a water solar collector system is used.

Present Air Fans Cost	\$ 12,890
Water Pump Cost Estimate	\$ 1,000
Air-Water Heat Exchanger Estimate	\$ 6,000
Effective Savings with a Water System	\$ 5,890

Therefore the estimated effective cost saving for the heat storage unit, plumbing and auxiliary equipment for a water

system, as compared to an air system, is \$16,230.

The present cost of the insulated solar collector in the proposed air system, without supports or manifolds, is \$5.52/sq. ft. The air-water heat exchanger is assumed to be 85% efficient because of the larger temperature rise requirement of the solar collector and corresponding lower collector efficiency. Thus, about 10% more collector area is required with a water system. The allowable cost of insulated water solar collectors is \$5.69/sq. ft.

Present Cost of Air Solar Collector	$\frac{121,522}{22,000 \text{ sq. ft.}}$	= \$5.52/sq.ft.
Allowable Cost of Water Solar Collector	$\frac{121,522 + 16,230}{(1.10)(22,000) \text{ sq. ft.}}$	= \$5.69/sq.ft.

Costs of Commercially Available Liquid Solar Collectors:

Sunworks	\$12.47/sq. ft.
Ying Manufacturing Corp.	11.14/sq. ft.
e and k Services Company	6.56/sq. ft.
Tranter Inc. Platecoil Division (without insulation or glazing)	8.10/sq. ft.
Garden Way Laboratories	6.30/sq. ft.
Sunwater	8.15/sq. ft.
Alanta (Towns) Solar Equipment	16.50/sq. ft.

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B. FAN AND DUCTING SELECTION:

1. FAN SELECTION: As shown in Figure V-B-1, three fans are used in the proposed solar food dehydration system in addition to the existing dehydration tunnel fan. Fan #1 maintains a constant flow rate with variations in system resistance, dependent upon the mode of operation. Ignoring duct losses and given adequate sun conditions, this fan must overcome a relatively large pressure loss through the collectors, in addition to the pressure loss through each side of the heat recovery unit. In the case of "no-sun" conditions, this fan must normally overcome a relatively small pressure loss through the storage unit, plus the pressure loss through each side of the heat recovery unit. Therefore fan #1 must be sized for the "sun" mode of operation (i.e. with operation for about six hours per day), but will operate during the main portion of the day (eighteen hours of "no-sun" condition) with relatively low system resistance.

Fans #2a and 2b must operate at various flow rates from zero to a design maximum and, ignoring ducting losses, must be sized to overcome the pressure loss through the rock storage and the collectors. The system resistance for fans #2a and 2b is not constant since the flow rates through the collectors and storage are different.

Centrifugal fans were chosen for the three fans because, compared to axial fans, they require a lower horsepower when run at lower flow rates. Other advantages of centrifugal fans are their lower

noise levels, especially at higher pressure differentials, the ease of replacement of bearings, and the lower variation in flow rate for a given pressure change.

Although of higher cost, backward inclined airfoil shaped centrifugal fans were selected in the design, in favor of forward curved bladed fans, because of the non-overloading characteristic, stability, and higher efficiency.

Variations in the rise of fan pressure may be obtained by the use of inlet guide vanes, dampers or both. Because of the low usage at non-rated conditions, the initial investment of inlet guide vanes for fans #2a and #2b is not economically warranted, and only damper control is provided. Decreased operating costs warrant the initial investment of inlet guide vanes for fan #1 instead of dampers.

In order to provide fan performance data for the computer simulation of the solar dehydration system, dimensionless performance characteristics were obtained for Buffalo backward inclined airfoil bladed centrifugal fans with inlet guide vane and damper control, and are presented in Figure V-B-2. This curve was obtained using Buffalo's dimensionless performance curve presented in Figure V-B-3.

In the case of Figures V-B-1 and V-B-2, the rated fan horsepower, static pressure and flow rate must be used. A tabulation of fan

Figure V-B-2

Percent rated horsepower as a function of percent rated pressure and percent rated capacity.

Typical Variable Inlet Vane Performance
Type "BL" Fan

Key:

- 1 = H. P. Pos "A"
- 2 = H. P. Pos "B"
- 3 = H. P. Pos "C"
- 4 = S. P. Pos "A"
- 5 = S. P. Pos "B"
- 6 = S. P. Pos "C"

Figure V-B-3

Dimensionless Fan Performance for
Buffalo "BL" Centrifugal Fans with Inlet Guide Vane Control.

discharge velocities corresponding to points of rated fan performance (i.e. maximum efficiency), as a function of fan pressure rise, is presented below.

Fan Pressure Rise (in W. G.)	Outlet Velocity at Rated Fan Performance (ft./min.)
0-3	2000
3-6	2400
6-8	2800
8-12	3200

Fan performance information is based on Buffalo's performance bulletin F-150B.

2. DUCT SIZE AND INSULATION THICKNESS: Optimum duct size and insulation thickness are primarily functions of the initial component costs, electrical costs, and the cost of the heat lost as compared to the infinite insulation case. In this section, duct costs are compared using various flow rates, modes of operation, and insulation thicknesses.

The nominal design flow rates through the ducts of the system, exclusive of the manifolds to and from the collectors (to be discussed in the following section of this report), are 20,000 cfm on a 24-hour per day basis and 30,000 cfm on a 6-hour per day basis. The nominal design air temperature in the ducts is

assumed to be either 150°F or 110°F for a flow rate of 30,000 cfm. Two conditions were investigated in detail. The duct cost per therm for a 100 ft. long round duct is presented as a function of duct diameter, insulation thickness and air temperature in Figures V-B-4, V-B-5, and V-B-6. The results were obtained using equation 1, which assumes a dehydrator heat need of 1.4×10^6 Btu/hr., and amortization of the initial cost over a 20 year life at 10% interest. Additional assumptions used in the calculations are listed below:

$$\text{Duct cost per therm} = \frac{\text{Duct Cost per year}}{\text{Therms per year}} \quad \text{Eqn 1a}$$

$$\text{Duct cost per year} = 0.116^* (C_{ID} + C_{II} + C_F) + C_{FO} + C_{LH} \quad \text{Eqn 1b}$$

*Annual amortizing constant for 20 years at 10% interest.

Assumptions used in the calculations are listed below:

Initial cost of galvanized steel duct installed (C_{ID})
(Spiral Lock Seam Duct)

Duct Dimension (in.)	Gage Galvanized	Cost/sq. ft. (\$/sq. ft.)
23-36	22	2.843
37-50	20	3.348
51-60	18	4.359

Initial insulation cost installed (C_{II})

Base cost of \$1.05/sq. ft. (surface preparation, \$0.15/ft. + weather proofing, \$0.90/sq. ft.), plus \$0.25 sq. ft. per inch of insulation thickness.

Duct pressure loss = 1.5 velocity heads + Moody smooth tube friction loss

Initial cost of fan (C_F) = \$ 80. per horsepower required to overcome duct pressure loss

Electrical cost = \$0.90 per therm

Fan efficiency = 60%

Therms per hour = 14 therms

Operating days per year = 200 days

Cost of lost heat (C_{LH}) = \$0.90 per therm

Thermal conductivity of insulation = 0.01416 Btu/hr. ft. °F

Ambient temperature = 69.4°F (average ambient temperature in Fresno for August, September, October, and half of November)

$$h_i = \frac{0.32 v^{0.8}}{D^{0.25}} \text{ Btu/hr. sq. ft. } ^\circ\text{F} \quad (1)$$

$$h_o = 0.0239 \left(\frac{v_\infty D_o}{\gamma_f} \right)^{0.805} \left(\frac{k_f}{D_o} \right) \text{ Btu/hr. sq. ft. } ^\circ\text{F} \quad (2)$$

$$\text{for: } 4 \times 10^4 \leq \left(\frac{v_\infty D_o}{\gamma_f} \right) \leq 4 \times 10^5$$

Ambient velocity (V_∞) = 8.65 ft./sec. (i.e. average ambient velocity in Fresno for August, September, October and the first half of November).

The initial cost for galvanized ducting was obtained from 1978 building construction cost data (3), using the thicknesses of galvanized steel ducting recommended by ASHRAE (4). This cost includes the cost of fittings and compares well with listed prices

of a manufacturer (5). Initial cost estimates for insulation (i.e. spray-on polyurethane foam) were obtained from contractors in the Fresno area.

The duct pressure loss of 1.5 velocity heads plus smooth tube Moody friction loss is an estimate of a typical pressure loss in 100 ft. of duct for the proposed system. This pressure loss, as well as the electrical cost, fan efficiency, and the number of operating days per year, is used to calculate the cost of fan operation (C_{FO}). The cost of heat lost through the duct is considered an operating cost. The value of this heat loss is \$0.90 per therm based on estimate of the cost per therm of the energy contributed by the solar portion of the food dryer.

As shown, in Figures V-B-4, V-B-5 and V-B-6, the optimum duct dimension for a minimum duct cost per therm, with a flow rate of 30,000 cfm and 6-hour per day operation, or with a flow rate of 20,000 cfm and 24-hour per day operation, independent of inside air temperature, is 4 ft. The duct cost per therm is also a function of insulation thickness, and inside air temperature. At a flow rate of 20,000 cfm, an optimum insulation thickness of 4 in. occurs with an inside air temperature of 150°F (i.e. very little difference in duct cost per therm occurs with insulation thicknesses between 3 in. and 5 in.), while the optimum insulation thickness with an inside air temperature of 110°F is 3 in. (i.e. very little.

Figure V-B-4

Duct cost per therm ($\$/1 \times 10^5$ BTU per 100 ft. duct)
vs. duct diameter for various insulation thicknesses.

Flow Rate = 20,000 cfm, 24 HR./day, 200 day/year

Temperature in duct = 150°F

Figure V-B-5

Duct cost per therm ($\$/1 \times 10^5$ BTU per 100 ft. duct)
vs. duct diameter for various insulation thicknesses.

Flow Rate = 30,000 cfm, 6HR./day, 200 day/year

Temperature in duct = 150°F

Figure V-B-6

Duct cost per therm (\$/1x10⁵ BTU per 100 ft. duct)
vs. duct diameter for various insulation thicknesses.

Flow rate = 20,000 cfm, 24 HR./day, 200 day/year

Temperature in duct = 110°F

difference in duct cost per therm occurs with insulation thicknesses between 2 in and 4 in.). At a flow rate of 30,000 cfm and 6-hour per day operation, with an inside air temperature of 150°F, the optimum insulation thickness is 2.5 in. (i.e. very little difference in duct cost per therm occurs with insulation thicknesses between 1 in. and 4 in.).

The value of the minimum duct cost for the operating conditions presented in Figures V-B-4, V-B-5 and V-B-6 varies from \$0.0145 to \$0.016 per therm per 100 ft. of duct.

The relative magnitudes of the initial duct and insulation costs, electrical cost, and heat loss cost are compared using a 24-hour per day operation at a flow rate of 20,000 cfm. At a duct dimension of 4 ft. with the optimum insulation thickness of 4 in., the initial cost of the duct is 44% of the total duct cost per therm. The insulation and electrical costs are 37% and 13%, respectively, of the total duct cost per therm, and the cost of the lost heat is 15% of the total duct cost per therm. The initial cost of a fan to overcome the duct pressure loss is 1% of the total duct cost per therm.

The pressure loss per 100 ft. of duct as a function of duct dimension is presented in Figure V-B-7 for flow rates of 20,000 cfm and 30,000 cfm. The pressure loss is the sum of pressure

losses due to friction and an assumed 1.5 velocity head loss caused by fittings. One further consideration for the choice of duct dimensions was that the total system pressure loss should be in the range of capability of the fans chosen. The optimum duct dimension on a cost basis resulted in suitably low duct pressure losses.

The air temperature loss per 100 ft. of duct, as a function of duct dimension and insulation thickness, is presented in Figure V-B-8, for flow rates of 20,000 cfm and 30,000 cfm, with an inside air temperature of 150°F. At a duct diameter of 4 ft. and optimum insulation thicknesses, the air temperature per 100 ft. of duct is about 0.25°F. The air temperature loss per 100 ft. of duct, as a function of duct diameter and insulation thickness, is presented in Figure V-B-9 for an air flow rate of 20,000 cfm and with an inside air temperature of 110°F. At a duct diameter of 4 ft. and optimum insulation thickness of 3 in., the air temperature drop per 100 ft. of duct is about 0.13°F.

The use of round ducting has several advantages as compared to square ducting. The comparisons made below assume the same cross-sectional area for round and square ducts. In view of the lower surface area, the initial cost of insulation and the cost of lost heat will both be about 13% lower for a round duct compared to a square duct. The pressure loss for straight

Figure V-B-7

Duct pressure loss (in H₂O)
vs. duct diameter (ft.)

Duct pressure loss due to Moody
Smooth Tube Friction plus 1.5
velocity head loss due to fittings

Temperature in duct = 150°F

Figure V-B-8

Temperature loss per 100 ft. of duct (°F)
vs. duct diameter (ft.) as a function of
insulation thickness.

Temperature in duct = 150°F

Figure V-B-9

Temperature loss per 100 ft. of duct (°F)
vs. duct diameter (ft.) as a function of
insulation thickness.

Temperature in duct = 110°F

Flow Rate = 20,000 cfm

Figure V-B-10

Schematic of Manifold for Solar Collectors

Collector	Supply Manifold Diameter (ft.)	Return Manifold Diameter (ft.)	Flow Rate Ratio
1	5	5	1
2	5	5	0.9970
3	5	5	0.9943
4	5	5	0.9919
5	5	5	0.9898
6	4	5	0.9791
7	4	5	0.9806
8	4	4	0.9844
9	4	4	0.9818
10	4	4	0.9797
11	3	4	0.9690
12	3	4	0.9713
13	3	4	0.9731
14	2	4	0.9624
15	2	4	0.9697
16	2	4	-

TABLE 3

round duct flow, and therefore the electrical operating cost, will be about 10% lower. The initial cost of round spiral ducting (5) assuming three 90° elbows per 100 ft. of duct, is about 24% lower than the estimated cost for square ducting (3).

3. SUPPLY AND RETURN MANIFOLD FOR SOLAR COLLECTORS: A schematic diagram of the designed manifold system which supplied and returns air from the solar collectors is presented in Figure V-B-10. A computer program was written to predict the flow rate through each collector as a function of the dimensions of the supply and return manifolds, the total flow rate through the manifold, the collector pressure loss, and the supply and return branch geometries and velocities. References (6), (7), and (8) were used to calculate the various fitting and duct losses.

A large number of inlet and exit duct dimension sets were compared first by using 90° supply and return fittings based on the 90° return branch loss procedure presented by Healy (9). A relatively large pressure loss in the return duct, and a large difference in the flow rates through each collector existed for this case. The use of 45° return branch fittings used in the design resulted in relatively lower pressure losses and more uniform flow rates between various collectors. After trying a large number of duct dimension sets, while keeping the duct dimensions as small as possible for low cost, and still have little flow rate non-uniformity between

various collectors, the duct dimensions listed in Table 3 were chosen. Also listed, are ratios of the flow rates through each collector to the flow rate through the first collector. With the designed system at a minimum total flow rate of 40,000 cfm, the maximum difference in flow rates between any two collectors is less than 4% and the pressure loss through the supply and return manifolds is 0.62 in. water column. Less flow rate non-uniformity and less pressure loss occurs at lower total flow rates.

Fan #1	
Buffalo 805 SWSI Class II, Arrangement 1 Type BL-Airfoil Centrifugal Fan	\$ 2,350
25 HP Motor	\$ 447
Inlet Guide Vanes	\$ 870
Belt Drive	\$ 315
Controls (estimate)	\$ 600
Freight	\$ 242
Fans #2a and 2b	
Buffalo 980 SWSI Class II Arrangement 1 Type BL-Airfoil Centrifugal Fan	\$ 3,660
40 HP Motor	\$ 699
Inlet Guide Vanes	\$ 1,090
Belt Drive	\$ 530
Controls (estimate)	\$ 1,000
Freight	\$ 375
Total for Three Fans	\$19,532

TABLE 4

Figure V-B-11

Branch Loss Coefficients (K_B) for Converging Flow Fittings

Figure V-B-12

Main Loss Coefficients (K_M) for Converging Flow Fittings

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C. ROCK STORAGE DESIGN:

1. INITIAL DESIGN: The rock storage design is based on the analysis methods and empirical results available in the literature. To assure stable temperature stratification in the rock, the storage is designed for vertical flow, with flow downward during the filling cycle, as is conventional practice. The simulation program was used to assist in determining the appropriate bed cross-sectional area (i.e. perpendicular to the air flow direction), bed depth, and rock size.

For given storage specifications, such as initial temperature profile and air input conditions as a function of time, the rock storage subroutine determines the rock storage temperature depth profile, outlet air temperature, and bed pressure drop, at one and one-half minute simulated time intervals. An analytical expression for the pressure drop through the rock bed was obtained from Bird (1). This empirical equation correlates bed pressure drop as a function of bed rock diameter, storage void fraction, length of bed, and the velocity, density, and viscosity of the air passing through the particles.

To determine bed temperature profiles, the rock bed simulation subroutine utilized the standard numerical procedure of simultaneously integrating the energy balance equations (i.e. one for the rock and one for the air passing through the rock). The equations used are those described by Duffie (2), with minor modifications. This model assumes the rock bed to have zero conductivity in the flow direction and infinite conductivity in planes normal to it. The validity of

these assumptions is still under some debate in the literature, but the assumptions were generally felt to be justified for design purposes, particularly with respect to ease of solution. The infinite lateral conductivity assumption tends to lead to a conservative design with respect to side heat losses, which are exaggerated, whereas the first assumption is likely to lead to a slightly optimistic design unless an air-to-rock heat transfer correlation is used to compensate for the neglected bed conductivity. On the basis of Close's work (3) the volumetric heat transfer coefficient correlation of Lf and Hawley (4), as given in Duffie (2), is expected to compensate for this effect. An additional assumption was made, that the rock had infinite conductivity. This was assumed to be valid as long as the rock diameter is small enough to assure that the rock particle Biot number is kept below 0.1, as suggested by Duffie (2).

With the aid of the simulation subroutine it was determined that the rock bed depth had only a small influence on the effectiveness of the rock bed as a regenerative heat exchanger, at least for the range of depths needed to keep a compact storage shape. This was expected, in view of the rock sizes and flow rates of interest and the attendant small heat transfer relaxation length (see Balcomb (5)) in the order of 1% or 2% of the rock bed depth.

Thus, the appropriate rock bed depth and rock diameter was determined by consideration of pressure drop, and the structural requirements of

the storage container. Too deep a bed, or too small a rock size was shown to lead to pressure drops resulting in prohibitively high fan operating costs, as well as increased construction costs due to vertical rock loads. On the other hand, too short a bed, besides resulting in a noncompact storage configuration, can result in pressure drops so low that nonuniform flow through the storage bed can occur due to small but overriding nonuniform pressures in the distribution plenums at the storage entrance and exit. Reducing these nonuniformities can be achieved by increasing plenum sizes, but at the expense of increased construction costs.

The storage unit shown in Figure III-2 was designed for 0.5 in. rock and a 7.5 ft. rock depth, which resulted in a predicted pressure drop of about 0.6 in. water column at the maximum expected storage flow rate during loading. The rock was specified to be natural (round) river gravel washed and screened to $1/2 \pm 1/8$ in. size. The preponderance of granite in the Fresno locality ensures a relatively high density and resultant high heat storage capacity. The top and bottom distribution plenums designed for the storage unit illustrated in Figure III-2 are shown in the construction drawings in Section VI. These plenums are tapered to reduce nonuniformity of flow through the storage due to nonuniform pressures in the plenums. At flow rates corresponding to the 0.6 in. water gauge pressure drop discussed previously, distributed plenum pressure nonuniformities are expected to result in flow differences between the central and end sections

Figure V-C-1

Equivalent Energy Cost and Percent Solar
Versus Storage Volume Using September
Fresno Simulation with 19500 ft.² Collector

of the beds of less than 5%.

To determine the optimum storage volume, the total annual equivalent system cost was determined as a function of storage size and oil cost, and is plotted in Figure V-C-1. Also plotted is the percent energy supplied by the sun as a function of storage volume.

These curves are similar to the cost versus collector size curves presented in Figure V-A-6. The results indicate that the system costs are minimized close to the design storage size of 12,500 cu. ft. chosen.

2. REVISED DESIGN: The storage design discussed above was based on the preliminary design conclusion that a block structure would be cheaper than a modified commercial silo structure. However, the simulation analysis showed that the storage volume could be significantly cheaper. The revised design, shown in Figure III-3, utilizes a commercially available silo, modified to handle larger inlet and exit ducting. The storage volume is 12,500 cu. ft. of rock with a depth of 9 ft. Tests recently performed on a sample rock screening from the Fresno area showed that packed rock void fractions near 0.33 can be expected, with bulk densities near 107 lb./cu. ft. Pressure drop measurements are shown in Figure V-C-2, where a comparison is made with pressure drop predictions based on the equation referred to previously (i.e. reference (1)). These results seem to substantiate

Figure V-C-2:
Measured and Predicted Pressure Drops for Sample Gravel

the prediction equation. Since the experimental void fraction is lower than anticipated in the preliminary design, higher pressure drops are expected through the revised design storage, particularly in view of the increased depth. This suggests that rock larger than the 1/2 in. previously specified should be used. It is currently anticipated that 1 to 1½ in. diameter rock will be employed, resulting in a pressure drop of from 1.25 to 0.8 in. water column respectively. Adjustments in rock size specifications are facilitated by the wide range of natural river rock sizes available from quarries in the extensive San Joaquin River system located near the Fresno site.

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- 4) Löf, G. O. G., Hawley, "Unsteady State Heat Transfer Between Air and Loose Solids", Industrial Engineering Chemistry, Vol. 40, June, 1948.
- 5) Balcomb, J. D. "Design Considerations of Air Cooled Collector/Rock-Bin Storage Solar Heating Systems", 1975, International Solar Energy Congress and Exposition, July 1975, LA-UR-1334.
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D. HEAT RECOVERY:

Four types of heat exchangers applicable to air-to-air heat recovery are, secondary fluid, heat pipe, recovery wheel and the plate-fin type heat recovery unit. The secondary fluid heat recovery unit, illustrated in Figure V-D-1, circulates a liquid through two coils by means of a pump. The heat pipe heat recovery unit is similar to the secondary fluid heat recovery unit, except that the heat pipes replace the need for a circulating fluid. The heat recovery wheel, illustrated in Figure V-D-2, contains a rotating porous thermal mass. It stores heat energy on the hot exhaust airstream side and releases this energy to the cool incoming air. In plate-fin heat recovery units, as illustrated in Figure V-D-3, hot and cold streams of air travel through alternate passages in cross-flow or counter-flow arrangements.

A comparison of recovery wheels (Wing and Carnes), heat pipes (Hughes and Q-Dot Corporation), and plate-fin (Allied Air Products) heat recovery units is presented in Attachment V-D-1. The cost per therm of the heat recovered by each unit, as calculated by equation 1 includes the initial cost of the heat recovery unit, the additional cost of a larger fan necessary for the heat recovery unit, and the installation cost, all amortized over a twenty year life at 10% interest. The cost per therm also includes the fan operating cost using an electrical rate of \$0.90 per therm. The energy obtained from the unit includes the effect of the heat obtained from the fan.

Figure V-D-1

Secondary Fluid Heat Recovery Unit

Figure V-D-2

Recovery Wheel

Figure V-D-3

Plate-Fin Heat Recovery Unit

$$\text{Cost per therm} = \frac{\text{Cost per season}}{\text{Therm obtained per season}} \quad \text{Eqn 1a}$$

$$\text{Cost per season} = 0.116^* (C_I + C_{LF} + C_{IN}) + C_{FO} \quad \text{Eqn 1b}$$

$$\text{Therms obtained per season} = \text{Energy from HRU} + \text{Heat equivalent of fan power} \quad \text{Eqn 1c}$$

*Annual amortizing constant for twenty years at 10% interest.

Assumptions used are listed below:

$$C_{LF} + C_{IN} = 0.25 C_I$$

$$\text{Fan efficiency} = 60\%$$

$$\text{Flow rate} = 20,000 \text{ cfm}$$

$$\text{Operating days per year} = 200 \text{ days}$$

$$\text{Electrical costs} = \$0.90/\text{therm}$$

$$C_{p_{\text{air}}} = 0.24 \text{ Btu/lb. } ^\circ\text{F}$$

$$T_{\text{ATM}} = 69.4^\circ\text{F} \text{ (Average temperature in Fresno for August, Sept., Oct., and first half of November)}$$

$$T_{11} = 120^\circ\text{F}$$

$$T_{10} = 150^\circ\text{F}$$

The portion of the heat needed by the dehydrator which is supplied by the heat recovery unit, ignoring the temperature increase due to fan operation, is given by equation 2.

$$\text{Portion of Heat Need Supplied by HRU} = \frac{N_{\text{HRU}} (T_{11} - T_{\text{ATM}})}{T_{10} - T_{\text{ATM}}} \quad \text{Eqn 2}$$

The cost per therm of the Carnes' 12 ft. diameter heat recovery wheel (\$0.0467 per therm) is 28% lower than that of Wing's heat recovery wheel, is 42% lower than the lowest cost heat pipe unit, and is 21% lower than the lowest cost plate-fin heat recovery unit. The cost per therm of three Carnes' heat recovery wheels of different sizes is about the same. Since the cost of the solar portion of the energy for the food dehydrator is significantly larger than that for the heat recovery unit, a high heat recovery wheel efficiency is desired. The Carnes' 12 ft. diameter heat recovery wheel has the highest efficiency of all the heat recovery units investigated, with its rated efficiency of 86%. It will supply 54% of the heat need of the food dehydrator.

Besides the major advantage of the Carnes' heat recovery wheel, such as relatively low cost per therm and high efficiency, the heat recovery wheel is semi-self-cleaning since the flow direction through the wheel alternates as the wheel turns from the incoming air to the exhaust airstream. The short length of the recovery wheel in the flow direction will facilitate easy steam or hot water cleaning. The major disadvantage of the heat recovery wheel as compared to other types of heat recovery units, is that some contamination of the supply air by the exhaust air will occur. The designed solar dehydration system using draw-through of the supply and blow-through of the exhaust is not recommended for low contamination. A modification to the designed system consisting of

the installation of a fan downstream of the exhaust airstream, with the use of a purge system of the heat recovery wheel, would reduce contamination to less than 1%. This is a relatively simple modification if found necessary. Data on the percent contamination of the fan configuration used in the designed solar dehydration system is not available. The purge system of the heat recovery unit should not be used with the designed dehydration system.

The Carnes "MS" wheel should be used since the heat transfer will be primarily sensible. Condensation may occur during late season night operation in the exhaust side of the heat recovery wheel, and provision for the removal of this moisture has been made.

DEFINITIONS

C_{FO}	Yearly cost of fan operation (\$)
C_I	Initial cost (\$)
C_{IN}	Installation cost (\$)
C_{LF}	Additional cost of larger fan (\$)
HRU	Heat recovery unit
T_{ATM}	Ambient temperature ($^{\circ}F$)
T_{10}	Air temperature into dryer ($^{\circ}F$)
T_{11}	Exhaust air temperature from dryer into heat recovery unit ($^{\circ}F$)
N	Efficiency (%)

ATTACHMENT V-D-1

Data for air-to-air heat recovery units, air flow rate = 20,000 cfm

Wing WC-3160 Correx Heat Recovery Wheel

Initial cost	\$ 14,550
Efficiency	78.4%
Pressure loss per side	0.52 in. W. G.
Cost per therm (\$ per therm)	\$ 0.0648/therm

Carnes Therm-O-Wheel (MS Type)

	#120	#132	#144
Initial cost	\$ 9204	\$10,608	\$11,908
Efficiency (%)	81%	84%	86%
Pressure loss per side (in. W.G.)	0.60	0.45	0.35
Cost per therm (\$ per therm)	0.0469	0.0461	0.0467

Hughes Heat Pipe-T 15-121.5 x 138-6R12P

Initial cost	\$ 17,241
Efficiency	67.0%
Pressure loss per side	0.6 in. W. G.
Cost per therm (\$ per therm)	\$ 0.0889/therm

Q-Dot Corporation Heat Pipe, 54" x 129" coil

	11 Rows 14 Fins	6 Rows 14 Fins
Initial cost	\$ 14,712	\$ 9,568
Efficiency	80%	62.8%
Pressure loss per side (in. W. G.)	2.55	1.4
Cost per therm (\$ per therm)	0.0801	0.0865

Applied Air Products 10 ft. Temp.-X-Changer

	2 Units	4 Units
Initial cost	\$ 9,920	\$ 19,840
Efficiency	73%	81%
Pressure loss per side (in. W. G.)	2.0	0.85
Cost per therm (\$ per therm)	0.0656	0.0885

E. CONTROL SYSTEM:

The following discussion defines the control system required for the solar dehydration facility. Included in the definition are:

- General location of control instrumentation (i.e. thermocouples and pressure transducer) and control actuators (i.e. dampers and fans).
- Instrumentation and actuator designator symbols.
- Control operating modes.
- Control logic.
- Controller input/output.

1. CONTROL INSTRUMENTATION AND ACTUATORS: Figure V-E-1 shows the general layout of the solar dehydration facility with the associated control/instrumentation locations. Thermocouples (T/C's) T2A and T2B provide the temperature of the outlet air in the two solar collectors. T/C T22 provides the temperature of the outlet air in the heat storage unit. T/C T25 provides the temperature of the air entering the tunnel dehydrator after some of the waste heat is recovered from the heat wheel. T/C T7 senses the temperature of the air entering the tunnel from the solar heating system. T10 is the reference or setpoint temperature which the dryer operator determines. It is the desired temperature of the drying air supplied by the solar heating system. It will be

150°F nominally. Differential pressure transmitter (D/P) P1 acts as the sensor for the dryer air flow rate (nominally 20,000 cfm). The corresponding air flow actuator is damper D3. The tunnel 12 fan contacts are used to sense the occurrence of dryer shut down by the operator for a product load change.

Operation of dampers D1A, D1B, D2A, D2B, D4A, D4B, and D5 depends upon the relative magnitudes of the temperatures sensed by the thermocouples and the T10 setpoint. The exact damper response to the various combinations of temperature magnitudes is discussed in the next section.

Dampers D1A and D1B operate in a "ganged" fashion with D1A opening when D1B is closing or vice versa (i.e. opposed operation). The D1 dampers route product air through storage and/or through the collectors for heating. Separate damper motors are used for D1A and D1B. The D1 dampers are operated open/closed or modulating depending on the system mode defined by the temperature sensors.

Dampers D2A and D2B are ganged, but open and close in unison. The D2 dampers in combination with fans F2 route air from storage to the collectors in "pump around" fashion. The "pump around" occurs when the solar heating at the collectors is in excess of that required by the dehydrator. The excess heat is absorbed by storage. The maximum flow rate through storage during "pump around" is

60,000 cfm. Separate damper motors are used for D2A and D2B.

The D2 dampers are operated open/closed or modulating depending on the system mode.

Dampers D4A and D4B are ganged, operated opposed, and have a single motor (due to their close proximity). The D4 dampers permit some or all of the product air to bypass the solar heating system when the solar heated air is too hot or too cold for the dehydrator. (Auxiliary gas heating is provided when solar heat either from the collectors or storage is depleted.) The D4 dampers are operated open/closed or modulating depending on the system mode.

Damper D5 serves to isolate the solar heating system from the dehydrator when the solar heat is depleted or when there is a product load change and solar heat is available for adding to storage. D5 operates only in an open/closed manner depending on the system mode.

Damper D3 in combination with fan F1 provide the necessary air flow rate for the dehydrator. D3 operates as a regulator for all system modes except when the dryer system is completely shut down. The nominal pressure drop sensed by D/F F1 across the heat recovery wheel is 1 in. of water column for a flow rate of 20,000 cfm.

Figure V-E-1
Control/Instrumentation Schematic

2. CONTROL OPERATING MODES: The objective of the automatic control system is to provide for maximum use of solar heat within the bounds of the collector/storage capability. The operating modes have been defined with this objective in mind. Table 1 shows the various system operating modes. For each anticipated combination of sensor conditions, the proper response of dampers and fans is defined.

In mode 1, corresponding to a nighttime condition, both the collectors (T2) and heat storage (T22) are cold. In this case the solar heating system is completely bypassed. All heating is provided by a separate auxiliary system.

Mode 2 represents a sunrise condition where the collectors are providing some heat for the dehydrator with the remainder made up by auxiliary heating (i.e. sufficient solar heat is not yet available for heating storage).

Mode 3 represents a similar condition to mode 2 except that there is still considerable heat available from storage. Such a condition would exist during intermittent cloudy periods. By modulating dampers D1, product air is divided between collectors and storage such that collector temperature equals storage temperature.

Modes 4 and 5 are also similar. Both exist during periods of strong solar insolation. In mode 4, both the dehydrator and storage receive solar heat. Damper D2 is modulated such that the "pump around" air flow and the dehydrator flow are just sufficient to maintain collector temperature T2 equal to the dehydrator setpoint T10. In mode 5, the "pump around" air flow has reached its maximum. Via damper D4, bypass air cools the air from the solar heating system. D4 is modulated to maintain dehydrator inlet air temperature T7 equal to the T10 setpoint. Excess solar heat is added to storage.

Modes 6 and 7 represent nighttime or overcast conditions when heat is available from storage but not from the collectors. When the storage temperature T22 is less than the setpoint T10 and yet T22 is greater than the incoming fresh air temperature T25, all the product air is routed through storage. Thus some of the required dehydrator heat is supplied by storage with the remainder supplied by auxiliary heating. This defines mode 6. In mode 7, the storage temperature is so high that some of the product air is bypassed. D4 is modulated to maintain the dehydrator inlet air T7 equal to the T10 setpoint.

Modes 8 and 9 occur when the operator opens the tunnel dehydrator door (and shuts off the tunnel 12 fan) to change the product load. Mode 8 represents the daytime occurrence. Solar insolation is

MODE	SENSOR CONDITIONS	D1A	F1B	D2A	D2B	D4A	D4B	D3	F2	F1	CONDITIONS TO MAINTAIN
1.	T2 & T22 < T25	Closed	Open	Closed	Closed	Closed	Open	Closed	Off	On	Bypass entire system.
2.	T25 < T2 < T10 & T2 > T22	Open	Closed	Closed	Closed	Open	Closed	Open	Off	On	All product air thru collectors only.
3.	T25 < T2 < T10 & T2 > T22	Mod	Mod	Closed	Closed	Open	Closed	Open	Off	On	Product air is divided between collectors and storage to maintain T2 = T22.
4.	T2 > T10 & D2 is not fully open	Open	Closed	Mod	Mod	Open	Closed	Open	On	On	Excess air thru collectors maintains T2 = T10.
5.	T2 > T10 & D2 is fully open	Open	Closed	Open	Open	Mod	Mod	Open	On	On	Maximum air thru collectors - bypass air maintains T2 = T10
6.	T2 < T25 & T22 > T25	Closed	Open	Closed	Closed	Open	Closed	Open	Off	On	All product air thru storage.
7.	T2 < T25 & T22 > T10 (dashed line)	Closed	Open	Closed	Closed	Mod	Mod	Open	Off	On	Product air divided between storage & bypass to maintain T2 = T10.
8.	Dryer door open & T2 > T22	Off	Off	Mod	Mod	Off	Off	Closed	On	Off	Modulate D2 to maintain T2 = T22.
9.	Dryer door open & T2 < T22	Off	Off	Off	Off	Off	Off	Off	Off	Off	Shut down system.

Table 1
Temperature Control Modes

added to storage. Only the "pump around" loop operates in mode 8. D2 is modulated appropriately to maintain collector temperature T2 equal to storage temperature T22. If T2 is less than T22 no heat can be added to storage and the entire system is shut down. This is mode 9.

3. CONTROL LOGIC: Figure V-E-2 shows the implementation of the control modes defined in Table 1. The various control modes are identified at the appropriate logic gates at the centerline of Figure V-E-2. Input sensor signals are shown at the top while output control signals are shown at the right. Besides T/C and D/P signals, inputs include feedback signals from the damper motor limit switches. These limit switches turn off the controller power to the damper motors when the dampers have reached their open or closed limits. Much of the control can be realized using simple "AND/OR" logic as Figure V-E-2 shows. However, some arithmetic functions must be performed in order to generate temperature and pressure error signals whose magnitudes and/or signs initiate the various control functions.

For example, mode 1 control is initiated by the error (difference) signals ϵ_1 and ϵ_2 , where

$$\epsilon_1 = T2 - (T25 - \Delta_1)$$

$$\epsilon_2 = T22 - (T25 - \Delta_1).$$

If ϵ_1 and ϵ_2 are both less than zero, it means that both the storage and the collectors are cold, and the entire solar heating system should be by-passed. Hence a logical "1" signal is generated at the "yes" outputs of the decision blocks for ϵ_1 and ϵ_2 . The logical "1" signals at the input to the MODE 1 "AND" gate generate a logical "1" at its output (if the tunnel 12 fan contacts are open). This logical "1" now appears at the "OR" gates for initiating the following motor action;

1. D1A - close
2. D1B - open
3. D2A - close
4. D2B - close
5. D4A - close (D4B - open)
6. D5 - close
7. F1 - on
8. F2 - off

The gates immediately below the MODE 1 "AND" gate labeled D1, D2, D4, and D5 are used to shut off the damper motor actions when the respective motors reach the limits of their travel.

An example of modulating control occurs in mode 3 where the D1 dampers are modulated to maintain collector temperature T2 equal to storage temperature T22 as discussed in the previous section. Mode 3 occurs when the collector temperature T2 is greater than

the fresh air T25 and when T2 is less than the T10 setpoint and when T2 is less than the storage temperature T22 and when the tunnel 12 fan contacts are open. The logical "1" out of the MODE 3 "AND" gate appears at the "OR" gates of D2A/D2B (i.e. closing them), D4 (i.e. opening D4A and closing D4B), D5 (i.e. opening it), F1 (i.e. turning it on), and at F2 inverted to logical "0" (i.e. turning it off). Also, the logical "1" output of the MODE 3 "AND" gate initiates modulation of the D1 dampers as shown by being put into two MOD PERMIT "AND" gates at the two D1 MOD control sequence lines towards the bottom in Figure V-E-2. Either the upper or the lower MOD PERMIT "AND" gates are turned on, depending on the sign of the error signal between T2 and T22. If T2 is less than T22, less air should flow through the collectors and more should flow through storage. Therefore, D1A should be closed slightly and D1B should be opened. This is permitted by the upper MOD PERMIT gate turning on, energizing the normally open relay, and permitting the error signal between T2 and T22 (ϵ_a) to be multiplied by a gain in K_a and then put into a timer whose time setting is $K_a \epsilon_a$. Thus the timer setting is proportional to the error between T2 and T22. The timer puts out a pulse of duration proportional to the error between T2 and T22. This pulse is used to drive dampers D1A and D1B in the appropriate way as discussed above. The opposite control action occurs if T2 is greater than T22. The proportional control for this case is shown by the sequence line containing the lower

MOD PERMIT gate. The output of the lower D1 MOD gate drives the D1 dampers in the opposite direction for a duration proportional to the error signal ϵ_D .

The arithmetic boxes in Figure V-E-2 show that differences between temperatures are to be taken with a "delta" or biasing value added or subtracted from the "reference" temperature. These deltas provide deadbands such that excessive control action (i.e. hunting) will be avoided either between modes or between open or close modulating action on the dampers. These deltas should be field adjustable. Also, the proportional gains (K_a , K_D , etc.) should be field adjustable, as well as the T10 and P₂ setpoints. These items will serve as tuning parameters when the control system is started up.

4. CONTROLLER INPUT/OUTPUT: Figure V-E-2 also illustrates the input signals to which the control logic must properly respond and the output signals required from the logic. The characteristics of those input/output signals are defined in this section.

Table 2 defines the characteristics of the controller inputs. The thermocouple inputs will require a reference junction at the controller. The temperature ranges sensed by the thermocouples will be 32° to 300°F. The corresponding arithmetic resolution in the controller should be $\pm 1^\circ\text{F}$. The P₁ signal will be 4-20mA DC

calibrated over the range of about 0.8 to 1.2 inches of water with a nominal value of 1 in. water column corresponding to 20,000 cfm air flow. The calibration will be done in the field because the pressure drop vs. flow characteristics of the heat recovery wheel are best established after installation in the actual system. The tunnel 12 fan contacts and the damper feedback signals are all 115 VAC line voltage on/off signals as Table 2 shows.

Table 3 defines the controller output signal requirements.

Table 4 shows the specifications for the sensors and the damper actuators (i.e. operators). These items are to be supplied by the facilities contractor. They are presented here for informational purposes.

SIGNAL DESIGNATOR	NATURE OF SIGNAL
T2A	Type J thermocouple (T/C) analog
T2B	Type J T/C analog
T22	Type J T/C analog
T7	Type J T/C analog
T25	Type J T/C analog
T10	Analog voltage calibrated over temp. range 100°F to 200°F.
P ₁	4-20 mADC analog
P ₂	Setpoint entered at controller
Tunnel 12 fan contacts	0 or 115 VAC on/off
D1A - open	0 or 115 VAC on/off
D1A - closed	" "
D1B - open	" "
D1B - closed	" "
D2A - open	" "
D2A - closed	" "
D2B - open	" "
D2B - closed	" "
D4A - open	" "
D4A - closed	" "
D5 - open	" "
D5 - closed	" "
D3 - open	" "
D3 - closed	" "

TABLE 2
CONTROLLER INPUTS

CONTROL SIGNAL DESIGNATOR	NATURE OF SIGNAL
D1A - Close "ON"	115 VAC, 0.2 Amp damper motor drive
D1B - Open "ON"	"
D1A - Open "ON"	"
D1B - Close "ON"	"
D2A - Open "ON"	"
D2B - Open "ON"	"
D2A - Close "ON"	"
D2B - Close "ON"	"
D4A - Open "ON"	"
D4A - Close "ON"	"
D5 - Open "ON"	"
D5 - Close "ON"	"
D3 - Open "ON"	"
D3 - Close "ON"	"
F1 - "ON"	115 VAC, energizes relay coil
F2 - "ON"	115 VAC, energizes relay coil

TABLE 3
CONTROLLER INPUTS

TEMPERATURE SENSORS:

Range to be covered ;	32°F to 300°F
Type of sensor ;	thermocouple, type J, iron-constantan, grounded
Suggested vendor ;	Omega Engineering..
Type of construction;	heavy duty cast iron screw type lock head thermocouple assembly with stainless steel sheath; must be suitable for outdoor use
Size ;	1/8 in. sheath diameter
Number required ;	5
Designation ;	T2(2), T22, T7, T25

DIFFERENTIAL PRESSURE SENSOR:

Range to be covered ;	0" to 3" H ₂ O
Type of sensor ;	Δ P to electrical transmitter, with 4-20 mA DC analog output
Suggested vendor ;	Leeds and Northrup
Suggested model ;	L&N 1912-2
Type of construction;	must be suitable for outdoor use
Power ;	115 VAC
Special requirements;	high pressure port vented to atmosphere

DAMPER OPERATORS (FOR RECTANGULAR DAMPERS):

Type of operator ;	reversing motor, floating action
Stroke ;	90 degrees
Power ;	115 VAC

TABLE 4

SENSOR AND DAMPER OPERATOR SPECIFICATIONS

Torque	;	150 in.-lb. normal running, 300 in.-lb. breakaway
Suggested vendor	;	Honeywell
Suggested model	;	Modutrol M944K with Q605 damper linkage
Type of construction;		must be suitable for outdoor use
Special requirements;		motor/damper will be operated as open/close or modulating depending on system mode. Motor should hold last position when motor power is turned off. Auxiliary limit switches are required. The operator for damper D4 must actuate two separate dampers ganged mechanically.
Number required	;	6
Damper designation	;	D1(2), D2(2), D4, D5

DAMPER OPERATOR (FOR VORTEX DAMPER ON F1 FAN):

Type of operator	;	reversing motor, floating action
Stroke	;	160 degrees, field adjustable
Power	;	115 VAC
Torque	;	200 in.-lb. normal running, 330 (or more in.-lb. breakaway
Suggested vendor	;	Honeywell
Suggested model	;	Actionator M630A
Type of construction;		must be suitable for outdoor use
Special requirements;		motor/damper will be operating in modulating mode continuously. Motor should hold last position when motor power is turned off. Auxiliary limit switches are required.
Number required	;	1
Damper designation	;	D3

TABLE 4
SENSOR AND DAMPER OPERATOR SPECIFICATIONS (continued)

F. SYSTEM SIMULATION:

The system described in the previous section is considered to be a continuous flow system with deterministic input conditions. To determine the dynamic system performance, equations relating input and output variables were developed for each component. These equations were derived by writing energy balances for the components. In general, the equations for the components are coupled, so that simultaneous solution is necessary. For example, the collector performance was determined to be a function of the temperature and flow rate of the fluid entering the collectors, as well as the other environmental factors. The independent variable, time, only appeared implicitly in these equations via determination of environmental input variables. Thus, heat capacity effects in the collectors were neglected. Coupling of the collector equations with those describing the rock storage occurs because the collector inlet temperature depends on the storage exit temperature (i.e. in certain modes of operation), the collector exit temperature determines the storage exit temperature and not only depends on the storage inlet conditions at any moment, but also on the recent past history of storage inlet conditions. Thus, it is not possible to separate storage performance from that of the collector.

System input conditions (i.e. insolation, mode changes, ambient air temperature, etc.) were assumed to be constant over each

successive 15 minute period of a 24 hour simulated day, as shown in Figure III-3. However, the storage finite difference equations were solved every one-and-one-half minutes for reasons of solution stability. Operational mode of the system was determined at the beginning of each 15 minute period. Energy quantities were integrated over a 24 hour period of time to assess energy distribution.

Although the model would accommodate weather tape type input simulating day-to-day climatic variation, computer and design time limitations constrained the studies to the simulation of repeating periodic day conditions. Such studies were pursued for "typical days", for various months and for geographic locations. Because of the transient nature of the storage profile, it was found necessary to run the program through a five day simulation period in order to reach steady periodic storage temperature profiles.

Instantaneous insolation (i.e. direct plus diffuse) values were determined by use of the ASHRAE algorithm (i.e. equation 5 et seq, Ch.22, ASHRAE Handbook of Fundamentals, 1972). The daily values of insolation determined by this means, were normalized to the Fresno insolation record in the case of the design site calculations and to the values supplied by ERDA in the case of the Albuquerque and Omaha calculations.

A sample portion of the simulation program output from which Figure III-5 and III-6 were plotted is shown in Figure V-F-1. This output tabulates daily total integrated energy input and output, and instantaneous values of system temperature (i.e. refer numbers and letters to Figure III-1), as well as collector performance, system flowrates, pressure drops, and power requirements.

The program is of substantial magnitude at around 2000 Fortran statements, of which nearly one-third are comments provided to aid readability and revision. The program is deliberately modular, with fifteen separate compilations. A number of philosophic considerations dominated program development and are important to the degree of confidence placed in program results, although any code of such magnitude constructed in such a short time span can be expected to have additional "bugs" crop up from time to time.

Of the fourteen subroutines, nine were developed independently and exercised extensively with separately written driver programs. The drivers tested limiting conditions as well as expected parameter ranges and provided considerable confidence in the individual segments prior to their integration into the system.

Nearly all communication is handled through COMMON storage and

identical COMMON blocks are used in each communicating routine to minimize this frequent source of error and to encourage consistent and meaningful mnemonic variable naming. Minimal language subsets allowed development under IBM FORTRAN G and WATFIV and an easy transplant to a CDC Cyber system in a time share environment. Subsequent modifications related to data input for timeshare use would require revision before returning to a batch environment.

In the interests of time savings, some of the subroutines require separate initializing calls. Debugging "dump" options have been left intact in most cases. As integration progressed and output formatting improved, more and more variables were added to COMMON communication areas, and this had some influence on final coding. Minor code optimizing opportunities were frequently ignored where time constraints precluded retesting with the drivers.

One arithmetic difficulty occurred with the language transplants. Minor inconsistencies in numbers of significant figures dictated some code sequences when the WATFIV compiler "helpfully" switched into double precision during some of the spherical trigonometric calculations associated with solar insolation. In general, these were held to single precision, and conversion to the much longer word length of the Cyber system then caused further problems near angular extremes. These were bypassed

with a function to truncate numbers greater than plus or minus one to seven significant figures prior to entering ARCSIN or ARCCOS routines, and no further trouble ensued.

Problems associated with numerical method instability led to the use of a relatively small integration step size in the ROKBED subroutine. Once a step size was selected which gave satisfactory results over a broad range of parameter values, no further attempts at step organization were made. This is a potentially profitable area of investigation if the program is to be used for further production processing, since considerably more than 50% of the running time is associated with the ROKBED routine.

The program has a great deal of flexibility in component description within the overall system design. Some sixty-five input parameters have been assigned default values which can be individually and optionally overridden on input for each model run. Many additional constants and tables are compiled in place but easily changed. In addition to running the program for widely varying system component characteristics and capacities, dates, climates, and geographic locations, patches were added to study parameterized performance of particular components within the overall system context.

A major system reconfiguration would require corresponding program

reconfiguration and revised operational mode descriptions. The modular program construction should prove helpful in such an effort. The model should be recognized to be a model of the particular system design developed for the task at hand, and there is no intent of direct applicability outside the current context.

NET COLLECTOR OUTPUT= 14285651.										GRS FAN ENERGY USE (1)= 1399684, (2)= 1014459													
NET SUN INPUT= 41106173.										NET ENERGY SUPPLIED= 18091682													
HEAT RECOVERED= 20483737.										NET BURNER USE= 1888332													
SOLAR= 78.963%, ELECTRIC= 12.010%, BURNER= 10.438%										Energy collected								Fan power and Pressure rise					
Hour of day with four 15 minutes segments.										Energy in storage													
Mode										Fuel Energy used													
HR	M ₁	1	2	7	14	20	22	0	P	ST	SUN	ETA	SHD	* FLOWRATE *	2	17	22	COL	STOR	BRN	* F1 *	* F2 *	
Percent sunlight																							
* FLOWRATE *																							
6	6	113.6	57.2	131.3	111.2	113.5	129.8	57.2	115.1	116.3	0.0	0	1.0	0.0	0.0	4.3	0	-3667	95	20	2.9	0	0.0
6	6	113.6	57.6	129.8	111.3	113.6	128.4	57.6	111.3	116.0	1.6	0	1.0	0.0	0.0	4.3	0	-3743	103	20	2.9	0	0.0
6	6	113.7	58.1	128.4	111.3	113.6	127.0	58.1	111.3	115.8	13.1	0	1.0	0.0	0.0	4.3	0	-3812	110	20	2.9	0	0.0
6	6	113.7	58.5	127.0	111.4	113.7	125.7	58.5	111.4	115.6	30.3	0	1.0	0.0	0.0	4.3	0	-3873	118	20	2.9	0	0.0
7	6	113.8	58.9	125.7	111.4	113.8	124.4	58.9	111.4	115.3	49.4	0	1.0	0.0	0.0	4.3	0	-3928	124	20	2.9	0	0.0
7	6	113.9	59.7	124.4	111.6	113.9	123.3	59.7	111.6	115.2	69.2	0	1.0	0.0	0.0	4.3	0	-3977	131	21	2.9	0	0.0
7	6	114.0	60.5	123.3	111.7	114.0	122.3	60.5	111.7	115.0	88.9	0	1.0	0.0	0.0	4.3	0	-4019	136	21	2.9	0	0.0
7	6	114.1	61.2	122.3	111.8	114.1	121.3	61.2	111.8	114.9	108.5	0	1.0	0.0	0.0	4.3	0	-4057	142	21	2.9	0	0.0
8	3	114.2	122.3	122.3	111.9	114.1	121.3	62.0	123.9	114.9	127.5	6	1.0	4.2	0.0	.1	40	-4059	142	21	3.2	0	0.0
8	2	114.4	129.9	129.9	112.0	114.1	121.3	63.1	131.1	114.9	146.0	11	1.0	4.3	0.0	0.0	79	-4059	103	21	3.2	0	0.0
8	2	114.6	137.1	137.1	112.2	114.1	121.3	64.3	138.1	114.9	163.7	14	1.0	4.3	0.0	0.0	115	-4060	65	21	3.2	0	0.0
8	2	114.8	143.9	143.9	112.4	114.1	121.3	65.4	144.8	114.9	180.6	17	1.0	4.3	0.0	0.0	149	-4060	31	21	3.2	0	0.0
9	4	115.0	149.8	149.8	112.5	114.1	123.4	66.5	150.5	114.9	196.7	19	1.0	4.5	0.0	.1	184	-4054	1	21	3.2	19	.3
9	4	118.7	149.4	149.4	112.7	114.0	137.9	67.8	149.1	115.3	211.7	25	1.0	7.2	0.0	2.8	261	-3935	2	22	3.7	32	.9
9	4	117.5	149.7	149.7	112.9	113.9	145.9	69.1	148.8	116.0	225.6	30	1.0	8.6	0.0	4.3	330	-3753	1	22	4.0	37	1.2
9	4	117.0	149.6	149.6	113.1	113.8	148.7	70.4	148.4	116.9	238.5	34	1.0	10.2	0.0	5.9	394	-3503	2	23	4.4	41	1.7
10	4	116.7	149.5	149.5	113.2	113.7	149.4	71.7	148.1	118.1	250.1	37	1.0	11.6	0.0	7.3	452	-3193	2	24	4.8	45	2.2
10	4	116.5	149.6	149.6	113.4	113.6	149.5	73.1	148.0	119.4	260.5	40	1.0	12.9	0.0	8.5	504	-2829	2	25	5.2	48	2.8
10	4	116.4	149.2	149.2	113.6	113.6	149.2	74.6	147.6	121.0	269.6	42	1.0	14.2	0.0	9.9	554	-2410	4	26	5.6	52	3.4
10	4	116.3	149.5	149.5	113.9	113.5	149.5	76.1	147.8	122.7	277.4	44	1.0	15.0	0.0	10.7	593	-1951	2	27	5.9	56	3.8
11	4	116.4	149.2	149.2	114.1	113.6	149.2	77.5	147.5	124.6	283.8	46	1.0	16.2	0.0	11.8	630	-1449	4	30	6.4	63	4.3
11	4	116.6	150.5	150.5	114.2	113.6	150.4	78.9	148.6	126.5	288.8	46	1.0	16.2	0.0	11.8	651	-930	0	30	6.4	63	4.3
11	5	116.6	151.4	150.0	114.4	113.7	151.4	80.3	149.5	128.5	292.4	47	1.0	16.2	.2	12.0	668	-391	0	29	6.3	64	4.3
11	5	116.7	152.2	150.0	114.6	113.9	152.1	81.7	150.2	130.6	294.6	47	1.0	16.2	.3	12.1	681	160	0	29	6.2	65	4.3
12	5	116.9	152.8	150.1	114.8	114.1	152.7	83.1	150.6	132.7	295.3	48	1.0	16.2	.3	12.2	689	720	0	29	6.2	65	4.4
12	5	117.1	153.0	150.1	115.0	114.3	153.0	84.1	150.9	134.8	294.6	48	1.0	16.2	.4	12.2	691	1283	0	29	6.2	65	4.4
12	5	117.3	153.1	150.1	115.1	114.7	153.1	85.1	150.9	136.9	292.4	48	1.0	16.2	.4	12.2	688	1842	0	29	6.2	66	4.4
12	5	117.6	153.1	150.1	115.3	115.4	153.1	86.2	150.8	138.9	288.8	48	1.0	16.2	.4	12.2	680	2392	0	29	6.2	66	4.4
13	5	118.2	152.9	150.1	115.4	116.5	152.9	87.2	150.7	140.9	283.8	48	1.0	16.2	.4	12.2	668	2926	0	29	6.2	66	4.4
13	5	119.0	152.7	150.1	115.5	118.1	152.7	87.9	150.5	142.8	277.4	48	1.0	16.2	.3	12.2	647	3433	0	29	6.2	66	4.4
13	5	120.3	152.5	150.1	115.6	120.5	152.6	88.6	150.5	144.6	269.6	47	1.0	16.2	.3	12.2	620	3906	0	29	6.2	66	4.4
13	5	122.1	152.5	150.1	115.7	123.5	152.5	89.3	150.6	146.2	260.5	46	1.0	16.2	.3	12.2	585	4337	0	29	6.2	66	4.4
14	5	124.4	152.7	150.1	115.8	127.2	152.7	90.0	151.0	147.6	250.1	45	1.0	16.2	.3	12.2	543	4720	0	29	6.2	66	4.4
14	5	127.2	152.9	150.1	115.8	131.2	152.8	90.2	151.4	148.8	238.5	42	1.0	16.2	.4	12.2	493	5049	0	29	6.2	67	4.4
14	5	130.2	153.0	150.1	115.9	135.2	153.0	90.5	151.8	149.8	225.6	40	1.0	16.2	.4	12.2	436	5321	0	29	6.2	67	4.4
14	5	133.3	152.9	150.1	115.9	138.9	152.9	90.7	152.1	150.7	211.7	37	1.0	16.2	.4	12.2	376	5537	0	29	6.2	68	4.4
15	5	136.1	152.4	150.1	115.9	142.2	152.5	91.0	151.9	151.2	196.7	33	1.0	16.2	.3	12.2	314	5695	0	29	6.2	68	4.3
15	5	138.3	151.3	150.0	115.9	144.8	151.4	90.7	151.1	151.6	180.6	28	1.0	16.2	.2	12.0	249	5795	0	29	6.3	67	4.3
15	4	140.0	149.6	149.6	115.9	147.0	149.7	90.5	149.7	151.8	163.7	23	1.0	16.2	0.0	11.8	184	5838	2	30	6.4	66	4.3
15	3	119.2	149.9	149.8	115.8	138.0	149.8	90.2	149.2	151.6	146.0	16	1.0	3.2	0.0	1.1	116	5796	0	21	3.0	0	0.0
16	3	118.2	149.7	149.8	115.8	124.8	150.2	90.0	153.0	151.0	127.5	1	1.0	.2	0.0	4.1	7	5640	1	21	2.9	0	0.0
16	6	118.1	140.2	150.0	115.7	120.2	150.6	89.4	141.6	150.4	108.5	0	1.0	0.0	.0	4.3	0	5473	0	21	2.9	0	0.0
16	6	118.0	128.5	150.0	115.6	118.7	151.0	88.8	129.0	149.8	88.9	0	1.0	0.0	.1	4.2	0	5306	0	21	2.9	0	0.0
16	6	117.9	88.2	150.0	115.5	118.2	151.3	88.2	115.5	149.2	69.2	0	1.0	0.0	.1	4.2	0	5140	0	20	2.8	0	0.0
17	6	117.8	87.5	150.0	115.5	117.9	151.6	87.5	115.5	148.5	49.4	0	1.0	0.0	.2	4.1	0	4972	0	20	2.8	0	0.0
17	6	117.7	86.6	150.0	115.3	117.7	151.8	86.6	115.3	147.9	30.3	0	1.0	0.0	.2	4.1	0	4805	0	20	2.8	0	0.0
17	6	117.5	85.7	150.0	115.2	117.6	152.0	85.7	115.2	147.3	13.1	0	1.0	0.0	.2	4.1	0	4637	0	20	2.8	0	0.0
17	6	117.4	84.7	150.0	115.1	117.5	152.2	84.7	115.1	146.7	1.6	0	1.0	0.0	.2	4.1	0	4468	0	20	2.8	0	0.0

Figure V-F-1:
Sample Simulation Output

G. MONITORING SYSTEM:

The monitoring system for the solar dehydration facility will allow on site observation with remote reading and analysing. The system will consist of an on site scanner, Mag Tape recorder, a Modem and fifty varying transducers. The scanner can be controlled remotely by telephone and the Mag Tape can be dumped every few days for analysing and permanent recording data.

Figure V-G-1 shows the general placement of the transducers needed for analysing the solar crop drying system. The description of the various transducers are shown in Table 1. The bulk of the transducers will be Copper-Constantan Thermocouples which are inexpensive, reliable and accurate enough for system evaluation. When fans F1 and F2 are on, an induction coil will be used to record the ampere load. Static and velocity pressures will be recorded from pressure taps and pressure transducers located on straight runs of ducting and at the fans. Damper positions will be recorded by slidewire pots attached to the damper drive. Ambient conditions will be monitored by wet and dry bulb temperatures, total and diffused radiation, and by wind speed and direction.

The various transducer leads will be fed through some 2,000 ft. of plastic pipe to a central location near the storage tank, where an instrument trailer will be housed for the duration of the project.

DUCT PLAN
CYLINDRICAL STORAGE

1. Ampers		
A101	Amp of F1	Moore Ind. Model ACT 0-150 AMP 0-10VDC/117VAL
A102	Amp of F2	Moore Ind. Model ACT 0-150 AMP 0-19VDC/117VAL
2. Temperature T.C. Type T.		
T101	Dry Bulb	120'
T102	Wet Bulb	120'
T103	Temp. into system	100'
T104	Temp. out of H.R. to collector	100'
T105	Out of tunnel to H.R.	100'
T106	Out of H.R. from tunnel	110'
T107	From H.R. duct to collector duct	50'
T108	Under heat storage	40'
T109	To collector N	250'
T110	To collector S	250'
T111	From collector N1	80'
T112	From collector N7	160'
T113	From collector N15	230'
T114	From collector S1	80'
T115	From collector S7	160'
T116	From collector S15	230'
T117	To storage from collector N	60'

TABLE 1
TRANSDUCER NOMENCLATURE

T118	To storage from collector S	40'	
T119	Top of storage	50'	
T120	From storage to tunnel	100'	
T121	In tunnel	200'	
T122	Storage 2/3 up N.C.	60'	
T123	Storage 2/3 up W.C.	60'	
T124	" " " C.C.	60'	
T125	" " " E.C.	60'	
T126	" " " S.C.	60'	
T127	" 1/3 " N.S.	60'	
T128	" " " W.C.	60'	
T129	" " " C.C.	60'	
T130	" " " E.C.	60'	
T131	" " " S.C.	60'	
		3230 ft.	2,000 ft. . of conduit

3. Milivolt readings

M101	Solar total, Epply #2	
M102	Solar difused Epply #2 with shadow ring	
M103	Ps at F1	Tavis Ind. P 8 Cs 0-6.0
M104	Ps at F2	Tavis Ind. P 8 Cs 0-6.0
M105	Pv from H.R. to storage	Ellison Mfg., Annubar Mod 75 Tavis P 8 Cs 0-2.77

TABLE 1, cont.
TRANSDUCER NOMENCLATURE

M106	Pv from H.R. to collectors	same as M105
M107	Pv from storage to tunnel	Ellison Mfg., Annubar Mod 76 Tavis P 8 Cs 0-2.77
M108	Pv from storage to tunnel	same as M105
4. Voltage Readings		
V101	Wind	Weather Measure Corp. W101
V102	Speed	P DC/540
V103	Position of D1A	Moore Ind. Slide Wire Pot PTT/10K/0-10VDC/117VAL STD/X
V104	Position of D1B	same as V103
V105	" " D2	" " "
V106	" " D3 inlet to F1	" " "
V107	" " D4A	" " "
V108	" " D4B	" " "
V109	" " D5	" " "

TABLE 1 cont.
TRANSDUCER NOMENCLATURE

VI. CONSTRUCTION COSTS

A. NEED FOR A CONSTRUCTION COST ESTIMATING SYSTEM:

The preparation of cost estimates in the construction industry is normally undertaken by the most experienced and senior staff members of a contracting company, with the assistance of detailed construction cost estimating manuals and virtually unlimited access to previous job costing information. Even under these favorable conditions, construction cost estimating is considered to be a most exacting task, requiring, in addition to experience and professional skill, an innate sense on the part of the estimator to foresee potential construction problems, operation sequences and future patterns of demand and supply in the labor and material sectors of the market.

The preparation of accurate cost estimates for components and systems for which no case history exists, is extremely difficult and often impossible. In the past, few solar energy research and development programs have included more than cursory studies of actual fabrication and implementation costs. Many of these cost estimates were prepared by researchers with no practical experience in the construction industry. In relatively few cases, where construction cost estimates were included in the published results of solar research and development projects, is there evidence to suggest that these estimates were used as a research tool during the evaluation of the design alternatives.

Previous investigations undertaken by this research and development team under National Science Foundation grant RANN/SE/G142944, in 1975, to determine the cost effectiveness of industrial solar crop drying installations indicated, that for mechanical crop dehydration installations in which more than 25% of the heat is supplied by means of solar collectors, costs tend to increase sharply due to the need for heat storage facilities.

Based on these findings and the objective of achieving a greater than 60% solar heat contribution, this research and development team could assume at the commencement of the investigation that the final cost effectiveness of the prototype solar dehydration system would be governed not only by the successful optimization of the various system parameters, such as collector tilt, air flow, storage temperature and heat recovery, but more critically by material selection and the development of low cost construction designs. In short, it was considered imperative that an accurate construction cost estimating tool should be available during both the system analysis and component design stages of the investigation.

B. ESTABLISHMENT OF COST ESTIMATING SYSTEM CRITERIA:

A number of design criteria were established for the development of the cost estimating system to extend its functions beyond the preliminary estimating capabilities of a normal construction cost estimating procedure. In particular, it was agreed that these

additional functions should include the establishment of component cost priorities and the ability to evaluate, the cost effectiveness of any individual item in relationship to any other item; the cost effectiveness compared to the component of which it forms a part; and in comparison to the solar system as a whole. More specifically the following design criteria were established:

- 1) Division of the construction project into an appropriate number of main components and sub-components, with the ability to establish separate material, labor, equipment and item quantities and rates at the subcomponent level.
- 2) Procedure for calculating the actual cost and the percentage cost at each level of quantity measurement.
- 3) Ability to update the material, labor, equipment and item quantities and rates of any or all sub-components, with minimum manual effort.
- 4) Provision of a permanent record of the precise nature, date and person responsible, for all changes in cost information.
- 5) Establishment of a subcontractor reference file, with the ability to cross-reference at the main component and subcomponent levels.

6) Minimization of the need for manual computation
and administration tasks.

It was recognized that the requirements for minimal manual intervention, repetitive computations and data management would be most easily satisfied by a computer based cost estimating system. In view of the desirability for on-line access to the cost information data base, an interactive timeshare computer system was selected.

C. DESIGN OF THE COST ESTIMATING SYSTEM (ASEID1):

As shown in the flow diagram (Figure VI-1), the construction cost estimating system was designed to consist of a single interactive computer program, ASEID 1, providing access to a data base consisting of four information files, namely; COMP.AR1 and COMP.AR2, containing the description, quantity, unit, rate and computation results for each subcomponent element; UNCOMP.DAT containing a permanent record of all cost information updates; and SUB1.DAT, an index file of sub-contractor information referenced by component and subcomponent.

For the sake of simplicity all data management operations, such as entry, storage, retrieval, correction and output reports are made available to the user through the program ASEID 1, in the form of options illustrated in Figure VI-2.

1. SITE PREPARATION:	4.3 Storage bin walls and floor
1.1 Collector site	4.4 Storage bin roof
1.2 Main duct site	4.5 Heat storage material
1.3 Heat storage site	4.6 I/O ducts
1.4 Tunnel dryer surround	4.7 I/O duct insulation
	4.8 I/O duct dampers
2. SOLAR COLLECTORS:	4.9 Storage bin insulation
2.1 Excavation	4.10 Fans
2.2 Footings	
2.3 Collector supports	5. DRYER FACILITY MODIFICATIONS:
2.4 Collector air ducts	5.1 Ducts
2.5 Collector glazing	5.2 Duct dampers
2.6 Collector weather proofing	5.3 Fans
2.7 Miscellaneous hardware	5.4 Duct insulation
2.8 Collector insulation	5.5 Heat exchange
2.9 Duct dampers	5.6 Miscellaneous paintwork
2.10 I/O Manifold ducts	
2.11 I/O Manifold insulation	6. CONTROL SYSTEMS:
	6.1 Thermistors
3. MAIN DUCT SYSTEM:	6.2 Wiring, conduits
3.1 Excavation	6.3 Air speed meters
3.2 Duct support footings	6.4 Switches, fuses, etc.
3.3 Duct supports	6.5 Motors
3.4 Main ducts	6.6 Differential thermostats
3.5 Main duct insulation	
3.6 Duct dampers	7. OVERHEAD AND CONTINGENCIES:
3.7 Main duct paintwork	7.1 Insurances
	7.2 Permits
4. HEAT STORAGE SYSTEM	7.3 Contingency
4.1 Excavation	7.4 Clean-up
4.2 Storage bin footings	7.5 Profit margin

TABLE 1

Subdivision of Construction Project
into Main Components and Subcomponents

The main component and subcomponent divisions shown in Table 1 were selected on the basis of the principal system elements defined earlier, with the addition of the separate components "site preparation" and "overhead and contingencies". It was felt that the degree of detail indicated at the subcomponent level (Table 1) would be necessary for the accurate pricing of alternative design solutions. The division of each subcomponent into separate material, labor, equipment, and item elements was dictated partly by the costing requirements established by ERDA and by the need for cross-referencing subcontractors on the basis of their ability to participate in only a portion of a subcomponent.

Final guidelines for the programmer included the stipulation that execution of the program and, in particular, the updating of data files should be sufficiently simple to be undertaken by a user without previous computer experience, and should require no user documentation in addition to instructions included in the program ASEID 1 itself.

D. IMPLEMENTATION OF THE COST ESTIMATING SYSTEM (ASEID1):

The development and coding of the construction cost estimating system was completed by an experienced student programmer in sixty-one hours, extending over a period of two weeks. For ease of access the program was implemented on an existing DEC PDP

Figure VI-1

Flow Diagram of Computer Based Construction Cost Estimating System

Figure VI-2
Program ASEID1 Principal User Options

U. T. - USER Terminal
In. - Input
Out. - Output
Proc. - Processor
Opt. - Optional

Figure VI-3

Proportional Cost of Major Construction Components

11/45 minicomputer timeshare system, supporting thirty-two ports with an 8K maximum user partition. BASIC-PLUS was chosen as the programming language since it was the only completely interactive language available on the computer system at that time. The program ASEID 1 consists of 263 statements and five associated data files, requiring a total of 136 blocks (i.e. approximately 270,000 alphameric characters) of disk storage.

One of the axioms established early in the research investigation was that the construction cost estimating system should be utilized not only during the design phase, but also during the theoretical analysis and optimization stages undertaken on the basis of a separate computer simulation model. Accordingly, the ASEID 1 program was implemented during the early stages of the development of the computer simulation model, and immediately focussed attention on the proportional costs of the various system components. For example, it became clear before the second update of the cost information data files that consideration should be given to the deletion of the heat recovery unit in favor of reduced ducting costs and slightly increased solar collector costs. The program also indicated that special attention would have to be given to the selection of thermal insulation materials and the development of suitable attachment details. With the cost of thermal insulation constituting approximately one-third of the cost of the entire construction project (i.e. approximately \$360,000 total construction

cost), the analysis team was persuaded to investigate the relationships between collector area, heat recovery and thermal insulation for a number of additional system configurations, utilizing the computer simulation model.

The ASEID-1 program also served as an excellent communication link between the University analysis team and the TRW design group. Design proposals made by either team were quickly evaluated, with a detailed cost estimate available within hours. In particular, the availability of percentage costs (Table 2) at the component and subcomponent levels proved to be of great benefit in team discussions dealing with design modifications and system variations.

The establishment, early in this research and development project, of the proportional cost of each major construction component had considerable influence on the scheduling and planning of individual research tasks. Firstly, it provided a convenient index for the efficient allocation of time and manpower, and secondly, it very much influenced the completion dates set for the various analysis, design and documentation stages of Phase I of the project. After a relatively brief exposure to the construction cost estimating system, it became apparent that individual team members acquired the ability to judge the general effect of design variations on the total cost of the project, without the need for a complete cost analysis. In particular, it was found to be of great value

Component and Subcomponent	Quantity	Unit	Rate (\$)	Cost (\$)	A (%)	B (%)
Heat Storage System				\$60201		18.4%
Excavation				\$ 340	1%	0.1%
MTR: 1' Deep Walls, Int. Supp.	56	cu.yd.	5.00	\$ 280	0%	0.1%
LBR: Set Out	6	HR.	10.00	\$ 60	0%	0.0%
Storage Bin Footings				\$ 2160	4%	0.7%
MTR: 1' x 3' Walls, Int. supp.	56	cu.yd.	30.00	\$ 1680	3%	0.5%
LBR: Pour & Finish	32	HR.	15.00	\$ 480	1%	0.1%
Storage Bin Walls and Floor				\$22577	38%	6.9%
MTR: 8" Conc. Blocks	4322	SQ.FT.	2.60	\$11237	19%	3.4%
LBR: Constr. Floor, Joists & M.G.	2160	SQ.FT.	0.60	\$ 1296	2%	0.4%
ITM: Conc. Slab, Joists & M.G.	2160	SQ.FT.	4.65	\$10044	17%	3.1%
Storage Bin Roof				\$ 4680	8%	1.4%
MTR: Prefab. Steel	2340	SQ.FT.	1.50	\$ 3510	6%	1.1%
LBR: Install beams, decking	2340	SQ.FT.	0.50	\$ 1170	2%	0.4%
Heat Storage Material				\$10100	17%	3.1%
MTR: Graded rock	926	cu.yd.	10.00	\$ 9260	15%	2.8%
LBR: Place, etc.	80	hr.	8.00	\$ 640	1%	0.2%
EQP: Conveyor rental	1	week	200.00	\$ 200	0%	0.1%
I/O Ducts				\$ 1520	3%	0.5%
MTR: Galv. Iron 6' Square	32	ft.	25.00	\$ 800	1%	0.2%
LBR: Assemble	48	hr.	15.00	\$ 720	1%	0.2%
I/O Duct Insulation				\$ 1824	3%	0.6%
MTR: 3"-8lb. mineral wool/alum	608	sq.ft.	3.00	\$ 1824	3%	0.6%
I/O Duct Dampers				\$ 2000	3%	0.6%
ITM: 2 to 4' Rnd. Duct Damper	2	No.	1000.00	\$ 2000	3%	0.6%
Storage Bin Insulation				\$12000	20%	3.7%
MTR: 2.5" Spray Polyurethane	8000	sq.ft.	1.50	\$12000	20%	3.7%
Fans (centrifugal)				\$ 3000	5%	0.9%
EQP: 40 h.p. centrifugal fan	1	40hp	3000.00	\$ 3000	5%	0.9%

KEY:

A = cost as a percentage of component cost

B = subcomponent cost as a percentage of total project cost

TABLE 2

Typical Cost Report for the "Heat Storage System" Component

by the investigators to have the ability to rapidly obtain individual material and labor costs by reference to past cost reports.

E. COST EVALUATION OF DESIGN ALTERNATIVES:

The ASEID 1 construction cost estimating system was utilized as an integral part of the design process in an effort to develop a minimum cost solar system. Attention was focussed, in particular, on three major components which constituted approximately 70% of the estimated total construction cost; namely, the solar collector array (40%), the main duct system between the collectors and the heat storage facility (10%), and the rock storage unit (20%).

1) SOLAR COLLECTOR ALTERNATIVES:

- a) INRYCO 3 in. deep (3NF) roof decking galvanized steel air ducts mounted in lengths of 200 ft. and 12 ft. wide on steel angle (1 1/2 in. x 1 1/2 in. x 3/16 in.) frames, set in concrete slab footings. Single glazed with corrugated Tedlar coated FILON panels and insulated with 3 lb./3 in. weatherproofed fiberglass board.
- b) Corrugated 1 3/4 in. deep aluminum (0.024 in.) roof decking air ducts mounted on plywood (3/8 in.) base and supported by redwood frames set in concrete pad footings. Single glazed with flat Tedlar coated FILON panels and insulated with 2 in. weather-

Solar Collector Construction Components	MATERIAL (\$/sq.ft.)	LABOR (\$/sq.ft.)	TOTAL (\$/sq.ft.)
Footings & Support Frames	0.32	0.18	0.50
Air Ducts	1.40	0.05	1.45
Glazing & Hardware	0.87	0.05	0.92
Collector Insulation	2.50	1.00	3.50
I/O Manifolds	1.08	0.30	1.38
	6.17	1.58	7.75
Contractor's Overhead & Profit (15%)			1.16
Cost per sq. ft. of Collector Area			\$ 8.91

Solar Collector Alternative (a)

Solar Collector Construction Components	MATERIAL (\$/sq.ft.)	LABOR (\$/sq.ft.)	TOTAL (\$/sq.ft.)
Footings & Support Frames	0.54	0.28	0.82
Air Ducts	1.06	1.06	2.12
Glazing & Hardware	1.14	0.46	1.60
Collector Insulation	0.91	0.62	1.60
I/O Manifolds	1.08	0.30	1.38
	4.73	2.72	7.52
Contractor's Overhead & Profit (15%)			1.13
Cost per sq. ft. of Collector Area			\$ 8.65

Solar Collector Alternative (b)

proofed (i.e. painted) polyurethane board. Manifold ducts insulated with 8 lb./3 in. mineral wool, complete with aluminum weatherproofing.

- c) Flat 28 g. glavanized steel 3 1/2 in. deep air ducts mounted in lengths of 180 ft. and 4 ft. wide on plywood (1/2 in.) base and supported by redwood frames set in concrete pad footings. Single glazed flate Tedlar coated FILON 542 panels set between prestressed triple 2 in. x 4 in. bolted redwood sideframes, and insulated with 3 in. weatherproofed sprayed polyurethane foam.

Solar Collector Construction Components	MATERIAL (\$/sq.ft.)	LABOR (\$/sq.ft.)	TOTAL (\$/sq.ft.)
Footings & Support Frames	0.32	0.17	0.49
Air Ducts	1.14	0.68	1.82
Glazing & Hardware	1.11	0.25	2.17
Collector Insulation	0.91	0.62	1.53
I/O Manifolds	1.08	0.30	1.38
	4.56	2.02	6.58
Contractor's Overhead & Profit (15%)			.99
Cost per sq. ft. of Collector Area			\$ 7.57

Solar Collector Alternative (c)

2) MAIN DUCT SYSTEM ALTERNATIVES:

- a) 400 ft. of galvanized steel air ductwork
(2 1/2 ft. to 6 ft. diameter range), supported
above ground on 2 in. diameter galvanized steel
pipe frames at 10 ft. centers for 150 ft., and
supported on concrete slab at ground level for
250 ft. The ductwork insulated with 3 in.
weatherproofed sprayed polyurethane foam and
including elbows and special fittings.

Main Duct System Construction Components	MATERIAL (\$/sq.ft.)	LABOR (\$/sq.ft.)	TOTAL (\$/sq.ft.)
Excavation, footings & Support Frames	0.06	0.04	0.10
Air Ducts	1.04	0.04	1.08
Duct Insulation	0.51	0.42	0.93
	1.61	0.50	2.11
Contractor's Overhead & Profit (15%)			0.32
Cost per sq. ft. of Collector Area			\$ 2.43

Main Duct System Alternative (a)

- b) 250 ft. of galvanized steel air ductwork (2 1/2 ft.
to 6 ft. diameter range) supported on concrete
slab at ground level; 150 ft. of double duct

insulated section located 5 ft. below ground level (spiral covered piping for external duct). Above ground duct section insulated with 3 in. weather-proofed sprayed polyurethane foam. All ductwork complete with elbows and special fittings.

Main Duct System Construction Components	MATERIAL (\$/sq.ft.)	LABOR (\$/sq.ft.)	TOTAL (\$/sq.ft.)
Excavation, Footings & Support Frames	0.12	0.05	0.17
Air Ducts	1.52	0.03	1.55
Duct Insulation	0.26	0.18	0.44
	1.90	0.26	2.16
Contractor's Overhead & Profit (15%)			0.32
Cost per sq. ft. of Collector Area			\$ 2.48

Main Duct System Alternative (b)

3) HEAT STORAGE UNIT ALTERNATIVES:

- a) Rectangular building with 8 in. concrete block walls, prefabricated steel roof and steel floor grading (3/4 R4) supported 1 1/2 ft. above ground level by concrete block piers at 6 ft. centers. Roof and walls insulated with 2 1/2 in. sprayed polyurethane foam. Input and output

air ducts constructed of lightweight timber frames with plywood lining and insulated with 3 in. weatherproofed sprayed polyurethane foam.

Includes one 40 H.P. centrifugal fan.

Heat Storage Unit Construction Components	MATERIAL (\$/sq.ft.)	LABOR (\$/sq.ft.)	TOTAL (\$/sq.ft.)
Excavation & Footings	0.10	0.03	0.12
Floor (and Plenum)	0.51	0.06	0.57
Walls & Roof	0.74	0.27	1.01
Insulation	0.45	0.15	0.60
I/O Ducts	0.14	0.11	0.25
Rock Material	0.47	0.06	0.53
Fans & Miscellaneous	0.34	0.05	0.39
	2.75	0.73	3.48
Contractor's Overhead & Profit (15%)			0.52
Cost per sq. ft. of Collector Area			\$ 4.00

Heat Storage Unit Alternative (a)

- b) Cylindrical 42 ft. diameter prefabricated BUTLER tank (galvanized steel) complete with conical roof. Perforated corrugated galvanized steel floor supported by 8 in. concrete blocks at 1 1/2 ft. centers on 6 in. concrete slab. Walls insulated

with 3 1/2 in. sprayed polyurethane foam separated from rock by 1/4 in. plywood lining. Roof insulated with 3 in. polyurethane boards attached to timber battens. Input and output air ducts constructed of galvanized steel around perimeters of tank and insulated with 8 lb./3 in. mineral wool complete with aluminum weatherproofing. Includes one 40 H.P. centrifugal fan.

Heat Storage Unit Construction Components	(\$/sq.ft.)	(\$/sq.ft.)	(\$/sq.ft.)
Excavation & Footings	0.09	0.05	0.14
Floor (and Plenum)	0.25	0.09	0.34
Walls & Roof	0.46	0.17	0.63
Insulation	0.30	0.25	0.55
I/O Ducts	0.14	0.08	0.22
Rock Material	0.47	0.06	0.53
Fans & Miscellaneous	0.34	0.05	0.39
	2.05	0.75	2.80
Contractor's Overhead & Profit (15%)			0.42
Cost per sq. ft. of Collector Area			\$ 3.22

Heat Storage Unit Alternative (b)

c). Rectangular, partially submerged building with 12 in. concrete block walls (reinforced) and three-ply composite roofing on prefabricated 3 in. TECTUM concrete panels supported by 14J3 steel joints; 10 in. concrete slab with wall beams to ground level, supporting timber joists with steel floor screen above. Walls insulated with 2 in. silicon sealed sprayed polyurethane foam. Input and output air ducts constructed of lightweight timber frames with plywood lining and insulated with 3 in. weatherproofed sprayed polyurethane foam. Includes one 40 H.P. centrifugal fan.

Heat Storage Unit Construction Components	MATERIAL (\$/sq.ft.)	LABOR (\$/sq.ft.)	TOTAL (\$/sq.ft.)
Excavation & Footings	0.41	0.12	0.53
Floor (and Plenum)	0.67	0.14	0.81
Walls & Roof	0.39	0.15	0.54
Insulation	0.33	0.20	0.53
I/O Ducts	0.14	0.11	0.25
Rock Material	0.47	0.06	0.53
Fans & Miscellaneous	0.34	0.05	0.39
	2.75	0.83	3.58
Contractor's Overhead & Profit (15%)			0.54
Cost per sq. ft. of Collector Area			\$ 4.11

Heat Storage Unit Alternative (c)

F. TOTAL CONSTRUCTION COST ESTIMATES:

Total construction cost estimates increased at an average rate of 8% from the first cost report of \$271,494 (August 31, 1976) to the lowest bid of \$420,000. Taking into account that the ASEID 1 program was first applied during the analysis stage of the project, before the commencement of any of the design tasks, a gradual escalation of successive construction cost estimates is to be expected. The ability of the ASEID 1 system to establish an accurate proportional cost for each major construction component was found to be of great value by the research development team. As shown in Figure VI-3 little variation in the proportional cost of the major construction components occurred after the fifth cost report. At that time, only approximately 50% of the design task had been completed, and yet the project management team was in a position to establish meaningful target costs for each major construction component to serve as a basis for final design decisions.

Based on the latest construction estimate of \$363,600 for a solar system incorporating a solar collector array of approximately 20,000 sq. ft., the total system cost per sq. ft. of collector surface is calculated to be \$16.83, as stated in the following table. A contractor's overhead and profit allowance of 15% is included in the cost estimate of each individual major construction component.

Construction Component	Cost Estimate	% of Total Cost	\$/sq.ft. of Collector
Site Preparation	\$ 7,500	2.06	0.35
Solar Collectors	129,600	35.64	6.00
Collector Supports & Installation	38,000	10.45	1.76
Storage System	33,800	9.30	1.56
Modification of Facility	31,325	8.62	1.45
Ducting	78,175	21.50	3.62
Fans & Dampers	22,000	6.05	1.02
Electrical	12,000	3.30	0.56
Controls	11,200	3.08	0.52
Totals:	\$ 363,600	100%	\$16.83

G. CONCLUSION:

The computer based construction cost estimating system (ASEID 1) described in this report, is certainly not recommended as a substitute for sound planning and design practices in large scale proof-of-concept research and development projects. However, it has been shown that when such a cost estimating system is implemented in combination with normal research procedures, and a computer with on-line facilities is available, the direct benefits derived greatly exceed the cost of the resources required for the development and operation of the system. Firstly, the development of the cost estimating system forces the research

and development team to identify the main components and sub-components of the construction project during the early stages of the investigation. Secondly, it establishes scheduling priorities on the basis of proportional cost. Thirdly, it generally assists the analysis and design teams in exploring the cost effectiveness of the design alternatives, in identifying potential construction problems, and in providing progressively more and more accurate construction cost estimates.

In addition to these direct benefits, there appear to be some significant indirect advantages. Apart from the confidence which is instilled in the research and development team by the existence of a continuously updated cost information data base, individual researchers tend to develop a higher level of cost consciousness which better prepares them for the technical decisions which have to be made during the design phase of the project.

THIRD PART

PHASE 1: Construction Documentation

VII. CONSTRUCTION DESIGN

The construction design proceeded from initial layouts and system cost trade-offs to detailed design. The working drawings are included at the end of this section and a detailed specification is available separately. Using these contract documents, firm construction bids were obtained from the construction industry in the Fresno area. Due to the unfamiliar character of the system, most of the construction industry was extremely conservative in their proposals. The resultant high cost of construction caused us (i.e. TRW) to re-evaluate the high cost elements and obtain firm bids on earlier configurations discarded in the trade-off studies. The final configuration is shown in Figure VII-1. Description of the elements of the design follow.

1. DESIGN PROCESS: The first step in the process of construction design consisted of producing layouts of the Lamanuzzi and Pantaleo site as it exists today. These drawings included details of the interfaces with the tunnel dehydrator, locations of produce truck tracks, building structures, concrete pavements and the elevations of the proposed field to be used for the solar system installation.

Layouts of various configurations of the following assemblies

-200-

were developed:

- Collectors and collector supports
- Storage building
- Underground vs. elevated ducts
- Fans and damper system configuration

These layouts were used in trade-off analyses to develop the lowest cost configuration. Cost considerations were not limited to their influence on the individual component but to their effect on the total system. In order to establish the cost of various configurations, discussions were held with potential materials suppliers, component manufacturers and construction contractors. Data developed were then fed into the cost, economic and system performance models for evaluation. System and component design criteria were then established so that detailed design could proceed.

2. SITE CONSIDERATIONS: The site is a relatively level plowed field draining to the southeast, located in an agricultural belt 10 miles northwest of the center of Fresno, California. The field is used for drainage of the quantities of water used in the production of L&P's food products. This drainage capability had to be maintained. The placement of the concrete block rock storage building in the midst of this field required the regrading of the site in both directions away from the storage building.

The final configuration shown in Figure VII-1 has placed the storage bin on the existing concrete pad, thus eliminating the need for regrading the site which will be used only for supporting the collectors and ducting.

3. COLLECTOR DESIGN: Numerous revisions to the collector configuration were examined in order to develop the lowest cost design, since collector cost was estimated to be 40% of the system cost. The baseline collector built and tested by the University and TRW during the NSF contract phase of this program was used for comparison.

Configurations consisting of inflatable plastic flow passages, corrugated aluminum absorbers, and all steel flat plate designs were examined. Performance evaluated during the trade-off studies included:

- Pressure drops
- Thermal efficiency of collection
- Fan power requirements
- Cost of construction
- Ease of installation
- Structural integrity
- Life

The 70-ft. long section of the selected design was constructed

and tested as described elsewhere in this report to substantiate the criteria and performance requirements. Actual construction of the collector verified concepts of the production of the unit. Thermal performance was as predicted and pressure drops were below those anticipated. Evaluation of the firm construction bids obtained have identified several small changes in the design as shown in Drawing 03. These changes will have no influence on the performance of the collectors. These changes include:

- Change specification of redwood material to one available at lower cost and equivalent strength
- Revise support installation to allow greater tolerance of support post placement

The final system performance analysis indicated that the optimum number of square feet of collector is achieved with 30 rows of collectors rather than the 32 rows shown on the working drawings. This small change will be incorporated into the drawings prior to construction.

The configuration selected for collector design consists of units 4 ft. wide by 180 ft. long manufactured in 30 ft. long segments and connected together in the field. The field connection provides flexible airtight joints which could accommodate any thermal stresses experienced in operation. Actually though, measurements made during thermal test indicate that this motion

is minimum. The collector contains two air passages 3 1/2 in. high by 21 3/4 in. wide under a thin steel, flat black absorber plate. The glazing is a single sheet of polyester reinforced with glass fibers manufactured under the trade name of Filon. This material maintains over 90% of its transmissibility at least for 20 years. The sides and bottom of the air passages are insulated with polyurethane foam and fiberglass. The collectors are supported at their 36° angle (latitude) off the ground on 6 in. x 6 in. posts imbedded in the ground. This support is designed to withstand wind and seismic loads. Both ends of each 180 ft. long collector are connected to the sheet metal manifold ducting. Configuration of these connections have taken into account the need to provide uniform flow among all of the collectors.

4. THERMAL STORAGE: Thermal storage is provided by a quantity of rock through which the heated air is passed in one direction to load the energy in the rock and, when the energy is extracted, the direction of air passage is reversed. A natural inclination would be to have the air pass through the rock horizontally. Past experience with horizontal flow has shown that this is unsatisfactory. Natural convection causes stratification of the heat with the top considerably warmer than the bottom. During extraction, too much of the energy is lost due to this stratified flow. For this reason, the storage system is designed with vertical flow.

The height of the rock storage is dictated by the pressure drop through the rock. Initial analysis defined the quantity of rock required for the selected percent solar as 25,000 cu. ft. Pressure drops defined the height of this quantity of rock as 12 ft. with a practical building width of 24 ft. The resulting storage facility is shown in the working drawings. Structural requirements required a practically solid concrete structure with considerable steel reinforcement.

The final system analysis showed that the storage capacity should be set at 12,500 cu. ft. This reduced quantity permitted using our first concepts for the storage building, the round silo-type of steel grain storage bin. The smaller size of this structure permitted placing it closer to the drying tunnel with the following advantages to the system design as shown in Figure VII-1.

- Storage placed on existing asphalt concrete pavement
- Collector fields can be placed closer together
- All fans and dampers are closer to storage unit
- Hundreds of feet of electrical power and control wiring eliminated
- Three ducts replaced with one duct
- Two F-2 fans replaced by one
- Reduced length of underground ducts
- Large reduction in storage building cost
- Need to regrade field under collectors eliminated

Figure VII-2
Rock Storage Bin

The storage bin selected is a 42 ft. diameter corrugated steel structure as seen in Figure VII-2.

Depth of the rock for the required quantity is 9 ft. The rock is supported by a perforated steel floor located 2 ft. above the concrete slab. The floor's perforations constitute 14% of the floor area. The drawings following the picture of the storage bin show details of construction of the foundation, the floor support beams and the floor itself. Openings in the side wall of the building provide the air flow from a plenum chamber attached to Fan F-2 for flow through the floor and rock. The building's height is 11 ft. to the eaves and this is the height of the rock. The area under the conical roof forms the plenum chamber at the top of the rock. Additional plenums are attached to the roof structure to facilitate attachment of the ducting.

The interior of the corrugated steel structure and the floor of the concrete foundation are sprayed with 2 in. of polyurethane foam for thermal insulation.

PLAN VIEW

SECTION "B - B"

FLAT SLAB FOUNDATION

DETAIL "E"

NOTES

1. FOUNDATION SITE SHOULD BE ON UNDISTURBED SOIL OR WELL COMPACTED BACKFILL. SITE SHOULD BE WELL DRAINED AND FREE FROM STANDING SURFACE WATER.
2. SOIL UNDER AND AROUND RING FOOTING SHOULD BE UNIFORM AND A TYPE OTHER THAN A VERY FINE SAND OR SILT. IF DETERMINATION OF SOIL TYPE IS A QUESTION, CONSULT A LOCAL ENGINEERING REPRESENTATIVE.
3. FOUNDATIONS ARE BASED ON UNIFORM SOIL BEARING CAPACITY OF 3700 LBS. PER SQ. FT. DESIGN IS BASED ON ULTIMATE COMPRESSIVE STRENGTH OF CONCRETE AT 28 DAYS OF 2500 LBS. PER SQ. INCH. SEE SLUMP NOTES.
4. (*) RE-ROD END LAP NOT INCLUDED. ADD 15" FOR EACH END LAP. FOR ESTIMATING, ADD 1/2 LB. PER 3/8" DIA. ROD END LAP, 1 LB. PER 1/2" DIA ROD END LAP.

CONCRETE FOUNDATION
'42' x 15' THRU 42' x 22'

BUTLER

**BUTLER MANUFACTURING
COMPANY**

GENERAL OFFICES - KANSAS CITY, MISSOURI 64108

DRAWN BY	CHECKED BY	SCALE	PART GROUP IN
SAL	GEM	None	207
DATE	SUPERSEDED		
1-20-75		D	435822

D-433126

TEL.
ROKEN LINES INDICATE
DOES OF SHEET
APPED UNDER
OP SHEET

FLANGE OF SUMP LAPS
BETWEEN COVER SHEETS
AND BEAM

SUMP ATTACHMENT DETAIL

SHEET LAYOUT

SHEET WITH LONGER OVERHANG FROM A BEAM GOES ON TOP
OF SHEET WITH SHORTER OVERLAP FROM A BEAM

SHEET LAP DETAIL

SE. F. DRILL SCREW
LOCATE THESE SCREWS
CENTERED BETWEEN THE
SUPPORT ANGLE BOLTS
IN THE TANK WALL. IN
LINE WITH THE NEAREST
HILL OF CORRUGATION

SUPPORT ANGLE INSTALLATION

HEX HEAD BOLT
5/16" DIA. X 3/4" WITH
WASHER ATTACHED

1/2" Approx.
SUPPORT
ANGLE

SCREWS ARE LOCATED
IN VALLEYS OF
CORRUGATIONS

SCREW INSTALLATION DETAIL

COVER SHEET ASSEMBLY INSTRUCTIONS 48" DIA. CUSTOM FLOOR			
CUSTOM 11		BUTLER MANUFACTURING COMPANY	
GENERAL OFFICE - ALABAMA CITY, OKLAHOMA 56		DATE	
BY	WLB	FOR	122
7/27/72			D 433523

5. DUCTING, FANS, DAMPERS AND CONTROLS: Construction of the manifold and main ducts are designed as standard sheet metal round ducts. Versions investigated before a selection was made include:

- Round paper tubes used for concrete column forms
- Rectangular sheet metal
- Inflatable plastic
- Rigid plastic

Here, again, the trade-offs included consideration of pressure drop, ease of connection to collectors, structural support requirements, interfaces with fans and dampers and cost. Ducts selected are round sheet metal with diameters ranging from 2 ft. to 6 ft.

Underground ducts consist of two concentric steel tubes separated by polyurethane foam insulation. This configuration was selected in order to ensure that the duct and insulation could withstand any loads due to heavy traffic on the surface above the buried ducts. The outer tube also provides protection for the thermal insulation from penetration by rocks and deterioration from underground water.

Thermal losses are minimized by the application of polyurethane foam sprayed all over the exterior of all ducts, manifolds, fans and plenum chambers. The exterior of all foam is protected with

a coating which prevents damage from moisture and ultraviolet attack. This type of thermal protection has been in use for many years on the wine storage tanks in California.

The fans and dampers are described in detail in the attached construction specification. The Ventilation section of this specification has not been revised to reflect the changes to the configuration resulting from the change to the storage building discussed earlier. Of major importance, fan F-2 will be a single unit of 60,000 cfm capacity driven by an 80 HP electric motor. Damper D-2 will correspondingly change to a single damper with dimensions to match F-2.

6. SPECIFICATION FOR CONSTRUCTION: The following construction specifications supplement the working drawings in defining the materials and methods of construction of the solar system. They also define the quality of workmanship required. These specifications were originally written to conform with the working drawings as presented here. However, in an effort to revise them to reflect the change in design discussed earlier, those sections dealing with construction tasks which have been deleted have been removed from the document. Additional revisions are required to make the specification reflect the current configuration. These include items such as the details of the new fan, F-2; damper, D-2, and the fact that dampers D-1 and D-4 have changed location.

A section must be added to define the storage bin and its plenum chambers. These items will be accomplished within the first month of the construction contract when the working drawings are revised to make the same revisions.

ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
CALIFORNIA POLYTECHNIC STATE UNIVERSITY - SAN LUIS OBISPO

SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT

DRAWING NUMBER	TITLE
00	TITLE SHEET, PERSPECTIVE, GENERAL CONSTRUCTION, NOTES, LOCATION MAP.
1	PLOT PLAN, LEGAL DESCRIPTION, SITE PLAN, GRADING, COLLECTOR SUPPORT DETAIL UNDERGROUND DUCT BENCHING SECTION.
2	CIVIL, PROFILE, DUCT SUPPORTS
3	SOLAR COLLECTOR DETAILS
4	HEAT STORAGE PLAN, FOUNDATION ELEVATIONS AND DETAILS
5	HEAT STORAGE ROOF PLAN SECTION AND DETAILS
6	HEAT STORAGE MISCELLANEOUS DETAILS
7	DUCT PLAN, ELEVATIONS AND MISCELLANEOUS DETAILS
8	DUCT SUPPORT LAYOUT
9	DRYING TUNNEL MODIFICATION, HEAT WHEEL
10	POWER PLAN, SINGLE LINE DIAGRAM AND DETAILS
11	CONTROL WIRING PLAN AND SCHEMATIC DIAGRAMS
12	CONTROL INTERCONNECTION DIAGRAM AND TERMINAL CABINET DETAIL

- GENERAL CONSTRUCTION NOTES
- MISCELLANEOUS
1. ALL MATERIALS AND WORKMANSHIP SHALL CONFORM TO THE REQUIREMENTS OF BOTH LOCAL AND UNIFORM BUILDING CODE (U.B.C.) STANDARDS.
 2. CONTRACTOR SHALL CHECK AND VERIFY ALL DIMENSIONS AND CONDITIONS ON THE JOB SITE AND REPORT ANY ERRORS, OMISSIONS, OR POSSIBLE DISCREPANCIES TO THE ENGINEER PRIOR TO PROCEEDING WITH THE WORK.
 3. CONTRACTOR TO OBTAIN CLEARANCE FROM TRW BEFORE PERMITTING THE REMOVAL OF SCRAP OR SAVABLE MATERIAL FROM THE JOB SITE. TRW SHALL DETERMINE THE ITEMS FOR RETENTION.
 4. THE CONTRACTOR SHALL PROVIDE SAFE AND ADEQUATE BRACES AND CONNECTIONS TO SUPPORT THE COMPONENT PARTS OF THE STRUCTURE UNTIL THE STRUCTURE ITSELF (INCLUDING THE FLOOR AND ROOF DIAPHRAGMS) IS COMPLETE ENOUGH TO ADEQUATELY SUPPORT ITSELF.
 5. ALL ELECTRICAL & UTILITIES ARE TO BE IN OPERATION DURING CONSTRUCTION PERIOD. THE CONTRACTOR SHALL OBTAIN PERMISSION FROM TRW PRIOR TO TURNING OFF ANY UTILITIES, AND SHALL MAKE CONNECTIONS TO MAIN SERVICES AT TIMES AND PLACES DESIGNATED.
- CONCRETE AND REINFORCING STEEL:
1. ALL CONCRETE AND REINFORCING STEEL SHALL BE MIXED, FORMED AND PLACED ACCORDING TO A.C.I. CODE, LATEST EDITION.
 2. CONCRETE SHALL HAVE A MINIMUM ULTIMATE COMPRESSIVE STRENGTH OF 2000 LBS. PER SQ. IN. AT THE END OF 28 DAYS, UNLESS NOTED ON PLANS.
 3. CONCRETE SHALL BE MACHINE-MIXED USING A MAXIMUM OF 7-1/2 GALLONS OF WATER PER SACK OF CEMENT. READY-MIX CONCRETE SHALL BE MIXED AND DELIVERED IN ACCORDANCE WITH ASTM C-94.
 4. REINFORCING STEEL SHALL BE INTERMEDIATE GRADE DEFORMED BARS CONFORMING TO ASTM A-615 GRADE 40. SPLICE BARS A MINIMUM OF 40 DIAMETERS OR 2'-0" MINIMUM.
- REINFORCING STEEL SHALL HAVE THE FOLLOWING MINIMUM CONCRETE COVER:
3"	... FOOTING CONCRETE POURED AGAINST EARTH.
2"	... CONCRETE POURED IN FORMS BUT EXPOSED FACE IN CONTACT WITH EARTH.
2"	... WALLS BELOW GRADE OR EXPOSED FACE IN CONTACT WITH EARTH.
1"	... SLABS AND INSIDE FACES OF WALLS.
1-1/2"	... BEAMS AND COLUMNS.
6. WIRE MESH SHALL CONFORM TO ASTM A-185 AND SHALL BE PLACED 2" FROM TOP OF SLAB, WITH A MAXIMUM TOLERANCE OF 3/4".
 7. AT ALL OPENINGS IN CONCRETE, CONCRETE BLOCK AND BRICK MASONRY PROVIDE TWO #5 BARS AT JAMBS, HEAD AND SILL, EXTENDING 2'-0" BEYOND EDGES OF OPENING.
 8. ALL WELDING TO REINFORCING STEEL SHALL BE DONE WITH LOW HYDROGEN ELECTRODES.
 9. ALL REINFORCING STEEL, ANCHOR BOLTS, DOWELS AND OTHER INSERTS SHALL BE SECURELY FASTENED IN THE FORMS BEFORE CONCRETE IS POURED. ADEQUATE CLEANOUTS SHALL BE PROVIDED IN THE BOTTOM OF CONCRETE FORMS FOR PROPER CLEANING AND INSPECTION.
 10. CEMENT FOR CONCRETE OR MASONRY MORTAR SHALL BE A STANDARD BRAND "PORTLAND CEMENT", MEETING THE REQUIREMENTS OF ASTM C-150.
 11. AGGREGATES FOR CONCRETE SHALL MEET THE REQUIREMENTS OF ASTM C-33.
 12. SLABS POURED ON GRADE SHALL BE LEVEL TO WITHIN 1/8" IN 8'-0" IN ANY DIRECTION EXCEPT AS NOTED OTHERWISE ON PLANS.
- CONCRETE BLOCK:
1. ALL CONCRETE BLOCK UNITS SHALL BE LIGHTWEIGHT GRADE "A" CONFORMING TO ASTM C-90.
 2. ALL CELLS CONTAINING HORIZONTAL OR VERTICAL REINFORCING STEEL, BOLTS, OR INSERTS SHALL BE GROUTED. REINFORCING STEEL, BOLTS, INSERTS, ETC., OCCURRING IN MORTAR JOINTS (OR ADJACENT THERETO) SHALL HAVE THE CELLS ON EACH SIDE OF SAID REINFORCING STEEL, ETC., GROUTED SOLID. USE WIRE MESH ONLY TO FORM HORIZONTAL CUTOFF POINT. (THE USE OF PAPER, CARDBOARD, ETC., NOT ACCEPTABLE.) WALLS SHALL NOT BE GROUTED SOLID UNLESS SPECIFIED ON THE PLANS.
 3. CLEANOUT OPENINGS SHALL BE PROVIDED AT BOTTOM OF ALL VERTICALLY GROUTED CELLS IF GROUT LIFT EXCEEDS 4'-0". MAXIMUM LIFT: 8'-0".
 4. CEMENT MORTAR: ONE CEMENT TO 3-1/2 SAND TO 1/4 LIME PUTTY.
 5. GROUT: ONE CEMENT TO 3 SAND TO 1/10 LIME PUTTY.
 6. IN GROUT SPACED 4" OR MORE IN BOTH HORIZONTAL DIMENSIONS, GROUT MAY CONTAIN AN ADDITION OF 3/8" PEA GRAVEL EQUAL TO NOT MORE THAN TWO PARTS BY VOLUME OF CEMENT USED.
 7. MORTAR AND GROUT SHALL HAVE A MINIMUM ULTIMATE COMPRESSIVE STRENGTH OF 2000 PSI AT 28 DAYS.
 8. ALL MASONRY WALLS IN EXCESS OF 10'-0" IN HEIGHT MUST BE ADEQUATELY BRACED TO RESIST A WIND LOAD OF 10 LBS. PER SQ. FT. APPLIED PERPENDICULAR TO WALL. BRACING TO REMAIN IN PLACE UNTIL ROOF DIAPHRAGM IS COMPLETE ENOUGH TO ADEQUATELY SUPPORT THE WALL.
- GROUT AND DRYPACK:
1. GROUT SHALL CONSIST OF 1 PART CEMENT, 4 NOT MORE THAN 3 PARTS SAND AND NOT LESS THAN 1 PART NOR MORE THAN 2 PARTS PEA GRAVEL BASED ON DRY LOOSE VOLUMES. GROUT TO BE OF FLUID CONSISTENCY. APPROVED ADMIXTURES MAY BE ADDED TO THE GROUT MIX. GROUT SHALL ATTAIN A MINIMUM ULTIMATE COMPRESSIVE STRENGTH OF 2000 PSI AT 28 DAYS.
 2. DRYPACK SHALL CONSIST OF 1 PART CEMENT, 4 PARTS SAND BASED ON DRY LOOSE VOLUMES AND NOT LESS THAN 1/4 PART NOR MORE THAN 1/2 PART LIME PUTTY OR DRY HYDRATED LIME. DRYPACK SHALL OBTAIN A MINIMUM ULTIMATE COMPRESSIVE STRENGTH OF 2000 PSI AT 28 DAYS.
- STRUCTURAL STEEL AND WELDING:
1. STRUCTURAL STEEL SHALL CONFORM TO ASTM A-36.
 2. PIPE COLUMNS SHALL CONFORM TO ASTM DESIGNATION A-53, GRADE "B", WITH SIX PHIR CONTENT NOT TO EXCEED 0.05%.
 3. STRUCTURAL STEEL SHALL BE FABRICATED AND ERECTED IN FULL COMPLIANCE WITH THE "AISC SPECIFICATION FOR THE DESIGN, FABRICATION AND ERECTION OF STRUCTURAL STEEL FOR BUILDINGS," LATEST EDITION.
 4. ALL WELDING TO BE PERFORMED BY CERTIFIED WELDERS AND SHALL BE PERFORMED AS REQUIRED BY THE A.W.S. SPECIFICATIONS FOR WELDING AND FABRICATION. REINFORCING STEEL SHALL BE WELDED USING LOW HYDROGEN ELECTRODES. WELDING SHALL BE PERFORMED BY CERTIFIED OPERATORS IN A LICENSED FABRICATING SHOP.

J. C. Chene Proj. Eng.

ERDA/CALPOLY SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT		TRW SYSTEMS GROUP	
TITLE SHEET			
PERSPECTIVE GENERAL CONSTRUCTION NOTES LOCATION MAP			
DATE	10 DEC 1976	DWG. NO.	6821.24-00
DRAWN	LS		
CHKD.	ES		
APPROVED	J. C. Chene	APPROV. - PROGRAM OFFICE

A
SECTION
SCALE: 1/4" = 1'-0"

B
WEST ELEVATION
SCALE: 1/4" = 1'-0"

FLOOR PLAN
SCALE: 1/4" = 1'-0"

1
WALL FOOTING DETAIL
SCALE: 3/4" = 1'-0"

C
SECTION
SCALE: 1/4" = 1'-0"

2
INTERIOR BEARING FOUNDATION
SCALE: 3/4" = 1'-0"

4
TYPICAL SIDE WALL FOOTING
SCALE: 3/4" = 1'-0"

FOUNDATION PLAN
SCALE: 1/4" = 1'-0"

3
PLENUM FOOTING DETAIL
SCALE: 1/2" = 1'-0"

5
DETAIL
SCALE: 1" = 1'-0"

D
TYPICAL SIDE ELEVATION
SCALE: 1/4" = 1'-0"

ERDA/CALPOLY SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT		TRW SYSTEMS GROUP	
HEAT STORAGE PLAN, FOUNDATION, ELEVATIONS, SECTIONS, AND DETAILS		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
DATE 10 DEC. 1976	DRAWN D. REWAY	CHKD. DS	APPRD. - PROGRAM OFFICE
PROJECT CAPTAIN		6821.24-04

ERDA/CALPOLY SOLAR ENERGY
FOR INDUSTRIAL DEHYDRATION PROJECT

HEAT STORAGE ROOF PLAN,
SECTION, AND DETAILS

DATE 10 DEC. 1976
DRAWN D. REVAY
CHKD. D. G.
APPROV. J. H. HUNT
PROJECT MANAGER

TRW
SYSTEMS GROUP

ENVIRONMENTAL, POWER
AND FACILITIES DEPARTMENT
SYSTEM INTEGRATION LABORATORY

UNIV. NO.
6821.24-05

APPROV. - PROGRAM OFFICE

(A) SECTION - WEST PLENUM
SCALE: 1/2" = 1'-0"

(C) SECTION - EAST PLENUM
SCALE: 1/2" = 1'-0"

(6) HEAD DETAIL
NO. SCALE

(B) TEMPERATURE SENSOR AT DUCTWORK
NO. SCALE

(E) UNISTRUT BASE
SCALE: 3/4" = 1'-0"

(1) SECTION THRU HATCH COVER
SCALE: 1/2" = 1'-0"

(5) PRECAST CONCRETE 'SPASH-BLOCK'
SCALE: 1/2" = 1'-0"

(7) THERMOCOUPLE DETAIL @ ROOF
SCALE: 3/4" = 1'-0"

(D) DELTA P MOUNTING DETAIL
SCALE: 1/2" = 1'-0"

(2) GLUTTER DETAIL
SCALE: 3/4" = 1'-0"

(D) SQUARE DUCT FOOTING
SCALE: 1 1/2" = 1'-0"

(H) TYP DAMPER ACTUATOR
SCALE: 3/4" = 1'-0"

(G) ACTUATOR ELEVATION
SCALE: 3/4" = 1'-0"

(3) WALL/DUCT PENETRATION
SCALE: 1/2" = 1'-0"

(F) AIR MIXING PLENUM ELEVATION
SCALE: 3/4" = 1'-0"

MARK	OPENING SIZE	THK	MATERIAL	HEAD	JAMB	SILL	W/VE SET	REMARKS
(1)	24x24	-	METAL FOAM	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	# WHERE OCCURS
(2)	40x54	1/2"	PLYWOOD	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	
(3)	80x86	1/4"	METAL CLAD	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	
(4)	12x12	-	METAL FOAM	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)	(B) (C) (D) (E) (F) (G) (H) (I) (J) (K) (L) (M) (N) (O) (P) (Q) (R) (S) (T) (U) (V) (W) (X) (Y) (Z)

(8) DETAIL @ ACCESS DOOR
SCALE: 3/4" = 1'-0"

(9) DETAIL @ ACCESS DOOR
SCALE: 3/4" = 1'-0"

(B) ACTUATOR ELEVATION
SCALE: 3/4" = 1'-0"

ERDA/CALPOLY SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT		TRW SYSTEMS GROUP	
HEAT STORAGE MISCELLANEOUS DETAILS		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
DATE: 10 DEC 1976	DRWN: D. REAY	CHKD: D. REAY	APPRD: PROGRAM OFFICE
DWG. NO. 6821.24-06

NOTE:

1. INSULATION NOT SHOWN ON THIS DRAWING FOR CLARITY
2. REFER TO SPECIFICATIONS FOR SCOPE OF THAT WORK.

ERDA/CALPOLY SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT
DUCT PLAN, ELEVATIONS &		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT
MISCELLANEOUS DETAILS		SYSTEM INTEGRATION LABORATORY
DATE 10 DEC 1976		DWG. NO. 6821.24-07
DRAWN KJC		
CHKD. DG		
APPROVD <i>[Signature]</i>	APPRVD. - PROGRAM OFFICE

DUCTWORK SUPPORT PLAN
SCALE: 1/4" = 1'-0"

DOOR 5
REFER TO
DOOR SCHEDULE
SHEET NO 00

(F) DRYING TUNNEL DUCT SUPPORTS
SCALE: 3/4" = 1'-0"

(A) 08 SUPPORT ELEVATION
SCALE: 1/2" = 1'-0"

NOTE:
INSULATION NOT SHOWN
ON THIS SHEET FOR CLARITY
REFER TO SPECIFICATIONS FOR
SCOPE OF THIS WORK

(B) 08 BASE PLATE DETAIL
SCALE: 3/4" = 1'-0"

(C) 08 HEAT WHEEL DUCT SUPPORT
SCALE: 3/4" = 1'-0"

(B) 08 SUPPORT DETAIL
SCALE: 3/4" = 1'-0"

END/CALPOLY SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT		TRW SYSTEMS GROUP	
DUCT SUPPORT LAYOUT		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
DATE 10 DEC 1976	DWG. NO.	6821.24-08	
DRAWN RJC	APPROVED	APPROVED - PROGRAM OFFICE	
CHKD. RJC	APPROVED - PROJECT CAPTAIN

NOTE:
1. INSULATION NOT SHOWN FOR CLARITY ON THIS SHEET.
2. REFER TO SPECIFICATIONS FOR SCOPE OF THAT WORK.

ERDA/CALPOLY SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT		TRW SYSTEMS GROUP	
DRYING TUNNEL MODIFICATION, HEAT WHEEL		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
DATE: DEC 1976 DRAWN: KJC, DRG CHKD: DS APPROVD: [Signature]	DWG. NO. 6821.24-09	APPROVD: [Signature] PROGRAM OFFICE

SCHEMATIC DIAGRAMS

SYMBOLS

- WIRING IN CONDUIT RUN EXPOSED
- WIRING IN CONDUIT RUN UNDERGROUND
- T- THERMOCOUPLE LEADS IN CONDUIT
- T- (1) SET OF LEAD
- 3T- (3) SETS OF LEADS
- Ⓢ2A DAMPER MOTOR, SUBSCRIPT INDICATES DAMPER DESIGNATION
- Ⓢ7 THERMOCOUPLE, NUMBER DENOTES SENSOR DESIGNATION
- Ⓢ HEAT EXCHANGER MOTOR
- ΔP PRESSURE DIFFERENTIAL SENSOR
- Ⓢ JUNCTION BOX
- N.I.C. NOT IN CONTRACT

NOTES

- EXCEPT NOTED OTHERWISE, CONTROL WIRING SHALL BE TYPE TW.
- EACH THERMOCOUPLE LEAD SHALL BE CONTINUOUS AND WITHOUT SPLICE FROM THERMOCOUPLE TO LOGIC CONTROLLER AT CONTROL CLOSET. PROVIDE 8 FEET ALLOWANCE IN WIREWAY FOR CONNECTION TO LOGIC CONTROLLER. TAPE ENDS OF LEADS TO PREVENT ENTRY OF MOISTURE.
- COIL 3 FEET LENGTH OF THERMOCOUPLE LEAD INSIDE J-BOX FOR CONNECTION TO FUTURE T10. TAPE END OF LEAD.
- TAPE ENDS OF CONDUCTORS AND COIL INSIDE J-BOX FOR CONNECTION TO T10 POWER SUPPLY.

ERDA/CALPOLY SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT		TRW SYSTEMS GROUP	
CONTROL WIRING PLAN AND SCHEMATIC DIAGRAMS		ENVIRONMENTAL POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
DATE: 10 DEC 1976	DRAWN: E.C.	DWG. NO.:	682-24-11
CHKD: E.C.	APPROV: [Signature]	PROGRAM OFFICE

FRONT VIEW (INSIDE)

SIDE VIEW

CONTROL TERMINAL CABINET
NO SCALE

LOGIC CONTROLLER
(N.I.C.)

LIST OF MATERIAL	
ITEM	DESCRIPTION
1	CABINET, 24" X 16" X 6", NEMA 4 ENCLOSURE, 14 GAUGE STEEL, CONTINUOUSLY WELDED, WITHOUT KNOCKOUTS, WITH 12 GAUGE NEOPRENE GASKETED HINGED DOOR COMPLETE WITH EASY TO OPERATE DOOR CLAMPS. HOFFMAN CAT. NO. A-24HIGALP OR APPROVED EQUAL
2	BACK PANEL, 12 GAUGE STEEL, 21" X 13" HOFFMAN CAT. NO. A-24PIG
3	TERMINAL BLOCK, TANG TERMINALS, 600 VOLT SPACING, #22 TO #8 WIRE RANGE, CONNECTRON CAT. NO. N3. (3 TERMINALS PER 1/4" SECTION)
4	MOUNTING RAIL, STAINLESS STEEL, 1/2" LONG, CONNECTRON CAT. NO. NR 12 1/2
5	MARKING STRIP, CONNECTRON CAT. NO. MT 12 1/2
6	JUMPER, 2-CIRCUIT, COPPER, CONNECTRON CAT. NO. JN3
7	PLASTIC WIRING DUCT, WITH REMOVABLE NON-SLIP COVER, RANDUIT 1" X 1"

NOTES

- DAMPER 4B IS MECHANICALLY LINKED TO DAMPER 4A. WHEN 4A IS OPEN, 4B SHOULD BE CLOSED AND VISE-VERSA.
- PROVIDE EACH END OF ALL CONTROL WIRES WITH WIRE MARKER DESIGNATED AS SHOWN IN THE INTERCONNECTION DIAGRAM. WIRE NUMBERS IDENTICAL TO TERMINAL BLOCK NUMBERS.

CONTROL INTERCONNECTION DIAGRAM

ERDA/CALPOLY SOLAR ENERGY FOR INDUSTRIAL DEHYDRATION PROJECT		TRW SYSTEMS GROUP	
CONTROL INTERCONNECTION DIAGRAM AND TERMINAL CABINET DETAIL		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
DATE 10 DEC 1976	DRAWN E.D.	CHKD. E.D.	APPRD. - PROGRAM OFFICE
		DWG. NO. 6821-24-12	

FOURTH PART

PHASE II: VII. Phase II Estimated Installed Solar System Cost

VII. PHASE II ESTIMATED INSTALLED SOLAR SYSTEM COST

For Phase II of the Application of Solar Energy to Industrial Drying or Dehydration Processes, the California Polytechnic State University Foundation will be the general contractor for the construction of the solar system at the Lamanuzzi and Pantaleo Dehydration Facility in Fresno, California. The estimated construction costs of the proposed installed solar system are noted below.

		<u>\$/sq.ft.</u>
1. Site Preparation	\$ 7,500	0.35
2. Collectors	\$129,600	6.00
3. Collector Supports & Installation	\$ 38,000	1.75
4. Storage System	\$ 33,800	1.56
5. Modification of Facilities	\$ 31,325	1.45
6. Ducting	\$ 78,175	3.62
7. Fans and Dampers	\$ 22,000	1.02
8. Electrical	\$ 12,000	0.56
9. Controls	<u>\$ 11,200</u>	<u>0.52</u>
TOTAL:	\$363,600	\$ 16.83