MASTER

MEDIUM-TEMPERATURE AIR-HEATER DEVELOPMENT PROGRAM

FINAL REPORT
FOR PERIOD OCTOBER 1, 1977 - DECEMBER 31, 1978

April 15, 1979

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PREPARED FOR THE
U.S. DEPARTMENT OF ENERGY
SOLAR ENERGY
UNDER CONTRACT EG-77-C-04-4126

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FOREWARD

This research and development program was conducted by the General Electric Company at Valley Forge, Pennsylvania. The technical monitor for the initial portion of the program was Walter Adams, Sandia Laboratories. In the fall of 1978, technical monitor responsibilities were assigned to Lyle Groome of the Solar Energy Research Institute. Their assistance and guidance is gratefully acknowledged.

This program was a continuation of the effort to develop commercial solar collectors using air as the working fluid and an evacuated tube absorber. It utilized the results of a previous program, SOLAR HEATER PROGRAM or SOLAIR, carried out on Contract EY-76-C-02-2705. The initial design concept, designated SOLAIR IA, originated in the original program and was evaluated on this effort. At the end of the performance period activities were underway to further investigate the commercial viability of the product concept with a test market program.

ACKNOWLEDGEMENT

"This work has been supported by the Solar Heating and Cooling Research and Development Branch, Office of Conservation and Solar Applications, U.S. Department of Energy."

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SECTION 1

INTRODUCTION

This is the final report for the Medium Temperature Air Heater Development Program, Contract No. EG-77-C-04-4126, conducted for the U.S. Department of Energy over the period from October 1, 1977 to December 31, 1979.

1.1 PROGRAM OBJECTIVES

The overall objective of this program was to develop a superior delivered-energy-per-unit-cost solar heater for domestic space heating and hot water applications. The end product was to be a non-concentrating medium temperature air heater that could be adopted to both applications. The intent was to design the system to meet requirements identified in a market analysis for modest residential heating demands. Also, the intent was to utilize the vacuum tube solar absorber previously developed by the General Electric Company for solar collectors. Cost effectiveness and producibility for volume were key considerations.

1.2 BACKGROUND

This product was an evolution from previous activities that utilized the General Electric vacuum tube solar absorber with air as the heat transfer medium. The absorber was originally developed for use in a liquid collector product (the TC-100 collector) and was in production at a pilot collector manufacturing plant at the Valley Forge, Pennsylvania facility. The absorber characteristics (low thermal losses at high temperature) were recognized as the potential basis for an effective air collector. This was verified by performance estimates for an initial design concept and led to the SOLAIR Heater Project (Contract EY-76-C-02-2705). Also recognized was that the vacuum tube air collector could be a low cost item because the absorber production could be highly automated for large volume production, and the absorber was made up of low cost materials. The use of the device for both air and liquid collectors would provide a larger production base and thereby decrease the unit cost.

The SOLAIR system used a bank of air collector assemblies in conjunction with a blower unit and rock thermal energy storage to provide solar heat. The unit was a free standing module, and field tests were conducted in Pennsylvania, North Dakota, and Arizona. These three systems operated successfully, but marketing and product assessments showed a limited commercial potential for the design. One of its problems was that it contained too much material to allow costs to be

reduced to a marketable level. Design concepts to minimize the materials content were investigated, and one designated SOLAIR IA was reduced to hardware on the previous program.

Additional market studies to provide guidance to this project indicated that the vacuum tube absorber could be the basis of a marketable solar heating system and that domestic hot water systems offered the best opportunity for an early product. Consequently this program was directed at an effective medium temperature air heater with system activities focused on the domestic hot water application.

1.3 SUMMARY

One initial activity on this program was the selection of a collector configuration. The cost and performance of the SOLAIR IA concept, a minimum material content design, was compared with the cost and performance of configuration similar to the liquid cooled collectors using the tubular vacuum absorber. The liquid cooled collector configuration was selected and is designated SOLAIR II. A modular design for domestic hot water systems has been developed and has undergone test verification. The design consists of a two tank system that can be used for both new construction and retrofit applications; the system diagram is shown on Figure 1-1. System elements specific to the SOLAIR II are the TCA solar collectors, the energy transfer module (ETM), interconnecting ductwork, and mounting hardware. The hot water tanks and plumbing hardware are standard commercial products and would typically be supplied by the housing contractor or the home owner.

The medium temperature air heater design is based on light gage galvanized steel sheet with foam insulation and uses the same vacuum absorbers as the General Electric TC-100 liquid cooled solar collector and the same a cusp shaped reflector. Figure 1-2 is a photograph of the air heater module. Two to four of these modules are used in the SOLAIR II domestic hot water system depending on geographic location and on the percentage contribution from solar desired.

The energy transfer module is also constructed of light gage galvanized steel with both foam and fiberglass mat insulation being used. A single motor drives the fan and water circulation pump. A photograph of the ETM is shown in Figure 1-3.

The system performance in terms of the percent of the total load supplied by solar was evaluated for a typical application in two regions of the nation. The evaluation used the following data and conditions:

- o $2.76 \times 10^{-6} \text{ m}^3/\text{s}$ @ 330°K (63 gal/day @ 140°F) water draw
- o Draw schedule per ASHRAE-95 solar DHW rating method (In preparation)
- o NOAA weather tapes (hourly computations, design year)
- o Collectors oriented south, tilted at latitude angle.

Figure 1-1. System Diagram for Solar Domestic Hot Water System

h-1

Figure 1-2. TCA solar Collector

Figure 1-3 Energy Transfer Module

Based on these conditions, the performance predicted for a DHW system using the SOLAIR II is given in Table 1-1.

TABLE 1-1 SOLAIR II SYSTEM PERFORMANCE

NUMBER OF MODEL TCA COLLECTORS	PERCENT SOLAR CONTRIBUTED	
	Boston, MA	Ft. Worth, TX
2	35	50
3	50	71
4	65	90

SECTION 2
PROGRAM SCOPE

The principal objective of the program was to design, fabricate, and verify by test a low cost (in volume production), modularized, practical solar air heater adaptable for new or retrofit space heating and hot water applications. Toward that objective, the following tasks were included:

- a) Requirements - determine cost effective heat energy requirements to satisfy modest space heating and hot water demands in conjunction with a market analysis for potential applications; determine modular concepts for establishing market acceptance.
- b) Optics Optimization - optimize energy gathered per unit cost by selection of tube spacing, reflector material, and configuration designs.
- c) Heat Transfer Optimization - optimize collector efficiency with parasitics (electrical) consumption by minimizing tube pressure drop and the corresponding blower power requirement. Evaluate various configurations of collector flow passages and potential tubulator designs. Perform verification tests on an individual collector element, and formulate an analytical model for test data correlation.
- d) Industrial Design - investigate concepts for cost effective support structures.
- e) Detailed Design - perform manufacturing engineering leading towards a commercial product.
- f) Fabrication and Assembly - fabricate and assemble two design verification air collector modules.
- g) Design Verification Test - prepare plan and conduct design verification testing to compare performance with design optimization predictions; both direct space heating and domestic hot water configurations to be tested.
- h) Pre-production Prototype Design - incorporate design modifications indicated by results of verification testing; develop fabrication drawings.

SYSTEM DESCRIPTION

3.1 GENERAL

The SOLAIR II medium temperature air heater is a two tank domestic hot water augmented system that can be used for both new construction and retrofit applications. The hardware elements that are unique to this system are:

- a) TCA solar collectors - evacuated tube collectors using air as the heat transfer medium in modules of 8 collectors.
- b) Energy Transfer Module - contains the air to water heat exchanger, the motor driven air blower and water pump on a common shaft and all controls for the DHW system.
- c) Interconnecting Ductwork - connects the collector modules (2,3, or 4) with the energy transfer module.
- d) Mounting Hardware - mounts the collector modules to the air ductwork and to the roof.

The hot water tanks and other plumbing hardware are standard commercial products that would have to be supplied by either the housing builder or contractor or the home owner.

3.2 SPECIFICATIONS

Table 3-1 presents the specifications for the SOLAIR II collector modules. The environments in which the collectors must operate effectively over long periods are presented in Table 3-2. The energy transfer module is factory assembled and incorporates the components for the control and transfer of collected energy. Its specifications and operational environment requirements are given in Table 3-3.

3.3 COLLECTORS AND MOUNT

The SOLAIR II water heating system uses the Model TC-100 vacuum tube absorber and

a cusp shaped reflector as the Model TCA solar collector. This collector is designed to operate efficiently in direct and diffused sunlight regardless of ambient air temperature or wind conditions.

The absorber consists of one glass cylinder inside another. The space between the tubes is evacuated. The outer cylinder serves as the window; the inner cylinder is coated with an optically selective coating having a solar absorptivity greater than 0.85 and an infrared emissivity less than 0.04 at 373°K (100°C) and serves as the absorber. Thermal energy is absorbed by the inner glass cylinder and a concentric but smaller diameter metal sleeve and then returning down the center of the metal sleeve or tube to an outlet header.

Table 3-1
SOLAIR II Collector Module Specifications

I. Vacuum-Absorber	
Envelope	5.3 cm (2.08 in.) dia.x1.25m (44.1 in.) long
Absorber Area	4.45cm (1.75 in.) dia.x1.14m (45 in.) long
Solar Absorptivity	> 0.85
IR Emissivity	< 0.04 at 373°K (100°C)
Stagnation Temperature Capability	755°K (900°C)
II. Reflector	
Aperture Area	1.71 m ² (18.4 ft ²)
Shape	Cusp
III. Absorber/Reflector Combination	
Concentration Ratio	1.09
Normal Incidence Absorbed Energy Fractions	
Direct to Absorber	40%
Reflected to Absorber	60%
IV. Energy Transport	
Air Flow	1.42 to 1.89 dam/s (3 to 4 cfm)(per tube)
Annulus Feed	3.18 cm (1.25 in.) dia. inner tube (metal)
Pressure Drop	62 to 87 N/m ² (0.25 to 0.35 in. H ₂ O)
Temperature Rise	180°K (100°F) at 945 W/m ² (300 Btu/hr-ft ²)
Turbulent Flow Conditions	
V. Collector Module	
8 Absorbers/Reflectors in Frame	
Parallel Flow	
Two Headers - Feed and Return	
Weight	≤ 18.2 kg (40 lb.)
Active Absorber Area	1.39 m ² (15 ft ²)

Table 3-2
Collector Operational Environments

Ambient Temperature Range	224 to 316°K (-20 to 110°F)
Relative Humidity	To 100%
Ambient Pressure	-183 < P < 1830m (-600 < P < 6000 ft) altitude above sea level
Hail	1.3 cm (0.5 inch)
Snow Load	To 960N/m ² (20 lb/ft ²)
Ice Load	To 2.5 cm (1 in.) thickness
Rain	4.4 μm/s (15 in. in 24 hr.) 35 μm/s (5 in. in 1 hr.)
Wind	To 40 m/s (90 mph) any direction
Salt Air	15 yr. life with yearly maintenance
Air Pollution	No significant degradation in performance or life

Table 3-3

Energy Transfer Module Specifications/Environments

Dimensions	0.61m (24 in.) long x 0.43m (17 in.) wide x 0.43m (17 in.) high
Weight	<13.6 kg (30 lb.)
Heat Exchanger Type	Air to water (finned tube)
Motor	1.7 A (1/8 hp), 115v, 60 Hz, single phase AC, RPM 3150
Air Blower	Fafco, 0.075 m ³ /s (160 cfm) at ΔP drop of 174 N/m ² (0.7 in. H ₂ O)
Water Pump	March Co. Model #2, rated for 0.32 dm ³ /s (5 gpm) with 3.96m (13 feet) head
Controls	Operate when insolation > 35 BTU/hr-ft ²

ETM Module Operational Environments

Ambient Temperature Range	244 to 344°K (-20 to 160°F)
Relative Humidity	To 100%
Ambient Pressure	-183 < P < 1830M (-600 < P < 6000 ft) altitude above sea level

The Model TCA collector contains eight vacuum tube absorbers parallel to one another, each at the focal point of a highly reflective, specially shaped reflector. The reflector is an inverted dual tube with a cusp located directly below and pointed toward the centerline and extending over the length of the absorber. This reflector provides a nine percent concentration of the solar radiation intercepted by the absorber. Figure 3-1 is a sketch of the parts of the absorber tube for air flow partition. The assembled TCA collector is shown in Figure 3-2.

The roof mount provided for the TCA collector is shown on Figure 3-3. It consists of four sheet metal brackets which bolt to the roof structure. Standard roofing construction techniques are used to provide water tight penetrations.

3.4 ENERGY TRANSFER MODULE (ETM)

The energy transfer module is factory assembled and incorporates the components for the control and transfer of collected energy. The module has been engineered for simplified installation. It is lightweight for ease of handling and compact for applicability in most new or retrofit applications. The module interfaces with the collectors with specially provided ducts; standard piping connects the module to the preheat tank. Overall dimensions of the module are shown in Figure 3-4. An assembled unit is shown in Figure 3-5.

3.5 INTERCONNECTING DUCTWORK

Interconnecting ducts are provided to transition from a standard 15.3 cm (6 inch) diameter duct to the rectangular ducts in the collectors. The inlet duct feeds into a transition section to two rectangular ducts. The rectangular ducts pass through the roof and, after turning 90°, mate with the collector ducts. The outlet duct is handled similarly. The four ducts are shown in Figure 3-6 which shows the collector interface.

3.6 ANCILLARY EQUIPMENT

Additional equipment and materials that would be required to complete the installation and operation of the SOLAIR II domestic hot water system include an insulated tank with a 30-52 gallon water capacity to serve as the preheat tank and other standard piping, wiring, ducting, and insulation as needed. This

Figure 3-1. Evacuated Tube Air Collector Flow Diagram

Figure 3-2. Assembled TCA Collector

Figure 3-3. Typical Roof Mount For TCA Collector (UF-79-97A)

Figure 3-4. ETM Installation Data

3-12

Figure 3-5. Assembled Energy Transfer Module

3-13

Figure 3-6. Ductwork for the SOLAIR System

material/equipment would be provided by the housing contractor or builder in the case of new construction or by the home owner in the case of a retrofit application.

3.7 INSTALLATION

A typical system installation is shown in Figure 3-7. A sheet metal enclosure is provided for the roof penetration ducts. A completed installation is illustrated in Figure 3-8.

SURE
NOT
ANYTHING

3-16

Figure 3-8. Residnetial Installation

SECTION 4

MARKET ANALYSIS

4.1 MARKET FORECAST

The market forecast methodology is shown in Figure 4-1. The total market of interest is determined. This represents the total number of single family residential hot water applications in the U.S. This total is reduced by applications that are unsuitable for solarization. The remainder represents the potential solar market. A market penetration/solar economic correlation is established based upon available market data. Using the product price history as given by the price goals, a market penetration history is developed. This penetration into the potential solar market determines the sales projection.

4.1.1 TOTAL SERVED MARKET OF INTEREST

Figure 4-2 shows the total U.S. market of interest. The present inventory is comprised of about 46 million single family hot water applications. Of these, 55% is serviced by utility gas, 11% by fuel oil, 27% by electric, and 4% by other means.

There is a large annual replacement market. With tank service lifetimes of about ten years, ten percent of the inventory would be expected to be replaced every year. The industry shipment data confirm this expectation. Shipments to the replacement market run over four million units per year.

The new construction market projections are shown to 1983 (data supplied by General Electric - Air Conditioning Business Division). This market is in excess of 1.2 million installations/year. Of these, over half will be electric water heaters. This increase in the use of electric systems is due to the fact that no new gas customers are allowed in many new construction areas. Also,

FIGURE 4-1. SOLAR MARKET FORECAST METHODOLOGY

FIGURE 4-2. DOMESTIC HOT WATER (DMW) MARKET FOR SINGLE FAMILY RESIDENCES

homeowners who opt for the electric heat pump must use electric hot water to keep an all-electric rate. Table 4-1 shows the estimated hot water total market of interest by region and energy type for the existing inventory and new market segments.

4.1.2 POTENTIAL SOLAR MARKET

The existing inventory includes many residences that cannot be feasibly solarized. Major conditions precluding the effective use of solar energy are:

- a. unacceptable orientation and/or
- b. site constraints and/or
- c. architectural constraints

The optimum azimuthal orientation for solar systems is due south. Analyses indicate that the limit of economic feasibility with respect to orientation is about $+ 45^{\circ}$ from due south. Assuming a random distribution of azimuthal orientation angles, this implies that about 50% of the existing inventory could not be feasibly solarized using a roof-mounted system. This dilution includes many houses that may be suitable because they have adjoining roofs perpendicular to the main roof. Also, many of the homes included in this dilution are amenable to ground installation. It is felt that these homes are counterbalanced by the ones with proper roof orientation that must be eliminated because of site or architectural constraints. These constraints consist of shadowing by trees or structures, insufficient contiguous roof area, inadequate insulation, etc. Based upon the above considerations, the existing inventory has been uniformly diluted by 50%.

The new construction market will also be affected by orientation, site, and architectural considerations. The impact upon market dilution will be smaller relative to the existing housing stock. New houses are well insulated, have large roof expanse, and usually are built on clear sites. It is felt that by the early 1980's, homes will be oriented for compatibility with solar systems.

TABLE 4-1. ESTIMATE OF HOT WATER MARKET OF INTEREST BY REGION

REGION	INVENTORY X 10 ³	SERVED BY (X 10 ³)			NEW CONSTRUCT X 10 ³ (1980)	SERVED BY (X 10 ³)		
		ELEC	UTIL. GAS	OIL		ELEC	UTIL. GAS	OIL
EAST NORTH CENTRAL	4000	1080	2200	440	240	137	79	24
S & S ATLANTIC	4400	1180	2420	484	225	128	75	20
PACIFIC	4500	1215	2475	495	160	91	53	15
N & NE	11200	3240	6600	1320	60	34	20	6
MID-ATLANTIC	5800	1566	3190	638	165	94	55	16
SOUTHWEST	600	162	330	66	60	34	20	6
SOUTH TROPIC	1000	270	550	110	25	14	9	2
CENTRAL	2200	590	1210	242	70	40	23	7
W.N. CENTRAL	2200	590	1210	242	40	23	13	4
E.S. CENTRAL	5500	1485	3025	605	110	63	36	10
NW PACIFIC	1100	297	550	121	25	14	9	2
W.S. CENTRAL	2800	756	1540	308	180	100	62	18

Based upon these considerations, the new construction market has been uniformly diluted by 25%. Table 4-2 shows the potential solar hot water market of interest by region and energy type for the existing inventory and new market segments.

4.1.3 MARKET PENETRATION/SYSTEM PAYBACK CORRELATION

Today's solar hot water systems generally provide marginal paybacks.

The primary reasons for solar system purchase are not economic. Market studies have shown that consumers can be separated into five categories, with respect to new product adoption rate. These are:

1. innovators (3-5% of market)
2. early adopters (10-15% of market)
3. early majority (35% of market)
4. late majority (35% of market)
5. laggards (5-16% of market)

The innovators are the first to adopt new products followed by the early adopters and so forth.

For a volume market, penetration into the early majority market segment would be a reasonable approach. Limited market data indicate that this segment has interest, has capital, but has priorities on how money is spent. For solar to have high priority with this market segment, the system must be economically attractive. Survey data indicate that the system must payback in 5-7 years to be considered economically attractive by this segment:

Using the above data base, a correlation between market penetrations and payback time has been established. This is shown in Figure 4-3.

TABLE 4-2. ESTIMATE OF POTENTIAL SOLAR HOT WATER MARKET BY REGION

REGION	POTENTIAL SOLAR INVENTORY $\times 10^3$	SERVED BY ($\times 10^3$)			NEW CONSTRUCT $\times 10^3$ (1980)	SERVED BY ($\times 10^3$)		
		ELEC	UTIL. GAS	OIL		ELEC	UTIL. GAS	OIL
EAST N. CENTRAL	2000	540	1100	220	180	100	60	20
S & S ATLANTIC	2200	590	1210	240	170	95	55	20
PACIFIC	2250	610	1240	250	120	70	40	10
N & NE	6000	1600	3300	650	45	25	15	5
MID-ATLANTIC	2900	800	1600	320	125	70	40	15
SOUTHWEST	300	80	150	30	45	25	15	5
SOUTH TROPIC	500	130	275	50	19	11	7	1
CENTRAL	1100	300	600	120	50	30	15	5
W. N. CENTRAL	1100	300	600	120	30	17	10	3
E. S. CENTRAL	2750	750	1500	300	80	45	25	10
N. W. PACIFIC	5500	150	275	60	19	11	7	1
W. S. CENTRAL	1400	380	770	150	135	75	45	15

FIGURE 4-3. MARKET PENETRATION/SOLAR SYSTEM PAYBACK CORRELATION

The correlation has limited utility for payback times greater than 15 years. The consumer is purchasing solar for reasons other than attractive economics (prestige, independence, etc.). Economics will not be a strong sales initiator until paybacks are less than 10 years. From this point, solar acceptance increases dramatically as payback time decreases. It is anticipated that a five year payback will capture 25% of the solar feasible market.

4.1.4 SALES PROJECTIONS

Using the above data base, sales projections for solar hot water systems have been determined. Table 4-3 shows the forecast for 1979. It is assumed that a government rebate program is in effect. At this point in time, the market will be fragmented with respect to market share. There will be many competitors; few will have a national distribution network. Many garage outfits will cover limited market areas.

By 1983, the market will follow the more traditional pattern. There will be price-demand elasticity. Many small outfits will drop out of the market by attrition and acquisition. There will probably be no more than 7-10 national brands and several regional brands competing at this price level.

Table 4-4 shows the industry forecast for 1983. Sales in excess of 275,000 units are projected. Figure 4-4 shows the forecast on a yearly basis.

4.2 SOLAR ECONOMICS

Solar economics will be considered from a consumer viewpoint. The following assumptions have been used based upon limited market data.

TABLE 4-3. SHIPMENTS FORECAST FOR 1979*

Region	Retrofit			New Construction			Total Shipments
	# Solar Add-ons To			# Solar with Backup			
	Electric	Gas	Oil	Electric	Gas	Oil	
EE. N. Central	750	440	90	400	180	60	1920
SS & S Atlantic	4700	500	100	1000	165	60	6525
Pacific	6100	1000	250	5200	800	100	13450
N and NE	3200	2600	1300	100	75	25	7300
Mid-Atlantic	1600	800	500	280	120	5	3305
Southwest	1000	200	-	1000	120	-	2320
South Tropic	1300	500	-	500	200	30	2530
Central	590	-	100	200	50	15	955
W. N. Central	300	-	-	160	30	10	500
E. S. Central	-	-	-	100	75	30	205
N. W. Pacific	-	-	-	-	-	-	-
W. S. Central	750	300	-	1000	120	75	2245
*assumes incentive program in effect						TOTAL	41255

FIGURE 4-4. SHIPMENT FORECAST SOLAR DOMESTIC HOT WATER SYSTEMS FOR TWO PRICING SCENARIOS

TABLE 4-4. SHIPMENTS FORECAST FOR 1983*

Region	Retrofit			New Construction			Total Shipments
	# Solar Add-Ons To			# Solar With Backup			
	Electric	Gas	Oil	Electric	Gas	Oil	
E. N. Central	7500	1100	450	3000	240	80	12370
S & S Atlantic	45000	1200	1000	20000	440	160	67800
Pacific	40000	8700	1750	17000	400	100	67950
N & NE	24000	6600	1300	2500	600	200	35200
Mid-Atlantic	12000	3200	650	7000	160	60	23070
Southwest	8000	300	50	6000	60	20	14430
South Tropic	13000	2200	400	2200	700	100	18600
Central	5000	600	120	3500	45	15	9280
W. N. Central	4000	600	120	1700	40	10	6470
E. S. Central	5200	-	-	1450	-	-	5650
N. W. Pacific	-	-	-	-	-	-	-
W. S. Central	3800	-	-	11000	100	30	14930
						TOTAL	275760

*assumes incentive program in effect

- a. The majority of consumers will consider economic attractiveness as the major justification for purchasing a solar system.
- b. The economics criteria will be based upon simple payback. The consumer will not use sophisticated cash flow analysis.
- c. The consumer will accept a yearly increase in energy savings due to price escalations.
- d. The consumer will discount energy savings claims when calculating payback. It appears that 50% solar contribution is a believable number. This is about 11 MM/BTU per year for an average family of 4.

The market is comprised of the following conventional systems: (1) electric, (2) oil fired, (3) gas fired. The costs of operation for each system vary with region and utility rate structure. To illustrate the analysis procedure, the mid-Atlantic region has been chosen as an example. Table 4-5 shows the energy price scenario used.

Tables 4-5 through 4-9 show the solar system yearly savings as perceived by the consumer for the various conventional options. Two starting dates are shown: installation in 1979 and installation in 1983.

TABLE 4-5. ENERGY PRICE SCENARIO F
ECONOMIC ANALYSIS

CONVENTIONAL SYSTEMS	PRICE (1978 \$)	ESCALATION RATE(%)
Electric (regular rate)	\$.06/KWH	8%
Electric (separate meter)	\$.035/KWH	8%
All Electric Rate (no separate meter)	\$.035/KWH (Oct-May) \$.06/KWH (June-Sept)	8%
Oil Fired	\$.50/gallon	8%
Gas Fired	\$3.00/MM BTU	10%

TABLE 4-6. CONSUMER EVALUATION OF SOLAR ECONOMICS VS.
ELECTRIC HOT WATER

- Believed Savings = 50% of yearly demand
- Savings escalation at 8%/yr
- Electric rate = 6¢/KWH in 1978, efficiency = 1

YEAR	SAVINGS	CUM. SAVINGS	CUM. SAVINGS 1983 INSTALLED
1979	210		
80	227	210	
81	245	437	
82	265	682	
83	286	947	
84	308	1233	286
85	333	1541	594
86	360	1874	927
87	389	2234	1287
88	420	2623	1676
89	453	3043	

TABLE 4-7. CONSUMER EVALUATION OF SOLAR ECONOMICS
VERSUS ELECTRIC HOT WATER

- Believed Savings = 50% of yearly demand
- Savings Escalation at 8%/year
- Electric Rate = 3.5¢/kwh in 1978, efficiency = 1

YEAR	SAVINGS	CUM. SAVINGS	CUM. SAVINGS 1983 INSTALLED
1979	124		
80	135	124	
81	145	259	
82	157	404	
83	170	561	
84	183	731	170
85	198	914	353
86	213	1112	551
87	231	1325	764
88	249	1556	995
89	269	1805	1244
90	291	2074	1513
91	314	2375	

TABLE 4. CONSUMER EVALUATION OF SOLAR ECONOMICS
VERSUS OIL FIRED HOT WATER

- Believed Savings = 50% of yearly demand
- Savings escalation at 8%/year
- Oil rate - 50¢/gallon in 1978, efficiency = .7

YEAR	SAVINGS	CUM. SAVINGS	CUM. SAVINGS 1983 INSTALLED
1979	56	56	
80	60	116	
81	65	181	
82	70	251	
83	76	327	76
84	82	409	158
85	89	498	247
86	96	594	343
87	103	697	446
88	112	809	558
89	121	930	679
90	130	1060	809
91	141	1201	950
92	152	1353	1102
93	164	1517	1266
94	178	1695	1444
95	192	1887	
96	207	2094	
97	224	2318	
98	241	2559	
99	211		

TABLE 4-9. CONSUMER EVALUATION OF SOLAR ECONOMICS
VERSUS GAS FIRED HOT WATER

- Believed Savings = 50% of yearly demand
- Savings escalation at 10%/year
- Gas Rate - \$3/mm BTU in 1978, efficiency = .7

YEAR	SAVINGS	CUM. SAVINGS	CUM. SAVINGS 1983 INSTALLED
1979	47	47	
80	52	99	
81	57	156	
82	63	219	
83	69	288	69
84	76	364	145
85	83	447	228
86	92	539	310
87	101	640	421
88	111	751	532
89	122	873	654
90	134	1007	788
91	148	1155	936
92	162	1317	1098
93	179	1496	1277
94	197	1693	1474
95	216	1909	
96	238	2147	
97	262	2409	
98			
99			

SECTION 5.0

DESIGN AND ANALYSIS

5.1 SYSTEM REQUIREMENTS

System design guidelines were established at the start of the program on the basis of market expectations and product planning. The parameters of marketability were incorporated with guidelines being established in each of the following categories:

- general
- performance
- physical characteristics
- operational characteristics
- reliability/durability
- ease of installation/interface
- maintenance/serviceability
- safety

Specific guidelines in each of these areas include the following:

General

- o The system shall operate in conjunction with conventional water heating systems.
- o System design shall be compatible with both retrofit and new construction applications.
- o The system shall be compatible with design requirements of an expanded system to provide space heating also.
- o The design shall minimize the number of different components.

- o Component modules shall be engineered to minimize ducting and field assembly/ installation.
- o The system shall be compatible with either a ground or a roof installation. Roof mounting shall require only one (1) major roof penetration.

Performance

- o The system shall provide at least 50% of the annual load of 333°K (140°F) water per ASHRAE draw schedule with less than 5.95m² (64 ft²) of collector area for all U.S. climatological regions excluding the Pacific Northwest.
- o Parasitic energy demand below 10% of the total yearly delivered energy is acceptable.
- o The system components shall perform at the above level for a period of 15 years after installation with recommended care and maintenance.

Physical Characteristics

- o Component size and weight shall be compatible with installation in standard residential construction.
- o The collector shall have a painted, attractive appearance. The frame and manifold shall be rust-proof for the collector design life.
- o The energy transfer module shall have a finished exterior appearance comparable to that associated with conventional HVAC indoor equipment.
- o All components shall be configured for easy handling, serviceability, and appearance. Components shall not require more than two men for handling.

Operational Characteristics

- o The system shall perform its function automatically.
- o The system shall not generate operational noise significantly higher than that from conventional fossil-fired hot water systems or that from conventional warm air heating systems.

Reliability/Durability

- o The failure rate shall be less than 3% for the first five years of operation.
- o Durability shall be such as to satisfy the design life requirement (15 years) and the operational environments set forth in Tables 3-2 and 3-3.

Ease of Installation/Interface

- o The system shall be designed to minimize installation time. Components shall present a simplified interface to each other.
- o Components shall be designed to interface with standard field supplied components.
- o Installation operations shall be performed using standard tools.

Maintenance/Serviceability

- o Maintenance shall be restricted to periodic inspection and cleaning of the collectors.
- o The ETM shall incorporate components that are non-corrosive and self-lubricating.
- o The system shall be designed for rapid removal and replacement of component parts.
- o The placement of components inside the ETM shall provide equal accessibility for either bottom or side entry.

Safety

- o The system shall be designed and fabricated in accordance with applicable industry and government safety standards and regulations.

5.2 DESIGN DEVELOPMENT

5.2.1 COLLECTOR OPTICS OPTIMIZATION

An analysis was performed to determine what effect variations in configuration, reflector properties and cost, and would have on the performance of an air collector system which utilized the GE vacuum shroud. The vacuum shroud employs a vacuum between the absorber and the ambient environment to reduce significantly the energy losses by convection and conduction.

The performance of the various collector configurations was defined using an hourly computer simulation at two reference cities; Fort Worth, TX and Boston, MA. These cities were selected because they represent two diverse climates which exist in the continental U.S. The hourly simulations were made utilizing the actual weather conditions of representative years to accurately assess the environment which the collectors would experience.

The collector panel for this analysis had a specific size constraint. The dimensions of the current GE TC-100 collector panel was specified to allow for future incorporation of this collector without modifying the present support hardware. This constraint limited the available concentration ratios obtainable to those which correspond to specific shroud configurations. Configurations of 8, 10, 12 and 15 shrouds per panel were simulated.

The reflector configuration associated with specific shroud numbers was defined using the tangency method. As shown in Figure 5-1, all rays entering the cusp of fewer than 10 shrouds/panels when the entrance angle is less than 90° would be reflected onto the shroud.

5.2.1.1 Collector Comparison

The different collector configurations were compared as a function of the equivalent cost of each system. The equivalent cost was defined as the cost of a system which would be required to collect a specified amount of energy.

Figure 5-1. Reflection Characteristics of Cusp Reflector

After the collector efficiency of each configuration was defined, the collector size was defined such that each configuration would collect the same amount of energy. This procedure would result in the electrical power requirements, storage capacity, and fuel costs of each configuration being nearly identical and therefore, not parameters to be considered. The costs were generated for the collector panels and headers only. No other equipment costs were factored into this analysis.

The equivalent cost of each system was based on the projected cost of each component utilizing 1977 dollars. Table 5-1 presents the cost data utilized. Two different shroud costs were incorporated into the analysis. These two values were used to determine the effect of a rapid development and expansion of the solar system sales (\$2/shroud) and the nominal predicted development and expansion rate (\$6.36/shroud). These costs were the actual material costs and did not have any margins incorporated into them. Profit, depreciation, overhead, benefits and labor costs were not part of this analysis.

TABLE 5-1

Cost of Material for SOLAIR Collector

	<u>Cost</u>
Manifold Frame	\$6.25/Linear Foot
Reflector	\$0.85/Square Foot
Metal Insert into Shroud	\$1.00 each
Shroud	
nominal growth	\$6.36 each
accelerated growth	\$2.00 each
	1977 dollars

5.2.1.2 Analysis

The collector efficiency, defined as the total energy deliverable to the storage tank divided by the total incident energy on the collector surface was generated for the cusp and flat reflector configurations at the two sites. The flat reflector configuration also had the reflectivity varied. Figure 5-2 shows the Boston domestic hot water system. The dashed line is the performance of a cusp configuration in a space heating application. While the magnitude is slightly below the domestic hot water level, the trends and the ratios of the relative values are the same. Therefore, the results obtained for the domestic hot water application are considered applicable to the space heating application.

The cusp efficiency starts a rapid decrease when fewer than 10 shrouds per panel are utilized. This trend is continued when the reflector is removed. The flat reflector performance appears to approach the cusp configuration. Variation in the flat reflector reflectivity produces a linear variation in the collector efficiency over the range of reflectivities examined.

The Forth Worth collector efficiencies are presented in Figure 5-3. This location has the same trends that were exhibited by the Boston system. A slight variation in the magnitude is noted due to the change in the direct to diffuse ratio and on the slight variation in ambient temperature. The ambient temperature variation will effect the insolation losses which are incorporated into the collector efficiency.

8-5

Figure 5-2. Optimum SOLAIR Performance Collector Collection Efficiency
Boston Domestic Hot Water

Figure 5-3. Optimum SOLAIR Performance Collector Collection Efficiency
Fort Worth Space Heating

The equivalent cost of each system was obtained based on systems supplying the same amount of energy to a storage tank. Utilizing the material costs of Table 5-1, the equivalent costs ratio curves (Figures 5-4 through 5-7) were generated. The cost ratio is the equivalent cost of each system divided by the lowest equivalent cost. Thus the ratio is simply a percentage increase in cost of the various configurations.

The lowest cost system for the Boston climate with a \$6.36 shroud would be an 8 shroud cusp configuration. The increased performance of the 10 shroud system (Figure 5-2), is not as large as the increased material cost. Usage of a flat reflector only increases the system cost. Reduction of the shroud cost to \$2 would increase the size of the optimum to 10 shrouds per panel.

Figure 5-5 shows the equivalent cost ratio for Boston as a function of the reflectivity. The lowest cost system is 10 percent higher than the cusp equivalent cost. Removal of the reflector would decrease the system costs but now an upper limit in the reflectivity would have to be established. Reflectivities greater than 0.5 would generally not be available from normal building materials. If special attempts were made to increase the reflectivity, the cost of these efforts would have to be incorporated into the equivalent cost of the system. With the nominal reflectivity, the \$2 shroud system would be economically viable when this reflectivity was greater than 0.4. Increasing the shroud cost to \$6.36 would eliminate the no reflector case from consideration. The Fort Worth system has the same trends (Figures 5-6 and 5-7) as the Boston system but has larger cost ratios.

Figure 5-4. Optimum SOLAIR Performance Equivalent Cost Comparison (Shroud Number) - Boston Domestic Hot Water

Figure 5-5. Optimum SOLAIR Performance Equivalent Cost Variation (Reflectivity) - Boston Domestic Hot Water

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Figure 5-6. Optimum SOLAIR Performance
Equivalent Cost Variation (Shrouds)
Fort Worth Space Heating

Figure 5-7. Optimum SOLAIR Performance
Equivalent Cost Variation (Reflectivity)
Forth Worth Space Heating

With the current predicted development rate of the glass shroud, the 8 shroud per panel cusp configuration appears to be the most cost effective system. Usage of a flat reflector would significantly increase the equivalent cost for both shroud cost conditions. The cost gain obtained by removing the reflector would not be as large as was observed for the Boston system. The reflector-less system would not be cost competitive even with a \$2 shroud cost system.

5.2.1.3 Other Considerations

The performance characteristics of several solar air collector configurations were generated to define a cost effective system. Variations in reflector reflectivity and shroud number (concentration ratio) were made to define the collector efficiency of each configuration. These collection efficiencies were then utilized to define the equivalent cost of each collector configuration. This method was used to compare the cost of each configuration on an equal basis to the others. The system having the lowest collector cost would be the most cost effective approach.

Comparisons of equivalent costs of systems operating in Forth Worth, Texas and Boston, Massachusetts, showed that 8 shroud assembly was the most cost effective system when shrouds cost \$6.36 each. Reduction in the cost to \$2 each would increase the system to 10 shrouds per panel.

The difference in costs between the 8 and 10 shroud systems was less than 10 percent. Thus, the labor and maintenance costs must be examined. A system having fewer parts (shrouds) will have lower installation and maintenance costs. The 8 shroud system would have this additional advantage which was not incorporated into the equivalent cost.

Removal of the reflective surface, while decreasing the collector cost, would penalize the collection efficiency to a larger extent. The net effect would be an increase in the equivalent cost of reflector-less systems. Modifications made to the roof to increase its reflectivity would have to be incorporated into the collector costs.

5.2.2 COLLECTOR HEAT TRANSFER OPTIMIZATION

5.2.2.1 SOLAIR 1A Test Data Evaluation

Test data obtained with the Solair 1A demonstration modules indicated collector efficiencies significantly less than expected. A comprehensive evaluation of all of the contributors to collector efficiency was undertaken in an effort to determine if changes in design, operational characteristics, experimental techniques, or a combination thereof could result in improved efficiency. The principal contributors to collector performance were identified as the following:

1. Vacuum tube performance
2. Air tube design/optimum heat transfer
3. Optimum flow rate, Δp , distribution
4. Reflector design performance
5. Duct insulation/air leakage
6. Experimental Measurement accuracy

Each of these considerations were investigated in depth to the extent that the design development tests were rerun in the laboratory to help satisfy Nos. 2, 3, and 4 above:

The forty-eight vacuum tubes used in the Solair 1A test modules were obtained from one production batch and were determined to be representative of production quality. Nine of the 48 air tubes were instrumented with thermocouples for the tests. In test measurements, these air tube temperatures were observed to be

consistently biased as much as +5% about the mean, which is considerably greater than standard experimental accuracy on thermocouples. With the ΔT 's ($\sim 40^\circ$) measured on Solair 1A, this variation in the air tube temperatures reflected about a +15% variation in heat transfer and likewise, a +15% on tube efficiency. A stagnation test was conducted and the resulting air tube temperatures were obtained as shown in Figure 5-8. Also, the operating air flow rate across the tube bank was determined from duct pressure distribution measurements. A definite correlation was observed between the variations in tube temperatures during operation and at stagnation, especially when accounting for the effects of variations in flow rate.

Therefore, it was concluded that under a given set of operating conditions, normal variations in tube quality could also produce significant variations in tube performance, especially with air applications at low ΔT 's. This conclusion is important in terms of the laboratory tests conducted with a single vacuum tube, in terms of evaluating individual tube data from Solair 1A tests and Solair II verification tests, and also in terms of evaluating the effects of diffuse reflectors which produce lower absorber temperatures and lower flow ΔT 's.

Heat Transfer. An evaluation of the air tube configuration used in Solair 1A indicated that, at the flow rates tested, the Reynolds number in the annulus were not high enough to provide turbulent heat transfer, primarily because the hydraulic diameter was too small. Also, existing laboratory test data was not adequate to define the conditions for optimum heat transfer and air tube designs. A revised test plan was proposed and tests were conducted as reported in paragraph 5.2.2.2:

Figure 5-8. SOLAIR 1A - Effect of Vacuum Tube Performance

Optimum Flow Rate - p - Flow Distribution. An increase in efficiency was anticipated by increasing the flow rate with a resulting increase in parasitics. Again, insufficient performance data existed to make the necessary trade-offs, and the necessary test requirements were included in the revised test plan. A revision in the module inlet and outlet duct design was recommended to overcome the large duct losses to aid in reducing the parasitics and to allow higher flow rates if necessary to increase efficiency. The large pressure losses measured in the Solair 1A ducts are indicated in Figure 5-9 with the resultant variations in the flow rates in the individual tubes. In order to provide a more uniform pressure distribution and reduce the losses in the inlet duct, an end-feed was recommended with a duct configuration as shown in Figure 5-10. This duct configuration provides a low aspect ratio inlet to minimize inlet losses, provides a minimum interference effect of duct flow with air tube penetrations, and also should reduce the annulus inlet and tube outlet losses.

Reflector Design Performance. The diffuse reflector design used in Solair 1A was a flat coilzak Aluminum sheet with a measured reflectance of about 0.80. In order to get a measure of the effectiveness of the diffuse reflector on Solair 1A, a test was conducted to obtain data with and without the reflector in place. The results are shown in Figure 5-11 and demonstrate a rather well defined change in performance with and without the reflector. The data indicates that the reflector contributes about 25% of the total energy collected at these operating conditions with good solar insolation. Comparative reflector performance tests were also recommended and included in controlled laboratory tests:

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Figure 5-10. Recommended SOLAIR Duct Design

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Figure 5-11. Effect of Diffuse Reflector on SOLAIR Performance (150 cfm)

Duct Insulation/Air Leakage. The possibility of large insulation losses or air leakage was thoroughly investigated because of some significant temperature differences from tube outlet to duct outlet measurements on Solair 1A. Air leakage and insulation losses were concluded to be small, but considerable heat transfer from the hot outlet duct to the cold inlet duct through the adjoining un-insulated wall was also concluded to occur. Although the energy cross coupling was not lost, it would tend to raise the inlet temperature and have a minor effect on reducing collector efficiency. The duct redesign for Solair II includes insulation between the hot and cold ducts.

Experimental Measurement Accuracy. Measurement accuracy and good instrumentation are prerequisites of any performance verification test. Two measurements in particular are important to determine the efficiency of the Solair 1A modules accurately. These are the solar insolation and flow rate measurements. Assessments indicated that the insolation measurements might be low by 0 to 20%, depending on the time of day. Two checks on the hot wire anemometer used to determine flow rates on Solair 1A indicated measurements could be 20 to 25% too low which would result in a 30 to 35% increase in efficiency from mid-to-low thirties to the mid-to-low forties. The recommendation was made that the instruments used in the Solair 1A test be checked for accuracy and that, in future verification tests, provisions would be made to use good instrumentation that would be adequately calibrated for determining measurement accuracy.

5.2.2.2 Design Development Tests

The objectives of the design development tests were to provide parametric data for the optimization of both optical and heat transfer characteristics. The laboratory tests were concerned primarily with performance optimization of the air tube and reflector design configurations as a function of flow rate, insolation level, and pressure drop. The acquisition of parametric performance

data would allow subsequent system cost trade-offs to provide the most cost effective design and operating parameters.

The design development tests were conducted in a laboratory using a tungsten-iodide lamp as a controlled solar simulation source and a single vacuum tube with both a cusp and a diffuse reflector as the optical test considerations. The radiation intensity and distribution of the lamp was measured and controlled using a hand-held pyranometer. Three air tubes in standard sizes (1", 1½", 1 3/8") were tested at flow rates from 2 to 6 cfm. Each air tube was installed concentrically within the vacuum tube so that the air was forced either into and down the annulus and out the tube or vice-versa. The air flow was provided by a laboratory industrial nitrogen supply and controlled and monitored with a parallel circuit of two valves and two air flow meters feeding the test. Selected air tubes were also modified and tested to evaluate the effects of turbulence generators (A. Knurled roughness; B. Spiral tube configuration; C. Rings (2) of holes) on the heat transfer characteristics in the annulus.

Measurements obtained during the tests included:

- a. Outcoming air temperatures (3 T/C's)
- b. Incoming air temperatures (1 T/C - supply)
- c. Annulus air temperatures (3 T/C locations)
- d. Static pressure drop (inlet to outlet)
- e. Flow rate pressure drop - cfm
- f. Insolation rate - BTU/FT²/HR

Data were obtained at two insolation levels and at selected flow rates for each tube/reflector configuration tested. Temperatures were monitored continuously, and data were taken when thermal equilibrium was obtained.

The pertinent laboratory test results are shown in Figures 5-12 through 5-16. Some caution is noted in the interpretation of these test results because of the validity of the solar simulation source and the resultant optical response. The energy wavelength spectrum and collimation does not accurately simulate incident sunlight, and the pyranometer readings only establish a level of radiation measurement. The test data indicated that the lamp is providing and the tube is absorbing about 50% more energy than in sunlight at the comparable solar pyranometer reading. Although the insolation level and optical response of the reflectors and absorber coating is in doubt, comparative temperature and heat transfer data are considered valid and do, in fact, provide adequate performance measurements to insure optical correction factor to the insolation measurements and allow a comparison of test variables as they affect collector efficiency.

The effects of air tube size and reflector configuration on the temperature rise across the collector tube is shown in Figure 5-12 for two insolation levels as a function of flow rate. The data indicates a significant effect of the reflector configuration on energy collected with the 1 3/8" air tube. Comparable temperature data derived from the Solair 1A tests with a diffuse reflector is shown superimposed at a measured flow rate of 4 cfm/tube, which apparently may have to be corrected to approximately 5.2 cfm. Although the temperature rise is in relatively good agreement between the lab test and Solair 1A test, the data does tend to indicate that the performance of the diffuse reflector in the laboratory environment is reduced as expected. Also, the measured ΔT in Figure 5-12 demonstrates that the 1 1/4" O.D. air tube performs best at both insolation rates and at all flow rates.

Figure 5-12: SOLAIR $\Delta T = (T_{out} - T_{in})$ - Across Tube vs. Cfm
 with Annulus Feed
 Turbsten-Iodide Lamp Solar Simulation

The effects of tube size on the heat transfer rates is shown in Figure 5-13. The highest heat transfer rates are shown to occur with the intermediate tube size ($1\frac{1}{4}$) at the lower flow rates. The size of the annulus and the appropriate hydraulic diameter have a big effect on the film coefficients and Reynolds number of the flow, and the results tend to indicate that more turbulent flow heating is achieved with the $1\frac{1}{4}$ " and 1" tubes than with the $1\frac{3}{8}$ " tube. Turbulent heating is also indicated with the 1" O.D. as expected, but the heat transfer coefficient is lower because of the increase in the hydraulic diameter of the annulus. The significant parameter in Figure 5-13 is the flow rate, and the results indicate that with a $1\frac{1}{4}$ -inch tube, there is no significant performance advantage in operating at flow rates above 3 cfm/tube.

The effects of the reflector configuration on the heat transfer rate is also shown in Figure 5-13, with a $1\frac{3}{8}$ " O.D. tube. Under constant test conditions, the difference in heat transfer rate is a measure of what is being absorbed by the vacuum tube, except for some minor effects of flow temperature.

The effects of annulus versus tube feed and the effects of knurled roughness on the heat transfer rates are shown in Figure 5-14 with the $1\frac{1}{4}$ " O.D. tube. The observed differences in heat transfer rate between annulus and tube feed at the same test conditions appear consistent with several test configurations indicating better performance with annulus feed. A logical explanation of the phenomena is not apparent but must be related to the difference in annulus entrance and exit effects. Effects of roughness (approximately 6 inches of knurled roughness on tube at the annulus entrance) was obtained with a $1\frac{1}{4}$ " O.D. tube. Only a slight difference in the heat transfer rate is observed, as would be expected, with the subsequent conclusion that turbulent heating is obtained with the $1\frac{1}{4}$ " O.D. tube without roughness at flow rates above 3 cfm.

Figure 5-13. SOLAIR - Heat Transfer Rate vs. Flow Rate
 $I' = \text{Const.} = 207 \text{ BTU/FT}^2$ Annulus Feed

Figure 5-14. SOLAIR - Heat Transfer Rate vs. Flow Rate

$I' = \text{Const.} = 207 \text{ BTU/FT}^2$

1 1/4" Tube Size

The static pressure drop across the tubes is shown in Figure 5-15, as a function of flow rate and tube size. Some effects of flow temperature and tube roughness are observed, but results appear consistent from test to test. The 1½" O.D. tube indicates the lowest pressure drop which is consistent with theoretical predictions that show minimum pressure drops with annulus-to-tube area and velocity ratios of 1.0.

In Figure 5-16 an attempt was made to correct the simulated insolation data and interpret the laboratory results in terms of collector efficiency. An insolation correction factor was obtained by assuming a maximum collector efficiency of 60% for the highest measured heat transfer rate at each level of insolation. Using the corrected values of insolation does collapse all of the high and low temperature data with relatively consistent trends as shown in Figure 5-16. Solair 1A test data with a diffuse reflector and an equivalent 12 tube, 4-foot module did indicate efficiencies from 30-37%. The primary conclusion derived from Figure 5-16 is that, on a comparative basis, it is not unreasonable to expect collector efficiencies from 50 to 55% with an 8 tube - .1X cusp reflector module (4' X 4') operating at 3 cfm and with a $\Delta T = 100^{\circ}\text{F}$ at the highest insolation rates.

Figure 5-15. SOLAIR - Pressure Drop ΔP vs. Flow Rate

NOTES : K_1 DETERMINED
BY LIMITING $\dot{q}/K_1 K_2 I'$ TO
A MAXIMUM OF 0.60

: I' IS AVERAGE OF
SEVEN (7) READINGS USING
HAND HELD PYROMETER

: VALIDITY OF DIFFUSE
REFLECTOR PERFORMANCE IN
LAB. TEST IS QUESTIONABLE

Δ 1 1/4" OD TUBE } $I' = 207$

$\square$ 1 3/8" OD TUBE

SHADED SYM. - $I' = 296$

FLAGGED SYM - WITH ROUGHNESS

I' = LAMP INSULATION MEAS.

K_1 = LAMP CORRECTION FACTOR

1.582 @ 207 BTU/FT²

1.463 @ 296 BTU/FT²

K_2 = CONCENTRATION FACTOR

1.77 = CUSP

1.23 = DIFFUSE

Figure 5-16. SOLAIR - Laboratory Simulation-Collector Efficiency
versus Flow Rate

5.2.2.3 Design Recommendations

- a) A redesign of the inlet/outlet duct geometry was undertaken to minimize the parasitic pressure losses at the inlet and through the inlet duct and to provide a more uniform flow distribution through the tubes. The redesign should reduce the blower power requirements by 30 - 40% and could also result in some improvement in performance.
- b) A redesign of the collector module was accomplished to incorporate the 1X cusp reflector into a nominal 4' X 4' module with 8 vacuum tubes instead of an equivalent 12 tube module with the diffuse reflector as used in Solair 1A. Laboratory test results indicated a large potential increase in collector performance at constant insolation which was in addition to a higher expected off-peak daily performance. Also, the 8-tube cusp reflector module was expected to result in about a 20% cost reduction.
- c) A 1 $\frac{1}{4}$ -inch OD air tube was incorporated to replace the 1-3/8-inch OD tube used in Solair 1A. Laboratory tests indicated the 1 $\frac{1}{4}$ " tube would provide higher performance than either the 1" or the 1-3/8" tubes and would also provide the lowest pressure drop. Standard size aluminum tubing was recommended without turbulence generators (added roughness, etc.) and without surface treatment or painting.
- d) A design flow rate of 3 cfm per tube with annulus feed was established to maintain good heat transfer characteristics with high fluid outlet temperatures (as desirable for hot water applications) as well as minimize the blower power requirements.

e) Design features were needed to insure that the air tube would be concentric within the vacuum tube when installed and that the length of the tube would allow a 2-inch clearance (as a minimum) at the end of the vacuum tube.

5.2.3 DESIGN DEFINITION

The above recommendations were incorporated into a design concept that was documented with drawings to the detail required to fabricate the development units described earlier in this report.

SECTION 6
FABRICATION

Components for two systems were fabricated to the prototype design definition. Sheet metal parts were procured from a sheet metal specialty shop and assembled at the Valley Forge facility. A specialty vendor provided services to foam in place the insulation in the header of the collector. Reflectors and evacuated tube absorbers are standard parts used in the TC-100 evacuated tube liquid solar collector and were purchased from stock. Assembly was carried out under the direction of the design engineer to provide feedback into the design for future builds.

Assembled units are shown in Figures 1-2, 1-3, and 3-6. The interior of the Energy Transfer Module is shown in Figure 6-1.

Figure 6-1. Interior of Energy Transfer Module

SECTION 7

DESIGN VERIFICATION TESTS

7.1 FUNCTIONAL TEST

7.1.1 Description

A prototype Solair II system, including four collector modules, an Energy Transfer Module, and the appropriate duct configuration was assembled to perform an operational checkout and evaluate system flow characteristics for the system. The prototype ETM employed a pump-motor-blower combination, assembled on a common shaft as supplied by MARCH, INC. The water pump was a MARCH Model #2 pump rated for 5 gpm with a 13 foot head. The blower was a FAFCO, rated for 160 cfm with a pressure drop of 0.7 inches of water. Lab tests were conducted to evaluate system air flow rates, pressure drops, pressure distributions, and parasitic power consumption as a function of 2, 3 or 4 collectors, assembled in parallel with a common ETM.

7.1.2 Test Set-Up

The system was assembled as shown in Figure 5-17. A six-inch diameter AIRMONITOR calibration duct was installed in the air supply duct to determine the air flow rates, using an inclined leg manometer for measuring the velocity pressure in the calibrated duct. The ETM water lines were connected to a water reservoir (bucket) to circulate water through the pump, but no effort was made to measure the water flow rates. The electrical connection to operate the ETM motor was monitored with a watt meter to determine the electrical power consumption. Holes were drilled in the collector supply and return ducts to measure the air pressure drop across the tubes and determine the pressure and flow rate distributions. Collector modules #1 and #4 were removed to test a two-collector system, and modules #1, #2, and #3 were used to evaluate the 3-collector system.

FIGURE 7-1. TEST SET-UP

7.1.3 Results

The system flow rates were measured for each of the 2, 3 and 4 collector configurations using the AIRMONITOR calibration duct, as shown in Table 7-1a.

The measured velocity pressure drop (ΔP) was converted to a flow velocity using the AIRMONITOR calibration curve shown in Figure 7-2. The flow rate in cfm was derived by multiplying the velocity by the cross-sectional area of the duct. With a uniform pressure distribution in the collector ducts being assumed, the cfm per tube was obtained by dividing the system flow rate by the number of tubes in each configuration. The duct pressures were measured, referenced to atmosphere for the 4-collector configuration as shown in Figure 7-1. The pressure variation in the supply duct from the center feed to the outer end of the collector duct was about 13% of the measured collector pressure drop of 0.47 inches H_2O , and the corresponding variation in the return duct was approximately 3%. Results are tabulated in Table 7-1b.

Table 7-1. Flow Data

a. System Flow Rate

NUMBER OF COLLECTORS	VEL. Δ P IN. H ₂ O	VEL. fpm	SYST. RATE cfm	RATE cfm/tube
4 (32 tubes)	.054	930	183	5.71
3 (24 tubes)	.050	900	177	7.36
2 (16 tubes)	.032	715	140	8.77

b. System Pressure Distribution

PRESSURE - IN. H ₂ O	END POSITION A	CENTER POSITION B	MEAN - DISTRIBUTION	
			% Δ P	% cfm
SUPPLY DUCT	0.40	0.46	13%	4%
RETURN DUCT	-.045	-.06	3%	1%
ACROSS TUBES	0.445	0.520	$\pm$ 8%	$\pm$ 3%

Figure 7-2. SOLAIR II - AIRMONITOR Calibration Duct

The purpose of the Solair II redesign was to minimize the duct losses and provide a uniform pressure and flow rate distribution through the tubes. By comparison, the variations in the supply and return duct pressure distributions for Solair 1A were 110% and 75%, respectively, with about a 50% variation in the ΔP across the tubes. The corresponding variation in flow rate would vary with the square root of the pressure drop. Therefore, the Solair II redesign provides a significantly better flow distribution through the collector tubes, which should enhance collector efficiency.

The system operating characteristics are plotted in Figure 7-3 as a function of cfm/tube for the three collector configurations. The pressure drop across the collector tubes was observed to be lower than indicated in the tube simulation tests. The lower pressure drop in the system tests was attributed to a better annulus inlet configuration and possibly to the system design modification to maintain the alignment and concentricity of the metal tube inside the vacuum tube. A significant reduction in the system flow rate was observed with the two collector system, which was due to the increase in pressure drop with fewer tubes and higher flow velocities in the tubes. The power consumption of the motor was a function of both the pressure drop and the flow rate of the air system and showed a slight decrease in the power requirements as a function of the number of collectors.

The resultant electrical parasitic energy requirements were evaluated qualitatively, without a detailed solar simulation study by assuming an average solar insolation of 250 BTU/FT²/HR during periods of operation and an average collector efficiency of 45%. The estimated parasitic energy requirements in terms of percent of collected energy are shown in Table 7-2 as a function of the number of collectors.

Table 7-2. Parasitic Energy Requirements

NUMBER OF COLLECTORS	AVERAGE ENERGY CAPTURE	PARASITIC POWER	PARASITIC ENERGY	% PARASITIC
	BTU/HR	KW	BTU/HR	%
4	6600	0.145	495	7.5
3	4950	0.143	488	9.9
2	3300	0.130	444	13.5

Figure 7-3, SOLAIR II System Operation Characteristics

7.1.4 Conclusions

The air flow tests provided some basic operational performance data which supported the design development of the Solair II system and which could be used for performance verification studies. The following conclusions were derived from the results.

- a. The Solair II design appears to provide good, uniform pressure and flow distributions through the collectors, relative to the Solair IA design approach.
- b. The pump-motor-blower combination appears to provide an acceptable level of parasitic energy requirements and appears to be adequate for prototype development and for the first generation TCA systems.
- c. The blower capability is more than adequate and can be used effectively with as many as six collectors.
- d. The current water pump capacity (5 gpm @ 13 ft. hd.) is excessive and can be reduced significantly if standard pumps are available, which will reduce the parasitic energy requirements and system cost.
- e. Likewise, the motor capacity may be excessive, along with the blower and water pump, for an efficient, cost-effective system, but the oversized capacities are advisable for the first generation TCA systems.

7.2 System Performance Tests

As the program hardware units were completed, the original testing plan was re-evaluated. It was concluded the planned mobile home test bed had some serious deficiencies. The season of the year (Fall) does not provide the best of weather for solar testing and it was anticipated that the 30 days planned for the test would not provide representative data. A solar domestic hot water system rating standard is being defined and it was considered desirable that the test data to be obtained be compatible with the standard. Statements of Work were generated and quotations obtained, but costs exceeded the program budget. Arrangements were made to characterize the performance unit at the National Bureau of Standards test facility. This will occur after the period of performance included in this report.

Likewise, the plan is to install the second sytem in a residence to provide realistic long term performance data. This will also occur after the period of performance included in this report.

SECTION 8

PRE-PRODUCTION DESIGN

During the program, the SOLAIR II design was identified as a marketable product for the domestic hot water application and non-contract commercialization activities were initiated. It was recognized that large scale production could not be justified without more confirmation of a large volume market. Production on a small scale was needed to provide units for field tests and market surveys. Consequently, an initial lot of six systems was fabricated. The design used for this pre-production lot was essentially the SOLAIR II design. Changes as noted during the fabrication of the initial two units, were incorporated in the design. These are listed in Table 8-1. This pre-production design will serve as the basis for future design evaluations as the product is introduced into the commercial market.

Table 8-1

Changes Incorporated into the Pre-production Design

<u>DESIGN</u>	<u>PURPOSE</u>
a. Solar Collector	
Increase Frame Gage to 24	Increased Stiffness
Increase Frame Flange Width	Increase Stiffness
Changed Length of Inner Duct	Allow for Gasket
Add Latches to End Cap	Eliminate Screws
Modify Method of Providing Shroud Clearance in Foamed Part	Simplify Fabrication Process
Modified Grommet Design	Increase Interference with Shroud
Changed Mounting Tab Dimensions	Correct Drawing Error
b. ETM	
Changed Method/Location of Inter-connection to Heat Exchanger and Pump	Allow Subassembly Technique
Changed Location of Pump/Motor	Simplify Ducting
Note: The following additional changes were made in Post Contract Prototype Assembly Activities	
Chnaged ETM Pump Model	Meet Operating Requirements
Changed Paint Color	Customer Preference
Reoriented Pump/Motor/Blower	Access to Lubrication Points
Added Motor Support Brackets	Increased rigidity