DOE/NASA CONTRACTOR REPORT

DOE/NASA CR-150613

PRELIMINARY DESIGN PACKAGE FOR SOLAR HEATING AND HOT WATER SYSTEM

Prepared by

Fern Engineering Company
P. O. Box M
Buzzards Bay, Massachusetts 02532

Under Contract NAS8-32246 with

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

For the U. S. Department of Energy

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Solar Energy

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This work was accomplished under the technical management of Mr. Mitchell Cash, George C. Marshall Space Flight Center, Alabama.

16. ABSTRACT

This report is a collection of documents that were submitted by the Fern Engineering Company for the preliminary design review on the development of two prototype solar heating and hot water systems. The information contained in this report includes system certification, system functional description, system configuration, system specification, system performance and other documents pertaining to the progress and the design of the system.

This system, which is intended for use in the normal single-family residence, consists of the following subsystems: collector, storage, control, transport, and Government-furnished Site Data Acquisition. One of the two prototype units will be installed in Lansing, Michigan, and the other in Tunkhannock, Pennsylvania.

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CONTRACT: NASA 8-32246 | Document No. 7002

SUBJECT: System Certification | Job No. 198

AUTHOR: P. Levine | Date: 12/30/76

1. Scope

This document partially fulfills the requirement that the system is certified to meet nationally recognized standards and codes, summarizing the results of the initial meeting with UL personnel.

2. Meeting with Underwriters Lab Personnel on 12/28/76

Mr. Enos Toomsalu, Associate Managing Engineer, Heating, Air Conditioning & Refrigeration, Dept. of UL, and Mr. David Engblom, Engineer of UL visited Fern Engineering on 12/28/76 to discuss the certification program and review currently available drawings and descriptions. The following schedule was set forth:

1) UL evaluation report by January 20, 1977.

2) UL representative will attend PDR on January 20th to review certification plan.

3) A final certification plan will be defined at the PDR covering:

Fire hazards
Chemical hazards
Mechanical hazards

The following information was noted at the meeting:

1) There is generally a 90°C limit on combustibles (e.g. wood) for continuous operation.

2) UL 883, Standard for Safety, "Fan-Coil Units and Room Fan-Heater Units" was felt to cover the Energy Transport Module functions.

3) The BOCA Basic Codes are recommended for regions east of the Mississippi; the IAPMO Codes are recommended west of the Mississippi.

4) NFPA Codes 90-A and 90-B are recommended as Fire Safety Codes.

5) Water tanks must be tested to 300 psi. A maximum temperature of 190°F is currently in use for mobile homes; if temperature control is better than $\pm 5°$ F then one can go to 200°F.

6) Use of Urethane/Isocyanurates in the collectors will be reviewed by their Fire Hazard Specialists. In general, it was felt that the solar collector is not merely an extension to the ventilation ducting system, as it is remote from fire sources; hence the use of Urethane appears acceptable if done properly.

7) The wind load resistance will be reviewed by their staff.

8) Installation of the collectors to the roof is felt to be covered by the Basic Boca Codes.

9) Automatic flue dampers for use with oil fired furnaces have been UL approved and a new standard, recently published, will be forwarded to Fern; use with gas fired systems is more hazardous because of the benign odor of gas fumes.

The complete system operation and drawings were reviewed and no significant defects were noted.

Document No. 001-4

CONTRACT : NASA 8-32246

Job No. 198

SUBJECT : System Functional Description

Date: December 1, 1976

AUTHOR : P. Levine

1. Scope

This document is intended to partially fulfill the data requirements for the System Performance Specification. The system configuration is defined, and criteria related to the configuration selection rationale are stated.

2. System Functional Description

The system configuration is depicted in Figure 1. The system consists of six modular subsystems :

1.) Back-up oil fired warm air furnace subsystem.

2.) Back-up electric domestic hot water heater.

3.) Solar collector subsystem

4.) Energy transport module subsystem.

5.) Energy storage subsystem.

6.) Control subsystem.

There are five primary functional modes, namely ;

Mode 1. Direct Solar Heating of air used for space heating.

Mode 2. Extract Stored Energy for space heating and/or for domestic hot water heating.

Mode 3. Storage of Solar Energy for later use for space heating and/or for domestic hot water heating.

Mode 4. Use of Back-up Furnace.

Mode 5. Use of Back-up domestic hot-water heater.

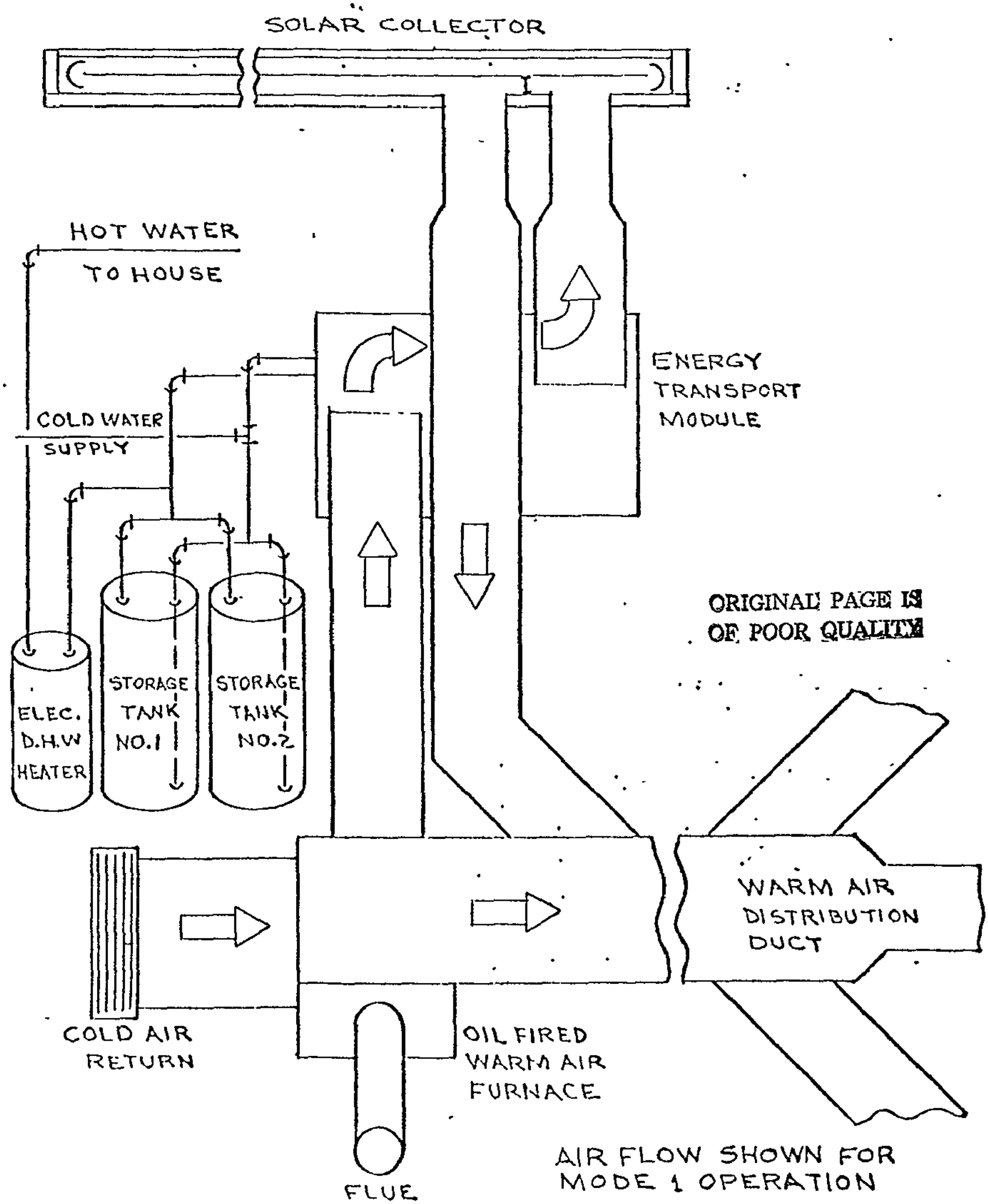
The system fulfills the entire demands for heating warm air and domestic hot water for a residence. When there is no solar input due to extended periods of cloudiness, the conventional oil-fired warm air furnace and the electric hot water heater supply the demand for heat. The electrical and mechanical interfaces between the solar and back-up systems are designed so that the back-up system is free to operate in a conventional way, totally unimpaired by the solar system.

The solar collector is a two-pass air heating collector so air enters and returns through the same end of the collector. The collector area is nominally 200 square feet. The solar collector can directly heat air being recirculated through the residence, if there is a heating demand. If the heating demand is larger than the solar collector can handle, the warm air furnace is automatically started, in which case any solar energy available is diverted to storage. If there is no demand for space heating, collector air flow is thermostatically controlled to recirculate through the collector and to transfer heat to storage.

The energy transport module is a prepackaged unit containing the pump and fan needed to transfer heat via air or water flow. An air-to-water heat exchanger is also contained in the module, so heat can be transferred from air-to-water or water-to-air. The pump and fan are controlled automatically by the state of thermostat sensors in the control subsystem.

The energy storage consists of two 120 gallon domestic hot water heater tanks. Cold municipal water is supplied to the storage tank, and DHW draw uses the solar heated water, reducing the demand on the back-up electric hot water heater. The pump and heat exchanger located in the energy transport module draws water from near the tank bottom, passes it through the heat exchanger and returns it to the top of the tank when storing energy. When extracting energy, the pump is reversed to draw water from the top of the tank, pass it through the heat exchanger and return it to a port near the tank bottom.

The control subsystem actuates the solar system fan, circulator , dampers and the back-up system in response to various sensor and relay states. Upon demand for heat, the first electrical signal from a two-stage thermostat activates the solar system and the fan in the back-up system ; a second signal activates the back-up heating system if the solar system is unable to meet the demand. A snap acting thermostat in the solar collector controls the state of a demand relay, so on a sunny day inside air is routed through the solar panel for direct heating. When there is no demand for space heating a differential thermostat, using collector and storage temperature sensors, activates the storage mode of the solar system automatically. An immersion thermostat in the storage subsystem controls a demand relay so air can be routed through the heat exchanger in the heat extraction mode.



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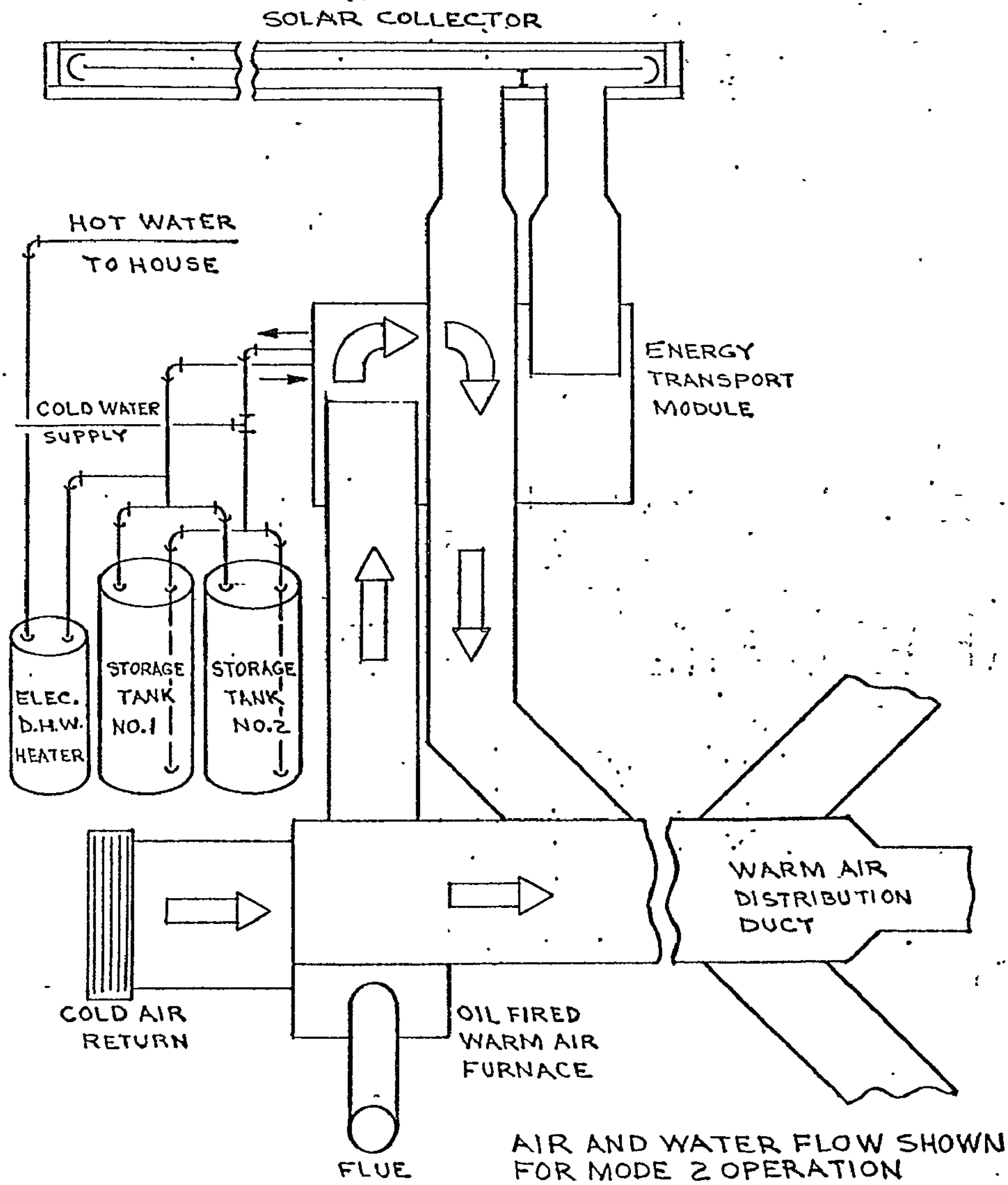
BUZZARDS BAY, MASSACHUSETTS
U.S.A.

DRAWN
APPD
DATE

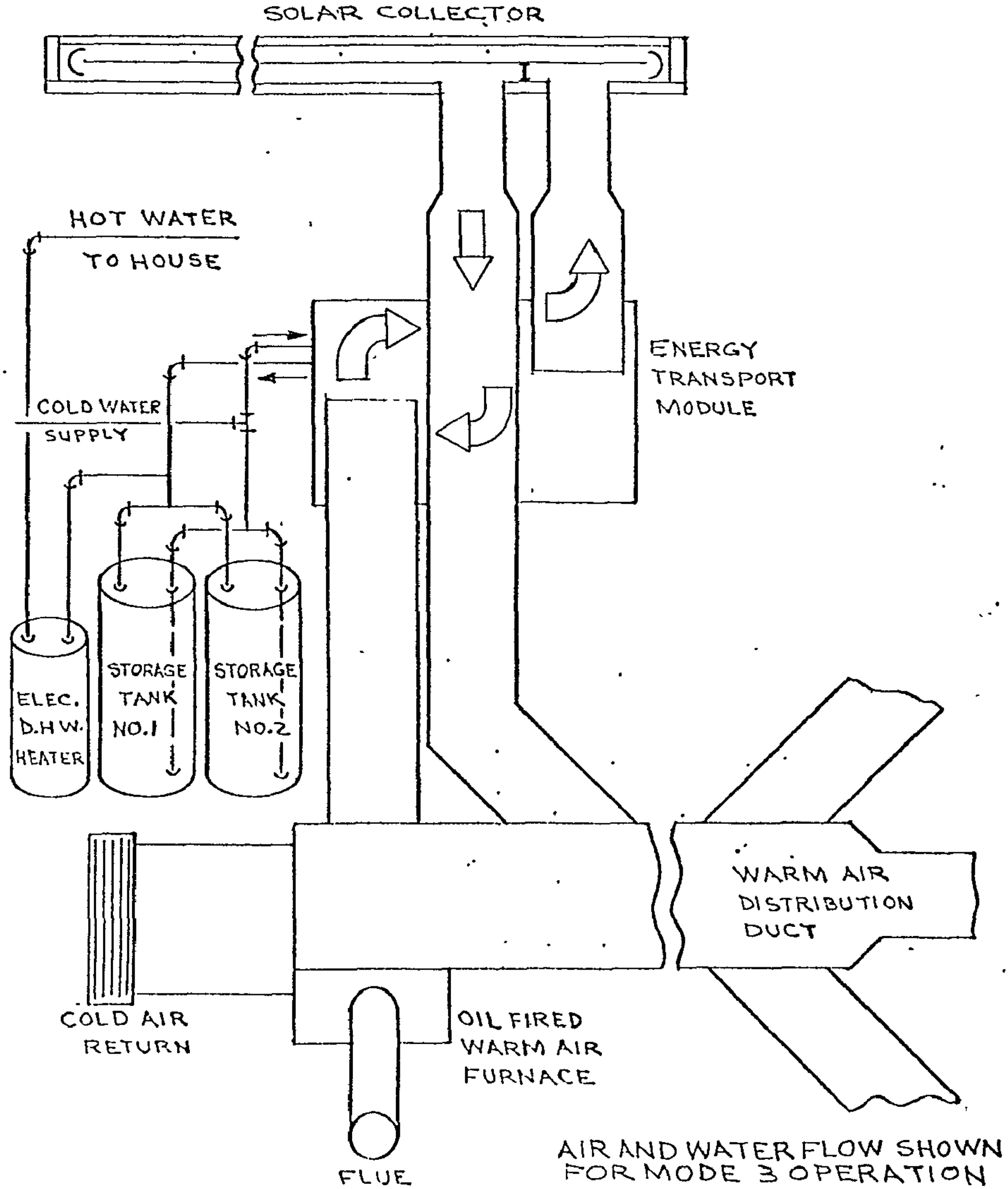
SYSTEM CONFIGURATION

DWG NO. FIGURE 1A
REV.



AIR AND WATER FLOW SHOWN
FOR MODE 2 OPERATION

FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	DRAWN APP'D DATE	SYSTEM CONFIGURATION DWG. NO. FIGURE 1B REV.



AIR AND WATER FLOW SHOWN FOR MODE 3 OPERATION

FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	DRAWN APPD DATE	SYSTEM CONFIGURATION DWG NO. FIGURE 1C	REV.

CONTRACT: NASA 8-32246 DOC. NO. 001-7

SUBJECT: SYSTEM CONFIGURATION RATIONALE JOB NO. 198

AUTHOR: P. Levine Date: December 1, 1976

The system arrangement has been developed to eliminate a long list of potential deficiencies associated with the alternative approaches depicted in Figure 1. Among them are the following:

1) Potential obstruction of back-up system ducting.
2) Non-participating elements in the air flow paths in the different modes - introduce objectionable pressure drops.
3) Operation of solar blower in parallel with furnace blower can result in load pulsation and related problems.
4) Operation upstream of furnace results in thermal losses thru furnace heat exchanger, flue and housing walls.

In addition system criteria considered included:

1) Locate blower on cold side of heat exchanger and collector in all pertinent modes for coolest operation.
2) Pre-package as much of system possible for economics of installation, marketing, maintenance and performance.

It is therefore desirable to place the solar interface with duct work downstream of furnace, operating furnace blower in distribution modes, and using the 4 damper reversing heat exchanger configuration which has been selected. This arrangement is selected and sized with careful treatment of the total economic impact of its cost and performance, although no economic presentation is made here.

The energy transport module is the "pre-packaged" assembly housing solar blower, heat exchanger, reversible water pump, 4 dampers, their 2 motors, and the control system. A complete description of energy transport module is included.

The energy storage system is a part of the domestic hot water system. Domestic water is used in the storage system, eliminating need for a second heat exchanger to pre heat DHW. The system is arranged to circulate cold water from bottom of tank to heat exchanger in storage mode and to circulate hot water from top of tank in recovering heat. This is accomplished by reversing pump flow.

Heat exchanger air flow reversal is coincidental.

ALTERNATE DUCTING ARRANGEMENTS

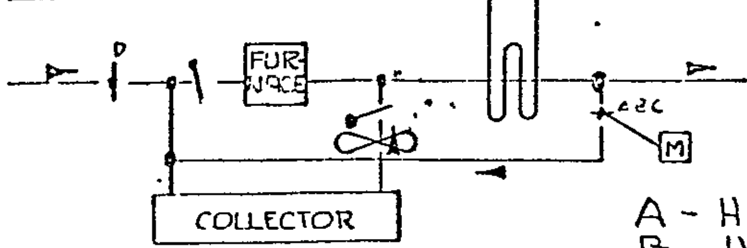
LEGEND

MOTORIZED DAMPER
BACKDRAFT DAMPER
WATER HEAT'G COILS
FAN

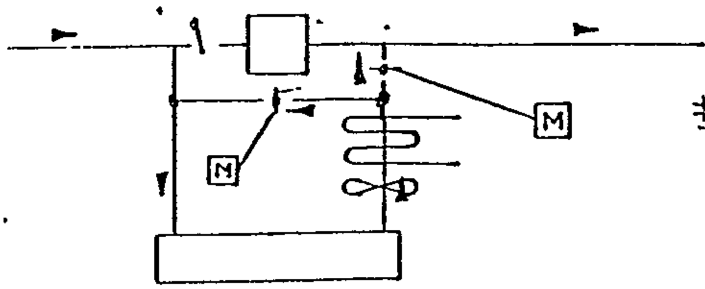
MODES

A - HEAT HOUSE FROM COLLECTOR
B HEAT HOUSE FROM STORAGE
C HEAT HOUSE FROM FURNACE
D STORE COLLECTED HEAT

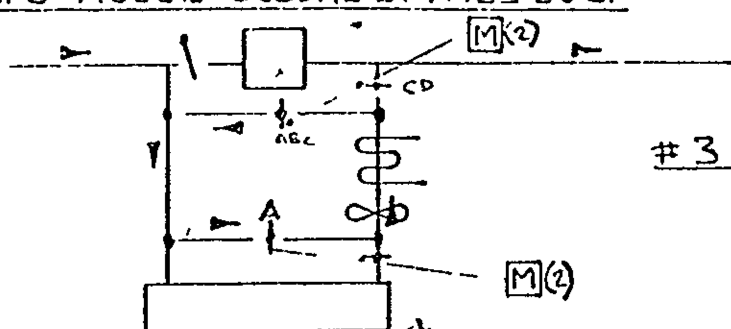
#1 BASE LINE



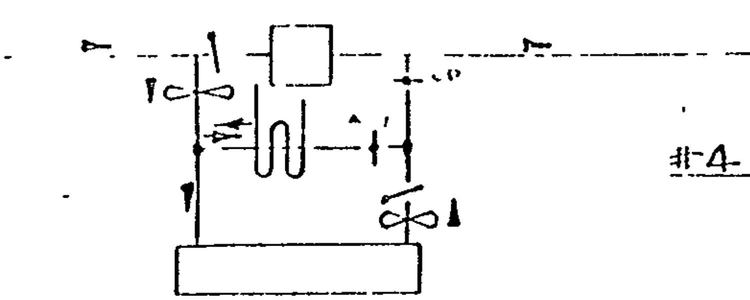
#2-RELOCATED WATER COILS OUT OF MN DUCT



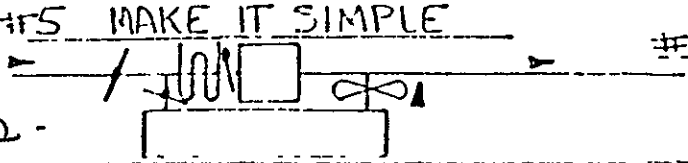
#3 ADDED SECOND BY-PASS DUCT



#4 REDUCE DUCT & DAMPER REQUIREMENT BY ADDING ANOTHER FAN



#5 MAKE IT SIMPLE



REMARKS

#2 AIR DRAWN OVER COLLECTOR IN MODE B NOT ACCEPTABLE

#3 A WORKABLE SYSTEM WITH 4 MOTORIZED DAMPERS

#4 A WORKABLE SYSTEM WITH 2 FANS

#5 A WORKAPLE SYSTEM

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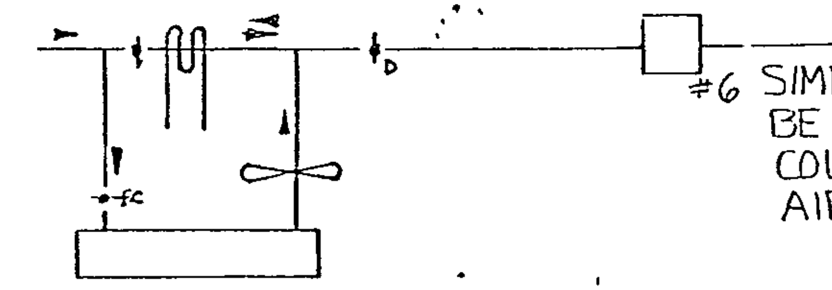
SOLAR HEAT DUCTING SYSTEMS - PARALLEL WITH FURNACE

DWG NO. FIGURE1
REV.

ALTERNATE DUCTING ARRANGEMENTS

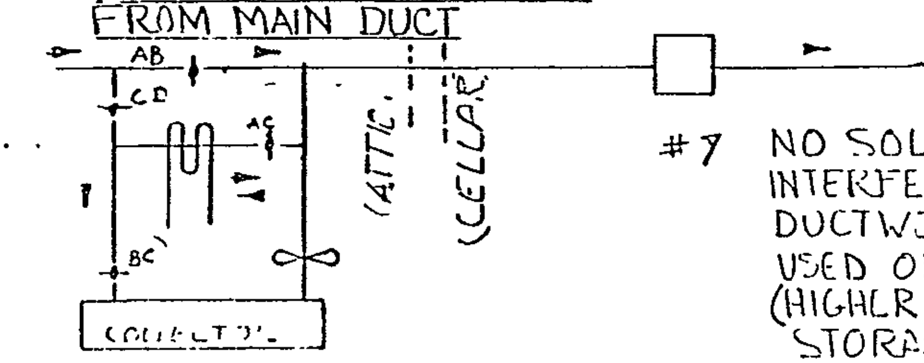
#6 LOCATED SOLAR HEATING SUBSYSTEM UPSTREAM OF FURNACE



REMARKS

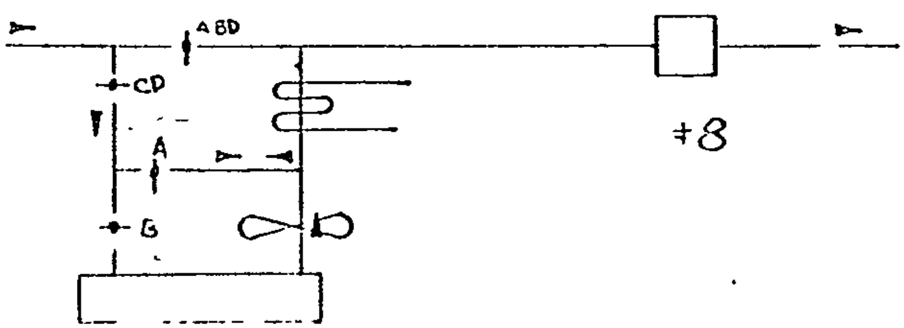
#6 SIMPLE SUBSYSTEM CAN BE INSTALLED IN ATTIC NEAR COLD AIR RETURN (STRATIFIED AIR PICK UP)

#7 REMOVED HTG COILS FROM MAIN DUCT



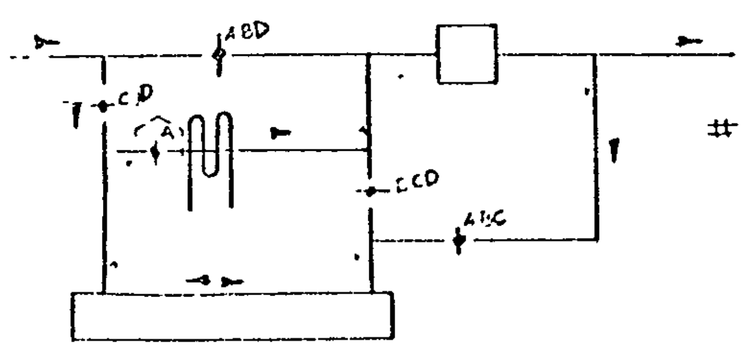
#7 NO SOLAR COMPONENTS INTERFERING WITH MAIN DUCTWORK. FAN IS USED ONLY TO STORE HEAT (MODE D) (HIGHER TEMP DESIRED FOR STORAGE CAN BE OBTAINED BY SMALLER FAN - DITTO FOR #6)

#8 PLACED HEAT EXCH OUT OF COLLECTOR BYPASS



#8

#9 USE FURNACE FAN ALONE



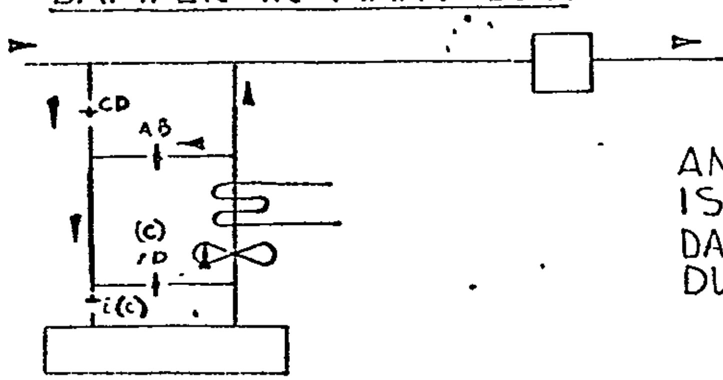
#9 MODE D CIRCULATES SOLAR HEAT THRU FURNACE PROBABLY LOSING IT TO HOUSE

FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	DRAWN APPD DATE	SOLAR HEAT DUCTING SYSTEMS - UPSTREAM OF FURNACE DWG NO. REV.

ALTERNATE DUCTING ARRGTS

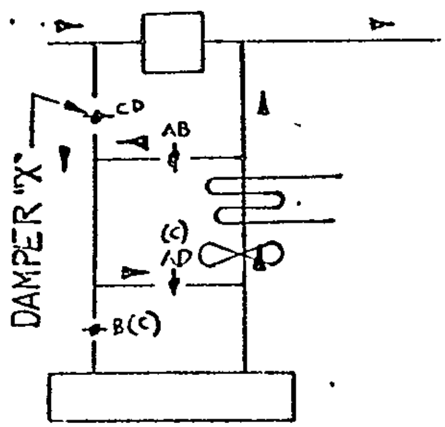
#10 UP/DOWNSTREAM OF FURN. W/ NO DAMPER IN MAIN DUCT

REMARKS



ANY DAMPER IN MAIN DUCT IS APPROX 20" SQUARE WHEREAS DAMPER IN SOLAR SUBSYSTEM DUCT IS 8" ROUND.

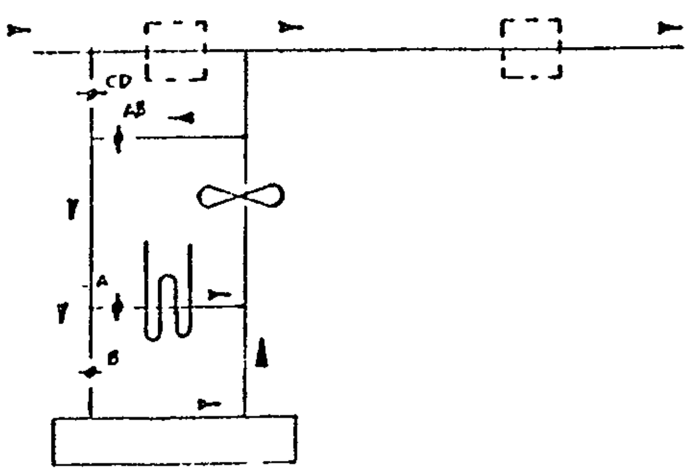
11 PARALLEL WITH FURNACE W/ NO DAMPER IN MAIN DUCT



IF FURNACE FAN FAILS IN MODE A & B OR IF DAMPER "X" FAILS IN MODE D THEN FILTERS UNLOAD INTO COLLECTOR/HEATING COILS. FILTER CAN BE RELOCATED TO RESOLVE THIS PROBLEM.

SEE AMCA PUB. 201 P42 - PARALLEL FAN INSTABILITY

#12 PARALLEL CONNECTION OF COLLECTOR & HEAT'G COILS.



THIS SYSTEM IS INEFFICIENT IN MODE D AS COOLED & WARMED AIR ARE MIXED

FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	DRAWN APVD DATE	SOLAR HEAT DUCTING SYSTEMS - WITH NO DAMPER IN MAIN DUCT DWG NO. REV.

ORIG 1

CONTRACT:	NASA 8-32246	DOCUMENT NO. 1004
SUBJECT:	System Performance Analysis	JOB No. 198
AUTHOR:	P. Levine	DATE: December 29, 1976

1. Scope

This document is intended to partially fulfill the data requirements for the System Specification. A yearly performance study was done for a single residence near Scranton, PA., an anticipated installation site.

2. System Description

The System Functional Description is given in Document No. 1001, System Functional Description. Alternate systems were configured as follows:

System	Collector Aperture	Water Storage	Air Flow	Reflector Enhancement Factor
1	172 ft^2	2000 lbs	400 CFM	1
2	172	2000	400	Variable, 1.6 max.
3	344	2000	800	1
4	344	4000	800	1

The following system insulation parameters were considered :

	Surface Area	R, $(Btu/ft^2\text{-hr }^oF)^{-1}$
Water Storage	.05 ft^2/lbm.	30
Ducting	.375 ft^2/CFM	13.5
Collector Edge	.24 ft^2/ft^2 Col.	9

The water storage and ducting were assumed to be in an unheated space with a temperature of 50 ^oF and no benefit of heat loss to the space was taken.

3. Residence Description

The residence features assumed are ,

Floor Area of 1080 ft^2

Heat Loss Factor 10 Btu/ft^2 -DD

4. Weather Data

The Ashrae Handbook and Product Directory, 1976 Systems, Chapter 43

provides the following degree day information, for Scranton, PA.

August Winter Temperature 37.2 °F
Yearly ToT. 6254 DD
Monthly

J	1156
F	1028
M	893
A	498
M	195
J	33
J	0
A	19
S	132
O	434
N	762
D	1104

The site latitude is nominally 41.5 DEG N.L.

Snow data indicates a Jan. mean of 10.8 inches and a seasonal mean of 51.8 inches (U.S. G.P.O. weather normals, means and extremes). The average wind speed in Jan. is 7.3 mph. The fastest mile is 78 mph. The sunshine data available is, based on annual mean number of days;

clear	68	cloudy	178
partly cloudy	119		

5. <u>Collector Performance</u>

The collector performance is as stated in Doc. No. 1002. The rear-face insulation losses are accounted for in the collector performance calculations. The collector edge losses are accounted for as noted in paragraph 2 above.

6. <u>Calculation Model</u>

Clear day calculations were performed, accounting for the earth's declination and reflections induced due to the angle between the sun-line and normal to the collector. The reflection calculations were used to modify (reduce) the insolation values given in NBS TN 899. The calculation was divided into hourly intervals. If the solar energy available was less than the demand, it was used for direct space heating ; if the solar energy available was larger than the demand,then the required demand time for direct heating was subtracted and the remainder of the time, the system heated the storage. The return air temperature to the collector was 65 $^{\circ}$F ; the initial water storage temperature was 90 $^{\circ}$F.

7. Calculated Results

Total Daily Solar Heat/Total Daily Demand

System	S	O	N	D	J	F	M	A	M
1	2.27	.93	.54	.37	.38	.40	.48	.76	1.38
2	2.90	1.37	.80	.61	.57	.60	.64	.93	1.38
3	3.86	1.55	.90	.64	.65	.68	.79	1.25	2.34
4	4.50	1.77	.97	.67	.68	.72	.85	1.39	2.67

Based on 50% sun-factor

System	Annual Percentage of Heat Demand
1	28
2	38
3	45
4	47

8. Conclusions

A reflector offers a potential for markedly boosting solar heat output without increasing the amount of basic solar system hardware. The weather pattern indicates many cloudy days, and the effectiveness of the reflector cannot be precisely defined. Increasing the system by 50% would yield a minimum annual percentage of 42% ; with reflectors the percentage could be as high as 57%. The collector area to house area is 0.24. Comparison with other predictions (Viz. " Pacific Regional Solar Heating Handbook", LA-6242-MS, Seattle, Vancouver) indicates good agreement for the basic operation mode.

The advantage of a reflector, that appear are :

a) Increased solar intensity results in higher air temperatures minimizing the storage size.

b) The annual solar energy production curve more closely matches the demand curve.

The exact setting of the collector and reflector angles involve consideration of the extent of cloudy weather. Tentative collector orientation is latitude + 30 degrees and the reflector at -10 degrees ; however, further study is planned to verify this choice.

CONTRACT: NASA 8-32246 DOCUMENT NO. 1006

SUBJECT: System Specification Update JOB NO. 198

AUTHOR: P. Levine DATE: December 30, 1976

1. Scope

This document presents a System Specification Update as required by Data Requirement 504-4

2. System Specification Update

Site: The site location is near Scranton, PA. with a yearly load of 6254 DD.

Heating Capacity: The system without a reflector will provide solar energy for 40% of the average total heating load based on the average monthly and yearly degree days given in ASHRAE 1976 Systems Handbook, Pg. 435, a floor area of 1080 square feet and heat loss factor of 10 Btu per sq. ft. per DD. With a reflector, the system will provide up to 60% of the average total heating load.

Auxiliary Energy: The Auxiliary Energy will be between 40% and 60% of the average total load depending on the performance of the reflector.

Physical Data: Collector: 260 Sq. Ft. aperture area

Two rows of panels	
Row 1	4-ft x 40 ft
Row 2	4-ft x 32 ft

Physical Data: cont'd

Reflector: One per row of panel 8-ft wide same length as panel.

Storage: 2000 lbs of water

General: The system is a pressurized water storage system as proposed. The collector size has been increased to allow for the more severe climate of the site and to accomodate the losses. It is assumed that duct and storage losses do not contribute to the useful solar contribution. The reflector is constructed in the field of standard building materials and sized using the results of McDaniels. et al " Enhanced Solar Energy Collector", Solar Energy, Vol. 17 , pp. 277-283.

The system performance calculations are given in Doc. 1004 which is enclosed.

The operating requirements design guideline has been to limit their energy consumption to less than 5% of the delivered solar energy.

Document No: 1003-2

CONTRACT : NASA 8 - 32246 Job No. 198

SUBJECT : Functional Description of Control Subsystem

AUTHOR : R. Meyers Date : December 2, 1976

1. Scope

This document is intended to partially fulfill the verification requirement 1.7 " Control " .

2. Control Subsystem Description

Wiring Diagram Fig. 1 depicts the logic which controls the heating system. The heating system always employs the back-up furnace blower to distribute heated air. The control system, except for probes, is housed in an electrical enclosure mounted in energy transport module.

The system operating modes are commanded by (4) thermostats. The room thermostat detects building temperature and calls upon the solar heating system to deliver heat, activating the thermostat T1 (See Fig' l.). If the collector is warm T1 will be closed and will engage relays K2 and K3 (See Fig. l), commanding the appropriate dampers, the solar blower and the furnace blower to deliver heat from the collector, (mode 1). If, however, T1 is (cold) open power from K1-6 activates the storage thermostat T3. If storage is warm, (T3 closed) K4 and K5 are engaged providing power to recover heat from storage, (Mode 2). If T3 is open, then the solar system will not deliver heat to building. A second stage of the above room thermostat will call for back up heat whenever heat provided (or not Provided) by solar system is inadequate to maintain building temperature.

When second stage closes, solar heating is locked out.

Power to thermostat T2 is interrupted only when the building is being heated from collector or storage. At other times, as long as collector temperature is sufficiently higher than storage by a pre-set increment T2 will be closed, relays K6 and K7 will be engaged, operating the storage mode (# 3). Thus it is possible to collect and store solar energy while furnace is operating.

System control is developed to be failsafe. No control elements are placed in the main duct where they could inhibit back up system performance. Dampers are arranged to prevent heat consuming flow loops in their de-energized modes, and are coupled in pairs, in such a way that one blower inlet damper and one discharge damper must always be open. Spring returns will be used so that de-energized dampers will run back freely to de-energized positions. An electrical interlock will prevent blower operation while D2 and D3 are both open, as they are when system is de-energized. This prevents blower motor overload in all damper failure modes, and provides a time delay between operation in the various modes.

The differential thermostat produces an on-off (not proportional) signal, and has the added feature of shutting off under extreme temperature conditions. When storage temperature reaches 160° or when collector temperature is below 80°, then the differential thermostat will not operate storage mode.

Solar System cont'd

The collector thermostat will operate at temperatures yet to be determined by more accurate system modeling. It will be of the snap-acting type, with a broad temperature differential. It will operate nominally at around 80°. The storage thermostat will operate at around 90° having a close temperature differential. The above figures are not fully developed, but are based on total operating cost/output/stability considerations.

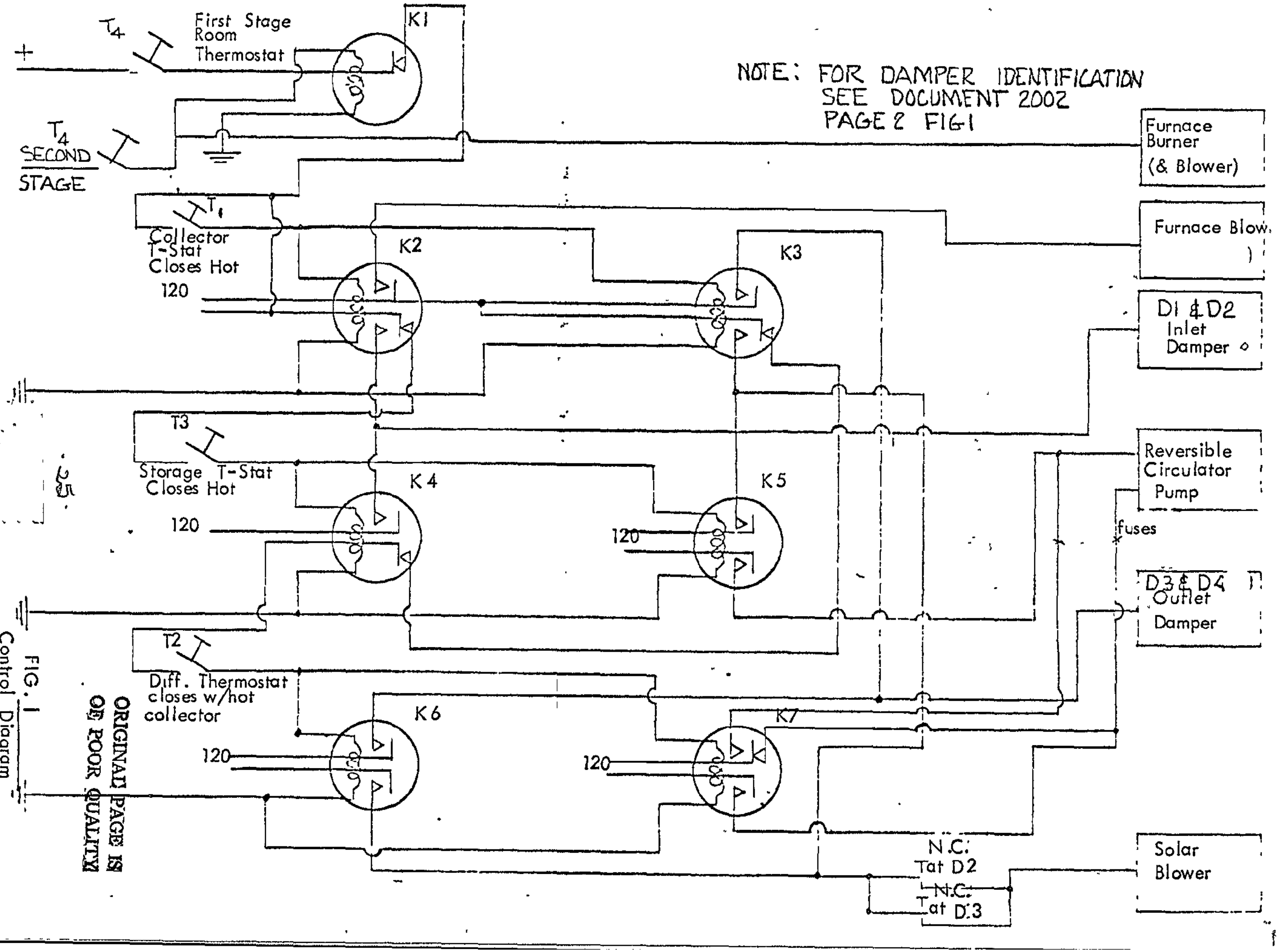


FIG. 1
Control Diagram

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Fig 2

Solar System Control Modes

Mode	Fans BF	Fans SF	Dampers D1	Dampers D2	Dampers D3	Dampers D4	Circulator CF	Circulator CR	Thermostats T1	Thermostats T2	Thermostats T3
1. Direct Solar Heating	On	On	O	C	C	O	Off	Off	C	N/A	N/A
2. Extract Stored Solar Heat	On	On	O	C	O	C	On	Off	O	N/A	C
3. Store Solar Heat	Off	On	C	O	C	O	Off	On	N/A	C	N/A

BF - Backup System Fan

SF - Solar System Fan

D1 - Admits Air Modes 1 & 2

D2 - Admits Air Mode 3

D3 - Air Discharge Mode 2

D4 - Air Discharge Mode 1 & 3

CF - Circulator For Heat Extraction Forward Power

CR - Circulator For Heat Storage - Reverse Power

T1 - Collector Thermostat

T2 - Collector/Storage Differential Thermostat

T3 - Storage Thermostat

O - Open State

C - Closed State

N/A - State Does Not Effect Mode

Document No. : 2002-2

CONTRACT: NASA 8-32246

Job No. : 198

SUBJECT: Mechanical Systems and Components Design Conditions

Date: December 1, 1976

AUTHOR: R. Meyer

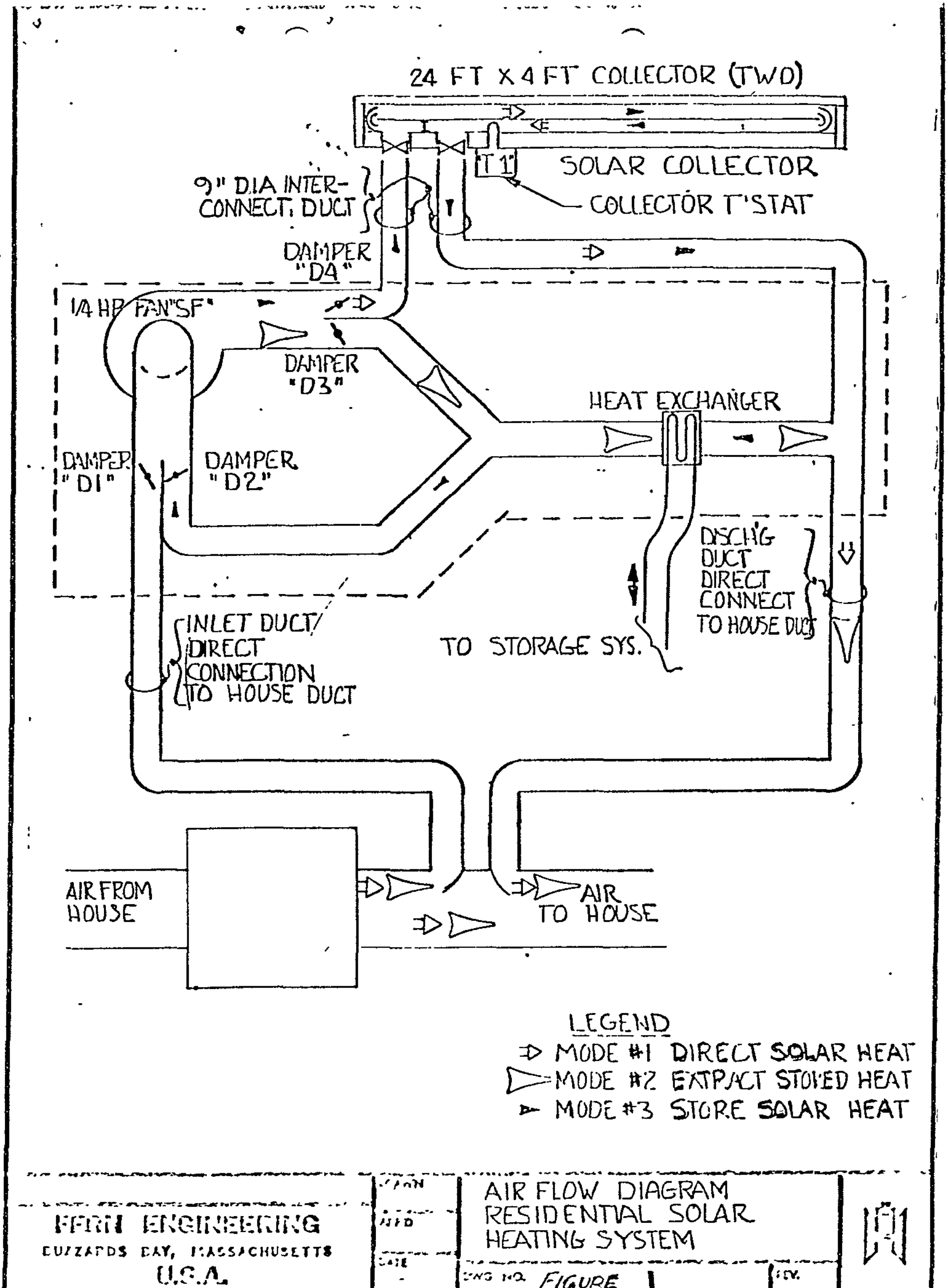
1. Scope

This document is intended to partially fulfill the verification requirement 2.1 "System Design Conditions". The mechanical design requirements and performance capabilities for the ducting, heat exchanger, pumps and fans are presented.

2. Ducting Design

The prepackaged solar energy transport module plays a major role in the ducting design as it houses the blower, 4 dampers, heat exchanger, and interfaces with the solar panels, storage and back-up furnace subsystems. A schematic diagram of the transport module functions is given in Figure 1. The housing for the transport module is designed to be floor mounted adjacent to the back-up oil-furnace and is constructed of 22 gauge galvenized steel. The remainder of the ducting will have thickness not less than 28 gauge. The pressure drop through the duct are based on 400 CFM standard air flow. The pressure drop through the energy transport module in the worst case, pressure drop made of storing energy is given below,

	ΔP, IN.W.G.
1) Heat Exchanger - using aerofin bulletin MP-72 Page 10 Table 5, 2 Row Heat Exchanger with 300 FPM face velocity; .074 in.w.g. ea./2 Req'd Back-To -Back	.148


24 FT X 4 FT COLLECTOR (TWO)
SOLAR COLLECTOR
T 1
COLLECTOR T'STAT
9" DIA INTER-CONNECT. DUCT
DAMPER "D4"
1/4 HP FAN "SF"
DAMPER "D3"
HEAT EXCHANGER
DAMPER "D1"
DAMPER "D2"
DISCH'G DUCT DIRECT CONNECT TO HOUSE DUCT
INLET DUCT/ DIRECT CONNECTION TO HOUSE DUCT
TO STORAGE SYS.
AIR FROM HOUSE
AIR TO HOUSE
LEGEND
MODE #1 DIRECT SOLAR HEAT
MODE #2 EXTRACT STORED HEAT
MODE #3 STORE SOLAR HEAT
FERN ENGINEERING
BUZZARDS BAY, MASSACHUSETTS
U.S.A.
AIR FLOW DIAGRAM RESIDENTIAL SOLAR HEATING SYSTEM
DWG NO. FIGURE 1

2) Inlet from Collector Duct , 0.3q .015

q = .051 in.wa, 9" Dia Duct and Transition Section

3) Damper Loss (2)

Based on sharp, edge orifice loss 1.2q

plus structure loss .07q .020

4) Internal Duct Losses

90° Miter 1.38q , q = .005 .018

90° Miter 1.38q , q = .008

5) Fan Inlet Loss .005

Fan performance based on AMCA tests with free air inlet and exit duct ; installed with inlet diffuser benefits rated capability . Adjustment is made to fan performance.

blower inlet q = .082 , q diffuser inlet = .005

Reduction is loss is .27 Δq

6) Fan Outlet Loss -.013

AMCA rating on straight duct outlet ;

Installation uses a baffled diffuser. The improved condition is pertinent to blower performance.

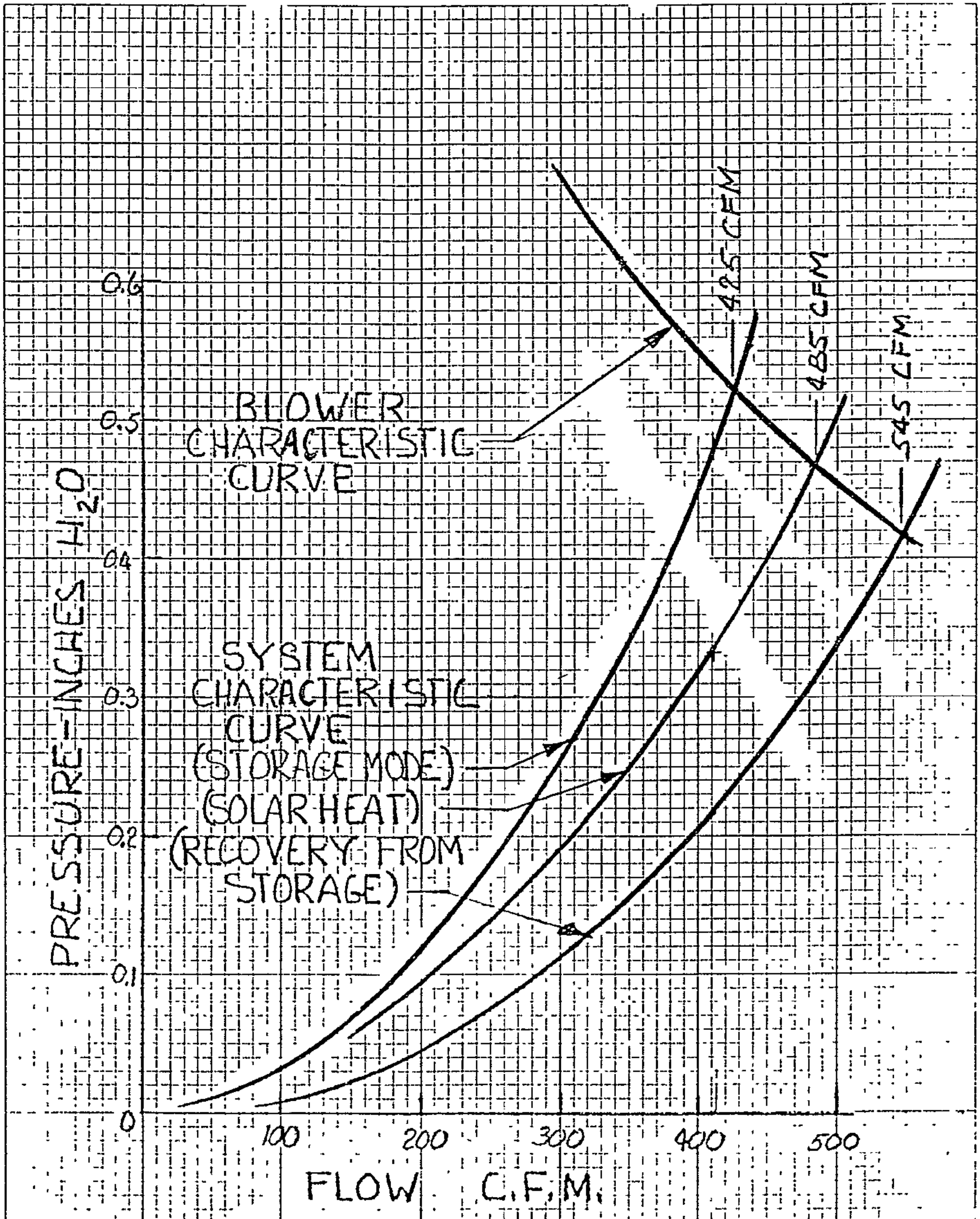
ΔP , Energy Transport Module Subtotal	.219 in. wg
The solar collector loss (Document 1002)	.170
Duct loss, 50 ft, 9" Dia	.050
8 Long radius elbows	.041
Total ΔP	.480 in wg.

3. Blower Selection

Lau Blower Model HV 7-4 has been selected for use in the solar air handler. A Dayton 1/4 hp 100 rpm permanent split capacitor motor Granger Stock No. 3M-339 selected to drive this blower, to yield maximum electrical efficiency. Improved performance of blower is obtained by use of inlet/outlet diffusers. At the inlet, a perfect diffuser raises blower pressure by .34 q and at the outlet .81q. The result is that a blower with inlet and outlet diffuser: can develop pressures of S.P. + 1.15q (Diffuser inefficiencies are accounted above.) Flow is plotted on Fig. 2 on following page.

4. Duct Heat Losses

In order to properly evaluate heat losses, it is necessary to review each mode separately. In the direct solar heating mode and in recovering heat from storage operating temperatures are low. In the mode of storing energy the air temperatures are highest and part of loss may be providing unwanted heat to house. It is helpful to review the path of air flow in this mode. The heated air flows from collector thru interconnecting ductwork to the center-top of the air handler (a nominal run of 25'). Thence flow is thru heat exchanger. Under worse case conditions the water is hot, so air temperature may drop as little as 10°. Cooled air then flows thru blower and out of energy tramsport module and back to the collector. The 9" duct has a 4" fiberglass jacket (50ft. x n x 13"/12"/ft.)/ 13.5 = 12.6 Btu/hr °F. The energy transport module air flow in this mode, is in direct contact with 31.7 square ft., having 1 1/2" fiber-glass jacket 31.7 ft.^{2}/ 5.8 = 6.2Btu/hr °F. The remaining 18 sq.ft. of surface is further insolated by a dead air space: 18/7.8 = 2.3Btu/hr °F.



SYSTEM OPERATING CURVE

FIG 2

The total loss is 21.1 Btu/hr. °F or with 110° differential 2321 Btu/hr, or 9.6 % 9.6% of collected energy. This figure represents a peak performance condition.

5. Liquid Energy Transport and Storage System

The heat exchanger is to be provided with flow of 2.9 gpm by a bronze rotary gear pump (Granger Stock No. 1P782) driven by 1/4 hp 1100 rpm permanent split capacitor motor (Granger Stock No. 3M 339) (same model is used for blower). The motor pump combination is devised to be reversible for the dual purpose of reversing pick-up/delivery to storage, and reversing flow thru heat exchanger. The pump works against a head of 8.6 ft. in heat exchanger and (ashrae P494 Fig. 1) 0.9" per foot of 3/4 O.D. Cwt. using a length, including piping into storage tanks, of 30 ft then pressure drop is 30 x 9/12 $\frac{in}{ft}$ = 2.3 ft, and total circulator head is 2.3 + 8.6 = 10.9ft. (.007hp) of water.

The two 120 gallon Sepco water storage tanks are arranged in parallel such that water can be extracted from bottom or top to take advantage of stratification. This is accomplished by reversing pump direction. The two 120 gallon Sepco storage tanks are porvided with 2" insulation. Although the tanks are well insulated and combined, will lose only 0.56% of stored energy per hour, this energy is equal to 5 % of a typical hour's collected energy. Although this 5% is associated with peak storage (temp. rise of 110°) it represents a system inefficiency which can be overcome. Therefore, a 4" fiberglass jacket will be added to storage tanks. With the jacket, loss is reduced to less than 2% of an hours collected energy; losses, therefore, do not have a substantial impact on system performance.

System 3/4" OD tubing is covered with 3/4 thk. Johns-Manville fiberglass "Flame-Safe" insulation, which at 160° will lose 1.6% of heat being collected, passing thru 30ft. of pipe. Again as with ducts, for lower temperatures the loss is proportionately lower.

6. Heat Exchanger Design

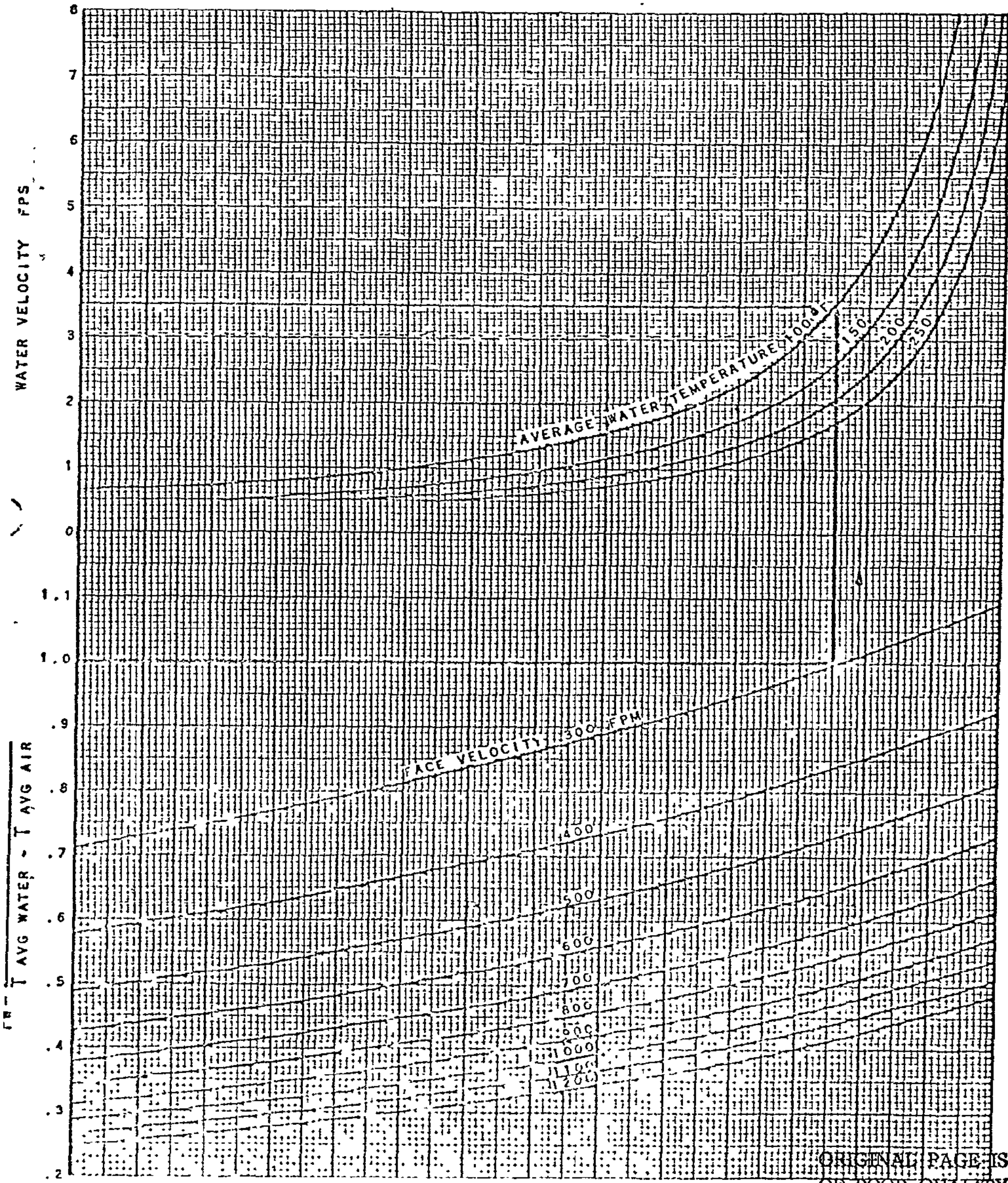
The heat exchanger is arranged to provide a counterflow arrangement. The circulator pump reverses in different modes to take advantage of the stratification in the water tank, and the air flow over the heat exchanger reverses, maintaining counterflow conditions. The heat exchanger is sized to provide a low air pressure drop, critical to system performance. The long, narrow configuration is selected to establish uniform air flow characteristics over the surface of the heat exchanger. The heat exchanger thermal effectiveness, which is crucial to the good performance of the collector, developed by the use of 2 back-to-back 2 row heat exchangers. The above conditions result in a 82.9% effective heat exchanger with air pressure drop of .15" and a flow rate under 3 gpm. The selected heat exchanger uses 5/8 OD copper tubing with helical aluminum fins.

Performance calculations follow using 2.9 gpm of water in 2 2 pass aerofin type MP, 6 tube face x 1'-9" N.T.L: Heat exchangers arranged for counterflow with 400 CFM standard air flow. Water velocity in single circuit 5/8 OD tube is 3.5 f/s. From table, performance curve gives heat exchanger characteristic F_W = 1.0 (For water temperature of 100 °F.)

$$F_W = \frac{\text{T leaving air} - \text{T entering air}}{\text{T average water} - \text{T average air}} = 1.0$$

Since nearly constant air flow and water flow pass over the heat exchanger, a constant performance characteristic results. The effectiveness "E" is directly related to the tabulated characteristic "F_W" but consideration must also be made for the counterflow interaction of the heat exchangers.

PERFORMANCE RATINGS
TYPE MP 2 ROW



AEROFIN BUL.MP-72

25

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The heat exchanger effectiveness is

$$E = \frac{T_{air_{in}} - T_{air_{out}}}{T_{air_{in}} - T_{water\ in}}$$

Typical conditions are

$$T_{air_{in}} = 160$$

$$T_{air_{out}} = 110$$

$$T_{water_{in}} = 100$$

$$T_{water_{out}} = 114$$

$$E = 0.83$$

DOC. NO. 1002-2

CONTRACT: NASA 8-32246

JOB NO. 198

SUBJECT: COLLECTOR PERFORMANCE

AUTHOR: P. Levine

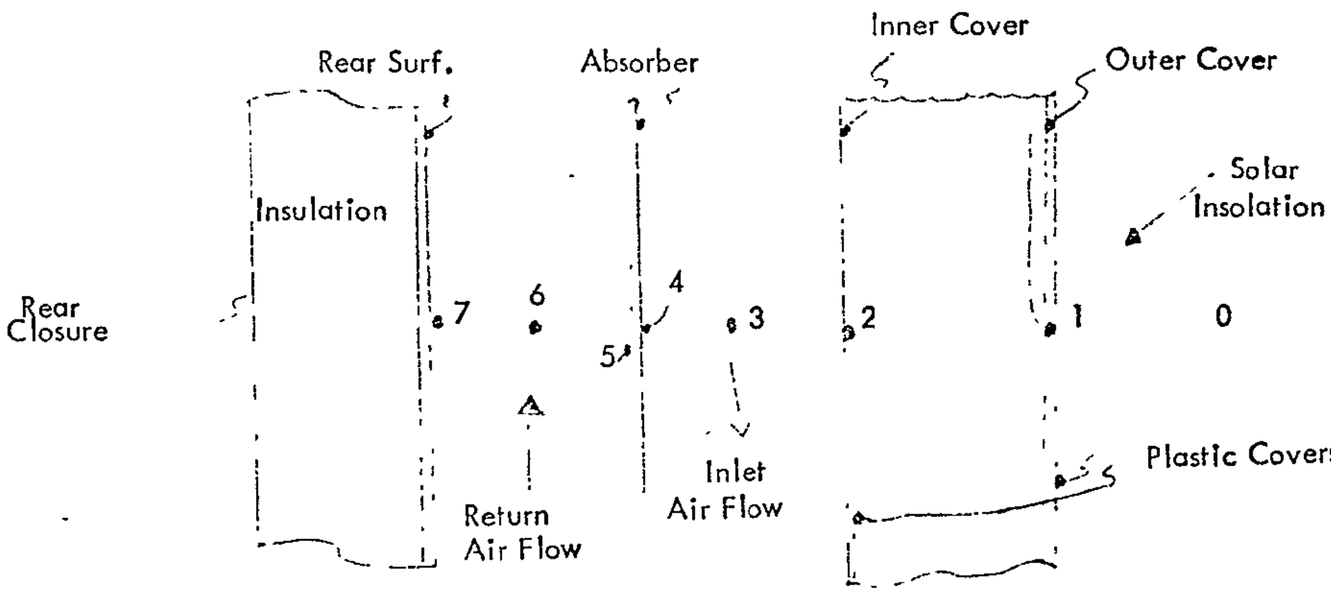
DATE: November 29, 1976

1. Scope

This document is intended to partially fulfill the "Verification Requirement No. 1.3 Collector Performance". A simplified preliminary collector performance calculation model is described and calculated performance results are presented. An improved collector performance calculation is also described and calculated performance results are presented.

2. Preliminary Calculation Model

Collector Station Numbers



Front Losses

The convective and radiation losses from the sun side of the collector are the predominant losses.

Hottel and Woertz[1] proposed a front loss formulation, later modified by Tabor[2],

$$Q_2 = \frac{(T_2 - T_o)}{\dfrac{1}{C_c \sqrt[4]{\dfrac{T - T_o}{1 + f_w}}} + \dfrac{1}{h_n}} + \frac{\sigma(\bar{T}_2^{\,4} - \bar{T}_o^{\,4})}{\dfrac{1}{E_c} + \dfrac{1 + f_w}{E_c} - 1} \qquad 1$$

The loss formula above neglects the effects of the cover transmittance and assumes the plastic covers are opaque to IR radiation. The terms in equation 1 are,

C_c = Free convection Loss coefficient

Tilt Angle w/Horiz.	C_c Btu/ft^2 - hr, $^oF^{5/4}$
0	.23
30	.20
60	.17
90	.16

The values of hw and f_w are[1]:

Wind Speed, MPH	hw, Btu/ft^2 - hr	f_w
0	1	0.76
10	4.07	0.36
20	7	0.24

The IR emissivity value used is 0. 88.

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Absorber Performance

The absorber is thin metal coated with a high absorptivity paint. Nextel 101-C10 velvet coating manufactured by 3 M is a candidate coating. The absorptivity is .98 and IR emittance is .89. based on manufacturers data given in Appendix 1, pg. 65.

The effects of incidence are based on the absorptivity curve given in Ref. 1 (Fig. 14).

Transmission Performance

The transmission data for the FRP covers are given in Appendix 11, page 69. The average solar transmittance is .875. The effects of weather, UV and aging on transmission properties are summarized in Appendix 11. The combined normal transmission absorption product for the two-cover glazing is 0.76. An alternate approach of an outer glass - cover and inner FRP was considered. The outer cover transmission data for this case is given in Appendix 111. The combined normal transmission absorption product is 0.78 for an outer glass cover, inner plastic cover and 3 M Nextel absorber.

Heat Balance

The heat balance equations used for the calculation of collector performance are summarized below :

$$Q_2 = \frac{(T_2 - T_o)}{\dfrac{1}{.2\sqrt[4]{\dfrac{T_2 - T_o}{1.36}}} + .25} + 2140\left[\overline{T}_2^{\,4} - \overline{T}_o^{\,4}\right] \qquad 1$$

$$Q_2 = A_c I + \left(\overline{T}_4^{\,4} - \overline{T}_2^{\,4}\right)2857 + h_2 \ (T_3 - T_2) \qquad 2$$

$$\frac{T_7 - T_o}{R7} = 3600 \frac{\left[\overline{T}_4^{\,4} - \overline{T}_7^{\,4}\right]}{\left[1/\varepsilon_7 + 1/\varepsilon_5 - 1\right]} - h_7 \ (T_7 - T_5) \qquad 3$$

$$QAir = h_2 \ (T2 - T3) + h4 \ (T4 - T3) + h5 \ (T5 - T6) + h7 \ (T7 - T6) \qquad 4$$

$$QAir = (A_2 + A_4) I - Q_2 - (T_7 - T)/R_7 \qquad 5$$

Forced Convection Heat Transfer

As the absorber forms only one side of the air flow duct, unsymmetrical heating occurs. The theory for unsymmetrical turbulent heating in rectangular ducts given in Ref. 5 was incorporated into the performance calculation method. The numerical results for Nusselt number and form factor were curve fit. The resulting expressions are :

$$Nu_6 = 27.8 * (Re_6/10^4)^{.72} \quad , \; Nu_3 = 27.8 \; (Re_3/10^4)^{.72}$$

$$B_6 = .22 * (10^4/Re_6)^{.15} \quad , \; B_3 = .22 \; (10^4/Re_3)^{.15}$$

D_3 , D_6 = Hydraulic Diameters

$$h_2 = \frac{Nu_3 \; K_3}{D_3} \left(1 + B_3 * \frac{(T_4 - T_3)}{(T_2 - T_3)} \right) / \left(1 - B_3^2 \right) \qquad 6$$

$$h_4 = \frac{Nu_4 \; K_4}{D_3} \left(1 + B_3 * \frac{(T_2 - T_3)}{(T_4 - T_3)} \right) / \left(1 - B_3^2 \right) \qquad 7$$

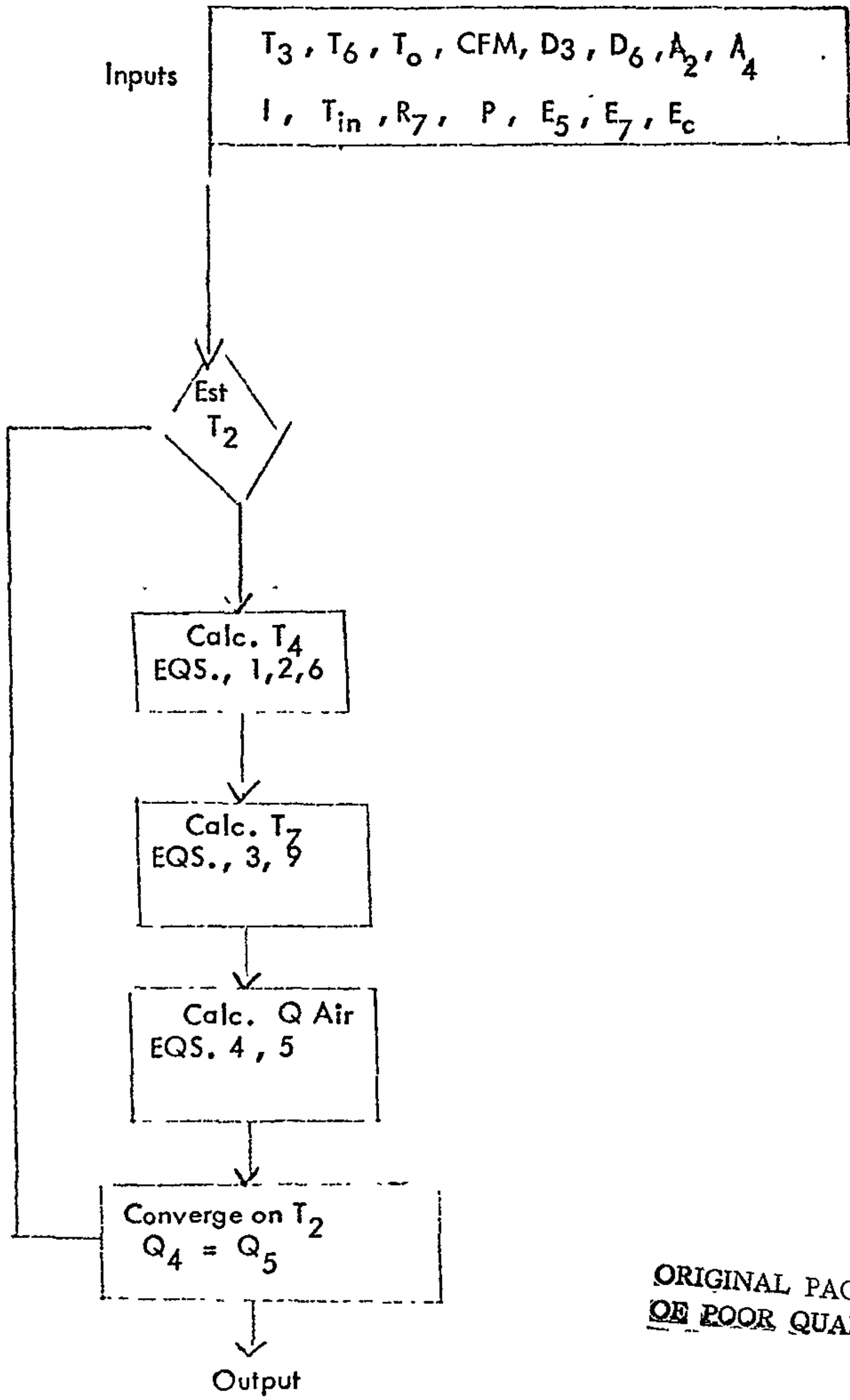
$$h_5 = \frac{Nu_6 \; K_6}{D_6} \left(1 + B_6 * \frac{(T_7 - T_6)}{(T_5 - T_6)} \right) / \left(1 - B_6^2 \right) \qquad 8$$

$$h_7 = \frac{Nu_6 \; K_6}{D_6} \left(1 + B_6 * \frac{(T_5 - T_6)}{(T_7 - T_6)} \right) / \left(1 - B_6^2 \right) \qquad 9$$

Method of Solution

The calculations are done at various stations along the length of the collector, and the resultant temperature rise of the air integrated along the flow path.

For each station the method of solution is as follows :



A computer analysis was designed capable to calculate collector performance for 3 - types of collectors : 1) A single pass on the front side of the absorber, 2) A single pass on the rear side of the absorber and 3) A two pass collector.

3) Results

Comparative performance of the 3 types of collectors were obtained.

The conditions assumed were :

FRP Covers

$T_1 = 65^\circ F$	$R_7 = 10\ ^\circ F/Btu\ ft^2\ hr$
$T_o = 50^\circ F$	$E_5 = E_7 = .9$
$C = 200$ CFM	$A3 = A6 = .319\ ft^2$ (1" x 46" Duct)
$D_3 = D_6 = 1/6$ ft	$P = 2120$ PSF
$I = 300\ Btu/ft^2\ hr$,	$A_c = 88.32\ ft^2$ (24-ft Long)

The results for the 3 types of collectors are given in Figures 1a and 1b. Inspection of the results indicates the double pass to have the highest efficiency. The inner cover temperature is lowest for the double pass, so the front losses are lowest. The rear duct wall temperature for the double pass concept lies between the two single pass concepts ; hence further optimization appears possible by increasing the rear side thermal resistance. Comparing the two single pass alternatives, the front side pass has the lower inner cover temperatures and hence the lower front loss ; however, the higher rear side temperature (absorber) associated with the front pass concept results in larger rear side loss. Increasing the thermal resistance (R_7) to 20 increases the efficiency to 58%.

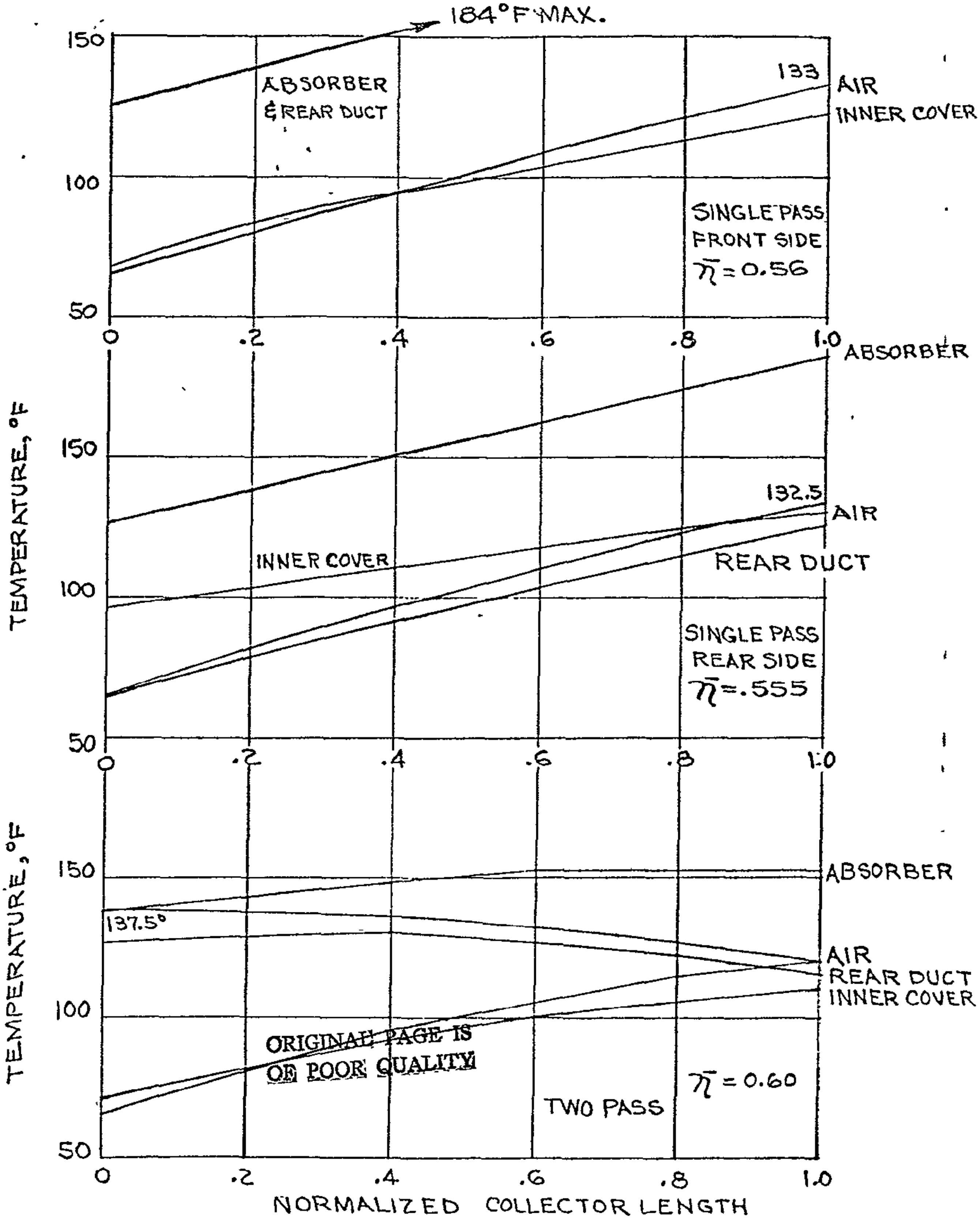


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FIGURE 1a COMPARISON OF ALTERNATIVE COLLECTOR DESIGNS AT I = 300, T_N = 65°F, T_o = 50°F, C = 200 CFM

	DRAWN	COLLECTOR COMPARISONS
FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	APP'D	
	DATE	DWG. NO. FIGURE 1 — REV.

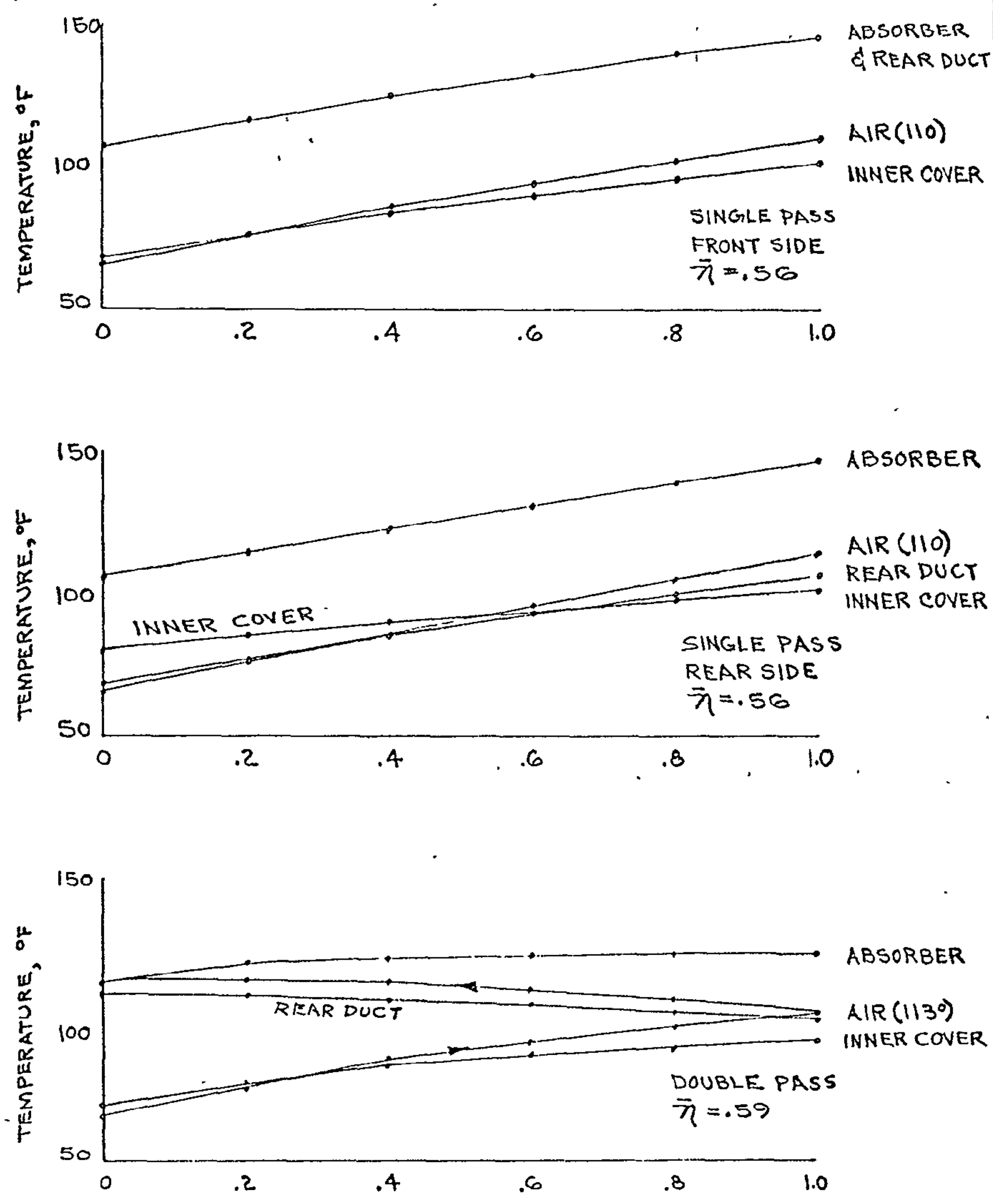

FIGURE 16: COMPARISON OF ALTERNATE COLLECTOR DESIGNS AT $I=200$, $T_1=65$, $T_0=50$, $C=200$, $D3=D6=2"$, $R_7=10$.

	DRAWN		
FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	APP'D		
	DATE	DWG NO.	REV.

Insulation and emissivity trade-offs were calculated for the two pass collector under the conditions given above yielding the following results:

E_5, E_7	R_7	n
.9	10	.60
.9	20	.61
.2	10	.61
.2	20	.62
.05	10	.61

Low emissivity surfaces in the rear duct are favorable. The effect of duct size of the efficiency of the two-pass collector was calculated yielding the following results:

$D_{3/2}$	$D_{6/2}$	n
.875"	.875"	.59
1.5	1"	.60
2"	1"	.61

The effect of Duct size on the efficiency appears to be small.

The effects of insolation and flow were calculated yielding the following results ($E_5 = E_7 =$ .05, $R_7 = 10$, $D_3 = D_6 = 2"$).

I	C	T_d	n
300	200	139	.61
200	200	113.5	.60
100	200	87	.55
300	250	127	.64
200	250	105	.62
100	250	84	.59

The results indicate that increases in the CFM reduces the temperature rise and increases the efficiency. The fan power consumption is of concern at the higher flow rates. The pressure drop through each collector is given by,

$$\Delta P = (F + f \frac{L}{D}) q_D$$

For 1-inch Duct Heights

$L/D = \underline{24} \times 6 = 144$

At 200 CFM, $Re = 10^4$, $f = .03$

Because of inlet and outlet losses and flow turning losses $F = 3$, so

$$\Delta P = 7.32 \; q_D$$

At 200 CFM

$$FP = \text{Fan Power} = \frac{\Delta P \times CFM}{N_F \; 33000}$$

The overall fan efficiency is estimated as 0.25. It must be remembered that the fan drive is electrical and the cost of this power is several times more than the fossil fuel costs. At 200 CFM,

$q = .13$ PSF , or .024" H_2o

$\Delta P = .95$ PSF , or .172" H_2o

FP = .046 , or 116 Btu/hr (2 collectors)

At a nominal delivery rate of 25,000 Btu/hr, the FP represents..46% of the collector output. The FP scales as ,

$$FP \propto CFM^3$$

for a fixed collector geometry. Hence at 250 CFM, the FHP is .89% of the collector output.

Collector efficiency can be increased by increasing the flow rate and simultaneously increasing the flow area so the Reynolds number says the same. The (L/D) ratio becomes more favorable, hence investigation of this approach appears worthwhile, Viz,

(1) Increase CFM

(2) Keep CFM/D = const. , or V = const.

Scaling CFM/D = const, yields the following fan - horsepower (FHP) results, discharge temperatures and efficiencies,

				I = 300		I = 100	
CFM	L/D	P, psi	F P	T_D	η	T_D	η
200	144	.95	.046	139	.61	87	.545
250	115	.83	.050	127	.64	84	.57
300	96	.76	.055	118	.657	81	.595
350	82	.71	.060	111	.665	79	.608
400	72	.67	.065	106	.678	77.5	.62

The higher CFM results given above indicate improved efficiency at lower temperatures. Many system considerations are involved in the selection of the CFM, including the collector size and cost, the fan size and cost, matching the collector with the furnace and delivered temperature distribution in the heated space, At the present time it is proposed to carry 200 CFM /panel preliminary designs.

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Performance of the collectors when storing energy was evaluated and typical results are given below :

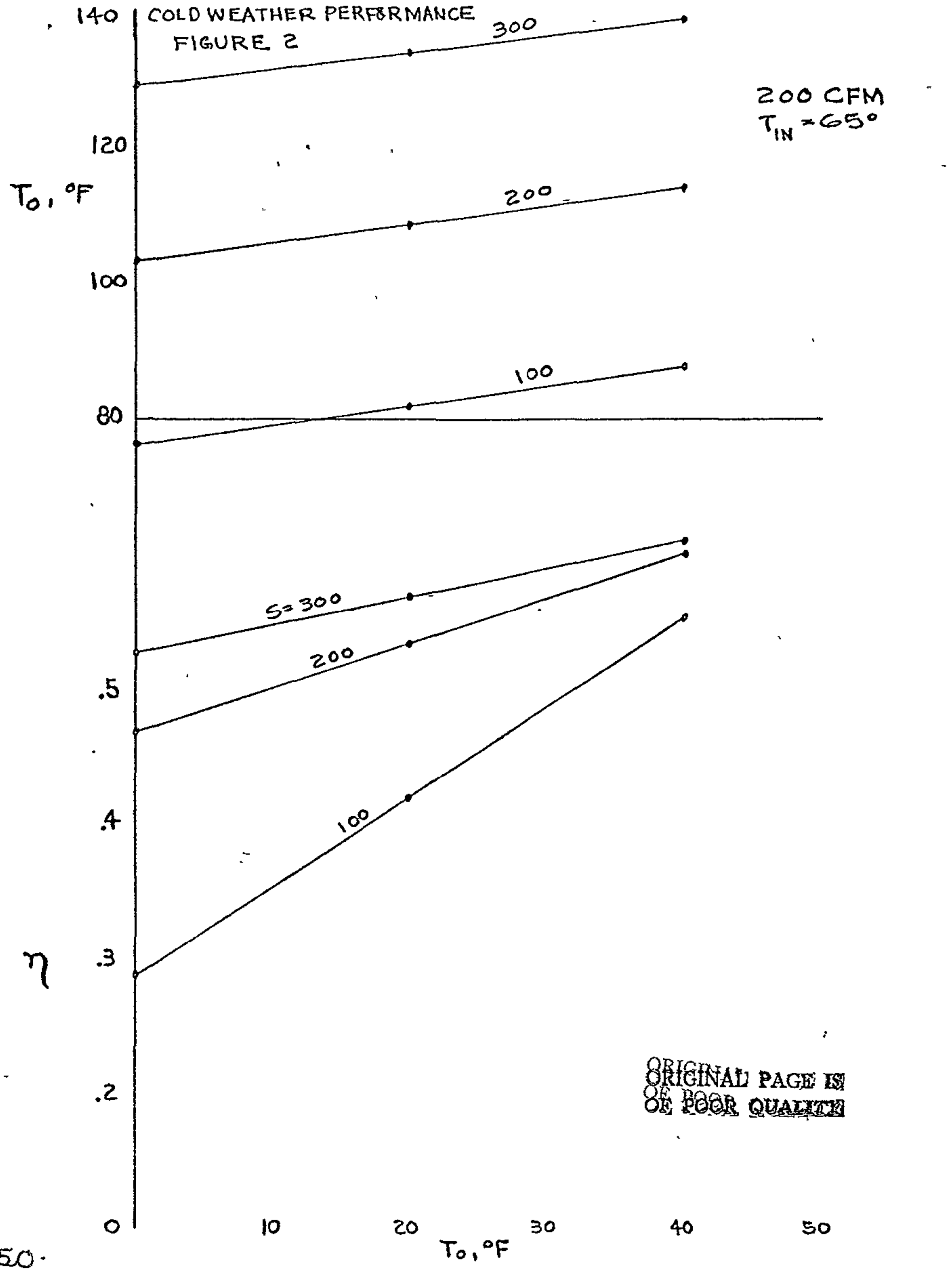
CFM	I	Tin	Tout	η
200	300	137	186	.40
400	300	151	175	.40

The inlet temperature Tin corresponds to air leaving the heat exchanger , hence the higher CFM is able to deliver the same amount of heat at higher water temperature. The results of cold weather calculations are given in Figure 2.

An analytical expression for the heat transfer is useful for preliminary design studies:

$$T_D = .26 I + .25 T_o + 48.5$$

$$\eta = \frac{2.474}{I} (.26 I + .25 T_o - 16.5)$$



FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	DRAWN APP'D DATE	FIGURE 2 COLD WEATHER PERFORMANCE DWG. NO.	REV.

4. Discussion of Simplified Calculation Model

Under a typical warm air operating conditions (65°F inlet) preliminary calculations indicate the inner cover (2) temperature ranges from 70°F to 110°F at an insolation of 300 Btu/ft^2-hr, in which case about 10% of the long-wave radiation would be transmitted. The absorber temperature ranges from 135 to 150°F. The transmitted IR heat is ,

$$Q_{IR.\ Trans.} = .1 \times 3600 \times .88 \left[\left(\frac{602}{1200} \right)^4 - \left(\frac{550}{1200} \right)^4 \right] = 6.1 \ \text{Btu/ft}^2\text{-hr}$$

The actual loss will be less, as 90% of this is absorbed in the front cover. As there are other complex design areas influencing the collector, the assumption made for preliminary design trade is that the covers are opaque to long-wave IR Radiation.

The loss formula (EQ.1) also neglects the effect of solar absorption, assuming that heat transport through each cover is due only to internal radiation and free convection. In order to assess this effect, the solar absorption was estimated. The average transmission of solar energy is 0.875 according to the manufacturer. For non absorbing covers, the reflection coefficient is

$$r = \left(\frac{n - 1}{n + 1} \right)^2$$

where n = index of refraction = 1.52 for the cover material.

Hence r = .0425 and $\tau = (1 - r_2) / (1 + r_2) = .918$

Since the actual transmission is 0.875, the difference of .043 is attributed to absorption. As the inner cover froms one side of the air duct, the absorbed fraction is available for air pick-up.

A heat balance on the first surface yields[3]

$$Q_2 + A_1 I = h_w (T_1 - T_o) + \frac{(\bar{T}_1^4 - \bar{T}_o^4)\ 3600}{\frac{1}{E_c} + \frac{1}{E_c} - 1}$$

For $h_w = 4$, (10 mph wind), $E_c = .88$ (Plastic Cover)

and $T_o = 50^{\circ}F$, we have

T_1	$h_w (T_1 - T_o)$	$\frac{3600(\bar{T}_1^4 - \bar{T}_o^4)}{(2/E_c - 1)}$	$Q_2 + A_1 I$
52	8	1.71	9.71
54	16	3.36	19.36
56	24	5.03	29.03

The above results indicate that an increase in the temperatures (T_1) of 2°, increases the heat loss by about 10 Btu/ft^2 -hr. The absorption term $A_1 I$ at I = 250 Btu/ft^2-hr is about 10 Btu/ft^2 -hr ; hence the absorption term tends to have a favorable effect in reducing the loss. Use of the loss formula as given by EQ.1 is conservative.

The absroption on the inner cover is included in the heat balance equation given in a subsequent section. For preliminary design purposes the solar absorption effects are neglected.

The agreement of the predicted free convection loss model with recently published loss models correlated with Rayleigh number was checked.

The Rayleigh Number is

$$R_a = g\beta \rho^2 \frac{\Delta T}{\mu^2} H^3 \underline{\quad P_r \quad}$$

For a typical case

$T_2 = 95\,^{\circ}F$, $T_1 = 63\,^{\circ}F$, $\bar{T}_{12} = 79\,^{\circ}F$

$H = 1/12$ ft

$R_a = 2.88 \times 10^4$.

The correlation given in Ref. 4 in this range

$N_u = .229 \; (R_a \cos \gamma)^{.252} \qquad 5900 < R_a \cos\gamma < 92300$

$N_u = .157 \; (R_a \cos \gamma)^{.285} \qquad 92300 < R_a \cos\gamma < 10^6$

For $\gamma = 60^{\circ}$

$N_u = 2.5$

$h = .46$ Btu/ft^2-hr

$Q_{loss} = .46\,(95 - 63) = 14.72$ Btu/ft^2 - hr

The free convection loss relationship used to derive Eq. 1, is ,

$Q_{loss} = .17 \; (95 - .63)^{5/4} = 12.94$ Btu/ft^2 - hr.

Hence the Rayleigh No. correlation predicts a loss 14% higher than that predicted by Eq. 1. Increasing the loss coefficient C_c by 14% yields $C_c = .19$.

A loss coefficient of $C_c = 0.20$ was used for conservatism.

5. Alternative Calculation Method

An alternate calculation method was set-up to include the effects of the collector cover absorption, transmission and spacing in more detail.

The addition equations are described below .

Heat balance on surface $\underline{\underline{1}}$

$$h_w \; (T_1 - T_o) + E_1 \, .3600 \, (\bar{T}_1^4 - \bar{T}_o^4) = \qquad (10)$$

$$\underbrace{.9 \times 3600 \; \frac{(\bar{T}_2^4 - \bar{T}_1^4)}{(1/E_2 + 1/E_1 - 1)}}_{\text{90\% of IR Transmitted}} + \underbrace{\frac{.09 \times 3600 \; (\bar{T}_4^4 - \bar{T}_2^4)}{1/E_4 + 1/E_2 - 1}}_{\text{90\% of IR absorbed}} + \underbrace{.04 I}_{\text{Solar-absorption}} + \underbrace{h_1 (T_2 - T_1)}_{\text{Free Convection}}$$

The value of h_1 is calculated via the Rayleigh No. correlation noted above and given in Ref. 4.

Equation 2 is modified ;

$$Q_2 = h_w \; (T_1 - T_o) + \mathcal{E}_1 (3600) \, (\bar{T}_1^4 - \bar{T}_o^4) + \frac{.1 \times 3600 \; (\bar{T}_2^4 - \bar{T}_1^4)}{(1/E_2 + 1/E_1 - 1)} \qquad (2a)$$

$$+ \frac{.01 \times 3600 \; (\bar{T}_4^4 - \bar{T}_2^4)}{(1/E_4 + 1/E_2 - 1)}$$

Equation 5 is modified

$$Q \text{ air} = (A_1 + A_2 + A_4) \, I - Q2 - (T_7 - T_o)/R_7 \qquad (5a)$$

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Calculations with the alternative formulation using a 2" space between the covers were done for

$T_o = 30^\circ$, $I = 300$, $C = 200$

$Tout = 134^\circ F$.

The previous calculations yielded Tout = 133, hence the calculated results are very close.

6. Performance With Glass Cover

A comparison of the exterior glass and plastic cover was made for :

T_{in} = $65^{\circ}F$

T_o = $50^{\circ}F$

I = 300 Btu/ft^2-hr

C = 200 CFM

	Tout	n
Glass	140	.615
FRP	137.5	.60

A comparison of the 3 collector approaches under cold weather conditions was also made for :

T_o = $10^{\circ}F$

I = 300 Btu/ft^2-hr

C = 200 CFM

Tin = $65^{\circ}F$

The results are :

	T out	n
Two-pass	130	.54
Single Front Pass	122	.47
Single Rear Pass	127	.51

The two-pass collector is superior to either single pass aproach. The previous two-pass calculation with a FRP outer cover gave n = .53 ; hence the glass cover appears beneficial to the collector performance.

7. Absorber Plate Emissivity

The potential benefits of incorporating a low emissivity absorber was evaluated. A high temperature inlet condition was evaluated with the following results ;

$I = 300$ Btu/ft^2-hr.

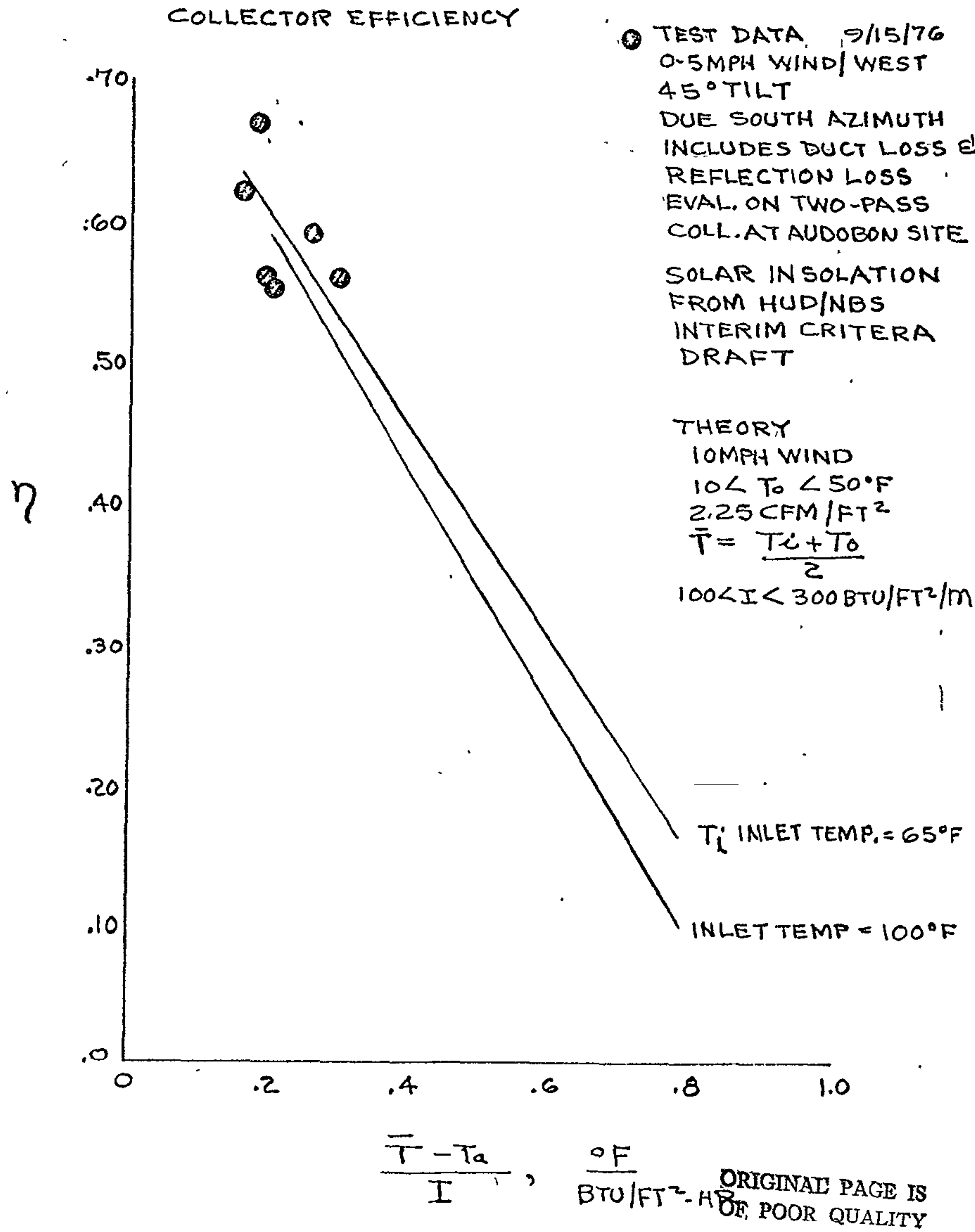
$C = 200$ CFM

E_4	T in, $^\circ$F	T out, $^\circ$F	n
.89	133	190.6	.41
.1	133	198.6	.47

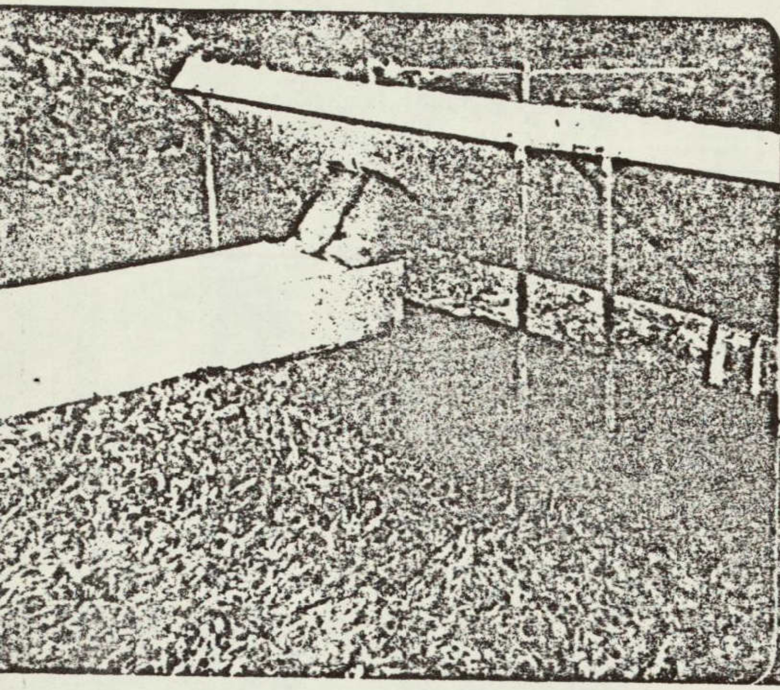
A 6% in efficiency is noted for the low emissivity absorber.

8. Performance Data

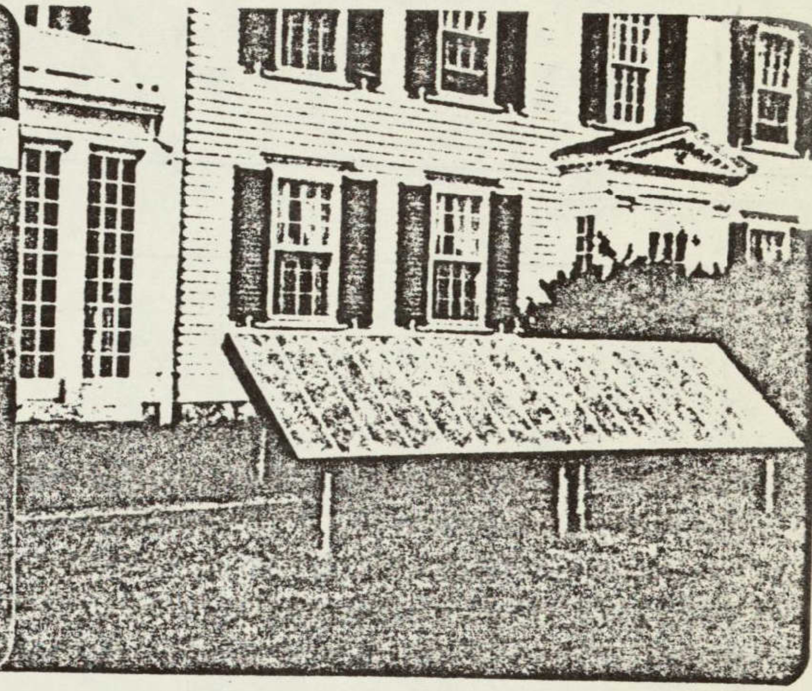
Performance Data is currently available on a two-pass air - heating collector being used to heat domestic hot water. Field measurements were taken on the air flow using a pitot-static tube and immersing a thermometer. Comparison of experimental data and theoretical performance calculations is shown in Figure 3 . Photographs of the test site are shown in Figure 4. Test data is given in Figure 5.



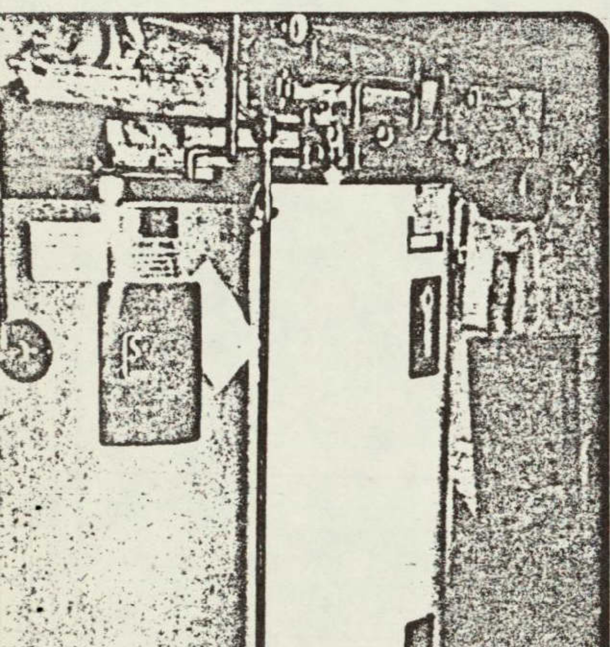
	DRAWN	COLLECTOR EFFICIENCY		
FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS **U.S.A.**	APP'D			
	DATE	DWG. NO. FIGURE 3	REV.	



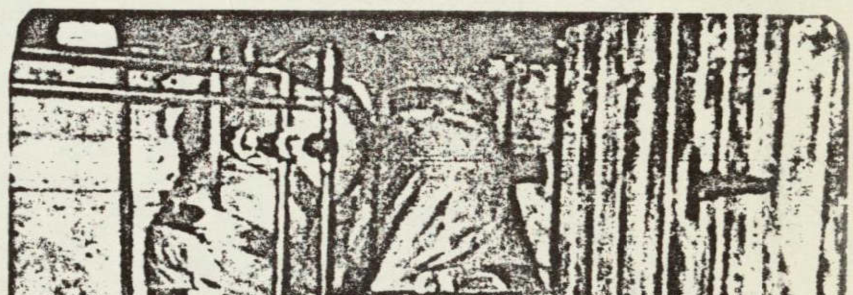
b) Air Ducting Hook-Up



a) Exterior View



(c) Solar Heat Storage



(b) Heat Exchanger

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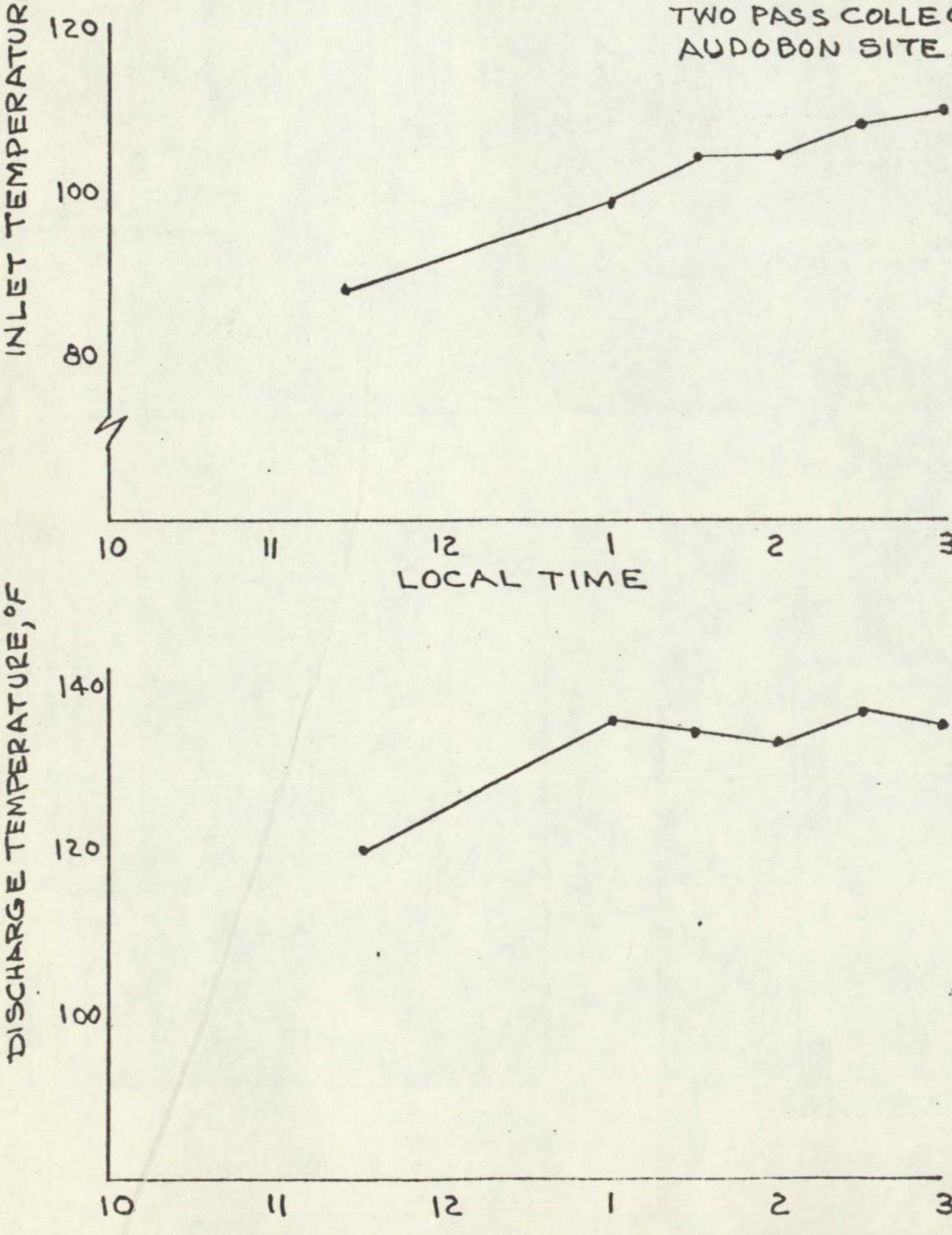
Figure 4. Fern Engineering Operational System for Mass. Electric at Audobon Society in Topsfield, Massachusetts

TEST DATA - 9/15/76

COLLECTOR DATA

0-5MPH WIND/WEST
45° TILT
DUE SOUTH AZIMUTH
INCLUDES DUCT LOSS &
REFLECTION LOSS
TWO PASS COLLECTOR AT
AUDOBON SITE.



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FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	DRAWN	TEST DATA
	APP'D	
	DATE	DWG. NO. FIGURE 5 REV.

Symbols

Station and surface numbers and subscripts

0 - ambient condition

1 - external cover

2 - surface of inside cover

3 - bulk air temperature between absorber and inside cover

4 - absorber surface facing cover

5 - absorber surface facing back of collector

6 - bulk air temperature in duct behind absorber
(not applicable to front-pass collector)

7 - surface of rear face insulation

A_c = collector net useful area, ft^2

A_3, A_6 = Flow cross section area of airducts ft^2

C = Air flow CFM at inlet temperature

Q_2 = heat loss to ambient from inside cover, Btu/ft^2-hr.

T_o, T_1, T_2, T_3, T_4, T_5, T_6, T_7 - Temperatures at various stations and surfaces, $^\circ F$

q_o = dynamic pressure, PSF

L = collector length, ft

P = pressure, PSF

FHP = fan power

F = Flow loss factor, accounts for turns

$\bar{T}$ = $(T + 460) / 1200$

Symbols continued

C_c = free convection coefficient, $Btu/ft^2-hr-{}^oF^{5/4}$

h_w = forced convection coefficient for winds, $Btu/hr-ft^2-{}^oF$

σ = Sletan Boltzman constant

$\epsilon_{1,2,4,5,7,}$ = emissivity of various surfaces

f_w = free convection coefficient defined in Ref. 1

f = friction coefficient

r = reflection coefficient of cover

n = index of refraction

τ = transmissivity of cover

$A_{1,2,4}$ = absorptivity of various surfaces

I = Solar insolation incident on collector

Tin = inlet air temperature to collector

Tout = discharge air temperature from collector

g = acceleration due to gravity

β = expansion coefficient in Rayleigh No., = 1/ (T+460)

μ = viscosity, lbm/ft - sec

K = conductivity, $Btu/hr - ft - {}^oF$

P_r = Prandtl No.

R_e = Reynolds No.

h_2, h_4, h_5, h_6 = film coefficients at various surfaces, $Btu/hr-ft^2$

Symbols continued

N_u = Nusselt No.

R_a = Rayleigh No.

γ = collector tilt angle

H = gap between covers, ft.

R_7 = thermal resistance $^{o}F/Btu/hr\text{-}ft^2$ of insulation

D_3, D_6 = hydraulic diameters for air ducts 3 and 6.

η = collector efficiency

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3. Goodman, R.D., and Menke, A.G. " Effect of Cover Plate Treatment on Efficiency of Solar Collectors ".
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4. Buchberg, H., Catton, I., Edwards, D.K., " Natural Convection in Enclosed Spaces - A review of Application to Solar Energy Collection " ,
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5. Kays, W.M., "Convective Heat and Mass Transfer "
McGraw Hill Book Co., 1966.

Product Bulletin 201

DECORATIVE PRODUCTS DIVISION

DATE: NOVEMBER 1973

"NEXTEL" BRAND VELVET COATING
SERIES 101

I. DESCRIPTION

"NEXTEL" Brand Velvet Coating Series 101 is an air dry enamel designed for spray application to properly primed surfaces. This coating uniformly scatters light regardless of incidence angle and provides a velvet-like appearance without the glare of ordinary flat and textured finishes. The resulting surface has a soft, velvety appearance and provides extremely uniform light diffusion over a wide range of viewing angles. Dirt and light abrasion are easily removed from the smooth surface without changing its original appearance.

Black 101-C10 is the recommended Velvet Coating Optical Black and is designed for application to interior surfaces or cameras, optical equipment and darkrooms where control of stray light reflection is required.

White 101-A10 is suitable for most optical white uses, however, White 202 A10 provides slightly more total reflectance.

Note: Not intended for interior architectural applications.

Typical uses include:

1. Optical and electro optical equipment, instrument interiors, and darkrooms – To minimize stray light reflections.
2. Instrument dials – For consistent contrast thru a broad range of viewing angles.
3. As a contrast with bright chrome and stainless steel – For unique accents, such as on exterior automotive trim.
4. Aircraft glare shield nose exteriors – To reduce glare.
5. Infrared absorbing coating – For thermal control.

	Colors	Primers
White	101 A10	Series 901
Black	101-C10	*DuPont 65 Line*
		Rinshed Mason U51C008 Clear Primer (for chrome and stainless steel)

This bulletin does not contain application procedures for the subject products The appropriate Instruction Bulletin(s) may be obtained by contacting your Decorative Products sales representative.

II. DURABILITY

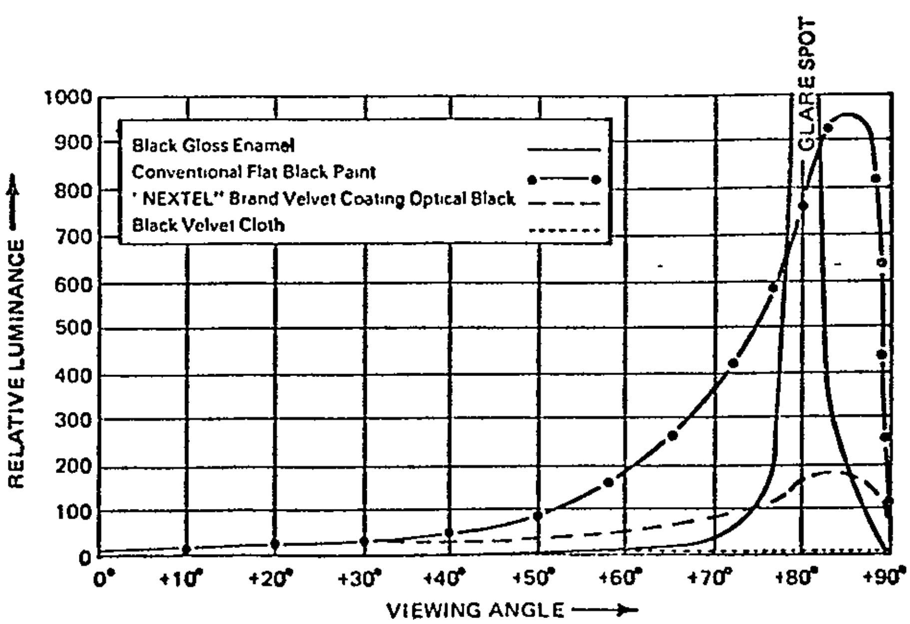
When applied in accordance with procedures recommended by 3M, the following exterior performance may be expected. Durability statements are based upon representative experience obtained from testing throughout the United States; however, actual durability will be determined by substrate selection and preparation, exposure conditions and maintenance of the coating.

White 101-A10	3 Years
Black 101-C10	3 Years

III. PROPERTIES

Values given are typical and not for use in specifications.

A. Optical



The graph shows the relative luminance of "NEXTEL" Velvet Coating Optical Black, conventional flat black paint, black gloss enamel, and black velvet cloth at various viewing angles when illuminated from - 80°. At viewing angles to +40°, all materials appear much the same. Beyond 40°, the conventional flat black appears lighter because of glare. For gloss black, this glare component is present over a much narrower range, but is many times more intense. "NEXTEL" Velvet Coating minimizes this glare and is effective as a light trap even when viewed directly opposite the source (+80°).

Initially, the 85° gloss of "NEXTEL" Velvet Coating is less than half that of conventional flat finishes. When rubbed, the advantage increases considerably, since ordinary flat finishes burnish to much higher gloss when rubbed or cleaned. Listed below are 85° Glossmeter readings of "NEXTEL" Velvet Coating and conventional flat finishes after various amounts of abrasion from a Gardner Laboratory Scrubbing Machine (Fed. Test Method Std. No. 141a, Method 6143).

85° GLOSSMETER READINGS				
	New	25 Scrubs	125 Scrubs	250 Scrubs
Commercial Flat Finish	5	10	12	14
"Nextel" Velvet Coating	2	2	2	2

PROPERTY	TEST METHOD	RESULTS
Gloss	ASTM D523-53T	
101-C10	60°	0
	85°	2
Total Reflectance	ASTM E97-55	
101-C10		<2 1/2%
101-A10		>85%
Solar Absorption		
101-C10		0.98
101-A10		0 21
Infrared Emittance	At 25°C.	
101-C10		0 89
101 A10		0 88
Infrared Reflectance	Fed. Test Method Std. No. 141a Method 6241 at 5% Incidence	
	Millimicrons	101-C10
	700 1400	1/2%
	1400 2500	2%
		101-A10
	600	90%
	800	88%
	1000	82%
	1500	70%
	2000	65%
	2600	50%

B. PHYSICAL

PROPERTY	TEST METHOD	RESULTS
Outgassing Weight Loss	10^{-5} to 10^{-6} torr. 84°F. (29°C), 20 hours	$1\ 3 \times 10^{-4}$ gm/cm^2
Thermal Conductivity		0.17 BTU/hr./ft.^{2} / ft/°F. (101 C10 only)

PROPERTY	TEST METHODS	English Units	Metric Units
Reverse Impact		60 in. lb.	6.8×10^7 dyne-cm.
Abrasion Resistance	Weight loss, 750 Taber Abraser cycles, CS 0 wheels, 1000 gm. load (Fed. Test Method Std. 141a Method 6192)	—	.01 gram
Coverage (Approximately)	On 2' x 3' (61 cm x 91 cm) Flat Panel	200 sq. ft./ Gal.	5 sq. meters/ liter
Temperature (Maximum recommended continuous service)	500 hours exposure		
	101-C10	300°F.	149°C.
	101-A10	150°F.	65°C.

IV. SAFETY

WARNING: FLAMMABLE. Contains-petroleum distillates. Keep from heat, sparks, and open flames. Use only in well ventilated areas Avoid prolonged breathing of vapors and repeated contact with skin. Keep closed when not in use. KEEP OUT OF REACH OF CHILDREN.

Attachment II

Appendix II

KALWALL CORPORATION

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P. O. BOX 237
MANCHESTER, N. H 03105
TELEPHONE A/C 603 627-3861

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SUN-LITE SOLAR COLLECTOR COVER MATERIAL BY KALWALL

In direct response to the needs of the solar energy industry, Kalwall Corporation has developed two special fiberglass sheets - Sun-lite Regular and Sun-lite Premium - for use as cover plates on solar collectors. Our primary objective has been to offer a cover material with the highest performance, consistent with low product and installation costs. Sun-lite meets this cost/performance objective.

Sun-lite, both Regular and Premium, has been tested by our own laboratories, and by many outside, independent groups doing research in the solar energy field. The general conclusion has been that Sun-lite performs at the same level or better than glass in all solar transmittance characteristics. Sun-lite thus creates the same "greenhouse" effect as is created with glass. Further, Sun-lite is impact-resistant and insensitive to thermal shock. This toughness is a genuine advantage in handling Sun-lite in a manufacturing plant where its convenient form minimizes storage space and inventory damage. The continuous strips of Sun-lite also eliminate the leaky and maintenance-prone horizontal joint often found in large glass collectors. In addition, since the coefficient of expansion of Sun-lite is virtually the same as that of aluminum, glazing details with Sun-lite are less complicated than they are with other plastics. Unlike plastic films, Sun-lite due to its stiffness, does not suffer from fatigue caused by wind flapping. The light weight of Sun-lite (1/10 that of 3/16" glass) is another distinct advantage in Solar Collector design, where lower weight will be significant in reducing the building design loads, glazing and installation difficulties, and packing and shipping costs.

Sun-lite Premium differs from Sun-lite Regular in that the Premium grade is more resistant to ultra-violet degradation than the Regular grade, and will retain its excellent solar transmission properties longer in normal service (we estimate 20 years compared with 7 years for Regular). However, Sun-lite Regular is slightly higher in initial solar energy transmittance as some of the special ingredients in the Premium grade, which account for its longer life, are not used in the Regular. Premium with its very advanced performance, is unlike any other fiberglass sheeting and is a Kalwall exclusive.

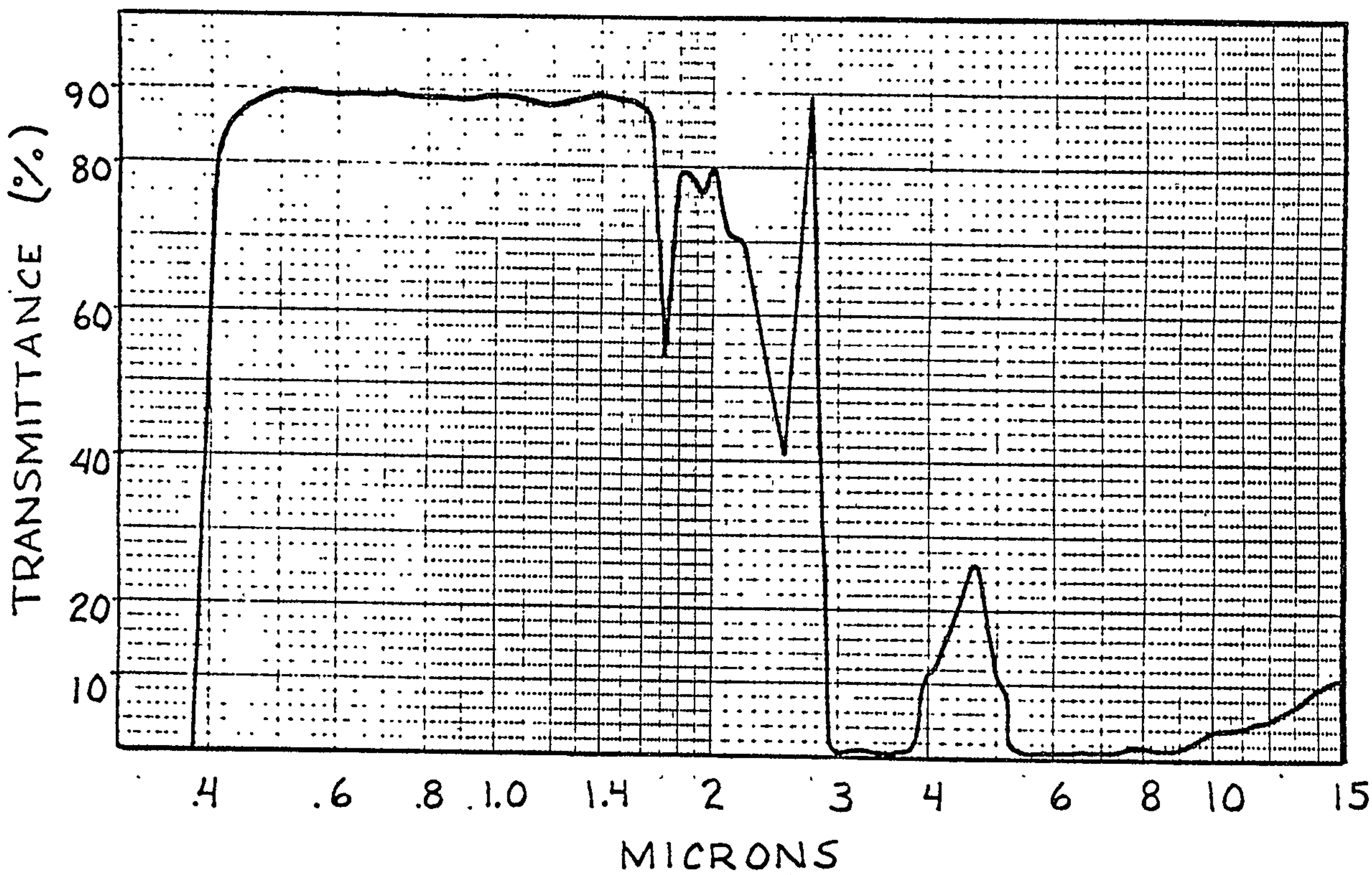
The Sun-lite resin/fiberglass system is quite stable at high temperatures which have little effect on its physical properties. As a thermosetting material, it will not melt, sag or cold flow. Its ignition temperature exceeds 900°F which is higher than that for wood. When exposed to temperatures of around 200°F, both Regular and Premium may show a reduced solar transmittance of up to 5% over their expected life time. This change will be gradual and, in many situations, the transmittance loss will not reach 5%. If the collector cover is going to be exposed to unusually high temperatures of over 300°F for extended periods of time, Sun-lite Premium should be used since it is more stable at these higher temperatures. Its solar transmittance decline will probably be around 10% after considerable exposure at these high temperatures. In our experience, the glazing surfaces of well-designed collectors will not be exposed to 300°F, and most designs will not exceed 200°F.

In addition, Kalwall is continually improving Sun-lite's properties through its research program. Sun-lite is unique, and from a cost/performance point of view, Sun-lite is without equal.

Form: SL-38
Date: 5/22/75

SPECTRAL TRANSMITTANCE

SUN-LITE PREMIUM FIBERGLASS SHEETING



SOLAR TRANSMITTANCE
VERSUS ANGLE OF INCIDENCE

SUN-LITE PREMIUM FIBERGLASS SHEETING

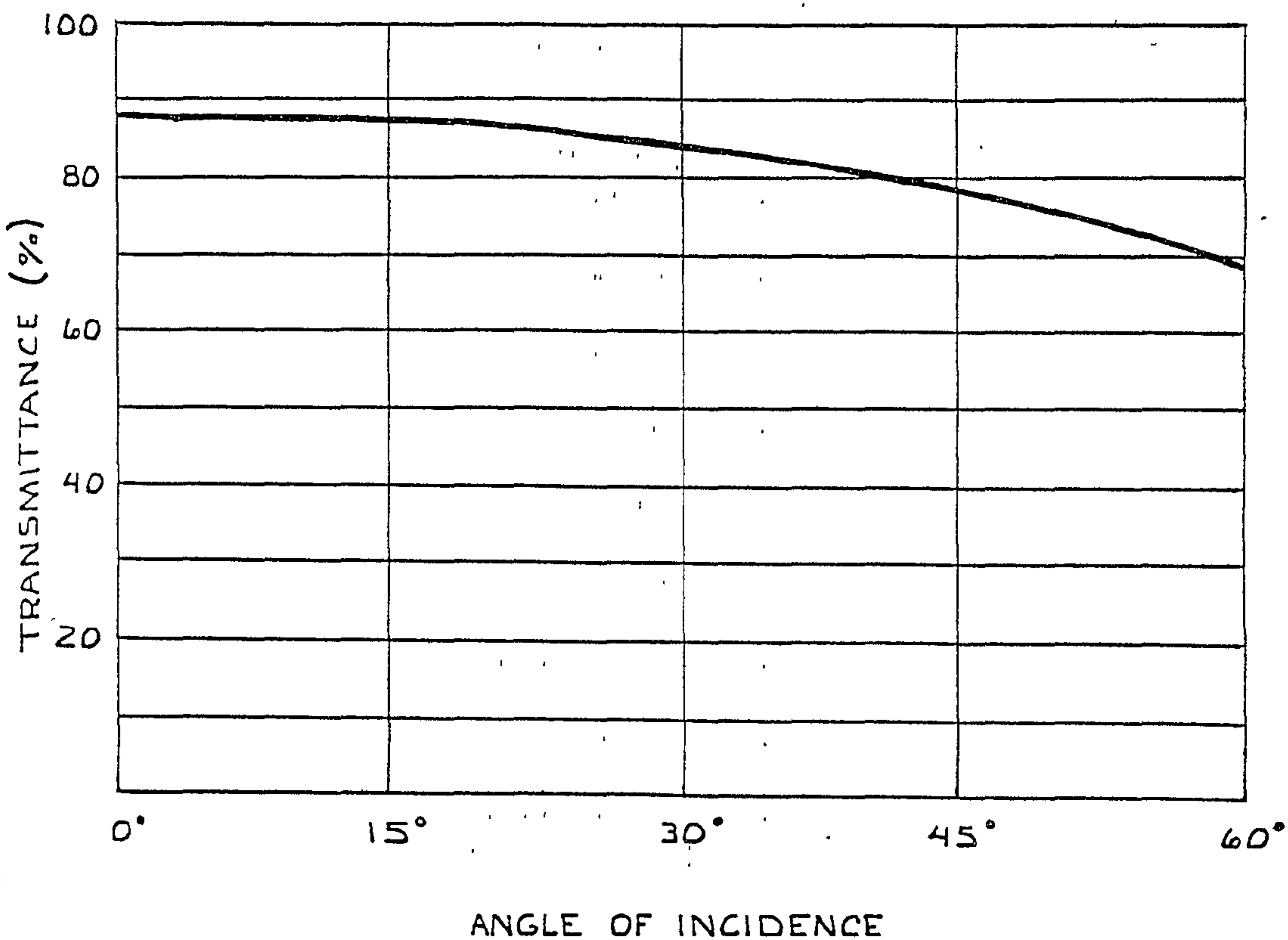


Table 1 - Glass and Sun-Lite Fiberglass Compared to Ideal

	Ideal	3/16" Glass	Sun-Lite
Transmittance, solar, ASTM E424	1.00	.83 to .87	.85 to .90
Transmittance, heat (5 $_u$M to 50 $_u$M)	0.00	.1	.1
Thermal Conductivity, (BTU-in/hr-ft^2-^{o}F)	0.00	6 to 7	.87
Thermal Expansion Coef., PPM*/^{o}F	12**	4 to 5	14
Impact Strength, SPI ball drop, ft-lbs	high	10	60
Ultimate Tensile Strength, ASTM D638, psi	high	10,000	16,000
Weight per sq ft (.040" Sun-Lite), lbs	low	2.4	0.3
Resistance to chemicals, UV & time	excellent	excellent	very good
Handling & Cutting	safe, easy	difficult	safe, easy
Cost (in quantity from distributor), \$/ft^2	low	.52	.28 to .46

* Parts per million ** To match aluminum used in most collectors.

Figure 1 Spectral Transmittance, Sun-Lite Fiberglass Sheet

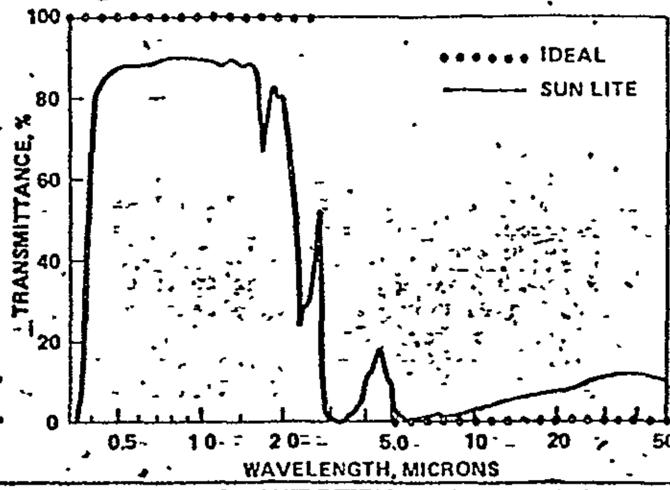
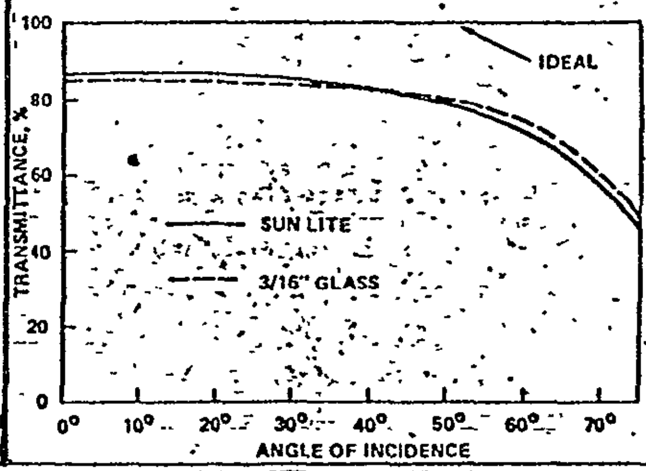


Figure 2 Solar Transmittance Versus Angle of Sun-Lite Fiberglass Sheet Incidence



SUN-LITE SOLAR COLLECTOR COVER MATERIAL

AVERAGE PHYSICAL PROPERTIES	METHOD	UNITS	SUN-LITE REGULAR	SUN-LITE PREMIUM
Solar Energy Transmittance	E 424 Method B	%	85%-90%	85%-90%
Estimated Solar Lifetime(1)		Years	7	20.
Thermal Sensitivity(2)	@	200^{o}F	Excellent	Excellent
	@	300^{o}F	Poor	Good
Heat Transmittance	5-20 Microns	%	10%	10%
Index of Refraction	D 542	Ratio	1.54	1.52
Tensile Strength	D 638	PSI	16,000	11,000
Flexural Strength	D 790	PSI	24,500	22,000
Flexural Modulus	D 790	PSI x 10^6	1.0	0.6
Shear Strength	D 732	PSI	14,000	12,000
Izod Impact	D 256	Ft.lb./In.	18	10
Water Absorption	D 570	%	0.20-0.33	0.50-0.60
Thermal Expansion	D 696	(In./In./^{o}F) x 10^{-5}	1.4	1.4
Thermal Conductivity	C 177	BTU-In./Hr./Ft.^{2}/^{o}F	.87	.87
Specific Heat	D 2766	BTU/lb./^{o}F	.35	.35
Specific Gravity	D 792	Ratio	1.4	1.4
Weight	NBS PS53	Oz./Ft.^{2}	2.8-4.7	2.8-4.7
Thickness	NBS PS53	Inches	.025 or .040	.025 or .040
Sheet Size	NBS PS53	Feet	4' or 5' wide, up to 1,200' long	

The above information is presented in good faith and believed to be correct to the best of our knowledge, but no warranty is expressed or implied.

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WEATHERABILITY OF FIBERGLASS SOLAR COLLECTOR COVERS

by

JAMES S. WHITE

Product Development Manager

KALWALL CORPORATION

James S. White, Product Development Manager for Kalwall Corporation, one of the leaders in research and development of fiberglass reinforced plastics for over 25 years, has been involved in the development of special FRP products for the past 8 years, including the development of the accepted industry testing procedures for load/deflection, shatter resistance, and the visual rating system for weathered FRP.

Mr. White is a registered professional engineer and an active member of the Society of the Plastics Industries, the Society of Plastics Engineers, American Society for Testing and Materials, Solar Energy Industries Association, and the American Society of Heating, Refrigeration and Air-Conditioning Engineers.

ABSTRACT

The objective of this paper is to demonstrate the proper method of evaluating the weatherability of fiberglass solar collector covers and to document the performance after weathering. The following discussion will cover such important features for solar collector covers as solar energy transmission, ultraviolet degradation, thermal degradation, surface erosion, impact resistance, and thermal shock.

Three basic types of fiberglass reinforced polyester sheet are considered. The first material (referred to as material #1) is a proprietary

material developed specifically for the solar energy industry by Kalwall Corporation called Sun-Lite Premium. The second material, (referred to as material #2) Sun-Lite Regular, is also manufactured by Kalwall Corporation but can be considered an acrylic modified, highly light stabilized polyester. The third material considered, for purposes of a bench mark for comparison, is the various types of standard grade fiberglass reinforced polyester sheet. (Referred to as material #3)

SOLAR ENERGY TRANSMISSION

When considering fiberglass reinforced polyester for a solar collector cover, one of the most important properties is solar energy transmission. Accurately measuring the solar energy transmission of diffuse materials such as fiberglass reinforced polyester has caused many researchers problems because of the light scattering phenomena. The preferred method of measuring the solar energy transmission of diffuse materials, according to the American Society for Testing and Materials, is ASTM E 424 (Test for Solar Energy Transmittance and Reflectance of Sheet Materials) Method B. This method requires a 28" x 28" sample to minimize the effect of light scattering. Initial solar energy transmission for a "super clear" fiberglass reinforced polyester can run between 80% and 90%.

Method A of ASTM E 424 has been used with widely varying degrees of success. Many problems have been encountered because of small samples and the location of the sensing device relative to the intergrading sphere surface. However, with proper care, solar energy transmission by wavelength curves can be generated. (Figure #1)

Fiberglass reinforced polyester has the desirable properties of very high transmission over the typical solar spectrum and near opacity in the longwave range for excellent heat trapping properties.

ULTRAVIOLET DEGRADATION

Degradation due to ultraviolet radiation has long been of great concern to those people designing or using products exposed to sunlight. Researchers in the fiberglass reinforced polyester industry have come a long way in retarding ultraviolet degradation. A typical non-light stabilized general purpose polyester can lose more than 15% transmission in just 50 hours of exposure to a sun lamp. One of the earliest attempts to improve the UV resistance of polyester was to add ordinary aspirin as a light stabilizing additive. After 50 hours exposure to a sun lamp, a general purpose polyester will only lose 5% transmission if aspirin is added as a light stabilizer.

Obviously, today's researchers have gone much beyond aspirin in the field of light stabilization. Altering the polyester backbone (modifying the glycols and acids which make up polyester), adding acrylic, adding sophisticated light stabilizers, and applying special coatings or films are necessary to produce a quality solar collector cover.

In order to facilitate research into UV degradation, several different weatherometers were developed and are in general use today. The most common are the Fluorescent, Carbon Arc, and Xenon weatherometer. The Fluorescent weatherometer has a high concentration of UV and causes more severe changes than the Carbon Arc or Xenon weatherometer.

Although it is extremely difficult to correlate weatherometer hours to real time outdoors, many researchers use 250-400 weatherometer hours as approximately 1 year actual weathering. (2,000 hours would approximately equal 5 years real time.)

Samples were exposed in a fluorescent weatherometer for 2,000 hours. Color change (ΔE) and light transmission readings were taken at 500 hour intervals.

COLOR CHANGE - Measurements were taken in accordance with ASTM D 1929. Material #1 had a total color change of 3.5 after 2,000 hours. Material #2 had a color change of 10. Depending on formulation, a standard grade fiberglass reinforced polyester sheet could have a total color change of around 28. (Figure #2)

In order to verify the weatherometer results, color change measurements were taken on a sample of material #1 weathered in South Florida for 5 years. The color change was found to be 4.4. (A specially coated piece of material #1 had a color change of only 1.1!)

South facing exposures in South Florida are considered the most severe natural environment in the United States because of large quantities of sunlight, heat, and moisture.

LIGHT TRANSMISSION - Light transmission measurements were taken on the same weatherometer specimens. Material #1 lost only 3% light transmission after 2,000 hours, while material #2 lost 11%. A standard grade fiberglass reinforced polyester can lose up to 20% light transmission in only 500 hours exposure time. (Figure #3)

It is apparent from the above data that it is extremely important to consider the grade of fiberglass reinforced polyester when trying to decrease ultraviolet degradation. Another area of extreme importance for solar collector covers is thermal degradation.

THERMAL DEGRADATION

In most efficiently operating flat plate collectors the cover temperature will not be above 200°F; therefore, tests were conducted on samples continuously aged in a 200°F oven for 1 year. The drop in solar energy transmission for both material #1 and material #2 was approximately equal (about 10%). However, the standard grade sheet lost more than 50% solar energy transmission in one year of continuous exposure. (Figure #4)

Cover plate temperatures higher than 200°F may occur during stagnation in collectors with improperly designed venting systems. Stagnation temperatures occur when no fluid (water or air) is flowing through the collector. For example, with 300 BTU/sq. ft./hr. insolation, the absorber plate could reach 380° and the inner cover of a double cover could reach 260° if the outside temperature is 60°. For this reason, short term tests were conducted at 300°F. (Figure #5)

After 300 hours (equal to 10 hours/day for 30 days), material #1 lost only 2% solar energy transmission, while material #2 lost 4%. Extending the test to 5,000 hours, material #1 lost approximately 10% solar energy transmission at 300°F. Material #2 lost 22% and standard grade material lost 40% under the same conditions.

SURFACE EROSION

(continued next page)

One of the weathering factors that should be considered for maintenance of long term performance is surface erosion. Surface erosion is the actual physical wearing away and oxidation of the surface. The result is exposed fibers on the surface sometimes called "fiber bloom." In order to measure the amount of surface erosion, measurements were taken with a Clevite 1200 Surfanalyzer. Both average roughness and peak-to-valley roughness were measured.

First consider average surface roughness (the average magnitude of all surface irregularities reported in microinches or 1/1,000,000 of an inch). The surface erosion for material #1 is not noticeable for the first three years of outdoor weathering in South Florida. However, at the end of four years, some surface roughness was noticeable, and after five years, there was about 55 microinches of average erosion. A standard grade of fiberglass can have more than 105 microinches of average erosion after only two years of South Florida exposure. (Figure #6) In order to halt this kind of surface erosion, a proprietary high temperature coating manufactured by Kalwall Corporation called Kalwall Weatherable Surface can be applied. After five years of weathering exposure, only 14 microinches (hardly noticeable to the human eye) of average erosion was measured on material #1 with this coating. (Material #1W) A more dramatic measurement is of peak-to-valley roughness. (Figure #7)

Using peak-to-valley measurements instead of average measurements, material #1 showed a maximum roughness of approximately 300 microinches change, while the coated sample showed only 100 microinches. Both samples had been weathered for five years in South Florida. A standard grade sheet can have more than 1,000 microinches of erosion after only two years to this same exposure.

IMPACT RESISTANCE

Embrittlement is often of great concern to people using plastics. One of the major reasons fiberglass reinforced polyester is used as a solar collector cover is its remarkable impact strength and shatter resistance. Unlike glass, which can be easily broken into dangerously sharp pieces, fiberglass reinforced polyester is completely shatter resistant. The best way to measure impact resistance for solar collector covers is to use the falling ball method.

To prove fiberglass does not lose its remarkable impact strength after many years of outdoor exposure, a 14 year old sample was taken from a building and tested. The control (un-weathered) sample required 25 foot pounds, (6.4 pound steel ball dropped from 4 feet) to cause a rupture of the material while the 14 year old sample required 32 foot pounds (6.4 pounds from 5 feet) to cause rupture.

Low temperature impact does not cause a problem. Tests have been conducted on fiberglass reinforced polyester at -40°F, and the results showed almost a 50% increase in the dynamic load required to cause failure. (Figure #8)

THERMAL SHOCK

The final property to be considered for a solar collector cover to be able to withstand the effects of weathering is thermal shock. Many times during the life of a solar collector, a rain storm or other rapid change in temperature may cause a severe thermal shock to a heated collector. To test fiberglass reinforced polyester's resistance to thermal shock, a sample was heated to 300°F and then quickly submerged in cold water. The thermal shock did not cause any harmful effects or noticeable degradation.

SUMMARY

It has been shown that high grades of fiberglass reinforced polyester exhibit excellent weatherability. Critical properties for solar collector covers such as solar energy transmission, ultraviolet and thermal degradation resistance, erosion resistance, impact resistance, low temperatures, and thermal shock have been examined and shown to be highly acceptable for safe and efficient use in the solar industry.

Special thanks are extended to the American Cyanamid Company and Owens-Corning Fiberglass Corporation for their generous technical assistance and testing.

Figure #1

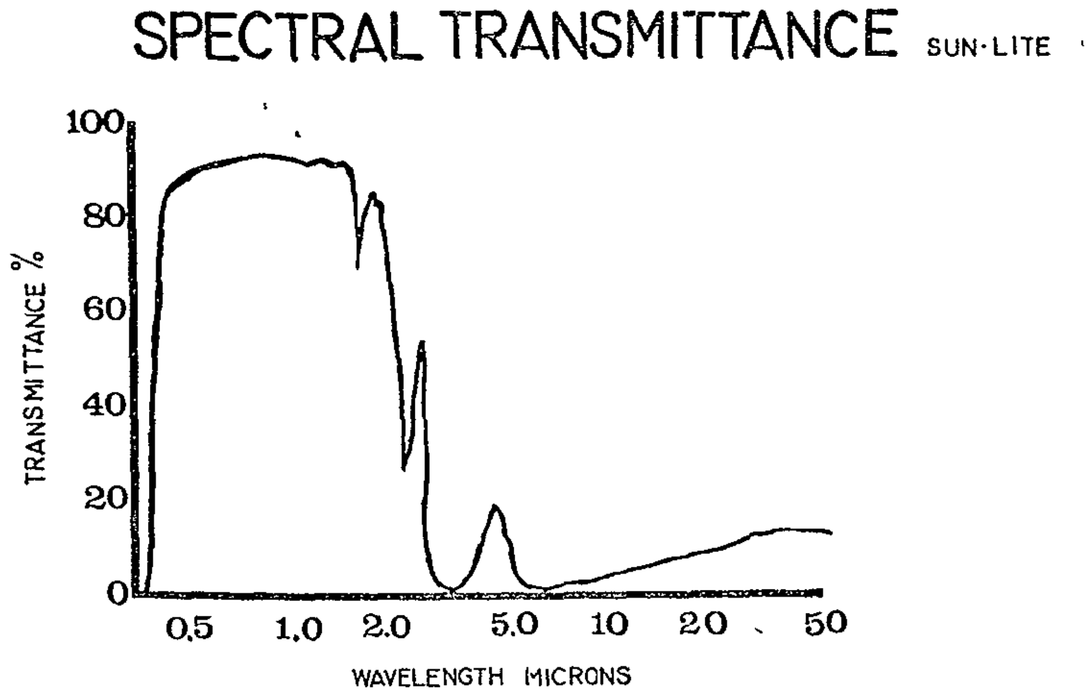

Figure #2

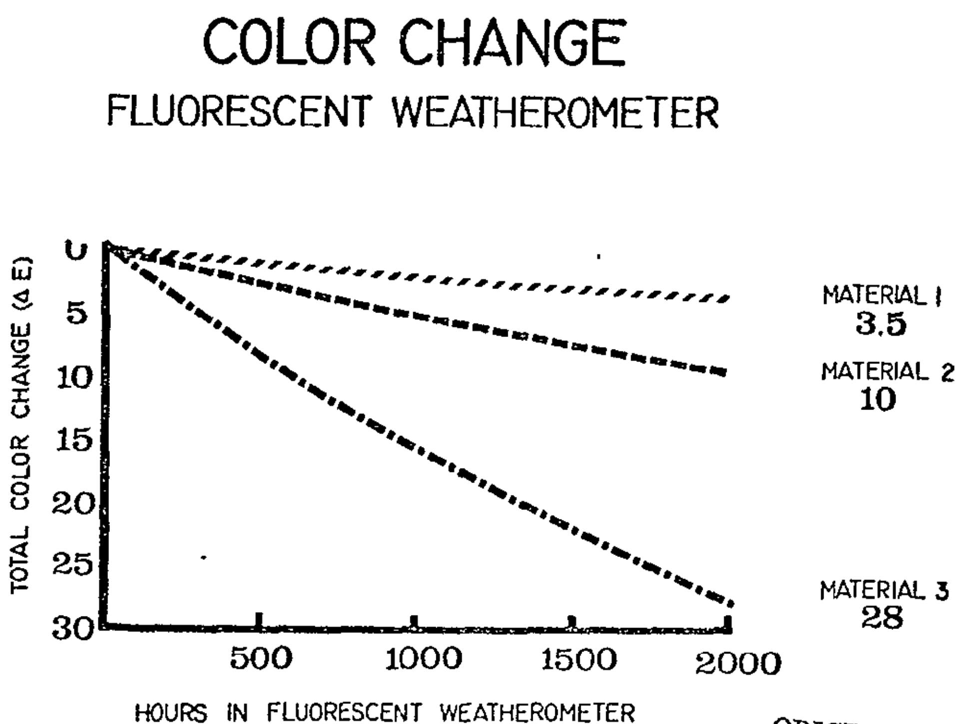

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Figure #3

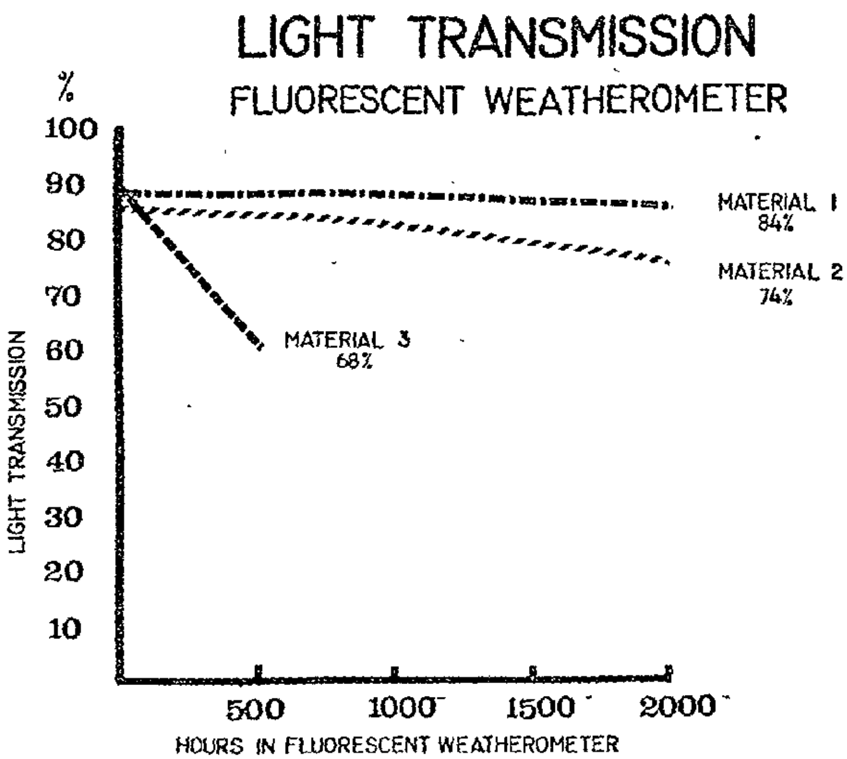


Figure #4

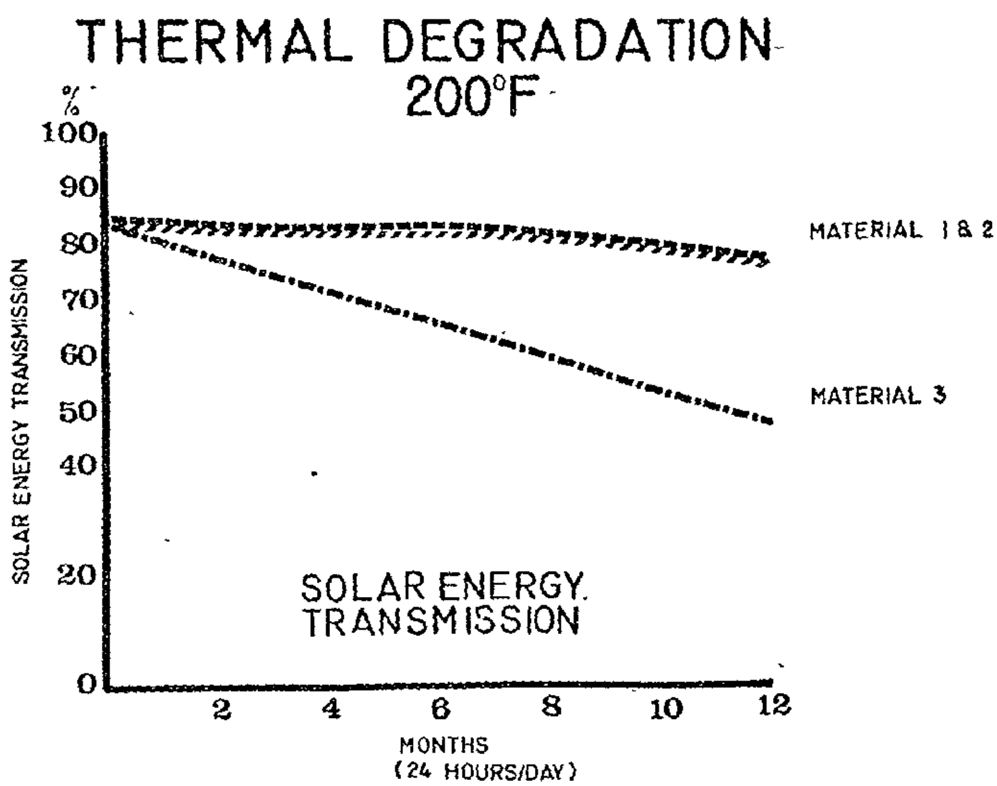


Figure #5

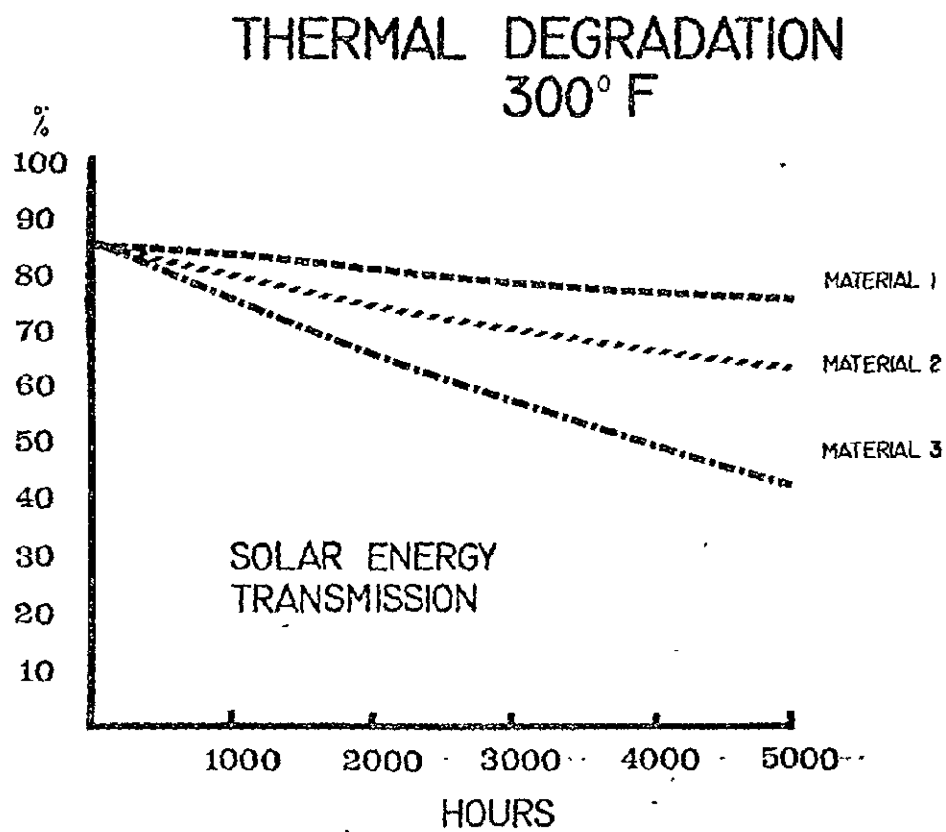


Figure #6

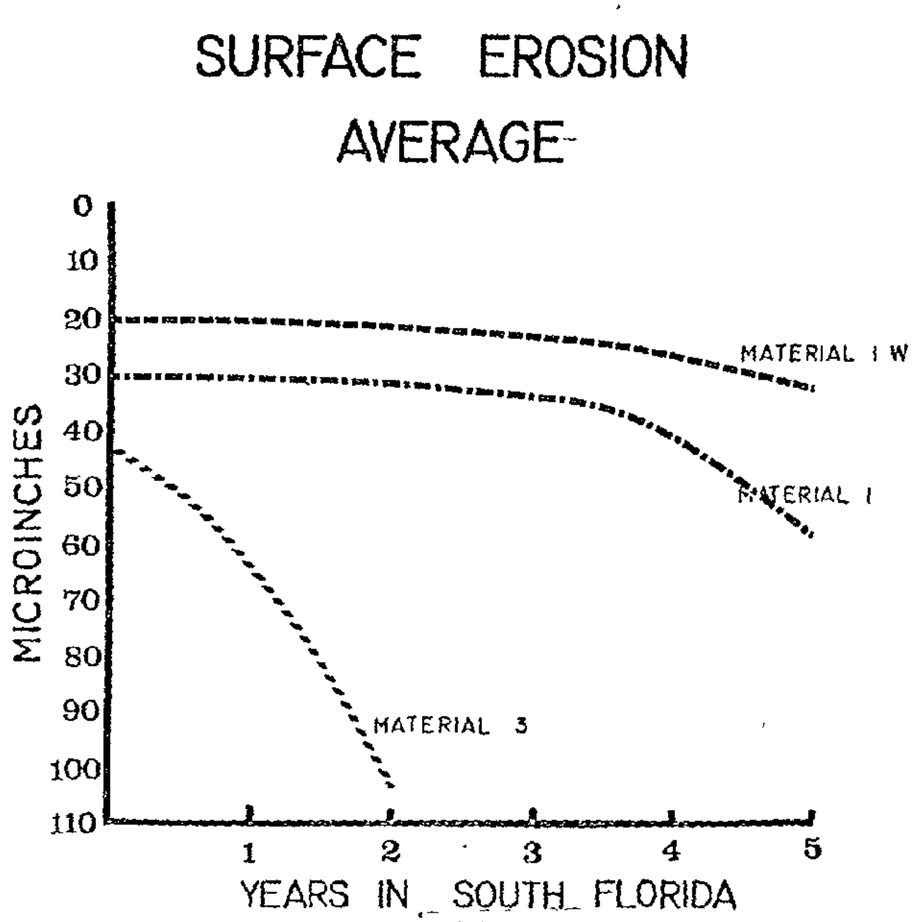


Figure #7

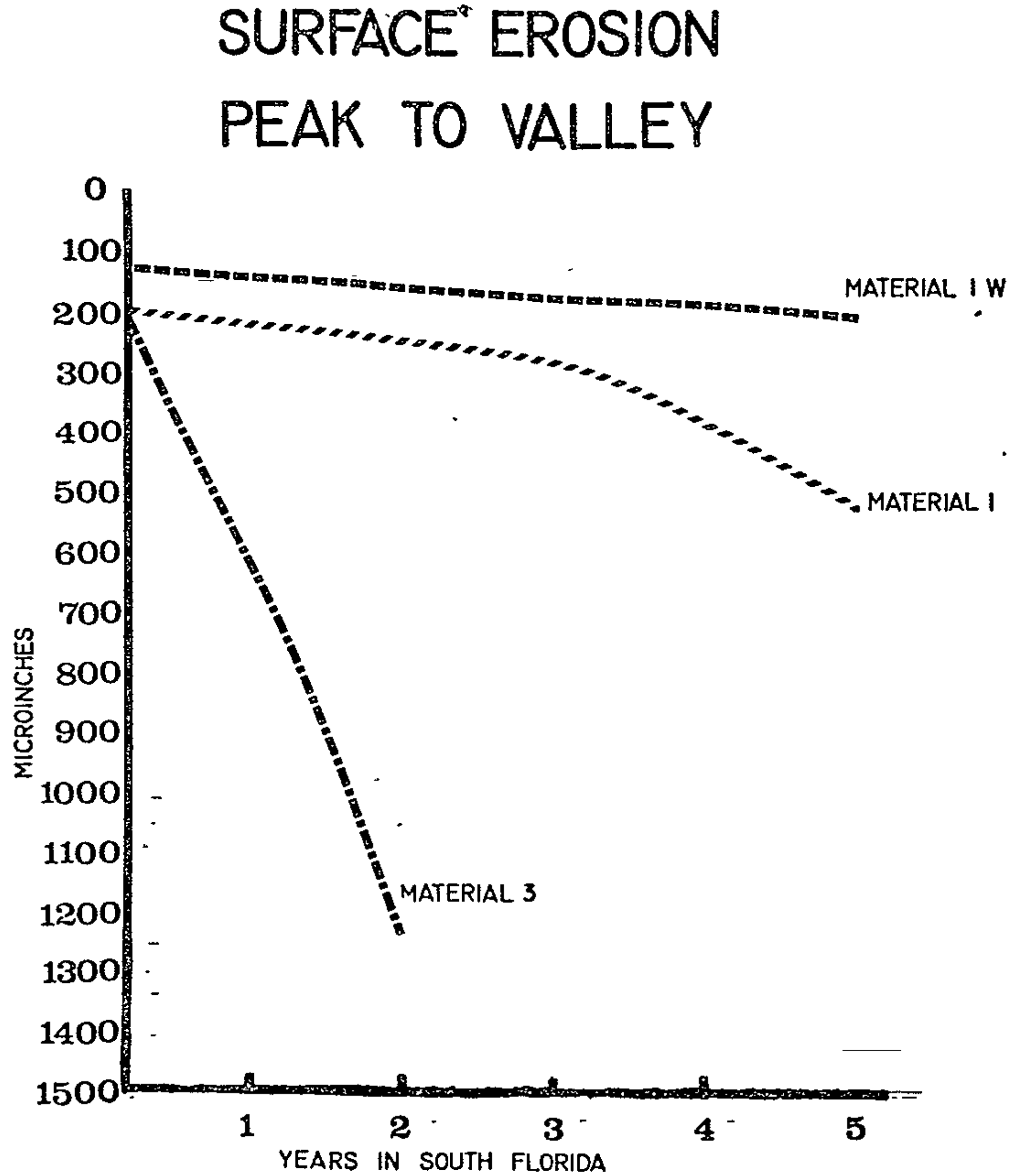
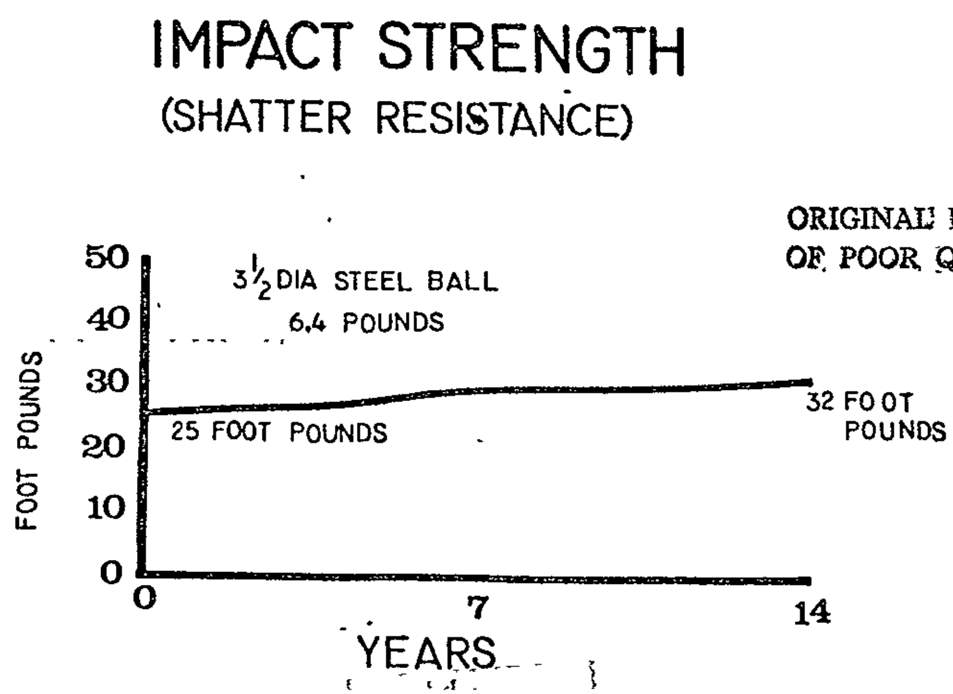


Figure #8



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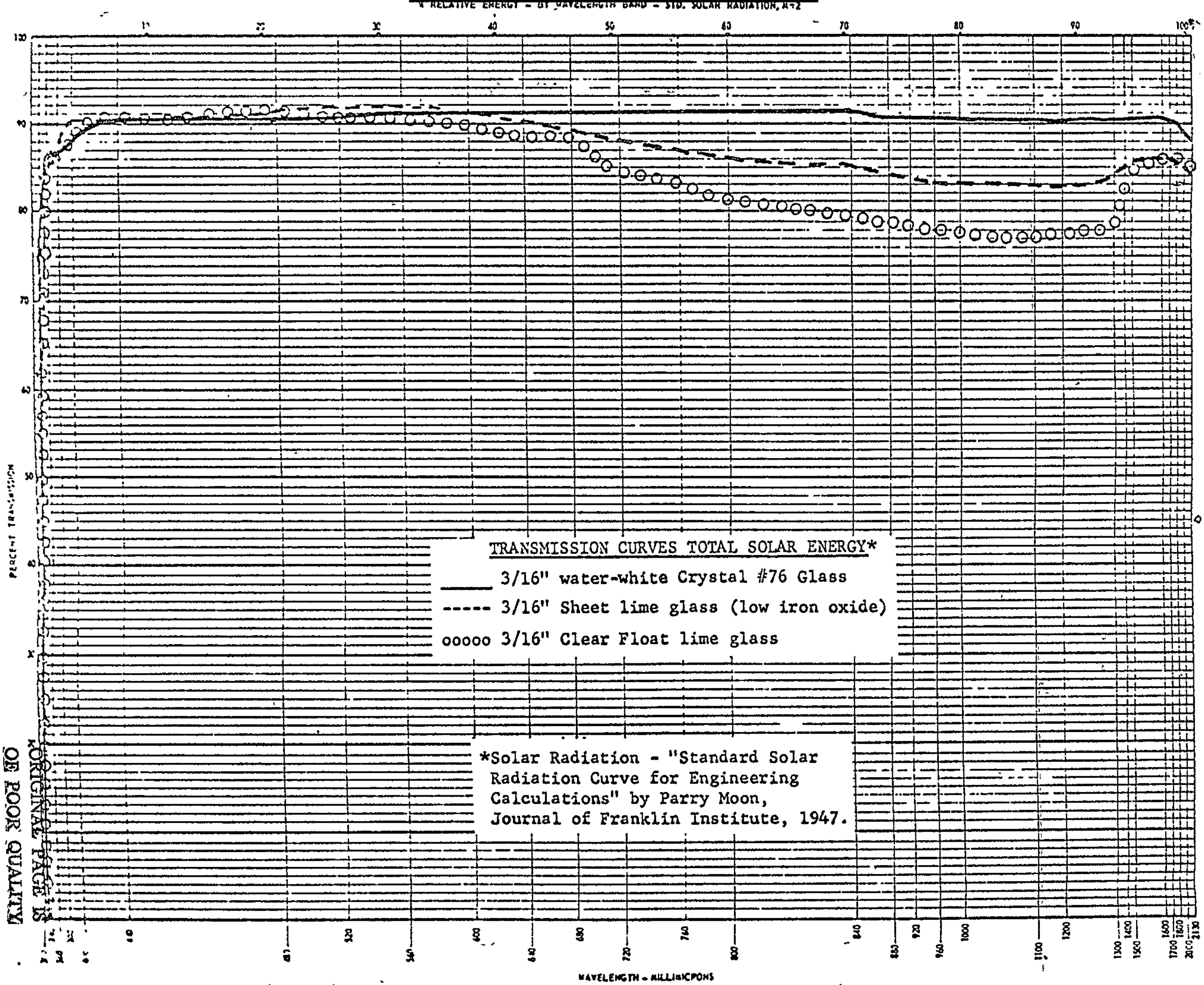
APPENDIX III

ASG Industries Inc.
The Glass Company
POST OFFICE BOX 929 KINGSPORT TENNESSEE 37662 615/245-0211

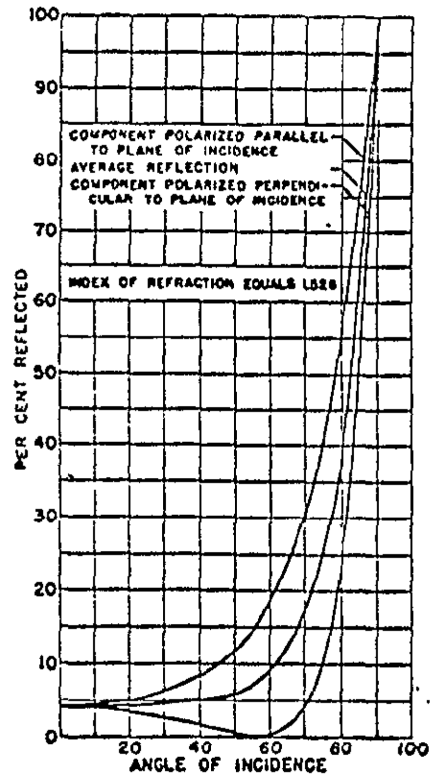
PRESENTATION AT INTERNATIONAL SOLAR ENERGY CONFERENCE

Ft. Collins, Colorado, August 21–23, 1974

TRANSPARENT GLAZING MEDIA FOR SOLAR ENERGY COLLECTORS



% RELATIVE ENERGY - BY WAVELENGTH BAND - STD. SOLAR RADIATION, M=2
PERCENT TRANSMISSION
WAVELENGTH - MILLIMICRONS
TRANSMISSION CURVES TOTAL SOLAR ENERGY*
_____ 3/16" water-white Crystal #76 Glass
----- 3/16" Sheet lime glass (low iron oxide)
ooooo 3/16" Clear Float lime glass
*Solar Radiation - "Standard Solar Radiation Curve for Engineering Calculations" by Parry Moon, Journal of Franklin Institute, 1947.



REFLECTION OF DIRECT SOLAR RADIATION FROM A SINGLE SURFACE AS A FUNCTION OF ANGLE OF INCIDENCE

PERTINENT PHYSICAL PROPERTIES OF TRANSPARENT GLAZING MEDIA

GLAZING MEDIA	REFRACTIVE INDEX	**SOLAR ENERGY TRANSMISSION (PER SHEET)	**SOLAR ENERGY LOSSES PER SHEET (2 SURFACE REFLECTION PLUS ABSORPTION)	***MAXIMUM OPERATING TEMP.
ORDINARY CLEAR LIME GLASS (FLOAT) (IRON OXIDE CONTENT 0.10% TO 0.13%)	1.52	1/8"(3.2 mm) - 85% 3/16"(4.8 mm) - 81% 1/4"(6.0 mm) - 78%	15% (8.2%R + 6.8%A) 19% (8.0%R + 11.0%A) 22% (7.9%R + 14.1%A)	400° F (204°C)
SHEET LIME GLASS (LOW IRON OXIDE CONTENT 0.05% TO 0.06%)	1.51	DS(3.2 mm) - 87% 3/16"(4.8 mm) - 85%	13% (8.1%R + 4.9%A) 15% (8.0%R + 7.0%A)	400° F (204°C)
WATER-WHITE CRYSTAL GLASS #76 (0.01% IRON OXIDE)	1.50	5/32"(4.0 mm) - 91% 3/16"(4.7 mm) - 90.5% 7/32"(5.5 mm) - 90%	9% (8.0%R + 1.0%A) 9.5% (8.0%R + 1.5%A) 10% (8.0%R + 2.0%A)	400° F (204°C)
*METHYL METHACRYLATE (100% ACRYLIC COLORLESS CAST SHEET)	1.49	1/8"(3.2 mm) - 89% 3/16"(4.7 mm) - 87% 1/4"(6.2 mm) - 85%	11% (7.6%R + 3.4%A) 13% (7.5%R + 5.5%A) 15% (7.4%R + 7.6%A)	180° TO 190° F (82° TO 93°C)
*POLYCARBONATE CLEAR SHEET (LIGHT STABLE TYPE)	1.586	1/8"(3.2 mm) - 81% 3/16"(4.7 mm) - 78% 1/4"(6.2 mm) - 74%	19% (9.8%R + 9.2%A) 22% (9.6%R + 12.4%A) 26% (9.4%R + 16.6%A)	250° TO 270° F (121°C TO 132°C)

*Data obtained from manufacturers' Literature and/or Photometric Testing.

**Solar Radiation Calculations based on "Standard Solar Radiation Curve for Engineering Calculations", by Parry Moon, Journal of Franklin Institute, 1947.

***Maximum Operating Temperature for Continuous Operation. For Annealed Glass Thermal Endurance of Cool Edges Dependent on Temperature Differential.

CONTRACT : NASA 8-32246 — Document No. 5001-2

SUBJECT : Solar Collector No-Flow Temperature — Job No. 198

AUTHOR : P. Levine — Date : December 7, 1976

1. CRITERIA

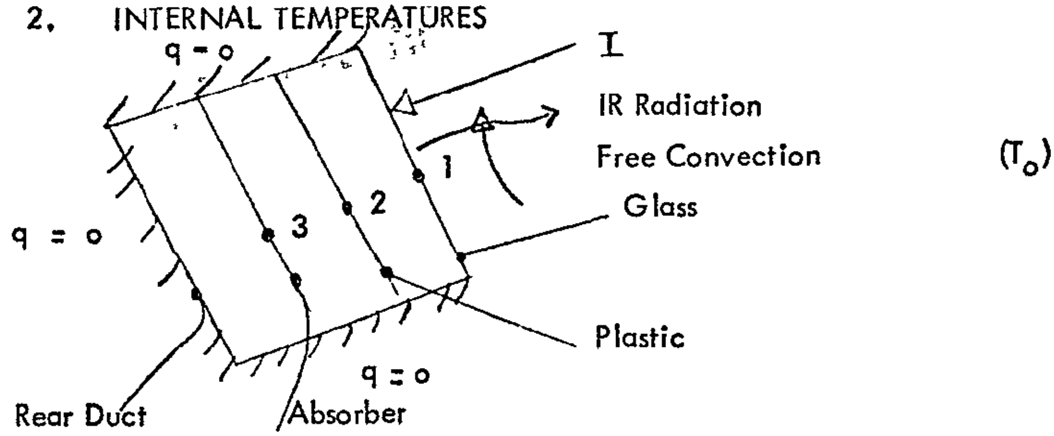
I_{max} = 340 Btu/ft^2 - Hr

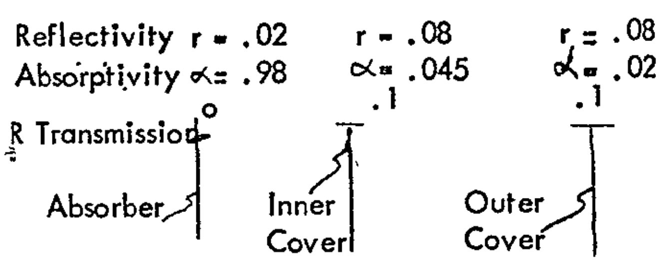
Arbitrary Installation Angle

Front-End Space Loss Only

T_o = 100^{o}F

2. INTERNAL TEMPERATURES





The total absorbed insolation is (.84 I) ; approximately 10% of this is transmitted as IR radiation through the front cover, and the remainder must be shed by the front cover, or 257 Btu/ft^2-Hr.

The emittance of the glass is taken as 0.88*

* H. Tabor "Radiation, Convection and Conduction Coefficients in Solar Collectors", Irael Bull Resch. Vol. 6C, 1958.

The free convection loss from the front, depends on the Grashof No.,

$$\rho^2 g B (T-T_o) L^3 / \mu^2$$

Values of $gB\rho^2 / \mu^2$ are ,

T	$gB\rho^2/\mu^2$, $1/F, ft^3$	Pr
100	1.76×10^6	.72
200	$.85 \times 10^6$	.72
300	$.44 \times 10^6$	.71

The short-side L = 4 - ft will be used as the length. The order of magnitude of the Grashof No. is ,

$$.85 \times 10^6 \times 100 \times 64 \times .72 = 4 \times 10^9$$

For $Gr > 10^9$, McAdams recommends ,

$$\overline{N}uL = 0.10 (Gr_L P_r)^{1/3} \qquad \text{Vertical Plate}$$

$$\overline{N}uL = .14 (Gr_L P_r)^{1/3} \qquad \text{Horizontal Plate}$$

For a conservative result, the vertical plate Nusselt No. is assumed.

The heat loss from the front side is then ,

$$q_1 = 25\% = \sigma_1 E_1 (\overline{T}_1^4 - \overline{T}_o^4) + h_L (T_1 - T_o)$$

Assuming $T_o = 100^\circ F$, then :

T_1	$G_r P_r$	N_{uL}	h_L ,	$h_L (T_1 - T_o)$	$\sigma_1 \epsilon_1 (\overline{T}_1^4 - \overline{T}_o^4)$	q_1
150	3.53×10^9	152	.61	30.5	61.4	91.9
200	6.01×10^9	182	.73	73	140	213
250	4.4×10^9	164	.70	105	238	343

The variation of T_1 with q_1 is shown in Figure 1. For the installation with perfectly insulated edges and rear, the front-heat loss is 257 Btu/ft^2-hr and $T_1 = 218^\circ F$.

The temperature of the inner cover depends on the I.R. radiation and free convection losses. As the front cover absorbs 2% of the incoming radiation, the 2nd cover must shed 250 Btu/ft^2-hr. The heat loss is,

$$q_2 = \sigma \frac{\overline{T_2}^4 - \overline{T_1}^4}{\frac{1}{E_1} + \frac{1}{E_2} - 1} + h_2 \ (T_2 - T_1)$$

As shown in document No. 1002-2, a conservative estimate on h_2 is $h_2 = .16 \ (T_2 - T_1)^{\frac{1}{4}}$; the emissivities $E_1 = E_2 = .88$. The heat loss variation with T_2 is,

T_2	h_2	$h_2 \ (T_2 - T_1)$	$\dfrac{\sigma (T_2^4 - T_1^4)}{1/E_1 + 1/E_2 - 1}$	q_2
250	.38	12	58	70
300	.48	39	167	206
310	.50	46	191	237
320	.51	52	217	269

The inner cover temperature of $315^\circ F$ results in the heat loss of 250 Btu/ft^2-hr.

The absorber must shed $0.78 I = 265$ Btu/ft^2-hr.

The heat loss formulation is the same as for the inner cover (under our assumption of perfect insulation), and $E_4 = .89$

T_4	h_4	$h_4 \; (T_4 - T_2)$	$\dfrac{\sigma \, (T_4^{\;2} - T_2^{\;2})}{1/E_4 + \frac{1}{E_2} - 1}$	q_4
400	.48	41	256	297
390	.47	35	222	257

The peak absorber temperature is 392 $^\circ$F , corresponding to a heat loss of 265 Btu/ft^2-hr.

3. INSULATION LOSSES

The calculation above assumes that as a result of design and/or installation the side and rear losses are negligible. An estimate of these losses is useful to evaluate the degree of conservatism.

The insulation between the two covers is 1-inch of high temperature isocyanurate with a $K = .15$ Btu/ft^2-hr.-$^\circ$F/inch. The total edge loss, prorated to collector area is,

$$q_{Edge} = \left(\frac{1.5}{12} \times 20\right) \frac{(1/9)\,(172)}{32} = 1.5 \text{ Btu/ft}^2\text{-hr.}$$

The sides of the air-ducts are insulated with 1 ½ inches of high temperature fiberglass insulation, similar to Certainteed K - 242. The conductivity at an estimated average temperature of 250°F is .35 Btu/hr - ft^2 $^\circ$F/inch, hence the heat loss is ,

$$q_{\text{ Side}} = \frac{4.5}{12} \times \frac{20 \times 292}{6.3 \times 32} = 11 \text{ Btu/ft}^2 \text{ - hr.}$$

The side and edge losses constitute about 3% of the front loss. The rear of the collector is insulated with 2.5 - inches of fiberflass ; the linear 'ra diation loss coefficient and free convection in the rear duct add more thermal resistance so,

$$q_{rear} = \frac{1}{11} (292) = 26.5 \text{ Btu/hr-ft}^2$$

If these losses are accounted for, then,

$$q_1 = 218 \text{ Btu/ft}^2\text{-hr}$$

and $T_1 = 203^\circ F$, indicating only a $15^\circ F$ decrease in temperature. The inner cover must shed 211 Btu/ft^2-hr , so,

T_2	h_2	$h_2(T_2 - T_1)$	$\frac{\sigma(T_2^4 - T_1^4)}{1/E_1 + 1/E_2 - 1}$	q_2
300	.51	51.	196	247
290	.49	44	173	216
285	.49	41	161	202

The inner cover temperature drops to $288^\circ F$ including losses.

The absorber must shed 226 Btu/ft^2-hr., so,

T_4	h_4	$h_4(T_4 - T_2)$	$\frac{\sigma(T_4^2 - T_2^2)}{1/E_4 + \frac{1}{E_2} - 1}$	q_4
350	.45	29	167	196
360	.47	35	197	232

The absorber temperature is $358^\circ F$.

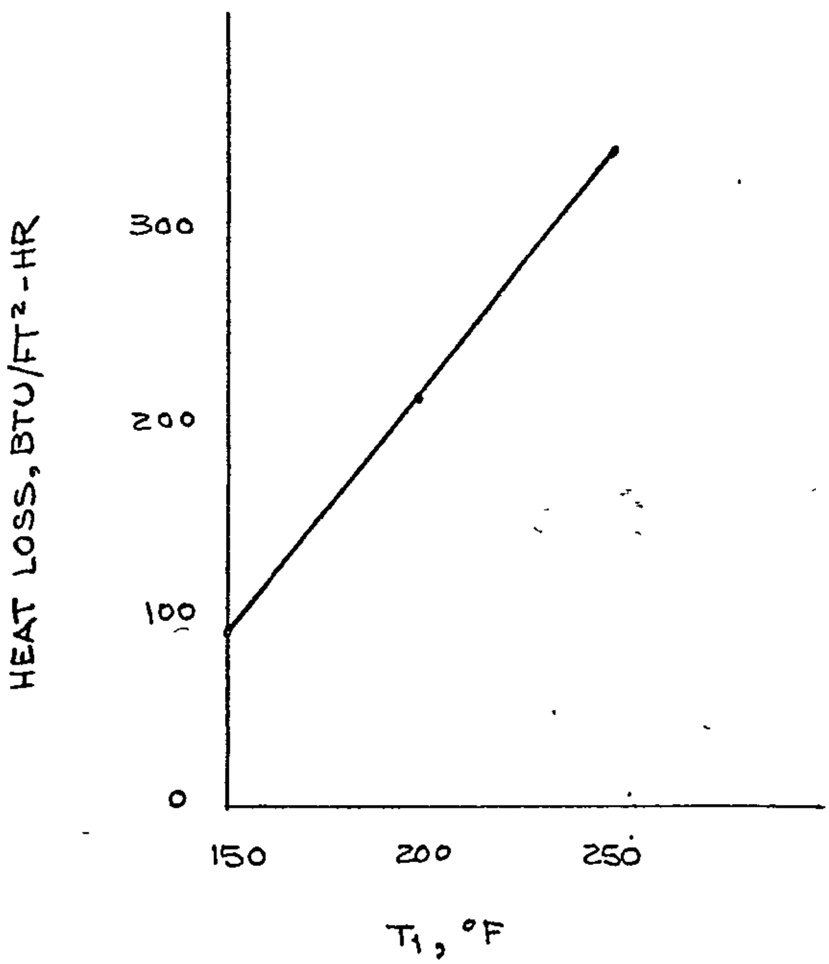


FIGURE 1, FRONT COVER HEAT LOSS

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FERN ENGINEERING
BUZZARDS BAY, MASSACHUSETTS
U.S.A.

Document No : 3001-2

CONTRACT: NASA 8-32246

Job No. 198

SUBJECT: Collector Wind Loads (Item 3.12)

Date: December 2, 1976

AUTHOR: P. Levine

1. Scope

   This document is intended to partially fulfill the verification requirement No. 3.1.2 service loads.

2. MPS Requirements (601 - 6.1)

   Minimum design wind pressure = 20 PSF

   Roofs : 1.25 times design wind pressure

   (Limited to 30 ft height) - outwards

3. Ultimate load combinations (Item 3.2.1)

   Factor of 1.7.

4. Design Loads Criteria

   Inwards 1.7 x 20 = 34 PSF Ultimate

   Outwards 1.7 x 25 = 42.5 PSF Ultimate

5. Underwriters Laboratory Classifications (U L 580)

Class	Uplift Pressure - PSF
30	8.1 - 27.7
60	16.2 - 55.4
90	24.2 - 56.5

CONTRACT: NASA 8-32246

SUBJECT: Plastic Cover Wind Load Capacity

AUTHOR: P. Levine

Document No: 3002- 2

Job No: 198

Date: December 1, 1976

1. Scope

This document is intended to partially fulfill the verification requirement 3.2 "Failure Loads and Load Capacity".

2. Wind Load

The Wind Loads in Document No. 001 indicate a load of 56.5 PSF inwards and outwards to meet UL Class 90 condition.

3. Material Properties

Collector cover is plastic of fiberglass reinforced polyester. Vendor data attached.

Degradation of properties due to UV and thermal effects negligible.

Cyclic fatigue effect limits flexural strength and available data attached.

Wood stiffeners are a candidate for the two-cover construction.

The "Timber Construction Manual", John Wiley & Sons, 1974 recommends the allowable unit stresses contained in the attached tables. Douglas Fir Larch No. 2, with repetitive member use has an allowable of 1650 Psi.

The use of aluminum and steel are also being considered as alternate approaches although they have high thermal conductivity. Comparative conductivity data are:

Material	K, Btu/hr-ft-°F
Wood (Fir)	.062
F R P	0.10
A L	118
Steel (Carbon)	26
Steel (SS)	8 - 10

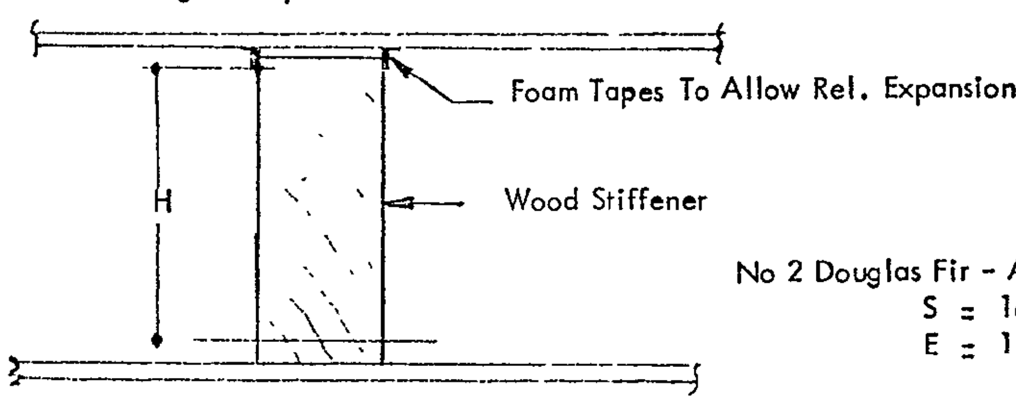
A low thermal conductivity is desirable to minimize front-side collector losses. Typical operating conditions indicate a 72°F temperature drop and heat loss of 88 Btu/hrft^2 at a solar insolation of 300 Btu/ft^2 hr and ambient temperature of 10°F. For a gap of 1",

$$K_{eq} = \frac{88}{72}\ \frac{1}{12} = .1$$

Hence wood and FRP materials have less or equal conductivity than that due to free convection and radiation between the covers for this case.

4. Wood Stiffener Analysis

The wood stiffener geometry is sketched below :



No 2 Douglas Fir - Allowables :
S = 1650 psi
E = 1.7 x 10^6

The flexure equation is,

$$S = \frac{Mc}{I}$$

The cover dimensions are 48" X 96", 48" wood spacers on 12" centers are used; the edges are lightly restrained and a simple support condition is assumed; assuming 8 wood stiffeners carry the total load, then:

$$M = \frac{56.5 \times 4 \times 8}{8} \times \frac{48}{8} = 1356 \text{ in-lb/ stiffener}$$

$$\frac{I}{C} = \frac{1}{12} \frac{bH^3}{H/2} = \frac{1356}{1650}$$

$$b H^2 = 4.93$$

b"	H"	% Blockage
.25	4.44	2
.5	3.14	4
1	2.22	8
.75	2.56	6.25

Parametric results for 8 stiffeners are shown in Figure 1 .

The simple beam results are very conservative and at odds with installation that survived 1-2 years along the coast, subjected to high winds.

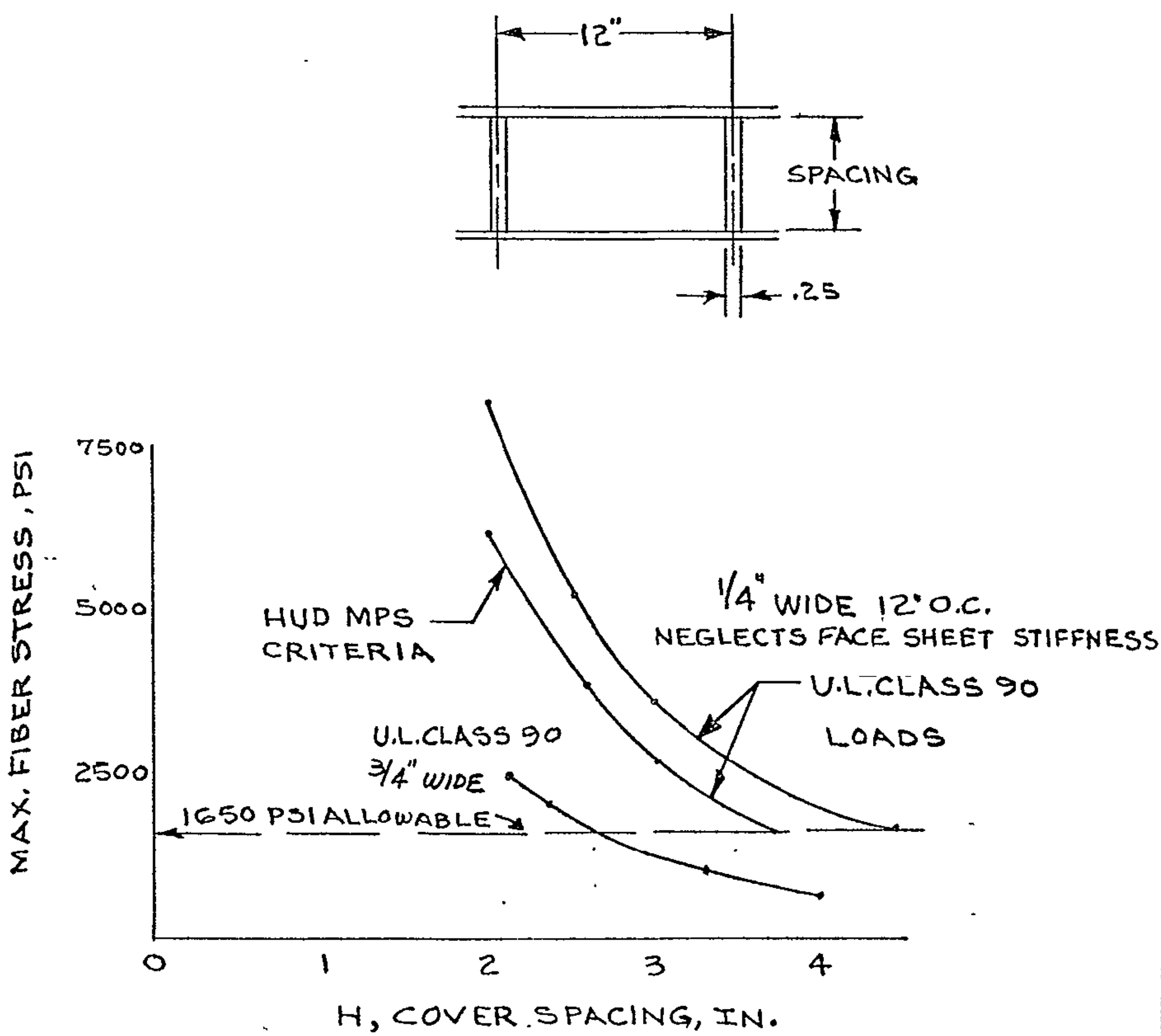
The collector cover is clamped to the collector frame via the side panels. On the basis of a clamped-end- condition,

$$b H^2 = 4.93 / 1.5 = 3.28$$

and the following results hold:

b	H
.25	3.62
.5	2.56
.75	2.09
1.0	1.81

Although the strength requirements are eased by the clamped condition, the results are still higher than what would be in-line with experience.



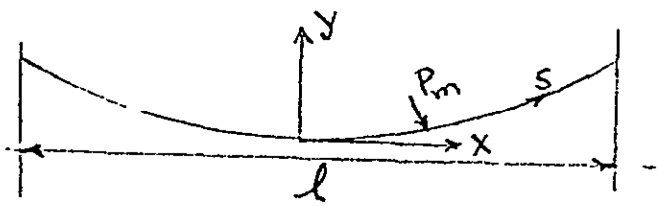
FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS **U.S.A.**	DRAWN / APP'D / DATE	WOOD STIFFENER TRADE-OFFS	
		DWG NO. FIGURE 1	REV.

A further structural model consideration, for use with small values of H, is to consider large deflections, and load sharing by the plastic cover and the stiffeners. The bond between the stiffeners and the cover is assumed to be soft, but with adequate tack strength to cause the stiffener to deflect with the cover. For a simply supported, uniformly loaded beams,

$$P_B = \frac{Y_m \,(384)\, EI}{5\,\ell^4 L}$$

L= length of cover supported per stiffener = 12"

For a section of the cover



Force Balance

$$S_m t \frac{dy}{dx} = P_m x$$

And

$$y_m = \frac{P_m (\ell/2)^2}{2 S_m t_c}$$

$$\epsilon = \Delta\ell/\ell/2$$

$$\Delta\ell = \int ds - \int dx \doteq \int \frac{1}{2}\left(\frac{dy}{dx}\right)^2 dx = \frac{1}{48}\left(\frac{P_m}{S_m t_c}\right)^2 \ell^3$$

Or since

$$S_m = \frac{E_m \epsilon}{1-\mu} = \frac{E_m}{(1-\mu)} \frac{1}{24} \ell^2 \left(\frac{P_m}{S_m t_c}\right)^2$$

$$P_m = y_m^3 \frac{E_m}{(1-\mu)} \frac{t_c}{\ell^4} \frac{64}{3}$$

Combining Loads,

$$P = P_B + P_m = \frac{y_m^3 E_m t_c}{(1-\mu)\ell^4} \frac{64}{3} + \frac{y_m \, 384 \, E_B I}{5 \ell^4 L}$$

Since $E_m = 10^6$, $E_B = 1.7 \times 10^6$

For $H = .75$, $b = .25$ $I \approx .0088$ in^4

$P = .39$ PSI , $\ell = 48"$, $t_c = .040"$ $L = 12"$

$$.39 = Y_m^3 \; (.229) + Y_m \; (.014)$$

$$Y_m = 1.18"$$

The beam carries the pressure

$$P_B = .0148 \text{ psi}$$

The stiffener load is .0148 PSI X 48 X 12 = 8.5 lbs.

$$S = \frac{8.5}{8} \times \frac{48}{.0088} \times \frac{.75}{2} = 2390 \text{ PSI}$$

The cover lifting stress in the tape is

$$S_t = \frac{8.5}{48 \times .25} = .7 \text{ PSI}$$

The stiffener stresses exceed the allowable stresses . The membrane stress is,

$$S = \frac{.39 \times \overline{24}^2}{2 \times 1.2 \times .04} = 2340 \text{ PSI}$$

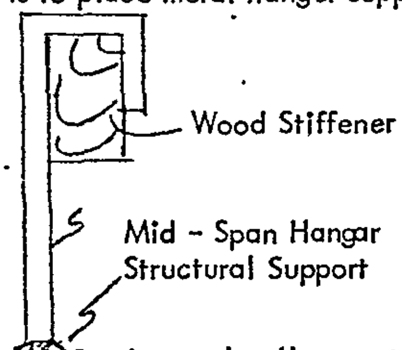
and is well within the allowable.

The wood stiffener, using small cross-section and relying on large deflections would require proof testing. Other approaches are to reduce the stiffener span and to introduce curvature into the cover.

The 48" span collector with wood stiffeners, using 1" board stock (.75")

yields a spacing of 2.56" and 6.25% blockage.

An alternate arrangement is to place metal hanger supports at mid-span, e.g.



The added blockage is small. But internal collector structure must be provided to carry the load. The moment is reduced by a factor of four and hence the thickness requirements as well.

Hence for ,

$b = .25"$ $\quad\quad H = 2.22"$

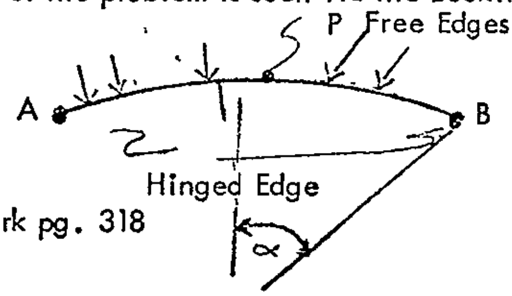
Important consideration is tape adhesion strength. As the collector cover cells are vented internally, the outer cover tape must transfer the load. For 1/4" wide stiffeners ,

$$S = \frac{56.5 \times 32}{8 \times 48 \times .25} = 18.8 \text{ psi}$$

A tape system , as 3 M 4282, a high strength neoprene foam tape is a good candidate, A high strength silicone adhesive is also a candidate.

The introduction of curvature into the external cover to take advantage of membrane stresses introduces the prospects of buckling when the air-load is reduced.

The criticality of the problem is seen via the buckling stress criteria for a curved plate,



Roark pg. 318

$$P_{cr} = \frac{Et^3\,(K^2-1)}{12r^3\,(1-\nu^2)} \qquad K = f(\alpha)$$

For $\alpha = 15^{\circ}$ $K = 17.2$, $r = 96''$, $E = 10^6$, $t = .04''$

$$P_{cr} = .002 \text{ psi}$$

Improvements by stiffeners to reduce the length and introde plate action will help, but the approach does not look attractive. Concave surfaces will be subject to outward loading, and collect snow.

In summary, the structural approach to the cover involves stronger stiffeners and better adhesives. The use of a FRP stiffener and high strength bond system may be an attractive approach.

Taking stock of the wood stiffener approach, we find :

1) The 48" span with a reasonable cover spacing of 2.25" results in an 8% blockage factor. The large gap is favorable thermally as the free convection is reduced (see ref. 4).

2) The mid-span tie-downs reduce the blockage to 2% for a 2.25" gap ; however collector construction is complicated.

3) Curvature in the cover could result in buckling under wind loads.

As plastic cover material is available in larger widths, to 5 feet, the blockage effect may be compensated by making the collector even wider.

An optimization of net collector area follows:

$$M = \frac{P.L.l.l.}{8.8} \, , \qquad P = \text{load, psi}$$

$L = 96"$, length

$l = \text{Span}$

8 Stiffeners assumed

$$\frac{I}{c} = \frac{M}{s} \qquad H = 2.25"$$

$$\frac{\text{Net Area}}{L} = l - 8\,bl = l - \frac{3}{4} \cdot \frac{l^3 P}{H^2 S} = S - \frac{S^3}{27801}$$

and $\frac{\text{Net Area}}{}$ is max at $l = 96"$. For $l = 60"$, the resulting net area is increased 10% over the $l = 48"$ collector, although $b = 1.5"$.

With increased span, the deflection increases sharply, as

$$y_m = \frac{5\, l^4 L P}{384\, E I}$$

For
$l = 60"$
$P = 56.5/144 = .39$ psi
$E = 1.3 \times 10^6$ psi
$L = 12"$ / per stiffener
$I = \frac{1}{12} \times 1.5 \times 2.25^3 = 1.42$

$y_m = .33$ inches

The weight of the wood is,

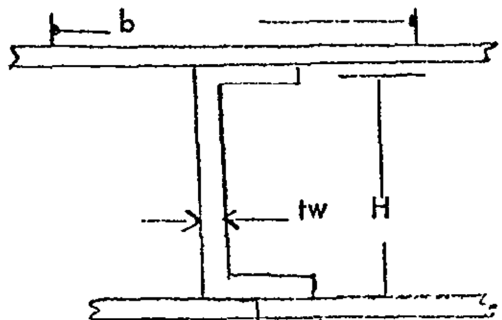
$$\frac{2.25 \times 1.5 \times 60 \times 36 \times 8}{1728} = 34 \text{ lbs.}$$

Various other techniques might also be used as using a composite metal/wood stiffener or FRP stiffener.

The 60" wide collector is attractive as its construction is relatively simple, and it is rugged. The wide surfaces provided by the stiffeners for tape adhesion, ease the adhèsive strength requirements also. The cost of the collector will likely increase about 10% ; non-standard widths, 5 ft vs 4 ft could lead to added costs.

5. <u>FRP Analysis</u>

The stiffeners are designed to closely match the thermal expansion coefficient of the cover material , 14×10^{6} in/in ^{o}F. A clear polyester resin is used so the resultant stiffener is translucent, in which case the blockage effect is offset by its translucent qualities. A stiffener configuration is shown below :



The stiffener is bonded to the covers, adding a fixity effect of the cover. As the cover moves with the stiffener, the effective width of the cover working with the stiffener is suggested by Perry " Aircraft Structures"

$$b = 1.45\, t \sqrt{\frac{E}{S}}$$

Cyclic fatigue could limit the stress to 4400 psi ; for $E = 10^{6}$, $t = .040"$,

$b = .874"$, $b/t = 21.85$

The moment of inertia is ,

$$I = 2\,b\,t_c \left(\frac{H}{2}\right)^2 + \frac{1}{12}\,tw\,(H - 2\,tw)^3 + 2\,L.tw\left(\frac{H}{2} - \frac{tw}{2}\right)^2$$

The effective width "b" depends on the stress in the covers. On the basis of 8 stiffeners on 12" centers.

Case 1. $\left(\frac{I}{c}\right)_{req} = \frac{1356}{S}$

L = .25

S = 10,000 psi

Results are shown in Fig. 2.

Case 2.

L = .5

S = 10,000 psi

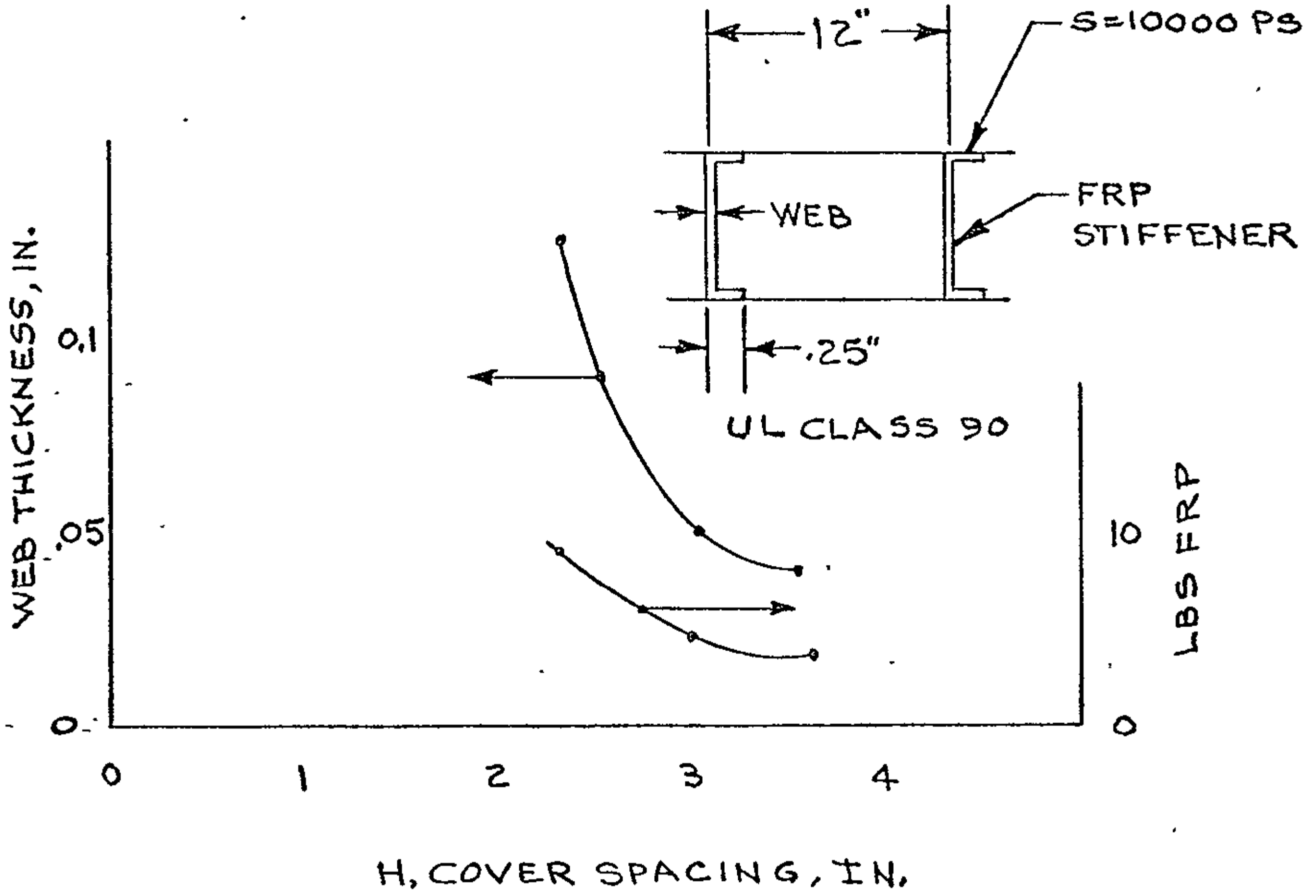
H	Tw	Weight, lbs
2"	.1"	5:0
2.5"	.05"	3:0

Case 3.

L = .5

S = 4400 psi (Strength Reduction due to fatigue)

H	Tw	Weight, lbs
2.5	.125	8.5



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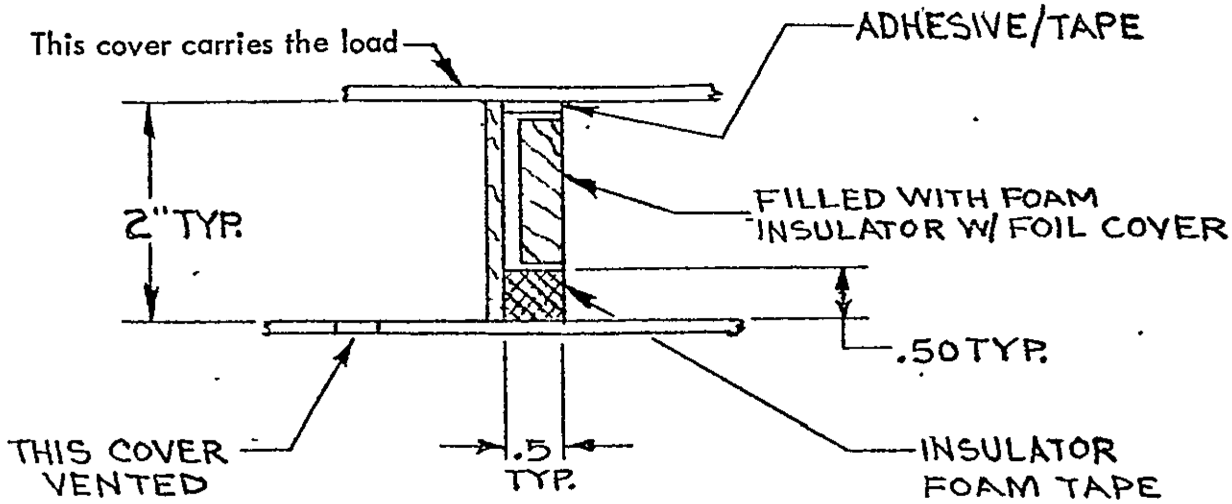
	DRAWN		
FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	APP'D	FRP STIFFENER TRADE-OFFS	
	DATE	DWG NO. FIGURE 2	REV.

6. Aluminum Stiffeners

The use of aluminum stiffeners alone introduces the problem of thermally short-circuiting the two covers ; however , an aluminum insulator composite is an alternative.



For 1/8" thickness,

$\frac{I}{c}$ = .15 in^4 ,

M = 135 6

S = 9040 psi , Ym = .215"

The stiffeners of 6061 - T6 could be made as thin as .060" , but the deflection would increase. The inner cover is softly restrained so it will easily follow the deflection without excessive stresses in the insulator.

7. Steel Stiffeners

The higher modulus for steel helps to reduce the deflections. A stainless steel channel, .062" thick, 2" depth and 0.5" flange has an I = .092.

$$S = \frac{1356 \times 1}{.092} = 14{,}739$$

$$\text{Deflection } Y_m = \frac{5 \times 48^4 \times 12 \times .39}{384 \times 28 \times 10^6 \times .092} = .1255"$$

The conductivity of 9 Btu/ft-hr is favorable as compared with aluminum. The thermal expansion coefficient does not match the cover material as well as aluminum, but the match is better than wood.

8. Preliminary Design Selection Alternatives

H = 2.5"	Flange	WEB	Deflection
Full Span FRP	.5"	.125"	.81"
Full Span Wood	.0"	.75"	.20"
Full Span Al + insul.	.5"	.063	.3514
Full Span S.S. + insul.	.5"	.063	.1255"

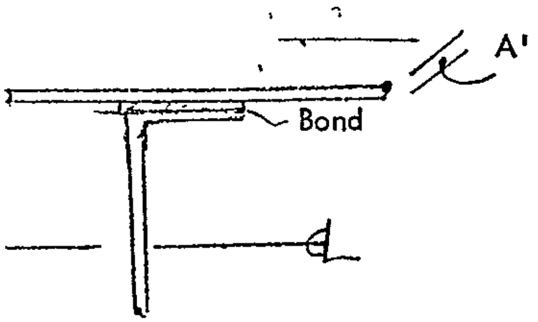
Tape or adhesive alternatives :

3 M neoprene foam rubber, 50 psi tension

Mod. Epoxy

Silicone Adhesives

Shear stress in the bond : if it transfers all shear, FRP ,



.5" Flange

H = 2.5"

I = .4

Effective Width

$$S_s = \frac{V A' \bar{Z}'}{I b} = \frac{1808}{2 \times 8} \times \frac{.04\,(.87)}{.5}\;\frac{1.25}{.4} = 25 \text{ psi}$$

Superimposed on this stress is the effect of thermal expansion differences ;

FRP and cover have thermal expansion coefficients that match closely.

ENGINEERING PROPERTIES OF LAMINATES*

Fatigue

When certain materials are subject to cyclic or repeated loads, fatigue failure can occur at stresses below the ultimate static strength. Fatigue failure can be due to the reversal of a tensile, flexural or torsional stress in a member. Reversal of stress can occur with or without an initial or "mean" stress. If a mean stress is present in a system that undergoes stress reversals, the stress at failure is further reduced. In other words, when a member is preloaded and then subjected to stress reversals, the fatigue strength of the member is less than the strength of the same member without preload. Fatigue failures are usually propagated by cracks in high tensile areas and can be accelerated by initial cracks, flaws, discontinuities, holes, notches, etc.

Fiberglass laminates are subject to fatigue failures. However, because of the nature of their composition, it is difficult to pinpoint what type of failure actually predominates, that is, tensile, shear or delamination.

Figs. 5-7 and 5-8 present SN curves or fatigue strengths as per cent of ultimate strengths for mat-polyester and 181-136 cloth-polyester laminates in the dry condition, tested parallel to warp and at 73 degrees F and 50 per cent relative humidity. Similar data for 10 ounce cloth and woven roving is presently unavailable. These curves indicated that both tensile and flexural fatigue strengths of fiberglass polyester laminates tend to level off at approximately 20 to 30 per cent of their ultimate strengths at 10 million cycles (9-16). This stress level defines the fatigue limit or the value at which the material can undergo stress reversals for an indefinite period.

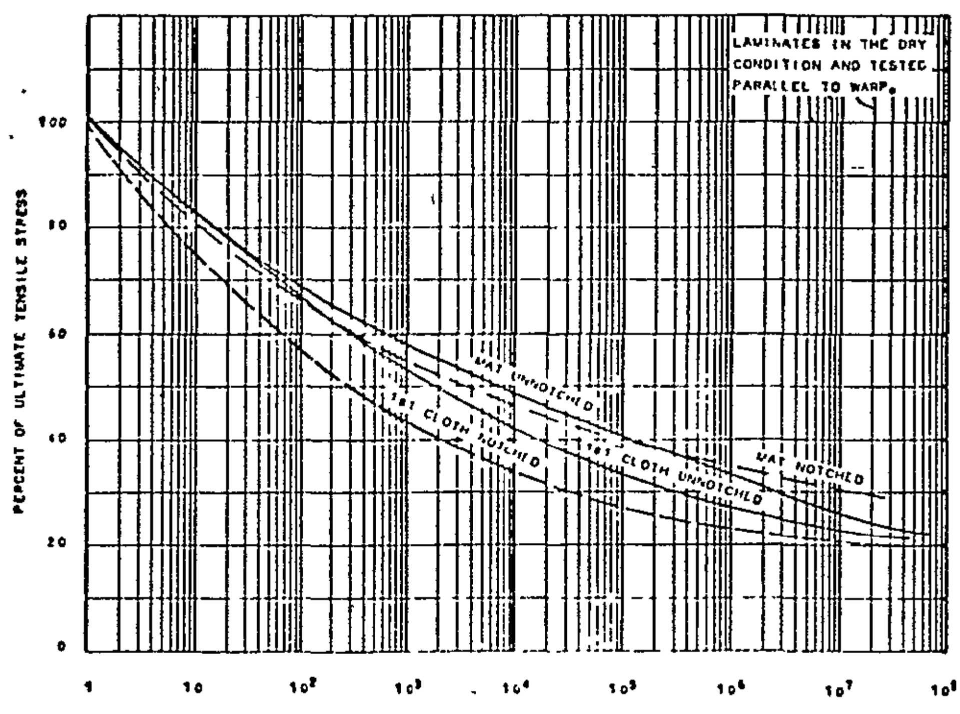


Fig. 5-7. Tensile Fatigue Strength of Fiberglass Polyester Laminates

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* "Marine Design Manual For FRP", McGraw-Hill.

TABLE 2.14 (Continued)

Species and Commercial Grade	Size Classification	Allowable Unit Stresses, psi: Extreme Fiber in Bending, F_b: Single-Member Uses	Extreme Fiber in Bending, F_b: Repetitive-Member Uses	Tension Parallel to Grain F_t	Horizontal Shear F_v	Compression Perpendicular to Grain $F_{c\perp}$	Compression Parallel to Grain F_c	Modulus of Elasticity E	Grading Rules Agency
Douglas fir larch (surfaced dry or surfaced green Used at 19% max. MC)									
Dense select structural	2 to 4 in thick 2 to 4 in. wide	2,450	2,800	1,400	95	455	1,850	1,900,000	West Coast Lumber Inspection Bureau and Western Wood Products Association (see footnotes [a] through [j])
Select structural		2,100	2,400	1,200	95	385	1,600	1,800,000	
Dense No 1		2,050	2,400	1,200	95	455	1,450	1,900,000	
No 1		1,750	2,050	1,050	95	385	1,250	1,800,000	
Dense No 2		1,700	1,950	1,000	95	455	1,150	1,700,000	
No 2		1,450	1,650	850	95	385	1,000	1,700,000	
No 3		800	925	475	95	385	600	1,500,000	
Appearance		1,750	2,050	1,050	95	385	1,500	1,800,000	
Stud		800	925	475	95	385	600	1,500,000	
Construction	2 to 4 in thick 4 in. wide	1,050	1,200	625	95	385	1,150	1,500,000	
Standard		600	675	350	95	385	925	1,500,000	
Utility		275	325	175	95	385	600	1,500,000	
Dense select structural	2 to 4 in thick 6 in. and wider	2,100	2,400	1,400	95	455	1,650	1,900,000	
Select structural		1,800	2,050	1,200	95	385	1,400	1,800,000	
Dense No. 1		1,800	2,050	1,200	95	455	1,450	1,900,000	
No 1		1,500	1,750	1,000	95	385	1,250	1,800,000	
Dense No. 2		1,450	1,700	950	95	455	1,250	1,700,000	
No 2		1,250	1,450	825	95	385	1,050	1,700,000	
No 3		725	850	475	95	385	675	1,500,000	
Appearance		1,500	1,750	1,000	95	385	1,500	1,800,000	

Table 1 - Glass and Sun-Lite Fiberglass Compared to Ideal

	Ideal	3/16" Glass	Sun-Lite
Transmittance, solar, ASTM E424	1.00	.83 to .87	.85 to .90
Transmittance, heat (5 $_u$M to 50 $_u$M)	0.00	.1	.1
Thermal Conductivity, (BTU-in/hr-ft^2-^{o}F)	0.00	6 to 7	.87
Thermal Expansion Coef., PPM*/^{o}F	12**	4 to 5	14
Impact Strength, SPI ball drop, ft-lbs	high	10	60
Ultimate Tensile Strength, ASTM D638, psi	high	10,000	16,000
Weight per sq ft (.040" Sun-Lite), lbs	low	2.4	0.3
Resistance to chemicals, UV & time	excellent	excellent	very good
Handling & Cutting	safe, easy	difficult	safe, easy
Cost (in quantity from distributor), \$/ft^2	low	.52	.28 to .46

* Parts per million ** To match aluminum used in most collectors.

Figure 1 Spectral Transmittance, Sun-Lite Fiberglass Sheet

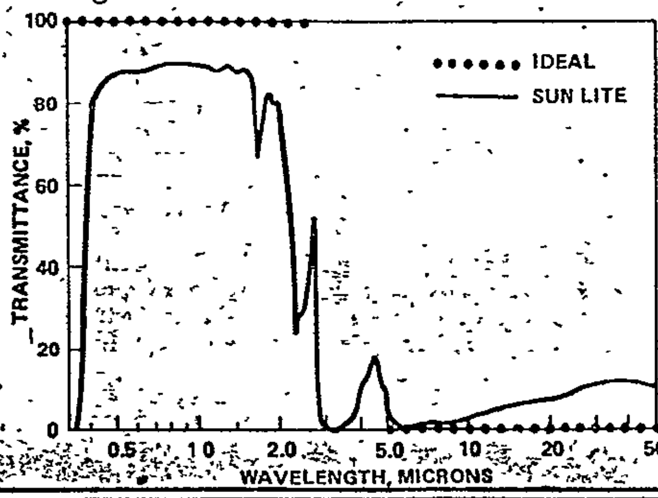
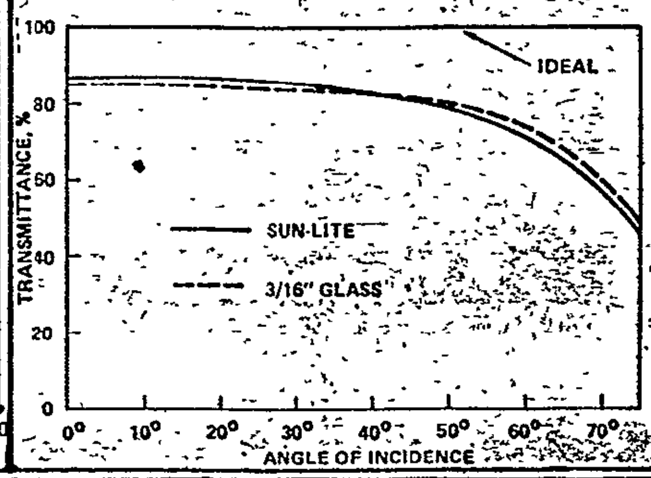


Figure 2 Solar Transmittance Versus Angle of Sun-Lite Fiberglass Sheet Incidence



SUN-LITE SOLAR COLLECTOR COVER MATERIAL

AVERAGE PHYSICAL PROPERTIES	METHOD	UNITS	SUN-LITE REGULAR	SUN-LITE PREMIUM
Solar Energy Transmittance	E 424 Method B	%	85%-90%	85%-90%
Estimated Solar Lifetime(1)		Years	7	20
Thermal Sensitivity(2)	@	200^{o}F	Excellent	Excellent
	@	300^{o}F	Poor	Good
Heat Transmittance	5-20 Microns	%	10%	10%
Index of Refraction	D 542	Ratio	1.54	1.52
Tensile Strength	D 638	PSI	16,000	11,000
Flexural Strength	D 790	PSI	24,500	22,000
Flexural Modulus	D 790	PSI x 10^6	1.0	0.6
Shear Strength	D 732	PSI	14,000	12,000
Izod Impact	D 256	Ft.lb./In.	18	10
Water Absorption	D 570	%	0.20-0.33	0.50-0.60
Thermal Expansion	D 696	(In./In./^{o}F) x 10^{-5}	1.4	1.4
Thermal Conductivity	C 177	BTU-In./Hr./Ft.^{2}/^{o}F	.87	.87
Specific Heat	D 2766	BTU/lb./^{o}F	.35	.35
Specific Gravity	D 792	Ratio	1.4	1.4
Weight	NBS PS53	Oz./Ft.^{2}	2.8-4.7	2.8-4.7
Thickness	NBS PS53	Inches	.025 or .040	.025 or .040
Sheet Size	NBS PS53	Feet	4' or 5' wide, up to 1,200' long	

The above information is presented in good faith and believed to be correct to the best of our knowledge, but no warranty is expressed or implied.

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00121-7

CONTRACT: NASA 8-32246 DOCUMENT NO. 5003-

SUBJECT: Collector Materials JOB 198

AUTHOR: P. Levine DATE: 12/7/76

There are two types of collectors being considered; a wood-frame plastic cover design and an aluminum frame glass cover design.

1) Wood-Frame Plastic Cover (Dwg. No. SK-198-8)

Wood frame Western Douglas Fir-Larch KD-2

Plywood rear cover, exterior A/C grade

Fiberglass insulation, Certainteed K-242

Isocyanurate TRYMER CPR 9545 (Upjohn)

"Nextel" Brand Velvet Coating Series 101-C10

Precoated Al. foil by 3M Company

Dow Corning 781 Sealant

No. 4282 Neoprene rubber foamtape (3M Co.)

3003-H14 Aluminum sheet

Kalwall Sunlite Premium FRP Covers

Weatherban brand sealant tape, 3M Co. 1202-T

2. Aluminum Frame & Glass Cover

Aluminum frame, 6063-T5 Mill finish

Al rear enclosure & ducts 3003-H14 sheet

Fiberglass insulation Certainteed K-242

Isocyanurate TRYMER CPR 9545 (Upjohn)

"Nextel" Brand Velvet Coating Series 101-C10

Precoated Al. foil by 3M Company

CONTRACT:	NASA 8-32246	DOCUMENT NO.	5002-2
SUBJECT:	Collector Materials	JOB	198
AUTHOR:	P. Levine	DATE:	12/7/76

Dow Corning 781 Sealant

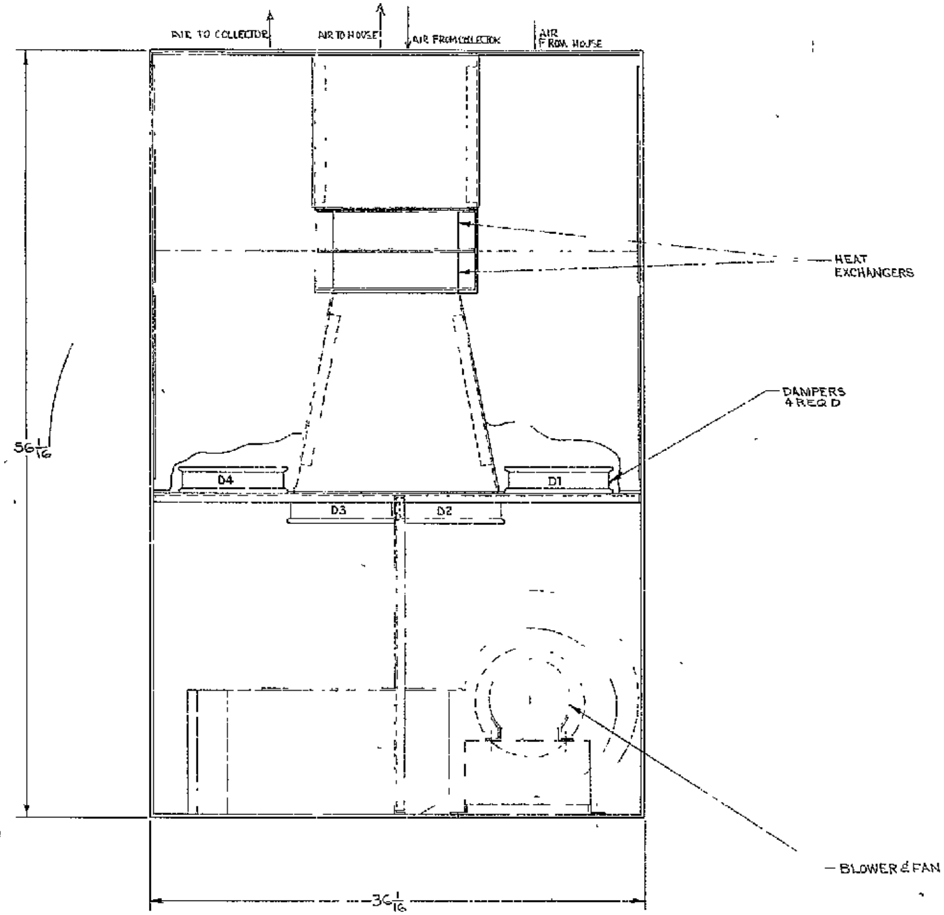
Kalwall Sunlite Premium FRP Cover

ASG tempered glass " Water White" Cover

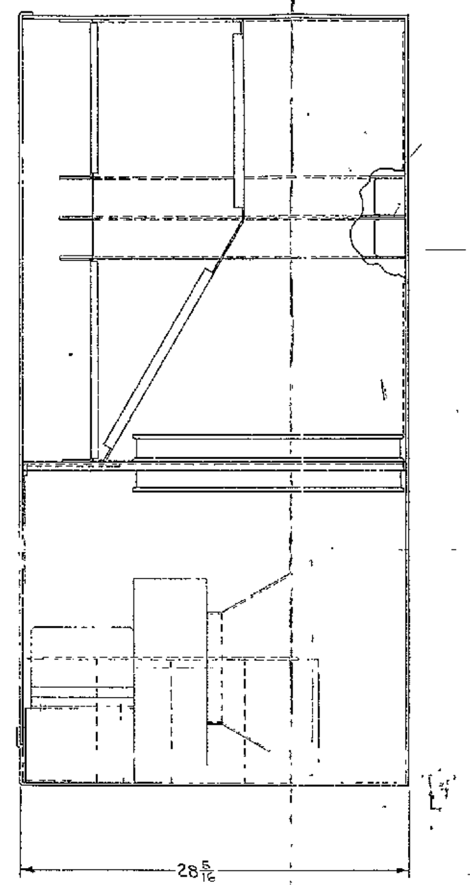
EPDM Glazing Gasket

APPENDIX A

PRELIMINARY DESIGN DRAWINGS



VIEW SHOWN WITH COVER REMOVED



OUTSIDE SKIN NOT SHOWN IN THIS VIEW

FOLDOUT FRAME 1

FOLDOUT FRAME 2

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NOTES:

1. INTERNAL SHEET METAL PARTS ARE 24GA COLD ROLLED STEEL (.024 THICK)
2. OUTSIDE SHEET METAL PARTS ARE 22GA COLD ROLLED STEEL (.030 THICK)

A-1-A

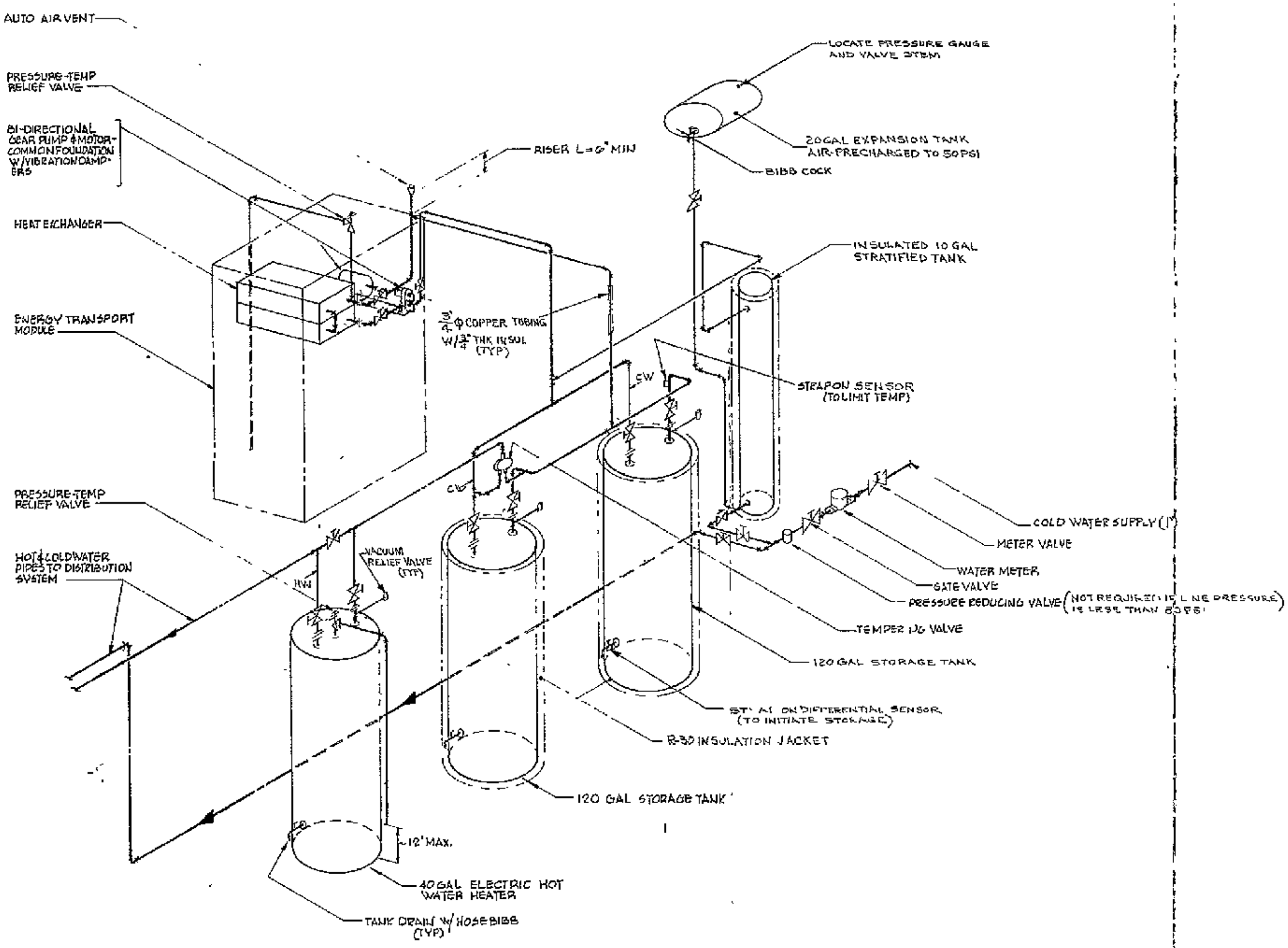
A-1-B

FERN ENGINEERING BUZZARDS BAY, MASSACHUSETTS U.S.A.	ENERGY TRANSPORT MODULE
DATE 11-25	SCALE 1/4
DRAWING NO.	SK-198-6

A-1

FOLDOUT FRAME 1

FOLDOUT FRAME 2



A	8/20/76		REVISED
NO.	DATE	BY	APPR.
	REVISIONS		

DWG. NO. — REFERENCE DRAWINGS

FERN ENGINEERING
BUZZARDS BAY, MASSACHUSETTS
U.S.A.

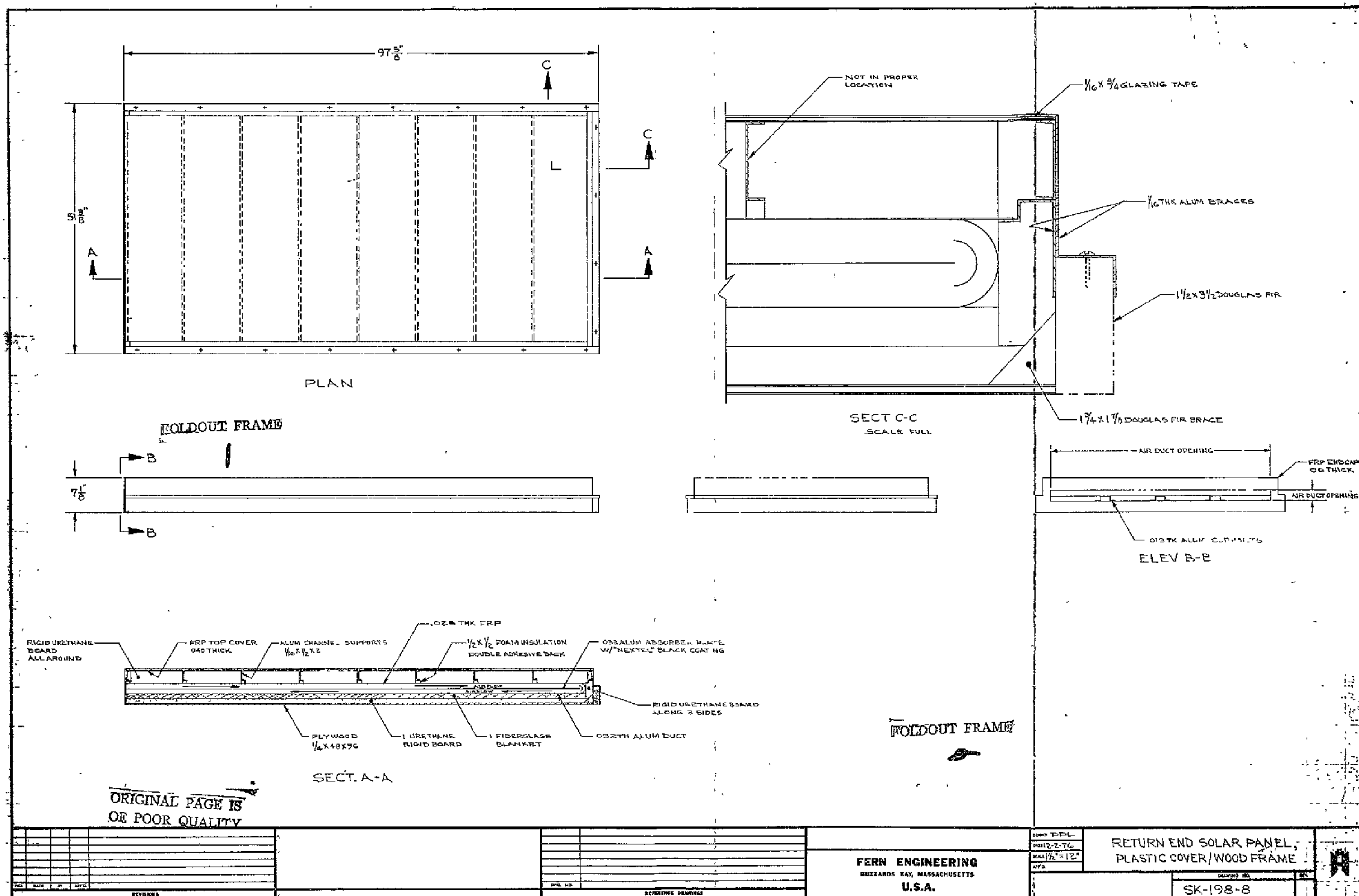
DRAWN J.F. PERRY
DATE 12-7-76
SCALE NONE
APP'D.

SOLAR HEATING SYSTEM
PIPING DIAGRAM

DRAWING NO. SK-198-7 REV. A

A-2-a

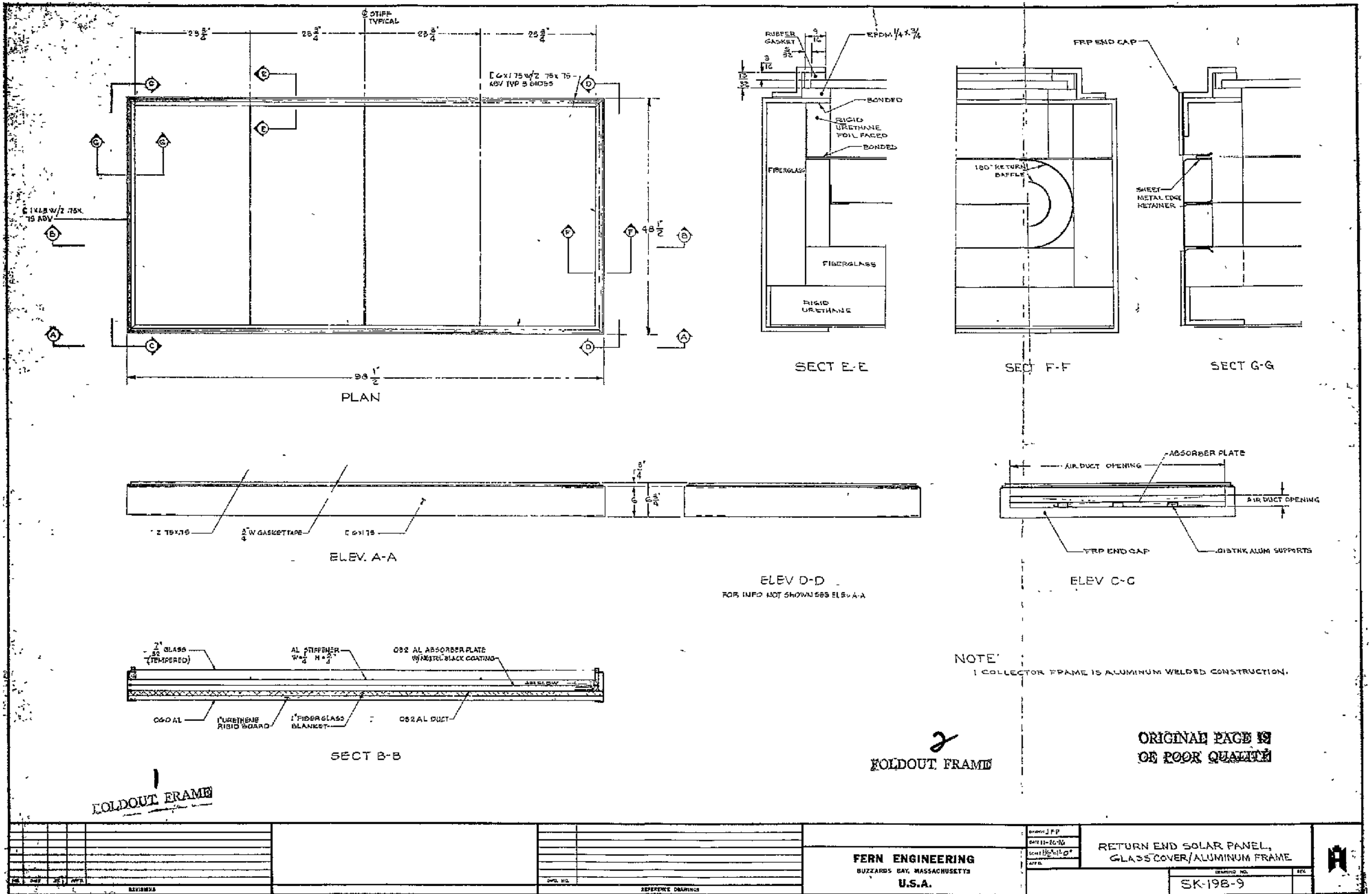
A-2-b



97 5/8"
5 3/8"
PLAN
FOLDOUT FRAME
7 1/8"
NOT IN PROPER LOCATION
1/16 X 3/4 GLAZING TAPE
1/16 THK ALUM BRACES
1 1/2 X 3 1/2 DOUGLAS FIR
1 3/4 X 1 7/8 DOUGLAS FIR BRACE
SECT C-C
SCALE FULL
AIR DUCT OPENING
FRP ENDCAP .06 THICK
ELEV B-B
RIGID URETHANE BOARD ALL AROUND
FRP TOP COVER .040 THICK
ALUM CHANNEL SUPPORTS 1/10 X 1/2 X 2
.025 THK FRP
1/2 X 1/2 FOAM INSULATION DOUBLE ADHESIVE BACK
.032 ALUM ABSORBER PLATE W/"NEXTEL" BLACK COATING
AIR FLOW
RIGID URETHANE BOARD ALONG 3 SIDES
PLYWOOD 1/4 X 48 X 96
1 URETHANE RIGID BOARD
1 FIBERGLASS BLANKET
.032 TH ALUM DUCT
SECT. A-A
FOLDOUT FRAME
ORIGINAL PAGE IS OF POOR QUALITY
REVISIONS
REFERENCE DRAWINGS
FERN ENGINEERING
BUZZARDS BAY, MASSACHUSETTS
U.S.A.
DRAWN DPL
DATE 12-2-76
SCALE 1/2" = 12"
RETURN END SOLAR PANEL, PLASTIC COVER/WOOD FRAME
SK-198-8

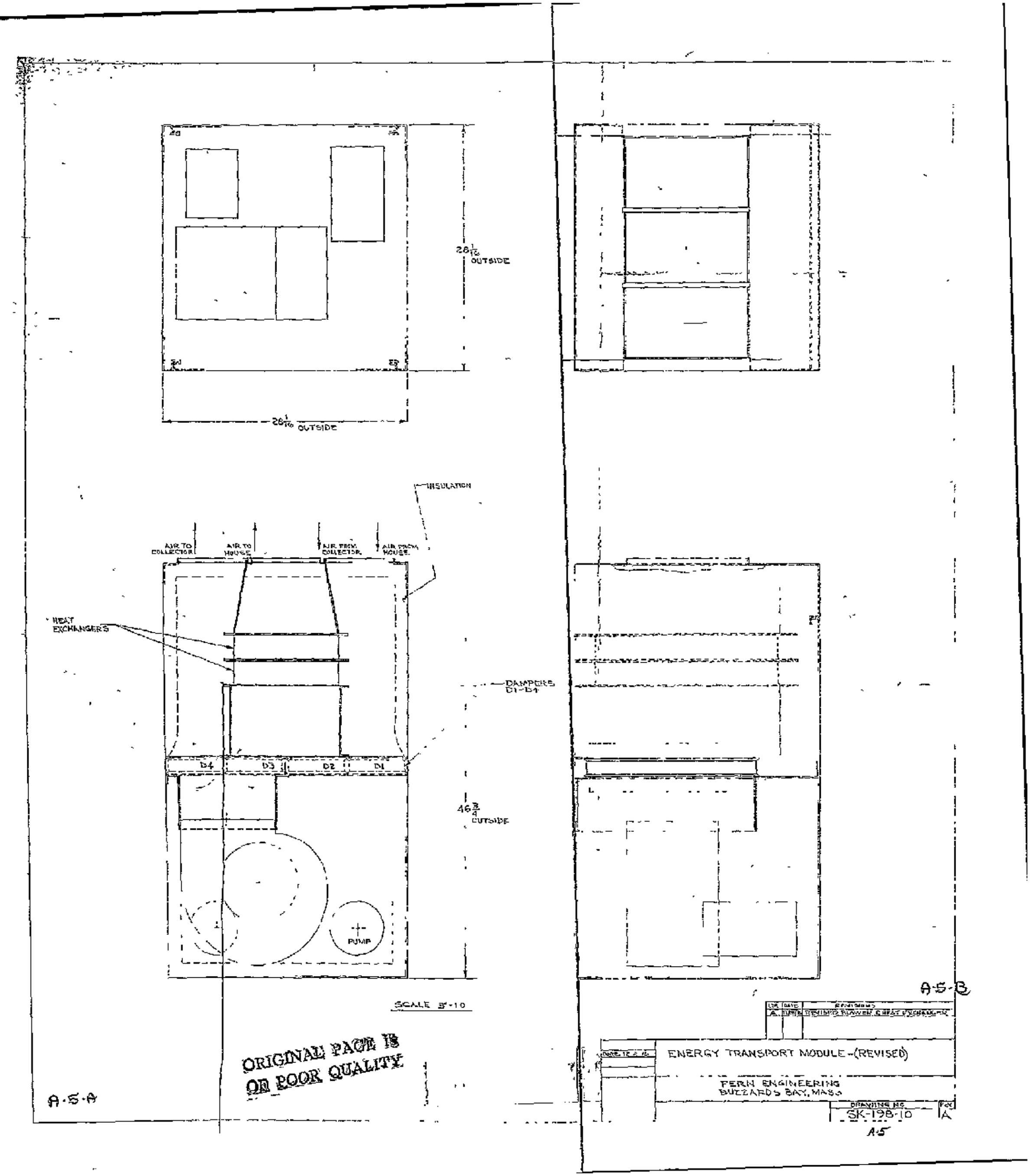
A-3-a

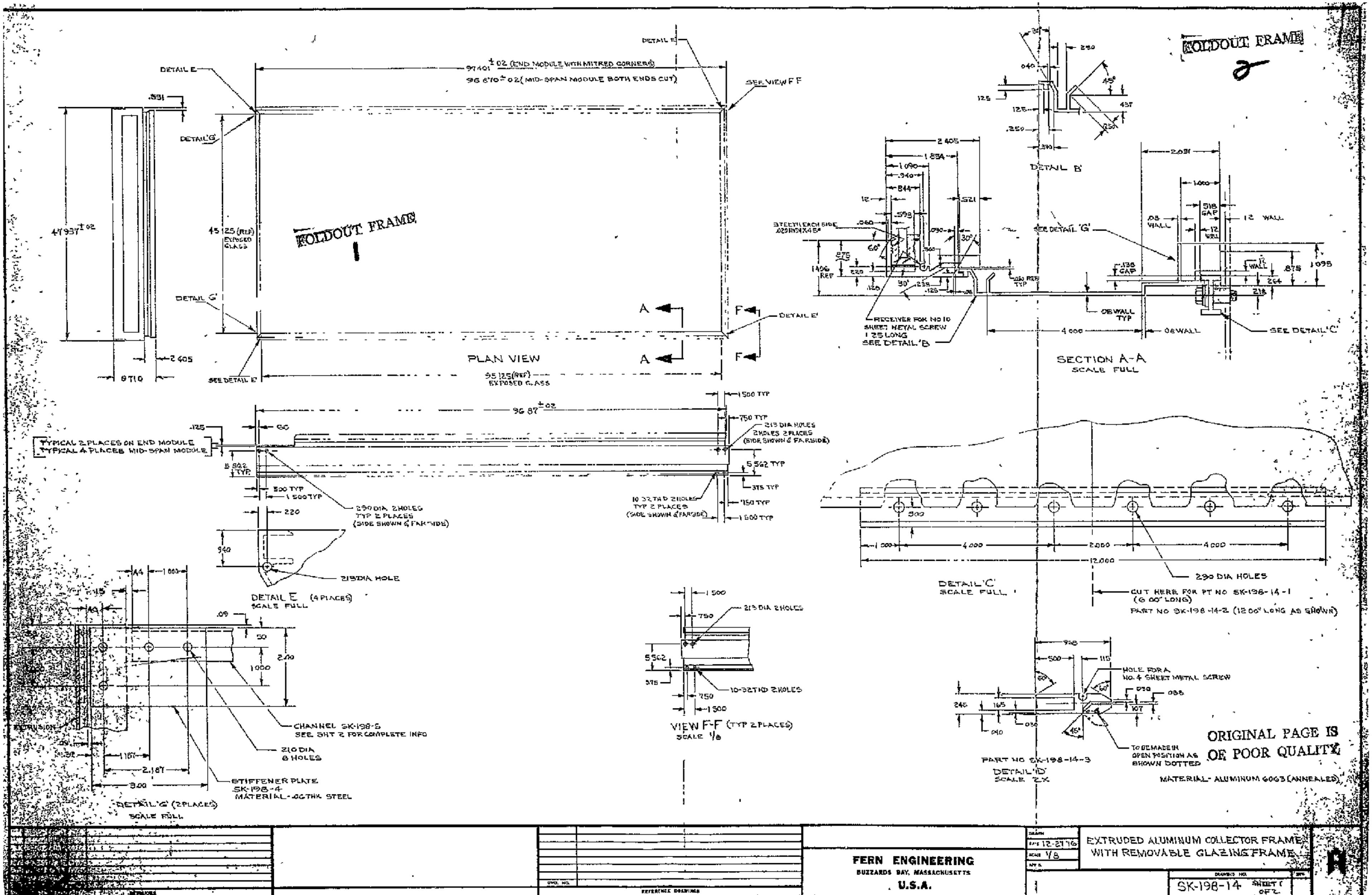
A-3-b

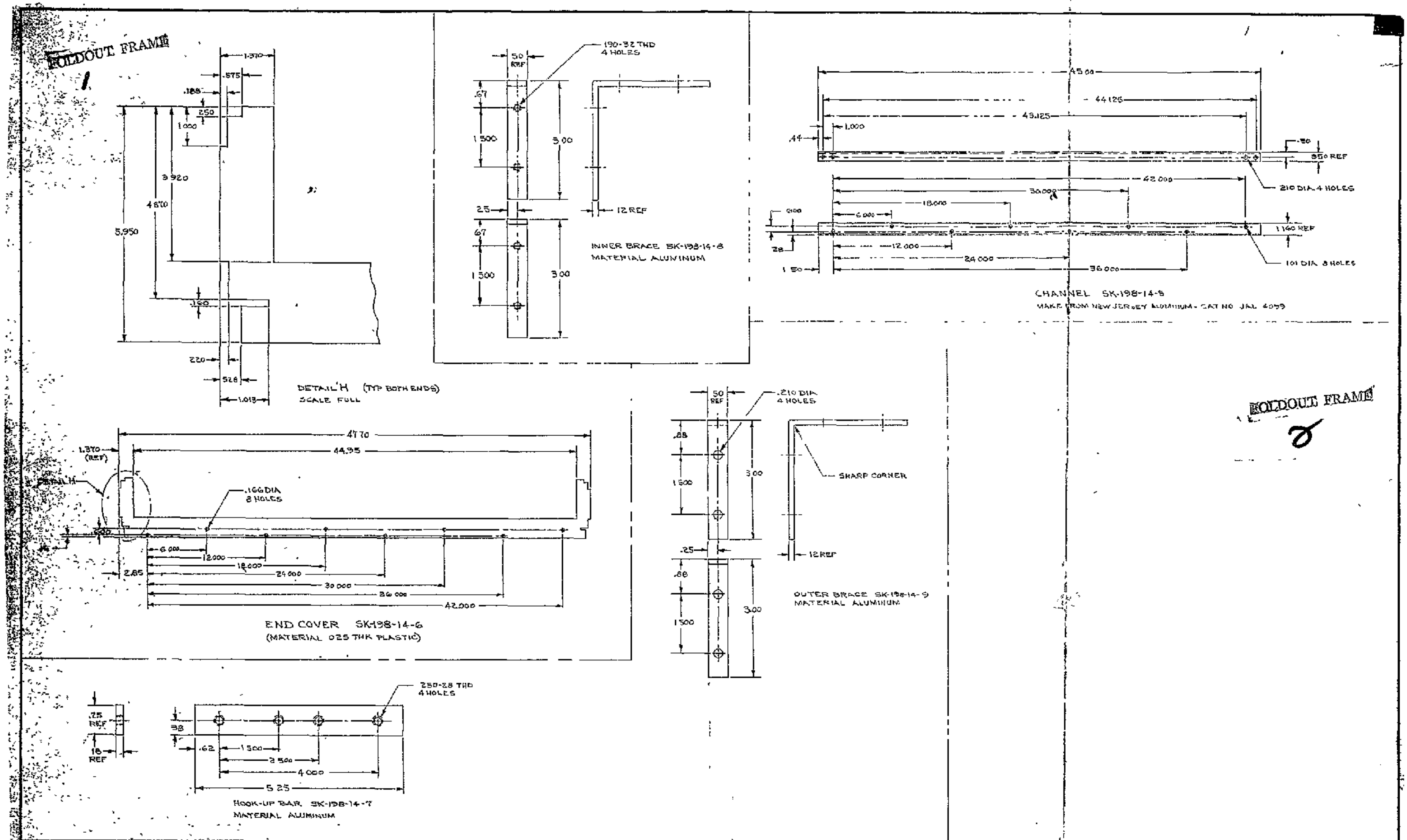

PLAN
SECT E-E
SECT F-F
SECT G-G
ELEV. A-A
ELEV D-D
FOR INFO NOT SHOWN SEE ELEV A-A
ELEV C-C
SECT B-B
RUBBER GASKET
EPDM 1/4 x 3/4
BONDED
RIGID URETHANE FOIL FACED
FIBERGLASS
RIGID URETHANE
FRP END CAP
180° RETURN BAFFLE
SHEET METAL EDGE RETAINER
ABSORBER PLATE
AIR DUCT OPENING
.016 THK ALUM SUPPORTS
3/4" W GASKET TAPE
7/32" GLASS (TEMPERED)
AL STIFFENER
.032 AL ABSORBER PLATE W/ NEXTEL BLACK COATING
.060 AL
1" URETHENE RIGID BOARD
1" FIBERGLASS BLANKET
.032 AL DUCT
NOTE:
1 COLLECTOR FRAME IS ALUMINUM WELDED CONSTRUCTION.
ORIGINAL PAGE IS OF POOR QUALITY
FOLDOUT FRAME 1
FOLDOUT FRAME 2
FERN ENGINEERING
BUZZARDS BAY, MASSACHUSETTS
U.S.A.
RETURN END SOLAR PANEL, GLASS COVER/ALUMINUM FRAME
SK-198-9
REVISIONS
REFERENCE DRAWINGS

A-4-a

A-4-b


28 1/16 OUTSIDE
28 1/16 OUTSIDE
INSULATION
AIR TO COLLECTOR
AIR TO HOUSE
AIR FROM COLLECTOR
AIR FROM HOUSE
HEAT EXCHANGERS
DAMPERS D1-D4
D4
D3
D2
D1
46 3/4 OUTSIDE
PUMP
SCALE 3"=1'0
A-5-A
A-5-B
ENERGY TRANSPORT MODULE-(REVISED)
FERN ENGINEERING
BUZZARDS BAY, MASS.
DRAWING NO.
SK-198-10
A
A-5


FOLDOUT FRAME 1
FOLDOUT FRAME 2
PLAN VIEW
97.401 ± .02 (END MODULE WITH MITRED CORNERS)
96.670 ± .02 (MID-SPAN MODULE BOTH ENDS CUT)
45.125 (REF) EXPOSED GLASS
95.125 (REF) EXPOSED GLASS
SEE VIEW F-F
DETAIL E
DETAIL G
SEE DETAIL E
TYPICAL 2 PLACES ON END MODULE
TYPICAL 4 PLACES MID-SPAN MODULE
.290 DIA. 2 HOLES TYP 2 PLACES (SIDE SHOWN & FAR SIDE)
.213 DIA HOLES 2 HOLES 2 PLACES (SIDE SHOWN & FAR SIDE)
10-32 THD 2 HOLES TYP 2 PLACES (SIDE SHOWN & FAR SIDE)
DETAIL E (4 PLACES) SCALE FULL
.213 DIA HOLE
CHANNEL SK-198-5 SEE SHT 2 FOR COMPLETE INFO
.210 DIA 8 HOLES
STIFFENER PLATE SK-198-4 MATERIAL-.06 THK STEEL
DETAIL G (2 PLACES) SCALE FULL
VIEW F-F (TYP 2 PLACES) SCALE 1/8
.213 DIA 2 HOLES
10-32 THD 2 HOLES
DETAIL B
SECTION A-A SCALE FULL
RECEIVER FOR NO 10 SHEET METAL SCREW 1.25 LONG SEE DETAIL B
SEE DETAIL G
SEE DETAIL C
.08 WALL TYP
.12 WALL
DETAIL C SCALE FULL
.290 DIA HOLES
CUT HERE FOR PT NO SK-198-14-1 (6.00" LONG)
PART NO SK-198-14-2 (12.00" LONG AS SHOWN)
HOLE FOR A NO. 4 SHEET METAL SCREW
TO BE MADE IN OPEN POSITION AS SHOWN DOTTED
PART NO SK-198-14-3
DETAIL D SCALE 2X
MATERIAL- ALUMINUM 6063 (ANNEALED)
ORIGINAL PAGE IS OF POOR QUALITY
FERN ENGINEERING
BUZZARDS BAY, MASSACHUSETTS
U.S.A.
EXTRUDED ALUMINUM COLLECTOR FRAME WITH REMOVABLE GLAZING FRAME
12-27-76
1/8
SK-198-14
SHEET 1 OF 2
A-6-a
A-6-b



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