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PERFORMANCE IMPROVEMENT OF A SOLAR HEATING SYSTEM
UTILIZING OFF-PEAK ELECTRIC AUXILIARY

2 JULY 1979 THROUGH JUNE 1980

DR. ADEL H. ELTIMSAHY, PRINCIPAL INVESTIGATOR
COLLEGE OF ENGINEERING
UNIVERSITY OF TOLEDO
TOLEDO, OHIO

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ABSTRACT

This project is concerned with the design and construction of a heat pump system suitable for incorporating in a space solar heating system utilizing off-peak storage from the electric utility. It is also the objective to experimentally evaluate the performance of the system.

The refrigerating capacity, heating capacity and compressor horsepower for a heat pump system using a piston type compressor are first determined. The heat pump design is also matched with the existing University of Toledo solar house heating system. The refrigerant is Freon-12 working between a condensing temperature of up to 172°F and evaporator temperature between 0°F and 75°F. The heat pump is then installed.

Performance indices for the heat pump and the heating system in general are defined and generated by the on-line computer monitoring system for the 1979/80 heating season operation. Monthly and seasonal indices such as heat pump coefficient of performance, collector efficiency, percent of heating load supplied by solar energy and individual components efficiencies in general are recorded.

The data collected is then analyzed and compared with previously collected data. The improvement in the performance resulting from the addition of a piston type compressor with an external motor belt drive is then evaluated.

Data collected points to the potentially improved operating performance of a solar heating system utilizing off-peak storage from the electric utility. Data shows that the seasonal percent of space heating load supplied by solar is 60% and the seasonal percent cost of space heating load supplied by solar is 82% with a solar collection coefficient of performance of 4.6. Data also indicates that such a system would pay for itself in 14 years when used in Northwest Ohio.

CHAPTER 1

INTRODUCTION

On 17 May, 1979, the U.S. Department of Energy awarded the College of Engineering at the University of Toledo a grant to design, install and test a solar assisted heat pump (capable of operating over a wide range of temperatures) when operated with the University of Toledo Solar House which utilizes off-peak electric auxiliary. This report describes the progress of the project during the first year.

The University of Toledo Solar House:

A block diagram of The University of Toledo experimental solar house system is shown in Figure 1.1. For energy storage the system uses two 3785 liter (1000 gallon) tanks. The first tank is connected to a maximum of 81.7 m^2 (880 ft^2) of flat plate solar collectors (two banks A and B) and is filled with a solution of water and ethylene glycol. The second auxiliary tank is charged by a 27 KW of electric resistance heating coils and contains water. To eliminate a more complex control problem during the experiment, each storage tank is connected to the 200 m^2 house space heating load by a fluid to air heat exchanger. A flexible fluid to fluid heat pump set-up is such that it allows the experimentation with different kinds of compressors, condensers and evaporators.

In the heat pump operating mode, heat is removed from the first tank and dumped into the second tank by the heat pump. This allows the solar collector circuit to operate at a reasonably efficient temperature ($<90^\circ\text{F}$) and the system to store energy in the second tank during off peak hours. The second tank must operate at a higher temperature (up to 170°F) if it were to supply the load for the next day under the worst conditions.

Figure 1.1 Block Diagram of UT Experimental Solar Heating System with an Off-Peak Electric Auxiliary.

The entire solar heating system is monitored by a computer based, data monitoring system. Data flow is controlled by a Data General Nova 2 minicomputer. A 16-channel input/output digital subsystem is used for external analog communication. Bulk storage is provided by a Zebec dual floppy disc storage unit. The computer monitoring system also uses a teletype for printing output data. The above solar heating system with its monitoring and control is placed in a University owned house on the main campus.

Scope of Investigations:

The main objective of this project was to design and build a heat pump system suitable for incorporating in a space solar heating system utilizing off-peak storage. It was the objective also to experimentally evaluate the performance of the space solar heating system. The project may be outlined by the following steps:

1. Find the size of a piston type compressor to be used with an external motor belt drive that is consistent with the evaporator and condenser in The University of Toledo system.
2. Replace the existing sealed type compressor by the piston type. This will allow operating the system in a wider range of temperatures.
3. Derive analytically the optimum bang-bang controllers settings for the heat pump system and the solar collector circuit. This will be done using the mathematical model derived for The University of Toledo system.
4. Define hourly, daily, monthly and yearly performance indices for the overall system and its components.
5. Develop the minicomputer necessary software to generate on-line, the system performance indices from the measured states.

6. Determine analytically the optimum solar collector sizing suitable for the house space heating load.
7. Operate the solar heating system with the piston type compressor and the optimum controllers settings during the 1979/80 season.
8. Analyze data collected and compare house performance to that of a conventional heating system, i.e. one that does not use off-peak storage and solar energy.
9. Determine the fuel/energy savings from the use of such systems in Ohio.

Significant Results:

Results indicate that the use of a high temperature solar assisted heat pump in a system utilizing off peak storage from the electric utility instead of a conventional direct solar heating system design would:

(a) Allow the flat plate solar collectors to operate at a much higher efficiency (34% for 1979/80 season, compared to 13% for the 1978/79 season when the system was operated without a heat pump).

(b) Minimize the thermal energy losses from the solar collectors storage tank.

(c) Allow the solar collectors during the cloudy days to act as heat exchangers to supply the heat pump evaporator with the necessary energy, thus permitting the solar assisted heat pump to act as an efficient conventional heat pump when solar energy is not available.

(d) Greatly improve the average coefficient of performance of the heat pump over the conventional one. Average C.O.P. during the 1979/80 season was 3.46.

(e) Eliminate the compressor overheating problem (by using a piston type compressor with an external motor belt drive) encountered in

a conventional heat pump operated at high temperatures needed for off-peak storage.

(f) Help create a much more uniform temperature in storage tanks, thus allowing better system control for human comfort.

(g) Reduce considerably the electrical resistance heating usage as a backup and as a result tremendously increase the percent of heating load supplied by solar energy (60% for the 1979/80 season).

(h) Make the effective solar portion supplied to the heating load higher than the measured since the auxiliary energy will be primarily at off-peak rates. The effective solar portion supplied to the heating load during the 1979/80 heating season was 82%.

(i) Permit the usage of the same system as a cooling one.

(j) Make the system economically feasible in the Northwest Ohio area.

CHAPTER 2

HEAT PUMP DESIGN

The refrigerating capacity and compressor horsepower for a heat pump system using a Victaulic (Lehigh) compressor turning at 544 rpm, working with Freon-12 between a condensing temperature of 172°F and evaporator temperatures of 0°F, 25°F, 50°F, and 75°F were determined. As a further variance, each of these were determined for 150°F, 115°F and 80°F temperatures for the high pressure liquid feed to the throttling valve. The compressor is a piston type with an external motor belt drive. It was installed with the existing evaporator and condenser in the University of Toledo Solar House. The evaporator used is a Dunham-Bush of capacity 21.62 KW (73,800 BTU/hr.), cooling a 50% ethylene glycol water solution by volume. The condenser is also a Dunham-Bush with a capacity of 23.6 KW (80,500 BTU/hr.) at 76.6°C (170°F) condensing temperature.

Assumptions and Data:

1. Displacement of compressor is $0.0216 \text{ ft}^3/\text{revolutions}$ (personal communication, Arthur Allen).
2. Volumetric efficiency of compressor is 85% for all evaporator pressures.
3. Overall compressor efficiency is 85%.
4. Point E is 10°F superheated above point D. (See Figure 2.1)
5. Pressure drop through the evaporator and condenser may be neglected.
6. Thermodynamic data is from "Thermodynamic Properties of Freon 12 Refrigerant" by E. I. Dupont.

The Cycle:

A typical heat pump or refrigeration cycle is shown in Figure 2.1, along with a typical flow diagram. Starting at point A, a high pressure, condensed and usually somewhat supercooled refrigerant is continuously expanded to low pressure in a valve. This partially vaporizes it and lowers its temperature, which in turn allows it to absorb heat at the lower temperature by vaporizing the liquid. The vaporized liquid is usually slightly superheated to point E and then compressed to point F. The condenser removes the superheat, condenses and supercools the high pressure fluid and returns it to the expansion valve.

Equations:

The amount of heat absorbed on the cold side (evaporator) is called the refrigeration effect, RE and is given by the equation,

$$RE = H_D - H_C, \text{ in BTU/lb} \quad 2.1$$

The amount of heat delivered to the hot side (condenser) will be called the heating effect, HE and given by the equation,

$$HE = H_F - H_G, \text{ in BTU/lb} \quad 2.2$$

The refrigerating effect, then, only includes the heat of vaporization, the superheat from D to E occurring in passage from the evaporator to the compressor.

The heating effect includes the work of compression and the latent heat of condensation, but not the supercooling from G to A, which happens outside the condenser.

The work of compression is given by the equation,

$$-W_C = H_F - H_E \quad \text{BTU/lb} \quad 2.3$$

The line E-F' represents the constant entropy path from point E to the higher pressure. The minimum, or isentropic work of compression is,

$$-W_{c \text{ ideal}} = H_{F'} - H_E \quad \text{BTU/lb} \quad 2.4$$

Overall compressor efficiency is given by,

$$\eta_c = \frac{\text{ideal work}}{\text{actual work}} \quad 2.5$$

which may be rearranged to give,

$$H_F = H_E + (H_{F'} - H_E) / \eta_c$$

Circulation rate $\dot{m}$, is given by the equation,

$$\dot{m} = \frac{\dot{V}}{V_E} = \frac{N V_{\text{comp}} 60}{V_E} \eta_v$$

where

$\dot{m}$ = lb refrigerant circulated per hour

$\dot{V}$ = cubic feet refrigerant circulated per hour

V_E = specific volume of refrigerant at condition E, ft^3/lb

V_{comp} = volume displaced by compressor piston each revolution,
 $\text{ft}^3/\text{revolution}$

N = compressor revolution per minute

η_v = volumetric efficiency of compressor, ft^3 gas circulated per
 ft^3 displaced by compressor piston.

From the manufacturer,

$$V_{\text{comp}} = 0.216 \text{ ft}^3$$

$$N = 544 \text{ RPM}$$

$$\eta_v = .85$$

$$\dot{m} = \frac{(544)(.0216)(60)(.85)}{V_E} = \frac{599.27}{V_E} \quad 2.6$$

On an hourly basis, then,

$$\text{Refrigerating capacity} = \dot{m} \times \text{refrigerating effect} \quad 2.7$$

$$\text{Heating capacity} = \dot{m} \times \text{heating effect BTU/hr.} \quad 2.8$$

$$\text{Compressor power} = \dot{m} \times \text{compressor work BTU/hr.} \quad 2.9$$

$$\text{Compressor horsepower} = \text{Compressor Power}/2544 \quad 2.10$$

Calculations:

Data

1. Evaporator temperature °F	0	25	50	75
2. Evaporator pressure, psia	23.85	39.31	61.39	91.68
3. Condenser temperature °F	172	172	172	172
4. Condenser pressure, psia	320	320	320	320
5. $T_E = T_D + 10$ °F	10	35	60	85
6. V_E (charts) ft ³ /lb	1.6411	1.0405	.66703	.4552
7. H_E (charts) BTU/lb	78.703	81.446	84.013	86.524
8. S_E (charts) BTU/lb°R	.17186	.1701	.16822	.16722
9. H_D (charts) BTU/lb	77.27	79.90	82.43	84.83
10. H_G (charts) BTU/lb	50.123	50.123	50.123	50.123
11. $H_A = H_B = H_C$ (from charts)				
BTU/lb for $T_A = 150^\circ\text{F}$	43.85	43.85	43.85	43.85
$T_A = 115^\circ\text{F}$	34.765	34.765	34.765	34.765
$T_A = 80^\circ\text{F}$	26.365	26.365	26.365	26.365

Compressor Calculations

12. $\dot{m} = 599.27/V_E$ lb/hr	365.16	575.94	898.41	1316.5
13. $H_{F'} = H$ at 320 psia with same S as S_E BTU/lb °R	101.45	98.11	96.897	96.24
14. $T_{F'},$ °F	205.5	200.0	194.7	191.8
15. $-W_{c \text{ ideal}} = H_{F'} - H_E$ BTU/lb	22.75	16.66	12.88	9.72
16. $H_F = H_E + \frac{H_{F'} - H_E}{\eta_c}$ BTU/lb	105.47	101.05	99.17	97.95
17. Compressor Horsepower = $\dot{m} (H_F - H_E)/2544$	3.84	4.44	5.35	5.91

Heating Calculation

18. Heating capacity = $\dot{m} (H_F - H_G)$ BTU/hr	20,207	29,331	44,064	62,964
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Refrigeration Calculations

19. Refrigerating Capacity = $\dot{m}$ $(H_D - H_C)$ BTU/hr for				
$T_A = 150^\circ\text{F}$	12,204	20,763	34,660	53,950
$T_A = 115^\circ\text{F}$	15,521	25,995	42,823	65,991
$T_A = 80^\circ\text{F}$	18,588	30,833	50,369	76,969
20. $T_F =$ temperature from chart at $P_{F'}, H_{F'}$ °F	244.8	213.6	204.9	199.3

Initial Heat Pump Operation:

During the month of September 1979, the heat pump system was tested separately (between the two tanks) and data was collected for several runs that took two weeks. Sample results are shown in Figures 2.1 - 2.3 and are in general agreement with calculations.

Figure 2.1a P-H Diagram of the Single Stage Refrigeration.

Figure 2.1b Flow Diagram of the Single Stage Refrigeration.

Figure 2.1 Heat Pump Test on September 21, 1979.

Figure 2.2 Heat Pump Test on September 25, 1979.

Figure 2.3 Heat Pump Test on September 27, 1979

CHAPTER 3

PERFORMANCE EVALUATION

During the months of September and October, components of the system were checked for operation. Minor leaks were found and corrected. Two expansion tanks were also added in the line. Insulation was checked for any moisture. Rates of flow were recorded in the different sections of the system so that it would be fed to the on-line computer simulation. Pressure levels in the heat pump section were checked to be normal. Temperature sensors and their associated electronics necessary to measure the energy states were assembled and calibrated. Some existing electronic components had to be replaced. In general all instrumentation was checked.

System Parameters Used During Operation:

It was decided to determine the best controller settings experimentally instead of mathematically. A preliminary economic feasibility study was also performed to determine the optimum area of solar collectors needed.

Collector Flow Rate: 14.6 gpm./15 solar panels

Number of Solar Panels Used: 30 panels (584 ft²)

Collectors Control: ON/OFF 6°C and 3°C for most of the time

Air Handler: ON/OFF 45°C and 40°C

Room Temperature Control: ON/OFF setting at 70°F

Water Heater Coils Control: ON/OFF 40°C and 45°C

Heat Pump Controller: Evaporator: ON/OFF varied

Condenser: ON/OFF varied

Fluid Used in Tank A: 800 gallons, 450 water and 340 Ethylene glycol.

System Performance Indices:

The following parameters were calculated each hour by the evaluation software during the heating season operation:

- average insolation
- average ambient temperature
- average room temperature
- average solar storage tank temperature
- average off-peak storage tank temperature
- energy collected from collector bank A
- energy collected from collector bank B
- energy delivered to the load from air handler
- energy delivered to the heater coils
- time collection circuit is active
- time air handler is active
- time heater coils are active
- time heat pump is active
- electrical energy delivered to collection circuit
- electrical energy delivered to air handler
- energy stored in solar storage tank
- energy stored in off-peak storage tank

The following parameters were calculated daily by the system software:

- electrical energy delivered to all of the devices
- energy delivered to the load from air handler
- energy stored in solar storage tank
- energy stored in off-peak storage tank
- insolation

energy collected from collector bank A
energy collected from collector bank B
energy delivered by the heater coils
energy delivered to the heat pump
energy delivered by the heat pump
heat pump coefficient of performance
average ambient temperature
percent of load delivered by solar
percent of load delivered by auxiliary
efficiency of collector bank A
efficiency of collector bank B
efficiency of both collectors combined
average load/hour
number of degree days

The following parameters were calculated monthly:

total insolation
average solar collector efficiency
thermal energy supplied to heat pump
electrical energy supplied to heat pump
thermal energy delivered by heat pump
heat pump coefficient of performance
electrical energy supplied by electric coils
thermal energy supplied to heating load
electrical energy supplied to complete system
percent of load delivered by solar

System Operation:

During the 1979/80 heating season the solar heating system was operated using the high temperature heat pump. The heat pump is used to transfer energy from the solar collection storage tank to the auxiliary storage tank, keeping the fluid temperature in the solar collectors tank as low as practically possible, thus increasing the collectors efficiency.

During January 1980, the average temperature was -1.2°C (29.8°F). This is compared with a -5.5°C (22°F) average based on a ten year average for Toledo. Insolation for this month was $20,901 \text{ BTU/ft}^2$ (65.9 KWH/m^2). The heat pump compressor consumed an average 34 KWH/day. Average energy supplied to all auxiliary pumps and air handler fan was 28 KWH/day. Percent of space heating load supplied by solar was determined to be 44% (electric energy supplied to pumps and air handler fan was not included). The solar collection coefficient of performance, which is defined as the ratio of energy at the load point from solar origin to the electric energy supplied to auxiliary pumps and fan, was measured to be 3.9. Average of 28 KWH/day of electric supply energy produced an average of 102.9 KWH/day of net usable energy of solar source.

During the 1979/80 heating season operation, the highest daily average heat pump coefficient of performance (C.O.P.) observed was 5.76 and the lowest was 2.5. The monthly average C.O.P. was 3.46. The monthly average solar collectors efficiency measured was 34% and that does not include the waste energy collected at nights or during cloudy days. During the nights and cloudy days the heat pump was able to maintain the solar collectors storage tank at relatively low temperatures, thus forcing the solar collectors pump to turn on and store waste energy from the ambient.

Table 3.1 shows the University of Toledo's Solar House data summary for the 1979/80 heating season. Figures 3.1 through 3.13 show the data plots.

Comparison With a Conventional System:

Table 3.2 shows the comparison between this year's operation of the solar house versus the cost to an electrically heated house of the same size without the solar system.

The first column gives the percent energy savings for this year. The value represents the actual percent savings in electricity that the solar system realized this year, as compared to the non-solar case. It is seen that on the average a 42% savings was realized.

The second column of Table 3.2 shows the additional cost savings that would have been realized by the system had we been able to take advantage of off-peak rates this season. On the average, a 25% increase in savings could have been seen, from this season's results.

It is seen that as the months get colder, the percent energy savings decreases. This is because during these months more auxiliary energy is needed to supply the heating load. However, during the colder months, the additional off-peak percent cost savings increases. This is because the system is in use a larger percentage of the time during these months, and more energy is used during off-peak periods.

Cost Effectiveness Study:

The method for calculating savings here is based on this season's results and projected over a 20 year period. These calculations are based on:

- (1) the percent of the heating load supplied by solar energy, as observed this season;

Table 3.1 1979-80 Heating System Data Summary

		NOV 79	DEC 79	JAN 80	FEB 80	MAR 80	APR 80	6-MON AVG
Average Ambient Temperature	°C °F	7.9 46	3.2 38	-1.2 29.8	-2.7 27	2.7 37	11.4 53	3.6 38
Total Insolation	KWH/m ² BTU/Ft ²	69 21,922	64.9 20,618	65.9 20,901	88.6 28,074	97.6 30,932	90.2 28,640	79.4 25,160
Elect. Energy Supplied to Heat Pump	KWH/DAY	55.6	55	34	42	47	33	45
Elect. Energy Supplied to Coils	KWH/DAY	11	33	108	110	56	16	55
Elect. Energy to Auxiliary Devices	KWH/DAY	23	29	28	34	30	16	27
Usable Energy of Solar Source	KWH/DAY	96.7	127	102.9	146.3	149.8	93.5	120
Solar Collection C.O.P.		4.22	4.38	3.9	4.3	4.9	5.9	4.6
Heat Pump C.O.P.		2.37	3.03	3.75	4.23	3.89	3.52	3.46
Solar Collectors Efficiency	%	34.5	33.5	37.5	30	36	33	34
Percent Solar	%	70.5	64	44	51	62	70	60
Percent Cost of Load Supplied by Solar	%	91	88	62	73	86	94	82

Figure 3.1 79-80 SEASON AVERAGES

Figure 3.2 NOVEMBER 11, 1979 DATA

Figure 3.3 NOVEMBER 1979 DATA

Figure 3.4 DECEMBER 27, 1979 DATA

Figure 3.5 DECEMBER 1979 DATA

Figure 3.6 JANUARY 3, 1980 DATA

Figure 3.7 JANUARY 1980 DATA

Figure 3.8 FEBRUARY 27, 1980 DATA

Figure 3.9 FEBRUARY 1980 DATA

Figure 3.10 MARCH 2, 1980 DATA

Figure 3.11 MARCH 1980 DATA

Figure 3.12 APRIL 5, 1980 DATA

Figure 3.13 APRIL 1980 DATA

Table 3.2 Systems Energy/Cost Savings

	Percent Energy Savings %	Percent Cost Savings as a Result of using off-peak rates %	Total Effective Energy Savings %
November	40	5	45
December	44	27	71
January	31	37	68
February	38	36	74
March	48	31	79
April	51	16	67
Seasonal Average	42	25	67

- (2) the present and future cost of electricity;
- (3) the cost of the solar heating system components;
- (4) the cost-effectiveness of the solar heating system over a 20 year period.

For each month of this season, two values for the percent of total load supplied by solar energy has been calculated. One value (percent solar) is the percent of thermal load which was actually replaced by solar energy. The other, effective percent solar is a conversion from dollars to BTU's, expressed as percent solar, of the savings from using off-peak electricity rates. The effective percent solar value implies that when using off-peak rates as opposed to regular rates, the same amount of heating dollars will supply more of the load, so overall energy costs are lower. Hence, savings are higher.

The effective cost of the system at the end of n years of operation is defined to be

$$C(n) = C_o(1 + i)^n - \sum_{t=1}^n S(t)$$

where

C_o = the initial cost

i = interest rate

S = savings from utility bill

When the value of $C(n) = 0$, the system is said to pay for itself in utility bill savings.

The cost effectiveness study was done using the existing system, with 584 sq. ft. of collector area. The parameters which are varied are:

- (1) Interest rate (8, 10, or 12%).

(2) Collector cost (100% to 10% of its present cost).

(3) Peak or off-peak utility rate structure.

The first set of data indicates that as collector cost decreases, system payoff time decreases. The second set of data indicates a reduction of 2-9 years in payoff time when using off-peak rates. At 12% interest rate, payoff time varies from 13 years at 100% collector cost, to 4 years at 10% collector cost. This compares to 21 and 13 years, respectively, for standard utility rates. Figures 3.14 - 3.16 show the relationship between percent of collector cost and time for the system to payoff for each interest rate. Notice that as the interest rate increases the effect of decreasing collector cost becomes more pronounced. That is, the slope of the line is greater, so for 12% interest rate a certain decrease in collector cost will result in a more substantial decrease in payoff time than for 8% interest rate.

Figure 3.14 TIME FOR SYSTEM TO PAY FOR ITSELF VS
% COLLECTOR COST AT 8% INTEREST RATE

Figure 3.15 TIME FOR SYSTEM TO PAY FOR ITSELF VS
% COLLECTOR COST AT 10% INTEREST RATE

Figure 3.16 TIME FOR THE SYSTEM TO PAY FOR ITSELF VS
COLLECTOR COST AT 12% INTEREST RATE

CHAPTER 4

CONCLUSIONS

Results indicate that the use of a high temperature solar assisted heat pump in a system utilizing off peak storage from the electric utility is technically and economically feasible for the Northwest Ohio area. According to the 1979/80 heating season data, such a system would pay for itself in 14 years when used only for space heating. Since the same system can be used for space cooling, the figure of 14 years is expected to be much lower. This is possible because:

(1) Flat plate solar collectors operated at a much higher efficiency (34% for 1979/80 season compared to 13% for the 1978/79 season when the system was operated without a heat pump). The thermal energy losses from the solar collectors storage tank were also minimized due to low temperature operation.

(2) During cloudy days the solar collectors acted as heat exchangers to supply the heat pump evaporator with the necessary energy, thus permitting the solar assisted heat pump to act as an efficient conventional heat pump when solar energy is not available.

(3) Average coefficient of performance of the heat pump during the 1979/80 season was 3.45 which is quite an improvement over the conventional one.

(4) The electrical resistance heating usage as a backup was considerably reduced, and as a result the percent of heating load supplied by solar energy (60% for the 1979/80 season) was tremendously increased.

(5) The effective solar portion supplied to the heating load is actually higher than the measured since the auxiliary energy will be

primarily at off-peak rates. The effective solar portion supplied to the heating load during the 1979/80 heating season was 82%.

Two problems remain: the first is the mechanical vibrations that appeared this year, which may be corrected by designing a vibration free mechanical frame to house the compressor, motor, evaporator, condenser and accessories. The second problem is the operation of the system as a space cooling one and calculation of the improvement in the economic feasibility.

APPENDIX A

COMPONENT SPECIFICATIONS

INSTALLATION AND START-UP DATA

EXTERNAL FAN COIL FILTER ASSEMBLY

MODELS

EFCF 36-90 EFCF 48-109 EFCF 60-136

ARKLA INDUSTRIES INC. • P.O. BOX 534 • EVANSVILLE, INDIANA 47704

PART NO. 14084-283-R4

PRINTED IN U. S. A.

REV: Jan., 1975

SPECIFICATIONS

MODEL	DIMENSIONS									
	A	B	C	D	E	F	G	H	J	K
EFCF 36-90	34 $\frac{1}{2}$	29 $\frac{1}{16}$	17 $\frac{3}{4}$	10 $\frac{5}{8}$	12 $\frac{3}{16}$	27	15	8 $\frac{7}{16}$	9 $\frac{3}{4}$	2 $\frac{1}{4}$
EFCF 48-109	41 $\frac{1}{2}$	33 $\frac{1}{16}$	22 $\frac{3}{4}$	11 $\frac{3}{4}$	13 $\frac{1}{2}$	31	20	9 $\frac{25}{32}$	10 $\frac{9}{16}$	3 $\frac{1}{4}$
EFCF 60-136	41 $\frac{1}{2}$	33 $\frac{1}{16}$	22 $\frac{3}{4}$	11 $\frac{3}{4}$	13 $\frac{1}{2}$	31	20	9 $\frac{25}{32}$	10 $\frac{9}{16}$	3 $\frac{1}{4}$

MODEL	DIMENSIONS						
	L	M	N	P	R	S	T
EFCF 36-90	2 $\frac{1}{8}$	1 $\frac{1}{2}$	10 $\frac{7}{8}$	13 $\frac{1}{4}$	2	$\frac{3}{4}$	$\frac{3}{4}$
EFCF 48-109	2 $\frac{1}{8}$	3 $\frac{5}{8}$	12 $\frac{3}{16}$	17 $\frac{1}{16}$	$\frac{9}{16}$	$\frac{3}{4}$	$\frac{3}{4}$
EFCF 60-136	2 $\frac{1}{8}$	3 $\frac{5}{8}$	12 $\frac{3}{16}$	17 $\frac{1}{16}$	$\frac{9}{16}$	$\frac{3}{4}$	$\frac{3}{4}$

DIMENSIONAL VIEWS

LEFT SIDE

FRONT

RIGHT SIDE

FAN MOTOR DATA

	EFCF 36-90	EFCF 48-109	EFCF 60-136
Type Fan	Centrifugal Wheel	Centrifugal Wheel	Centrifugal Wheel
Horse Power	$\frac{1}{3}$	$\frac{1}{3}$	$\frac{3}{4}$
Voltage	115/230*	115/230**	115/230**
Cycle	60	60	60
Phase	Single	Single	Single
Maximum External Resistance At Rated Air Flow (Total Supply & Return)	.33"	.53"	.50"
Net Weight Lbs.	137	166	174

FILTER DATA

Type	Permanent	Permanent	Permanent
Number	2	2	2
Size	13 $\frac{7}{8}$ x 15	16 x 20	16 x 20
Arkla Part No.	14898-32	14898-16	14898-16

* Shipped with motor wired for 115 volts

** Shipped with motor wired for 230 volts

OPERATING INSTRUCTIONS & PARTS LIST

CLOSE-COUPLED CENTRIFUGAL PUMPS (Bronze)

MODELS 1P787 THRU 1P793

FORM
5S1164

DAYTON ELECTRIC MANUFACTURING CO. CHICAGO 80648

0374/068/5N
2 **Description**

Teel Centrifugal Pumps use an open type, curved vane impeller centrally located and rotating in an efficiently designed volute housing. The medium to be pumped enters the eye of the impeller and is picked up by the vanes. It is then accelerated to a high velocity by rotation of the impeller and discharged by centrifugal force into the volute and out the discharge. This simplicity of operation affords operation under a wide variety of conditions. Centrifugal pumps when properly installed and maintained will operate trouble-free over long periods of time.

Quiet, continuous, high volume flow are features of your Teel Pump. The complete absence of contacting parts assures long life. The Close-Coupled Pump and Motor Unit operates at 3450 RPM to give maximum design efficiency. Maximum horsepower is required with a wide open discharge line. Increasing the pressure head (discharge restriction) decreases the power required.

Specifications

Dayton Motor		60 Hz. Volts	Inlet	Outlet	High	Wide	Long
Model	HP Type						
1P787	1/3 Split Phase	115	3/4"	1/2"	6"	6 1/2"	12 1/2"
1P788	1/2 Capacitor	115/230	1	3/4	6 1/4	6 3/4	12 1/2
1P789	3/4 Capacitor	115/208-230	1	3/4	6 1/4	6 3/4	13 1/4
1P790	1 Capacitor	115/208-230	1 1/4	1	8 1/2	6 3/4	14
1P791	1 1/2 Capacitor	115/208-230	1 1/4	1	6 1/4	6 3/4	14
1P792	3/4 3-Phase	208-220/440	1	3/4	6 1/2	6 3/4	13 1/4
1P793	1 1/2 3-Phase	208-220/440	1 1/4	1	6 1/2	6 3/4	14

Performance

Model	PUMP HEAD IN FEET								
	5'	10'	15'	20'	30'	40'	50'	60'	70'
1P787	29	26	25	23	17	12	2	—	—
1P788	43	37	32	28	20	2	—	—	—
1P789	83	78	75	68	57	47	22	1	—
1P790	89	86	83	76	67	54	37	16	—
1P791	95	94	92	88	77	66	52	32	—
1P792	83	78	75	68	57	47	22	1	—
1P793	95	94	92	88	77	66	52	32	—

Safety

When wiring motor, follow all local electrical and safety codes, as well as the National Electrical Code (NEC) and the Occupational Safety and Health Act (OSHA).

Motor must be securely and adequately grounded. This can be accomplished by wiring with a grounded, metal-clad raceway system, by using a separate ground wire connected to the bare metal of the motor frame, or other suitable means.

Always disconnect power source before working on or near a motor or its connected load. If the power disconnect point is out-of-sight, lock it in the open position and tag to prevent unexpected application of power.

Be careful when touching the exterior of an operating motor — it may be hot enough to be painful or cause injury. With modern motors this condition is normal if operated at rated load and voltage — modern motors are built to operate at higher temperatures.

Do not insert any object into motor.

Thermal Protection

Motor is equipped with an automatic reset thermal protector, and may restart unexpectedly when motor and protector cools after tripping. Protector tripping is an indication of motor overloading as a result of operating the pump at low heads (low discharge restriction), excessively high or low voltage, inadequate wiring, incorrect motor connection, or a defective motor or pump.

Installation

Improper installation may cause poor pump efficiency, increase power consumption and decrease operating life. Position your pump as close as possible to the source of liquid. Keep the suction line as short and direct as possible. Plan and install the suction line so that air pockets cannot form in it. Use a minimum of elbows and fittings and use a length of straight pipe at least 6" long for the entrance into the pump suction. Never use pipe smaller than the pump port sizes and preferably use the next larger size. A suitable strainer should be installed in the suction line. The net area of the strainer must be at least four (4) times that of the suction pipe.

Installation Instructions

SERVICE REFERENCE

DIV. 4	SEC. A	NUMBER 6001
SALES REFERENCE		M40411
		161-048353-001
DATE November, 1975		

Chromalox Adapt-O-Matic Water Heater Element

(Catalog Numbers AH and AL)

And Adapt-O-Flange Kits

(Catalog Number ADF)

General —

This universal replacement is intended to be used with Adapt-O-Flanges to fit nearly every water heater. (See illustrations of available Adapt-O-Flanges on reverse side.)

This element is also intended to be used as a direct replacement (without flange) for Rheem, Westinghouse, Kelvinator, Norge, Montgomery Ward, and Ruud water heaters originally equipped with screw-in type heating elements.

Installation —

1. Disconnect power at service switch.
2. Disconnect lead wires on element to be replaced.
3. Shut off water supply and drain tank below level of element opening.
4. Remove old element from water heater. Clean tank flange face or threaded opening. Remove accumulated lime deposits from tank bottom.
5. Select Adapt-O-Matic Element which agrees with wattage and voltage of element to be replaced. (For 208 volt service, use 240 volt element rated approximately 33% higher. Example: 208V, 1000 watts, use 240V, 1250 watts, or 240V, 1500 watts.)
6. To replace flanged water heater element — select Adapt-O-Flange that matches the flange of the old element. (See sketches on back.)
 - a. ADF-1 Flange fits all tank flanges that have four bolts on 2½" bolt circles and the Twist-Lock design. It will replace Cutler-Hammer, General Electric, Heatube, Thermalink, TK, Vulcan, Westinghouse and Wiegand models that have four holes on 2½" bolt circles. See

diagram 1 for conventional replacement; see diagram 2 for replacement of Twist-Lock design and see diagram 3 for replacement of TFI unit.

- b. ADF-2 Flange fits 4-bolt 2¾" Bolt Circle Elements. Used on Westinghouse Water Heaters.
- c. ADF-3 Flange fits 4-bolt 3¾" Tank Bolt Centers used on Frigidaire and Thermogray Water Heaters.
- d. ADF-4 Flange fits 6-bolts 2½" Tank Bolt Centers used on White Water Heaters.
- e. ADF-5 Flange fits 2-bolt 2¾" Tank Bolt Centers. Replaces older types of elements in field.

Note: All Adapt-O-Matic Flange Kits except the ADF-1 are furnished with a specially designed adhesive gasket to insure proper gasket seating. It is always accurately centered in installation. The seating pressure is concentrated around the tank opening because much gasket area has been eliminated.

Strip protective paper from adhesive side of gasket. Place gasket on Adapt-O-Flange, with adhesive side of flange. Flange is now ready for installation.

7. Assemble Adapt-O-Matic element to flange and tighten securely. If flange has boss at threaded opening, boss goes inside tank.
8. Bolt assembly to tank using one of the gaskets supplied with flange. For TFI and Twist-Lock Replacement use Rubber Gasket supplied. We recommend using pipe joint compound on Adapt-O-Flange and tank flange mating surfaces.
9. Turn on water supply and fill tank with water, checking for leaks. Purge all air from tank by opening hot water faucet until water flows evenly.

WILKIE-DUSTON CO.
Manufacturer's Representatives
2124 HOLLAND-SYLVANIA ROAD
TOLEDO, OHIO 43615
Telephone (419) 531-3588

DUNHAM-BUSH

**'CSTC' CLEANABLE SHELL AND
TUBE CONDENSERS**

INTRODUCTION

The Dunham-Bush CSTC Cleanable Shell and Tube Condenser is a compact, heavy-duty, highly efficient water cooled refrigerant condenser.

Available in capacities from 5 to 150 tons, using R-12, R-22 or R-502 refrigerants, these condensers are designed for use with city water, tower water, or sea water (marine models). Both water heads are removable to provide easy access to the straight-thru, smooth-wall water tubes for cleaning purposes. It is designed and circuited to provide subcooled refrigerant and has adequate pumpdown capacity.

FEATURES

- SEAMLESS STEEL SHELL WITH FLUSH TUBE SHEETS.
- 3/8" TUBES ARE EXPANDED INTO ALUMINUM FIN IN BANKS OF FIVE. EACH BANK IS SECURED WITH BRACKETS WHICH ARE WELDED TO THE SHELL.
- ELECTRONIC CONTROL IS USED TO ROLL ALL TUBES INTO HEAVY FIXED TUBE SHEETS.
- SMOOTH WALL WATER TUBES ARE EASILY CLEANED.
- ALL CONDENSERS ARE CONSTRUCTED IN ACCORDANCE WITH ASME CODE.
- MODELS CSTC 842 AND LARGER ARE ASME CERTIFIED ON BOTH REFRIGERANT AND WATER SIDES AND CARRY THE ASME STAMP.

'CSTC' STANDARD MODELS

COPPER WATER TUBES

STEEL TUBE SHEETS

CAST IRON HEADS

DESIGNED FOR USE WITH REFRIGERANTS R-12, R-22 AND R-502.

'MCSTC' MARINE MODELS

CUPRO-NICKEL TUBES

CUPRO NICKEL CLAD STEEL TUBE SHEETS

BRONZE HEADS

DESIGNED FOR USE WITH REFRIGERANTS R-12, R-22 AND R-502.

SPECIAL CONDENSERS ARE AVAILABLE TO MEET USCG OR NAVY APPROVAL - CONSULT FACTORY.

MODEL CSTC OR MCSTC	DESIGN WORKING PRESSURE	
	WATER SIDE	REFRIGERANT SIDE
548 THRU 1460	125 PSIG	400 PSIG
1648 THRU 1860	125 PSIG	300 PSIG

OPTIONAL ACCESSORIES

- FLANGED COMPANION VALVE OR FLANGED COUPLING (with gasket). Select size and part number from Table 1. See dimensions for flange details.
- RELIEF VALVE ASME rated. Specify proper pressure rating and part number. (See Table 2).

TABLE 1

R-OUT FLANGE SIZE	R-OUT FLANGED VALVE		R-OUT FLANGED COUPLING	
	SIZE IDS	P/N	SIZE IDS	P/N
#1	7/8	VAL 372	7/8	0121P01
	1-1/8	VAL 373	1-1/8	0121P00
	1-3/8	VAL 374	1-3/8	0121P02
#2	1-1/8	VAL 315	N/A	—
	1-3/8	VAL 83	1-3/8	0121P10
	1-5/8	VAL 59	1-5/8	0121P06
#3	2-5/8	VAL 54	N/A	—

TABLE 2

400 PSIG	P/N 0609P65	CSTC 548 Thru 1460
300 PSIG	P/N 0609P45	CSTC 1648 Thru 1860

Note: If local codes require a lower rating so specify.

- 1/4" SHUT OFF VALVE - for purging, charging or connecting to a water regulating valve cap tube. (P/N 25P135)
- SPIRAL TUBE CLEANER

DIMENSIONS

Dunham-Bush reserves the right to make changes in specifications and design without notice.

NOTE: for cooling tower operation (2 pass) connect W-OUT and W-IN together to form W-IN connection from cooling tower. For city water operation (4 pass) plug cooling tower conn.

REFRIGERANT OUTLET FLANGE DETAILS

FLANGE DETAIL No. 1

FLANGE DETAIL No. 2

FLANGE DETAIL No. 3

GENERAL DATA			CONNECTION DATA					DIMENSIONAL DATA													Pump Down Cap. Lbs. 80% R-12 90°	Pump Down Cap. Lbs. 80% R-22 R-502 90°	
Model No. CSTC	Nom. Tons	No. Tubes	Ref. Conn.		Water Conn.		Twr. Conn.	INCHES															
			IN	OUT	F.P.T.			F.P.T.	A	B	C	D	E	F	G	H	J	K	L	M			N
			IDS	Style	IN	OUT																	
548	5	20	1½	½ F.P.T.	1"	1"	1¼"	5½	54½	32½	7½	22½	48½	—	—	—	4½	2	1½	1½	1½	31#	28#
842	7½	40	1½	FLG. #1	1"	1"	1½"	8½	51½	30½	8½	5½	44½	1½	¼	—	6½	3½	¼	2½	1½	70#	63#
848	10	40	1½	FLG. #1	1"	1"	1½"	8½	57½	33½	8½	8½	50½	1½	¼	—	6½	3½	¼	2½	1½	92#	83#
1042	15	80	1½	FLG. #1	1¼"	1¼"	2"	10½	52½	30½	8½	6½	44½	1½	½	—	6½	4½	—	3½	2½	107#	96#
1048	20	80	1½	FLG. #1	1¼"	1¼"	2"	10½	58½	33½	8½	8½	50½	1½	½	—	6½	4½	—	3½	2½	122#	110#
1060	25	80	1½	FLG. #1	1¼"	1¼"	2"	10½	70½	40½	8½	14½	62½	1½	½	—	6½	4½	—	3½	2½	151#	136#
1248	30	120	1½	FLG. #1	1½"	1½"	2"	12½	58½	33½	15½	12½	46½	3½	¼	—	7½	5	1½	3½	3½	171#	154#
1260	40	120	1½	FLG. #2	1½"	1½"	2"	12½	70½	40½	9½	13½	58½	3½	¼	—	7½	5	1½	3½	3½	214#	193#
1454	50	180	2½	FLG. #2	3"	3"	—	14	66	37½	9½	10 ½	57½	—	—	2½	8½	5½	—	—	—	226#	204#
1460	60	180	2½	FLG. #2	3"	3"	—	14	72	41½	10½	14½	59½	—	—	2½	8½	5½	—	—	—	250#	225#
1648	75	290	2½	FLG. #2	4"	4"	—	16	62½	35½	10½	9½	52½	—	—	2½	8½	6½	—	—	—	237#	214#
1660	100	290	3½	FLG. #2	4"	4"	—	16	74½	42½	12½	15½	61½	—	—	2½	8½	6½	—	—	—	296#	267#
1848	125	390	3½	FLG. #3	4"	4"	—	18	61½	35½	10½	9½	52½	—	—	3½	11½	7½	—	—	—	332#	300#
1860	150	390	3½	FLG. #3	4"	4"	—	18	73½	42½	10½	14½	64½	—	—	3½	11½	7½	—	—	—	415#	374#

NOTES:

1. Dimensions subject to change without notice. Certified dimensional prints can be supplied upon request.
2. All dimensions remain the same for marine models (MCSTC) except for the following. Dimension B should be increased by 1/4", and Dimensions C, D, E, and F should be increased by 1/8".
3. For companion flanges or flanged valves for the R-Out connections see Optional Accessories

WILKIE-DUSTON CO.
Manufacturer's Representatives
2124 HOLLAND-SYLVANIA ROAD
TOLEDO, OHIO 43615
Telephone (419) 531-3588

FORM NO. 8010F

JANUARY 1971

SUPERSEDES FORM 8010E
DATED APRIL 1969

50

DUNHAM-BUSH

**'CH' & 'DCH' INNER-FIN
WATER CHILLERS**

INNER-FIN WATER CHILLERS

'CH' and 'DCH' Models

TYPICAL CHILLER CONSTRUCTION BRAZED OR WELDED HEAD MODELS

'CH' MODELS—as shown

'DCH' MODELS—same except refrigerant side of chiller is dual circuited with twin inlet and outlet connections

The Dunham-Bush line of 'CH' and 'DCH' chillers offers the optimum in efficient heat transfer. Patented Inner-Fin, the forerunner of all other types of internal extended surface in refrigeration tubing, is still "second to none" in overall performance. For any given shell diameter, Dunham-Bush chillers effect more net refrigeration per lineal foot than any other chiller available today.

Check these other all important features;

- Heavy shell available in brass or steel up to and including 6" dia. Steel shell is furnished on 7" diameter and larger. Stainless steel or brass shells are available for low temperature applications (below -20°F)
- Brass segmental baffles precision machined to ID of shell
—selection of baffle spacing is available to meet a wide range of water pressure drop requirements.
- Copper chiller tubes silver brazed into brass tube sheets for permanent leak proof joints
- Versatility
—in addition to conventional water chilling, Dunham-Bush 'CH' and 'DCH' chillers have numerous other product cooling applications—a partial listing includes:
Liquid refrigerant sub-coolers (for 2 stage refrigeration systems)
Brine coolers
Oil coolers
Jet fuel
Gas coolers —Air —Nitrogen —Helium —Natural gas —Ozone —Carbon Dioxide

Generally speaking, any low viscosity fluid, fluid mixture, or gas that has no corrosive reaction with copper and steel may be successfully cooled with these units. Where corrosion is a factor, Dunham-Bush Stainless Steel Inner-Fin Chillers should be considered—see Page 11.

DESIGN PRESSURES

TEMPERATURE CONDITIONS	SHELL SIDE (LIQUID)		TUBE SIDE (REFRIGERANT)	
	DESIGN PRESSURE (PSIG)	TEST PRESSURE (PSIG)	DESIGN PRESSURE (PSIG)	TEST PRESSURE (PSIG)
-20°F OR ABOVE	200	300 (HYDROSTATIC)	300	375 (PNEUMATIC)
STEEL SHELL below -20°F	80		120	
STAINLESS STEEL OR BRASS SHELL below -20°F	200		300	

'CH' INNER-FIN WATER CHILLERS

Single Circuit Models

Nominal 1.5 thru 50 Ton Models

Nominal 60 thru 115 Ton Models

NOM. TONS	MODEL	A	B	C	E	F	G		H	J	K	L	M	W-IN W-OUT O.D. S (NPT)	R-IN O.D. (I.D.)	R-OUT O.D. (I.D.)	VOL. OF SPACE IN SHELL GALS.	MIN. LBS. REFRIG. CHG.	TOTAL INS., WGTS. WITH WATER LBS.	SHIP. WGT. INS. LBS.
							STD. INSULA- TION	HEAVY INSULA- TION												
1.5	CH224	35 3/8	31 3/8	29 1/8	23 1/8	4 1/8	4 1/2	5 3/8	2 3/8	7 1/8	4	—	11 1/8	3/8	3/8	1 1/8	.20	1.0	40	52
2	CH324	34 3/8	30 1/2	29 1/8	22 3/8	3 3/8	5 3/8	6 3/8	3	6 3/8	4 1/8	—	11 3/8	1 1/8	3/8	1 1/8	.30	1.0	48	70
4	CH336	46 3/8	42 1/2	41 1/8	34 3/8	3 3/8	5 3/8	6 3/8	3	6 3/8	4 1/8	—	17 3/8	1 1/8	3/8	1 1/8	.44	1.4	63	90
5	CH348	58 3/8	54 1/2	53 1/8	46 3/8	3 3/8	5 3/8	6 3/8	3	6 3/8	4 1/8	—	23 3/8	1 1/8	3/8	1 1/8	.60	1.9	79	114
3.5	CH424	34 3/8	31 1/4	29 1/8	22 1/8	4 1/2	6 1/4	8	4 1/8	7 3/8	3 1/2	—	11 1/8	1 3/8	3/8	1 3/8	.55	1.8	76	97
7.5	CH436	46 3/8	43 1/4	41 1/8	34 3/8	4 1/2	6 1/4	8	4 1/8	7 3/8	3 1/2	—	17 1/8	1 3/8	3/8	1 3/8	.82	2.7	98	110
10	CH448	58 3/8	55 1/4	53 1/8	46 3/8	4 1/2	6 1/4	8	4 1/8	7 3/8	3 1/2	—	23 1/8	1 3/8	3/8	1 3/8	1.10	3.7	130	142
11	CH536	48 1/8	43 3/8	42 1/8	33 3/8	4 3/8	7 1/2	9	5 3/8	8 3/4	4 1/8	—	16 3/8	1 3/8	3/8	2 3/8	1.44	4.5	149	158
15	CH548	60 1/8	55 3/8	54 1/8	45 3/8	4 3/8	7 1/2	9	5 3/8	8 3/4	4 1/8	—	22 3/8	1 3/8	3/8	2 3/8	1.93	6.0	182	190
24	CH560	72 1/8	67 3/8	66 1/8	57 3/8	4 3/8	7 1/2	9	5 3/8	8 3/4	4 1/8	—	28 3/8	1 3/8	3/8	2 3/8	2.42	7.5	228	230
20	CH636	48 3/8	44 1/4	42 3/8	33 3/8	5 1/2	8 3/8	10 3/8	6 1/2	7 1/2	4 3/4	—	16 3/8	(2)	(1 1/8)	(2 1/8)	2.00	6.7	204	200
25	CH648	60 3/8	56 1/4	54 3/8	45 3/8	5 1/2	8 3/8	10 3/8	6 1/2	7 1/2	4 3/4	—	22 3/8	(2)	(1 1/8)	(2 1/8)	2.73	9.0	254	250
35	CH660	72 3/8	68 1/4	66 3/8	57 3/8	5 1/2	8 3/8	10 3/8	6 1/2	7 1/2	4 3/4	—	28 3/8	(2)	(1 1/8)	(2 1/8)	3.42	11.2	318	300
47	CH760	74 3/8	69 1/4	68 3/8	56 3/8	8 3/8	9 3/8	12 3/8	7 3/4	9 3/8	5 3/8	—	28 1/8	(3)	(1 1/8)	(2 3/8)	5.00	15.6	426	397
50	CH860	75 3/8	70 3/4	69 3/8	53 1/2	8 3/8	11 1/8	14	8 3/8	10 1/2	5 3/8	—	26 3/4	(3)	(1 1/8)	(2 3/8)	6.25	40.0	545	614
60	CH1048	65 1/2	61 1/4	59 1/4	41 1/2	9 3/8	12 3/8	14 3/8	10	11 1/2	6 1/2	2 1/8	20 3/4	(4)	(1 3/8)	(3 3/8)	7.5	45.0	564	545
75	CH1060	77 1/2	73 1/4	71 1/4	53 1/2	9 3/8	12 3/8	14 3/8	10	11 1/2	6 1/2	2 1/8	26 3/4	(4)	(1 3/8)	(3 3/8)	9.4	56.3	706	655
90	CH1160	77 1/2	73 1/2	71 1/2	53 1/2	10	14	15 1/4	11 3/8	12 3/8	7	2 1/8	26 3/4	(4)	(1 3/8)	(3 3/8)	11.1	67.5	830	855
100	CH1254	73 1/2	68 3/8	66 3/8	47 1/2	10 1/2	14 3/8	16 3/8	12 1/4	12 3/4	7 3/8	2 1/8	23 3/4	(4)	(1 3/8)	(4 1/8)	12.0	74.0	850	910
110	CH1260	79 1/2	74 3/8	72 3/8	53 1/2	10 1/2	14 3/8	16 3/8	12 1/4	12 3/4	7 3/8	2 1/8	26 3/4	(4)	(1 3/8)	(4 1/8)	13.3	82.5	954	988
115	CH1360	79 1/2	76 3/8	74 3/8	53 1/2	11 3/8	16 3/8	18 3/8	13 3/8	13 3/8	8 3/8	3 1/4	26 3/4	(4)	(1 3/8)	(4 1/8)	17.6	104.0	1140	1200

* Based on: Water in 55°F; Water out 45°F; Refrigerant 35°F.

AAIT

NOTE: Chillers are designed for simple thermal valve refrigerant feed. Valves should be of the externally equalized type and be set for 5° superheat and suction-liquid heat interchangers must be used to attain maximum capacity. See Form No. 8050 or 8051 for selection of proper heat interchangers.

INSULATION: Markedly superior Urethane* "foam-in-place" insulation enclosed in 20 gauge galvanized sheet metal jacket. Finish is attractive blue lacquer.

Standard insulation is ample for commercial temperature applications. Heavy insulation is available for low temperature applications.

*For comparative purposes, "k" factors are listed below for popular insulating materials.

Urethane	.13
Expanded Polystyrene	.25
Rockcork	.35 (avg.)

In addition to having a low "k" factor, Urethane by virtue of its closed cell construction, is an excellent vapor barrier with high structural strength.

Design Pressures: Standard construction is suitable for R-12, R-22, or R-502 Systems.

ORDERING INSTRUCTIONS

Specify quantity and complete model number. Example:

CH 6

Shell
Diameter
6 1/2" OD
6" NOM.

36

Nominal
Tube Length
36"

A

Baffle Spacing A
(See rating
tables for proper
selection)

I

Add I to Model
No. as shown if
insulation is
desired. Add II
to Model if
Heavy Insulation
is desired.

T

Add T for Top
Connection, B for
Bottom, RS for
Right Side, or
LS for Left Side,
with respect to
R-in end.

Chillers CH636 and larger are constructed of ASME approved materials and will be furnished with code stamp and national board number.

Chillers CH560 and smaller are covered by Underwriters' Laboratory Approval and Listing.

Flat-Plate Solar Collector

Dimensions

Specifications

Cover glass (2) (fully tempered)	- 1/8" TUF-FLEX® tempered safety glass
Exposed absorber plate area	- 19.47 sq. ft.
Absorber plate	- all copper
Heat transfer fluid system	- all copper
Frame and back panel	- aluminum
Back insulation	- 3" treated fiber glass
Recommended fluid flow rate	- 0.5 gpm
Pressure drop at recom. flow (glycol/water)	- 0.5 psi
Design pressure	- 100 psi
Test pressure	- 200 psi
Burst pressure (calculated)	- 450 psi
No-flow protection required	- none
Maximum internal temperature (no-flow)	- 450° F
Performance test specification	- NBSIR 74-635
Environmental conditions	- -40°F. to +120°F.
Maximum wind or snow load	- 30#/SF
Fluid connections	- brass 45° flared
Installation orientation	- short side horizontal
Min. tilt angle to allow complete draining of panel	- 5° (from horizontal)
Maximum tilt	- 90°
Mounting	- 4 point, on brackets
Panel weight (filled)	- 150 lbs.
Installed load	- 7.5#/SF
Shipping weight	- 160 lbs.

APPENDIX B

SYSTEM SOFTWARE

The software used to monitor and evaluate the solar heating system contains four levels of programming as shown in Figure B.1. This design was used this year due to its success in the previous years' operations. A complete listing of the software is given at the end of this Appendix along with the program variables and flow charts. One disk storage file is maintained by the system software. On this file are stored instantaneous sensor readings at selected intervals, the hourly and the daily average values of insolation, temperatures, efficiencies, and BTU measurements. The majority of the software is written in Fortran, however, assembly language handlers are used to communicate with the real time clock and the analog to digital convertor.

The first level of programming consists of the main program MAINTST. A flow chart of the main program is shown in Figure B.2. Its function is to provide for operator commands and overall program operation.

A flow chart of the second level of programming is shown in Figure B.3. This subroutine, called MEASURE is used for monitoring and control. MEASURE provides five modes of control: (1) system monitoring, (2) initialization, (3) file changing, and (4), (5) two modes of file printing.

- (1) The monitoring mode is completed automatically without operator interaction every control time interval. Every control time interval, all monitored variables are read and the system evaluation software (3rd level) is executed. It is commanded by the main program.

- (2) Initialization allows the operator to set the real time clock and initialize variables.
- (3) In the file changing mode, storage files are opened and closed by the operator. Also, the control and storage time intervals are chosen by the operator. Each storage time interval, all of the system monitored variables are recorded on the disk storage file.
- (4) File printing mode prints out the contents of the last data file.
- (5) Special file printing mode allows a print-out of the evaluation data that has been calculated for the fraction of the hour which has already passed. This had become a useful feature this season because of the necessity of taking some measurements by hand (power consumption). The problem was that the evaluation software records device run times, BTU calculations, and other accumulated evaluation data only on-the-hour. The student taking readings by hand could not always wait for on-the-hour times, so this print mode commands the evaluation software to record its data at any time requested during any hour. This allows an easy correlation between the software - accumulated data and the hand - accumulated data.

The third level of programming is flow charted in Figure B.4. This level which contains several subroutines, is used to supervise and calculate performance evaluation indices. The following parameters are calculated each hour and totalled each day by the evaluation software:

average insolation (BTU/ft^2)

average ambient temperature ($^{\circ}\text{C}$)

average room temperature ($^{\circ}\text{C}$, $^{\circ}\text{F}$)

average solar storage tank A temperature ($^{\circ}\text{C}$)
 average off-peak storage tank B temperature ($^{\circ}\text{C}$)
 energy collected from each collector bank (BTU)
 efficiency of each collector bank (%)
 coefficient of performance of each collector bank
 energy delivered to the load from air handlers (BTU)
 energy delivered to evaporator from solar tank A (BTU)
 energy delivered to load tank B from condenser (BTU)
 energy delivered to heater coils (BTU)
 time collection circuit is active (min)
 time air handlers are active (min)
 time heat pump is active (min)
 time heater coils are active (min)
 electrical energy delivered to collection circuit (converted
 to BTU)
 electrical energy delivered to air handlers (converted to BTU)
 electrical energy delivered to heat pump circulating pumps
 (converted to BTU)

The third level evaluation software is performed every control interval.
 The decoder input is read each time to see which devices if any are active.
 If a device is active, a fourth level of programming is performed.

The fourth level of programming makes device calculations. A flow
 chart of a typical device subroutine is shown in Figure B.5. Each time a
 device subroutine is executed, the time the device is active for the hour
 is incremented by the amount of the control interval. Appropriate tem-
 perature measurements are made, and energy flow calculations are calculated
 as though the device has been active for the entire control interval.

The control interval must be kept relatively small so that numerous device readings are taken each hour. Typically, the control time interval is two minutes. Energy flow calculations made in the fourth level are summed over the entire hour to obtain an hourly heat flow rate. Appendix B gives a sample of software output for instantaneous, hourly, and daily situations. A 24-hour day this season was established to be from 9:00 AM one day to 9:00 AM the next. This was due to the necessity of correlating software and hand-accumulated data. It would have been impractical to demand midnight readings, so the "day" was shifted in this way.

SOFTWARE CALCULATIONS

The monitoring and evaluation software receives external information from the digital subsystem's analog to digital convertor. A 16-channel input bus is used to read the processed analog information. Figure B.6 lists each channel and its corresponding monitored variable. Individual channels are read by calling the digital converted subroutine handler in the system software. The convertor converts an analog signal with a maximum magnitude of 10 volts to a 12 bit integer. A gain constant of $2^{12}/10$ (integers/volts) is experienced in each measurement.

There are three calculations that the program must make from the raw input data which provide the basis for ease of understanding the house performance. To be described below, these are: (1) insolation levels, (2) decoder, and (3) temperature sensor readings. Upon these are based the remaining calculations of solar house parameters which are: (4) collector bank heat, (5) heat to air handler B, (6) heat delivered to evaporator, (7) heat taken from condenser, (8) circulating pumps motor energies

(9) heater coils energy, (10) collector efficiency and coefficient of performance, (11) coefficient of performance for heat pump, (12) degree days, (13) % solar, system COP, (14) off-peak % solar.

(1) Insolation Levels:

When the insolation channel is read, the integer described previously must be multiplied by a conversion factor that accounts for both the interfacing circuit gain and the conversion gain.

conversion gain = interface gain x conversion gain x unit conversion

$$= 8.0018 \times 10^{-3} \frac{\text{volts}}{\text{watt-m}^{-2}} \times \frac{2^{12}}{10} \frac{\text{integers}}{\text{volts}} \times 3.153 \frac{\text{watt-m}^{-2}}{\text{BTU/ft}^2\text{-hr}}$$

$$= 10.33 \frac{\text{integers}}{\text{BTU/ft}^2\text{-hr}}$$

Finally,

$$\text{insolation} = \text{integer read} \times \frac{1}{10.33} \frac{\text{BTU/ft}^2\text{-hr}}{\text{integer}}$$

$$= [\text{integer read} \times 0.0967] \frac{\text{BTU}}{\text{ft}^2\text{-hr}}$$

(2) Decoder:

Similarly, the decoding circuit channel must be multiplied by the analog to digital conversion factor:

System <u>Monitored State</u>	Decoding CKT <u>Output (Volts)</u>	Monitored <u>State Integer</u>
all units off	0.02	8
collector circulating pump	0.14	57
heat pump	0.20	81
air handler A	0.53	217
air handler B	0.94	385
heater coils	1.83	750

The decoder output is a sum of all the active device integers. This gives a unique value for each combination of active devices. The combinations used this season are listed in Figure B.7.

(3) Temperature Sensor Readings:

Figure B.8 illustrates the temperature transducer circuit's output voltage as a function of temperature. The software must use an offset value as well as a gain factor to convert the analog signal into temperature degrees.

$$\begin{aligned}
 \text{temperature in } ^\circ\text{C} &= \text{offset value} - \frac{\text{integer read}}{\text{gain factor}} \\
 &= 115 ^\circ\text{C} - \frac{\text{integer read}}{\frac{212}{145^\circ\text{C}}} \\
 &= 115 ^\circ\text{C} - \text{integer read}/28.248
 \end{aligned}$$

The nonoperative parameters such as insolation and monitored temperatures are recorded each control interval. These are used to perform the remaining calculations.

(4) Collector Bank Heat:

The liquid solution which flows through the collector banks originates from the solar storage tank A. This tank is filled with 350 gallons of ethylene glycol and 450 gallons of water. To determine the amount of heat absorbed by this mixture, the following is found first:

$$\begin{aligned}
 \text{(a) Specific heat of water : } C_{p_w} &= 1.00 \frac{\text{cal}}{\text{g}^\circ\text{C}} \times \frac{1}{860.4} \frac{\text{KWH g}}{\text{cal Kg}} \times \frac{3412 \text{ BTU}}{\text{KWH}} \\
 &= 3.965 \frac{\text{BTU}}{\text{Kg } ^\circ\text{C}}
 \end{aligned}$$

$$\text{density of water : } \rho_w = 1.00 \text{ Kg}/\ell$$

	$^{\circ}\text{C}$	$\frac{\text{cal}}{\text{g } ^{\circ}\text{C}}$	$\frac{\text{BTU}}{\text{Kg } ^{\circ}\text{C}}$
(b) Specific heat of ethylene glycol :	0	0.547	2.169
	20	0.569	2.256
	40	0.591	2.344
	60	0.612	2.427

density of ethylene glycol : $\rho_e = 1.113 \text{ Kg/l} \pm 1\%$ over the same temperature range.

(c) By volume, the ethylene glycol/water mixture contains:

$$350/800 = 43.75\% \text{ ethylene glycol}$$

$$450/800 = 56.25\% \text{ water}$$

Since specific heat is expressed in terms of weight, the fractional weights must be found.

The density of each liquid multiplied by its % volume in the mixture gives the weight per unit volume of each liquid:

$$\rho_w \times \%_w = 1.00 \frac{\text{Kg}}{\text{l}} \times (.5625) = 0.5625 \frac{\text{Kg water}}{\text{liter mixture}}$$

$$\rho_e \times \%_e = 1.113 \frac{\text{Kg}}{\text{l}} \times (.4375) = 0.4869 \frac{\text{Kg ethylene glycol}}{\text{liter mixture}}$$

Per liter, the mixture contains $0.5625 + 0.4869 = 1.0494 \text{ Kg}$,

$$\text{consisting of } \frac{0.5625}{1.0494} \% = 53.6\% \text{ water}$$

$$\frac{0.4869}{1.0494} \% = 46.4\% \text{ ethylene glycol,}$$

(d) The value of specific heat is expressed per unit weight, so now a value for specific heat of the mixture can be found by the % weight method. Since the specific heat of ethylene glycol varies with temperature, so would that of the mixture. We have:

	<u>% by weight</u>	<u>C_{p_e}</u>	<u>% by weight</u>	<u>C_{p_w}</u>	<u>C_{p_m}</u>	
at 0°C	(0.464 x 2.169) + (0.536 x 3.965)	=	3.132	$\frac{\text{BTU}}{\text{Kg } ^\circ\text{C}}$		
20°C	(0.464 x 2.256) + (0.536 x 3.965)	=	3.172			
40°C	(0.464 x 2.344) + (0.536 x 3.965)	=	3.213			
60°C	(0.464 x 2.427) + (0.536 x 3.965)	=	3.251			

If the values for C_{p_m} are graphed as in Figure B.9, the specific heat for the mixture at any temperature from 0 - 60°C can be read off the graph. The equation for this straight line is:

$$C_{p_m}(T) = [0.00203 T + 3.132] \text{ BTU/Kg } ^\circ\text{C}$$

where T = temperature in $^\circ\text{C}$

C_{p_m} = specific heat of mixture

- (e) Using the above data, specific heat calculations can be made using the thermodynamic relation:

$$C_{p_m} = \int_{T_1}^{T_2} C_p dT$$

we have

$$\begin{aligned} C_{p_m} &= \int_{T_1}^{T_2} [0.00203T + 3.132] dT \\ &= \left[\frac{0.00203}{2} T^2 + 3.132 T \right]_{T_1}^{T_2} \end{aligned}$$

$$C_{p_m} = \left[1.015 \times 10^{-3} (T_2^2 - T_1^2) + 3.132(T_2 - T_1) \right] \frac{\text{BTU}}{\text{Kg}}$$

- (f) So finally, collector bank (A or B) heat is calculated in the software using the formula:

$$\dot{Q} = \int_{T_1}^{T_2} \dot{m} C_p dT = \dot{m} \int_{T_1}^{T_2} C_p dT$$

where $\dot{Q}$ = BTU/min

$\dot{m}$ = Kg/min

The measured flow rate through each collector bank is 14.6 gpm.

Therefore,

$$\dot{m} = 14.6 \frac{\text{gal}}{\text{min}} \times \frac{1 \ell}{.26417 \text{ gal}} \times \frac{1.0494 \text{ Kg}}{\ell} = 58 \text{ Kg/min}$$

and

$$\begin{aligned} \dot{Q} &= 58 \frac{\text{Kg}}{\text{min}} [1.015 \times 10^{-3} (T_2^2 - T_1^2) + 3.132 (T_2 - T_1)] \frac{\text{BTU}}{\text{Kg}} \\ &= 0.05887 (T_2^2 - T_1^2) + 181.6 (T_2 - T_1) \frac{\text{BTU}}{\text{min}} \end{aligned}$$

where T_2 = collector bank A or B discharge manifold temperature

T_1 = tank A bottom temperature.

(5) Heat to Air Handler B:

This air handler uses water as the heat transfer fluid. The flow rate through the unit was measured to be 81.243 l/min. Therefore, using

$$\dot{Q} = \dot{m} C_p \Delta T, \text{ we have}$$

$$\begin{aligned} \dot{Q} &= \left[81.243 \frac{\ell}{\text{min}} \times \frac{1 \text{ Kg}}{\ell} \right] \times \left[3.965 \frac{\text{BTU}}{\text{Kg } ^\circ\text{C}} \right] \times [T_2 - T_1] ^\circ\text{C} \\ &= 322.8 (T_2 - T_1) \text{ BTU/min} \end{aligned}$$

where T_2 = Tank B hot outlet

T_1 = Air handler B outlet.

(6) Heat Delivered to Evaporator:

The heat transfer fluid which comes to the evaporator from the top of solar tank A is the same fluid which travels through the collectors, so the same heat equation applies. The flow of this fluid through the evaporator has been measured at $61.347 \frac{\ell}{\text{min}}$.

$$\begin{aligned}\dot{Q} &= \left[61.347 \frac{\ell}{\text{min}} \times \frac{1.0494 \text{ Kg}}{\ell} \right] \times \left[1.015 \times 10^{-3} (T_2^2 - T_1^2) + 3.132 (T_2 - T_1) \right] \frac{\text{BTU}}{\text{Kg}} \\ &= 0.06534 (T_2^2 - T_1^2) + 201.6 (T_2 - T_1) \text{ BTU/min}\end{aligned}$$

where T_2 = Tank A top (hot)

T_1 = Evaporator outlet

(7) Heat Taken From Condenser:

Water travelling at $51.122 \frac{\ell}{\text{min}}$ is the heat transfer medium for the condenser.

$$\begin{aligned}\dot{Q} &= \left(51.122 \frac{\ell}{\text{min}} \times \frac{1.0 \text{ Kg}}{\ell} \right) \times \left(3.965 \frac{\text{BTU}}{\text{Kg } ^\circ\text{C}} \right) \times [T_2 - T_1] ^\circ\text{C} \\ &= 203.0 [T_2 - T_1] \text{ BTU/min}\end{aligned}$$

where T_2 = Condenser outlet

T_1 = Tank B cold outlet

(8) Circulating Pump Motor Powers:

The Kw rate of each circulating pump motor was measured, and this value was converted to BTU/min for the software evaluations:

$$\text{Collector Pump : } 0.740 \text{ Kw} \times \frac{3412 \text{ BTU}}{\text{KWH}} \times \frac{1 \text{ hr}}{60 \text{ min}} = 42.08 \text{ BTU/min}$$

$$\begin{aligned}\text{Evaporator Plus : } (0.648 + 0.592) \text{ Kw} \times \frac{3412 \text{ BTU}}{60 \text{ Kw min}} &= 70.51 \text{ BTU/min} \\ \text{Condenser Pumps}\end{aligned}$$

$$\text{Air Handler B} : (0.632 + 0.640) \text{ Kw} \times \frac{3412 \text{ BTU}}{60 \text{ Kw min}} = 72.33 \text{ BTU/min}$$

(Pump plus fan)

(9) Heater Coils Power:

The power to the heater coils was also measured and converted to BTUs as the circulating pump motor powers were:

$$\dot{Q} = \left(4.5 \frac{\text{Kw}}{\text{coil}} \right) \times (6 \text{ coils}) \left(\frac{3412 \text{ BTU}}{60 \text{ Kw min}} \right) = 1535.4 \frac{\text{BTU}}{\text{min}}$$

(10) Collector Efficiency and Coefficient of Performance:

CT = total possible collected energy (BTU) per bank.

$$CT = \sum_{\text{day}} \text{insolation} \left(\frac{\text{BTU}}{\text{ft}^2} \right) \times \text{Area of Collector Bank (ft}^2\text{)}$$

$$\text{COLLECTOR A or B EFFICIENCY} : \frac{\sum_{\text{day}} \text{collector (A or B) BTU}}{CT}$$

$$\text{COLLECTOR SYSTEM EFFICIENCY} : \frac{\sum_{\text{day}} \text{collector (A + B) BTU}}{(2 \times CT)}$$

$$\text{COLLECTOR A or B COEFFICIENT OF PERFORMANCE} : \frac{\sum_{\text{day}} \text{collector (A or B) BTU}}{1/2 \sum_{\text{day}} \text{power to collector pump (BTU)}}$$

$$\text{COLLECTOR SYSTEM COEFFICIENT OF PERFORMANCE} : \frac{\sum_{\text{day}} \text{collector (A + B) BTU}}{\sum_{\text{day}} \text{power to collector pump (BTU)}}$$

(11) Coefficient of Performance for Heat Pump:

$$\text{C.O.P.} = \frac{\sum_{\text{day}} \text{BTU transferred from condenser to tank B}}{\sum_{\text{day}} \text{power to compressor (BTU)}}$$

The coefficient of performance represents the amount of energy transferred per unit of electrical energy consumed.

(12) Degree Days:

Degree days are defined as the difference between the average daily ambient temperature and 18.3°C (65°F):

$$(^{\circ}\text{C}): 18.3 - (\text{Avg. Ambient in } ^{\circ}\text{C})$$

$$(^{\circ}\text{F}): 65 - (\text{Avg. Ambient in } ^{\circ}\text{F})$$

(13) Percent Solar, System C.O.P.:

$$(a) \% \text{ SOLAR} = \frac{\sum_{\text{day}} \text{evaporator BTU absorbed}}{\text{above} + \sum_{\text{day}} (\text{BTU to compressor} + \text{heater coils})} \%$$

or, if the BTUs to the heater coils equal zero, the heat pump C.O.P. can be used:

$$(b) \% \text{ SOLAR} = \frac{\text{C.O.P.} - 1}{\text{C.O.P.}} \%$$

This equation is true because of the unitless nature of the COP value. The definition of COP is that for every x units of heat transferred from the condenser, 1 unit of electricity is used. Therefore, the evaporator energy must have been x - 1 units, assuming perfect efficiency.

(c) The preceding methods for calculating % solar were used to obtain ballpark daily figures. A more realistic method was used to calculate a monthly % solar value. This method is not performed by the software, but is done by hand. It is best explained by an example, so the actual figures from the March 1980 data will be used here.

$$\begin{aligned} \text{STEP 1: } & (47.3 \frac{\text{KWH}}{\text{day}} \text{ used by compressor}) \times (0.85 \text{ est. motor efficiency}) \\ & \times (0.85 \text{ est. compressor efficiency}) \end{aligned}$$

= 34.2 KWH/day electrical energy transferred through the heat pump to the condenser.

STEP 2: $(47.3 \frac{\text{KWH}}{\text{day}} \text{ used by compressor}) \times (3.89 \text{ C.O.P. heat pump})$
 = 184.0 KWH/day energy seen from condenser.

STEP 3: Using the results of the last two steps, the estimated amount of solar energy which was obtained through the heat pump cycle is equal to $184.0 - 34.2 = 149.8$ KWH/day. (This value represents the energy seen at the evaporator). Also, during the month of March 1980, the heating coils used 56.3 KWH/day.

Therefore,

$$\% \text{ SOLAR} = \frac{149.8}{149.8 + 34.2 + 56.3} \% = 62\%$$

STEP 4: System COPs were based on the amount of solar energy transferred per unit electrical energy required by the circulating pumps. For March 1980, 30.4 KWH/day was used by the circulating pumps. Hence,

$$\text{COP}_{\text{system}} = \frac{149.8}{30.4} = 4.9$$

(14) Off-Peak Percent Solar:

This value represents an estimated change in the % solar value if off-peak utility rates are used. Actually, in terms of BTUs consumed, the % solar value would not change at all because this value does not depend on the cost of the power. However, when using off-peak rates, the consumer would see a decrease in the energy bill. One method to simulate this savings is to discount the actual energy used by the off-peak rate. Doing

this will increase the % solar value. Thus the off-peak % solar value is really the % savings value. As energy costs decrease, the apparent % solar increases. This is the strategy behind this calculation. For March 1980, the % solar value increased from 62% to 86% using the off-peak corrections.

Figure B.1 FOUR LEVELS OF SOFTWARE PROGRAMMING

Figure B.2 FIRST LEVEL OF PROGRAMMING

Figure B.3 SECOND LEVEL PROGRAMMING

Figure B.4 THIRD LEVEL OF PROGRAMMING

Figure B.5 FOURTH LEVEL OF PROGRAMMING

COMPUTER INPUT CHANNEL	MONITORED VARIABLE
0	insolation
1	collector bank B plate temperature
2	collector bank A manifold temperature
3	collector bank B manifold temperature
4	tank A outlet to heat pump temperature
5	tank A bottom temperature
6	tank B top temperature
7	tank B bottom outlet to condenser temperature
8	heat pump evaporator outlet temperature
9	heat pump condenser outlet temperature
10	room temperature
11	air handler B outlet temperature
12	ambient temperature
13	spare
14	tank B top outlet to air handler B temperature
15	decoder input

Figure B.6 COMPUTER INPUT CHANNEL AND MONITORED VARIABLE

DECODER OUTPUT

<u>COLL</u>	<u>HP</u>	<u>AMB</u>	<u>HEATERS</u>	<u>AHA</u>	<u>READING</u>
57	81	335	750	217	
					8
X					57
	X				81
X	X				138
				X	217
X				X	274
		X			335
X	X			X	355
X		X			392
	X	X			416
X	X	X			473
			X		750
X			X		807
	X		X		831
X	X		X		888
			X	X	967
		X	X		1085
X		X	X		1142
	X	X	X		1166
X	X	X	X		1223

Figure B.7 DECODER COMBINATIONS

Figure B.8 TRANSDUCER VOLT. VS TEMPERATURE

SPECIFIC HEAT $\frac{\text{BTU}}{\text{Kg } ^\circ\text{C}}$

$\frac{350}{450}$ by volume $\frac{\text{ethylene glycol}}{\text{water}}$

Figure B.9 SPECIFIC HEAT OF COLLECTOR FLUID

Figure B.10 Main Program: MAINTST

```
      CALL MEASURE (2)
20  CALL FCKBK (N)
      IF (N.EQ.0) GO TO 10
      TYPE "HELIOS: 1980"
      TYPE "2 = INIT, 3 = CF, 4(5) = PF?"
      ACCEPT IJ
      TYPE "IJ = ", IJ
      CALL MEASURE (IJ)
10  CALL MEASURE 1
      GO TO 20
      END
```

Figure B.11 Subprogram: MEASURE
(MDUMPZ:FR)

```

SUBROUTINE MEASURE(CTR)
COMMON/MEAS/CT(16),FR(2)
COMMON/TEAKE/MONTH(2)
COMMON/MEAS2/ICT,IST,INC,IS
COMMON/CAL/AC(10),Q(30),QIN(10)
COMMON/ICAI/IA,IQ(15),IC
DIMENSION IPEN(9),MON(2),THOLD(36),IPAC(9)

C MEASUREMENT ROUTINE
4   DO 5 I=1,5
    IF(CTR.EQ.1) GO TO 6
5   CONTINUE
    TYPE "INVALID-TYPE 2,3,4,5"
    ACCEPT "MODE? ",CTR
    GO TO 4
6   GO TO (10,20,30,40,500),CTR

C INITIALIZATION
20  TYPE "MON(XXX) ? "
    CALL FCNTAL(MON,4,IER)
    IF(IER.NE.0) GO TO 20
    ACCEPT "D,Y,H,M,S? ",ID,IY,IM,IS
    MONTH(1)=MON(1)
    MONTH(2)=MON(2)
    CALL TIME(3,MON,ID,IY,IM,IS)
    WRITE(10,70)IM,IS,MON,ID,IY
    DO 19 I=12,30
19  Q(I)=0.
    IC=1
    DO 22 I=1,5
    QIN(I)=0.
22  IQ(I+9)=0.
    CALL INIT

C CHANGE FILES
30  CALL FCLFL(4,IER)
    IF(IER.EQ.0) GO TO 65
    TYPE "CLOSE ERROR",IER
65  ACCEPT "C I ",ICT," S I ",IST
    DO 68 I=2,60,2
    IF(CT1.EQ.1) GO TO 69
68  CONTINUE
    TYPE "INVALID C I"
    GO TO 65
69  DO 690 I=2,60,2
    IF(IST.EQ.1) GO TO 66
690  CONTINUE
    TYPE "INVALID S I"
    GO TO 65

```

Subprogram: MEASURE (cont'd)

```

66      TMC=0
      CALL TIME(1,MON,TD,TY,TH,IM,IS)
      WRITE(10,70)TH,IM,IS,MON,TD,TY
70      FORMAT(" CM ",I2," ",I2," ",I2," ",I2,1X,2A2,I2,I5)
80      TMC=TMC+TCT
      IF(TMC.LE.IM) GO TO 80
      IF(TMC.GE.60) TMC=0
      TMS=0
90      TMS=TMS+TST
      IF(TMS.LE.IM) GO TO 90
      IF(TMS.GE.60) TMS=0
      TYPE "HELLO "
      CALL FCNIN(1PEN,18,TER)
      IF(TER.EQ.0) GO TO 230
      TYPE "QUIT DATA RECORDING"
      TMS=61
      RETURN
230     CALL FDEF(1PEN,TER)
      CALL FOPF(1PEN,4,TER)
      TYPE "OPEN OK"
      IF(TER.EQ.0) GO TO 10
      TYPE "OPEN ERROR",TER
      GO TO 30
C DUMP FILE
40      TYPE "FILE TO DUMP "
      CALL FCNIN(1PEN,18,TER)
      IF(TER.EQ.1) RETURN
      CALL FOPF(1PEN,5,TER)
      IF(TER.EQ.0) GO TO 130
      TYPE "OPEN ERROR",TER
      GO TO 40
130     ICTR=1
160     READ(5,140,END=150,ERR=150) THOLD
140     FORMAT(36A2)
      TI=36
350     IF(THOLD(1).EQ.0) GO TO 300
      GO TO 310
300     TI=TI-1
      IF(TI.EQ.0) GO TO 340
      GO TO 350
310     WRITE(10,141)(THOLD(I),I=1,TI)
141     FORMAT(1X,36A2)
340     CALL MEASURE(ICTR)
      CALL FCPBK(TER)
      IF(TER.EQ.0) GO TO 160
150     CALL FCLFL(5,TER)

```

Subprogram: MEASURE (cont'd)

C MEASURE CHECK

```

10      CALL TIME(1,MON,TD,TY,TH,TM,TS)
      IF (IM.NE.INC) RETURN
      IF (IM.NE.INS) GO TO 190
      DO 200 I=1,14
      CALL ATD(I,THOLD(1))
200     TT(I)=115.-FLOAT(THOLD(1))/28.24138
      CALL ATD(0,THOLD(1))
      G=0.09679*FLOAT(THOLD(1))
      CALL ATD(15,THOLD(1))
      T15=THOLD(1)
      MON(1)=MONTH(1)
      MON(2)=MONTH(2)
      TY=1980
      WRITE(4,180)TH,TM,MON(1),MON(2),TD,TY,G,(TT(I),I=1,7),T15,
$      (TT(I),I=8,14)
180     FORMAT(1X,I2," ",I2,1X,2A2,I3," ",I4,F9.2,3F6.1,4F5.1,/,
$      1X,I5,2X,5F5.1,2F6.1)
      INS=INS+1*ST
      IF (INS.GE.60) INS=0
190     CALL CALC
      INC=INC+ICT
      IF (INC.GE.60) INC=0
      RETURN
500     IQ(15)=CTR
      CALL CALC
      IQ(15)=0
      RETURN
      END

```


Figure B.12 Subprogram: NONP

```

SUBROUTINE NONP
COMMON/CAL/A(10),Q(30),QTM(10)
COMMON/ICAL/IN,IO(15),IC
IN=IN+1
CALL ATD(4,IN)
CALL TMP(1,IN,T1)
A(7)=A(7)+T1
CALL ATD(5,IN)
CALL TMP(1,IN,T2)
A(8)=A(8)+T2
A(5)=A(5)+(T1+T2)/2.
CALL ATD(6,IN)
CALL TMP(1,IN,T1)
A(9)=A(9)+T1
CALL ATD(7,IN)
CALL TMP(1,IN,T2)
A(10)=A(10)+T2
A(6)=A(9)
CALL ATD(0,IN)
G=0.09679*FLOAT(IN)
A(1)=A(1)+G
CALL ATD(12,IN)
CALL TMP(1,IN,D)
A(2)=A(2)+D
CALL ATD(10,IN)
CALL TMP(1,IN,D)
A(3)=A(3)+D
CALL TMP(2,IN,D)
A(4)=A(4)+D
RETURN
END

```

```

>>C CALC2:FR/XDP1,/TTC,5
SUBROUTINE CALC
COMMON/CAL/A(10),Q(30),QIN(10)
COMMON/TCAL/IN,TQ(15),TC
COMMON/MEAS2/TC,IST,INC,INS
DIMENSION MON(2)
IF(TQ(15).EQ.5) GO TO 90
CALL ATD(15,TD)
IF(TD.LT.25) GO TO 80
IF(TD.LT.70) GO TO 10
IF(TD.LT.100) GO TO 38
IF(TD.LT.150) GO TO 14
IF(TD.LT.230) GO TO 42
IF(TD.LT.300) GO TO 16
IF(TD.LT.362) GO TO 30
IF(TD.LT.370) GO TO 22
IF(TD.LT.410) GO TO 12
IF(TD.LT.440) GO TO 32
IF(TD.LT.500) GO TO 20
IF(TD.LT.780) GO TO 44
IF(TD.LT.820) GO TO 18
IF(TD.LT.850) GO TO 40
IF(TD.LT.910) GO TO 24
IF(TD.LT.990) GO TO 46
IF(TD.LT.1115) GO TO 34
IF(TD.LT.1160) GO TO 26
IF(TD.LT.1190) GO TO 36
IF(TD.LT.1250) GO TO 28
GO TO 80
28 CALL HEAT
20 CALL COL
32 CALL ATRB
38 CALL HP
GO TO 80
26 CALL COL
34 CALL ATRB
44 CALL HEAT
GO TO 80
24 CALL HEAT
14 CALL HP
10 CALL COL
GO TO 80
36 CALL HP
CALL HEAT
30 CALL ATRB
GO TO 80
22 CALL HP
16 CALL COL
42 CALL ATRA
GO TO 80
12 CALL COL
CALL ATRB
GO TO 80

```

Subprogram: CALC (cont'd)

```

18      CALL COI
      CALL HEAT
      GO TO 80
40      CALL UP
      CALL HEAT
      GO TO 80
46      CALL HEAT
      CALL ATRA
80      CALL MONP
      IF (INC.EQ.0) GO TO 90
      RETURN
90      IF (IN.EQ.0) RETURN
      DO 91 I=1,10
91      ACT=ACT/IN
      CALL TIME(1,MON,TD,TY,TH,TA,TS)
      IF (TO(15).EQ.5) GO TO 150
      IF (TH.EQ.0) GO TO 92
      TH=TH-1
      GO TO 94
92      TD=TD-1
      TH=23
94      TX=TH+1
      DO 110 I=1,6
      IF (TO(I).NE.0) GO TO 120
110     CONTINUE
      WRITE(4,93) TH, TX, (ACT), I=1,6
93      FORMAT(" HOUR ",I2,"-",I2,6F8.2," *A:LOFF*")
      GO TO 130
120     WRITE(4,95) TH, TX, (ACT), I=1,6
95      FORMAT(" HOUR ",I2,"-",I2,6F8.2)
122     WRITE(4,96) (Q(I), I=1,7), (TO(I), I=1,6)
96      FORMAT(1X,7F9.1,/,1X,6F9)
      IF (A(1).LT.10..OR.Q(1).LE.0) GO TO 140
      Q(8)=A(1)*292.
      PCA=Q(1)/Q(8)
      PCB=Q(2)/Q(8)
      COPA=(2.*Q(1))/(TO(1)*42.08)
      COPB=(2.*Q(2))/(TO(2)*42.08)
      WRITE(4,98) PCA, COPA, PCB, COPB, Q(8)
98      FORMAT(1X,5F8.2)
130     IF (TO(15).EQ.5) RETURN
      CALL EVAL
      CALL INIT
      RETURN
140     Q(8)=0.
      Q(1)=0.
      Q(2)=0.
      GO TO 130
150     WRITE(4,160) TH, TA, (ACT), I=1,6
160     FORMAT(" VALUES FOR HOUR",I3," UP TO MIN",I3,/,1X,6F8.2)
      DO 170 I=1,10
170     ACT=ACT*IN
      GO TO 122
      END

```

Figure B.14 Subprogram: EVAL
(EVALD.FR)

```

SUBROUTINE EVAL
COMMON/CAL/A(10),Q(34),QTM(10)
COMMON/MEAS2/ICT,IST,INC,INS
COMMON/ICAL/IN,IO(15),IC
DIMENSION MON(2)
CALL TIME(1,MON,TD,TY,TM,TS)
Q(12)=Q(12)+QTM(1)+QTM(4)
Q(13)=Q(13)+QTM(1)+QTM(2)+QTM(3)+QTM(4)
Q(14)=Q(14)+Q(3)
Q(15)=Q(15)+Q(4)
Q(16)=Q(16)+Q(8)
CT=2.*Q(16)
Q(17)=Q(17)+Q(5)
Q(18)=Q(17)*0.0002931
Q(19)=Q(19)+Q(6)
Q(20)=Q(20)+Q(7)
Q(21)=Q(21)+A(1)
Q(22)=Q(22)+Q(1)
Q(23)=Q(23)+Q(2)
Q(24)=Q(24)+QTM(4)
Q(25)=Q(25)+A(2)
Q(26)=Q(26)+A(3)
Q(27)=Q(27)+A(4)
Q(28)=Q(22)+Q(23)
TQ(10)=TQ(10)+TQ(1)
TQ(11)=TQ(11)+TQ(3)
TQ(12)=TQ(12)+TQ(4)
TQ(13)=TQ(13)+TQ(5)
TQ(14)=TQ(14)+TQ(6)
TF(TM,ED,9) GO TO 40
TC=TC+1
RETURN
40 DO 30 I=25,27
30 Q(I)=Q(I)/TC
Q21=Q(21)/316.97
WRITE(4,1)Q(21),Q21,Q(16),Q(22),Q(23),Q(15),Q(19),Q(20),
$ Q(17),Q(25),Q(26),Q(27),TQ(10),TQ(12),TQ(13),TQ(14),TC
1 FORMAT(' DAILY TOT',F11.1,F5.1,3F11.1,/,1X,4F11.1,F8.2,
$ /,1X,6F8.2," ***",I2," PWS***")
TC=1
DO 95 I=12,30
95 Q(I)=0.
DO 96 I=1,5
QTM(I)=0.
96 TQ(I+9)=0.
RETURN
END

```

Figure B.15 Subprogram: COL

```

SUBROUTINE COL
COMMON/CAL/A(10),Q(30),QIN(10)
COMMON/MEAS2/ICT,IST,INC,INS
COMMON/ICAL/IN,IQ(15),IC
IQ(1)=IQ(1)+ICT
IQ(2)=IQ(2)+ICT
CALL ATD(2,IST)
CALL TMP(1,IST,T1)
CALL ATD(5,IST)
CALL TMP(1,IST,T2)
Q(1)=Q(1)+(0.05887*(T2**2-T1**2)+181.6*(T1-T2))*FLOAT(IST)
CALL ATD(3,IST)
CALL TMP(1,IST,T1)
Q(2)=Q(2)+(0.05887*(T2**2-T1**2)+181.6*(T1-T2))*FLOAT(IST)
QIN(1)=QIN(1)+42.08*FLOAT(IST)
RETURN
END

```

Figure B.16 Subprogram: HP

```

SUBROUTINE HP
COMMON/CAL/A(10),Q(30),QIN(10)
COMMON/MEAS2/ICT,IST,INC,INS
COMMON/ICAL/IN,IQ(15),IC
IQ(6)=IQ(6)+ICT
CALL ATD(4,IST)
CALL TMP(1,IST,T1)
CALL ATD(8,IST)
CALL TMP(1,IST,T2)
Q(6)=Q(6)+(0.06534*(T2**2-T1**2)+201.6*(T1-T2))*FLOAT(IST)
CALL ATD(9,IST)
CALL TMP(1,IST,T1)
CALL ATD(7,IST)
CALL TMP(1,IST,T2)
Q(7)=Q(7)+203.6*(T1-T2)*FLOAT(IST)
QIN(4)=QIN(4)+70.51*FLOAT(IST)
RETURN
END

```

Figure B.17 Subprogram: AIRB

```

SUBROUTINE AIRB
COMMON/CAL/A(10),Q(30),QIN(10)
COMMON/MEAS2/ICT,IST,INC,INS
COMMON/ICAL/IN,IQ(15),IC
IQ(4)=IQ(4)+ICT
CALL ATD(14,IN)
CALL TMP(1,IN,T1)
CALL ATD(11,IN)
CALL TMP(1,IN,T2)
Q(4)=Q(4)+322.8*(T1-T2)*FLOAT(ICT)
QIN(3)=QIN(3)+72.33*FLOAT(ICT)
RETURN
END

```

Figure B.18 Subprogram: HEAT

```

SUBROUTINE HEAT
COMMON/CAL/A(10),Q(30),QIN(10)
COMMON/MEAS2/ICT,IST,INC,INS
COMMON/ICAL/IN,IQ(15),IC
IQ(5)=IQ(5)+ICT
Q(5)=Q(5)+1535.3*FLOAT(ICT)
RETURN
END

```

Figure B.19 Subprogram: INIT

```

SUBROUTINE INIT
COMMON/CAL/A(10),Q(30),QIN(10)
COMMON/ICAL/IN,IQ(15),IC
IT=0
IN=0
DO 10 I=1,9
10 IQ(I)=0
DO 11 I=1,10
11 Q(I)=0.
A(I)=0.
RETURN
END

```

Figure B.20 Subprogram: TMP

```

SUBROUTINE TMP(ISCAL,E,IVAL,TEMP)
C   ISCALE=1: CELSIUS TEMP
C   ISCALE=2: FAHRENHEIT TEMP
IF(ISCAL.EQ.2) GO TO 2
TEMP=115.-FLOAT(IVAL)/28.24138
RETURN
2   TEMP=239.-FLOAT(IVAL)/15.69
RETURN
END

```

Figure B.21 Subprogram: AIRA

```

SUBROUTINE AIRA
COMMON/ICAL/IN,TQ(15),TC
COMMON/MEAS2/ICT,IST,INC,INS
TQ(3)=TQ(3)+ICT
RETURN
END

```

Figure B.22 Analog to Digital Handler
(ATOD:SR)

```

>>C ATOD:SR/XDPI,/TTO,S
;ANALOG TO DIGITAL
;CALL ATD(ICHANNEL,IVALUE)
;IVALUE=RETURNED 12 S. B.
;ICHANNEL= 0 TO 15
;ICHANNEL= 16 =RESET ALL UNITS
;ERROR RETURN, ICH.=-1
.NREL
.TITE ATD
.ENT ATD
.EXTD .FRET,.CPYL
FS.=0
T.=-167
EIO=54
FS.

ATD: JSR 0,CPYL ;GET ARGUMENTS
LDA 0,PT.+0,3 ;GET ICH.
LDA 1,CMS ;GET MASK 177740
AND# 0,1,SZR ;ICH.>16?
JMP ERR ;YES
LDA 1,RMS ;RESET MASK 20
AND# 0,1,SZR ;RESET?
JMP RES ;YES
MOVZL 0,0 ;MOVE
MOVZL 0,0 ;ADDRESS
MOVZL 0,0 ;INTO
MOVZL 0,0 ;POSITION
LDA 1,COUT ;GET SLOT ADDRESS
ADD 1,0 ;FORM COMMAND
DOAP 0,EIO ;START CONVERSION
LDA 2,CNT ;SET UP DELAY
STA 2,CNTR
DSZ CNTR
JMP --1

AGN: DIAP 0,EIO ;GET STATUS
LDA 1,PM ;GET POWER MASK
AND# 0,1,SNR ;CHECK POWER
JMP ERR ;POWER DOWN
SUBZL 1,1 ;GENERATE READY MASK
AND# 0,1,SNR ;READY?
JMP AGN ;NO
DIRP 0,EIO ;GET VALUE
MOVZR 0,0 ;MOVE RESULT
MOVZR 0,0 ;INTO POSITION
MOVZR 0,0
MOVZR 0,0
STA 0,PT.+1,3 ;RETURN VALUE
JSR 0,FRET ;RETURN

```


Analog to Digital Handler (cont'd)

```

RES:   DOC      2,EIO+1 ;RESET ALL
        JSR      @.FRET ;RETURN
ERR:   ADC      0,0      ;GENERATE -1
        STA      0,@1.+0,3 ;RETURN ERROR CODE
        JSR      @.FRET ;RETURN
CMS:   17774M          ;CONSTANTS
RMS:   20
COUT:  1001
PM:    10
CN1:   10
CNTR:  0
        .END

```

Figure B.23 Interrupt Processor
(INTPR:SR)

```

C INTPR:SR/XDPI,7F10,2
        ;INTERRUPT PROCESSOR
        .TTL     INTPR
        .EN1     CNTR,INTPR,CMASK,.REL
        .EXTN     CLOCK
        .LOC      0
RET:     0          ;RETURN ADDRESS
        INTPR      ;ADDRESS INT. PROCESSOR
        .ZREL
CMASK:   177777      ;INT. MASK
        .NREL
INTPR:   SKPDZ     RTC      ;REAL TIME CLOCK?
        JMP      0,CH      ;YES
        STA      0,H00      ;ILLEGAL INTERRUPT
        STA      1,H01      ;STORE ACCUM.
        INTA      0          ;FIND TROUBLE MAKER
        LDA      1,INST
        ADD      0,1          ;GENERATE INSTRUCTION
        STA      1,FIX
FIX:     0          ;CLEAR TROUBLE MAKER
        ISZ      CNTR      ;RECORD EVENT FOR DEBUG
        INC#     0,0        ;DUMMY INST.
        LDA      0,H00      ;RECOVER ACCUM.
        LDA      1,H01
        INTEN
        JMP      @RET      ;RETURN
CH:      CLOCK      ;CLOCK ADDRESS
H00:     0          ;HOLD ACC
H01:     0          ;HOLD ACT
INST:    NI OC      0        ;PARTIAL INSTRUCTION
CNTR:    0          ;ILLEGAL INTERRUPT COUNTER
        .END

```

Figure B.24 Clock Handler
(TIME2:SR)

```

>>C TIME2:SR/XDP1,ZTTC,5
;CLOCK HANDLER
;VERSION 3
;WRITTEN BY: R.G. MOLYET
;ENTERED BY: RAY BARNES
;    DATE: 11/1/79
;CALL TIMEC(MON,TD,TY,TH,TM,TS)
;63 HZ
;1-CONTROL VALUE
;1-1:RETURN COUNT AND CONTINUE
;1-2: ZERO AND START TIMING:TD,TH,TM,TS ONLY
;1-3: USE SUPPLIED VALUES AND START
.NREI
.TT11    TIME
.CNT      TIME,CLOCK
.EXITD    .FRET,.COPY,.CMASK,.RET
ES.-7
ES.
TIME:    .SR      0,COPY    ;GET ARG.
        LDA      0,0-167,3 ;GET '
MOV#     0,0,SR    ;CHECK IF 1-0
        MP       ERR     ;ERROR IF YES
        LDA      1,C3    ;GET MASK 3
AND#     0,1,SR    ;CHECK IF 1>3
        MP       ERR     ;YES
        LDA      1,INST  ;GET PARTIAL INST.
ADD      0,1       ;FORM COMPLETE INST.
        STA      1,NXT   ;STORE INST.
NXT:     0          ;GENERATED INST.
1:       READ      ;RETURN VALUE
2:       ZS        ;ZERO AND START
3:       RS        ;RESET AND START
CMb:     177773
ERR:     ADC      0,0     ;GENERATE -1
        STA      0,0-167,3 ;RETURN ERROR CODE
        SR       0,FRET  ;RETURN
INST:    MP       00,1    ;PARTIAL INSTRUCTION
READ:    LDA      0,TD    ;GET DAY
        STA      0,0-165,3 ;RETURN DAY
        LDA      0,TY     ;GET YEAR
        STA      0,0-164,3 ;RETURN YEAR
        LDA      0,TH     ;GET HOUR
        STA      0,0-163,3 ;RETURN HOUR
        LDA      0,TM     ;GET MIN.
        STA      0,0-162,3 ;RETURN MIN.
        LDA      0,TS     ;GET SEC.
        STA      0,0-161,3 ;RETURN SEC.
        SR       0,FRET  ;BACK TO CALLER
7S:      SUBC     0,0     ;CLEAR ACM
        STA      0,0-165,3 ;ZERO TD
        STA      0,0-164,3 ;ZERO TY
        STA      0,0-163,3 ;ZERO TH
        STA      0,0-162,3 ;ZERO TM
        STA      0,0-161,3 ;ZERO TS

```

Clock Handler (cont'd)

```

RS: LDA 0,C60 ;GET LIMIT
STA 0,RPC ;SET REVERSE PULSE COUNT
LDA 1,0-161,3 ;GET INITIAL SEC.
STA 1,R'S ;SET SEC. COUNTER
SUB 1,0 ;GET REVERSE SEC. COUNT
STA 0,R'S ;SET REVERSE SEC. COUNT
LDA 0,C60 ;GET LIMIT
LDA 1,0-162,3 ;GET INITIAL MIN.
STA 1,R'M ;SET MIN. COUNTER
SUB 1,0 ;GET REVERSE MIN. COUNT
STA 0,R'M ;SET REVERSE MIN. COUNT
LDA 0,C24 ;GET LIMIT
LDA 1,0-163,3 ;GET INITIAL HOUR
STA 1,R'H ;SET HOUR COUNTER
SUB 1,0 ;GET REVERSE HOUR COUNT
STA 0,R'H ;SET REVERSE HOUR COUNT
LDA 1,0-165,3 ;GET INITIAL DAY
STA 1,R'D ;SET DAY COUNTER
JMP STRT

LDA 1,0-164,3 ;GET INITIAL YEAR.
LDA 0,C1900 ;MAKE SURE
SUB7# 1,0,S7C ;11 IS IN
ADD 0,1 ;THE FORM
STA 1,R'Y ;19XX.
SUBC 0,0 ;CLEAR ACC
STRT: SUBC 0,0 ;CLEAR ACC (GO 47)
DOAS 0,RTC ;START CLOCK.
LDA 0,CMB ;GET MASK BIT
LDA 1,CMASK ;GET CURRENT MASK
AND 0,1 ;MASK OUT BIT
MSK 1 ;SEND OUT MASK
STA 1,CMASK
;TEN
ISR 0,FRET ;BACK TO CALLER
RPC: 0 ;REVERSE PULSE COUNTER
R'S: 0 ;SECOND COUNTER
R'S: 0
R'M: 0 ;MIN COUNTER
R'M: 0
R'H: 0 ;HOUR COUNTER
R'H: 0
R'D: 0 ;DAY COUNTER
R'D: 0
RMON: 0 ;MONTH COUNTER
RMON: 0
R'Y: 0 ;YEAR COUNTER
HAG: 0 ;HOLD ACC
ARG: 0 ;CONSTANTS
C24: 24.
C3: 177774
C60: 60.

```

```

MASP:      177400
            .TX1M    1          ;PACK 1 TO R
MAN1:      MAN1
            .TXT     @ 'AFEMAAPMAJULIAUSECCODE@
MAN2:      MAN2
            .TXT     AN B R K Y N L G P T V C @
CLOCK:     STA      0,HAM      ;SAVE ACM
            MTC      RTC        ;SET UP CLOCK
            DSZ      RPC        ;REVERSE PULSE COUNTER
            MP       RESTR      ;RESTORE AND LEAVE
            LDA      0,C60
            STA      0,RPC      ;RESET LIMIT
            ISZ      RTS        ;FORWARD SEC COUNTER
            DSZ      RTS        ;REVERSE SEC. COUNTER
            MP       RESTR      ;RESTORE
            SUBC     0,0         ;CLEAR ACC
            STA      0,RTS      ;RESET SEC COUNTER
            LDA      0,C60
            STA      0,RTS      ;RESET SEC LIMIT
            ISZ      RTM        ;FORWARD MIN COUNTER
            DSZ      RTM        ;REVERSE MIN COUNTER
            MP       RESTR      ;RESTORE
            SUBC     0,0         ;CLEAR ACC
            STA      0,RTM      ;CLEAR MIN COUNT
            LDA      0,C60
            STA      0,RTM      ;RESET MIN LIMIT
            ISZ      RTH        ;FORWARD HOUR COUNTER
            DSZ      RTH        ;REVERSE HOUR COUNTER
            MP       RESTR      ;LEAVE
            SUBC     0,0         ;CLEAR ACC
            STA      0,RTH      ;INIT. HOUR COUNTER.
            LDA      0,C24
            STA      0,RTH      ;RESET HOUR LIMIT.
            ISZ      FID        ;FORWARD DAY COUNTER.
            TAC#      0,0        ;DUMMY INSTR.
RESIR:     LDA      0,HAM      ;RESTORE ACC.
            TAIEA      ;ION
            MP       ORET
DTBL:      DTBL           ;DAYS IN MONTHS.
            31.
            28.
            31.
            30.
            31.
            30.
            31.
            31.
            30.
            31.
            30.
            31.
G12:       12.
G1:         1.
G1904:     1900.
            .END

```

Figure B.25 LIST OF VARIABLES FOR FORTRAN PROGRAMS

ADC	degree days ($^{\circ}\text{C}$)
ADF	degree days ($^{\circ}\text{F}$)
ALC	average house load (BTU/hr - $^{\circ}\text{C}$)
ALF	average house load (BTU/hr - $^{\circ}\text{F}$)
A()	average hourly value
A(1)	insolation (BTU/ft ² -hr)
A(2)	ambient temperature ($^{\circ}\text{C}$)
A(3)	room temperature ($^{\circ}\text{C}$)
A(4)	room temperature ($^{\circ}\text{F}$)
A(5)	tank A temperature ($^{\circ}\text{C}$)
A(6)	tank B temperature ($^{\circ}\text{C}$)
A(7)	tank A top temperature ($^{\circ}\text{C}$)
A(8)	tank A bottom temperature ($^{\circ}\text{C}$)
A(9)	tank B top temperature ($^{\circ}\text{C}$)
A(10)	tank B "T" convection temperature ($^{\circ}\text{C}$)
COPA	C.O.P. collector bank A
COPB	C.O.P. collector bank B
COPS	C.O.P. collector system
CT	total collected energy per collector bank (BTU/bank-day)
D	temporary temperature variable
DT	temperature difference
G	insolation (BTU/ft ² - hr)
I	A/D dummy variable
IC	evaluation timer for daily averages
ICT	control interval (minutes)
ICTR	control command variable
ID	integer day; decoder output integer (CALC)
IER	output integer error variable
IH	integer hour
IHOLD	output character array
IJ	A/D dummy variable
IL	output character integer
IM	integer minute
IN	NONP counter for hourly averages in CALC
INC	control interval timer (minutes)
INS	storage interval timer (minutes)
IPFN	input character array
IQ(x)	minutes (x) is on
IQ(1)	collector A/hr
IQ(2)	collector B/hr
IQ(3)	air handler A/hr
IQ(4)	air handler B/hr
IQ(5)	heater coils/hr
IQ(6)	heat pump/hr
IQ(7-9)	not used
IQ(10)	collector A/day
IQ(11)	air handler A/day
IQ(12)	air handler B/day

IQ(13) heater coils/day
 IQ(14) heat pump/day
 IQ(15) inter-hour marker (unitless)
 IS integer second
 IST storage interval (minutes)
 IY integer year
 MONTH, MON temporary month storage
 PA % of load delivered by solar
 PB % of load delivered by auxiliary
 PCA collector bank A efficiency
 PCB collector bank B efficiency
 PCS collector system efficiency
 Q(1) collector A BTU/hr
 Q(2) collector B BTU/hr
 Q(3) air handler A BTU/hr
 Q(4) air handler B BTU/hr
 Q(5) heater coils BTU/hr
 Q(6) heat pump evaporator BTU/hr
 Q(7) heat pump condenser BTU/hr
 Q(8) average insolation per collector bank (BTU/hr)
 Q(9-11) not used
 Q(12) electrical input to collector, evaporator, condenser pumps (BTU)
 Q(13) electrical input to air handler plus Q(12) (BTU)
 Q(14) air handler A BTU/day
 Q(15) air handler B BTU/day
 Q(16) total insolation per collector bank (BTU/day)
 Q(17) heater coils BTU/day
 Q(18) heater coils KWH/day
 Q(19) heat pump evaporator BTU/day
 Q(20) heat pump condenser BTU/day
 Q(21) insolation (BTU/ft² - day)
 Q(22) collector A BTU/day
 Q(23) collector B BTU/day
 Q(24) electrical energy to evaporator and condenser pumps (BTU/day)
 Q(25) average ambient temperature (°C/day)
 Q(26) average room temperature (°C/day)
 Q(27) average room temperature (°F/day)
 Q(28) collector total BTU
 Q(29-30) not used
 QIN(1) electrical energy to collector pump (BTU/hr)
 QIN(2) electrical energy to air handler A (BTU/hr)
 QIN(3) electrical energy to air handler B (BTU/hr)
 QIN(4) electrical energy to evaporator and condenser pumps (BTU/hr)
 T1 temporary temperature variable
 T2 temporary temperature variable
 TT temperature sensor reading (°C)

XDOS ROUTINES

FCKBK(IER) - checks for control-D. NO if IER=0
FCLFL(#, IER) - close file in location #
FCNIN(NAME, #, IER) - assign (#-1) alphabetic characters to
NAME array - two characters per word
FDFFL(NAME, IER) - define file NAME
FOPFL(NAME, #, IER) - open file NAME in location #
IER - error code