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SOLAR ASSISTED HEAT PUMP SYSTEM WITH VOLUME SOLAR COLLECTOR

Technical Report

By

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September 1978

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Solar Energy

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ABSTRACT

The object of this project was to analyze by computer simulation the performance of a solar assisted heat pump system with a volume collector. The system uses the attic of the house with a large south facing window as the solar collector. An air-to-water heat pump uses the attic air as a heat source to heat a volume of storage water during the heating season. During the cooling season the attic is ventilated and the heat pump uses the attic air as a heat sink while cooling the storage water.

The computer program was developed to include a heat exchanger in the attic which could by-pass the heat pump condenser cooling water, thus permitting direct heat exchange between the attic air and the storage water whenever a favorable temperature existed. The program also accounts for the effect of the incidence angle of insolation and the effect of the number of glass plates on the transmittance and absorptance of the collector and windows. Other refinements include: the use of a sophisticated nighttime setback thermostat, account of internal heat generation and infiltration loss.

Among all of the parameter variations, the use of an attic heat exchanger resulted in the maximum savings in the heating/cooling energy consumption of the house. The use of double-glazed windows too, resulted in substantial energy savings. The total energy consumption was found to depend strongly on the infiltration rate.

The program was also used to simulate the same system under weather conditions existing at several different geographic areas.

For a 1500 ft² house in the Boston area, with the best approximation for air infiltration, the total annual energy consumption is about 10,000 KWH. In a conventional electrically heated and cooled house, the energy consumption would be more than twice as much.

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Chapter 1

INTRODUCTION

Solar energy offers an attractive means to improve the performance of a heat pump used for space heating. Various configurations have been tried to achieve this objective. One unique system utilizing a "volume collector", has been investigated by Sabnis [1]. This system incorporates a reversible air-to-water heat pump with water storage for a typical one-family residence. The attic of the house has a large south facing window which causes the temperature of the attic air to rise during the sunny periods. During the heating season, this attic air is the heat source for the heat pump which delivers heat to the storage water. During the cooling season, the attic is well ventilated and the attic air is used by the heat pump as a heat sink while it cools the storage water. The water in the storage tank is circulated through a coil in a conventional air handling unit to provide the necessary heating or cooling to the house. The heat pump is located in the attic and the water from the storage tank is circulated through the water coil of the heat pump with a circulating pump. The attic is sealed from the rest of the house as well as the outdoors.

Fig. 1.1 is a schematic diagram of the system. While working in the heating mode, closed loop circulation of attic air over the evaporator coil causes the moisture in the attic air to condense out after a few defrost cycles, resulting in relatively dry air circulating over the evaporator coil. This eliminates the need for further defrost cycles in the heat pump operation. This is a significant advantage as frosting of the evaporator coil usually poses a performance penalty in the heating mode operation of an air source heat pump. The dampers in the duct work make it possible to supply either the attic air or the ambient air to the heat pump evaporator coil. When the temperature of the attic air falls below the ambient air, these dampers supply

FIG. 1.1 SCHEMATIC DIAGRAM OF THE SOLAR ASSISTED
HEAT PUMP SYSTEM WITH VOLUME SOLAR
COLLECTOR

the ambient air to the evaporator coil. Thus the heat pump always has source air at the maximum available temperature. When sufficient heat is not available from the solar assisted heat pump, auxiliary energy is supplied in the form of direct resistance heating.

In the cooling mode operation, the heat pump can be operated mostly during the night. This results in better heat pump performance due to lower sink temperatures and also permits the use of electrical energy at off-peak hours.

A basic computer routine to simulate this system on a digital computer was first developed by Sabnis [1]. Then the computer program was improved to account for some effects neglected in the first approximation and also to test the effect of including new components in the system so as to reduce the overall energy consumption.

In the initial analysis, no account was made for heat generation within the house due to lighting, appliances, etc. When considered, this factor is expected to decrease the energy consumption of the house during the heating season and increase it during the cooling season. Similarly the first program did not make any attempt to account for the night time set-back of thermostat setting during the heating season.

The detailed results of the simulation reveal that the heat pump is pumping heat from a higher temperature to a lower temperature in two cases, (i) some hot afternoons and (ii) during the change-over period between heating and cooling seasons and vice versa. The storage water temperature must change drastically before the beginning of the next season's heat pump operation. A conventional air-to-water heat exchanger coil was, therefore, included in the system which makes it possible to have direct heat exchange between the storage water and the attic air whenever a favorable temperature difference exists.

Other needed improvements include improved logic for selecting the proper mode of operation, either heating or cooling, for the first day of each month, and the use of double glass windows. The directional properties of reflectance, absorptance, and transmittance are also incorporated into the simulation procedure.

Simulation is carried out for a residential building in the Boston, Massachusetts area. Input data includes latitude and weather conditions characteristic of Boston. Simulation is carried out for the first five days of each month of the year. The results of the fifth day are fairly free of the effect of initial conditions and are used to estimate the energy consumption for the month.

Chapter 2

PRELIMINARY SIMULATION PROCEDURE

The preliminary simulation program was designed so that the behavior of each component, except the attic, is represented by a subroutine, with additional routines to predict solar radiation and weather conditions. The simulation of the attic is done in the main program itself. The inputs to the main program are the geometric parameters, the initial conditions and the constants required to predict the weather conditions at the location of the house. For the sake of flexibility, the thermal resistance of walls and windows, and the characteristics of the heat pump are also maintained as inputs to the main program and then carried into the appropriate subroutines as parameters. The main program calls the subroutines in the proper sequence as many times as necessary for the simulation before printing a summary of the results. The program listings are available on request. Some important details are given below.

The system is divided into four subsystems, viz. the house, the storage system, the heat pump and the attic. The house and the storage system are simulated by routines HOUSE and STORAG respectively. The heat pump is simulated by two routines HTPMPC and HTPMPH which predict the performance of the heat pump in cooling and heating modes respectively. The attic heat balance requires iteration on attic air temperature, which in turn affects the various heat fluxes to and from the attic greatly. Since the attic heat balance has to call other subroutines repeatedly it was found to be more convenient to treat the attic heat balance in the main program itself.

A quasi steady model has been used for the system. The temperature of the storage water is assumed to remain constant through one time interval and the added heat is assumed to change the water temperature at the

end of the interval. This assumption is not too unreasonable if storage capacity is sufficiently large. However, no such assumption can be made about the temperature of attic air as its heat capacity is relatively very small. Hence attic air temperature for any interval is determined by iteration as that temperature at which the net heat flux into the attic less the heat stored in the attic is zero. The effect of heat storage in the attic walls has not been accounted for. A detailed analysis of transients in the attic was not attempted because other parameters such as weather conditions and heat losses from the house, which would affect the results of simulation, can not be analyzed very accurately anyway. Thus a more detailed model of the attic will not improve the accuracy of the results but will increase the cost of computation substantially.

A flow chart of the preliminary simulation procedure may be seen in figure 2.1. This procedure does not model the attic heat exchanger. The inputs to the basic simulation loop for any time interval are the temperatures of storage water and attic air, and the mode in which the heat pump should be operating. The heat pump can operate in three modes, and those are represented in the program by the value of the variable MODE. $\text{MODE} = 0$ corresponds to no operation of the heat pump and heating and cooling mode operations of the heat pump are represented by $\text{MODE} = 1$ and $\text{MODE} = 2$, respectively. The program starts with $\text{MODE} = 0$ and then keeps a continuous check on the daily maximum ambient temperature. If this value exceeds 75°F then MODE is assigned a new value 2 and if this value is less than 65°F then MODE is assigned a new value 1.

The program now generates the ambient temperature for the time interval and checks on MODE. If the heat pump is found to be in its cooling mode then the program calls the routine HTPMPC such that it uses ambient temperature as the temperature of entering air over the air coil and storage water temperature from the previous hour to calculate the heat

Fig. 2.1
FLOW CHART FOR
SYSTEM SIMULATION PROCEDURE

Fig. 2.1 (contd.)

Fig. 2.1 (contd.)

removed from storage tank (QWT) and the electrical energy consumption (KWH). Since, when the heat pump operates in the cooling mode, the attic is assumed to be perfectly ventilated, the attic air temperature is set equal to the ambient air temperature. The routine HOUSE is now called which evaluates the net heat flux into the house depending on the weather conditions and the solar flux. Finally, the program calls the routine STORAG which evaluates the heat removed from house air circulating over the air-to-water heat exchanger and thus the storage water temperature for the subsequent time interval.

However, if the first check on MODE reveals that the heat pump is not operating in the cooling mode, the program initializes a new variable INAIR with a value 1. The variable INAIR = 1 represents the condition when the heat pump uses attic air circulated over the evaporator coil. INAIR = 0 means that the evaporator has ambient air circulating over it. Now the heat removed from the attic (QAT), the heat added to storage water (QWT) and electrical energy consumption are set at zero. A trial balance temperature for attic (TTRY) is taken to be the attic air temperature for the previous interval. The program now checks values of INAIR and MODE. If both are 1 then the routine HTPMPH is called such that it uses TTRY as entering air temperature and calculates the values for QAT, QWT and KWH. This is not done if both INAIR and MODE are not 1 because in that case QAT is zero. The program now evaluates the net heat flux into the attic by adding QAT to the heat exchange between the attic and the house, the ambient air and the solar flux (if any). The program evaluates the difference between net heat flux into attic and heat stored in attic air by virtue of a rise in temperature from TAT to TTRY. This difference is the variable DQ in the program. Ideally this should be zero, and if it is not, then the program evaluates the next best guess for TTRY using

Newton's method and transfers control back to the point where the check on MODE and INAIR was made.

If, however, DQ is found to be small (a limit of $DQ \leq 25$ was found to be good enough), then the balance attic air temperature (TBAL) is assigned the value TTRY and a check on MODE is made. MODE = 0 means that the heat pump did not operate for this interval and then control is transferred to the point where routine HOUSE is called and what follows is similar to the cooling mode loop. However if the heat pump did operate then a check is made on INAIR. If INAIR = 0 then routine HTPMPH is called such that it uses ambient air temperature to evaluate QAT, QWT and KWH. However actually QAT was zero since the heat pump extracted heat from ambient air so QAT is set at zero. The procedure that follows is identical to what follows if the previous check had revealed MODE = 0.

If the check on INAIR reveals that the heat pump did use attic air, then TBAL is compared with the ambient temperature. So long as TBAL is greater than ambient temperature, use of attic air for heating the evaporator is justified and the program proceeds with setting TAT = TBAL etc. But if TBAL is less than the ambient temperature, then one would be better off using the ambient air as the heat source. Hence INAIR is set at zero and the control is transferred back to the point where QAT, QWT and KWH were set at zero.

An hourly time interval was chosen for the runs of the program and simulation was done for the first five days of each month. The results for the fifth day were found to be fairly free of the effect of initial conditions and were used for estimating the energy consumption of the month.

Chapter 3

COMPONENT MODELING AND EQUATIONS

3.1 The Heat Pump

The simulation of the system requires as one input, characteristics of reversible air-to-water heat pump. The commercially available air-to-water heat pumps have their air coil indoors and hence the literature available on the operating characteristics of the air-to-water heat pumps has a very restricted range for entering air temperature. However, one would expect the temperature of the attic air to vary from 20°F to 140°F depending upon the weather. Thus the data available on operating characteristics of air-to-water heat pumps is of very little use to this simulation. The operating characteristics of an air-to-air heat pump are not of much use either as even in these characteristics, the temperature of entering air on one of the coils (the indoor coil) is assumed to be constant. Hence the operating characteristics of the heat pump for this simulation have been generated separately from the operating characteristics of the Carrier MC40 compressor (see figs. 3.1 and 3.2). These operating characteristics can be represented as a function of the evaporation temperature in the following polynomial form :

$$Q_{zc} = AQZ_1 + AQZ_2 T_e + AQZ_3 T_e^2 \quad \dots (3.1)$$

$$Q_{rc} = AQR_1 + AQR_2 T_e + AQR_3 T_e^2 \quad \dots (3.2)$$

where

Q_{zc} = cooling capacity (Btu/hr)

Q_{rc} = heat load on condenser (Btu/hr)

T_e = evaporation temperature (°F)

AQZ's and AQR's depend on condensation temperature and are tabulated in table 3.1. For intermediate values of condensation temperature, Q_{zc} and Q_{rc} should be linearly intrapolated. Equations 3.1 and 3.2 represent the

MODEL: MC40-HI-EFF'Y
REFRIG: 22
3.33 H.P. MOTOR
230-1-60
SUBCOOLING: 0 DEG F
SUPERHEAT: 20 DEG G

FIG. 3.1 VARIATION OF COOLING CAPACITY WITH
EVAPORATION TEMPERATURE FOR CARRIER
MC40 COMPRESSOR

Table 3.1
COEFFICIENTS FOR COMPRESSOR CHARACTERISTICS
POLYNOMIALS (Eqns. 3.1 and 3.2)

T_c °F	AQZ ₁	AQZ ₂	AQZ ₃ × 10 ³	AQR ₁	AQR ₂	AQR ₃ × 10 ³
90	15.1228	0.47036	7.16071	23.338	0.58911	6.32291
100	12.9854	0.45621	6.6250	21.2863	0.5965	5.78415
110	10.7683	0.43536	6.33929	18.9050	0.60767	5.36439
120	9.24969	0.35966	6.7614	16.9522	0.56778	5.66316
130	7.26122	0.33908	6.30612	14.2598	0.58865	5.03876
140	7.0000	0.22000	7.14286	13.1882	0.49740	5.96486
155	3.33717	0.22632	6.03094	8.5733	0.53262	5.01894

least square error polynomials fitting the actual data in figs. 3.1 and 3.2.

For the purpose of this simulation, the most convenient way to represent the heat pump was to have the heat rejected at the condenser and the heat removed at the evaporator represented as functions of entering air and entering water temperatures. The operating characteristics of the compressor however, give the cooling capacity and the power required as functions of evaporation temperature (T_e) and condensation temperature (T_c). T_e and T_c are in turn determined by entering air and entering water temperatures in conjunction with the evaporator and condenser. For any particular set of entering air and water temperatures, there will be a unique set of evaporation and condensation temperatures where heat transfer in the evaporator will equal the compressor cooling capacity and the heat transfer in the condenser will equal the sum of compressor capacity and electrical energy input. These balance temperatures can be found by iteration if the performance of the evaporator and the condenser could be predicted accurately. Unfortunately, not enough data could be obtained on evaporator and condenser performance hence hypothetical units were assumed to be available and their analysis is as follows.

Since the evaporator and condenser are only hypothetical, it is convenient to start with a design condition operation of the heat pump, and size the evaporator and condenser to yield the desired capacities with the selected compressor. It has been assumed that once these components are sized, UA remains constant for each of these throughout the operating range. Although the heat transfer coefficients in boiling and condensation are bound to vary, the influence of these fluctuations on U , the over all heat transfer coefficient for heat exchange, will not be very large. The reason for this is the fact that U is governed largely by the heat transfer coefficient on the side of the transfer fluid (air or

water as the case may be) rather than that on the refrigerant side. The heat transfer coefficient on the transfer fluid side is a function of the flow geometry, the properties of the transfer fluid and its flow rate, and neither of these is likely to change appreciably.

In the analysis of condenser and evaporator that follows, it has also been assumed that both these heat exchangers are essentially constant temperature heat exchangers. This would be truly so only in case where there is no superheating in the evaporator and similarly no de-superheating and subcooling in the condenser. Both these assumptions result in a model which is not an ultimately refined one, but one which is sufficiently accurate for the technical and economic scope of this investigation. Use of a highly refined model of heat pump will not improve the reliability of the results substantially because this model of the heat pump is a subsystem of the main program where other subsystems cannot be modelled more accurately.

3.1.1 Heat Pump in Heating Mode

The sizing of condenser and evaporator requires first, the specifications of the heat pump. These specifications give the flow rate of water ($\dot{m}_w$), flow rate of air ($\dot{m}_a$), entering water temperature (T'_{w1}), entering air temperature (T'_{a1}) and the heating capacity (Q'_r). With these specifications and the compressor characteristics, one can proceed to size the condenser as follows :

3.1.1.1 Sizing of Condenser :

For a heat exchanger, the effectiveness is defined as

$$\epsilon = \frac{\text{Actual Heat Transfer}}{\text{Maximum Possible Heat Transfer}}$$

Considering the condenser as a constant temperature heat exchanger, the temperature profiles would be as shown in fig. 3.3, and

Fig. 3.3 Temperature profiles in the condenser

$$\epsilon_c = \frac{(\dot{m} c_p)_w (T_{wo} - T_{wi})}{(\dot{m} c_p)_w (T_c - T_{wi})}$$

or,

$$\epsilon_c = \frac{\Delta T_w}{T_c - T_{wi}}$$

Hence, at designed operating conditions,

$$\epsilon_c = \frac{\Delta T'_w}{T'_c - T'_{wi}}$$

where,

$\Delta T'_w$ = designed change in water temperature

T'_c = condensation temperature at designed operating conditions

It can also be proved that [2]

$$\epsilon_c = 1 - \exp(-NTU_c) \quad \dots (3.3)$$

where,

$$NTU_c = \frac{(UA)_c}{(\dot{m} c_p)_w}$$

Hence,

$$-(UA)_c = (\dot{m} c_p)_w \cdot \ln \left(1 - \frac{\Delta T'_w}{T'_c - T'_{wi}} \right) \quad \dots (3.4)$$

and $\Delta T'_w$ can be calculated using

$$(\dot{m} c_p)_w \cdot \Delta T'_w = Q'_r \quad \dots (3.5)$$

Thus a design point condensation temperature (T'_c) can be selected which would give the UA for the condenser. The effectiveness of the

condenser, ϵ_c , can now be calculated using eqn. 3.3.

3.1.1.2 Sizing of Evaporator.

Once a design point condensation temperature has been selected, compressor operating characteristics can be used to determine the design point evaporation temperature T'_e such that at the combination T'_c and T'_e , the compressor capacity added to the electrical power input to the motor equals the desired heating capacity. For the evaporator,

$$\epsilon_e = \frac{\Delta T'_a}{T'_{ai} - T'_e} = 1 - \exp(-NTU_e) \quad \dots (3.6)$$

where,

$\Delta T'_a$ = change in air temperature at designed operating conditions

$$NTU_e = \frac{(UA)_e}{(\dot{m} c_p)_a}$$

Thus,

$$-(UA)_e = (\dot{m} c_p)_a \cdot \ln \left(1 - \frac{\Delta T'_a}{T'_e - T'_{ai}} \right) \quad \dots (3.7)$$

$\Delta T'_a$ can be determined from Q'_z , the compressor capacity at T'_c and T'_e , and $(\dot{m} c_p)_a$ using

$$(\dot{m} c_p)_a \cdot \Delta T'_a = Q'_z \quad \dots (3.8)$$

3.1.1.3 Matching at Off Design Conditions

Assuming that UA for condenser and evaporator remain constant throughout the operating range implies that ϵ_c and ϵ_e remain constant. Thus if the condensation temperature be T_c at an entering water temperature T_{wi} , the heat transfer in the condenser, Q_{rcnd} , will be given by

$$Q_{rcnd} = \epsilon_c \cdot (\dot{m} c_p)_w \cdot (T_c - T_{wi}) \quad \dots (3.9)$$

Similarly at these conditions if the evaporation temperature is T_e with an

entering air temperature T_{ai} , then the heat transfer in the evaporator will be

$$Q_{zevp} = \epsilon_e \cdot (\dot{m} c_p)_a \cdot (T_{ai} - T_e) \quad \dots (3.10)$$

However, at T_c and T_e the compressor characteristics yield the compressor capacity and the heat load on the condenser. Obviously for a particular set of T_{wi} and T_{ai} , there will be a unique set of T_c and T_e where the compressor capacity would match the heat transfer in the evaporator and the heat load due to the compressor on the condenser will match the heat transfer in the condenser. A search for these balance point temperatures could be carried out using a suitable iteration scheme. Fig. 3.4 shows the flow chart of the iteration scheme used to generate the heating mode operating characteristics of the heat pump. Detailed listings of APL Program MATCHHEAT and its sub-routines used to generate the operating characteristics are available in Appendix C of reference [1], and the resulting operating characteristics can be seen in figs. 3.5 and 3.6. For the purpose of generating these characteristics, the specifications of the heat pump used were the following:

Heating capacity (Q_r)	= 36,000 Btu/hr.
Entering air temperature (T'_{ai})	= 50° F.
Air flow rate 1040 SCFM i.e. $(\dot{m} c_p)_a$	= 1173.41 Btu/hr° F.
Entering water temperature (T'_{wi})	= 90° F.
Water flow rate 5 GPM i.e. $(\dot{m} c_p)_w$	= 2502.93 Btu/hr° F.

The generated operating characteristics can be represented as functions of T_{ai} by the following least squared error polynomials :

$$Q_r = AQHR_1 + AQHR_2 T_{ai} + AQHR_3 T_{ai}^2 + AQHR_4 T_{ai}^3 \quad \dots (3.11)$$

$$Q_z = AQHZ_1 + AQHZ_2 T_{ai} + AQHZ_3 T_{ai}^2 + AQHZ_4 T_{ai}^3 \quad \dots (3.12)$$

Fig. 3.4
FLOW DIAGRAM FOR GENERATING HEATING HEATING
MODE CHARACTERISTICS OF HEAT PUMP

Fig. 3.4 (contd.)

Fig. 3.5 Variation of heating capacity of the heat pump with entering air temperature.

Fig. 3.6 Variation of COP of the heat pump in heating mode with entering air temperature.

where,

Q_r = heating capacity (Btu/hr)

Q_z = refrigeration (Btu/hr)

T_{ai} = entering air temperature ($^{\circ}$ F)

The coefficients AQHR's and AQHZ's depend on entering water temperature and are tabulated in Tables 3.2 and 3.3. For intermediate values of entering water temperature, Q_r and Q_z should be linearly intrapolated.

3.1.2 Heat Pump in Cooling Mode

In the cooling mode, the evaporator and condenser exchange their functions. Hence most logical method to generate the cooling mode characteristics would be to evaluate U for the evaporator hardware working as condenser and vice versa. This however, would require a detailed design of the condenser and evaporator hardware. Even if a detailed design be available, predicting heat transfer coefficients for two phase flow would require a lot of computation which would be undesirable. Therefore the cooling mode operating characteristics have been generated using a procedure similar to one used while generating heating mode characteristics. However, it would now be necessary to start with sizing of evaporator first as the specifications would now give Q'_z rather than Q'_r . The equations change very little and have been reproduced below.

Evaporator Sizing :

$$(\dot{m} c_p)_w \cdot \Delta T'_w = Q'_z \quad \dots (3.13)$$

$$-(UA)_e = (\dot{m} c_p)_w \cdot \ln \left(1 - \frac{\Delta T'_w}{T'_{wi} - T'_e} \right) \quad \dots (3.14)$$

$$\epsilon_e = 1 - \exp (- NTU_e) \quad \dots (3.15)$$

where,

$$NTU_e = \frac{(UA)_e}{(\dot{m} c_p)_w}$$

Table 3.2
COEFFICIENTS FOR HEATING CAPACITY
POLYNOMIAL (Eqn. 3.11)

$T_{wi} \text{ } ^\circ\text{F}$	$AQHR_1$	$AQHR_2 \times 10$	$AQHR_3 \times 10^3$	$AQHR_4 \times 10^6$
70	18.577	3.1401	1.5683	- 3.8761
80	17.196	3.3499	1.3605	- 3.1478
90	15.900	3.4929	1.2180	- 2.6225
100	14.431	3.6546	1.0317	- 1.6426
110	13.219	3.6389	1.1596	- 2.3828
120	11.529	3.8142	0.9928	- 1.7579
130	10.640	3.6068	1.3496	- 3.2879
140	8.920	3.7256	1.3232	- 3.4182
150	7.081	3.9237	1.1832	- 3.0932

Table 3.3
COEFFICIENTS FOR REFRIGERATION IN
HEATING MODE POLYNOMIAL (Eqn. 3.12)

$T_{wi} \text{ } ^\circ\text{F}$	$AQHZ_1$	$AQHZ_2 \times 10$	$AQHZ_3 \times 10^3$	$AQHZ_4 \times 10^6$
70	11.758	2.3636	1.7535	- 4.6743
80	10.357	2.4347	1.5689	- 3.9422
90	9.003	2.4597	1.4224	- 3.3196
100	7.000	2.4321	1.3350	- 2.8036
110	6.957	2.1462	1.5396	- 4.2390
120	5.770	2.0891	1.5091	- 4.3325
130	5.792	1.5089	2.2436	- 7.1385
140	4.431	1.4797	2.2168	- 7.5558
150	2.562	1.5873	1.9840	- 7.2050

Condenser Sizing :

$$(\dot{m} c_p)_a \cdot \Delta T'_a = Q'_r \quad \dots (3.16)$$

$$(UA)_c = (\dot{m} c_p)_a \cdot \ln \left(1 - \frac{\Delta T'_a}{T'_c - T'_{ai}} \right) \quad \dots (3.17)$$

$$\epsilon_c = 1 - \exp(-NTU_c) \quad \dots (3.18)$$

where,

$$NTU_c = \frac{(UA)_c}{(\dot{m} c_p)_a}$$

Matching at Off Design Conditions :

$$Q_{zevp} = \epsilon_e \cdot (\dot{m} c_p)_w \cdot (T_{wi} - T_e) \quad \dots (3.19)$$

$$Q_{rcnd} = \epsilon_e \cdot (\dot{m} c_p)_a \cdot (T_c - T_{ai}) \quad \dots (3.20)$$

The logic for matching is quite similar and requires no detailed description. Detailed listing of the APL program used to generate cooling mode characteristics are available in Appendix C of reference [1], and the resulting characteristics can be seen in figs. 3.7 and 3.8. For the purpose of generating the cooling mode characteristics, the following heat pump specifications were used.

Cooling capacity (Q'_z) = 25,000 Btu/hr.

Entering air temperature (T'_{ai}) = 90° F.

Air flow rate 1040 SCFM i.e. $(\dot{m} c_p)_a = 1173.41$ Btu/hr ° F.

Entering water temperature (T'_{wi}) = 50° F.

Water flow rate 5 GPM i.e. $(\dot{m} c_p)_w = 2502.93$ Btu/hr ° F.

The cooling mode operating characteristics thus generated can be represented as functions of T_{ai} by the following least squared error polynomials.

$$Q_z = AQCZ_1 + AQCZ_2 T_{ai} + AQCZ_3 T_{ai}^2 \quad \dots (3.21)$$

Fig. 3.7 Variation of cooling capacity of the heat pump with entering air temperature.

$$Q_r = AQCR_1 + AQCR_2 T_{ai} + AQCR_3 T_{ai}^2 \quad \dots (3.22)$$

where,

Q_z = cooling capacity (Btu/hr)

Q_r = heat rejected in the condenser (Btu/hr)

T_{ai} = entering air temperature ($^{\circ}$ F)

Coefficients AQCR's and AQCR's depend on entering water temperature and have been tabulated in Tables 3.4 and 3.5. For intermediate values of entering water temperature, Q_z and Q_r should be linearly interpolated.

3.2 The House

The heat exchange between the house and environment has been assumed to comprise of the following parts :

1. Heat exchange with ambient air due to conduction and convection through walls.(q_1).
2. Heat exchange with ambient air due to conduction and convection through windows.(q_2).
3. Heat gain due to solar radiation transmitted through the windows into the house.(q_3).
4. Sensible heat exchange with the ambient due to infiltration of outside air.(q_{ins}).
5. Latent heat exchange with the ambient due to infiltration of outside air.(q_{inl}).
6. Heat exchange with attic through the attic floor.(q_{at}).
7. Heat exchange with the ground through the floor.(q_{gr}).

Apart from these, a part of the solar radiation incident on the walls is also transmitted into the house and this accounted for by using the 'Sol-air temperature' method [3]. Sol-air temperature is that temperature of the ambient air, which in the absence of the solar radiation would result in the same rate of heat exchange between ambient and house air as would

Table 3.4
 COEFFICIENTS FOR COOLING CAPACITY
 POLYNOMIAL (Eqn. 3.21)

T_{wi} °F	AQCZ ₁	AQCZ ₂ × 10	AQCZ ₃ × 10 ⁴
40	37.104	- 1.6958	- 0.4449
45	39.566	- 1.7563	- 0.5513
50	42.030	- 1.8069	- 0.7144
55	44.437	- 1.8354	- 1.0063
60	46.802	- 1.8410	- 1.4556
65	49.084	- 1.8163	- 2.1104
70	51.290	- 1.7635	- 2.9535

Table 3.5
 COEFFICIENTS FOR COOLING MODE CONDENSER
 HEAT LOAD POLYNOMIAL (Eqn. 3.22)

T_{wi} °F	AQCR ₁	AQCR ₂ × 10	AQCR ₃ × 10 ⁴
40	44.283	- 0.8314	- 3.1389
45	47.207	- 0.8996	- 2.7995
50	50.210	- 0.9695	- 2.4547
55	53.120	- 1.0065	- 2.2783
60	56.144	- 1.0564	- 2.0398
65	59.138	- 1.0828	- 1.9460
70	61.968	- 1.0593	- 2.1325

exist with the actual combination of the incident solar radiation and convective heat exchange with the out-door air. A heat balance at a sunlit surface gives the heat flux into the surface as

$$\frac{q}{A} = \alpha I_t + h_o \cdot (t_o - t_s) \quad \dots (3.23)$$

where,

$\frac{q}{A}$ = net heat flux into the surface.

α = absorptance of the surface.

I_t = total solar radiation incident on the surface.

h_o = heat transfer coefficient for heat exchange between surface and ambient air.

t_o = temperature of the ambient air.

t_s = temperature of the surface.

Eqn. (3.23) can be rewritten in terms of sol-air temperature, t_e , as

$$\frac{q}{A} = h_o \cdot (t_e - t_s) \quad \dots (3.24)$$

and,

$$t_e = t_o + \frac{\alpha I_t}{h_o} \quad \dots (3.25)$$

In the program, all the heat fluxes are calculated as heat fluxes into the house. Thus a negative value for any heat flux denotes heat loss. The net heat flux into the house is then, the sum of all the individual heat fluxes. The various heat fluxes are discussed below in greater details.

3.2.1. Heat Fluxes Through the Walls.

Of the seven heat fluxes listed, the first three take place through the walls, and they can be estimated as follows :

Considering the heat flux through the wall exclusive of window area,

$$q_1 = U_{\text{wall}} \cdot A_{\text{wall}} \cdot (t_{e1} - t_i) \quad \dots (3.26)$$

where,

U_{wall} = over all heat transfer coefficient for wall.

A_{wall} = area of the wall exclusive of window area.

t_i = indoor temperature (assumed 70°F).

t_{e1} = sol-air temperature for the wall.

U_{wall} can be calculated using

$$\frac{1}{U_{\text{wall}}} = \frac{1}{h_o} + (R_{\text{th}})_{\text{wall}} + \frac{1}{h_i} \quad \dots (3.27)$$

where,

h_o = outdoor heat transfer coefficient.

h_i = indoor heat transfer coefficient.

$(R_{\text{th}})_{\text{wall}}$ = thermal resistance of unit area of the wall.

Considering the convective/conductive heat flux through the window area in the wall,

$$q_2 = U_{\text{window}} \cdot A_{\text{window}} \cdot (t_{e2} - t_i) \quad \dots (3.28)$$

where,

U_{window} = overall heat transfer coefficient for window.

A_{window} = window area in the wall .

t_{e2} = sol-air temperature for the window surface.

U_{window} can be calculated using,

$$\frac{1}{U_{\text{window}}} = \frac{1}{h_o} + (R_{\text{th}})_{\text{window}} + \frac{1}{h_i} \quad \dots (3.29)$$

where,

$(R_{\text{th}})_{\text{window}}$ = thermal resistance of unit area of window.

The solar flux transmitted through the window area would be,

$$q_3 = I_t \cdot A_{\text{window}} \cdot \tau_g \quad \dots(3.30)$$

where,

τ_g = transmittance of the window glass.

The sum of q_1 , q_2 and q_3 would result in heat flux into the house through the particular wall, and adding this sum for all the external walls of the house yields the net heat flux into the house through the walls.

The program uses the input array WALLS for evaluating this heat flux. WALLS has four rows, each for one outer wall. The first coloumn has the area (sq.ft.) of each wall. The second column lists the fractional window area on each wall. Columns three and four list the absorptance of wall surface and window surface respectively and the last column lists the azimuth angle of each wall (due south being zero and east positive).

Other parameters viz. $(R_{th})_{\text{wall}}$, $(R_{th})_{\text{window}}$, h_o and h_i have also been kept as input parameters to the program. $(R_{th})_{\text{wall}}$ and $(R_{th})_{\text{window}}$ can be calculated for the desired construction of the wall and the window. For the type of construction selected for this simulation, $(R_{th})_{\text{wall}}$ was 21.58 hr - sq.ft.-°F/Btu. The details of wall construction and calculations of $(R_{th})_{\text{wall}}$ can be seen in Appendix B of reference [1]. $(R_{th})_{\text{window}}$ was selected to be 0.024 hr - sq.ft.-°F/Btu. The values for h_o and h_i were also selected from the ASHRAE Handbook of Fundamentals to be 5 Btu/hr - sq.ft.-°F and 1.46 Btu/hr - sq.ft.-°F respectively. In reality, h_o will depend on the wind speeds, wind direction and type of surface at the external wall. But wind speed and wind direction would be rather difficult to estimate. Hence average values have been chosen. However, at a later stage, if it is thought to be necessary, the program can very easily be modified to have a subroutine to calculate h_o as a function of several variables.

3.2.2 Heat Fluxes due to Infiltration.

To estimate the heat fluxes into the house due to infiltration of ambient air, the 'Air Change' method [4] has been used. It has been assumed that there is half air change every hour. Thus

$$Q_{inf} = 0.5 \cdot \text{Volume of the House cu.ft./hr} \quad \dots (3.31)$$

where Q_{inf} is the rate of ambient air infiltration into the house.

Infiltration results in latent and sensible heat exchange with the ambient and the two can be estimated as follows :

$$\text{Sensible heat gain, } q_{ins} = \rho \cdot c_p \cdot (t_o - t_i) \cdot Q_{inf} \dots (3.32)$$

where,

ρ = density of air (lbm/cu.ft.)

c_p = specific heat of air (Btu/lbm-°F)

Substituting the appropriate values for ρ and c_p in eqn. 3.32

yields

$$q_{ins} = 0.018 \cdot (t_o - t_i) \cdot Q_{inf} \quad \dots (3.33)$$

Similarly the latent heat gain would be

$$q_{inl} = h_{fg} \cdot \rho \cdot Q_{inf} \cdot (W_{am} - W_{in}) \quad \dots (3.34)$$

where,

h_{fg} = latent heat of condensation for water (Btu/lbm)

W_{am} = humidity ratio for ambient air

W_{in} = humidity ratio of indoor air which has been assumed to be 0.0076 lbm of water/lbm of dry air

Substituting values for h_{fg} , ρ and W_{in} in eqn. 3.34 yields

$$q_{inl} = 79.5 \cdot (W_{am} - 0.0076) \cdot Q_{inf} \quad \dots (3.35)$$

The total heat flux into the house due to infiltration would be the sum of q_{ins} and q_{inl} .

3.2.3 Heat Fluxes through Ceiling and Floor

The heat flux into the house from the attic can be estimated as

$$q_{ai} = A_c \cdot U_{ai} \cdot (t_{at} - t_i) \quad \dots (3.36)$$

where,

q_{ai} = heat flux from attic to house

A_c = ceiling area

t_{at} = temperature of the attic air

U_{ai} = overall heat transfer coefficient for heat exchange between the house and the attic

U_{ai} depends on the construction of the ceiling and good insulation should help. For the purpose of this simulation, the total resistance of ceiling was assumed to be 38.45 hr - sq.ft. - °F/Btu, which corresponds to roughly 12 inches thick R-11 insulation. The heat transfer coefficient on either side was assumed to be 1.4 Btu/hr - sq.ft. - °F. These values result in $U_{ai} = 0.0805$ Btu/hr - sq.ft. - °F.

Similarly the heat flux into the house through the floor is estimated from

$$q_{gi} = A_f \cdot U_{gi} \cdot (t_g - t_i) \quad \dots (3.37)$$

where,

A_f = floor area

U_{gi} = overall heat transfer coefficient for heat exchange between the house air and the ground

q_{gi} = net heat flux into the house from the ground

t_g = temperature of the ground

ASHRAE Handbook of Fundamentals [5] recommends $U_{gi} = 0.1$ Btu/hr-sq.ft. - °F and $t_g = 50^\circ\text{F}$. Thus,

$$q_{gi} = -2 \cdot A_f \quad \text{Btu/hr} \quad \dots (3.38)$$

3.3 The Storage Tank

In this simulation the storage tank is assumed to be thoroughly mixed. The heat capacity of the storage water is entered into the program as an input. It has also been assumed that there are no heat losses from the storage tank. Thus the storage water exchanges heat only at two points; with the refrigerant in the heat pump coil and with the house air in the air-to-water heat exchanger.

The heat flux into the storage water from the refrigerant is estimated by the routine that simulates the heat pump. The air-to-water heat exchanger is simulated by the routine STORAG itself. The effectiveness of the heat exchanger is entered into the program as an input along with the flow rates of water and air.

For the air-to-water heat exchanger, the effectiveness, ϵ , will be

$$\epsilon = \frac{C_w \cdot \Delta T_w}{C_{\min} \cdot (T_{wi} - T_{ai})} \quad \dots (3.39)$$

where,

ϵ = effectiveness of air-to-water heat exchanger

ΔT_w = change in temperature of water

T_{wi} = inlet temperature of water

T_{ai} = inlet temperature of the air

C_w = capacity rate of water stream

$C_{\min}$ = minimum capacity rate

Thus,

$$\Delta T_w = \left(\epsilon \cdot \frac{C_{\min}}{C_w} \right) \cdot (T_{wi} - T_{ai}) \quad \dots (3.40)$$

Eqn. 3.40 can be used to calculate the change in temperature of water as it flows through the heat exchanger. In the program, the factor $\left(\epsilon \cdot \frac{C_{\min}}{C_w} \right)$ is represented by the variable FHE, which is calculated in the

main program and then carried into the routine STORAG.

Figs. 3.9 and 3.10 show a schematic of the modelling strategy used in simulating the routine STORAG. The logic for the simulation of the tank and heat exchanger depends to a certain extent on whether the heat pump operates in the heating mode or the cooling mode. Fig. 3.9 shows the strategy when the heat pump either does not operate or operates in the heating mode, while fig. 3.10 shows the strategy when the heat pump operates in the cooling mode.

Since the inlet air temperature is assumed to be 70°F, the storage water cannot provide any heating if storage water temperature is less than 70°F, while the heat pump is in the heating mode. Similarly the storage water cannot provide any air cooling if storage water temperature is greater than 70°F, while the heat pump is in the cooling mode. Hence in either of these cases the heat exchanger is not simulated at all. The routine then only calculates the auxiliary heat required in the heating season and returns control to the main program. If however, the storage water can provide some heating/cooling, then the routine calculates the maximum possible heat transfer in the water-to-air heat exchanger and compares it with the heating/cooling demand for that time interval.

During the heating season if the maximum possible heat transfer in the heat exchanger is less than the heating demand, then the house would require auxiliary heat. The routine then calculates the auxiliary heat requirement and calculates the new storage water temperature corresponding to the net heat removal from the tank equal to the difference between the heat added by the heat pump and the maximum possible heat transfer in the heat exchanger. If on the other hand, the maximum possible heat transfer in the heat exchanger exceeds the heating demand, then it would be reasonable to switch the blower off after the heating demand is satisfied.

Fig. 3.9
FLOW CHART FOR ROUTINE STORAG :
HEAT PUMP IN HEATING MODE OR NOT OPERATING.

Fig. 3.9 (contd.)

Fig. 3.9

Fig. 3.10
 FLOW CHART FOR ROUTINE STORAGE :
 HEAT PUMP IN COOLING MODE

Hence the routine assumes that the heat removed from the storage water is equal to the heating demand and calculates the new temperature of the storage water.

During the cooling season a similar logic is followed with very little change. The heat is added to the storage tank through heat exchange between water and air in the heat exchanger and heat removed by the heat pump as the circulating water evaporates the refrigerant. A negative value for auxiliary heat can exist in the cooling season. This feature has been retained deliberately, to find out when the available cooling is not sufficient.

3.4 The Attic

Analysis of the attic is quite similar to that of the house as far as the estimation of heat exchange with the ambient is concerned. Of course, the 'walls' of the attic are not all vertical and one is a completely glass wall. The temperature of the attic air is not constant, but fluctuates considerably. Hence it is necessary to carry out iterations. The details of the attic heat balance are carried out in the main program itself, and have already been discussed in Chapter 2.

3.5 Solar Radiation

The amount of solar radiation incident on any surface depends on a number of parameters. Most of these parameters determine the orientation of the surface with respect to the sun. The method used in this simulation is based on the analysis outlined in ASHRAE Handbook of Fundamentals [6] and by Duffie and Beckman [7].

The total shortwave radiation reaching a surface is given by

$$I_t = I_{DN} \cdot \cos\theta + I_d + I_r \quad \dots (3.41)$$

where,

I_{DN} = direct normal irradiation

θ = angle of incidence between solar radiation and normal to the surface

I_d = diffuse sky radiation

I_r = solar radiation reflected from surrounding surfaces

I_{DN} can be calculated using

$$I_{DN} = \frac{A}{\exp(B/\sin\beta)} \quad \dots (3.42)$$

where,

A = apparent solar irradiation at air mass = 0

B = atmospheric extinction coefficient

β = solar altitude

Both A and B vary during the year because of changing earth-sun distance and seasonal changes in the dust and water vapour content of the atmosphere. These constants are calculated for each day by the routine SOLKON. The method used for the calculation is discussed by Ucar [8].

The solar altitude β can be calculated using

$$\sin\beta = \cos\phi \cdot \cos\delta \cdot \cos\omega + \sin\phi \cdot \sin\delta \quad \dots (3.43)$$

where,

ϕ = local altitude

ω = hour angle = $0.25 \cdot$ (Number of minutes from local solar noon)

δ = solar declination angle

The solar declination angle δ varies during the year. Duffie and Beckman [7] recommend use of the following equation to calculate δ for each day :

$$\delta = 23.45 \cdot \sin \left[360 \cdot \frac{284 + n}{365} \right] \quad \dots (3.44)$$

where n is the day of the year ($n = 1$ for January 1 and $n = 365$ for December 31).

In the program, δ is calculated using eqn. 3.44 by the routine SOLKON.

The angle of incidence θ can be calculated using the following equation.

$$\begin{aligned} \cos\theta = & \sin\beta \cdot \cos\lambda + \sin\lambda \cdot \cos\delta \cdot \sin\phi \cdot \cos\gamma \cdot \cos\omega \\ & + \sin\lambda \cdot \cos\delta \cdot \sin\gamma \cdot \sin\omega - \sin\lambda \cdot \sin\delta \cdot \cos\phi \cdot \cos\gamma \quad \dots (3.45) \end{aligned}$$

where,

λ = angle between the normal to the surface and vertical

γ = surface azimuth angle; due south being zero and east positive

Obviously if θ is greater 90° then the surface is actually in the shade, in which case the direct solar radiation on the surface is zero. Hence the program assumes $\cos\theta = 0$ if eqn. 3.45 yields a negative value of $\cos\theta$.

The diffuse solar radiation on a surface from clear sky is given approximately by

$$I_d = C \cdot I_{DN} \cdot F_{ss} \quad \dots (3.46)$$

where,

C = diffuse radiation factor

F_{ss} = shape factor of sky with respect to the surface.

F_{ss} can be calculated using

$$F_{ss} = (1 + \cos\lambda) / 2 \quad \dots (3.47)$$

The value of C changes during the year and is calculated for each day by the routine SOLKON using the method discussed by Ucar [8].

The total solar radiation incident on a horizontal surface is

$$I_{tH} = I_{DN} \cdot (C + \sin\beta) \quad \dots (3.48)$$

Thus the ground reflected solar radiation falling on the surface is

$$I_r = I_{DN} \cdot (C + \sin\beta) \cdot \rho \cdot (1 - F_{ss}) \quad \dots (3.49)$$

where ρ is the ground reflectivity.

The calculation of I_t is performed by the routine SOLRAD every time it is called during simulation with the surface geometry included in the parameter list.

3.6 The Temperature Functions

To predict the ambient temperatures and humidity, the generalized temperature functions as discussed by Card et al [9] are used for their simplicity and lower computation cost as compared to the weather tapes. The routine DAVG uses these functions to calculate the daily average dry bulb temperature and dew point temperature. The routine DAVG then calculates the humidity ratio using the following equation.

$$\begin{aligned}
 W_{am} = & 7.916411 \times 10^{-4} + 3.548228 \times 10^{-5} \cdot DPT + 1.615477 \times 10^{-6} \cdot DPT^2 \\
 & + 4.064036 \times 10^{-10} \cdot DPT^3 + 1.025598 \times 10^{-10} \cdot DPT^4 \\
 & + 1.202073 \times 10^{-12} \cdot DPT^5 \quad \dots (3.50)
 \end{aligned}$$

where,

DPT = dew point temperature (°F)

W_{am} = humidity ratio

Equation 3.50 was arrived at by numerical analysis of data from psychrometric tables and is reasonably accurate for dew point temperatures between 0°F and 100°F.

Chapter 4

MODIFICATIONS TO SYSTEM AND COMPONENT MODELING

4.1 Internal Heat Generation

Internal heat is generated within the house for various reasons such as the people's use in the house of electrical and heating equipment, lights, etc. This affects the heating and air conditioning demands of the house. During the heating season, this heat generation reduces the heating requirements of the house. In the cooling season, however, it increases the air conditioning load thus increasing the total energy consumption. The effect of this factor has been included in the program by including an hourly energy consumption schedule for all the lighting and other appliances in the heat balance for the house.

4.2 Thermostat Logic

Initially the thermostat of the house is set at 70° for both day and night. During the winter nights, however, the thermostat can be set back to a lower temperature to reduce the heating requirements of the house.

The set back temperature is included in the simulation program by replacing the fixed value of 70°F with the variable TSET. This variable is assigned the value TLO for hours between 11:00 p.m. and 6:00 a.m. during the heating season. Otherwise, TSET is assigned the value THI. TLO and THI have been maintained as input variables for the program.

4.3 Attic Heat Exchanger

To reduce the energy consumption of the system, an air-to-water heat exchanger is installed in the attic. It is installed in the condenser cooling water line as a by-pass to the condenser. This attic heat exchanger allows direct heat exchange between the storage water and the attic air whenever a favorable temperature difference is available.

During the change-over from heating to cooling seasons, the storage water temperature must decrease from a high value required for heating to a much lower value required for summer cooling. During the change-over from cooling season to heating season an opposite situation exists. Use of the heat pump to effect these large temperature changes causes a severe energy penalty. The attic heat exchanger can be used to minimize this. During the change-over period from heating to cooling season, this heat exchanger can be used during nights to cool the storage water. Similarly during the change-over period from cooling to heating season, the attic heat exchanger can be used during sunny afternoons to warm up the storage water. In order to decide which one of these two actions is to be carried out, a check is made on the previous days maximum and minimum temperatures (TAMAX and TAMIN). When $60^{\circ}\text{F} \leq \text{TAMAX} \leq 80^{\circ}\text{F}$, the heat pump stops operating. Further if $\text{TAMIN} < 50^{\circ}\text{F}$, the heat exchanger is used to heat the storage water, otherwise it is used to cool the storage water.

Another advantage of the heat exchanger is to use the heat stored in the attic air during hot afternoons when operation of the heat pump is prohibited by compressor overload due to high evaporator temperatures. When overload limitations occur the heat exchanger operates, either heating or cooling the water, depending upon the mode of the system at that time.

In the heating and cooling modes, the heat pump's operation is also limited by the storage water temperature, to prevent boiling or freezing of the storage water. The heat exchanger is used in these situations too, to effect direct heat exchange between the storage water and the attic air.

For the purpose of simulation, the attic heat exchanger is represented by its effectiveness ϵ , which is an input parameter to the program. If the attic air temperature and storage water temperature be T_a and T_w respectively,

then the heat transfer to the storage water, Q_w is given by

$$Q_w = \epsilon(\dot{m}c_p)_s (T_a - T_w)$$

The program at a given period allows heat transfer only in one direction, which is determined by the conditions described earlier. When the heat exchanger is needed to cool the storage water, Q_w is allowed only negative values and similarly when the heat exchanger is needed to heat the storage water, Q_w is allowed only positive values.

Fig. 4.1 shows the decision tree used in the program for using the heat pump and the attic heat exchanger. Use of the attic heat exchanger, when the heat pump operation is prevented by overload protection, has been included in the subroutines HTPMPH and HTPMPC. To simulate the use of the attic heat exchanger during changer-over period, a new subroutine ATHTEX has been included in the program. Fig. 4.2 shows the flow chart for modified system simulation procedure. The flow chart for the routine ATHTEX is included in Fig. 4.3.

4.4 Directionality of Optical Properties of Glass

Initially the transmittance, absorptance and reflectance of window and collector glass were assumed to be constant. In reality these properties depend strongly on the angle of incidence, especially at higher values of the incidence angle. To account for these variations a subroutine TAUABS has been included in the program. This routine calculates the optical properties of the window and collector glass as a function of the incidence angle, plate thickness and also the number of glass plates. The dependence of the optical properties on incidence angle is described by Whillier [10] as follows:

$$\tau = e^{-KL} \frac{(1-\rho)}{(1+\rho)} \quad (4.1)$$

(TAMAX/TAMIN : Max./Min. Outdoor Temp. of Previous Day)

Fig. 4.1 Logical Decision Tree for the Simulation

FIG. 4.2 FLOW CHART FOR MODIFIED SYSTEM SIMULATION PROCEDURE

CALCULATE ALL HEAT FLUXES, (INCLUDING SOLAR)
TO AND FROM ATTIC FOR ATTIC TEMPERATURE AT
TTRY

Fig. 4.2 (contd.)

Fig. 4.2 (contd.)

FIG. 4.3 FLOW CHART FOR ROUTINE ATHTEX

where,

τ = transmittance of glass

L = length of the path of the light ray (in)

K = extinction coefficient for glass (in^{-1})

ρ = reflectance of glass

The path length L can be calculated using

$$L = t / \cos \theta'$$

where,

t = glass thickness

θ' = angle of refraction

Reflectance ρ depends on incidence angle θ as follows

$$\rho = \frac{1}{2} \left[\frac{\sin^2(\theta - \theta')}{\sin^2(\theta + \theta')} + \frac{\tan^2(\theta - \theta')}{\tan^2(\theta + \theta')} \right] \quad (4.2)$$

The angle of refraction θ' depends on the incidence angle θ and the refractive index of glass n as follows:

$$n = \frac{\sin \theta}{\sin \theta'} \quad (4.3)$$

Properties of the glass such as K , n , t , etc., have been maintained as input parameters to the program.

The dependence of the optical properties on the number of glass plates can be analyzed assuming the glass plates have large areas. The analysis involves accounting for all the reflected and absorbed fractions of multiply reflected radiation. The result of the analysis are given below:

$$\tau_n = \tau^n \left[1 + \frac{\rho^2}{1 - \rho^2} \sum_{m=1}^n (n-m) \tau^{m-1} \right] \quad (4.4)$$

$$\rho_n = \rho \left[1 + \frac{1}{1-\rho^2} \left(\sum_{m=0}^{n-1} \tau^{2m} \right) + 1 \right] \quad (4.5)$$

where τ_n and ρ_n are the effective transmittance and reflectance of n glass plates. Equations (4.4) and (4.5) make it possible to account for the effect of double pane windows more accurately.

The routine SOLRAD was modified to give the beam radiation and diffuse radiation separately along with the incidence angle for the beam component. This with Equations (4.1) through (4.3) estimates the solar radiation passing through window and collector glass more accurately.

4.5 Other Modifications

While simulating the system on the first day of each month, MODE was set at zero initially because in five days per month simulation, there is no record of the previous days maximum and minimum temperatures. This introduced some error although its effect on the results of the fifth day would be small. This was modified by calculating the maximum and minimum temperatures for the previous day using the subroutine MAXMIN.

The program was also reorganized in a better way for data input. The data input is now subdivided into groups containing input parameters for various subsystems. Thus the program listings are now self-explanatory for preparation of input data.

Chapter 5

RESULTS AND DISCUSSION

5.1 Comparison with Conventional Systems

The preliminary program developed was used to simulate the full year performance of the system for a 1500 ft² floor area house in Boston. The summary of the house geometry is listed in Table 5.1.

The program prints out detailed results of system performance for each interval of simulation. Although the result of greatest interest is the energy consumption, the program output also includes the attic air temperature, the storage water temperature, the net heat flux into the house, the heat added to attic air by the heat pump, the heat added into storage water by the heat pump, the energy consumption of the heat pump and the auxiliary heat requirement. While printing the daily summary and estimates for the month, only the heat pump electrical energy consumption and auxiliary energy consumption are printed.

It is neither practical nor necessary to reproduce all this information, hence only the monthly energy demands are presented in this chapter, and the monthly energy demands of the proposed system have been compared with the monthly energy demands for the same house with two other possible means of heating and air-conditioning, viz. (i) an air-to-air reversible heat pump and (ii) electric heating with window air-conditioners for summer. Fig. 5.1 shows a plot of the monthly total energy demand of the proposed system. For comparison, the monthly total energy demand of the same house with all electric heating and also with a conventional air-to-air reversible heat pump have been plotted. The latter two plots were generated using the same computer program with little changes. These energy demands are also listed in Table 5.2, where the energy consumption of the heat pump and the auxiliary heat can be seen.

TABLE 5.1

SUMMARY OF SYSTEM GEOMETRY

HOUSE ORIENTATION AND DIMENSIONS	50ft. (NORTH - SOUTH) x 30ft. (EAST - WEST)
HOUSE HEIGHT	10ft.
PERCENT WINDOW AREAS IN THE WALL FACING:	
(i) EAST	15%
(ii) NORTH	10%
(iii) WEST	15%
(iv) SOUTH	30%
COLLECTOR LENGTH	50ft.
COLLECTOR AREA	50ft ²
COLLECTOR TILT	47.22° (LATITUDE +5°)
STORAGE CAPACITY	1600 GALLONS OF WATER
STORAGE WATER CIRCULATION RATE	5 GPM
HOUSE AIR CIR- CULATION RATE	1000 SCFM

TABLE 5.2

MONTHLY ENERGY CONSUMPTIONS FOR THE THREE SYSTEMS
PRELIMINARY SIMULATION

MONTH	ELE. HEAT AND AIR CONDITIO- NER (kwh)	REV. AIR-TO-AIR H.P.		PROPOSED SYSTEM	
		HEAT PUMP (kwh)	AUX. (kwh)	HEAT PUMP (kwh)	AUX. (kwh)
JAN	4,577	1,312	1,569	1,538	32
FEB	4,007	1,176	1,415	1,304	-----
MAR	3,740	1,376	977	1,332	-----
APR	2,748	1,453	218	1,293	-----
MAY	1,735	1,610	-----	881	-----
JUN	781	-----	781	-----	-----
JUL	2,210	2,210	-----	961	-----
AUG	2,222	2,222	-----	1,055	-----
SEP	2,103	2,103	-----	800	-----
OCT	1,497	-----	1,497	-----	1,508
NOV	2,695	1,490	35	1,430	-----
DEC	3,917	1,414	797	1,554	-----
TOTAL (kwh)	32,232	16,366	7,289	12,148	1,540
TOTAL FOR H. P. AND AUX. HEAT	32,232	23,655		13,688	

The monthly energy demands of an all electric house and a house with air-to-air reversible heat pump are identical for the months July through September. This is to be expected since even an all electric house would have an air-conditioner during summer. The energy demand of the proposed system during the month of June drops to zero due to the fact that the low heating demands on the house during this month are satisfied by simply circulating the hot water from the storage tank. In addition to reducing the energy demand for June, this also helps to reduce the temperature of the storage water, hence the 'Shake down' losses are less. If the heat pump is run during June, not only will there be a higher energy demand, but the storage water temperature will also remain high. This would force the heat pump to remove this extra stored heat from the water when in July, the house has air-conditioning demand and the stored water must be maintained at a low temperature.

During October, a similar situation arises and the heat pump need not operate. The heating demands of the house are met by using auxiliary heating. This is the reason why the energy demand for all the three systems is almost the same for this month. The proposed system has slightly higher energy demand due to the fact that with one side of the attic made of glass, the heat loss from attic during nights is higher thus resulting in lower attic air temperatures. Lower attic temperatures cause higher heat loss from the house to attic thus increasing the energy consumption for heating.

During summer months, the lower energy demand is mainly due to the effect of storage. The detailed output of the simulation shows that the heat pump in the proposed system tends to operate mainly during night time when the ambient temperature is lower. This reduces the energy consumption of the heat pump.

The energy consumption of the proposed system in November is quite close to that of a conventional heat pump. This results from the fact that in the proposed system, the heat pump works to heat the storage water which was initially at a lower temperature suited to summer.

5.2 Effects of the Modifications

The modified simulation program was used to study the effect of various control parameters - such as storage water temperature limit for heat pump cut off, the nighttime setback of thermostat, number of glass plates in windows and collector, etc., - on the energy consumption of the same house. As all the modifications to the program described in Chapter 4 were included one at a time, the effect of each modification on the energy consumption of the house is available separately. However, it would not be practical to reproduce all this information in this report and hence only a qualitative description of each effect is described below.

5.2.1 Effect of Internal Heat Generation

The inclusion of internal heat generation in the heat balance for the house has resulted in higher energy consumption during the cooling season and lower energy consumption during the heating season, as expected. The reason for this has already been discussed in Chapter 4.

5.2.2 Effect of Thermostat Setback During Night

The setback of the thermostat to a lower value during winter nights has resulted in reduced heat loss from the house during nights. This has reflected in lower energy consumption for the house during heating season.

5.2.3 Effect of Attic Heat Exchanger

The attic heat exchanger was found to be the most effective modification resulting in maximum savings in energy consumption. This was because use of

this heat exchanger requires virtually no energy and thus the overall consumption decreases. The use of this heat exchanger during change-over periods is found to eliminate the energy penalty paid by the heat pump to effect the large temperature change in the storage water.

The use of the heat exchanger in the heating season can be increased by lowering the storage water temperature for heat pump cut off. This stops the heat pump operation during sunny afternoons and the heat exchanger can transfer heat from hot attic air to the storage water. However, lowering this limit excessively would result in insufficient heat transfer to house air and thus would increase the auxiliary energy consumption. Thus an optimum value for this limit exists and can be found by trying various values. For this house a limit of 105°F was found to be optimum. It should be noted however, that this limit will vary with a large number of parameters for the house such as infiltration, water circulation rate, effectiveness of the air handler coil, etc.

The improvements discussed so far reduced the energy consumption of the house from about 13,700 kWh to about 7,200 kWh.

5.2.4 Effect of Double Pane Windows

Double pane windows increase the thermal resistance of the house thus resulting in energy savings. The results of simulation showed that a major part of the savings was realized in the heating season and this reduced the annual energy consumption of the house to about 5,200 kWh.

Table 5.3 lists the monthly heating/cooling energy consumption figures for the house with all these modifications. For the purpose of comparison, similar energy consumption figures for the basic design are also included in the same table.

5.2.5 Effect of Infiltration

With all the modifications described earlier, the energy consumption of the house was reduced a very low value. One possible source of error in the simulation is estimation of infiltration rate in the house. Previously it was assumed that the house has half an air change per hour. A simulation run with 1.5 air changes per hour resulted in the annual energy consumption of the house shooting up to about 13,100 kWh. Noting the strong dependence of the energy consumption on the infiltration rate, a more accurate estimation of the infiltration was made using the ASHRAE Handbook of Fundamentals [4]. It was assumed that the house would have two rooms with doors or windows on two exterior walls, and three rooms with doors or windows on only one exterior wall. Then each of the first two rooms has 1.5 air changes per hour and the other three rooms would have one air change per hour. Thus, on the average, there would be 1.2 air changes per hour. Use of this number for infiltration loss estimation resulted in annual energy consumption of about 10,200 kWh. Table 5.4 lists the monthly energy consumption figures for this case. The last column shows the number of hours of attic heat exchanger operation on the fifth day.

5.3 Comparison with 365 Days Run

An effort was made to determine the reliability of 5 days per month simulation by making one run for 365 days. This was made for an infiltration rate of 1.5 air changes per hour. The results show that the estimates based on fifth day results are far from accurate for months with mild climate. This is due to the fact that the first five days of the month do not represent the weather conditions for the month. Thus it is recommended that the 5 days per month program be used only for preliminary simulations and finally a run should be made to simulate the system for 365 days for more reliable results. Table 5.5 shows the monthly energy consumption figures for both runs.

TABLE 5.3

MONTHLY ENERGY CONSUMPTION WITH AND WITHOUT MODIFICATIONS

Month	Without Modifications			With Modifications		
	Heat Pump (KWH)	Auxiliary (KWH)	Total (KWH)	Heat Pump (KWH)	Auxiliary (KWH)	Total (KWH)
January	1,538	32	1,570	669	-	669
February	1,304	-	1,304	513	-	513
March	1,332	-	1,332	263	-	263
April	1,293	-	1,293	-	-	-
May	881	-	881	-	-	-
June	-	-	-	-	100	100
July	961	-	961	2,011	-	2,011
August	1,055	-	1,055	1,170	-	1,170
September	800	-	800	-	4	4
October	-	1,508	1,508	-	-	-
November	1,430	-	1,430	-	-	-
December	1,554	-	1,554	468	-	468
TOTAL	12,148	1,540	13,688	5,094	104	5,198

TABLE 5.4

RESULTS FOR INFILTRATION RATE OF 1.2 AIR CHANGES PER HOUR

MONTH	HEAT PUMP ENERGY (KWH)	AUXILIARY ENERGY (KWH)	TOTAL ENERGY (KWH)	HOURS OF HEAT EXCHANGER OPERA- TION ON FIFTH DAY
JANUARY	1323.98	268.67	1592.65	4
FEBRUARY	1125.77	240.62	1366.39	4
MARCH	912.98	64.32	977.30	6
APRIL	493.07	0.00	493.07	7
MAY	0.00	0.00	0.00	8
JUNE	0.00	0.00	0.00	12
JULY	2411.14	0.00	2411.14	0
AUGUST	1541.73	0.00	1541.73	0
SEPTEMBER	0.00	0.00	0.00	11
OCTOBER	0.00	0.00	0.00	8
NOVEMBER	570.76	0.00	570.76	6
DECEMBER	1119.70	54.07	1173.77	4
TOTAL	9499.13	762.29	10261.42	

TABLE 5.5

COMPARISON OF 365 DAYS RUN WITH FIVE DAYS PER MONTH RUN

MONTH	HEAT PUMP ENERGY (KWH)	AUXILIARY ENERGY (KWH)	TOTAL ENERGY (KWH)	HEAT PUMP ENERGY (KWH)	AUXILIARY ENERGY (KWH)	TOTAL ENERGY (KWH)
JANUARY	1259.51	1374.37	2633.88	1194.89	1424.17	2619.06
FEBRUARY	1085.21	1180.64	2265.85	1009.97	971.84	1981.81
MARCH	939.97	585.49	1525.46	816.97	272.13	1089.10
APRIL	603.83	0.00	603.83	389.17	2.01	391.18
MAY	0.00	0.00	0.00	0.00	2.12	2.12
JUNE	0.00	150.73	150.73	0.00	42.48	42.48
JULY	2016.95	0.00	2016.95	1349.71	0.00	1349.71
AUGUST	1546.19	0.00	1546.19	693.21	0.00	693.21
SEPTEMBER	0.00	0.00	0.00	0.00	118.92	118.92
OCTOBER	0.00	0.00	0.00	69.70	0.00	69.70
NOVEMBER	669.32	0.00	669.32	946.66	120.32	1066.98
DECEMBER	1225.48	485.84	1711.32	1247.84	880.92	2128.76
TOTAL	9346.46	3777.07	13123.53	7718.12	3834.91	11553.03

5.4 Simulation for Different Cities

The program was used to simulate the system in different weather conditions using the weather functions developed by Card, et al [4]. Table 3.4 lists all the results for an infiltration rate of 1/2 air change per hour and single pane windows. This table and also the detailed results of simulation can be used only to predict the general trend for cities other than Boston, because the system has been optimized for Boston weather. The detailed results do give an indication of the required changes to be made in the control parameters for each city. The last column in Table 5.6 shows the ratio of the average operating period of the attic heat exchanger to that of the heat pump, and thus represents the usefulness of the attic heat exchanger. Seattle uses the heat exchanger for the maximum period due to its moderate weather. Cities with hot climates do not find much use for the heat exchanger because it is most effective for the heating mode. These cities use the heat exchanger mostly during the change-over period. It is recommended that the system should be optimized for the climate of a particular area in order to estimate the energy consumption of the system in that area.

TABLE 5.6
RESULTS FROM DIFFERENT CITIES

CITY	HEAT PUMP ENERGY COOLING (KWH)	HEAT PUMP ENERGY HEATING (KWH)	AUXILIARY ENERGY (KWH)	TOTAL ENERGY (KWH)	<u>HEAT EXCHANGER HOURS</u> HEAT PUMP HOURS
SEATTLE	0	2530	589	3119	2.553
BOSTON	3339	3497	354	7189	0.700
CLEVELAND	2682	4300	283	7265	0.615
BINGHAMTON	2678	4394	419	7486	0.595
INDIANAPOLIS	4318	3056	117	7491	0.532
NEW YORK CITY	4351	2958	209	7518	0.596
WASHINGTON	5999	1582	268	7850	0.627
BUFFALO	2588	4086	516	8190	0.615
FRESNO	8256	338	0	8594	0.372
CHARLESTON	9248	0	0	9248	0.336
KANSAS CITY	6300	2675	417	9392	0.417
DALLAS	9747	0	0	9747	0.341
MINNEAPOLIS	2756	4682	2522	9960	0.566

Note: 1) The last column shows the average ratio of hours of attic exchanger operation to the hours of heat pump operation.

2) 0.5 air changes per hour assumed.

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