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STRATIFIED THERMAL STORAGE IN RESIDENTIAL SOLAR ENERGY APPLICATIONS

by

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ABSTRACT

The benefits of thermal stratification in sensible heat storage were investigated for several residential solar applications. The operation of space heating, air conditioning and water heating systems with water storage was simulated on a computer. The performance of comparable systems with mixed and stratified storage was determined in terms of the fraction of the total load supplied by solar energy. The effects of design parameters such as collector efficiency, storage volume, tank geometry, etc., on the relative advantage of stratified over well-mixed storage were assessed.

The results show that significant improvements in system performance (5-15%) may be realized if stratification can be maintained in the storage tank. The magnitude of the improvement is greatest and the sensitivity to design variables smallest in the service hot water application. The results also show that the set of design parameters which describes the optimum system is likely to be substantially different for a system employing stratified storage than for a mixed storage system. In both the water heating and space heating applications collector flow rates lower than currently suggested for mixed storage systems were found to yield optimum performance for a system with stratified storage.

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NOMENCLATURE

A_c - collector area

c_p - specific heat

C_{min} - minimum of $\dot{m}_\ell c_p$ and $\dot{m}_r c_{pr}$

D - tank diameter

e - heat exchanger effectiveness

f - fraction of total load supplied by solar

F_R - collector heat removal factor

F' - collector efficiency factor

H_T - solar insolation on tilted collector surface

H - tank height

$\dot{m}$ - mass flow rate

q - heat transfer rate

N - number of tank segments

q_u - useful energy gain in collector

T - temperature

T_a - ambient outside temperature

T_ℓ - temperature of fluid returning to tank from load

T_N - tank bottom temperature

T_{r_i} - fluid inlet temperature on room side of load heat exchanger

T_{r_o} - fluid outlet temperature on room side of load heat exchanger

T_1 - tank top temperature

U_L - collector loss coefficient, surface to environment

U_o - collector loss coefficient, fluid to environment

$(UA)_{\ell}$ - total conductance for heat loss load

$(\tau\alpha)$ - collector effective transmittance - absorptance product.

Subscripts

c - collector

i - ideally stratified storage

ℓ - load

m - mixed storage

p - partially stratified storage

INTRODUCTION

Solar energy systems which employ sensible heat storage may benefit in two ways if the collected energy is not degraded by mixing during storage. First, the effectiveness with which the energy can be used will be improved if it is supplied to the load at the temperature at which it was collected rather than at a lower mixed-storage temperature. Second, the amount of energy collected may be increased if the collector inlet fluid temperature is lower than the mixed-storage temperature. The absolute and relative importance of these effects depends on the details of the solar system design and application.

The advantages of stratification are qualitatively illustrated in the results presented by Brumleve (1) of calculations comparing stratified and mixed storage under conditions of fixed return temperatures from the load and from the collector. A quantitative comparison is reported by Duffie and Beckman (2) who simulated the operation of a solar water heating system over a one week period. They found a 9% increase in the fraction of the total load carried by solar when a three-segment stratified storage tank was substituted for mixed storage. Although this increase is certainly significant the single point comparison provides no information on the sensitivity of the advantage of stratification to design variables.

The objective of the work reported here was to assess the potential benefits of stratified storage in typical residential solar energy heating and cooling applications. The approach taken was to simulate the operation of a system first with well-mixed storage, then with stratified storage. The figure of merit used for comparison was the fraction of the total heating or cooling load provided by solar over the simulation period.

Major design parameters were varied to determine their influence on the relative performance of the two systems.

Three separate solar applications were investigated: space heating, service water heating and air conditioning. The systems employed to perform these functions used liquid cooled flat plate collectors and sensible heat storage in water. Each system was modelled mathematically to give in an initial value problem for the temperatures at various points in the system subject to the forcing functions of insolation, ambient temperature and load requirements and modified by the control strategy. The problem was solved numerically on a digital computer.

SOLAR SYSTEMS

There are a number of system parameters which must be set before a simulation can be made, and each of these may influence the relative advantage of stratified over mixed thermal storage. A major portion of this study was devoted to an assessment of the influence of these parameters. In order to accomplish this task without an excessive expenditure of computer time, simplified systems and models were used. Schematic diagrams of the three basic solar systems studied are shown in Figure 1. Components common to all three include the collector, the storage tank, the collector circulating pump and the pressure relief valve. The collector coolant (water) is introduced directly into the storage tank. Alternatively, this configuration can be viewed as one in which a primary collector coolant transfers heat to the storage water in a heat exchanger of unity effectiveness with matched stream capacity rates.

The flat plate collector was modelled using equations which are developed in Duffie and Beckman (2). The rate of heat addition to the collector fluid is given by,

$$q_u = A_c F_R [H_T(\tau\alpha) - U_L(T_N - T_a)]$$

where,

$$F_R = \frac{\dot{m}_c c_{pc}}{U_L A_c} [1 - \exp(-U_L A_c F' / \dot{m}_c c_{pc})]$$

and

$$F' = U_o / U_L \quad .$$

The collector loss coefficient U_L and the collector efficiency factor F' are considered as input parameters. In this linearized representation of the collector, changes in U_L with absorber plate temperature are neglected. Thus, the predicted improvement in collector performance with stratified storage will be slightly underestimated.

A constant rate of coolant flow is passed through the collector whenever q_u is positive. If the calculated collector outlet temperature at this flow rate exceeds the boiling point it is reset to 100°C thus simulating the operation of a pressure relief valve. Loss of mass through the valve is considered negligible.

The thermal capacity of all of the components of these systems, except for the storage tank, was neglected. The temperature within the stratified tank was considered to vary with vertical position as well as with time. A finite difference model for the tank, similar to that described by Close (3), was developed. The tank is divided into a number of horizonal slices or segments each of which is characterized by a single temperature. A well-mixed tank is represented by a single segment. In what is referred to here as an ideally stratified tank, fluid from the collector and the return flow from the load are introduced into the tank without mixing at the location where the difference between the tank temperature and the incoming fluid temperature is a minimum. The degree to which the numerical model approximates this ideal case improves as the number of tank segments increases. The sensitivity of the results to the number of storage segments was investigated as a part of this study. Experiments (4) have shown that the stratification realizable with passive storage systems is somewhat less than ideal when the inlet conditions to the tank are variable. A more realistic model for such a system is one which provides ideal stratification when the incoming water is at or beyond the storage tank temperature extremes but mixes completely otherwise. This condition, referred to as partial stratification, was modelled by introducting the collector return flow always into the top of the tank and the load return flow into the bottom of the tank. If the collector return temperature is as hot or hotter than the tank top temperature, no mixing occurs. If, however, the collector return temperature is cooler than the tank top temperature, the return flow and tank water mix downward until the mixed temperature is hotter than the rest of the tank. A similar upward mixing occurs if the load return temperature is warmer than the tank bottom temperature.

In all cases the collector is supplied with water taken from the bottom of the tank and the load is supplied with water removed from the top of the tank. Vertical conduction of heat through the tank walls and through the storage fluid as well as vertical mixing of the water due to destabilizing vertical temperature gradients induced by external heat losses are included in the stratified tank model. A detailed description of the storage tank model may be found in reference 5.

The Space Heating System

The collector and storage components are coupled with a heat exchanger, circulating pump and auxiliary heat source to form the space heating system (Figure 1a). The heating load, to be supplied either by solar or auxiliary or by a combination of the two, is calculated from

$$q_\ell = (UA)_\ell \, (T_r - T_a)$$

where the building loss coefficient $(UA)_\ell$ is held constant throughout the simulation. That portion of the load supplied by solar is transferred to the building through a water to air heat exchanger of constant effectiveness. The equations

$$q = \dot{m}_\ell C_{p\ell} \, (T_1 - T_\ell) = \dot{m}_r Cp_r \, (T_{r_o} - T_{r_i})$$

$$= e \, C_{min}(T_1 - T_{r_i})$$

with C_{min} assumed equal to $\dot{m}_r Cp_r$ model the process.

Two different system control schemes were investigated. The primary mode of operation, control mode 1, was used to generate most of the comparative data. This control scheme is widely used at the present time. In mode 1 a minimum hot air supply temperature of 40°C is specified. From this minimum temperature and the specified heat exchanger effectiveness a minimum storage tank supply temperature or reference tank temperature is calculated.

As long as the minimum specified outlet temperature can be provided from storage the load pump stays on whether the energy available from storage is sufficient to supply the entire load or not. The auxiliary heater provides the additional energy as needed. When the top tank segment temperature drops below the previously calculated reference tank temperature the load pump is turned off and all of the required energy is supplied by auxiliary.

In mode 2 the reference tank temperature, which is calculated in mode 1, is specified by the user. A reference air temperature is also specified, as in mode 1, so that operation in mode 2 does not sacrifice human comfort if the reference air temperature is set sufficiently high. If the reference tank temperature in mode 2 is set equal to the room temperature all of the available energy in the storage tank may be removed. Operation in mode 2 is exactly the same as in mode 1 when the reference tank temperature is set equal to 40°C and when the heat exchanger effectiveness is unity. The auxiliary supplies additional energy as required by the load or to meet the required reference air temperature.

The Water Heating System

In the water heating system, shown in Figure 1b, water is removed from the top of the storage tank to supply the demand. Auxiliary heat is added to this flow or cold water is mixed with it as required to meet the demand which is specified in terms of a flow rate schedule at a fixed delivery temperature. Cold make-up water at a specified, constant temperature is added directly to the storage tank. In this application the load return, or make-up, water temperature is always cooler than any stored water and is thus always added at the bottom of the stratified tank.

The Absorption Air Conditioning System

In the air conditioning system, Figure 1c, hot water from the top of the storage tank is supplied to an Arkla WF36 chiller. The operation of the chiller was simulated using a computer model taken from Leflar (6). The coefficient of performance (C.O.P.) and capacity of the chiller are specified as functions of the hot and cold water supply temperatures as shown in Table 1. The cold water temperature depends on the outside air dry bulb and wet bulb temperatures. The flow rate of water supplied to the load from the storage tank is fixed at 2498 kg/hr (11 gpm) whenever the chiller is operating. The temperature of the water returning to the storage tank is then calculated from the chiller characteristics and the working conditions. If the chiller capacity exceeds the load requirement, the chiller operates only part of the time. Auxiliary cooling is provided if the chiller cannot meet the load requirement. The auxiliary cooling unit is considered to be completely independent of the solar cooling unit; thus operation of this parallel system does not affect the solar system.

In absence of an auxiliary system the room air temperature would rise above the desired set point until the load balanced the capacity of the solar chiller. The required cooling load was calculated on the basis of a specified building loss coefficient.

Time response of the chiller was not considered. The Arkla unit requires hot water at 76.7 C or greater for operation. Higher temperatures increase the performance of the chiller up to 96.1 C; above that temperature the chiller C.O.P. and capacity are independent of the hot water temperature. The chiller C.O.P. varies between .85 and .44 depending upon the hot water temperature and the condensing water temperature. The leaving mass flow rate and temperature of the chilled water are 1634 kg/hr and 7.2°C respectively at rated capacity. The chilled water return temperature is assumed to be constant at 12.8 C.

SIMULATION

The coupled set of algebraic and ordinary differential equations describing the temperatures at various locations in any one of these systems was solved numerically on a CDC 6400 digital computer. This initial value problem was solved using an explicit marching procedure in time.

Hourly measured insolation and air temperature data for seven days in January in Boulder, Colorado were used as input to the space heating simulations. These same insolation data were used for the water heating simulations and are summarized in the Appendix. The hot water demand flow schedule of Davis and Bartera (7) was scaled down to a single family dwelling, for which the total daily load was 250 kg of 60°C water. Seven days of July weather data for Washington, D.C. were used for the air conditioning simulations. These water heating and air conditioning inputs are also documented in the Appendix.

The simulation period for most cases was seven days. For some studies the simulation period was extended to fourteen days by repeating the weather data.

Several checks were made to establish confidence in the predictions of the computer codes. The space heating code was validated by checking it against a similar simulation using TRNSYS (8) for the mixed storage case. The solar load fractions predicted by the two codes agreed to within 1/4% and the final storage tank temperatures differed by only 1/2°C. The water heating code was checked against the TRNSYS water heating simulation presented in Duffie and Beckman (2). The solar load fractions agreed to within 1%. The predictions of the models for the stratified tank have been compared with the results of experiments and are described in reference 4.

RESULTS

The comparison between systems with stratified and mixed storage is based on the fraction of the total load carried by solar energy, f, over the simulation period. For the space heating application the total load is equal to the heat lost from the building with constant indoor air temperature. Similarly, for the air conditioning simulations the total load is equal to the heat gained by the building maintained at constant temperature. The total load for the water heating application is defined as the energy required to raise the temperature of the demand flow from the make-up water temperature to the desired supply temperature. Although the heat loss from the storage tank was based on a room temperature environment this heat loss was not normally subtracted from the heating load nor added to the cooling load.

A base system was established for each of the applications by choosing values for all of the design parameters consistent with current design practice. From these base systems, variations in parameters were made to determine their effect on system performance and on the benefit of stratification. The base parameters for the three systems are listed in Table 2. The results of the one week simulations are summarized in Table 3. The benefit of ideal stratification, defined as the difference in the solar fractions for systems with ideally stratified and mixed storage divided by the mixed storage fraction, was greatest in the water heating application at nearly 16%. The benefit of ideal stratification for space heating and air conditioning was 6% and 7.5% respectively. In both the space heating and air conditioning applications the improvement in performance realized with a partially stratified tank was essentially the same as with an ideally stratified tank.

In order to determine the sensitivity of these improvements to various design and computational parameters, additional simulations were run. In each simulation only one parameter was varied from the values selected for the base systems. The results of these variations are presented in the following paragraphs.

Number of Tank Segments

The degree to which the modelled storage tank approximates an ideally stratified tank depends on the number of segments into which the tank is divided. The results, shown in Figure 2 for the space heating system, indicate little change in the computed load fraction for the stratified base system if five or more segments are used. A twenty segment tank model was used in subsequent space heating simulations while a ten segment model was used in the water heating and air conditioning simulations.

Collector Mass Flow Rate

As expected, increasing the collector mass flow rate improves the performance of the mixed storage system. The improvement, shown in Figures 3, 4 and 5, is due to the reduction in collector losses associated with the lower average collector temperature. The effect noted is due solely to the change in the fluid temperature rise through the collector since heat transfer coefficient in the liquid carrying tubes, as characterized by F', was held constant. The performance of the stratified system was found to be much less sensitive to changes in collector flow rate. A much lower flow rate than that chosen for the base system could be used without substantially changing the overall performance. In fact, for both the space heating and water heating applications an optimum collector flow rate was found which yields a maximum solar fraction for the ideally stratified tanks. In these simulations the solar fraction increases with decreasing flow rate until the point is reached at which the collector outlet temperature reaches the boiloff limit of 100°C over some portion of the simulation period. In the air conditioning application some boiloff occurs even at the highest flow rate investigated.

Collector Characteristics

The collector was characterized by two parameters, the collector loss coefficient, U_L, and the collector efficiency factor, F'. The effect of each of these parameters on the performance of the space heating system was studied independently.

The results obtained by varying the collector loss coefficient U_L with fixed F' and $\tau\alpha$ are shown in Figure 6. This can be viewed as an indication of the effect on system performance of the efficiency of collectors of similar design. Thus, a poor collector having a high U_L might have fewer glass cover plates than a more efficient collector but also a lower surface absorptivity and thicker absorber plate so that F' and $\tau\alpha$ would be comparable for the two collectors. In this comparison the advantage of stratified storage increases as collector efficiency decreases or U_L increases. This is due to the direct relation between absorber surface temperature and collector coolant temperature for constant F' and to the increased dependence of q_u on absorber plate temperature as U_L increases. On the other hand, the relative advantage of the stratified storage system over the mixed storage system increases with increasing collector efficiency when U_L and $\tau\alpha$ are fixed and F' is varied as shown in Figure 7. The so-called collector efficiency factor F' can be viewed as the ratio of U_o, the loss coefficient from the collector fluid to the environment, to U_L, the loss coefficient from the absorber surface to the environment. An increase in F' for constant U_L represents a decrease in the thermal resistance between the absorber surface and the collector fluid. Such a decrease in resistance might, for example, be associated with an increase in absorber plate thickness for a given collector configuration. These results show that stratification is most beneficial when the absorber surface temperature and the coolant temperature are tightly coupled, that is, when F' is high.

An alternative approach was adopted for the water heating and air conditioning application. Values of the collector parameters were chosen, with the guidance of reference 9, to represent typical collector design. The results, presented in Tables 4 and 5 show that in general stratification is most beneficial in systems with inefficient collectors but that a significant improvement in performance can be realized even with very efficient collectors.

The relatively small influence of collector design on the air conditioning system performance is due to the fact that the collector area far exceeds the system requirements for the period simulated. This may occur in situations where the collectors are sized to supply the winter heating demand.

Load Requirements

The relative size of the solar system and the load was varied by changing the building loss coefficient in the space heating and air conditioning applications. The results, shown in Figure 8 and 9, indicate that the advantage of stratification is relatively insensitive to the size of the load over a large range. Obviously, if the load becomes small enough such that a system with mixed storage can supply the entire demand, then the advantage of stratification in terms of increasing f must fall to zero.

Relative size of the load and the source in the water heating application was varied in several ways. The source size was varied by changing the collector area. The results shown in Figure 10 indicate a similar insensitivity of the advantage due to stratification to the magnitude of the load fraction as was found in the other two applications. Because the storage volume and load are both held fixed in this presentation, one cannot be assured that the load fraction will approach unity as the collector area approaches infinity; thus the gap between mixed and stratified performance may not close.

The effects of changes in the magnitude of the load as well as changes in the temperature levels are reflected in the results of changes in desired hot water supply temperature and make-up water temperature presented in Figures 11 and 12. Again, the advantage due to stratification is relatively insensitive to these variables.

Storage Tank Volume

The fraction of the total load carried by solar for both the mixed and stratified space heating systems increases with increasing storage volume as shown in Figure 13. The advantage due to stratification also increases with increasing tank volume. For zero storage volume the two systems would, of course, be identical.

The magnitude of the solar load fractions predicted by this relatively short simulation can be expected to be sensitive to the assumed initial tank conditions. This sensitivity will increase as the storage volume increases. All simulations began with the tank at a uniform temperature. In order to test the effect of initial conditions on the results, the simulations with variable tank volume were continued for two weeks by repeating the weather data used for the first week. It was found that for storage volumes up to $12m^3$ the tank temperature profile at the end of the second week matches the profile at the end of the first week to within 0.1°C. The load fractions for the second week are significantly different for the two systems from those for the first week, as shown by the dashed lines in Figure 13. The advantage due to stratification is seen still to increase with increasing tank volume, but the magnitude of the advantage is somewhat lower than for the first week. Similar trends are seen for the air conditioning application as shown in Figure 14, where only the second week results with no net storage are shown.

An optimum tank volume was found in the water heating application, Figure 15. Here, the results for the second week of the simulation show that heat loss to the environment becomes a significant factor for smaller systems. If perfect insulation is assumed, the optimum disappears.

Reference Tank Temperature - Control Modes

The minimum temperature at which stored energy can be used greatly affects the advantage due to stratification in the space heating application as shown in Figure 16. All runs in this sequence were made in control mode 2 with the reference air temperature set at 40°C. The results for a tank reference temperature of 40°C are identical to those obtained for the base system in mode 1. When the reference tank temperature is set to room temperature, 20°C, the advantage due to stratification drops to about 1%, but the fraction of load carried by solar for both the mixed and stratified systems increases dramatically from its level in control mode 1. It appears that for low temperature heating applications there may be more to be gained by adopting the proper control scheme than by using stratified storage.

It should be noted that these results were obtained with a load side heat exchange effectiveness of unity. In control mode 2 the effect of reducing the heat exchanger effectiveness would be to shift the curves in Figure 8 to the left. The effect of decreasing the effectiveness in control mode 1 is the same as increasing the reference tank temperature in control mode 2. For example, the load fractions indicated on Figure 16 for reference tank temperatures of 42, 45 and 53°C correspond to those in control model 1 with heat exchangers having effectiveness values of 0.9, 0.8 and 0.6 respectively.

In the air conditioning application the tank reference temperature is replaced by the chiller performance map (Table 1). In the water heating application the demand is always supplied with water taken from the storage tank; in effect the tank reference temperature is equal to the make-up water temperature.

Tank Height to Diameter Ratio

Increasing the tank height to diameter ratio for a fixed tank volume improves the performance of the stratified tank by reducing the effects of vertical conduction. On the other hand, variations from optimum configuration of H/D = 1 will increase external heat losses from the tank for a constant thickness of insulation. The results of simulations with H/D ranging from 1 to 8 indicate that vertical conduction is not a significant factor in the space heating and air conditioning applications. The optimum H/D for these applications is one.

The optimum for water heating, however, is not one. The larger temperature differences maintained in the water heating storage tank cause the optimum to change to about 3 for ideal stratification and about 2 for partial stratification as shown in Figure 17. The difference in performance between the optimum ratio and other ratios between 1 and 10 is very small.

Tank Losses

For air conditioning applications the tank heat losses may add to the heat load if the tanks are located in the cooled space. To check on the magnitude of this effect and on several other aspects of the tank performance some additional air conditioning simulations were run. The results of these simulations, tabulated in Table 6, show that the addition of tank losses to the load decreases the solar fractions by 2%. The effects of vertical conduction through the water and the tank walls are verified to be insignificant by calculations which show the solar load fractions to be unchanged to the third decimal point when the conductivities are set to zero.

Stratification Model

The previously described results show a considerable variation in the relative importance of the stratification model used in the simulations. This is because the base systems for the three applications benefit from stratification in different ways. The sole effect of stratification in the heating applications is to increase the amount of energy collected. This may be the result of increased collector efficiency due to lowered average surface temperatures or of reduced boiloff losses.

In the water heating application the coupling between the load side and collector side of the system is minimal. The makeup water temperature is fixed and the load side flow rate is altered only if the tank temperature exceeds the desired supply temperature. Because the collector efficiency depends directly on the tank bottom temperature one can get a good idea of the importance of the stratification model in this application by studing the tank temperature histories. A comparison between the ideal and partial stratification models and mixed storage is shown in Figure 18 where the temperatures at the top, center and bottom of the tank are compared over a 24 hour period. At the beginning of this period the time is midnight, and the tanks are discharging slowly due to heat losses and a slight demand. At time 55 hours the collector pump starts as the sun rises. Initially, the outlet temperature from the collectors is less than the top temperature in the stratified tanks. In the ideally stratified tank this water is inserted at the appropriate location near the center of the tank, and stratification is preserved. In the partially stratified tank this water enters at the top of the tank and mixes downward until gravitational stability is restored. About three hours after sunrise an equilibrium charging rate is reached for the tanks with the bottom of the ideally stratified tank slightly cooler than the bottom of the partially stratified tank and both of these temperatures significantly lower than the mixed tank temperature. The tank temperatures diverge again near sunset when some mixing occurs in the partially stratified tank. Judging from the relative levels of these temperatures during the period of insolation for this day one would anticipate the relative load fractions shown in Table 3. The early morning mixing, which increases the bottom temperature of the partially stratified tank, enables the collector to come up to temperature sooner and provide slightly hotter water during most of the day than with ideally stratified storage. This has no effect on the solar load fraction for the water heating application but it may for other applications.

The tank temperature histories in the space heating application, Figure 19, are similar to those for water heating except that the temperature differences are smaller. The morning and evening mixing in the partially stratified tank consequently have a smaller influence on the collector performance. In this application the net effect of stratification is still just to increase the collection efficiency, and the tank bottom temperature remains a good indicator of system performance. The connection between the tank bottom temperature and the stratification model is, however, more subtle than in the water heating application. This is because the load-side flow, which influences the stratified tank temperature profile, depends on the tank temperature profile through the tank reference temperature. Depending on the histories of the collector and load flows and the insolation, the temperature at the bottom of the partially stratified tank may at times be lower than that at the bottom of the ideally stratified tank and if these times are correlated with periods of peak insolation the result may be higher solar load fractions for the partially stratified tanks.

The principal effect of stratification in the air conditioning application simulated was to improve the performance of the chiller. The tank temperature histories shown in Figure 20 indicate that over much of the collection period shown the mixed tank temperature and the temperatures at the bottom of the stratified tanks were almost the same, indicating that the collectors were operating almost identically. In fact, for much of the day the collector outlet temperature is 100°C for all systems, and energy is being thrown away by boiloff. The stratification realized during this period is due to the temperature drop through the chiller. The improvement in performance realized is due to the fact that the systems with stratified tanks are delivering water at 100°C to the chiller while the system with mixed storage is providing 93°C water to the chiller. Thus the maximum usable capacity of the chiller in the stratified systems is greater than that in the mixed system. See Table 1. In this application the system with partial stratification gets up to temperature faster than the system with ideal stratification and can thus carry more of the load when required in the morning hours.

Clearly, these one week simulations do not encompass every possible combination of operating conditions and solar system characteristics. They do, however, reveal certain trends which allow one to anticipate critical combinations. For example, from the foregoing discussion one would anticipate that the performance of ideally and partially stratified systems would diverge, in an application in which the collector performance was important, as the temperature differences increased and as the frequency of variation of insolation increased. Thus the mixing associated with an intermittent cloud cover would be expected to reduce the relative performance of a partially stratified system. To test this hypothesis the space heating simulation was run with the base insolation multiplied by the function $(0.6 + 0.4 \sin \omega t)$. The results for various "cloud frequencies" and two collector flow rates are shown in Table 7. For a given collector flow rate the improvement in performance of ideal over partial stratification is relatively insensitive to frequency for $2\pi \leq \omega \leq 8\pi$ but is greater than that for $\omega=0$. For frequencies much greater than 8π (corresponding to a period of 1/4 hr) the collector thermal inertia, neglected in this model, would tend to drive the performance of the ideally stratified tank down to the level of the partially stratified tank. The performance of the stratified tank is sensitive to both collector and load flow rates since both have a strong influence on the maximum temperature difference across the tank.

It is clear from these data that the set of parameters which describe an optimum system will be different depending on whether or not the storage tank is stratified or mixed. It appears that the impact of the stratification model on the system performance will be more evident when these parameters are closer to the optimum for a stratified system,but even then partial stratification should provide a considerable improvement over mixed storage.

CONCLUSIONS

The results of these simulations of solar water heating, space heating and air conditioning applications show that improved performance will be realized if stratification can be maintained in the storage tank. The magnitude of the improvement depends strongly on certain design parameters such as collector efficiency, collector coolant flow rate, tank volume, etc. When these parameters are chosen on the basis of current design practice an improvement in solar load fraction of 5 to 15% is predicted compared with an identical system using mixed storage. The benefits of stratification may be further heightened if the design parameters are chosen to take advantage of the special characteristics of stratified storage. Lowering the flow rates on both the collector and load side of the tank increases temperature differences and may improve the absolute as well as relative performance of a stratified system. This improvement must be balanced against possible increases in the size of heat exchangers, and ultimately capital cost considerations must be included in the parameter selection procedure. The purposes of this study were only to establish a base for comparison and to identify the sensitive parameters.

In some applications partial stratification, as may be achieved in a tank with fixed inlet locations, provides as much improvement over mixed storage as does ideal stratification. This level of stratification can easily be obtained using passive devices. Vertical conduction of heat through the stored water and through the tank walls has a negligible effect on stratification in the applications simulated.

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TABLE 1. Arkla Chiller Performance

Data - Standard Model WF-36

Hot water inlet temp. (F)	Cond. water inlet temp. (F)	Capacity (B/HR)	COP
170	75	15,600	0.70
	80	9,700	0.59
	85	6,400	0.44
	90	3,000	0.35
175	75	24,000	0.80
	80	17,300	0.73
	85	13,100	0.61
	90	8,400	0.50
180	75	31,200	0.84
	80	24,400	0.78
	85	19,400	0.67
	90	14,200	0.60
185	75	37,200	0.85
	80	31,100	0.81
	85	25,600	0.71
	90	19,300	0.63
190	75	42,000	0.84
	80	36,800	0.80
	85	31,300	0.73
	90	23,800	0.63
195	75	42,000	0.80
	80	40,600	0.76
	85	36,000	0.72
	90	27,600	0.62
200	75	42,000	0.75
	80	41,800	0.71
	85	40,200	0.72
	90	30,500	0.60
205	75	42,000	0.68
	80	42,000	0.66
	85	42,000	0.69
	90	32,500	0.58

TABLE 2. Base System Parameters

	Space Heating	Water Heating	Air Conditioning
Storage Tank			
Volume (m^3)	4.0	0.25	4.0
H/D	2.0	2.0	2.0
Wall thickness (m)	.0015	.0015	.0015
Wall Conductivity (kJ/m hr °C)	171.4	171.4	171.4
Heat loss coeff. (kJ/m^2 hr °C)	1.6	1.44	1.44
Number of segments	20	10	10
Initial tank temperature (°C)	50.0	60.0	80.0
Reference tank temperature (°C)	40.0	--	--
Collector			
Mass flow rate (kg/hr)	2500.0	150.0	2500.0
Area (m^2)	75.0	4.0	75.0
F'	0.876	0.9	0.9
$(\tau\alpha)$	0.84	0.8	0.8
U_L kJ/m^2 hr °C	17.9	15.0	15.0
Angle with horizontal (degrees)	40.0	45.0	45.0
Load			
Mass flow rate (kg/hr)	2500.0	variable	2498.0
$(UA)_\ell$ (kJ/hr °C)	1000.0	--	7000.0
Room temperature (°C)	20.0	20.0	22.0
Heat exchanger effectiveness	1.0	--	--
Air mass flow rate (kg/hr)	2500.0	--	--
Minimum air outlet temp. (°C)	40.0	--	--
Hot water supply temp. (°C)	--	60.0	--
Make-up water temp. (°C)	--	15.0	--

TABLE 3. Simulation Results for Base Systems.

System	f_i	f_p	f_m	$\frac{f_i - f_m}{f_m}$ (%)	$\frac{f_p - f_m}{f_m}$ (%)
Space heating	0.57	0.57	0.54	6	6
Water heating	0.64	0.62	0.55	16	12
Air conditioning	0.57	0.57	0.53	8	8

TABLE 4. Effect of Collector Design on System Performance, Water Heating.

Collector	F'	$\tau\alpha$	U_L (kJ/m^2 °C hr)	f_m f_p f_i (%)	$\frac{f_p - f_m}{f_m}$ $\frac{f_i - f_m}{f_m}$ (%)
one cover nonselective Base System	0.90	0.80	15.0	55.2 61.8 64.0	 11.9 15.8
one cover Selective	0.94	0.80	8.3	65.3 70.0 72.8	 7.2 11.4
two covers nonselective	0.94	0.72	8.5	60.2 65.2 67.7	 8.4 12.4
two covers selective	0.96	0.72	5.7	65.4 69.6 72.3	 6.4 10.5

TABLE 5. Effect of Collector Design on System Performance, Air Conditioning

Collector	F'	$\tau\alpha$	U_L (kJ/m^2 °C hr)	f_m f_p f_i (%)	$\frac{f_p-f_m}{f_m}$ $\frac{f_i-f_m}{f_m}$ (%)
one cover				53.3	
nonselective	0.90	0.80	15.0	57.3	7.5
<u>Base System</u>				57.4	7.6
one cover				55.7	
selective	0.94	0.80	8.3	59.6	7.0
				59.5	6.8
two covers				54.8	
nonselective	0.94	0.72	8.5	58.8	7.2
				58.7	7.1
two covers				55.9	
selective	0.96	0.72	5.7	59.8	6.9
				59.7	6.8

TABLE 6. Effect of Storage Tank Parameters on System Performance, Air Conditioning.

	f_m f_p f_i (%)	$\frac{f_p - f_m}{f_m}$ $\frac{f_i - f_m}{f_m}$ (%)
Base system	53.3 57.3 57.4	 7.5 7.6
Tank wall k = 0	53.3 57.3 57.4	 7.5 7.6
Water k = 0	53.3 57.3 57.4	 7.5 7.6
Tank losses = 0	54.4 58.2 58.3	 7.0 7.2
Tank losses added to load	51.3 55.2 55.3	 7.6 7.6

TABLE 7. Effect of Insolation Modulation and Flow Rates on Space Heating System Performance.

ω (rad/hr)	$\dot{m}_c$ (kg/hr)	$\dot{m}_\ell$ (kg/hr)	f_i	f_p	f_m	$\frac{f_i - f_m}{f_m}$ (%)	$\frac{f_p - f_m}{f_m}$ (%)
0	2500	2500	.351	.349	.321	9.4	9.0
2π			.384	.380	.349	9.9	8.8
4π			.385	.380	.350	9.8	8.5
8π			.385	.381	.350	9.8	8.6
0	1000		.350	.348	.306	14.3	13.7
2π			.380	.369	.332	14.2	11.0
4π			.381	.370	.333	14.3	11.1
8π			.381	.371	.333	14.3	11.2
0	2500	1000	.383	.376	.321	19.5	17.3
2π			.416	.402	.350	19.1	15.1
0	1000		.390	.388	.306	27.2	26.6
2π			.421	.408	.333	26.4	22.7

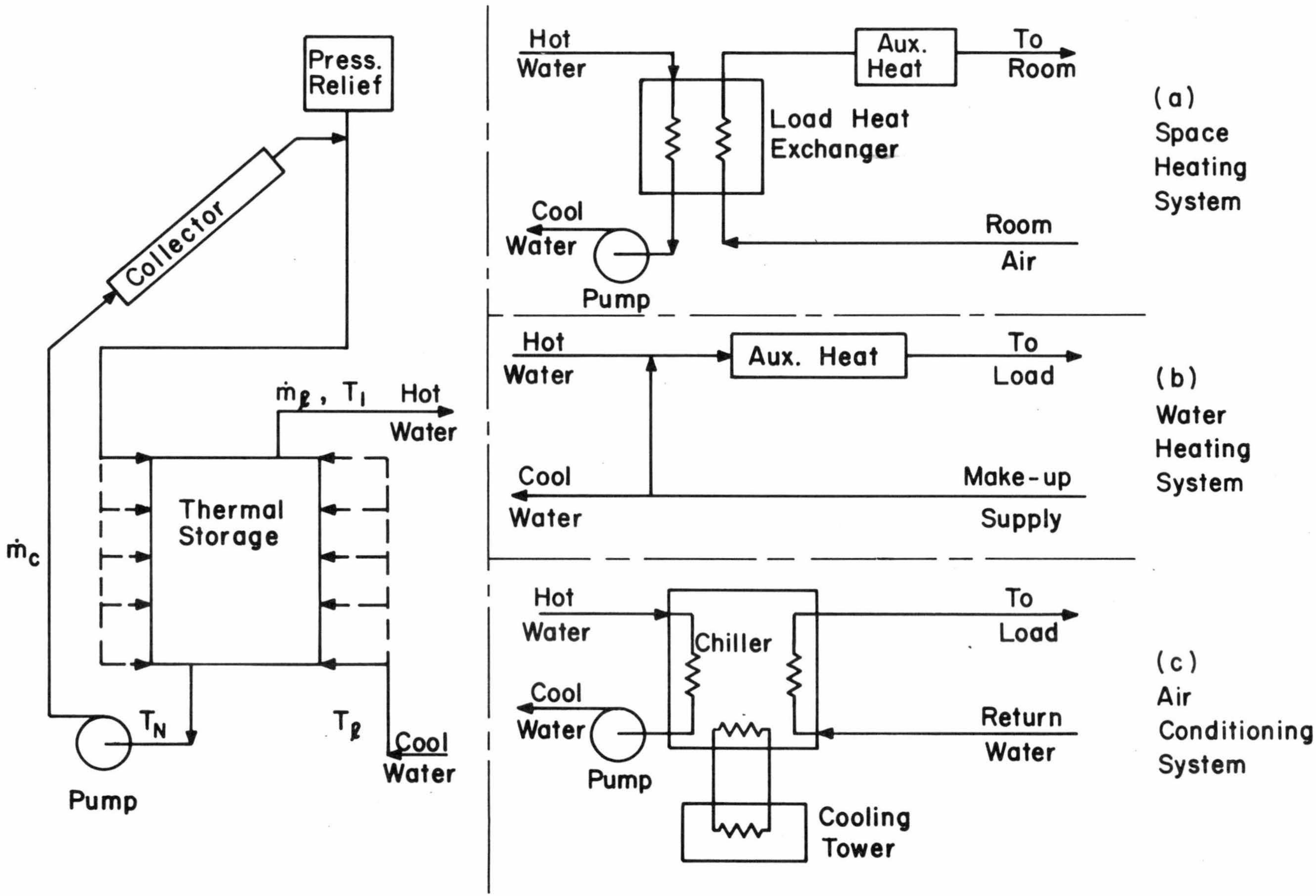

Fig. 1 System schematics.

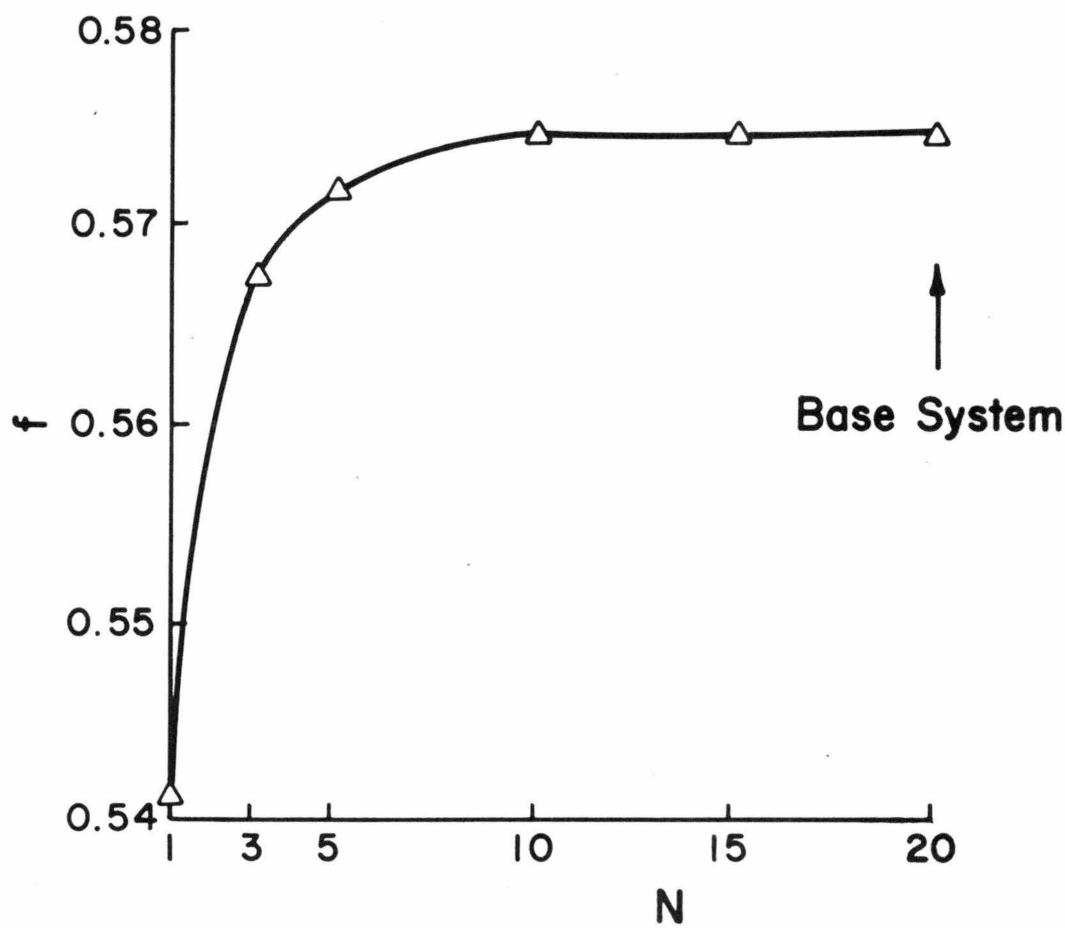

Fig. 2 Solar load fraction versus number of stratified tank segments, space heating.

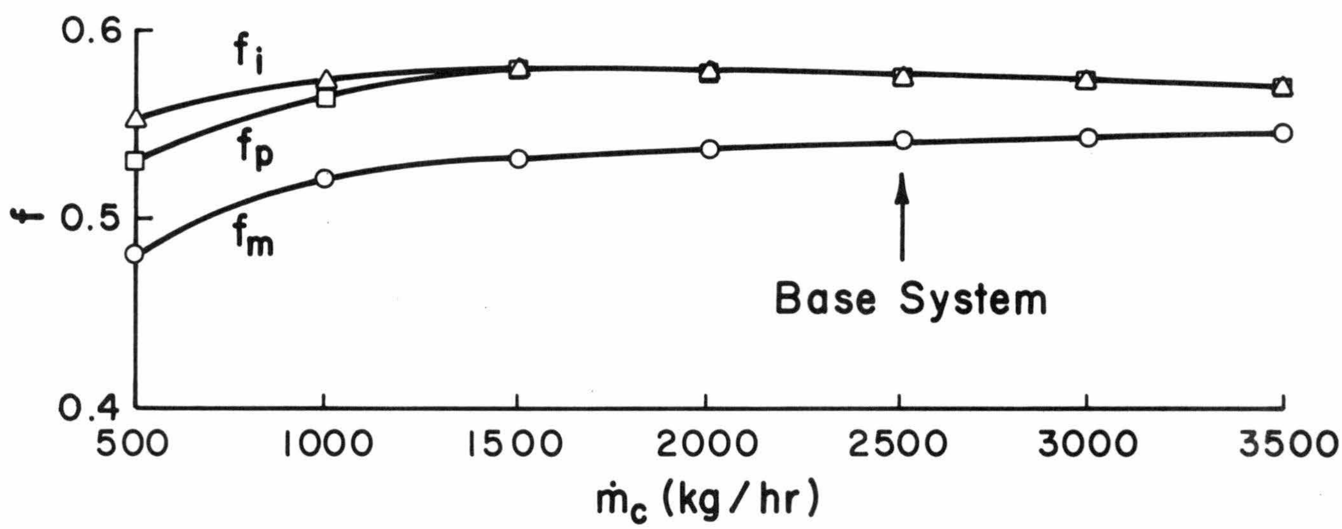

Fig. 3 Solar load fraction versus collector mass flow rate, space heating.

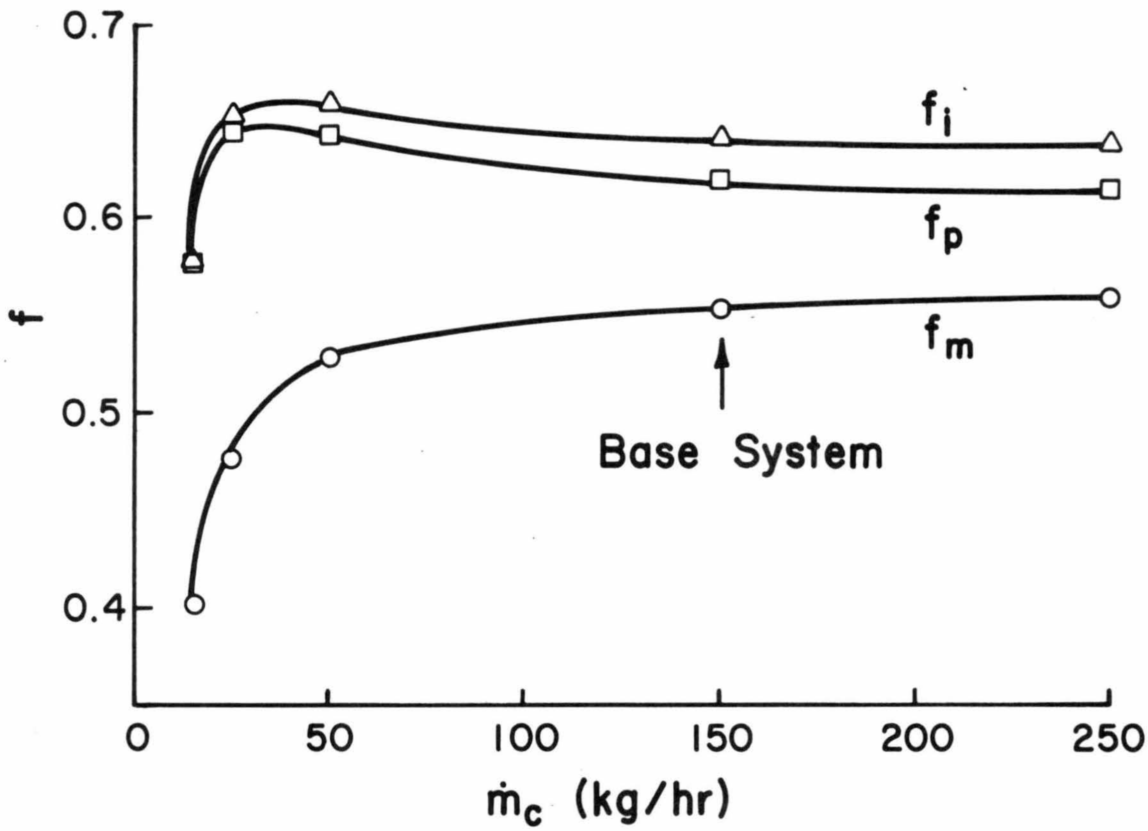

Fig. 4 Solar load fraction versus collector mass flow rate, water heating.

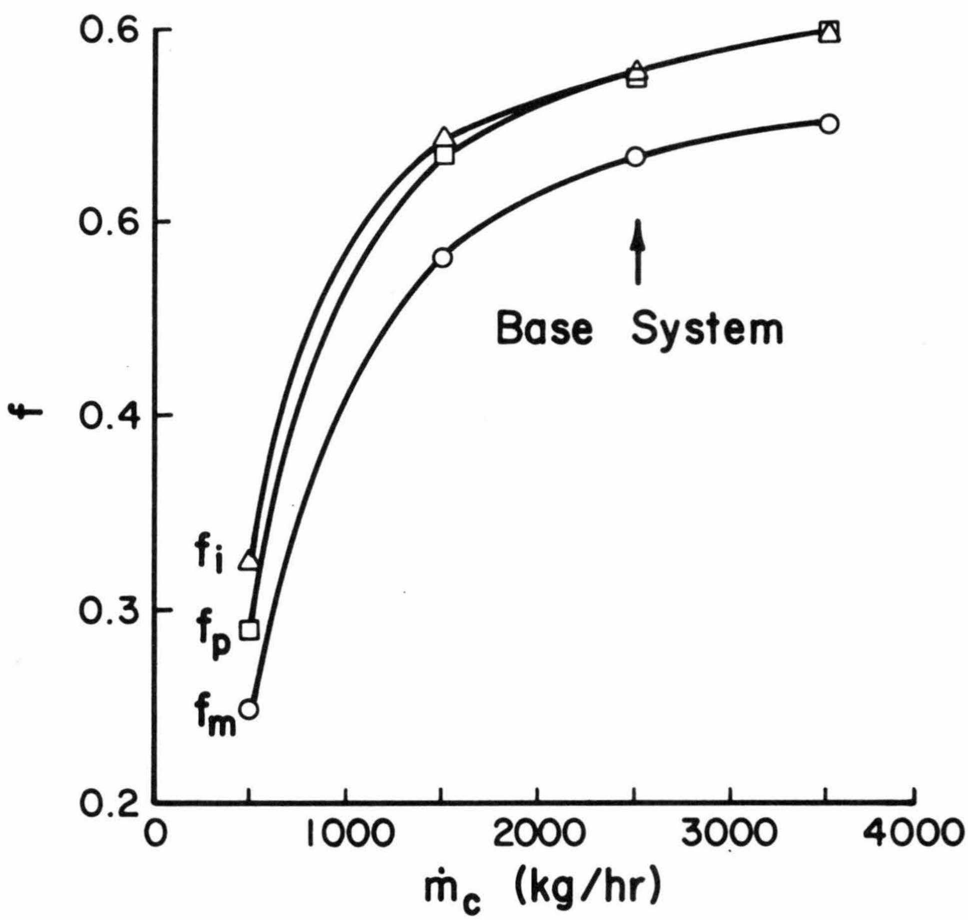

Fig. 5 Solar load fraction versus collector mass flow rate, air conditioning.

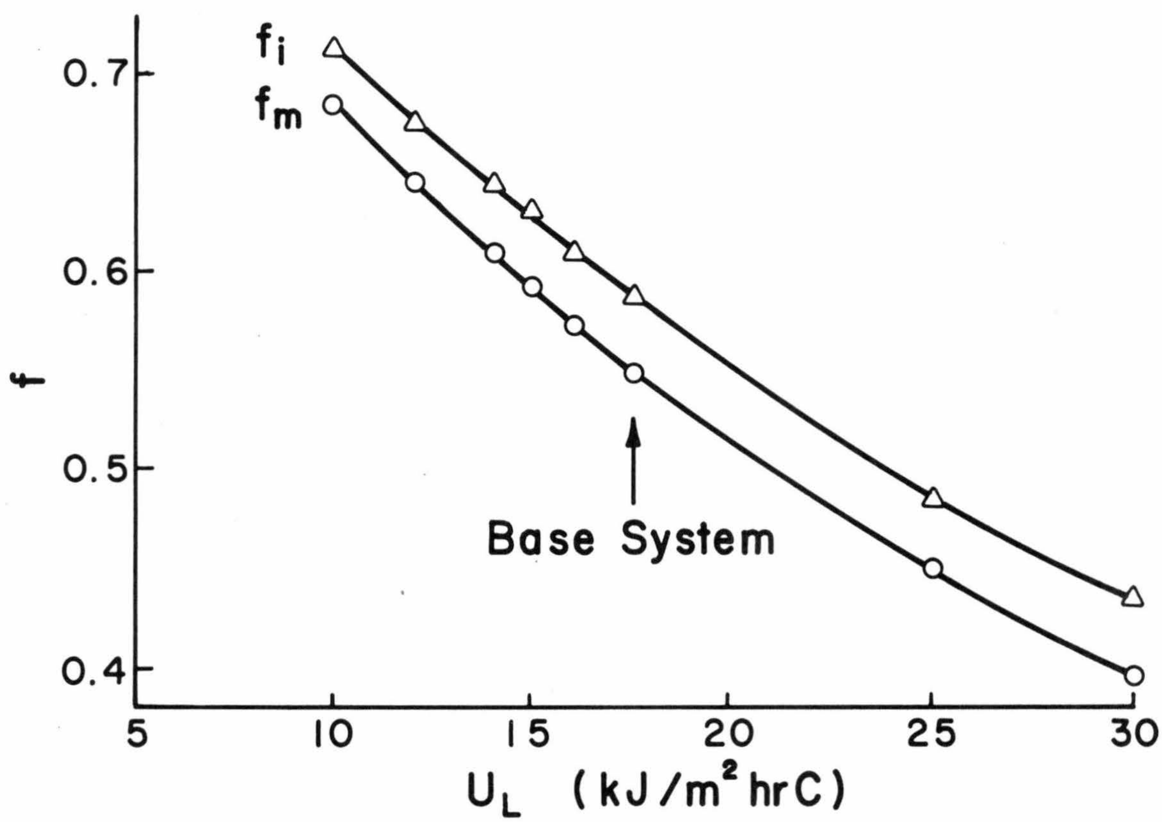

Fig. 6 Solar load fraction versus collector loss coefficient, space heating.

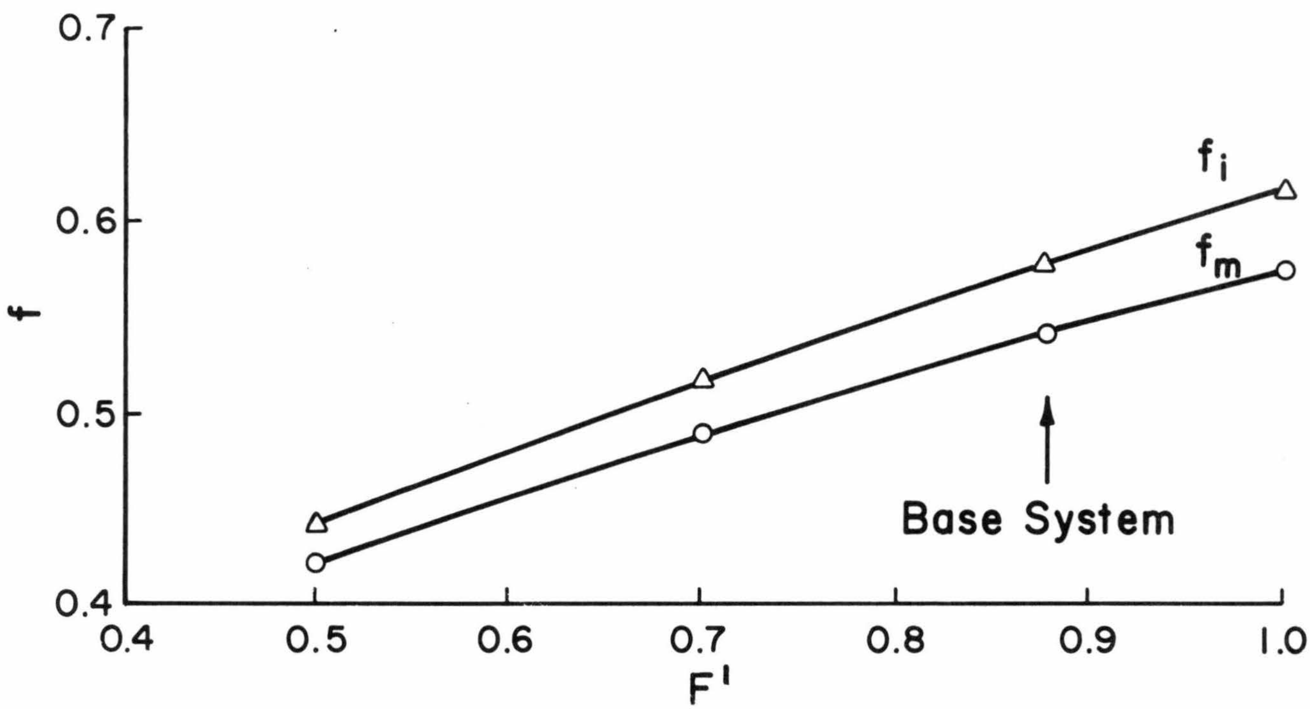

Fig. 7 Solar load fraction versus collector efficiency factor, space heating.

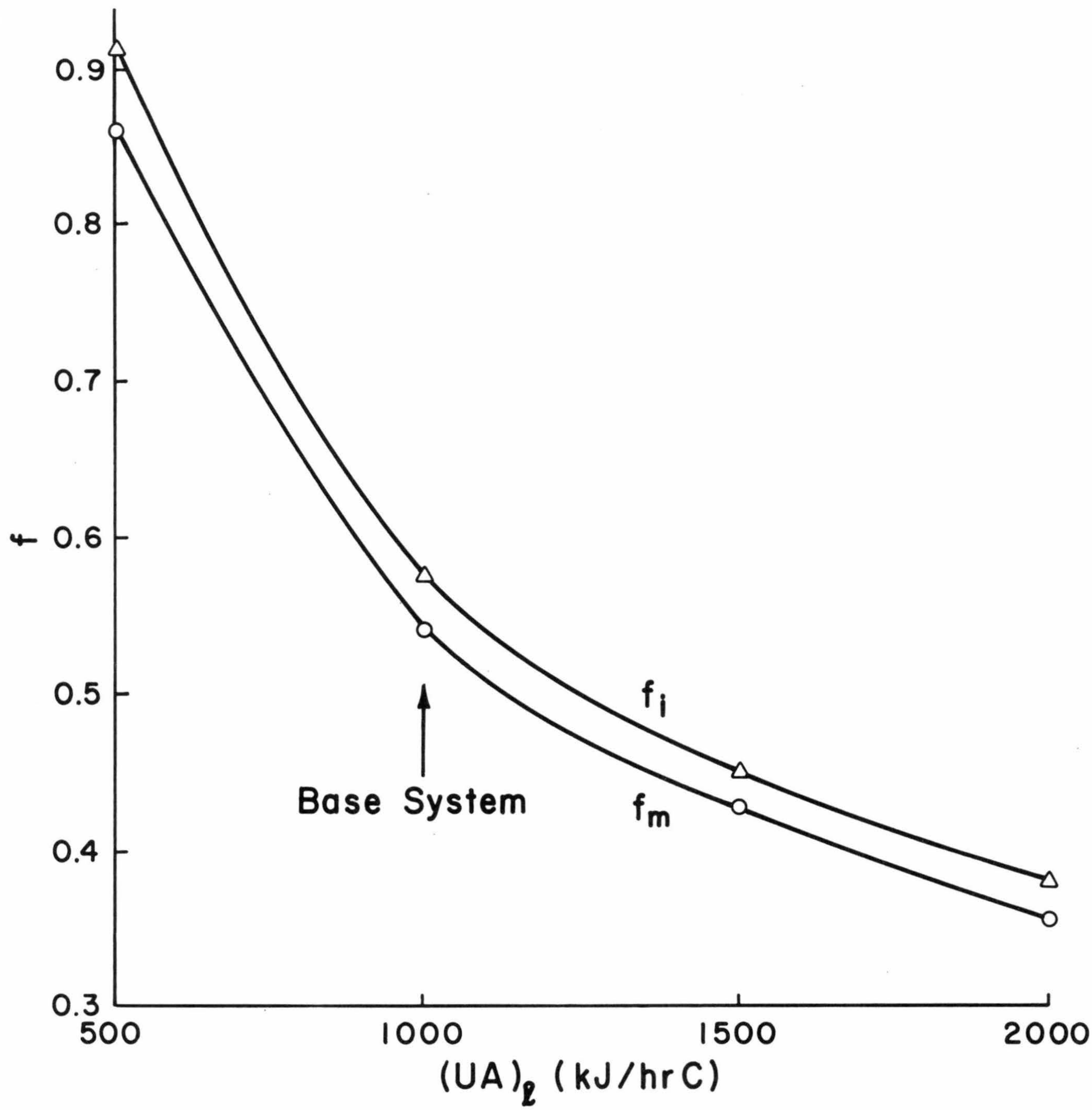

Fig. 8 Solar load fraction versus building loss coefficient, space heating.

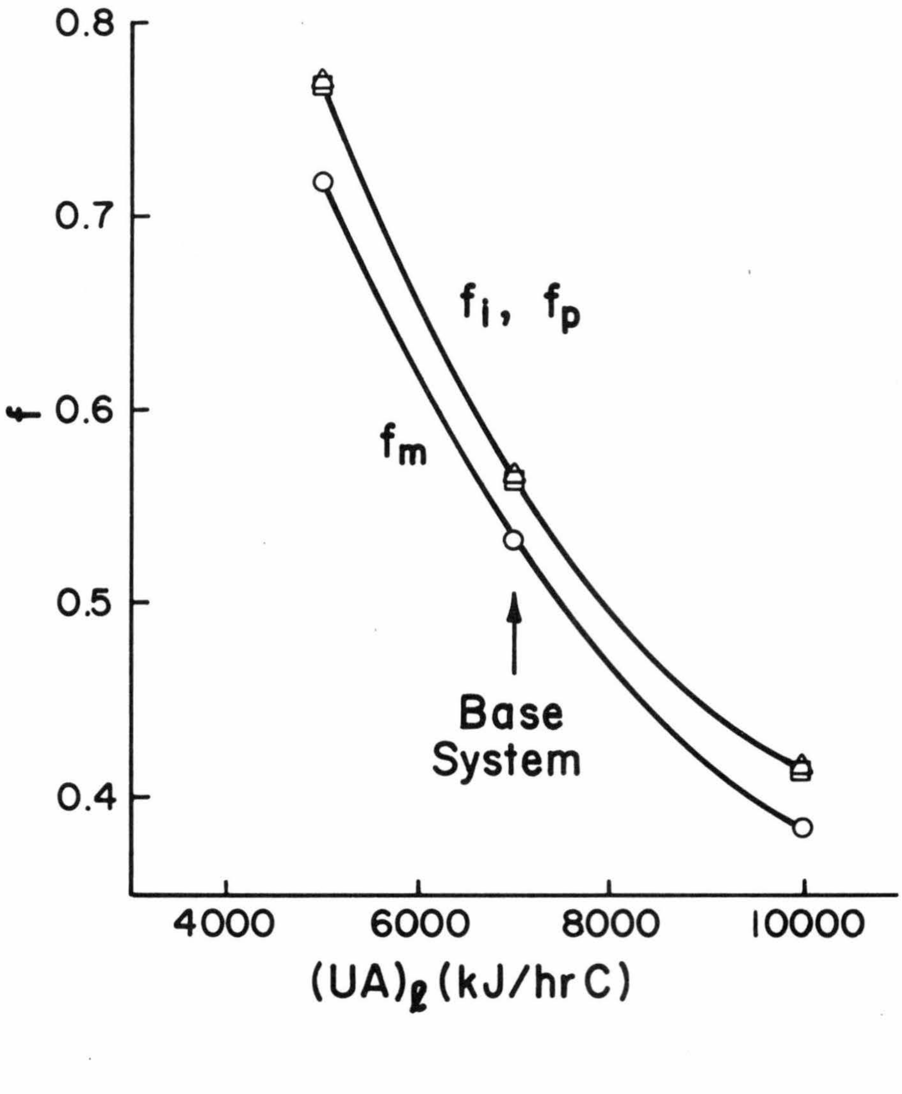

Fig. 9 Solar load fraction versus building loss coefficient, air conditioning.

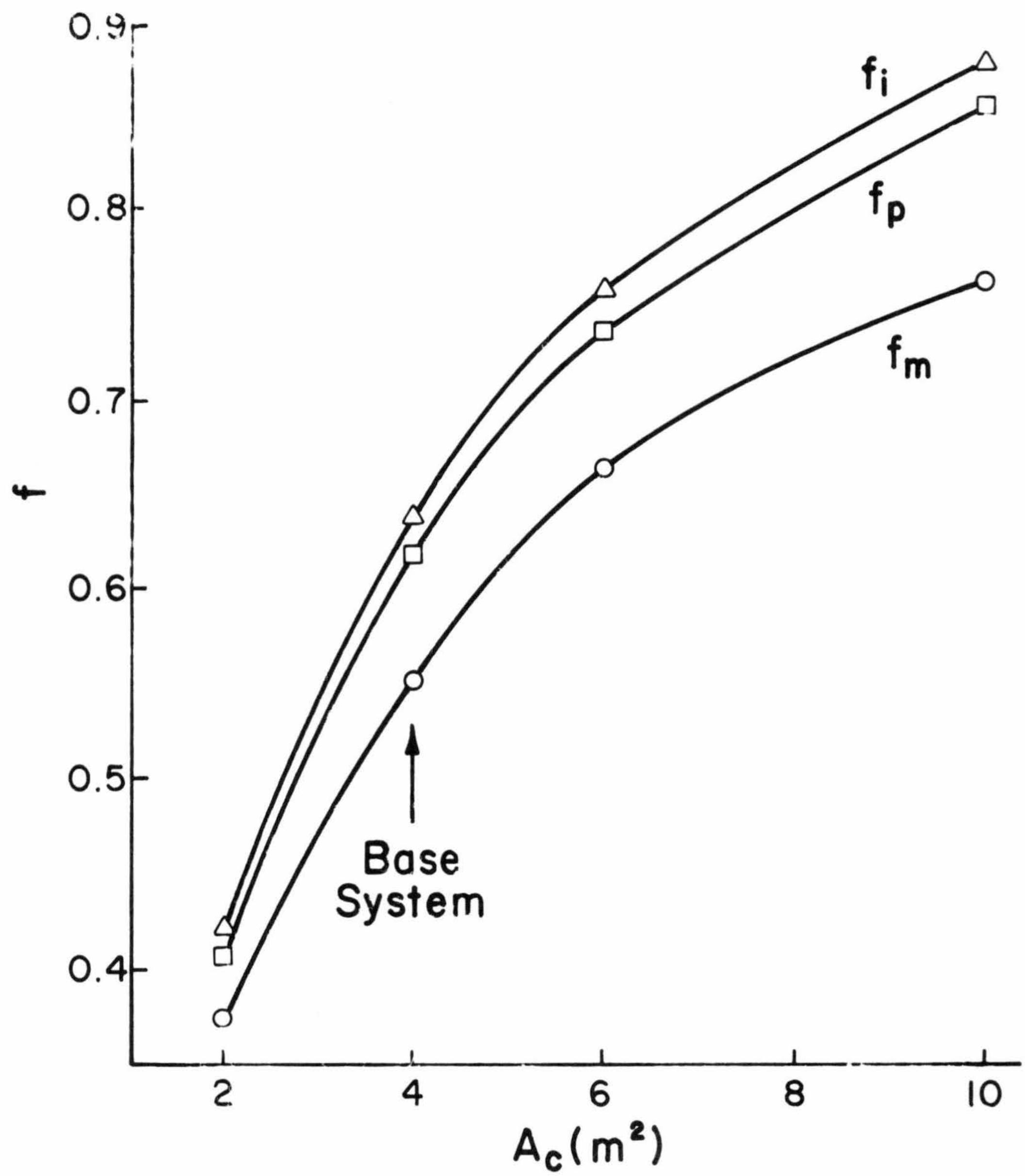

Fig. 10 Solar load fraction versus collector area, water heating.

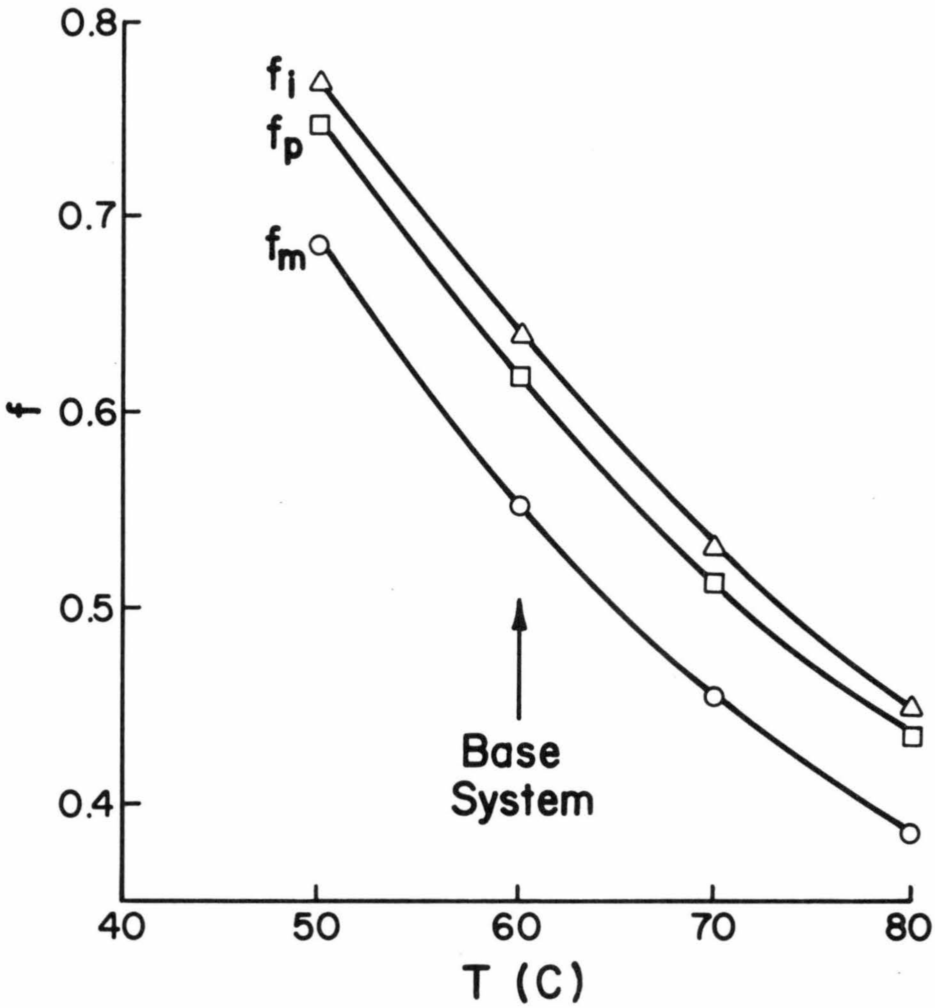

Fig. 11 Solar load fraction versus desired hot water temperature, water heating.

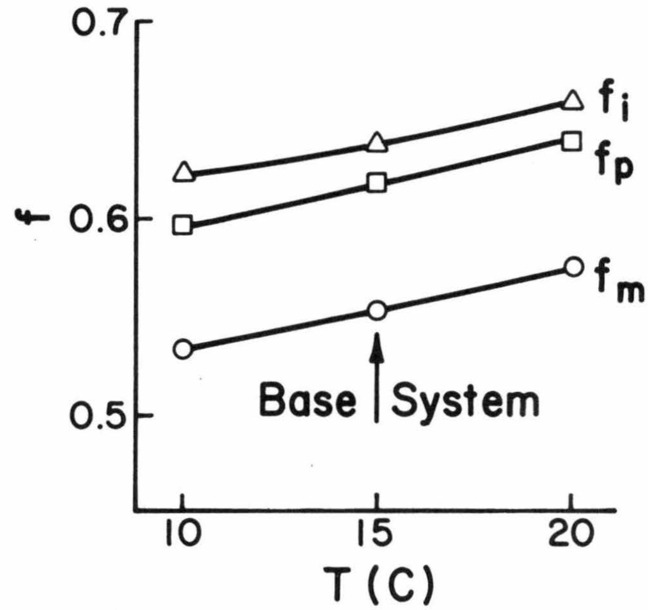

Fig. 12 Solar load fraction versus make-up water temperature, water heating.

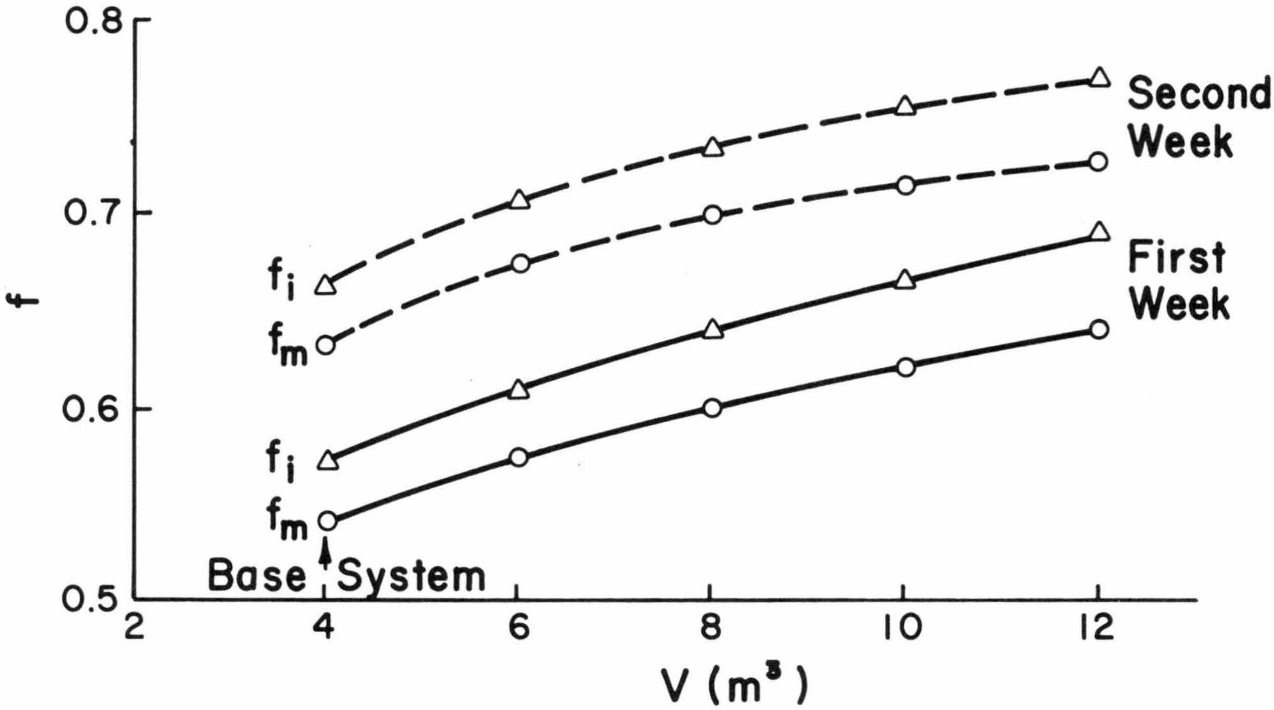

Fig. 13 Solar load fraction versus tank volume, space heating.

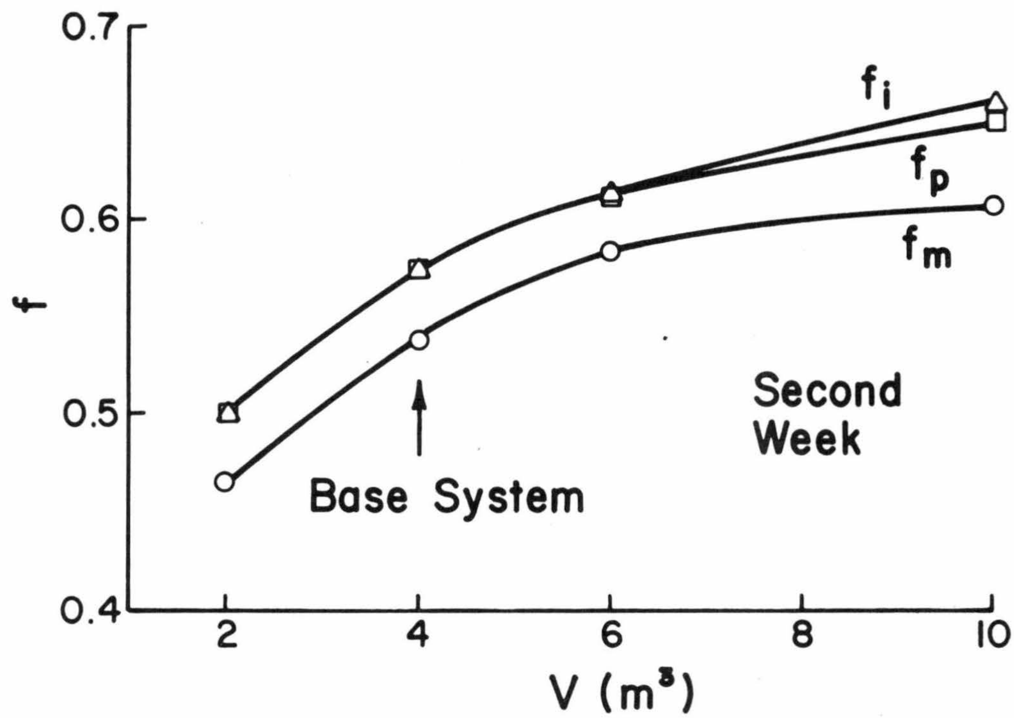

Fig. 14 Solar load fraction versus tank volume, air conditioning.

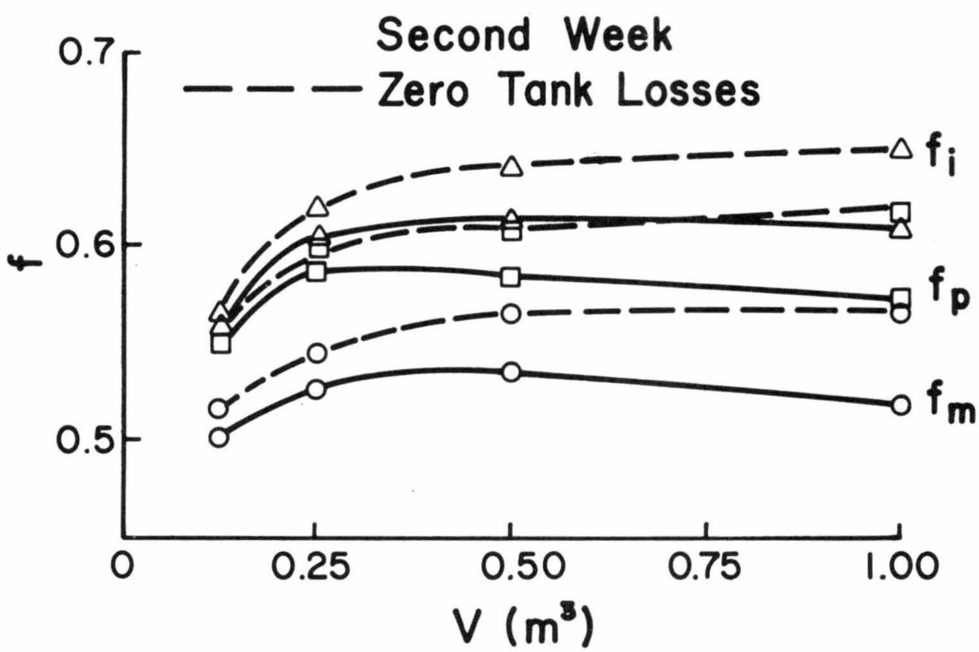

Fig. 15 Solar load fraction versus tank volume, water heating.

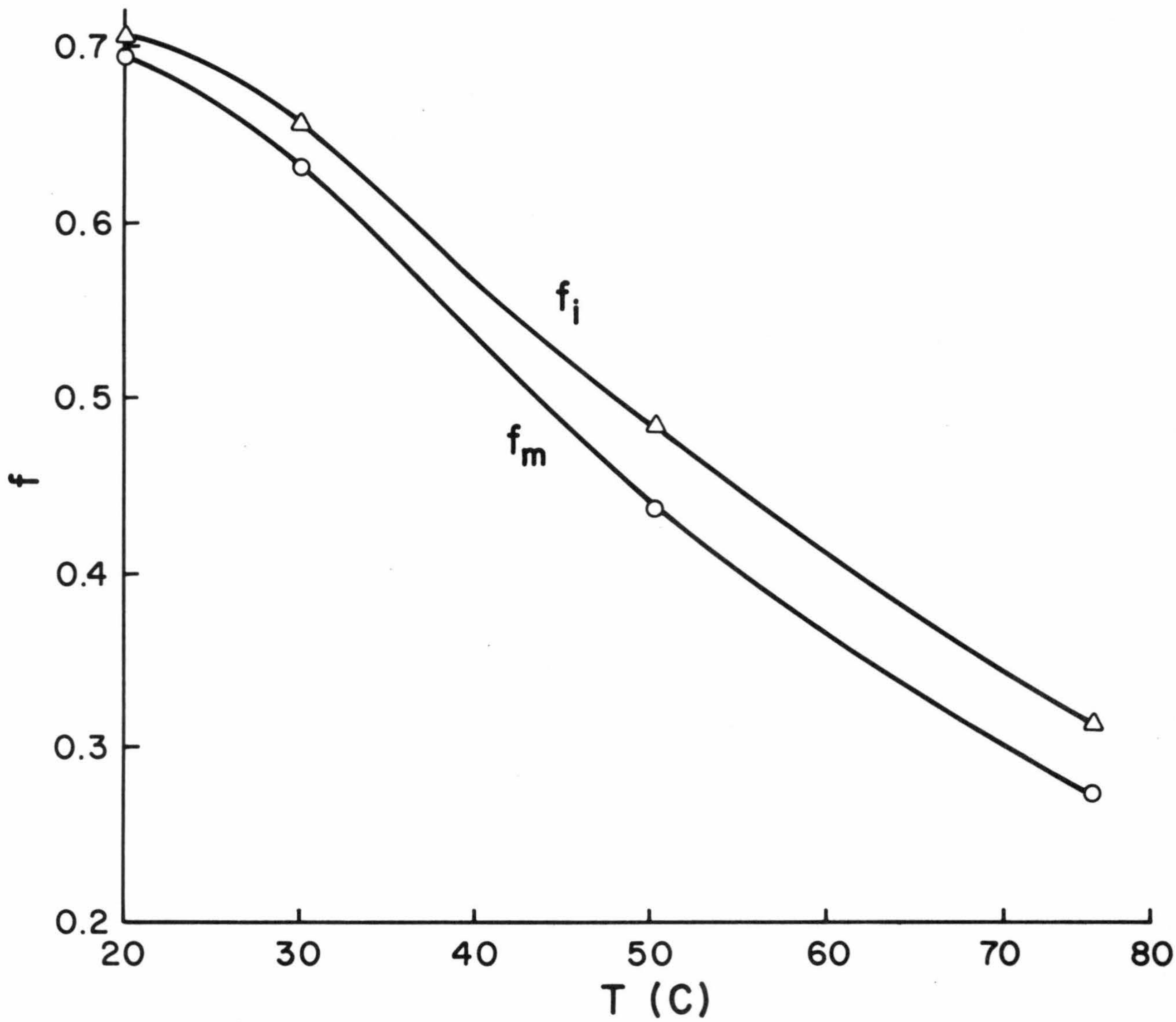

Fig. 16 Solar load fraction versus storage tank reference temperature, space heating.

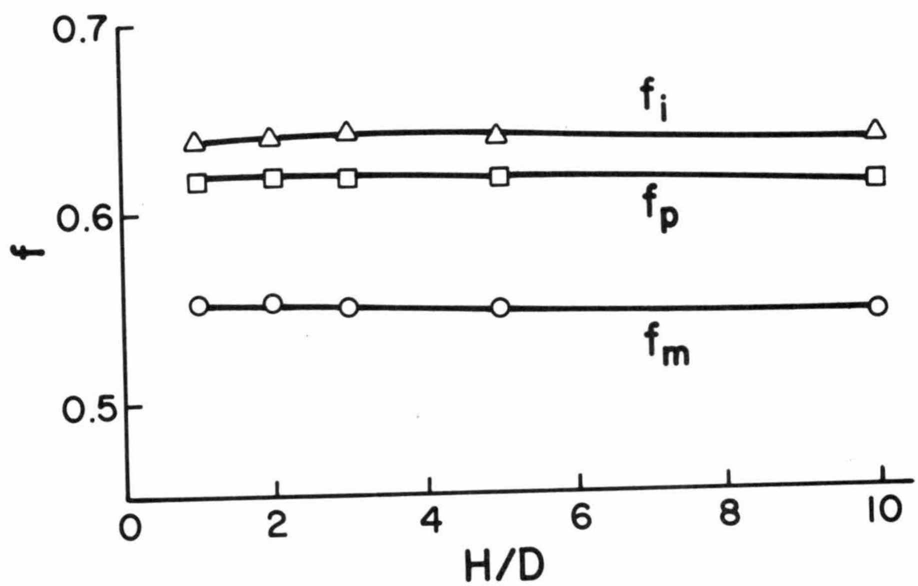

Fig. 17 Solar load fraction versus tank height to diameter ratio, water heating.

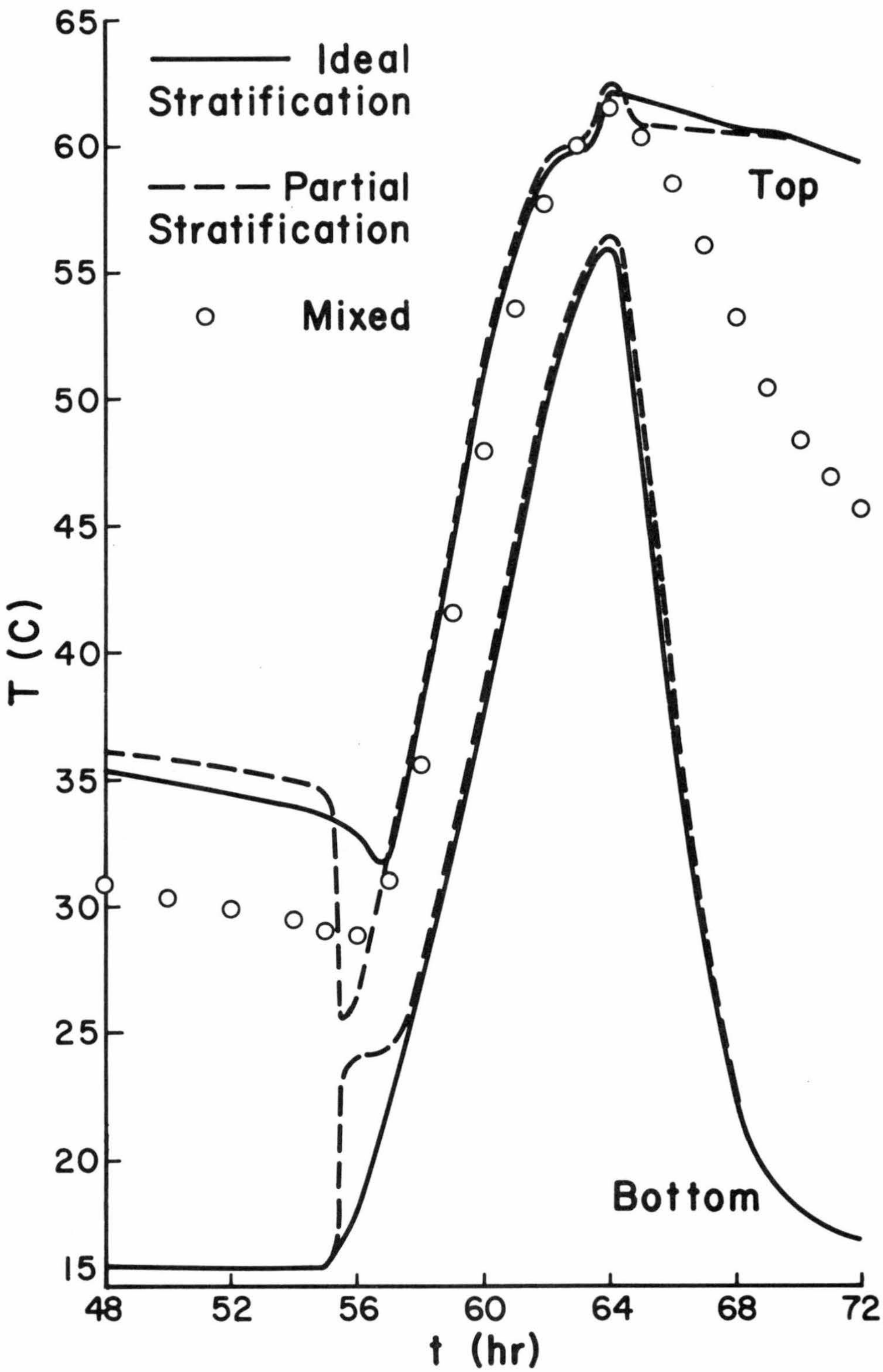

Fig. 18 Tank temperature histories, water heating.

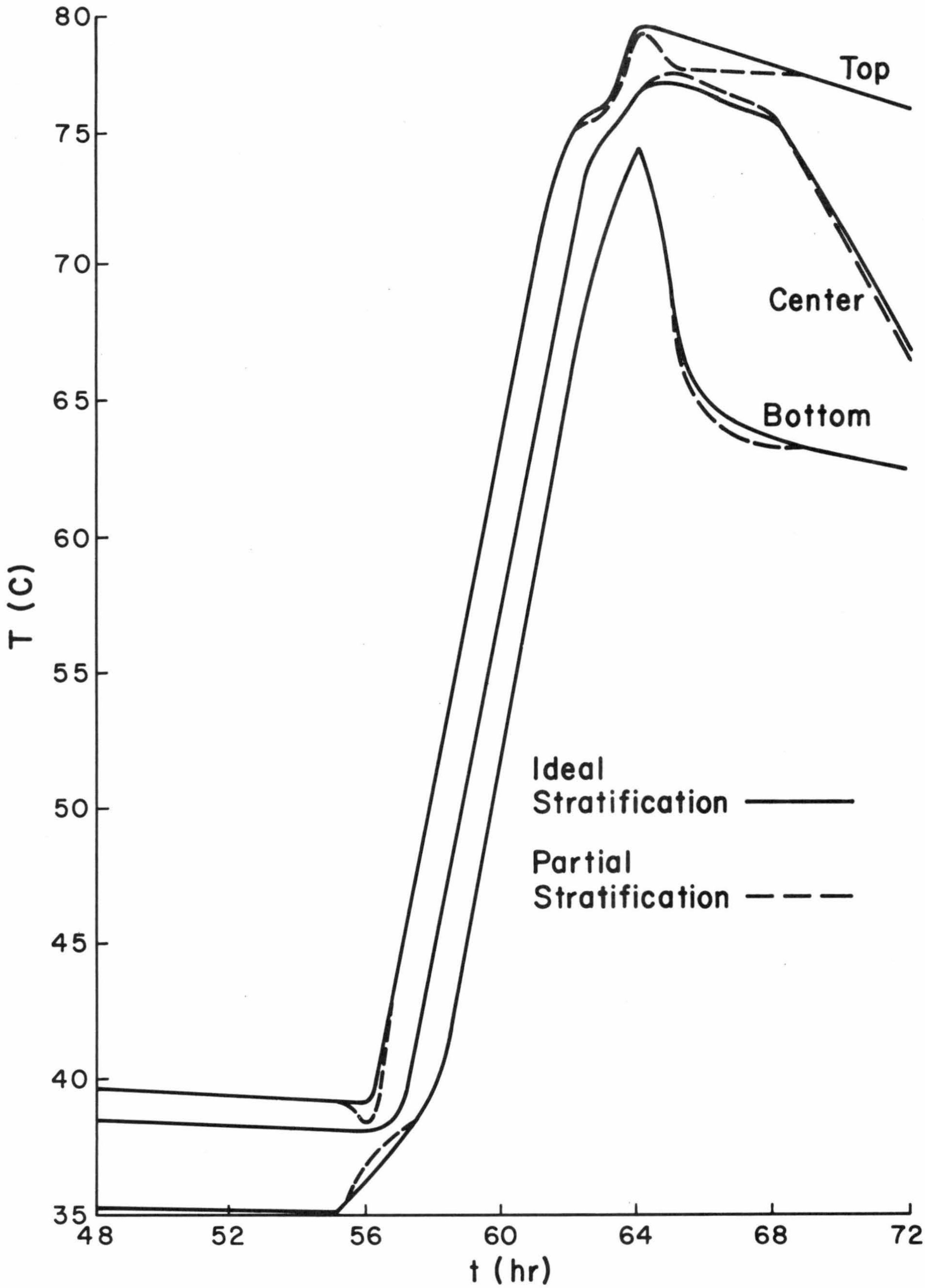

Fig. 19 Tank temperature histories, space heating.

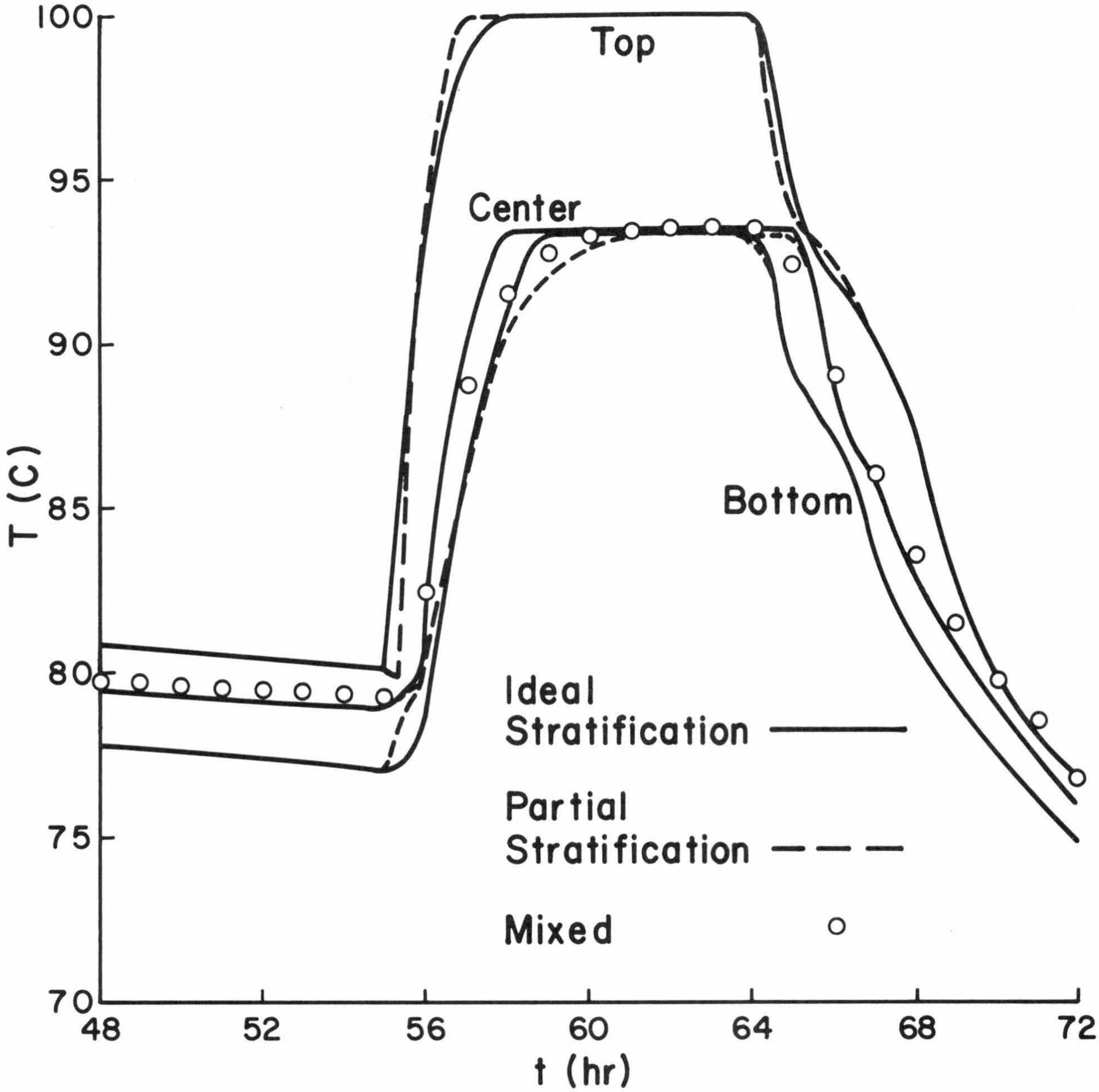

Fig. 20 Tank temperature histories, air conditioning.

Weather and Load Data

The weather data for both the space heating and the water heating simulations are for one week in January in Boulder, Colorado. These data are tabulated in reference 2 and are plotted here for convenient reference. The total insolation is shown in Figure A1 for the 75 m^2 of collector surface used in the space heating simulation. The corresponding ambient temperature variation is depicted in Figure A2. This was a mostly sunny, fairly cold week with two cloudy days with low insolation.

The water heating load is presented in Figure A3 in terms of the power required to heat the demand flow rate from 15°C to 60°C. The demand flow rate, shown in Figure A4 for a 24 hour period, was scaled down from the flow rate schedule presented in Davis and Bartera to give a total daily load of 250 kg. Their 2.1 minute long "demand events" each quarter hour were smoothed into the continuous demand profile shown.

Seven days of Washington, D.C. July weather data were used for the air conditioning simulation. The total insolation on the collector surface and the dry bulb and wet bulb temperatures are plotted in Figures A5 through A7.

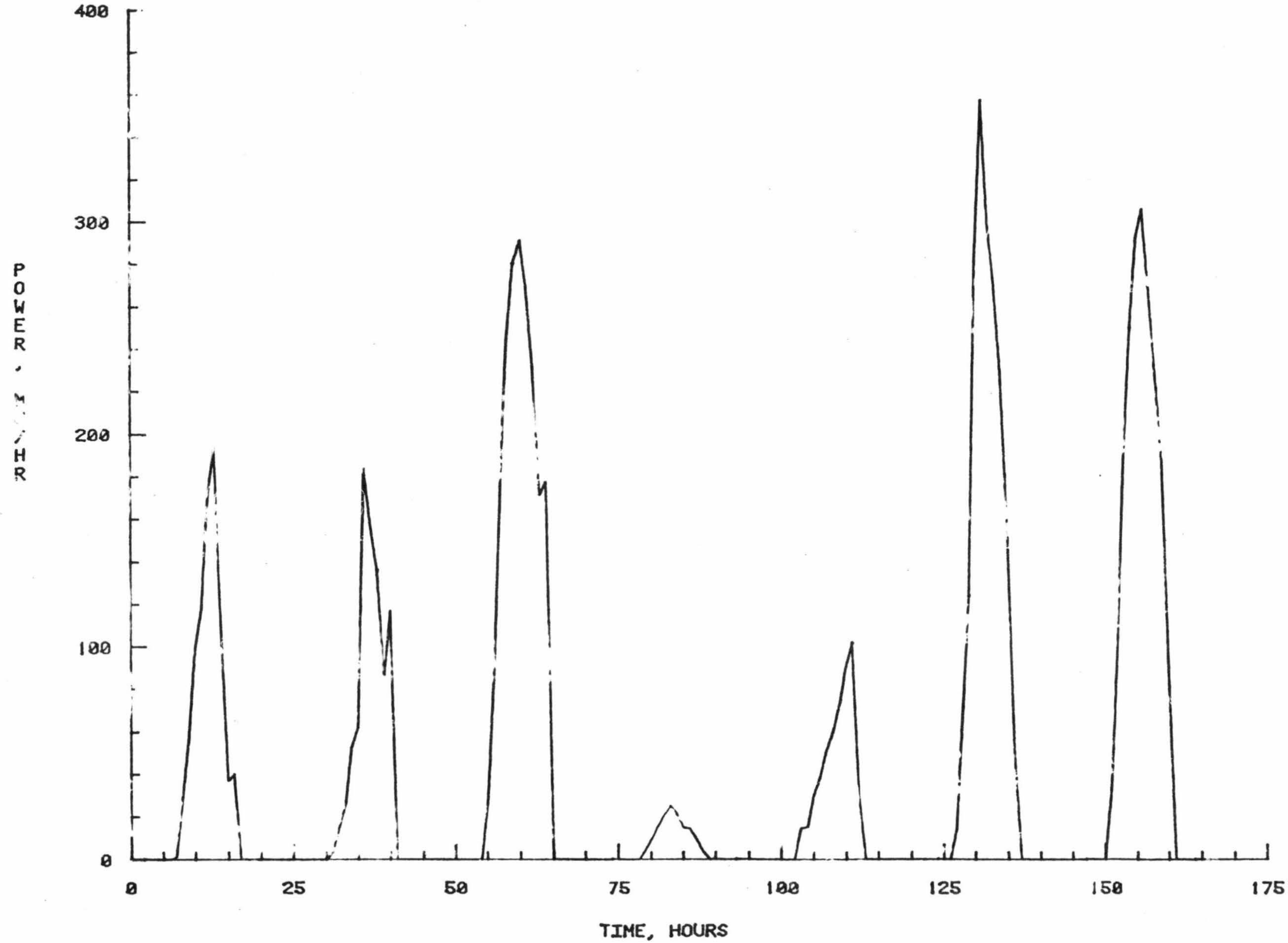

Fig. A1 Total insolation on collector, space heating.

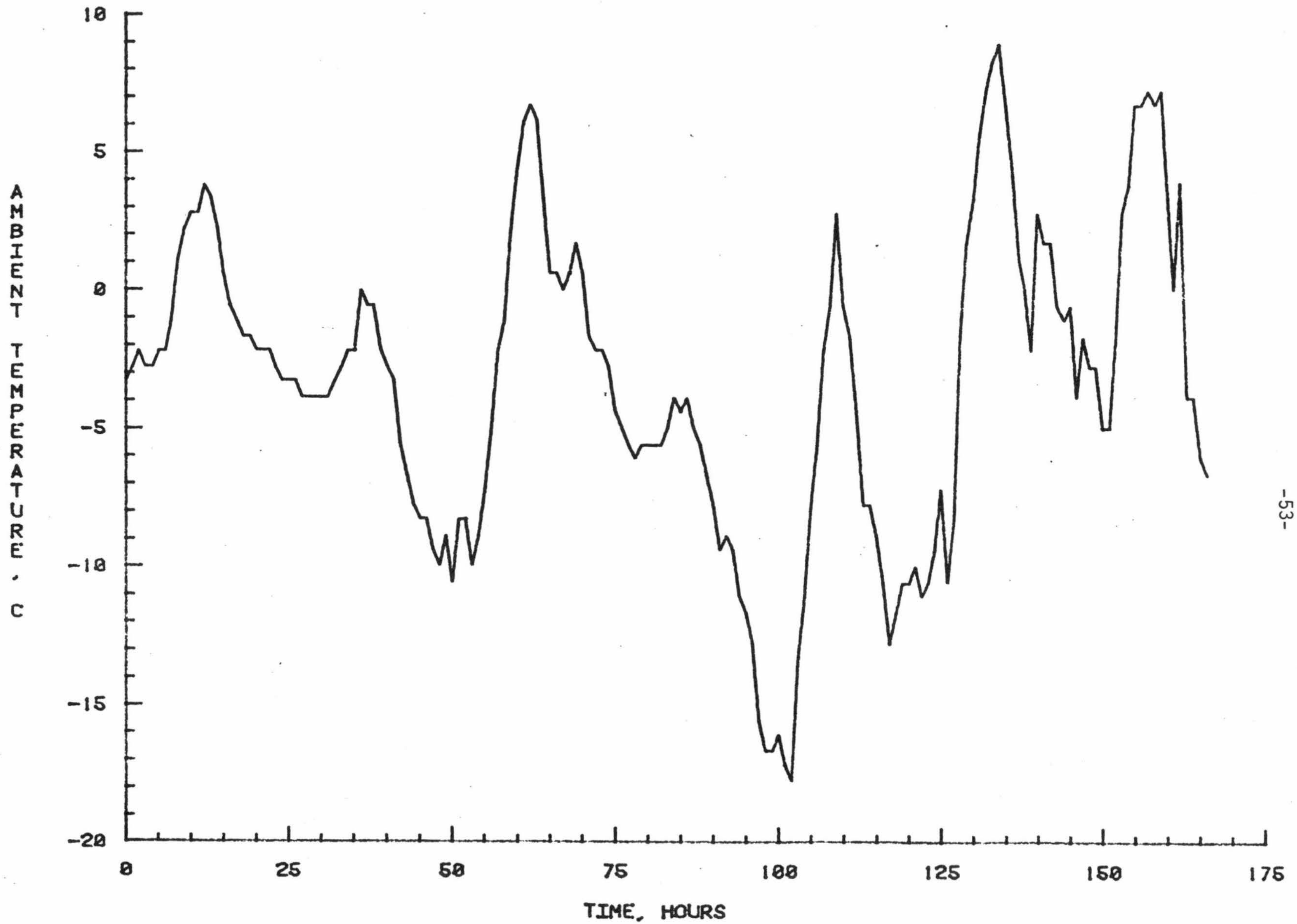

Fig. A2 Ambient temperature, water and space heating.

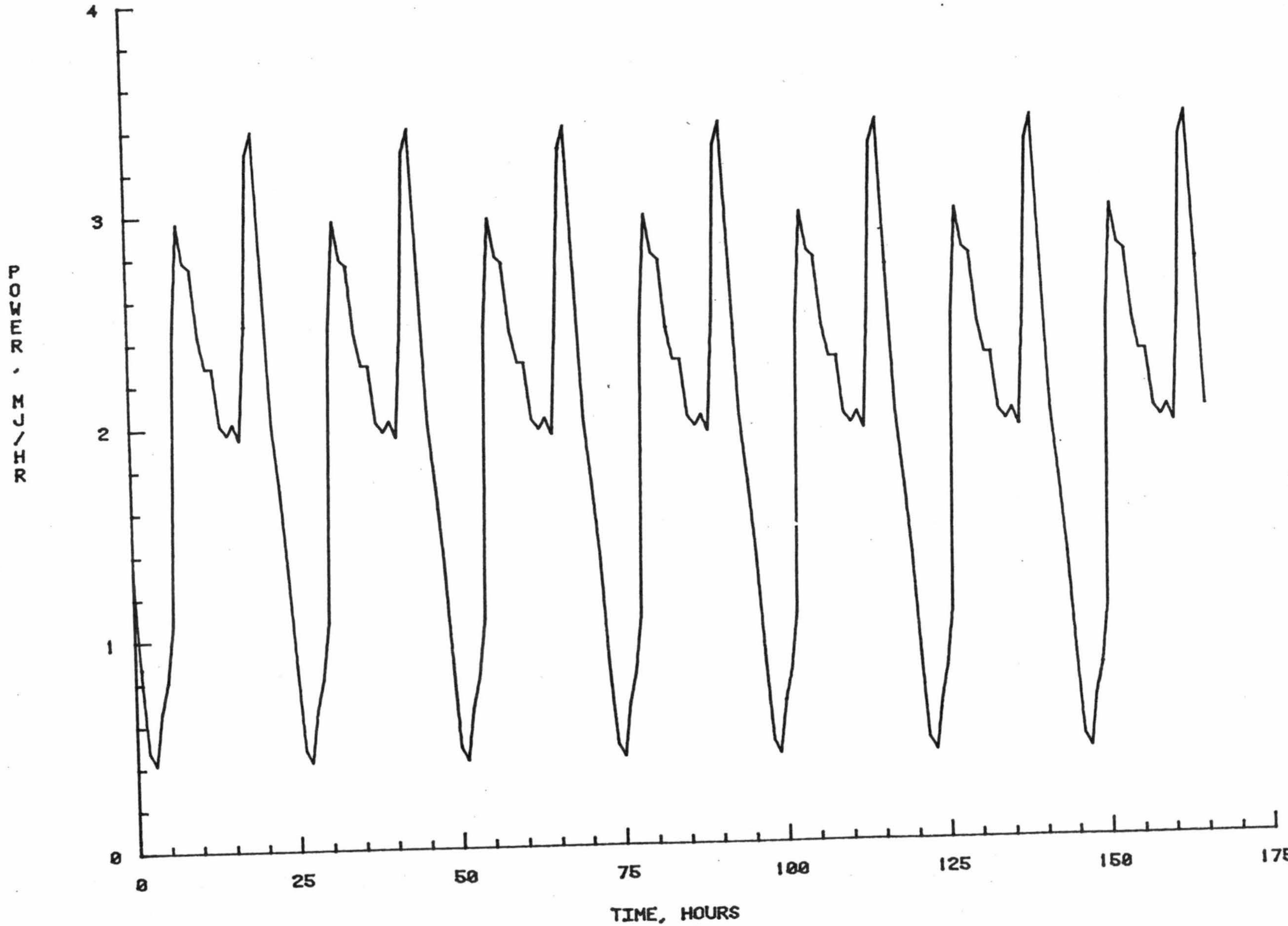

Fig. A3 Water heating load.

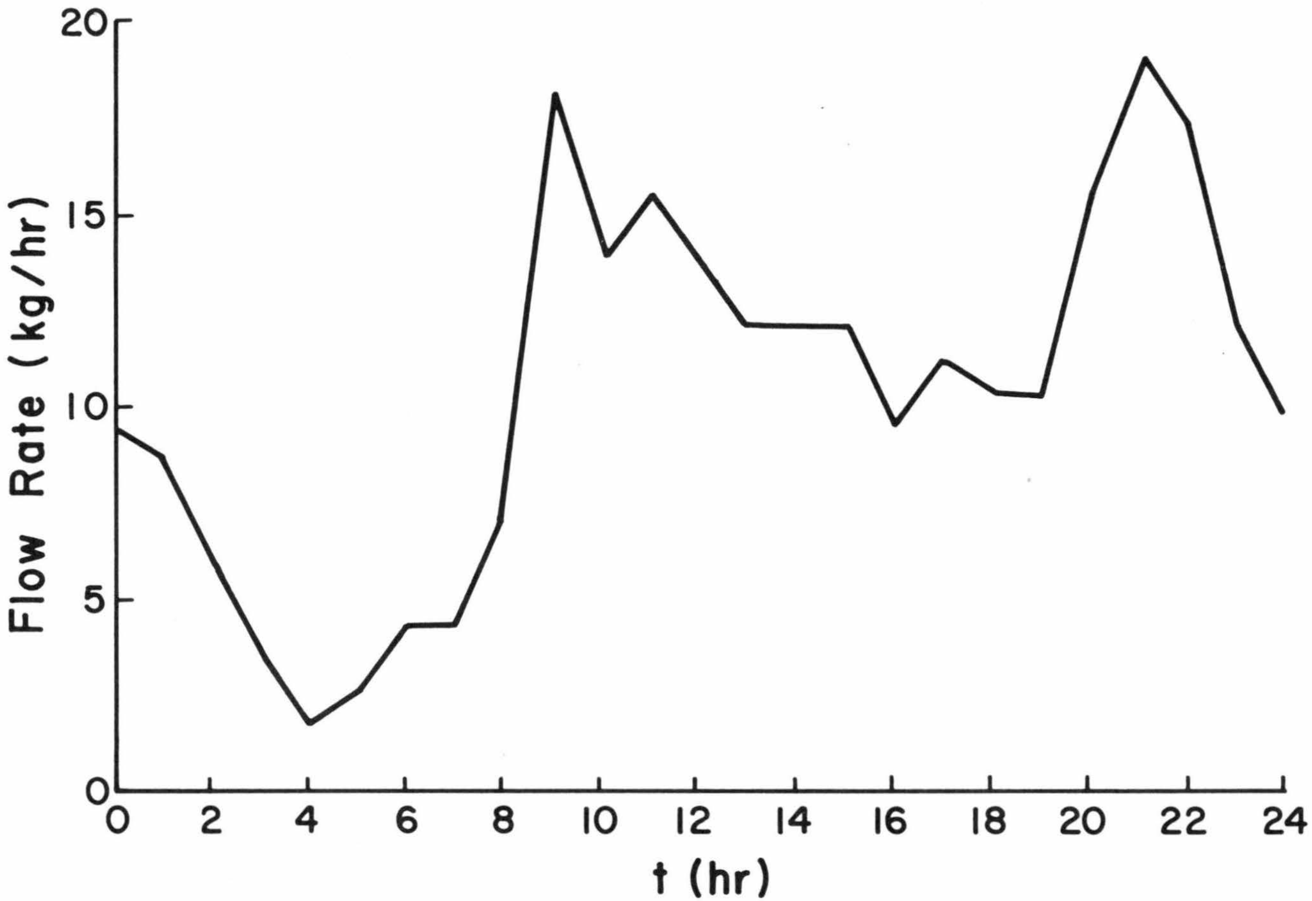

Fig. A4 Hot water demand schedule.

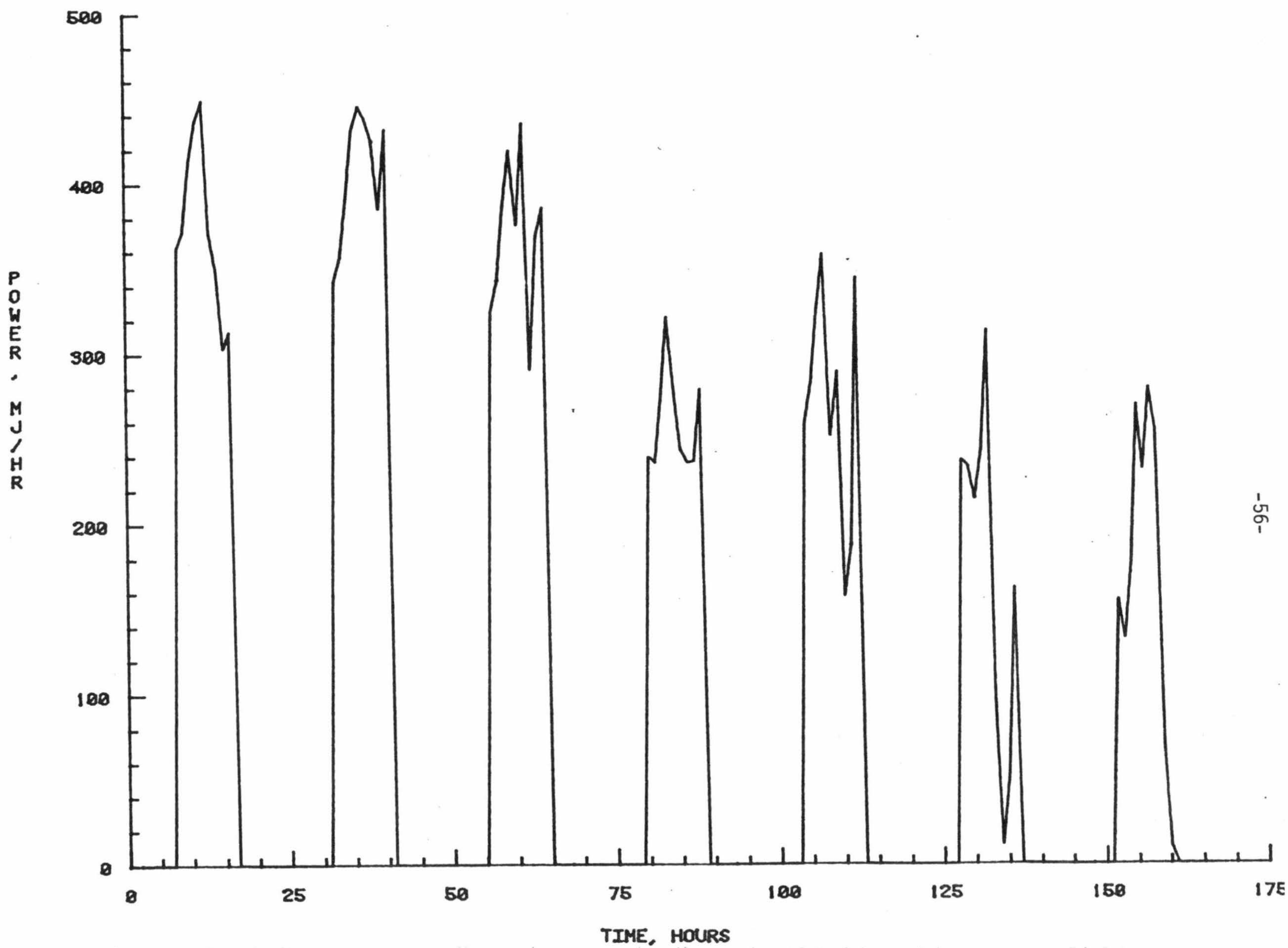

Fig. A5 Total insolation on collector,
air conditioning.

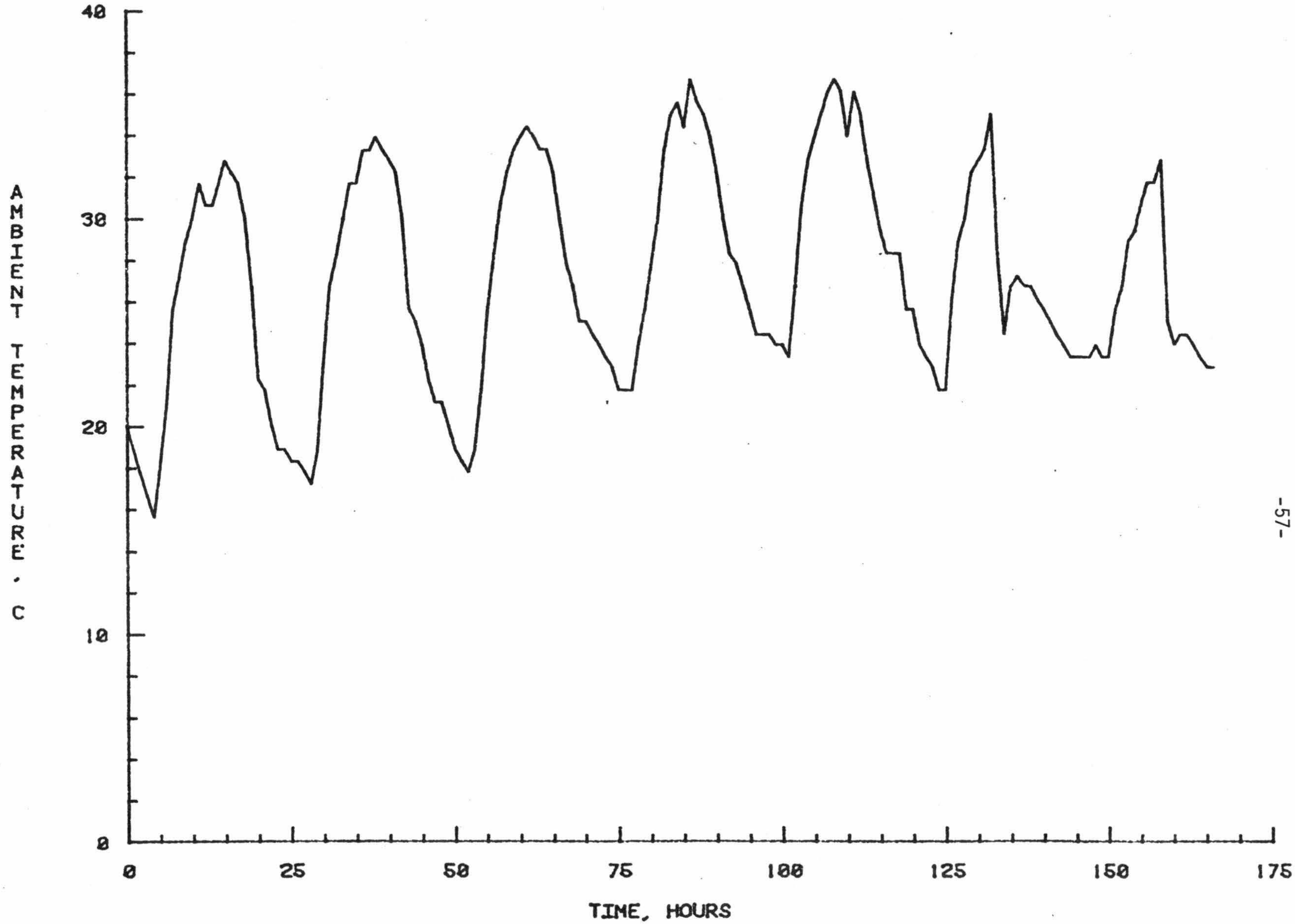

Fig. A6 Ambient temperature, air conditioning.

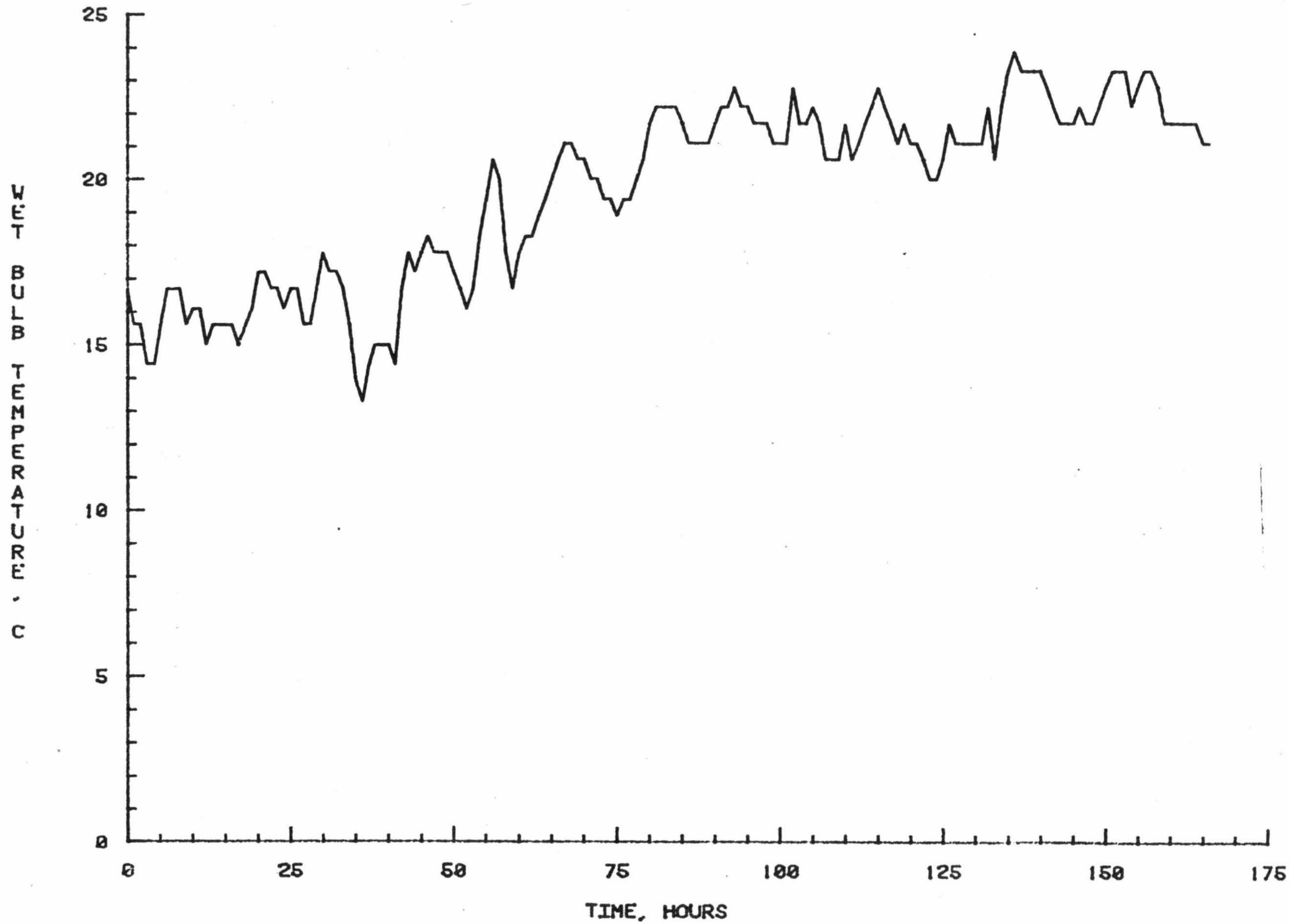

Fig. A7 Wet Bulb temperature, air conditioning.