THE DESIGN AND PREDICTED PERFORMANCE OF ARLINGTON HOUSE

BY

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Si - Slope of packed bed temperature profile at the center of node i [°C m^{-1}]

T - Local fluid temperature at position x [°C]

T_A - Ambient temperature [°C]

T_b - Local temperature of the bed packing at position x [°C]

$Tenv$ - Temperature of the surroundings [°C]

T_R - Room temperature [°C]

U - Energy loss coefficient from the packed bed to the surroundings [kJ hr^{-1} m^{-2} °C^{-1}]

V_H - Defined in equation (2.3-9) [dimensionless]

x - Position along the bed in the flow direction [m]

Greek

β - Defined in equation (2.3-8) [dimensionless]

γ - Control signal from the thermostat [dimensionless]

Δ - Denoting a difference or increment [dimensionless]

δ_{ij} - Kronika delta function, $\delta=0$ if $i \neq j$, $\delta=1$ if $i=j$ [dimensionless]

ε - Void fraction of the packed bed [dimensionless]

η - Effectiveness [dimensionless]

θ - Time [hr]

ρ - Density [kg m^3]

τ - Defined in equation (2.3-6) [dimensionless]

$\tau\alpha$ - Collector cover glass transmittance-absorptance product [dimensionless]

SUMMARY

The UW Solar Energy Laboratory, in cooperation with the College of Agriculture, Wisconsin utilities and the State of Wisconsin, intends to build a solar heated farmhouse near Arlington, Wisconsin. The purpose of the experiment is three-fold. First, Arlington House will demonstrate the practicality of domestic hot water heating and space heating with solar energy in a Wisconsin climate. Next, the problem of utilities supplying auxiliary energy when the solar system cannot meet the load will be addressed. Auxiliary energy supply can be either in the form of on-site storage (oil, LP gas) or demand matching from a central source (electricity, natural gas); this experiment is concerned with how to best incorporate offpeak electric auxiliary into a solar air heating system. Lastly, Arlington House will serve as a testing ground for advanced concepts such as the cylindrical evacuated glass tube collector.

As part of the Arlington House design effort, this thesis had two goals:

1) To develop a general capability to model air systems with the existing transient system simulation program, TRNSYS.

2) To model several potential systems for Arlington House and select from those a tentative design which was optimized with further simulations.

and with cylindrical evacuated glass tube collectors. A system using glass tube collectors and one gravel bed storage unit was singled out for further study. The effects of collector flow rate and tube spacing, domestic hot water heat exchanger position, simplified controls, and storage unit size on system performance have been studied through simulation. From this information a final system design has tentatively been selected which meets the goals of the Arlington House experiment.

1.0 INTRODUCTION

1.1 Socio-Economic Background

The technology required to heat and cool buildings with solar energy has existed for years. Recently the federal government has attempted to create a favorable climate for solar energy development by allocating funds for research and demonstration projects and by re-evaluating price control policies which have kept the cost of fossil fuels artificially low. State and local governments have introduced tax incentives to encourage the development of solar energy. In some parts of the country where oil and natural gas are unavailable for space heating and cooling, solar energy has been competitive with electric heating and private solar energy businesses have gained a foothold. A diverse group of well informed businessmen, government officials, engineers, and architects are viewing the trends in the solar field with great optimism.

Unfortunately the essence of the last paragraph is in the word "viewing". Major manufacturers, the construction industry and investors are all waiting for the market to materialize while the general public is unfamiliar with or skeptical about the practicality of solar heating and cooling in extreme climates. In addition, the need for a full-sized, conventional (auxiliary) heating and cooling system

University of Wisconsin Solar Energy Laboratory, in cooperation with the College of Agriculture, Wisconsin utilities, and the State of Wisconsin, intends to build a solar heated farmhouse near Arlington, Wisconsin to demonstrate: 1) the practicality of solar heating and domestic hot water systems in a Wisconsin climate, 2) the integration of offpeak electric auxiliary into a solar system, and 3) the use of advanced concepts in solar space heating design. "Arlington House" will have a well instrumented, experimental system suitable for providing the type of engineering data needed to verify computer models and convince the public that solar energy is feasible in Wisconsin.

1.2 Technical Background

Solar systems which supply space heat can be classified as passive or active; active systems can be further divided into air or water systems. A passive system is an architectural design using the building as a combined solar collector and thermal store. Active refers to mechanical solar heating and cooling systems characterized by appliance-like components which are assembled together to perform the functions of solar energy collection, storage and supply to the conditioned space. Air and water are the possible transfer fluids used in active systems.

period of time will slowly lower the temperature of the structure and the air in it. The larger the acceptable temperature swing, the more effective a passive system will be. In temperate climates a passive system cannot meet 100% of the heating load for a reasonable cost; therefore, a full-sized back-up system is required.

Active systems have the flexibility to be occupant controlled or controlled automatically. They also require a back-up or auxiliary system sized to meet the worst case load. Room temperature can be maintained near a set value, but relaxing this requirement will save auxiliary energy. As mentioned earlier, active systems are of either the water or air type. An economic analysis, based on the costs of hardware required to satisfy the restraints of climate and occupant preferences, should decide which active system is to be used.

Water systems have received the most attention since the latest rebirth of interest in solar energy because of their similarity to smaller commercial service hot water heating systems developed earlier. Since water can be circulated from the storage tank to the load and collectors simultaneously, simple control schemes can be used which operate pumps in the collector and load loops independently of one another. The power required to pump water is small compared to that needed to pump air;

combines the functions of heat exchange and thermal store. A disadvantage is that for the same storage capacity, a gravel bed must have a larger volume than a water tank. Although absorption air-conditioners are not easily integrated into air systems, night-time cooling of the packed bed with outside air (possibly boosted with evaporative or open-cycle coolers) provides an alternative.

An active solar air heating and service hot water system, integrated with offpeak electric auxiliary, was chosen for Arlington House. An active system was chosen because it is generally believed that they will gain acceptance with the public before passive systems, which require a floating room temperature and occupant participation in control. Arlington House will have an air system to avoid the practical problems which most water systems have in extreme climates. Cooling will not be provided because although Madison has a moderately hot and humid summer the cooling season is short and central cooling is not necessary. Furthermore, Madison's humidity and the absence of large diurnal temperature differences complicates solar air system cooling techniques while intelligent architectural design can utilize the forces caused by wind and air density gradients to induce sufficient ventilation for summer time comfort. An interesting concept is a passive architectural design

The solar air heating system for a CSIRO Laboratory in Australia was designed and studed by Close _et al_. [6]. They concluded that air flow modulation to obtain a constant collector outlet temperature was not necessary in a system with a gravel bed storage unit.

The Desert Grassland Station [7] solar air heating system (in Arizona) was designed to meet 100% of the structure's heating load. Bliss suggested that a smaller solar installation with an auxiliary furnace would have been less expensive.

The University of Delaware's Solar I has a unique system incorporating flat plate air collectors, phase change storage and photovoltaic cells. Böer _et al_. [8] reports that air flow modulation through the collectors is necessary to achieve the temperature level required to drive the reaction in the heat of fusion storage bin throughout the day. At the time of the study, no degradation of the storage material after two years of operation had been noticed, making Solar I the longest successful experiment with a phase change storage unit.

The Norwich House in Vermont and the Alumni Conference Center in Albany, New York, each have offpeak electric auxiliary integrated into solar heating systems [9]. The Norwich House has a very large gravel bed store which is heated to 30°C between 9 PM and 7 AM on a coherent form useable for designing the Arlington House System; the objective has been to arrive at a final design.

Other work at the U. W. Solar Energy Laboratory during recent years has been toward the development of a modular approach to digital computer simulation of solar energy systems. In general, simulation methods are employed to compare different system configurations and control strategies or to generate charts useful for designing standard solar heating systems such as those to be used in residences. Design charts for building solar heating systems of conventional design, and with water as the heat transfer fluid, have already been constructed [17]. The method used relied on long term simulations of these systems, following sensitivity analyses on system parameters. Therefore, a secondary objective of this study was to develop general air system simulation capability so that the same simulation applications described above can be used for air system design, including the design of Arlington House.

tem components, air systems can be simulated using TRNSYS as the driver-integrator program.

2.1 Problems and Simplifications

There are two classes of problems which complicate the simulation of solar air systems; those associated with transfer mechanisms that are difficult to model and those deriving from the limitations of finite difference digital simulation. Some of these problems are unique to solar air systems while others simply cause no difficulty in water system simulation.

2.1.1 Transfer Mechanisms Difficult to Model

2.1.1.1 Duct Heat Loss

Both solar air and water heating systems experience heat losses from their fluid distribution systems. In water systems, the losses are usually small or easily controlled because of the small surface area of piping which needs to be insulated while in air systems large duct surface areas make losses more significant. Although ducts within the conditioned space do not really "lose" heat, uncontrolled heat gains in duct stacks and basements are not very useful from a comfort standpoint. Ducts exposed to unconditioned spaces (attic, outdoors, crawlspaces) can lose significant amounts of energy even

2.1.1.2 Air Leakage

Although water leakage from hydronic heating systems is not tolerated, conventional forced air heating systems have historically been constructed under less rigid standards because air leakage was not considered critical. According to ASHRAE [20], well designed and installed low pressure air systems will leak 1-3% of the total flow rate while untaped systems can leak up to 20%. Since solar air heating systems must operate at higher pressures than conventional systems to overcome gravel bed store and collector pressure drops, air leakage may be a serious problem. This fear is verified by early solar air heating experiments [3,4,5] which were discussed in Section 1.3.

To predict duct air leakage accurately, a computer model must calculate average pressure differences (or potentials) between the duct interior and atmospheric for each length of duct in the system, as well as make a judgment gauging the workmanship used during installation. TRNSYS components would be required to model the fluid dynamic and thermal performance of their corresponding real components so that these potentials could be estimated. In addition, the potentials of each component must be known so that air leakage from them to the environment can be estimated. The resulting complication

As the temperature within a system component changes, the pressure drop incurred by air flowing through it also changes. The resulting fluid dynamic transients will cause a fan-duct system to "search" for a new equilibrium operating condition, thus causing a change in mass flow. The maximum mass flow rate variation which a solar air system can experience (without dampers changing position) is on the order of 10%; that extreme difference occurring when all of the circulating air is at 20°C as opposed to 150°C. As explained in Section 3.2.2, solar system performance is insensitive to such small changes in mass flow.

Convective heat transfer is governed by the magnitude of the Reynolds and Prandlt Numbers. The Prandlt Number varies less than 1.5% over the 20-150°C temperature range. The Reynolds Number can change significantly, but by evaluating it at an average temperature, mean convective heat transfer coefficients can be calculated.

In general, property variations are unimportant in low temperature thermal systems. This is a valid assumption for solar air system modeling, particularly since the effect of property variations on simulated performance is expected to be small compared to the influences of duct heat loss and air leakage.

Components with time constants much smaller than one hour for all practical purposes respond instantaneously and thus can be modeled with algebraic equations. Components with time constants on the order of one hour must be modeled with the differential equations which describe their thermal behavior. To accurately solve these differential equations the simulation must proceed at a time step which is smaller than the smallest time constant of any component which must be modeled with differential equations. If this time step is less than one hour, weather data values are commonly linearly interpolated between to obtain a forcing function at the required intervals.

Experience from solar water system simulation shows that collectors, heat exchangers and usually space heating loads can be modeled with algebraic expressions while the storage unit must be modeled with differential equations. It is reasonable to believe that the same will be true for air systems.

Figure 1 shows line diagrams of typical solar water and air heating systems. The method of adding auxiliary energy is identical in both. Water and air systems differ in that water systems can only supply solar energy to the load from the tank while air systems can supply solar energy to the load from the gravel bed or directly from the collector. In both systems room temperature is controlled by a 2-stage thermostat; 1st stage is solar heating and 2nd stage is solar heating boosted with auxiliary energy. Under normal operation the fans and pump in both systems will cycle on and off to satisfy the thermostat.

In finite-difference simulation time is divided into discrete intervals called timesteps. Fans and pumps are usually modeled as either on or off during a timestep; when on, the mass flow rate is a constant maximum value, and when off, the mass flow rate is zero. The simulation has difficulty whenever the load demands heat (and thus the load pumps or fans are on for a timestep), but the supply fluid stream is hotter than need be to meet the load. In water systems this situation occurs when the tank is at a high temperature while in air systems it occurs when the gravel bed is at a high temperature or when the collector experiences high radiation levels.

As shown in Figure 1, in a solar water system energy is usually transferred from the storage tank to the load via a crossflow heat exchanger. Whenever the situation described in the last paragraph occurs, the load pump will cycle on and off as the thermostat alternately calls for heat and is satisfied. A simulation cannot follow this cycling because more than one cycle may occur in a timestep and decreasing the timestep is too costly. The rapid cycling of the load pump can be approximated by assuming the pump remains on throughout the timestep. Then the water returning to the tank is given an artificially high temperature so that the energy transferred across the heat exchanger is forced to equal the load over the timestep. Because of the high flowrates required in solar water systems the tank is usually not highly stratified; therefore, the artificial load return temperature does not cause inaccuracies in modeling the thermal behavior of the storage tank.

In air systems, when the load requires energy it may come from either the gravel bed store or the collector as shown in Figure 1. The gravel bed store can be highly stratified and collector performance is usually very sensitive to inlet temperature. Therefore, to accurately simulate the thermal behavior of these components the correct inlet temperatures are needed. Thus, the

the timestep, to that which would be supplied if flow from the energy source were on for the whole timestep, equals the fraction of the timestep during which the system is in the energy source-to-load mode of operation. The rest of the timestep the system can be idle or in some other active mode of operation, depending on the system state at the beginning of the timestep.

The difficulty with this second approach is that TRNSYS components are separate subroutines whose outputs must converge to a unique value each timestep, and it is impossible to have two different values for the same output simultaneously. This situation would be required whenever two active modes of operation share the same timestep. One alternative for achieving the required result would be to have each component model calculate outlet temperatures for all possible inlet temperatures every timestep. A second possibility would be for TRNSYS to iterate, and converge on two different operating states, one after the other, each timestep. Both alternatives require drastic modification of TRNSYS.

To avoid revising TRNSYS, a capacitance load model has been used for solar air system simulations. A detailed description of this model is given in Section 2.3.2.

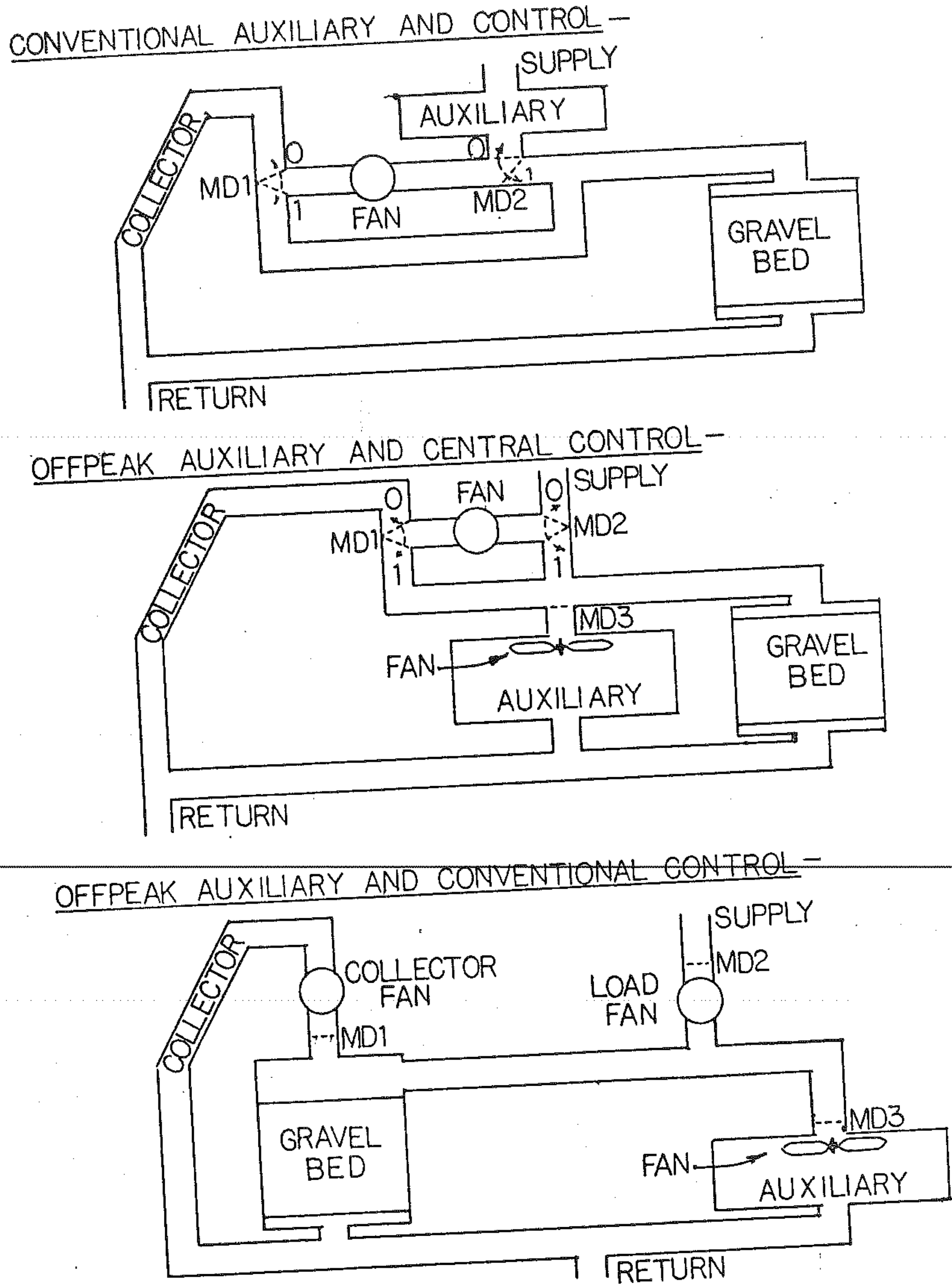


FIGURE 2 SEVERAL RESIDENTIAL SOLAR AIR HEATING SYSTEMS

ential controller and thermostat, respectively. Thus central control is necessary.

The last system shown in Figure 2 is an offpeak auxiliary system which does not require central control. The collector fan and MD1 are controlled by the collector-storage differential controller. In addition, when second stage heating is required the collector fan will shut off and MD1 will close. The load fan and MD2 are controlled only by the room thermostat. The auxiliary furnace (and fan) and MD3 are controlled only by a time clock and a differential controller indicating whether the storage unit needs to be charged. Notice that one extra fan is required in order to use conventional controls in the offpeak system. This suggests the following generalization. Although many types of solar air systems can be controlled with conventional devices, the more complicated the system, the more fans and dampers that could be saved if central controllers were used instead. Furthermore, it is conceivable that some solar air systems in large buildings will be so complicated that central controllers will have to be used. Simulation methods should be as general as possible so that both complicated and simple solar systems can be modeled. All solar air systems, whether controlled with conventional devices or with a central controller, can be simulated using a central controller model. The reverse is not true. Therefore, a central controller model has been used in the Arlington House simulations.

correctly. A second advantage is the ease with which the amount of time spent in each mode of operation (during a simulation) can be calculated. Lastly, it is very simple for other components to use the single integer output to insure that air flow follows the correct path for each mode of operation.

In a real system air follows the path of least resistance. Dampers are opened or closed as the system switches modes so that the desired air flow path is always the path of least resistance. In a simulation the method employed to force air to flow through the correct components can be explained as follows. Each time-step the coded integer sent out by the controller is received by a set of "numerical" dampers (described in Section 2.3.6) which are placed in each possible path the air could take while traveling from the fan outlet to fan inlet. Through the use of parameters, the dampers are individually programmable so that air is forced to flow through the correct circuit of components.

After system control and air flow is taken care of the remainder of the simulation model consists of subroutines which describe the thermal performance of system components. Standard TRNSYS component models can be used for those system components which are common to both air and water systems. Non-standard components written lation, the approximation has been used in a parametric study which indicates that the simple packed bed model described in Section 2.3.1.3 is usually adequate for this purpose.

2.3.1.1 Review of Analytic Solutions

The partial differential equations which describe the thermal performance of a packed bed with forced fluid flow are given below together with the assumptions required to derive them. If it is assumed that:

1) The bed material has infinite conductivity in the radial direction and the fluid is in plug flow,
2) The bed material has zero conductivity in the axial direction,
3) No fluid phase axial dispersion, or conduction takes place,
4) The system has constant properties,
5) No mass transfer occurs,
6) No heat losses to the environment occur,

then the equations describing the system can be written,

$$A\rho_f C_f \varepsilon \frac{\partial T}{\partial \theta} = -\mathring{m} C_f \frac{\partial T}{\partial x} + h_v A (T_b - T) \qquad (2.3\text{-}1)$$

$$A\rho_b C_b \,(1-\varepsilon) \frac{\partial T_b}{\partial \theta} = h_v A \,(T - T_b) \qquad (2.3\text{-}2)$$

London [22]. τ has units of time and represents the packing material capacitance per unit length, multiplied by the bed length, divided by the capacitance rate of the fluid.

In order to verify the accuracy of numerical approximations to the packed bed equations, an analytic solution is needed. The earliest solutions to equations (2.3-1) and (2.3-2) were by Anzelius, Nusselt, Hausen and Schumann as reported in Jakob [23]. All of these solutions were for the single blow case but Hausen also solved the equations analytically for cyclic operation. Anzelius, Hausen and Schumann assumed uniform solid temperature at the start, while Nusselt assumed the initial bed temperature to be an arbitrary function of x. Nusselt, however, neglected the heat capacity of the fluid which is not always valid. Schumann [24] assumed his solution was good only for incompressible fluids but Hausen [23] showed that the influence of temperature on the specific volume of the fluid can be neglected in practical cases.

More recent solutions are by Goldstein [25], Rizika [26], and Myers [27] et al., with Rizika's slightly in error. These solutions illustrate the identical nature of the packed bed, regenerator, and insulated duct problems for the case of a step change in the inlet fluid temperature with all temperatures initially at a single and

$$V_H = \frac{\rho_b C_b (1-\varepsilon)}{\rho_f C_f \varepsilon} = \frac{\tau \dot{m}}{\rho_f \varepsilon A L} \tag{2.3-9}$$

NTU can be estimated from equation (2.3-5) using either the correlation for h_v reported by Löf and Hawley [29] or the correlation from Dunkle and Ellul [30]. When used in the Schumann Model equations, Jeffreson shows by comparing predictions of packed bed behavior with experimental results, that the modified NTU or NTU_c adequately accounts for these additional effects. Jeffreson suggests that the Peclet Number correction term in equation (2.3-7) may not be necessary. This is the case unless the heat transfer correlation employed was made with a mass transfer experiment, as heat transfer experiments cannot separate dispersion effects from the fluid-to-particle convection mechanism. The Biot Number correction, however, appeared to be valid for values of Bi up to about 4.

Concerning assumption 4 it is reasonable to assume that material properties are constant over the range of temperatures encountered in most solar energy thermal systems. The assumption of no mass transfer is correct for beds of non-adsorbing solids such as glass spheres, but most types of gravel adsorb significant amounts of water. Close et al. [31] have observed experimentally the increased storage capacity effect which mass transfer environment. This approximation will over-estimate the loss since radial temperature gradients within the bed, needed to cause radial heat flow, exist but are neglected. The error introduced by this approximation is very small since typical storage units are well insulated so that over a period of time losses are small compared to energy leaving the bed to heat the load. With this addition, equations (2.3-3) and (2.3-4) become

$$\frac{\partial T}{\partial (x/L)} = NTU_c \ (T_b - T) + \frac{UPL}{\dot{m}C_f} \ (Tenv - T) \qquad (2.3\text{-}10)$$

$$\frac{\partial T_b}{\partial (\theta/\tau)} = NTU_c \ (T - T_b) \qquad (2.3\text{-}11)$$

2.3.1.2 Numerical Approximations to the Analytic Solution

For use in simulations, an algorithm is needed which can solve the Schumann Model equations thousands of times with very little computer time. Brute force methods such as fourth order Runga-Kutta or Predictor-Corrector are ruled out on cost considerations alone. The lower order TRNSYS Predictor-Corrector algorithm cannot be used because the Schumann Model equations require very small timesteps, and to force the simulation to proceed with such small stepsizes would be far too costly. Consequently, the approach has been to use a separate integra-

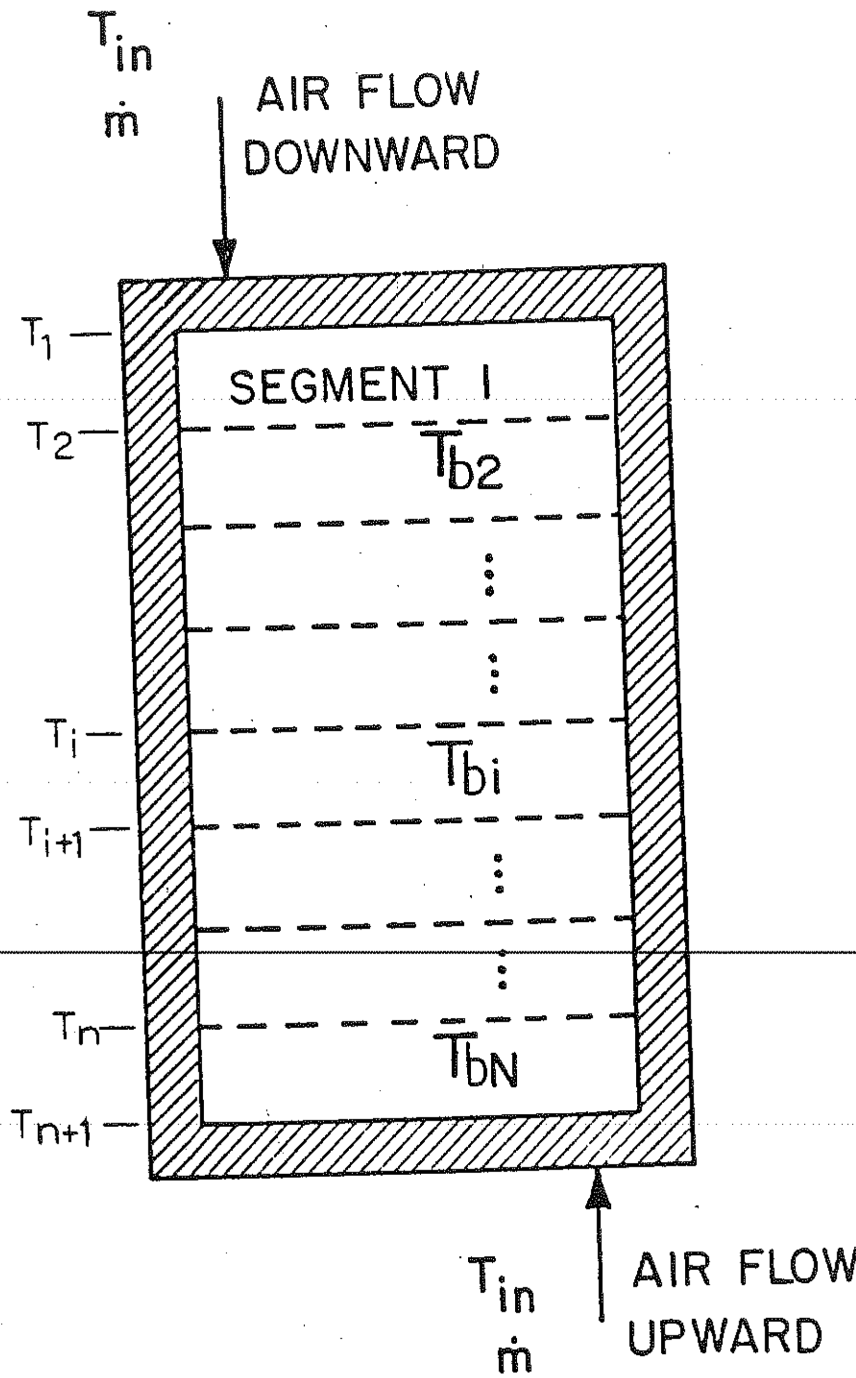


FIG. 3 . PACKED BED DIVIDED INTO N SEGMENTS

$$\frac{\partial T_b}{\partial(\theta/\tau)} = (L/\Delta X)(\eta)(T_i-T_{bi})+\frac{UPL}{\dot{m}C_f}(Tenv-T_i) \quad (2.3\text{-}13)$$

As before the exit fluid temperature of node i can be calculated after the new bed temperature is determined.

This approximation also reproduces the Schumann Model test case with small time-distance stepsizes. Slight improvement was obtained by averaging the energy transfer rates occurring at the beginning and end of the timestep as in Crank-Nicolson formulations.

A better approximation can be obtained if a new expression for η is derived assuming the bed temperature has a linear slope, s, throughout each node while the fluid temperature has an exponential distribution. If ~~s of node i~~ is assumed to be the average of the slopes between T_{bi} and the bed temperatures of adjacent nodes, the resulting expression is

$$\eta = 1.-e^{-NTU_c(\Delta X/L)} - \left\{\frac{S_i\Delta x}{2(T_i-T_{bi})}\right\}\left\{e^{-NTU_c(\Delta X/L)}+1.\right\}$$

$$-\left\{\frac{S_iL}{NTU_c(T_i-T_{bi})}\right\}\left\{e^{-NTU_c(\Delta X/L)}-1.\right\}$$

(2.3-14)

By substituting this into (2.3-13) and approximating in the Crank-Nicolson manner, the effectiveness-NTU method to as the "leap frog" method.

The temperature difference approximations in the two NTU_c terms are not taken at the same time-distance location; therefore, energy conservation is not an algebraic reality but depends on the stepsizes and NTU_c. When the energy balance closes, equations (2.3-15) and (2.3-16) reproduce the Schumann Model test case; therefore, the energy balance can be used as a criterion to determine the most accurate stepsizes for each NTU_c. Since it is easier to check accuracy with an energy balance than to compare calculated temperature profiles to the test case analytic solution, (2.3-15) and (2.3-16) have been used to describe the thermal behavior of the packed bed store with forced fluid flow.

As mentioned in Section 2.3.1, the non-flow condition simplifies to a one dimensional conduction problem with insulated end boundary conditions if natural convection is suppressed. Katto et al. [34] experimentally verify a critical Rayleigh Number criterion for the onset of natural convection in packed beds heated from below and according to this criterion, thermal stores of practical size will experience natural convection. Since flow through the bed is arranged so that generally higher temperature gravel is at the top of the bed, this situation will not be considered.

with analytic solutions.

2.3.1.3 Infinite NTU Model

The packed bed component model developed in the last section is costly when included in long term simulations of solar air heating systems. A simpler model has been developed by Hughes, Klein and Close [37] and a brief summary of how the model was derived is included below.

Many long term simulations of the solar air heating system labeled "Conventional Auxiliary and Control" in Figure 2 were performed with a gravel bed model based on equations (2.3-15), (2.3-16) and (2.3-17). All of the important parameters describing the behavior of solar air heating systems, including NTU_c, were varied independently of each other for these simulations. Figure 4 illustrates the sensitivity of the solar air system performance to the parameter NTU_c. The cross-hatched region represents how other system parameters interact with NTU_c to affect system performance. The arrow indicates that any change in other system parameters which results in larger collector losses will make system performance more sensitive to NTU_c. The most significant information to be gained from Figure 4 is that system performance is insensitive to NTU_c, no matter what other system parameters are, if NTU_c is greater than 10. This is important since gravel beds of practical size have NTU_c's much larger than 10.

Since NTU_c does not affect the performance of air systems of practical design for values of NTU_c larger than 10, the same simulated performance is predicted if NTU_c were infinity. Assuming NTU_c is infinity equations (2.3-15) and (2.3-16) can be simplified to

$$\frac{\partial T}{\partial(\theta/\tau)} = L\,\frac{\partial T}{\partial x} + \frac{UPL}{\dot{m}C_f}\,(Tenv - T) \qquad (2.3\text{-}20)$$

Equation (2.3-20) can be approximated numerically using the finite-difference method. Sufficient accuracy is obtained using large time and distance stepsizes. Figure 5 compares the accuracy of this method with that of the Schumann Model in long term simulations; N is the number of nodes used. Since the simple model results in a significant cost savings, it has been used in the Arlington House simulations. A more detailed description of the model appears in the TRNSYS manual [19].

2.3.2 One Lump Capacitance Load Model

The one lump capacitance load model calculates a load using the UA degree-hour concept. As explained in Section 2.1.2.1, the model must include a capacitance effect in order to output the correct return temperature to other components in the simulation. The load is calculated as follows

$$Q_L = UA(T_R - T_A) - Q_{gen} - Q_{gain} \tag{2.3-21}$$

where

Q_L = space heating load rate,

UA = overall building conductance-area product adjusted to include infiltration,

T_R = room temperature,

T_A = ambient temperature,

Q_{gen} = time varying internal heat generation rate such as losses from storage units,

Q_{gain} = constant internal heat generation rate due to lights, people, etc.

An artificial load capacitance is assumed, which ensures numerical stability and approximates the actual house capacitance. An energy balance on the conditioned space yields

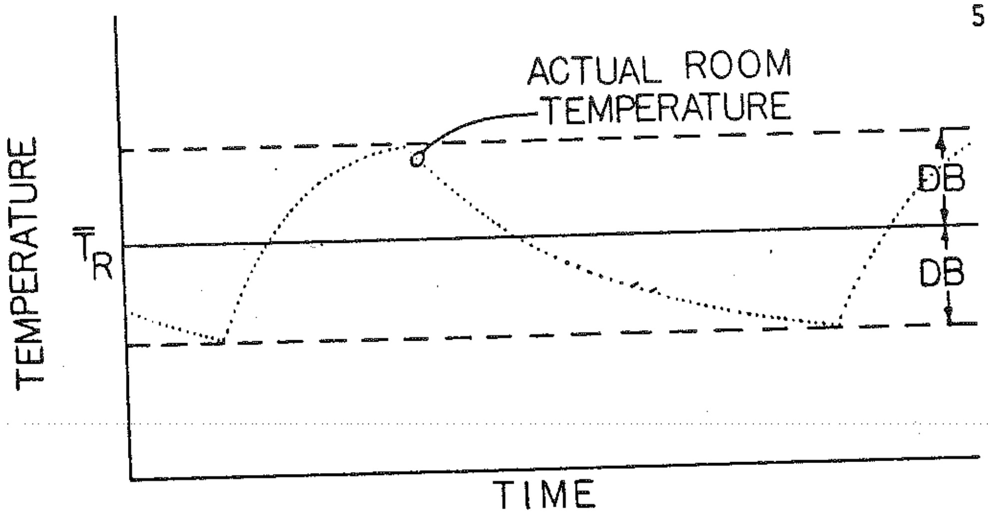


FIGURE 6 ROOM TEMPERATURE FLUCTUATIONS

The thermostat control function, γ, is generated by tracing room temperature using CAP as estimated above. Whenever room temperature falls below $\overline{T}_R$-DB, γ=-1 meaning 1st stage heating is required. 1st stage heating means the load will receive energy from the solar gravel bed or the collector. If room temperature remains below $\overline{T}_R$-DB for a timestep, γ=+1 meaning second stage heating is required. 2nd stage heating means the load will receive energy from the auxiliary gravel bed or auxiliary furnace. γ will remain -1 or +1 until room temperature is greater than $\overline{T}_R$+DB, at which time γ becomes zero.

The TRNSYS one lump capacitance load model requires Q_{gen}, T_A, the inlet fluid temperature and the incoming fluid mass flow rate as inputs. It outputs Q_L and γ and has Q_{max}, DB, $\overline{T}_R$ and C_f as parameters.

2.3.5 Air Collectors

2.3.5.1 Flat Plate Collectors

The air heating flat plate collector model calculates the solar energy gain, Q_u, using the relationship developed by Hottel, Whillier and Bliss [38,39,40] which is

$$Q_u = A_c F_R [H_T(\tau\alpha) - U_L(T_{in} - T_A)] = \dot{m}C_f(T_{out} - T_{in}) \quad (2.3\text{-}25)$$

where

$$F_R = \frac{\dot{m}C_f}{A_c U_L}[1. - \exp(\frac{F' U_L A_c}{\dot{m}C_f})] \quad (2.3\text{-}26)$$

Although the expression was developed for a water collector of tube and sheet construction, it holds for most other flat plate collector designs provided that a new expression for the collector efficiency factor, F', is derived for each [38,39,40]. The above model is an algebraic expression, Klein et al. [41] have verified that thermal capacitance effects can be neglected in collectors of standard design.

H_T, the total solar radiation incident on the tilted surface, is estimated from data available for solar radiation incident on a horizontal surface. Beam and diffuse components are estimated from the total radiation on a

An Owens-Illinois collector is constructed from two co-axial, elongated glass tubes sealed together with an evacuated space in between as in Figure 7. The inside tube is the absorber and has a selective coating on its outer surface which is protected from degradation by the vacuum. Typically, tubes are mounted as pairs projecting from opposite sides of a manifold with a common delivery tube down the center as in Figure 8. When air is the transfer fluid, manifolds are usually split so that air passes through one tube pair while flowing from the supply to return manifolds as shown in Figure 8. The tubes and manifolds are mounted above a diffusely reflecting backing surface, usually the roof.

Horizontal surface total radiation data is converted to tilted surface beam and diffuse components as explained in the last section. The energy incident on the absorber tubes is a complex function of declination, latitude, backing surface slope and reflectance, hour angle and glass optical properties. Briefly, the analysis considers beam and diffuse radiation directly intercepted by the absorber tube and beam and diffuse radiation diffusely reflected from the backing surface which is intercepted by the absorber tube. The total of these is multiplied by an effective transmittance-absorptance product to yield the energy incident on the absorber tube.

Energy balances on the absorber tube, air in the annular space, delivery tube, and air within the delivery tube yield a set of simultaneous equations. The numerical solutions of the resulting equations are too costly for long term simulations; therefore, a three dimensional performance map of a single tube pair is made for a given flow rate. Given the energy incident on the absorber tube per unit area, the air inlet temperature and ambient temperature, an interpolation routine calculates the efficiency. The useful energy gained by the collector bank equals the product of efficiency and the energy incident on all of the absorber tubes in the bank.

A method of controlling when the O-I collector should be on or off is desired. With the flat plate collector, a differential controller between the collector plate (thermistor located near the outlet manifold) and the cold end of the gravel bed is sufficient. The thermistor on the collector plate senses plate temperature with no air flow and the outlet air temperature with flow. In Section 2.3.7 a simple algebraic relation is developed between the plate temperature and the outlet temperature (if air flow were turned on) which allows the designer to calculate the deadbands for the differential controller. Such a relation has not yet been

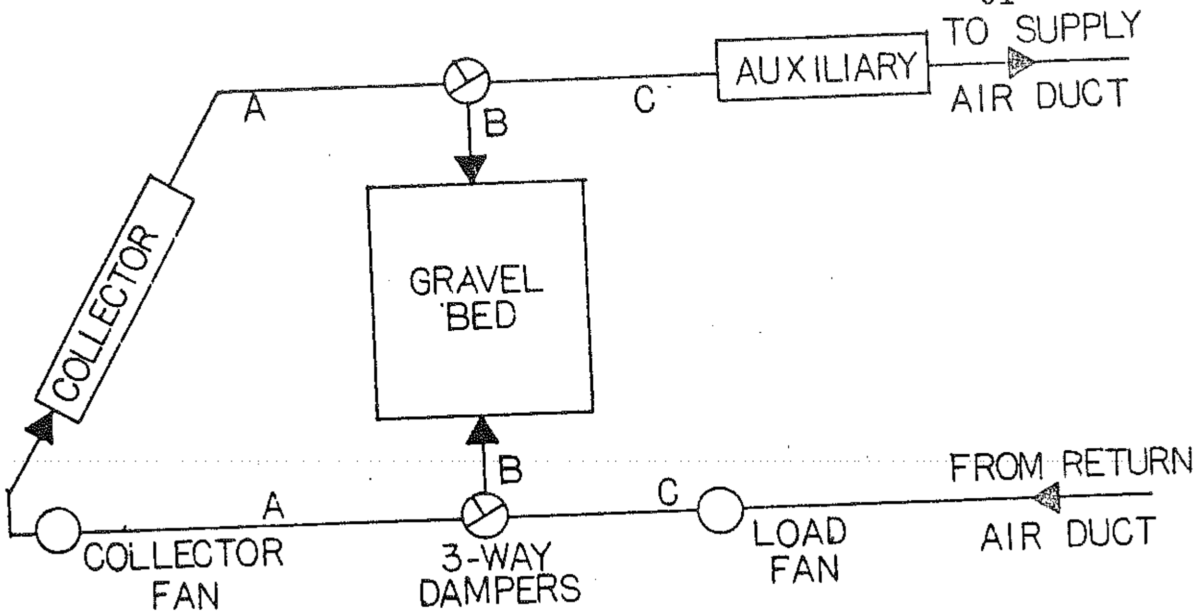


FIGURE 9 DUCT SECTIONS REQUIRING 'NUMERICAL' DAMPERS

flowrate, and all of the possible temperatures, depending on mode, which the inlet air could have. With the use of parameters, the damper is programmed to pass along the correct mass flow rate, air temperature and air flow direction as outputs. Of the air system component models written to date, only the packed bed requires the direction of air flow as an input.

2.3.7 Central Controller

The central controller is not a general component and therefore it must be rewritten to control different systems or to approximate different strategies. Figure 10 is a schematic of the controller used for some of the Arlington House simulations. It was written with enough flexibility to simulate the three alternative systems described in the next chapter. Definitions of the variables introduced in Figure 10 are as follows

γ -- control function from the thermostat

$\gamma=0$ no heat required

$\gamma=-1$ first stage heat required

$\gamma=+1$ second stage heat required

TS -- temperature at cold end of gravel bed store

TC1 -- collector outlet temperature with TS as the inlet temperature

TC4 -- collector outlet temperature with room temperature air at the inlet

TAVE -- average temperature of gravel bed

TMIN -- minimum average temperature of the gravel bed required at the end of the offpeak heating period to ensure there is enough energy in storage to heat the house the following day

DBHFC, DBCS -- control temperature deadbands, see discussion

IOPK -- offpeak auxiliary option

IOPK=1 offpeak auxiliary

IOPK=2 normal auxiliary

TIME1 -- beginning of offpeak period

TIME2 -- end of offpeak period

CS -- Collector to Storage

Collector with flow

$$\dot{m}C_f(T_{out}-T_{in}) = F_R A_c [H_T(\tau\alpha)-U_L(T_{in}-T_A)] \quad (2.3\text{-}27)$$

By combining (2.3-26) and (2.3-27) it is seen that air flow should remain off until

$$(T_p-T_{in}) > \left(\frac{\dot{m}C_f}{F_R A_c U_L}\right) \times (T_{out}-T_{in})_{min} = TD \quad (2.3\text{-}28)$$

where $(T_{out}-T_{in})_{min}$ is the lower deadband.

Since the flowrate through the collector, air properties, and collector parameters are all nearly constant,the upper deadband, used when $\dot{m} = 0$, can be defined as a constant multiple of the lower deadband which is DBHFC for the HFC mode and DBCS for the CS mode.

For the HFC mode

$$\begin{aligned} TD &= DBHFC \text{ if } \dot{m} > 0 \\ TD &= (DBHFC)(DBNF) \text{ if } \dot{m} = 0 \end{aligned} \quad (2.3\text{-}29)$$

and for the CS mode

$$\begin{aligned} TD &= DBCS \text{ if } \dot{m} > 0 \\ TD &= (DBCS)(DBNF) \text{ if } \dot{m} = 0 \end{aligned} \quad (2.3\text{-}30)$$

where

$$DBNF = \frac{\dot{m}C_f}{F_R A_c U_L} \quad (2.3\text{-}31)$$

conditions air leakage and heat losses from ducts in unconditioned spaces such as in the attic may cause a net energy loss to the system even when the collector outlet temperature is higher than its inlet temperature. Also, running the collector with a low DBCS serves only to push the high temperature section in the rock pile further down the bed until it eventually comes out the other end whereas leaving it idle until the collector becomes hotter tends to preserve stratification in the rock pile. An economic analysis cannot take these effects into account in a straightforward manner; therefore, DBCS cannot be calculated with much accuracy. DBCS = 5°C was used for the Arlington House simulations.

The above analysis for predicting outlet air temperatures from plate temperatures only applied for the flat plate collector. Owens-Illinois collectors were controlled by setting DBNF = 1 and requiring the Owens-Illinois collector model to always calculate the outlet air temperature as if it were on. But since there are two possible collector inlet temperatures, room and storage, the component must calculate two outlet temperatures each timestep. This explains why TC1 and TC4 are needed. When controlling a flat plate collector, TC1 and TC4 are identical.

interpolates for values between the hourly data intervals.

The service hot water load is modeled as a constant 20 Kg/HR mass flowrate of water required at 60°C for 13 hours of the day. The supply main or well water temperature is assumed to be 11.4°C throughout the year. A more sophisticated hot water load distribution throughout the day has little effect on system performance.

T_{min} is the minimum average temperature of the gravel bed needed at the end of the offpeak period to ensure that the following days load will be met. It is introduced to the simulation as a constant which can change every month. The monthly increment is an attempt to include in the control scheme the knowledge that the maximum possible daily load changes with time of year. T_{min} is used to control the offpeak heating of the gravel bed storage unit as described in Section 2.3.7.

Knowing the size of the thermal store, the number of offpeak hours in a day, and T_{LD} and T_{LH} (from weather bureau monthly summaries of the design year months) as defined below, T_{min} is calculated as follows

$$T_{MU} = 20. + \frac{UA(18.3-T_{LH})}{\dot{m}C_f} \qquad (2.3\text{-}32)$$

Table 1. T_{min} for 10 Offpeak Hours and 22 m^3 Storage for the Arlington House

Month	T_{LD}(°C)	T_{LH}(°C)	T_{MU}(°C)	T_{min}(°C)
Oct 1955	1.1	0.0	30.1	37.3
Nov 1949	-11.7	-18.9	40.6	53.1
Dec 1949	-20.5	-24.4	43.7	59.9
Jan 1954	-20.0	-26.1	44.7	60.7
Feb 1955	-20.5	-23.9	43.5	59.7
Mar 1954	-9.4	-12.8	37.3	48.9
Apr 1956	-1.1	-2.8	31.7	39.8

2.3.8.2 Standard TRNSYS Components

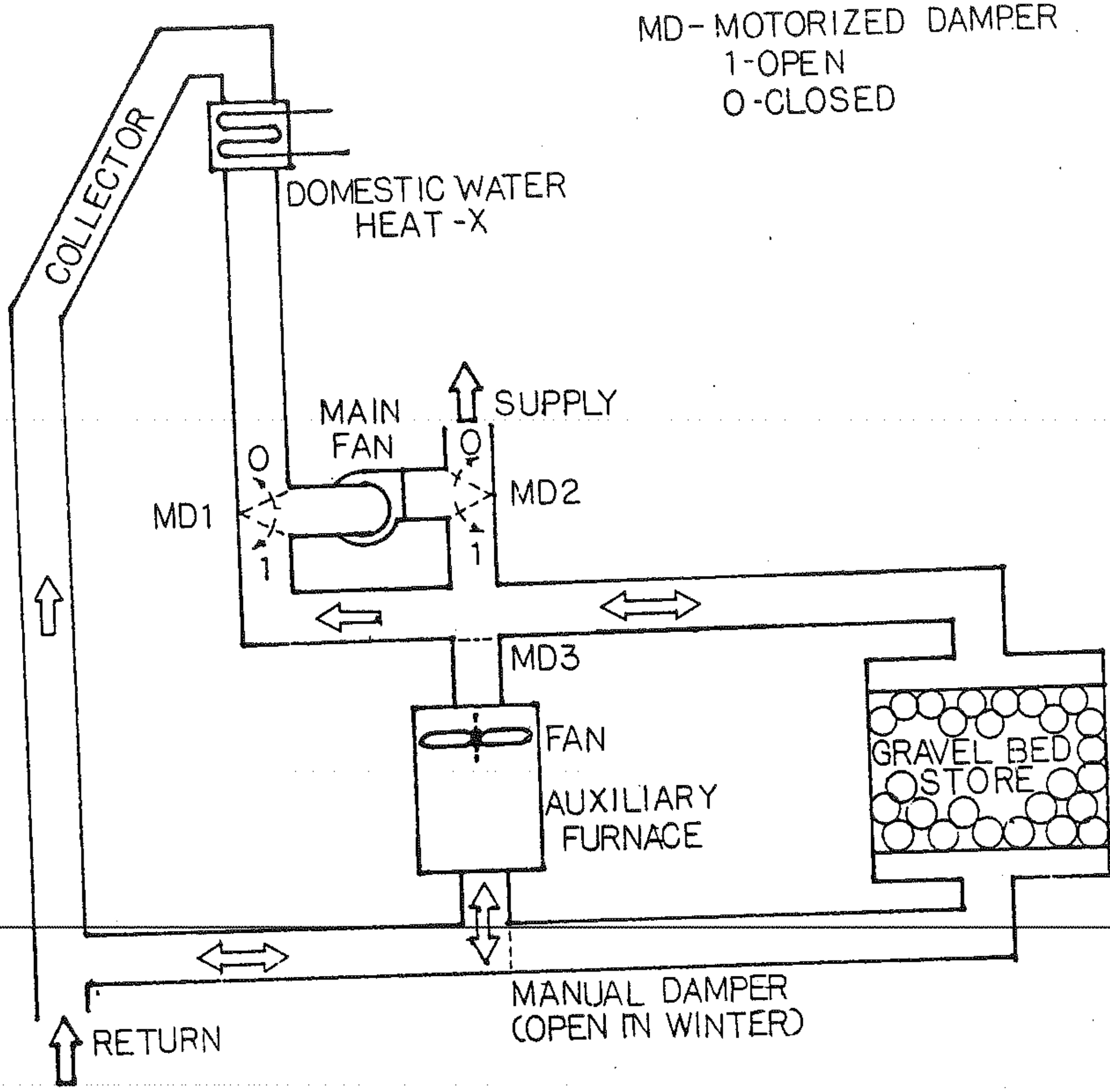
Descriptions of all of the standard TRNSYS components are contained in the TRNSYS manual [19]. Standard components used in the Arlington House simulations include the card reader, cross flow heat exchanger, on-off controller, water pump, water tank, modulating hot water auxiliary boiler, integrator and printer.

tral store with commercially available offpeak electric "heat banks" in each room. In either case the stored auxiliary energy does not affect the temperature at which the solar portion of the system will operate.

Figure 11 illustrates a design for the dual gravel bed store system. This design would require non-conventional controls (perhaps a central digital controller) because some damper states depend on more than one input. If more fans and dampers were used, each could be controlled with one input. Also, if more fans were used, fewer motorized dampers would be required assuming the method of control remained the same. In general, the best compromise between fans, motorized dampers and controls is determined by economics.

In winter operation, whenever there is energy to collect and the building demands heat, the main fan circulates air through the collectors to the load. After the load is satisfied, air will circulate down through the solar gravel bed and back to the collectors. When there is no energy to collect and the building demands heat, the main fan circulates air up through the solar gravel bed to the load. If the solar storage unit is depleted, air is diverted through the auxiliary gravel bed and then to the load. During offpeak hours whenever the main fan is off and the auxiliary store is not fully charged, the electric furnace is turned on and air is circulated through it to the auxiliary gravel bed by the furnace fan. In the summer, the load thermostat is disabled and the manual damper is closed to allow energy for domestic hot water to be collected without heating the gravel bed store.

A differential controller between the collector outlet and the cold end of the solar gravel bed determines when there is energy to collect. A two-stage thermostat in the conditioned space controls room temperature. The first stage calls for heat from the collector or solar gravel bed while the second stage calls for heat from the auxiliary gravel bed. When the main fan is off, offpeak charging of the auxiliary store is controlled by a time clock and a differential controller which compares the average gravel bed temperature obtained from a thermopile (or several thermocouples) to a set minimum average temperature which can be changed each month. A differential controller between the preheat tank and air upstream from the heat exchanger controls the operation of the pump which circulates water through the cross-flow heat exchanger. The fan and damper states, when System A is controlled as described above, are given in Figure 11. The abbreviations for the modes of operation have been defined in Section 2.3.7.



FAN AND DAMPER STATES -

MODE	MD1	MD2	MD3	MAIN FAN	FURNACE FAN
CS	1	0	0	1	0
HFS	0	1	0	1	0
HFC	1	1	0	1	0
HSA	1	1	1	0	1
IDLE	-	-	-	0	0

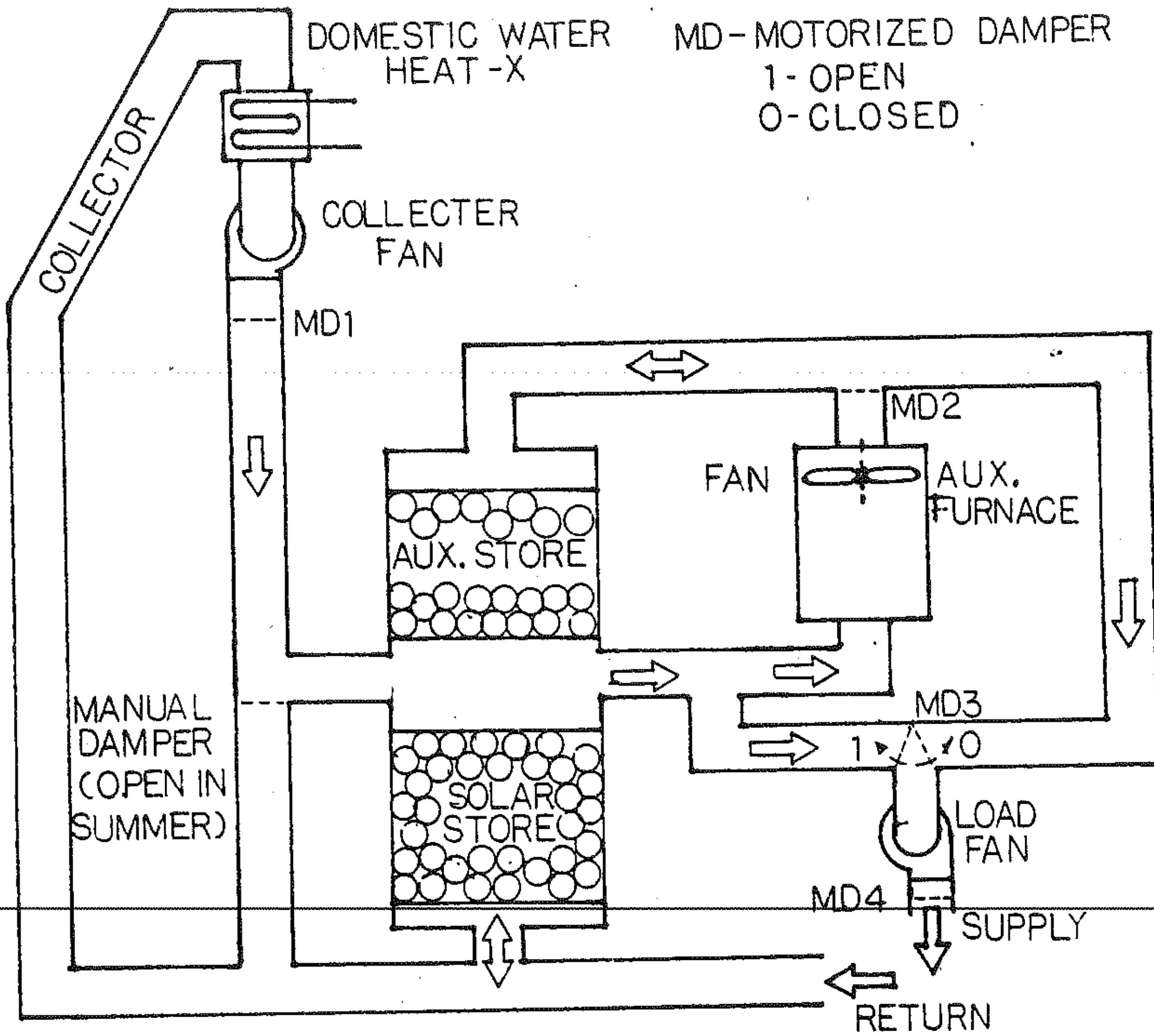
FIGURE 12 SYSTEM B

several thermocouples) to a set minimum average temperature which can change each month. A differential controller between the preheat tank and air upstream from the heat exchanger controls the operation of the pump which circulates water through the cross-flow heat exchanger. The fan and damper states, when System B is controlled as described above, are given in Figure 12. The abbreviations for the modes of operation have been defined in Section 2.3.7.

System B can be simulated using the central controller model shown in Figure 10, including the portion within the dotted lines. The minimum average storage temperature required at the end of the offpeak period in order to ensure that enough energy is available to meet the next day's load is introduced to the simulation as a function of month as described in Section 2.3.8.1.

3.1.3 System C

System C is a two gravel bed system devised by Austin Whillier. It can be controlled with conventional devices and has some of the characteristics of Systems A and B. If all of the system parameters are the same, System C is thermally identical to System A except that air flows through both gravel beds when second stage heating is required while there is no energy to collect.



FAN AND DAMPER STATES -

MODE	MD1	MD2	MD3	MD4	COLLECTER FAN	LOAD FAN	FURNACE FAN
CS	1	0	-	0	1	0	0
HFS	0	0	0	1	0	1	0
HFA	0	0	1	1	0	1	0
HFC	1	0	-	1	1	1	0
HSA	0	1	-	0	0	0	1
IDLE	-	-	-	-	0	0	0

FIGURE 13 SYSTEM C

3.2 Simulation Results

3.2.1 Effect of System Configuration and Collector Type

Simulations, from October through April using the Madison design year weather data and Arlington House parameters, were performed with Systems A, B and C employing both Owens-Illinois (O-I) and standard two cover non-selective flat plate (FP) collectors. A complete list of parameters used appears in Appendix A. Table 2 contains the results; runs are coded with the system letter followed by the collector abbreviation.

The Arlington House roof has room for 50.6 m^2 of flat plate collectors in .92 by 1.98 m. modules, or 194 O-I collector tube pairs when spaced one diameter apart. Work by Grunes [46] and Eberlein [45] indicates that 8.18 Kg/hr of air per tube pair (4 SCFM/tube pair) is a reasonable compromise between pressure drop and thermal performance. Fortunately this results in a total air flow through the collector bank which is adequate to achieve the recommended diffuser velocities for comfort air distribution in the 116 m^2 house. To prevent the minor effect of gravel bed stratification from clouding the desired comparisons, the same total air mass flow rate was used through the flat plate collector bank. This is equivalent to 31.4 Kg/hr-m^2 of collector (1.5 SCFM/ft^2).

Aside from the collectors, all other system parameters were kept the same in all of the simulations where possible. The volume of the solar gravel bed store in System A was 22 m^3, as was the volume of the gravel bed in System B. The combined volume of the split gravel bed of System C, including a plenum space of .21 m between the sections, was also 22 m^3. All of the flow rates in all of the systems were identical. The space and domestic water loads, as well as all control temperature differences were also the same.

As can be seen from Table 2, System A performed the best, followed by C and B. As mentioned in Section 3.1.3, System C is thermally identical to A except that room air passes through the solar gravel bed on its way to the auxiliary store in the HFA mode. This suggests that System C should be superior to A since the solar gravel bed is always driven to the lowest possible temperature; however, the larger volume of the solar gravel bed in System A more than offsets this effect. System B was not expected to perform as well as the others because of the consistently higher temperature at which the gravel bed store, and thus the collectors, must operate. This is not to say that System B with a very large storage unit would not perform as well as the other systems.

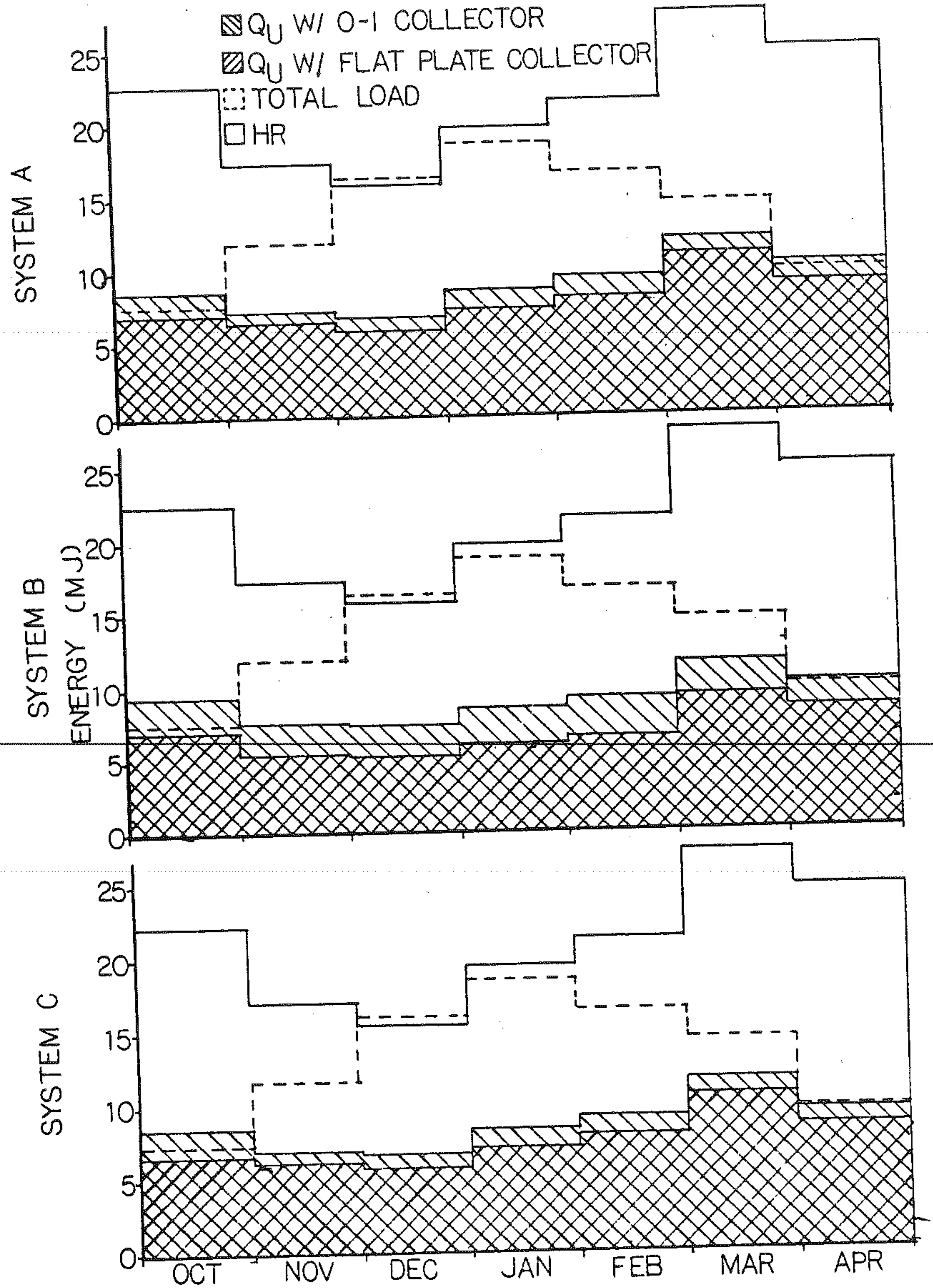


FIGURE 14 MONTHLY PERFORMANCE OF SYSTEMS A,B AND C

Table 3

Effect of Tube Spacing and Flow Rate on the Performance of System B with O-I Collectors

Run	Tube Pairs	$\dot{m}_{coll.}$ (kg/hr)	Q_u	Q_L	Q_{AUX}	Q_{WL}	Q_{WAUX}	% By Solar
BOIT5	156	1595.	53.15	81.65	38.85	12.18	2.26	56.2
BOI	194	1587.	59.14	81.65	33.90	12.78	1.88	62.1
BOIH3	260	1595.	64.75	81.65	29.60	13.40	1.59	67.2
BOI5	194	1984.	58.15	81.65	34.60	12.59	2.02	61.1
BOI	194	1587.	59.14	81.65	33.90	12.78	1.88	62.1
BOI3	194	1190.	58.42	81.65	35.05	13.07	1.72	61.2

3.2.3 Effect of Domestic Hot Water Heat Exchanger Position

For System B as shown in Figure 12 it is not immediately obvious how domestic hot water can best be obtained. One possibility is to embed a water pre-heat tank in the gravel bed store and thus avoid the need for a cross-flow heat exchanger. Unfortunately this requires charging the gravel bed in the summer to obtain hot water and the resulting heat gains would cause occupant discomfort.

If a heat exchanger is used it should be in the collector "loop" so that heat can be transferred to the pre-heat tank while the collector operates. Normally the heat exchanger is placed at the collector outlet but since the storage unit in System B operates at a high temperature, it may be better to place it at the inlet. Simulations were performed to determine the best location and from Table 4, it appears not to matter where in the collector loop the heat exchanger is placed.

With the heat exchanger at the outlet more energy is transferred to the domestic water system so that, on the average, the rest of the system operates at a lower temperature. The heat exchanger at the inlet drives down the collector inlet temperature somewhat but, on the average, not as low as before since less energy is transferred to the domestic water system. Since O-I collector performance is rather insensitive to inlet temperature, the difference between these two cases is small.

3.2.4 Effect of Simplified Controls

In solar air heating systems the collector receives air from either the conditioned space or the cold end of the gravel bed. Ideally the collector should be controlled by comparing the outlet-inlet air temperature difference to a minimum required dead band. It requires two differential controllers to compare the collector outlet temperature to room and cold end gravel bed temperatures, respectively. Typical solar air collectors have only one controller because these temperatures are nearly identical throughout the winter, while in the spring and fall the system is over-designed so that collector control is not critical. However, in offpeak auxiliary systems similar to B the cold end gravel bed temperature is generally not equal to room temperature so simulations were performed to determine the performance penalty incurred by using one rather than two differential comparators to control the collector.

As mentioned in Sections 3.1.2 and 2.3.7, the offpeak charging of the gravel bed is controlled by comparing the average gravel bed temperature obtained from a thermopile

Table 5

Effect of Simplified Controls on the Performance of System B

Run	No. collector controllers	T_{min}	Q_u	Q_L	Q_{AUX}	Q_{WL}	Q_{WAUX}	% By Solar
BOI	1	constant	57.9	81.65	36.65	13.30	1.60	59.7
BOI	1	varies monthly	59.14	81.65	33.90	12.78	1.88	62.1
BOI	2	varies monthly	61.04	81.65	32.10	12.83	1.98	63.9

Table 6

Effect of Storage Volume

Run	Storage Volume	Q_U	Q_L	Q_{AUX}	Q_{WL}	Q_{WAUX}	% By Solar
BOI	16 m^3	58.64	81.65	34.90	13.11	1.77	61.3
BOI	22 m^3	59.14	81.65	33.90	12.78	1.88	62.1
BOI	28 m^3	59.40	81.65	33.60	12.61	1.94	62.3

requires the least hardware but System C has simpler controls. Since a central digital control box was being considered for Arlington House for display and instrumentation reasons anyway, System B was chosen to save on equipment costs.

O-I collector tubes are presently expensive and therefore spacing them far apart, over a large area, is desirable. Since the Arlington House roof area was fixed, the tube spacing chosen was a compromise between cost and the portion of the load met by solar. A spacing of one tube diameter has been chosen, requiring 388 tubes total. Table 3 shows that the thermal performance of the tube bank within a system is not a strong function of flow rate; therefore, 8.18 Kg/hr (4.0 SCFM) of air per tube pair has been chosen based on pressure drop considerations.

A schematic of System B appears in Figure 12. Table 4 suggests that the domestic hot water heat exchanger should be placed at the collector outlet. System B requires three motorized dampers and two fans (one in the auxiliary furnace). In addition, there is a manual damper to allow a permanent by-pass of the storage unit in the summer. Depending on how badly the system leaks air, a backdraft or motorized damper may also be required in the return air duct. The domestic hot water heat exchana manual over-ride will allow heating directly from the auxiliary furnace. When the main fan is off, offpeak charging of the gravel bed store is controlled by a time clock and differential controller which compares the average gravel bed temperature to a minimum required value. Table 5 indicates that the minimum required average temperature can remain constant throughout the year at the value required in the coldest month. The central control box will be designed to allow changing the minimum average gravel bed temperature conveniently, however, so that the occupants or researchers can use daily weather forecasts to prevent adding unneeded auxiliary energy. All differential controllers and the time clock will be built into the central control box.

O-I collectors or the auxiliary furnace may heat the upper portion of the gravel bed to very high temperatures. Air supplied to the load at a very high temperature can be a safety hazard or cause discomfort. This can be prevented by placing a small fan with solenoid damper in the supply duct to mix basement air with supply air if the latter is above 60°C.

2) To simulate solar air systems a capacitance load model is required so that a floating room temperature can be used to control heat addition to the load. In this way the temperature of the air returning to the system from the load during heat addition can always be room temperature while maintaining the overall energy balance between the system and the load.

3) Other components needed to simulate solar air systems such as the fan, damper, and controller models have been written. The simple gravel bed model, and other model simplifications, allow solar air systems to be modeled using TRNSYS at a cost similar to that for solar water systems.

4) Two basic offpeak electric auxiliary systems have been studied; the system where the storage of auxiliary is isolated from the storage of solar energy and the system where all energy is stored together. Both types of systems can be controlled with conventional devices or with central digital controllers. System A is the dual storage system with central control and ignoring heat losses from the auxiliary storage unit, its performance is identical to that of a non-offpeak auxiliary system with the same system parameters. System C is the

Offpeak heating of the gravel bed is controlled by heating it to a minimum required average temperature by the end of the offpeak period to ensure that enough energy is stored to meet the load until the beginning of the offpeak period the following day. Initially the minimum required temperature was changed every month so that the storage unit would not have to be maintained year-round at the high temperature required in the coldest month. However, simulations indicate that the minimum average gravel bed temperature can remain the same year-round because the O-I collector is relatively insensitive to inlet temperature and in a Madison climate the solar system is oversized in the spring and fall so that storage is normally hotter in those months than required in the winter anyway.

The effect of controlling the collector, when it is heating the load and when it is heating storage, with a different differential controller in each case was studied. It was found that only a small performance penalty was incurred by controlling both operations with one differential controller between the collector outlet and cold end of the gravel bed.

meet generating capacity. Whether space heating, water heating and other loads which can be shifted to offpeak periods make up a large enough portion of a utility's total load to allow ripple control to save the expense of increasing generating capacity is a question that only the individual utility can answer.

3) Before offpeak auxiliary solar systems for wide-spread use can be designed, much information must be obtained from electric utilities. Further work is needed to identify the advantages and disadvantages of various central site control schemes, various proposed rate schedules and various time distributionsof customer demand from the perspective of solar energy development.

FLAT PLATE COLLECTORS

Efficiency factor, F'	.80
Transfer fluid specific heat, C_f	1.012 KJ/Kg-°C
Plate emittance, ϵ_p	.94
Plate absorptance, α_p	.94
Number of glass covers, N	2.
Cover system transmittance, τ	.8
Bottom and edge loss coefficient, U_{be}	2.0 KJ/hr-m^2-°C
Tilt from horizontal (due south), s	59. degrees
Air flow rate per collector area, G	32. Kg/hr-m^2
Collector area, A_c	50.6 m^2

Service Hot Water System

Preheat tank volume, v	.454 m^3
Tank overall loss coefficient, U	1.5 KJ/hr-m^2-°C
Cross-flow heat exchanger effectiveness, ϵ	.52
Water side flow rate, $\dot{m}_w$	450. Kg/hr
Air side flow rate, $\dot{m}_a$	1600. Kg/hr
Auxiliary heater set temperature, T_{set}	60. °C
Mains water temperature, T_{main}	11.4 °C
Flow rate consumed, 6 AM-7 PM, $\dot{m}_{taps}$	20. Kg/hr

Space Heating Load

Average overall loss coefficient-envelope area product (includes .6 air change/hr infiltration), UA	900. KJ/hr-°C
Constant internal generation rate, Q_{gen}	1500. KJ/hr
Thermal capacitance of building, CAP	25000. KJ/°C
Average room temperature, $\bar{T}_R$	20. °C
Room temperature deadband (Figure 6), DB	1. °C

Auxiliary Furnace

Capacity, Q_{max}	200,000. KJ/hr

Fan

Mass flow rate (flow rate for all modes assumed to be the same), $\dot{m}$. Most simulations with O-I collectors	1587. Kg/hr
Simulation w/ O-I collectors, $\dot{m}$ = 8.18 Kg/hr per tube pair and 260 tube pairs	1984. Kg/hr
Simulation w/ O-I collectors, $\dot{m}$ = 8.18 Kg/hr per tube pair and 156 tube pairs	1190. Kg/hr
All simulations with flat plate collectors	1632. Kg/hr

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