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ECONOMIC EVALUATION OF THE ANNUAL CYCLE ENERGY SYSTEM (ACES)

FINAL REPORT

Volume II — Detailed Results

ENERGY RESOURCES CENTER
HONEYWELL INC.
2600 RIDGWAY PARKWAY
MINNEAPOLIS, MINNESOTA 55413

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ENERGY RESOURCES CENTER
HONEYWELL INC.
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SECTION 1.0

INTRODUCTION

1.1 CONCEPT

At its most basic level, the Annual Cycle Energy System (ACES) concept consists of extracting heat from a storage tank by a unidirectional heat pump. As the heat is extracted during the heating season, or at other times to provide domestic hot water, ice is formed within the tank. This ice is then used for air conditioning during the summer. Thus the water's heat of fusion is available as a heat source in the winter and a heat sink in the summer. It is this interseasonal transfer of energy that allows both the heating and cooling outputs of the heat pump to be used to satisfy the building loads. The result of this unique feature of the ACES is that the annual coefficient of performance (COP) may be considerably higher than that of a conventional system.

In northern areas of the United States, where heating loads predominate, storage of all of the ice produced during the heating season would result in excess ice at the end of the summer, which would have to be removed. In these areas, it is desirable to store only the volume of ice needed for summer cooling. To prevent the accumulation of excess ice during the heating season, a collector/convector panel is used to supply solar and convective energy as needed and as available to melt ice.

In climates where cooling loads predominate, the ACES configuration utilizes a bin sized to store all of the ice generated during the heating season. This amount of ice will not be sufficient to meet all of the summer cooling needs. Therefore, after the stored ice has been depleted, the ACES heat pump is operated at night to generate ice to meet the cooling requirements of the next 24-hour period. The heat extracted from the water by heat pump operation is rejected to the environment by means of an air-cooled fan coil. The

advantages of nighttime operation of the compressor are that the dissipation of waste heat is more efficient at lower nighttime temperatures and that off-peak electric power rates may be utilized in those areas where they are in effect.

For capital cost or space limitation reasons, it may not be feasible to provide a bin large enough to store energy for an annual cycle. In such cases, a smaller bin is provided and a greater dependence is placed on solar assist in winter and nighttime compressor operation for cooling in the summer.

Another option available for reducing initial capital costs is to incorporate a dual source heat pump into the ACES design. With the dual source system energy can be extracted not only from the ice storage bin, but also from the ambient air. The optimum choice of when to use the air source evaporator or water source evaporator will result in the selection of the mode that yields the most cost-effective system operation.

Although the goal of the ACES is to maximize interseasonal energy transfer and hence energy savings, it also offers a unique potential for load management. Because of its configuration and storage capability the ACES is ideally suited to take advantage of off-peak rates, time-of-day pricing, and other rate structures proposed by the utilities for effective load management.

1.2 SYSTEM DEFINITION

For the purpose of this study, three ACES configurations are defined. All of them will incorporate an electrically-driven unidirectional heat pump and a water-ice storage tank. The ice-maker plates of the heat pump will be located above the ice storage tank, and flake or sheet ice upon harvest will fall by gravity into the storage bin.

The ACES configurations analyzed in this study are defined as follows:

- Full ACES -- This configuration provides the heating, cooling, and domestic hot water demands of the building while minimizing total energy consumption. The water-ice storage volume is selected to maximize interseasonal energy transfer. The Full ACES utilizes an electrically driven unidirectional heat pump, a water source evaporator (ice maker), water-ice storage tank, provision for producing and storing hot water, and solar panels and outdoor air coils as needed.

- Minimum ACES -- This configuration satisfies the heating, cooling, and domestic hot water demands of the building under the restriction of minimum water-ice storage capability, which is defined as storage sufficient to provide heating for 12 consecutive sunless days during the coldest month. The components of the system are the same as for the Full ACES.

- Cost-Optimized ACES -- This configuration provides the heating, cooling, and domestic hot water for the building at minimum total cost over the life of the system. It may contain all of the components of the Full ACES, and may, in addition, incorporate an air source evaporator that is automatically used when it is more cost-effective for the system to operate as an air-to-air heat pump than as a water-to-air heat pump. The use of stored sensible heat as well as heat of fusion will be considered as an option for reducing life cycle costs.

The ACES concept involves the interseasonal transfer of energy. This concept is arbitrarily defined to include all ACES designs ranging from the Full ACES to the Minimum ACES. Since no constraints will be imposed on the search for the Cost Optimized ACES, it is possible to find a configuration of components that no longer exhibits interseasonal energy transfer. Should this situation

exist, the resulting system cannot be classified as an ACES according to the above definition. Therefore, the following designation is made throughout this report: the cost optimized configuration of components that meets the requirements of minimum interseasonal energy transfer is termed a Cost Optimized ACES. The cost optimized configuration of components that does not meet the requirements of minimum interseasonal energy transfer is termed a cost optimized system.*

1.3 SCOPE

The objective of this study is to determine the energy effectiveness and the economic viability of the ACES concept. To perform this task proficiently, the ACES must be studied in a variety of different applications and compared to a number of conventional systems. The different applications to be studied can be catagorized in two groups: the class of building into which the ACES is incorporated, and the climatic region in which the ACES is located. To cover a reasonable range of possible applications, three different classes of buildings are investigated. They include:

- Single-family residence
- Multi-family residence
- Commercial office building

The application of ACES to each of these building types is studied in three different climatic regions. These regions were chosen to represent the major climatic conditions that would be encountered in the continental United States. The regions include:

- Minneapolis, Minnesota
- Atlanta, Georgia
- Philadelphia, Pennsylvania

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*Bin volumes smaller than those of the Minimum ACES are not investigated in the search for the cost optimized configuration of components. The reason why thses volumes are not investigated is that by definition they cannot meet the minimum requirements of interseasonal energy transfer.

For each building type in each geographic location, the economic evaluation of the ACES is based on a comparison of the present worth of the ACES to the present worth of a number of conventional systems. The conventional systems chosen represent four major systems that are currently in extensive use. They include:

- Electric resistance heating, electric air conditioning, and electric domestic water heating.
- Air-to-air heat pump and electric domestic water heating.
- Oil-fired furnace, electric air conditioning, and electric domestic water heating.
- Gas-fired furnace, electric air conditioning, and gas domestic water heating.

Both the single-family and multi-family residential ACES are compared with all four conventional systems in the three geographic regions.

Due to the extremely small number of commercial air-to-air heat pump systems now in operation, the air-to-air heat pump is not included in the list of conventional commercial systems. Therefore, the commercial ACES is compared to the remaining three conventional systems in the three geographic regions.

SECTION 2.0
COMPUTER SIMULATION PROGRAM FOR RESIDENTIAL BUILDINGS

2.1 GENERAL DESCRIPTION

The computer program used to simulate and evaluate residential ACES and conventional heating and cooling systems, ACESIM, is comprised of four distinct modules. The modules are:

- Loads Module
- Design Module
- Simulation Module
- Economic Module

The first module is used to calculate the time dependent thermal loads imposed on the user-specified structure. If these loads are already known or have been previously determined, the loads module can be bypassed.

The second module reads the required load information from a permanent record file or directly from the load module and then determines the sizes of the various ACES components. If the ACES components have already been correctly sized or if the user wishes to evaluate "off-sized" components, this phase can be bypassed.

The third module reads information generated by the first two modules and then simulates either one of the ACES systems or one of the four conventional systems. The output of this phase is the time dependent history of all gas, oil and electric consumption of the system being simulated.

The final module reads the energy use history of the system being simulated and all capital costs associated with equipment, as well as projected escalation rates, discount rates, and time dependent utility rates. The final outputs of the program are: 1) the present worth of all cash outlays for the system discounted over an appropriate system lifetime and 2) the payback of incremental ACES cost compared to the conventional systems.

2.2 SYSTEM DESCRIPTION

A schematic representation of a typical ACES system is shown in Figure 2-1. The system components include: the collectors that can be either absorbers or a glazed cover system; the tank that can be located in the basement, below ground or outside; and the heat pump that has waterside evaporator plates and an airside condenser. The airside condenser has the capability to reject heat to either the house or to the ambient. The heat pump may have an airside evaporator that can be used to complement heat pump operations when conditions permit. In addition to these major components, there are also fans, pumps, and controls associated with efficient system operation.

2.2.1 Controls

2.2.1.1 Conditioning System--Control of the space conditioning system is accomplished by the following logic. The thermostat of the house is set to the desired setpoint. If the space temperature falls below the minimum setpoint the heat pump is turned on. If the outdoor temperature is above a critical value (determined by the simulation) then the airside evaporator will be activated (if the heat pump utilizes this mode), otherwise the water source evaporator plates will be activated. With the heat pump on, energy is transferred from the source to the freon in the evaporator. This energy transfer creates ice or causes the source temperature to be lowered. More energy

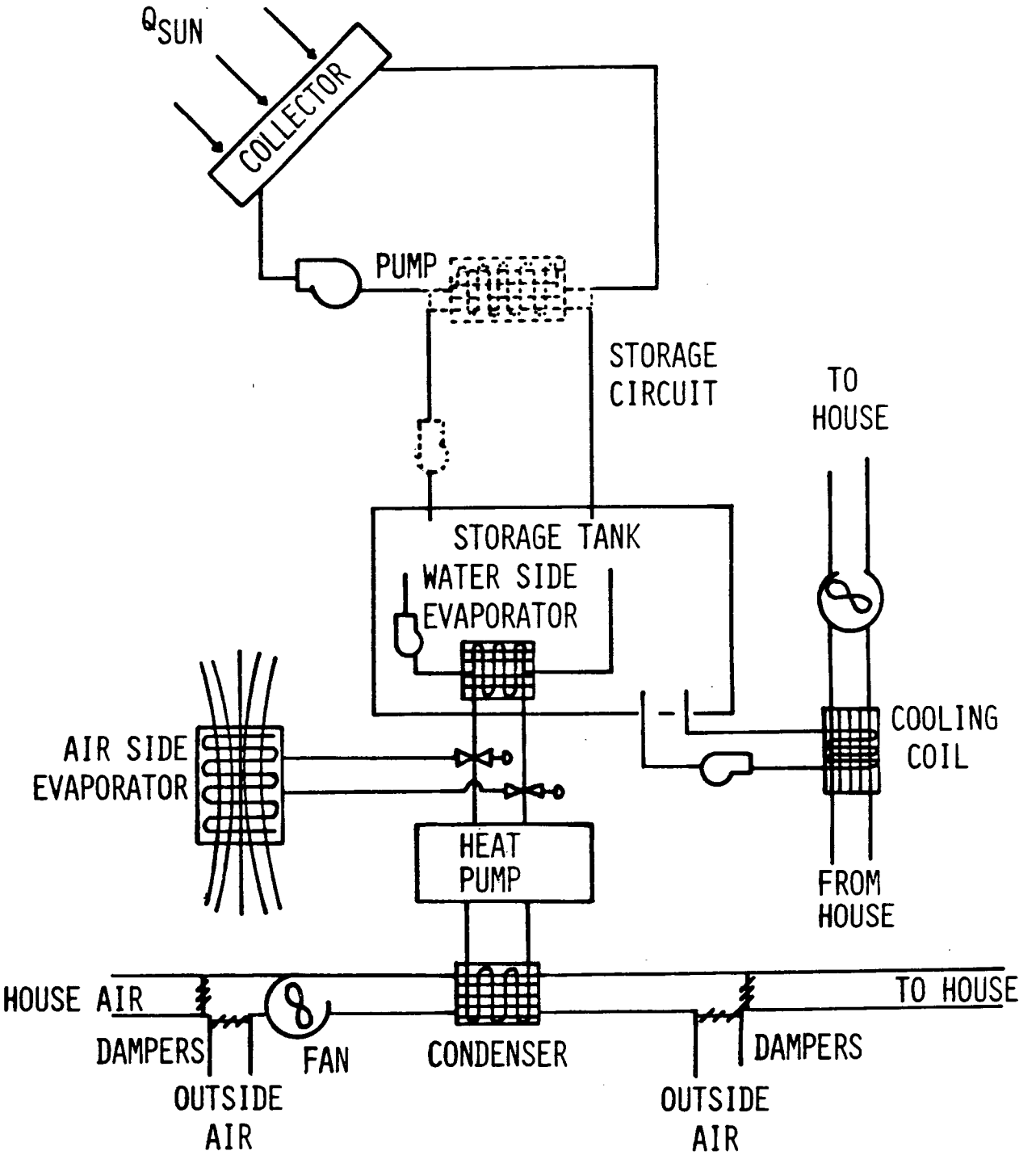


Figure 2-1. Typical Residential ACES

is then added to the freon in the compressor of the heat pump. The freon leaving the compressor is used to heat domestic water as well as the house itself. If there is no need for space heating but the domestic hot water temperature is below a prescribed value, then the heat pump will be turned on and the entire condenser will be used to reject heat to the domestic hot water system.

If the space temperature rises above the maximum setpoint the cooling coil will be activated. As warm air is passed over the cooling coil, heat is transferred from the air to the fluid flowstream. This heat transfer causes the air temperature to decrease and if conditions are correct will cause condensation of some of the water vapor in the air flowstream. The energy transferred from the warm airstream is rejected to the tank thereby increasing its temperature or decreasing the ice inventory.

If there is a need for cooling and the tank temperature is above a predetermined limit (value depends upon the design of the cooling coil and the design cooling load) the heat pump will run during the night and reject the discharge heat to the ambient, thereby lowering the tank temperature.

2.2.1.2 Panels--Control of the panels is accomplished by monitoring the difference between the storage temperature and panel temperature. When the collector temperature is five degrees above the storage temperature, the flow field is opened. As the fluid runs through the collector, heat is transferred from the panels to the water, thereby increasing its temperature. The warmer fluid is fed back to the tank where the energy is stored by increasing the tank temperature or by decreasing the amount of ice stored within the tank.

Although this is the basic mode of panel control, there are some periods when this mode is overriden. The various modes of panel operation are:

- During the fall when a cooling load still exists, the panels will be

turned off when the tank temperature exceeds the maximum allowable inlet temperature to the cooling coil.

- Once there is no longer a need for cooling, the panels will operate in the basic mode collecting energy whenever possible, thus storing sensible heat or decreasing the ice inventory.
- The panels will operate in the modulating mode when it is time to allow the tank to accummulate maximum ice for summer operation.
- During the cooling season, the panels are shut off.*

2.3 MATHEMATICAL MODEL

Any complex system can be described by subdividing that system into the constituent elements. Once these elements have been defined, the mathematical relationships governing the response and interaction of these elements can be derived. The following section gives a brief description of the analysis and methodology of solution of the basic elements of the ACES and conventional systems.

2.3.1 Energy Flow

To predict system response, the dynamic behavior of each state variable must be modeled. This modeling can be accomplished by describing the system as a set of subsystems and performing a control volume energy analysis of each of the subsystems.

The general expression for energy interaction within a system is:

$$\frac{\partial E_{sys}}{\partial \theta} = -\nabla \cdot \rho \nu (U + 1/2\ V^2) \qquad -\nabla \cdot q$$

Rate of energy gain	Rate of internal and kinetic energy input	Rate of heat energy output

*Unless it is feasible to have an oversized tank and perform summer meltdown.

$$+ \rho \, (\nu \cdot q) \qquad\qquad - \nabla \cdot p\nu$$

Rate of work done by gravitational forces

Rate of work done by pressure force

$$- \nabla \cdot [\tau \cdot \nu]$$

Rate of work done by viscous forces

(2-1)

while the modified form of this expression for a control volume is:

$$\frac{d\,E_{cv}}{d\,\theta} = Q_{NET} - W_{NET} + \Sigma(U + \frac{1}{2}\,\frac{V^2}{g_c} + P\,\nu + Z\,\frac{g}{g_c})\,\frac{dm}{dt}_{in}$$

Rate of energy change of control volume | Net heat input to control volume | Net work done by control volume | Net energy associated with mass entering control volume

$$- \Sigma\,(U + \frac{1}{2}\,\frac{V^2}{g_c} + P\,\nu + Z\,\frac{g}{g_c})\,\frac{dm}{dt}_{out} \qquad (2\text{-}2)$$

Net energy associated with mass leaving control volume

If potential and kinetic energies are neglected, equation (2-2) can be written as:

$$\frac{dE_{cv}}{d\theta} = Q_{NET} - W_{NET} + \Sigma\,\dot{m}_i h_i - \Sigma\,\dot{m}_o h_o \qquad (2\text{-}3)$$

which is the expression used in this study to predict the response of all global state variables such as the space temperature, tank temperature, and furnace temperature.

Heat transmission in this model is based on the conduction equation:

$$\frac{\partial^2 T}{\partial X^2} = \frac{1}{\alpha} \frac{\partial T}{\partial \theta} \tag{2-4}$$

where:

T = Temperature

X = Distance along the heat flow path

α = Thermal diffusivity = $K/c\rho$

θ = Time

Subject to the boundary conditions:

$$\theta \geq 0 \qquad x = 0;\ K \left.\frac{\partial T}{\partial x}\right|_{x=0} = h\,(T_{x=0} - T_\alpha) = f\,(\theta)$$

$$x = L;\ K \left.\frac{\partial T}{\partial x}\right|_{x=L} = h\,(T_{x=L} - T_\alpha) + r\,q'' = f\,(\theta)$$

where:

L = Total distance along heat flow path

K = Thermal conductivity

h = Linearized convective radiative transmission coefficient

r = Absorptivity of surface

q" = Incident radiant energy

Due to the nonlinear time dependent boundary conditions encountered in actual practice, a closed form solution of equation (2-4) is impractical. Therefore, this expression is represented by a numerical approximation that can be solved explicitly for nonlinear boundary conditions. The numerical method used in this model is based on evaluating the first and second partial derivations in equation (2-4) by a finite difference approximation. The approximation can be derived by a Taylor series expansion, hence:

$$U_{i-1,\,J} = U_{i,\,J} - \Delta X\, U_x + \frac{(\Delta X)^2}{2\,!}\, U_{xx} - \frac{(\Delta X)^3}{3\,!}\, U_{xxx} + \frac{(\Delta X)^4}{4\,!}\, U_{xxxx} \ldots \tag{2-5}$$

$$U_{i+1,\,J} = U_{i,\,J} + \Delta X U_x + \frac{(\Delta X)^2}{2\,!}\, U_{xx} + \frac{(\Delta X)^3}{3\,!}\, U_{xxx} + \frac{(\Delta X)^4}{4\,!}\, U_{xxxx} \ldots \tag{2-6}$$

where:

$$U_x = \frac{\partial U}{\partial X^1} \quad U_{xx} = \frac{\partial^2 U}{\partial X^2} \quad \ldots$$

If the higher order terms in equation (2-6) are neglected, the expression for the first partial derivation would be:

$$U_{i+1,\,J} = U_{i,\,J} + \Delta X \frac{\partial U}{\partial X} \tag{2-7}$$

or

$$\frac{\partial U}{\partial X} = \frac{U_{i+1,\,J} - U_{i,\,J}}{\Delta X} \tag{2-8}$$

Adding equations (2-5) and (2-6) and neglecting terms greater than second order gives the expression for the second partial derivative as:

$$U_{i-1,\,J} + U_{i+1,\,J} = 2\,U_{i,\,J} + 2\,\frac{\Delta X^2}{2\,!}\,\frac{\partial^2 U}{\partial X^2} \qquad (2\text{-}9)$$

or

$$\frac{\partial^2 U}{\partial X^2} = \frac{U_{i-1,\,J} + U_{i+1,\,J} - 2\,U_{i,\,J}}{\Delta X^2} \qquad (2\text{-}10)$$

Inserting equations (2-8) and (2-10), with the correct time and space subscripts, into equation (2-4) and substituting the time, temperature and space variables where required in the resulting equation gives the approximation to the conduction equation on a node basis as:

$$\frac{\alpha}{\Delta X^2}\,(T_{J-1,i} - 2\,T_{J,i} + T_{J+1,i}) = (T_{J,i+1} - T_{J,i})\,\frac{1}{\Delta\theta} \qquad (2\text{-}11)$$

which upon rearranging gives:

$$T_{J,\,i+1} = T_{J,\,i} + \frac{\alpha\Delta\theta}{\Delta X^2}\,(T_{J-1,i} - 2\,T_{J,i} + T_{J+1,i}) \qquad (2\text{-}12)$$

with

$$q = K\,\frac{\partial T}{\partial X}$$

or

$$q = K\,\frac{(T_{J+1,\,i+1} - T_{J,\,i+1})}{\Delta X} \qquad (2\text{-}13)$$

where:

J = Node of interest

i = Time at which quantity is evaluated

K = Thermal conductivity

q = Heat flow per unit area

The form of equation (2-12) will vary at interfaces between heat flow paths and at all boundaries. The required expression(s) can be derived from equation (2-3) when the work and flow terms are neglected. For example, the temperature of the node on one side of the interface of two different materials would be:

$$\frac{dE_{cv_2}}{d\theta} = \frac{K_1 A}{\Delta X} (T_1 - T_2) - \frac{K_2 A}{\Delta X} (T_2 - T_3) \qquad (2\text{-}14)$$

$$C \rho A \Delta X \frac{dT_2}{d\theta} = \frac{K_1 A}{\Delta X} T_1 + \frac{K_2 A}{\Delta X} T_3 - T_2 \left(\frac{K_1 A}{\Delta X} + \frac{K_2 A}{\Delta X} \right)$$

$$T_{2,\ i+1} = T_{2,\ i} + \frac{\Delta\theta}{C\rho\Delta X^2} [K_1 T_{1,\ i} - T_{2,i} (K_1 + K_2) + K_2 T_{3,i}] \qquad (2\text{-}15)$$

A similar expression can be derived for the temperature of a node at a convective and/or radiative boundary by replacing the appropriate terms in equation (2-14) with the required boundary expression.

Derivations obtained from equation (2-3) and equation (2-8) are equally valid for predicting directional heat flow. This can be shown by replacing the K_2 term in equation (2-14) by K_1. The resultant expression is:

$$\frac{E_{cv_2}}{d\theta} = \frac{K_1 A}{\Delta X} (T_1 - T_2) - \frac{K_1 A}{\Delta X} (T_2 - T_3)$$

$$C \rho A \Delta X \frac{dT_2}{d\theta} = \frac{KA}{\Delta X} (T_1 - 2 T_2 + T_3)$$

$$T_{2,\ i+1} = T_{2,\ i} + \frac{\Delta\theta K}{C\rho\Delta X^2} (T_{1,\ i} - 2 T_{2,\ i} + T_{3,\ i}) \qquad (2\text{-}16)$$

where

$$\frac{K}{C\rho} = \alpha$$

or

$$T_{2,\ i+1} = T_{2,\ i} + \frac{\alpha\Delta\theta}{\Delta X^2} (T_{1,\ i} - 2 T_{2,\ i} + T_{3,\ i}) \qquad (2\text{-}17)$$

which is identical to equation (2-12) if node J is equal to two. The number of equations needed to represent the system depends on the number of nodes (size of the grid spacing) and the number of heat flow paths.

Included in the model are the directional heat flows through walls, roofs, ceilings, basement, internal partitions, tank and surrounding ground as well as the time dependent boundary conditions that include radiation impingement on any surface. The equations defining heat flow are coupled to one another and to the global energy equations defining system response. These equations can be condensed to the form:

$$\dot{\underline{X}} = [A]\ \underline{X} + [B]\ \underline{V} \qquad (2\text{-}18)$$

where:

$\underline{X}$ = The system state vector

$\dot{\underline{X}}$ = The time rate of change of the state vector

A = n x n nonlinear time dependent coefficient matrix

$\underline{V}$ = Input or disturbance vector

B = m x n nonlinear time dependent coefficient matrix

n = number of system states

m = number of inputs

The solution of this set of simultaneous coupled differential equations will completely describe the system or any element of the system.

Due to the assumption made in deriving equations (2-8) and (2-10), there are two parameters that can have a profound effect on the solution of the energy flow equations: the step size, $\Delta\ \theta$, and the grid spacing, $\Delta\ X$. The effect that these parameters have on the solution to the describing equations depends largely on the system being studied and the accuracy of the desired results.

Figure 2-2 shows the results of the transient response of the center temperature of the wall subject to the boundary conditions.

$$\left.\frac{\partial (T - T_o)}{\partial X}\right|_{X = 0} = 0$$

$$\left. T - T_o \right|_{x = L} = 0$$

with the exact temperature profile given by:

$$T\ (X, \theta) = T\ (o) + 2\ [T\ (i) - T\ (o)] \sum_{n = o}^{\infty} e^{-[(n + 1/2)\pi\ /L]^2\ \alpha\theta}$$

$$\cdot \frac{(-1)^n}{(n+1/2)\pi} \cdot \cos\ (n + 1/2)\pi\ X/L$$

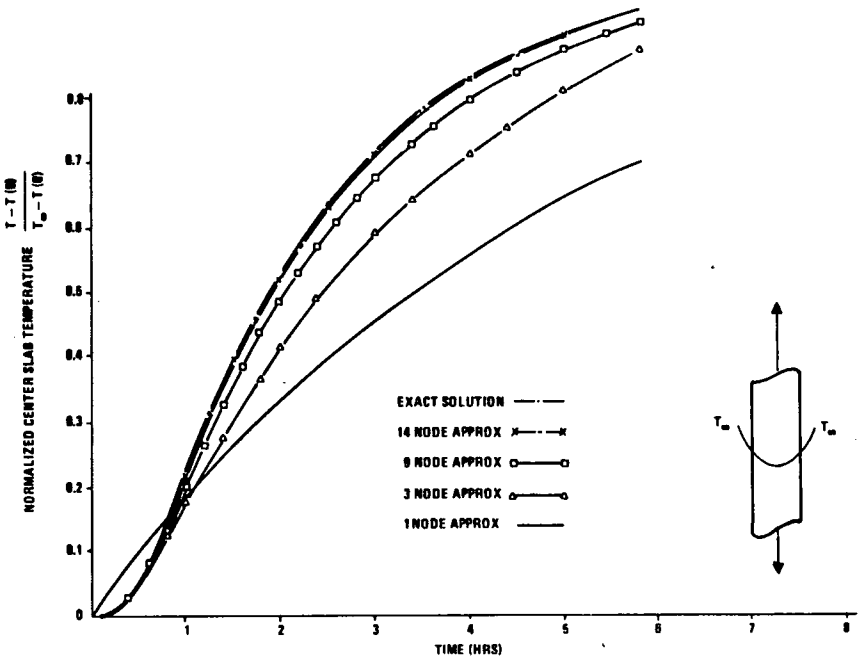


Figure 2-2. The Effect Of Grid Spacing On The Prediction Of The Transient Response of A Wall (Boundary Condition I)

The results show that the finite difference approximation gives essentially the same results as equation (2-19) when the appropriate grid spacing has been chosen. As the grid spacing increases, the accuracy of the approximation decreases. Figures 2-3 and 2-4 show the effect of grid spacing on the solution to the transient response of the same wall with different boundary conditions. These figures show that accuracy does, in fact, depend upon the system of interest (a change in the boundary conditions results in a change in accuracy for a given grid spacing).

Figure 2-5 shows the effect of grid spacing on the air temperature response of a thick walled building. Although this response is heavily influenced by the dynamics of the heating system, the grid spacing has little effect on the resultant accuracy of the air temperature response. Figure 2-6 shows the effect of step size on the air temperature response of the same thick walled building for the stated grid spacing. A large step size results in an unstable

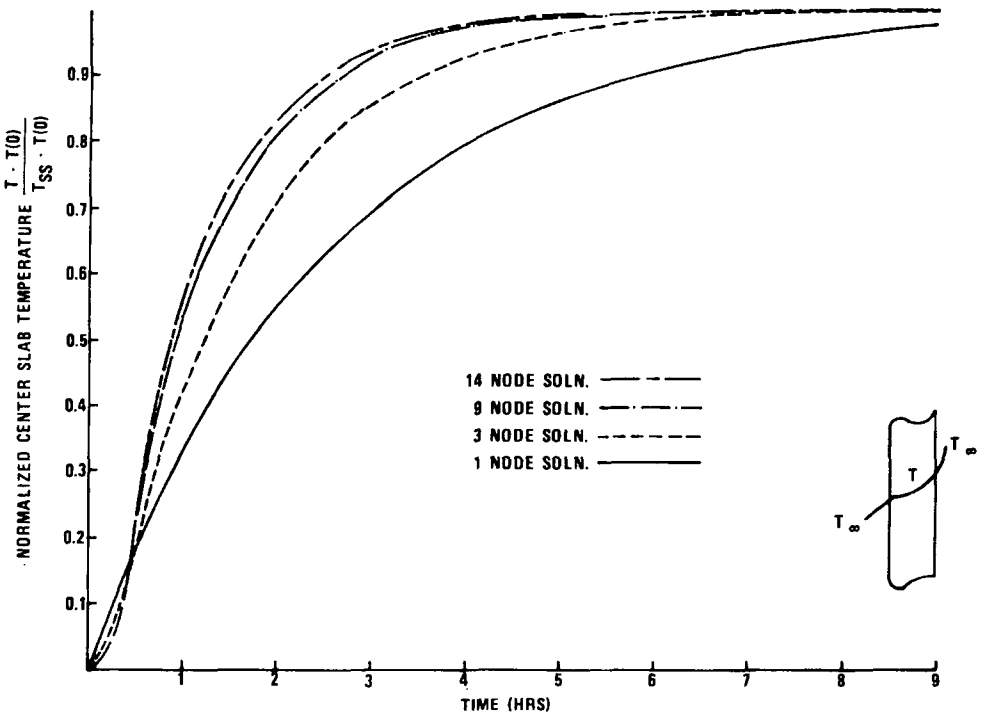


Figure 2-3. The Effect Of Grid Spacing On The Prediction Of The Transient Response of a Wall (Boundary Condition II)

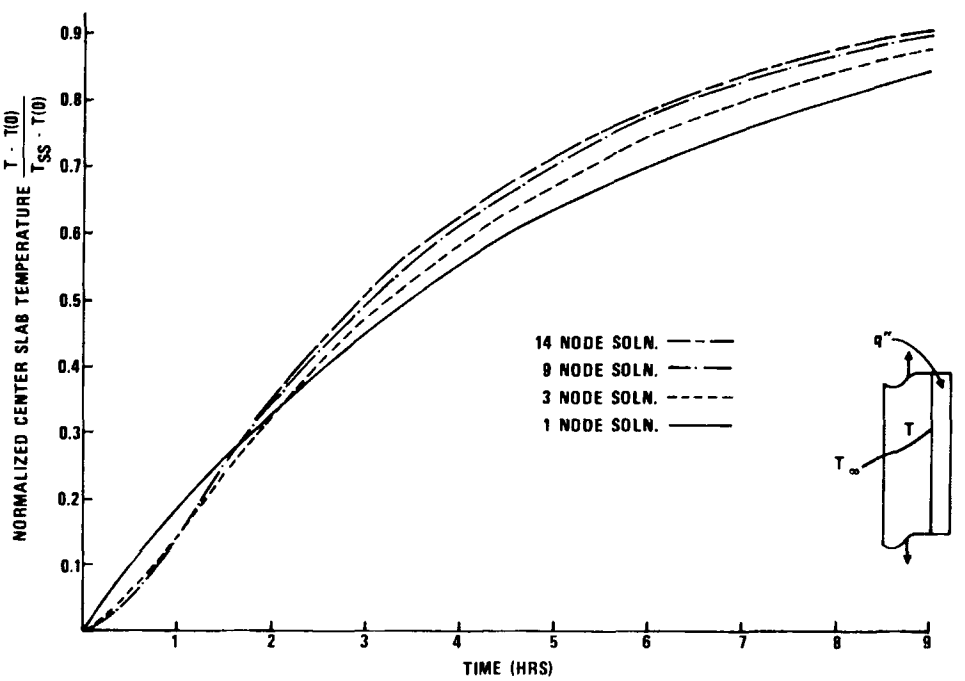


Figure 2-4. The Effect Of Grid Spacing On The Prediction Of The Transient Response Of A Wall (Boundary Condition III)

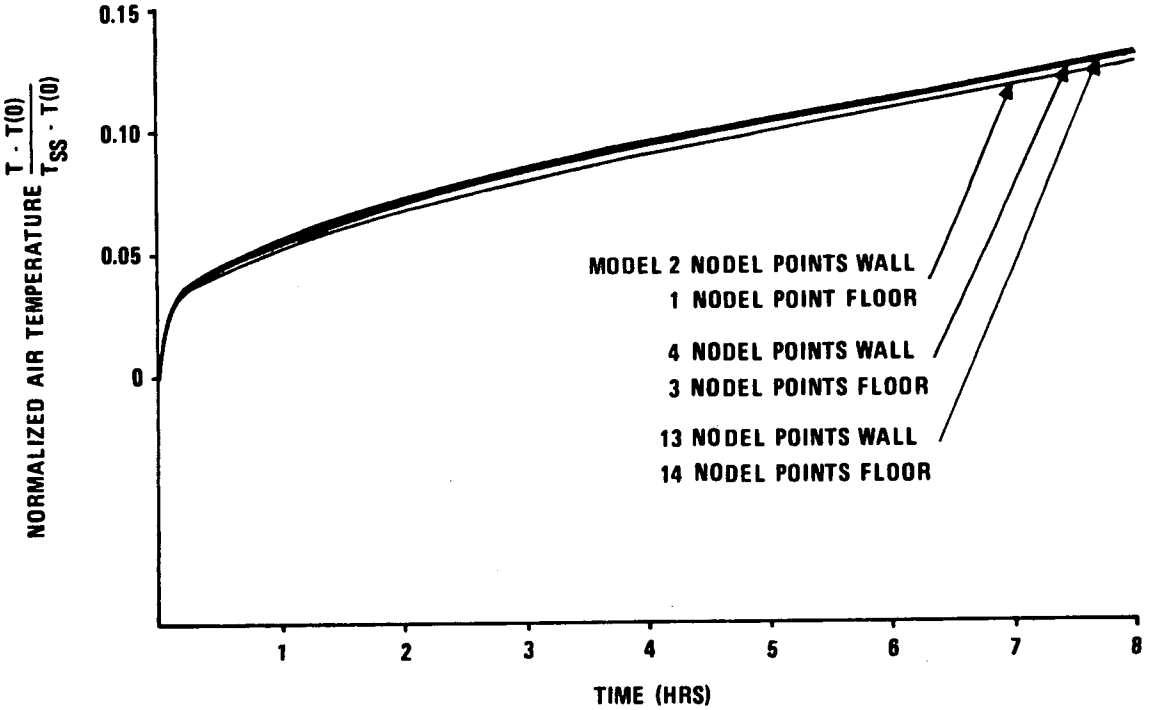


Figure 2-5. Temperature Response Of A Thick Walled Building

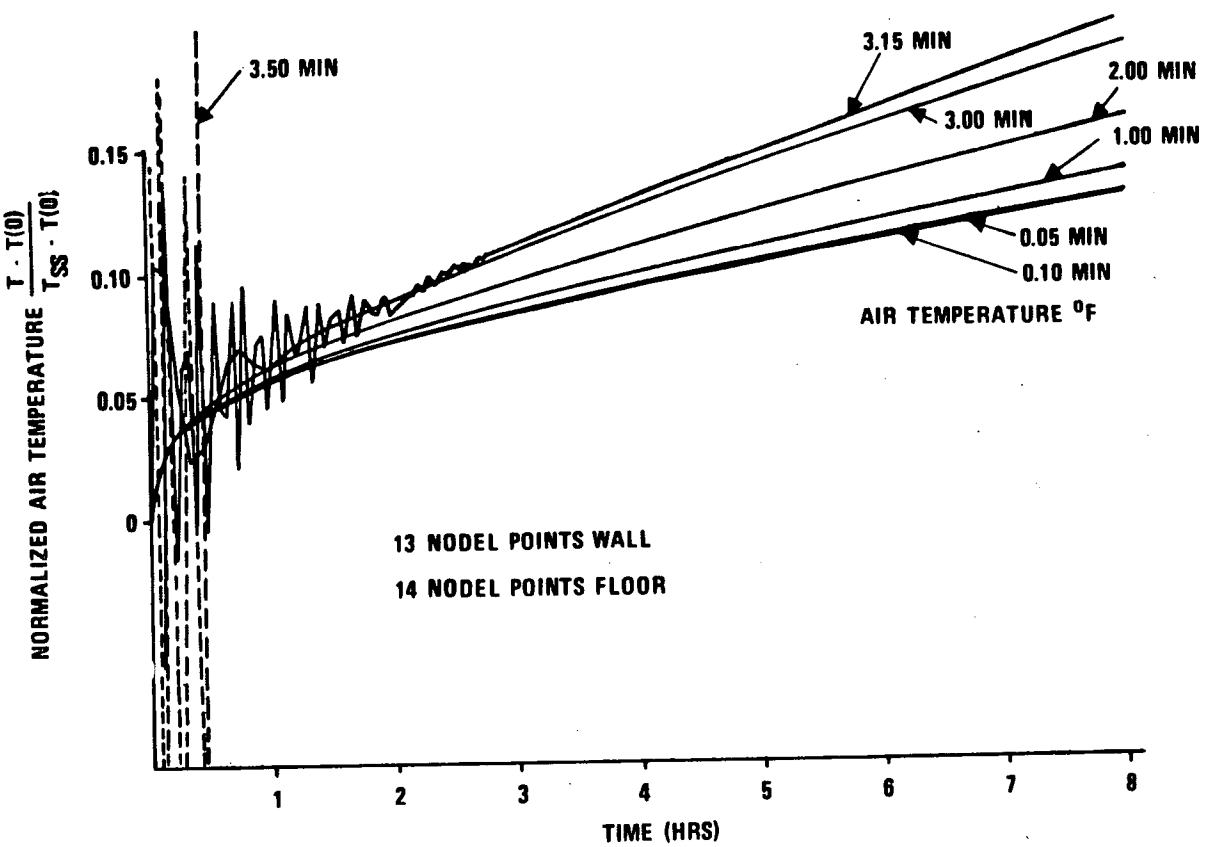


Figure 2-6. Effect Of Step Size On Response Of A Thick Walled Building

solution. As the step size is decreased the solution becomes stable; however, it remains inaccurate. Finally, there exists a step size beyond which there is no change in the accuracy of the air temperature response. The results in Figures 2-2 through 2-6 show that it is imperative to determine the appropriate step size and grid spacing for each system being analyzed.

Figure 2-7 shows the air and wall temperature response of the Minneapolis single-family residence. These curves show the effect of thermal capacitance on system response time and, hence, can be used to show how thermal storage affects the required heating or cooling load. The combination wall and space temperature curve represents the effective temperature that a thermostat would sense and, therefore, can be used to design and evaluate different control strategies. Figure 2-8 shows the differences in response times of some state variables associated with the house. Except for the air temperature curve, all curves represent midplane temperatures.

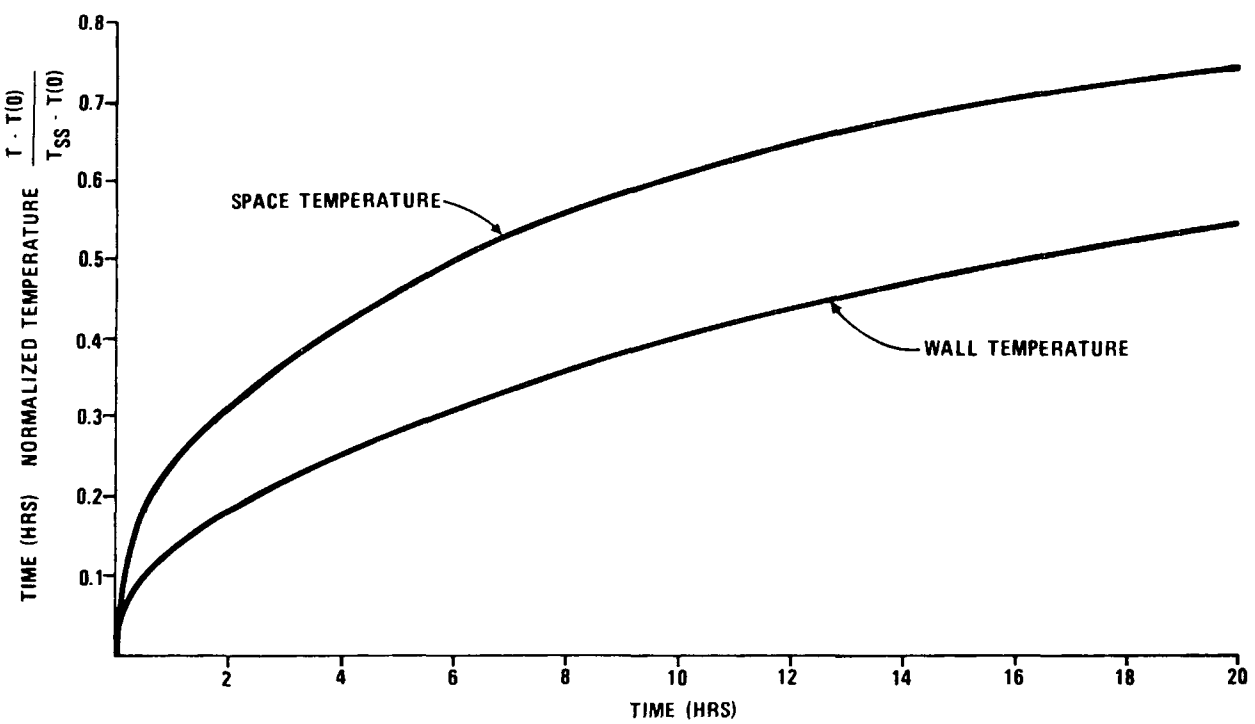


Figure 2-7. Air And Wall Temperature Response Of The Minneapolis Single-Family Residence

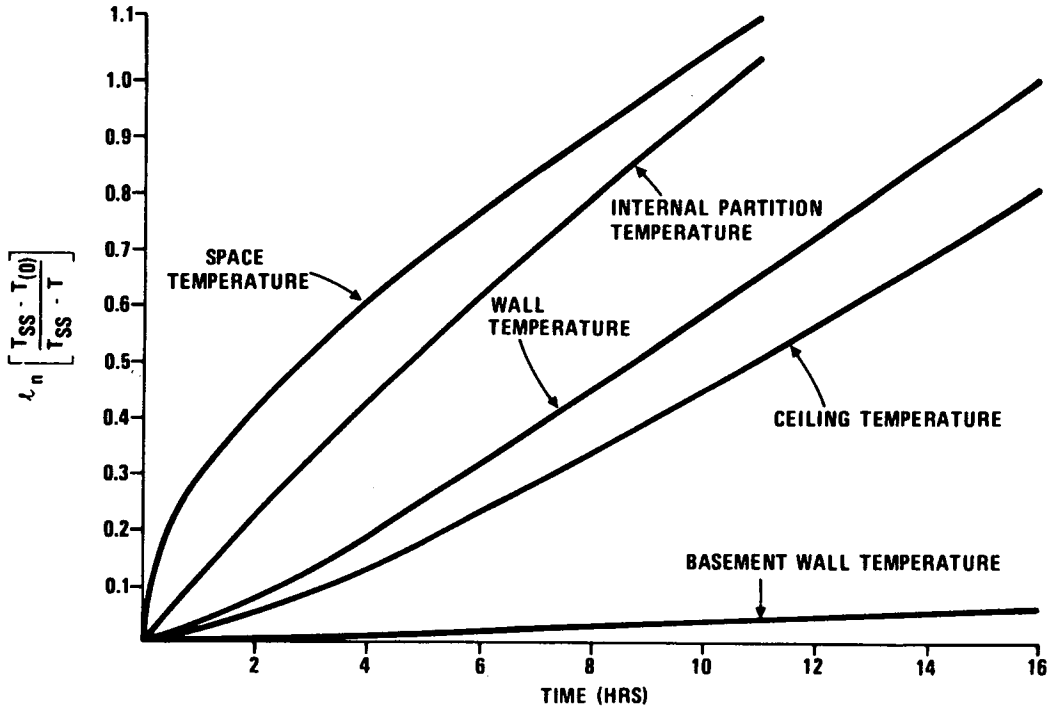


Figure 2-8. Midplane Temperature Response for the Minneapolis Single-Family Residence

2.3.2 Solar Radiation

Radiation impingement on any given surface is a function of the radiation that reaches the earth's surface and the geometrical relationship between the sun and the surface of interest. The geometrical relationships are a function of three basic angles: the hour angle, ω; the declination angle, δ; and the latitude angle, ϕ. The relationship used in this model can be derived by four coordinate transformations, which relate the equatorial reference frame to the solar reference frame.

The equatorial reference frame is shown in Figure 2-9:

where,

X_E = Direction positive to the vernal equinox

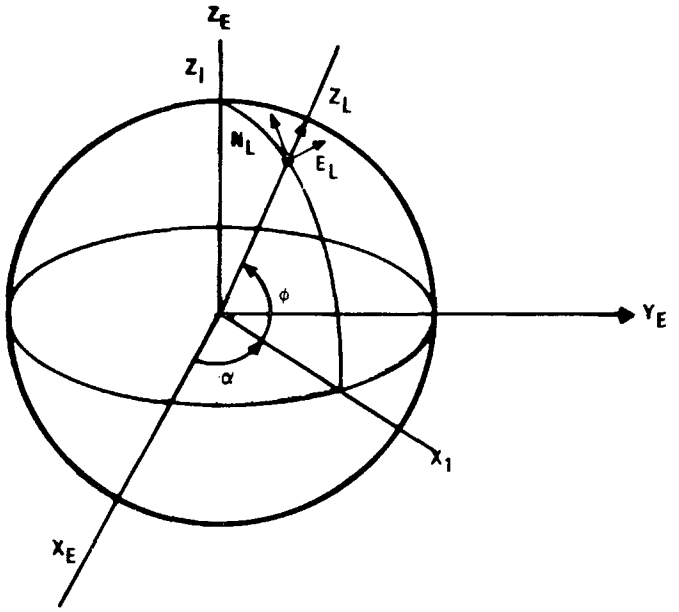


Figure 2-9. Equatorial Reference Frame and Coordinate Transformations α and ϕ

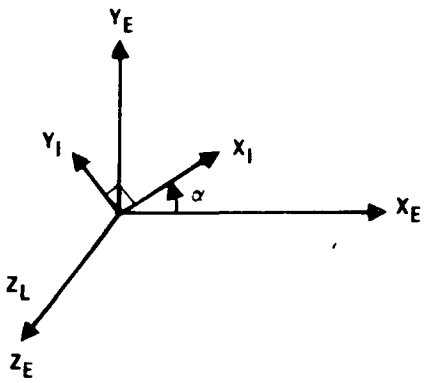


Figure 2-9a. Transformation α

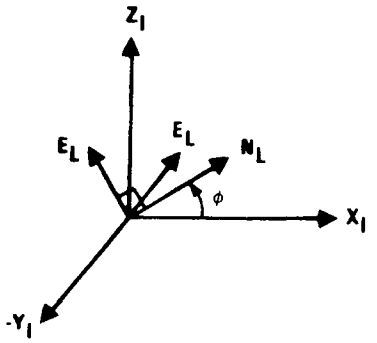


Figure 2-9b. Transformation ϕ

Z_E = Direction positive to the north pole

Y_E = Third coordinate of a right-handed system

X_1, Z_1, Y_1 } Coordinate reference system containing point of interest

Z_L = local normal

E_L = local east

N_L = local north

The local reference frame can be obtained from the equatorial reference frame through two rotations: α and ϕ.

The first rotation, α, shown in Figure 2-9a is:

$$X_1 = X_E \cos\alpha + Y_E \sin\alpha + 0$$

$$Y_1 = -X_E \sin\alpha + Y_E \cos\alpha + 0$$

$$Z_1 = 0 + 0 + Z_E$$

or

$$\begin{bmatrix} X_1 \\ Y_1 \\ Z_1 \end{bmatrix} = [\alpha] \begin{bmatrix} X_E \\ Y_E \\ Z_E \end{bmatrix} \tag{2-20}$$

The second rotation, ϕ, shown in Figure 2-9b is:

$$Z_L = X_1 \cos\phi + 0 + Z_1 \sin\phi$$

$$E_L = 0 + Y + 0$$

$$N_L = X_1 \sin \phi + 0 + Z_1 \cos \phi$$

or

$$\begin{bmatrix} Z_L \\ E_L \\ N_L \end{bmatrix} = [\phi] \begin{bmatrix} X_1 \\ Y_1 \\ Z_1 \end{bmatrix} \tag{2-21}$$

The relationship between the sun and the local reference frame can be seen in Figure 2-10 and can be obtained through two rotations ω, and δ.

The first rotation, ω, shown in Figure 2-10a is:

$$X_1 = X' \cos \omega - Y' \sin \omega + 0$$

$$Y_1 = X' \sin \omega + Y' \cos \omega + 0$$

$$Z_1 = 0 + 0 + Z^1$$

or

$$\begin{bmatrix} X_1 \\ Y_1 \\ Z_1 \end{bmatrix} = [\omega] \begin{bmatrix} X^1 \\ Y^1 \\ Z^1 \end{bmatrix} \tag{2-22}$$

The final rotation, δ, shown in Figure 2-10b is:

$$X^1 = X_s \cos \delta + 0 - Z_s \sin \delta$$

$$Y^1 = 0 + Y_s + 0$$

$$Z^1 = X_s \sin \delta + 0 + Z_s \cos \delta$$

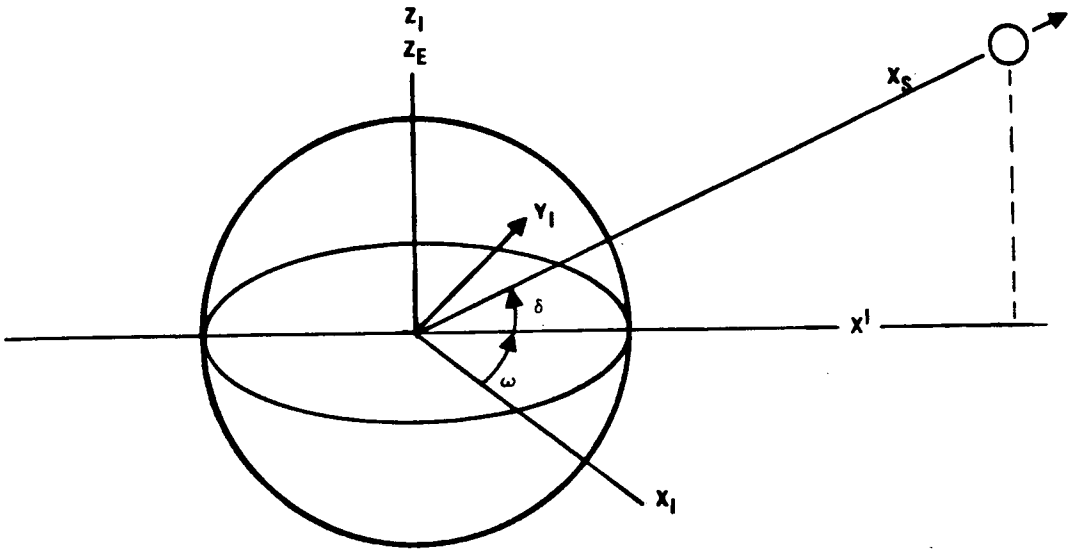


Figure 2-10. Relationship Between the Sun and the Local Reference Frame Coordinate Transformation ω and δ

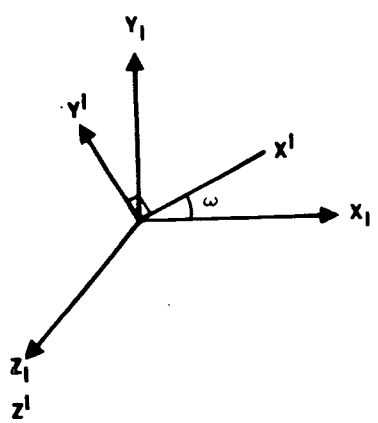


Figure 2-10a. Transformation ω

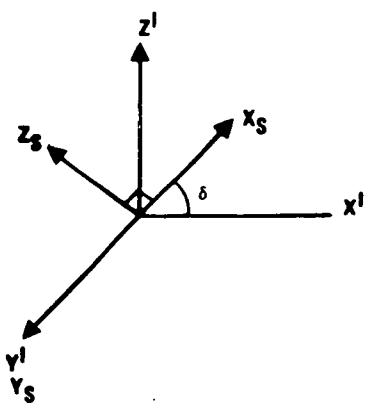


Figure 2-10b. Transformation δ

or

$$\begin{bmatrix} X^1 \\ Y^1 \\ Z^1 \end{bmatrix} = [\delta] \begin{bmatrix} X_s \\ Y_s \\ Z_s \end{bmatrix} \tag{2-23}$$

The unit vector pointing toward the sun from the local reference frame is thus:

$$\begin{bmatrix} Z_L \\ E_L \\ N_L \end{bmatrix} = [\phi] \quad [\omega] \quad [\delta] \quad \begin{matrix} [1] \\ [0] \\ [0] \end{matrix} \tag{2-24}$$

or

$$Z_L = \cos\phi \cos\omega \cos\delta + \sin\phi \sin\delta$$

$$E_L = \sin\omega \cos\delta$$

$$N_L = \cos\phi \sin\delta - \sin\phi \cos\omega \cos\delta$$

These are the basic relationships that are used for all solar calculations. From these relationships, and similar transformations, the solar altitude and azimuth angles may also be calculated. The cosine of the angle of incidence, θ, between the normal to the surface of interest and the sun's rays can be written as:

$$\cos\theta = E_L \sin\psi \sin\gamma + \sin\psi \cos(180 - \gamma) N_L + \cos\psi Z_L$$

or

$$\cos\theta = \sin\psi \sin\gamma \sin\omega \cos\delta + \cos\psi\,(\cos\theta \cos\omega \cos\delta + \sin\phi \sin\delta) - \sin\psi \cos\gamma\,(\cos\phi \sin\delta - \sin\phi \cos\omega \cos\delta) \tag{2-25}$$

ψ = Angle of the surface with respect to ground

γ = Angle of the surface with respect to local south (east positive, west negative)

The radiation intensity on a cloudless day is defined by the expressions:

$$I_{DN} = KA\, e^{-B/\sin\beta} \tag{2-26}$$

$$I_{DH} = \sin\beta\; I_{DN} \tag{2-27}$$

$$I_{ds} = c\; I_{DN}$$

$$I_{dr} = \rho\,(C + \sin\beta)\; I_{DIV} \tag{2-28}$$

where:

I_{DN} = Direct normal radiation

I_{DH} = Direct horizontal

I_{ds} = Diffuse radiation on a horizontal surface

I_{dr} = Diffuse ground reflected radiation

β = Solar altitude angle

ρ = Ground reflectivity

K = Clearness factor

A = Solar irradiation at air mass = 0

B = Atmospheric extinction coefficient

C = Diffuse radiation factor

While the total horizontal radiation is defined by:

$$I_{TH} = I_{DH} + I_{ds}$$

or

$$I_{TH} = (c + \sin \beta)\ KA\ e^{-B/\sin\beta} \tag{2-29}$$

The diffuse and direct radiation components on a cloudy day are given by the expressions:

$$I_{DIF} = [I_{TH}\ CCF - I_{DH}\ (1 - .09CC)]\ [1 - (1 - \cos \psi/2)] + I_{TH}\ \rho\ (1 - \cos \psi/2)\ CCF \tag{2-30}$$

$$I_{DIR} = I_{DN}\ (1 - .09CC)\ \cos \theta \tag{2-31}$$

where:

I_{DIF} = Diffuse radiation cloud modified

I_{DIR} = Direct radiation cloud modified

CC = Cloud cover

CCF = Cloud cover factor

The available energy incident on a surface is therefore:

$$Q_{INC} = I_{DIR} + I_{DIF} \tag{2-32}$$

2.3.3 Heat Pumps

A schematic representation of the heat pump modeled in this study is shown in Figure 2-11. There are a number of modes associated with the operation of this advanced heat pump. The modes are:

- Operation to supply heat to living space only, using the water source evaporator. This mode is accomplished when the compressor runs, Pump 2 (P2) is OFF, Control Valve 1 (CV1) is activated a-b, Coil 2 (C2) is used as a condenser, CV2 is activated a-b, CV4 is activated a-d, P1 is ON, CV5 is activated a-b, and Fan 1 is run in high speed mode.
- Operation to supply heat to living space and domestic hot water using water source evaporator. The operation of this mode is identical to the previous mode except P2 is turned ON.
- Operation to supply domestic hot water only, using the water source evaporator. The operation of this mode is the same as the previous mode except CV1 is activated a-d, and Fan 1 is OFF.
- Operation to supply heat to living space only, using air source evaporator. This mode is accomplished when the compressor runs, Fan 2 is ON, P2 is OFF, CV1 is activated a-b, Fan 2 is ON in the high speed mode, CV2 is activated a-d, CV3 is activated b-c, P1 is OFF, and CV4 is activated b-d.
- Operation to supply heat to living space and domestic hot water using the air source evaporator. Operation of this mode is identical to the previous mode except P2 is ON.

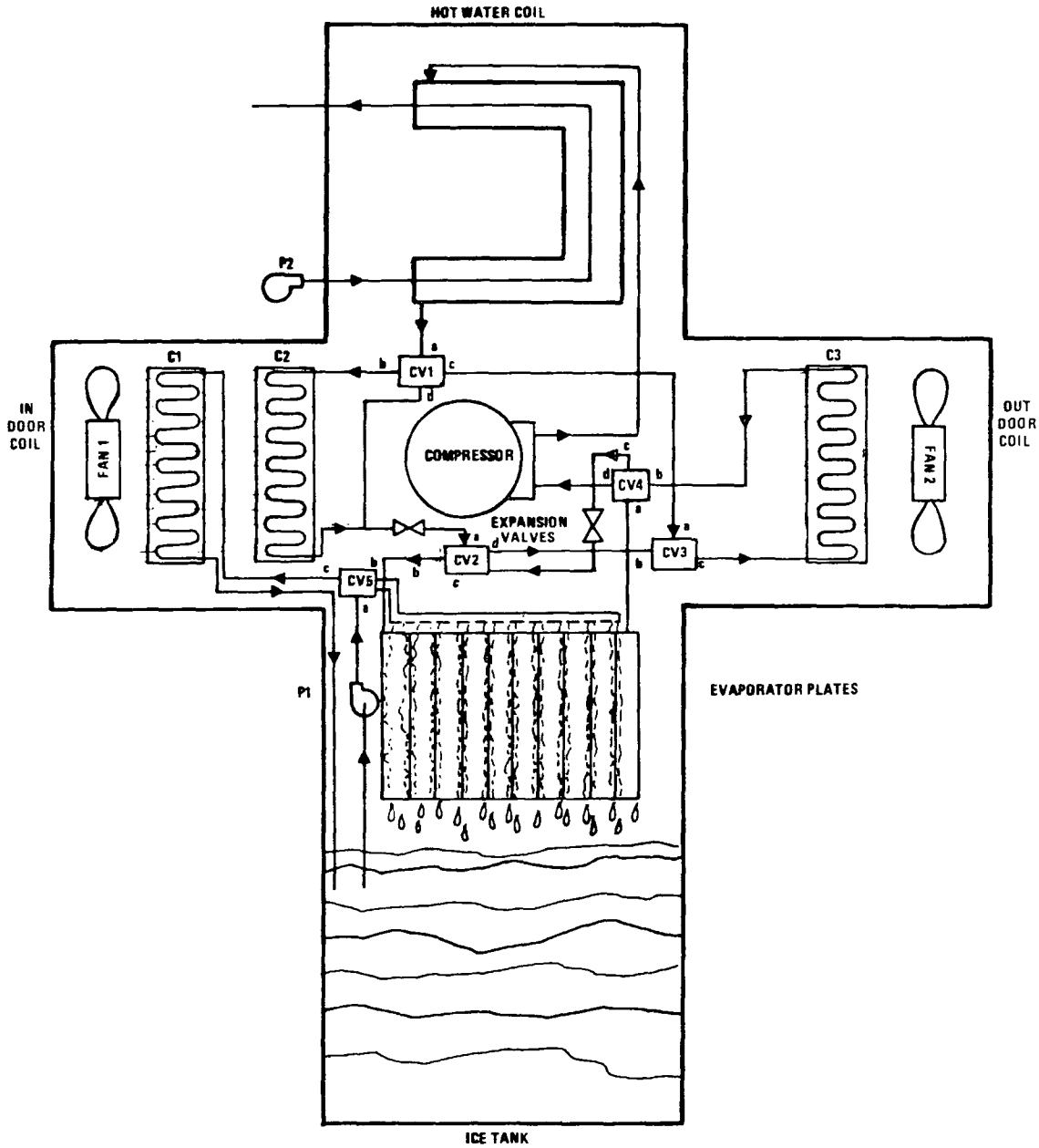


Figure 2-11. Schematic Representation of the Heat Pump

- Operation to reject heat from storage during summer nights. This mode is accomplished when the compressor runs, P2 is OFF, CV1 is activated a-c, Fan 1 is OFF, CV3 is activated a-c, C3 is used as the condenser, CV4 is activated b-c, CV2 is activated c-b, CV4 is activated a-d, P1 runs, CV5 is activated a-b, and Fan 2 is ON.
- Operation to reject heat from storage during summer nights while providing domestic hot water. This operation is similar to the previous mode except P2 is ON.
- Operation to provide cooling. In this mode, the compressor is OFF, P1 is ON, CV5 is activated a-c, and Fan 1 is run in the low speed mode.

These various modes are comprised of three basic heat pump systems: water-to-water, water-to-air, and air-to-air. The steady state performance of the high efficiency heat pump used in this model as a function of inlet conditions and the operational modes can be seen in Figures 2-12 through 2-14. The performance shown in these figures is based on compressor power only. Table 2-1 shows the effect of parasitic power on the performance of these heat pumps.

The heat pump model takes into account dynamic cycling losses (where applicable), energy needed to defrost (where applicable), and blower and pump power. The heat of energy associated with the parasitic power is rejected to a combination of the fluid flowstreams, the surrounding air or the ambient air (outdoor air).

The cycling losses can be determined by relating the instantaneous heat output to the steady state heat output. Assuming the response of the heat pump is first order, then:

$$Q_i(\theta) = Q_{ss}(1 - e^{-\theta/\tau}) \qquad (2\text{-}33)$$

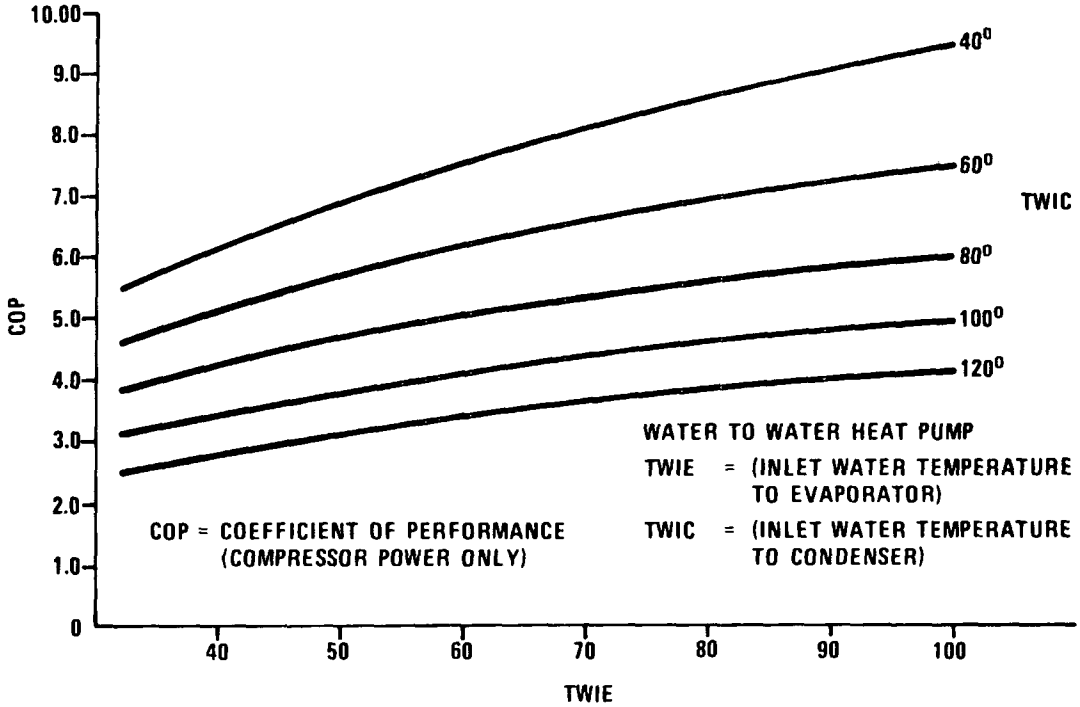


Figure 2-12. Steady State Performance of Water-to-Water Heat Pump as a Function of Inlet Conditions

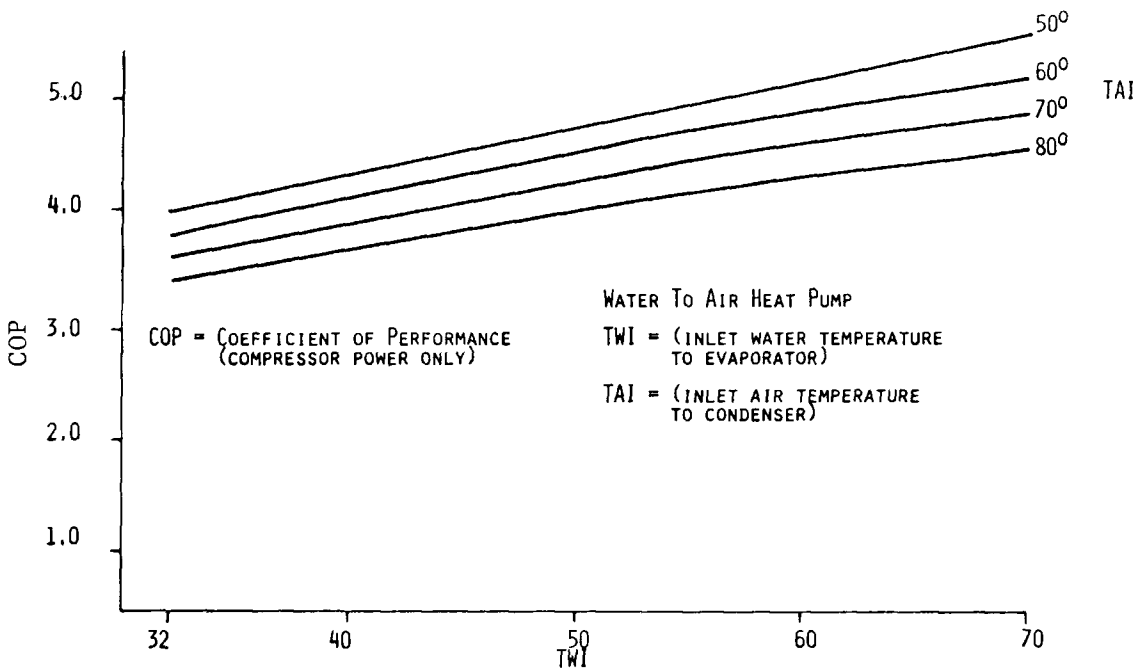


Figure 2-13. Steady State Performance of Water-to-Air (Ice-Maker Mode) Heat Pump As A Function of Inlet Conditions

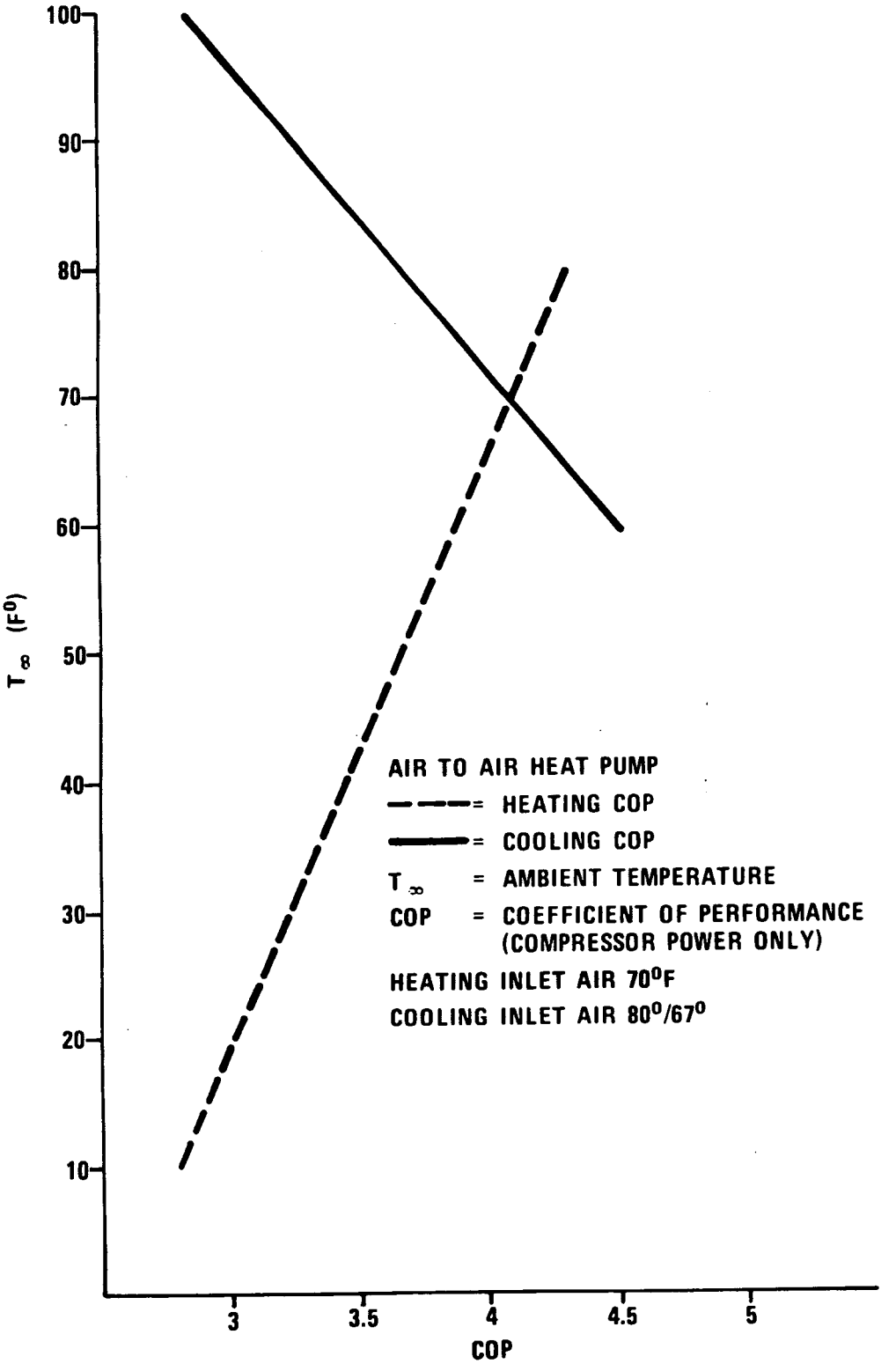


Figure 2-14. Steady State Performance of Air-to-Air Heat Pump as a Function of Inlet Conditions

Table 2-1. Effect of Parasitic Power Consumption on COP of Heat Pump

CONDITION / HEAT PUMP CONSUMPTION	ICE-MAKER HEAT PUMP		AIR-TO-AIR HEAT PUMP (HEATING)		AIR-TO-AIR HEAT PUMP (COOLING)	
	TAI=70° TWI=32°	TAI=70° TWI=45°	TAC=70° TAE=17°	TAC=70° TAE=42°	TAC=70° TAE=95°	TAC=70° TAE=85°
PRIMARY ELECTRIC ENERGY REQUIRED/BTU HEAT REJECTED OR ABSORBED	.281	.248	.336	.276	.328	.289
PARASITIC ENERGY REQUIRED/BTU HEAT REJECTED OR ABSORBED	.086	.064	.190	.112	.115	.109
COP OF HEAT PUMP EXCLUDING PARASITICS	3.55	4.02	2.97	3.61	3.04	3.46
COP OF HEAT PUMP INCLUDING PARASITICS	2.71	3.19	1.89	2.56	2.25	2.51

TAI = entering air temperature to the indoor coil of the ice-maker heat pump.

TWI = entering water temperature to the evaporator coil of the ice-maker heat pump.

TAC = entering air temperature to the indoor coil of the air-to-air heat pump

TAE = entering air temperature to the outdoor coil of the air-to-air heat pump

where:

Q_i = Instantaneous heat output

Q_{ss} = Steady state heat output

τ = Time constant of heat pump

Integrating over the ON period gives:

$$Q = \int_{\theta_o}^{\theta_{on}} Q_i \, d\theta \tag{2-34}$$

where

Q = Heat delivered during a cycle.

Then, substituting (2-33) into (2-34) and integrating gives:

$$Q = Q_{ss} \; \theta_{on} - K \left(1 - e^{-\theta_{on}/\tau}\right) \tag{2-35}$$

Defining the part load factor (PLF) as the amount of heat delivered during the ON period to the amount of heat delivered at steady state conditions during the ON period gives:

$$PLF = \frac{Q}{Q_{ss}} \; \theta_{on} \tag{2-36}$$

Substituting equations (2-35) into (2-36) gives the part load factor as:

$$PLF = 1 - \frac{\tau}{\theta_{on}} \left(1 - e^{-\theta/\tau}\right) \tag{2-37}$$

or for large $\frac{\theta}{\tau}$:

$$PLF = 1 - \frac{\tau}{\theta} \tag{2-38}$$

If the time fraction is defined as the ratio of ON time to cycle time, then

$$T_F = \left(\frac{\theta_{on}}{\theta_{on} + \theta_{off}}\right) \tag{2-39}$$

and, if a degradation coefficient is defined as:

$$C_D = \frac{\tau}{\theta_{on}\,(1-T_F)} \tag{2-40}$$

then the PLF is:

$$PLF = 1 - C_D\,(1 - T_F) \tag{2-41}$$

and the cycle loss is:

$$q_{cc} = \frac{q_{net}}{COP_{ss}} \quad (1/PLF - 1) \tag{2-42}$$

where:

q_{cc} = Cycle loss in BTU

q_{net} = Net thermal load in the space

COP_{ss} = Steady state COP

Defrost is accomplished by reversing the heat pump flow field so the condenser becomes the evaporator and the evaporator becomes the condenser. The model accounts for the energy needed to run the compressor and blowers during the defrost period as well as the energy removed from the space through the evaporator. The number of defrost cycles per hour is a function of the outdoor temperature and the relative humidity, and has been correlated to experimental data.

2.3.4 Cooling Coil

The cooling coil operates as described in the previous section on system control. The coil is modeled by relating heat flows to and from the water and air stream, respectively. Thus:

$$Q_{coil} = \dot{m}_a (h_{1_a} - h_{2_a}) = \dot{m}_w (h_{1_w} - h_{2_w})^* \qquad (2\text{-}43)$$

where:

Q_{coil} = heat transfer through the coil

$\dot{m}$ = mass flow rate

h = enthalpy

subscript w refers to water;

subscript a refers to air

and

$$h_a = C_{p_a} T_a + \omega(.45\ T_a + 1061)$$

$$h_w = C_p T$$

ω = humidity ratio

*The heat storage term in this equation has been neglected since the response of the coil is fast in comparison to the rest of the system.

An expression for the coil capacity can be derived by defining the effectiveness of the coil as the ratio of actual heat transfer to the maximum possible amount of heat transfer (based on the second law of thermodynamics), thus:

$$\varepsilon = \frac{Q_{actual}}{\dot{m}\, C_{p_a} (T_{a_i} - T_{w_i}) + \dot{m}\, C_{p_a} [\omega_{i_D} (1061 + .45\, T_{a_i}) - \omega_{o_{@T_{w_c}}} (1061 + .45 T_{w_i})]} \qquad (2\text{-}45)$$

where:

subscript i refers to inlet water
subscript o refers to outlet state
subscript D refers to design condition

In general:

$$\varepsilon = f(\dot{m}, T, w)$$

For constant flow rate, assume that:

$$\varepsilon \cong K$$

Therefore, the time dependent capacity of the coil is:

$$Q_{CAP}(\theta) = \varepsilon\, \dot{m}_a [(C_{p_a} + .45\, \omega_i(\theta)) T_{a_i}(\theta) + 1061\, \omega_i(\theta)] - [(C_{p_a} + .45\, \omega_{e_{@T_{w_i}}}(\theta)] \cdot T_{w_i}(\theta) + 1061 \omega_{e_{@T_{w_i}}}(\theta) \qquad (2\text{-}45)$$

with

$$\omega_i = \frac{\dot{m}_{INF}\, \omega + Q_L/1096 + \dot{m}_a\, \omega_e}{\dot{m}_{INF} + \dot{m}_a} \qquad (2\text{-}46)$$

and

$$\omega_{e_{@T_{w_i}}} = \omega\, f\,(T_{wi}) \tag{2-47}$$

where:

ε = Constant determined at some specified condition

$\dot{m}_{INF}$ = Infiltration flow rate

ω_∞ = Outdoor humidity ratio

ω_e = Exit humidity ratio

Q_L = Internal latent load

The exist coil enthalpy is thus:

$$h_{a_e} = C_{p_a}\ T_{a_i} + w_i\ (.45\ T_{a_i} + 1061) - Q_{cap}\ (\theta)/\dot{m}_a \tag{2-48}$$

Equations (2-45) through (2-48) are solved by guessing the exit humidity ratio and solving for h_{a_e}. Knowing h_{a_e}, the actual humidity ratio can be determined. A numerical method is used for determining the exact coil exit conditions that satisfy the global energy equations between coil, surroundings (house), and environment (outdoors). With the exit conditions known, the total, sensible, and latent loads, as well as the operating time of the coil, can be determined.

2.3.5 Collectors

The first law of thermodynamics applied to the solar collector gives an expression for energy flow within the system as:

$$A_c\,[HR\,(\tau\,\alpha)_d + HR\,(T\,\alpha)_D = Q_c + Q_L + Q_s \qquad (2\text{-}49)$$

Incident Energy — Collected Energy — Lost Energy — Stored Energy

Since the time constant of the collector is relatively small and cloud cover and radiation data are only available on an hourly basis, the fast transient response of the collector has been neglected. Defining F_r as the actual useful gain of the collector to the useful gain if the whole collector were at the inlet fluid temperature gives:

$$Q_c = A_c F_r\,[HR\,(\tau\,\alpha)_T - U_L\,(T_{in} - T_\alpha)] \qquad (2\text{-}50)$$

Collected energy — Available energy — Lost energy

with

$$\eta_c = \frac{Q_c}{Q_{inc}}$$

therefore:

$$\eta_c = \frac{F_r\,HR(\tau\,\alpha)_T}{Q_{inc}} - \frac{U_L\,F_r\,(T_{in} - T_\alpha)}{Q_{inc}}$$

or

$$\eta_c = K_1 - \frac{K_2\,(T_{in} - T_\infty)}{Q_{inc}}$$

The performance curves of the collectors used in this study can be seen in Figure 2-15.

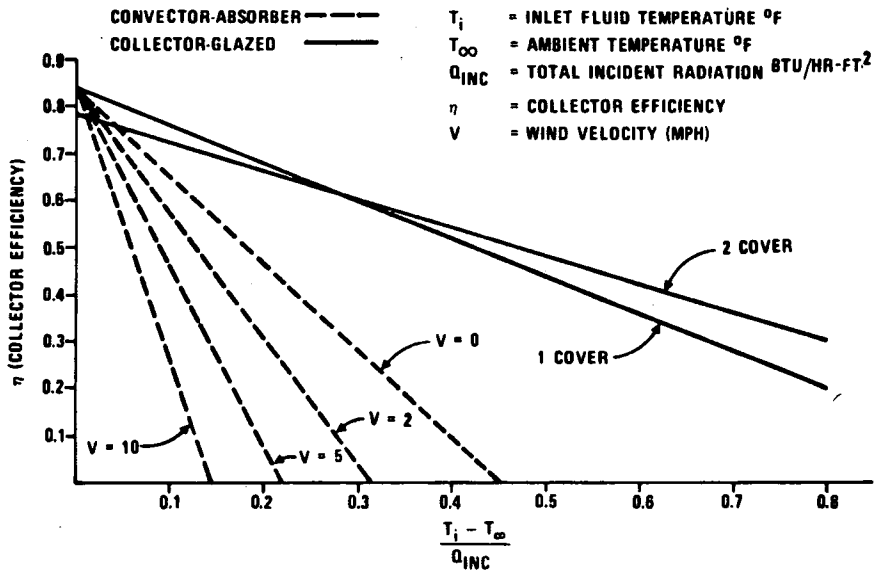


Figure 2-15. Performance Curves of Solar Collector with 0, 1 and 2 Covers

2.3.6 Tank

The tank has been modeled based on the methods presented in the energy flow section of this report. The global energy equations as well as the conduction equations describing tank response allow the user to evaluate heat transfer to or from the tank on a dynamic basis. This dynamic approach also allows the user to evaluate the effect that the tank has on the building loads (if the tank is within the conditioned space) and the ground temperature profile. The tank may be located in the basement, outside, or entirely below ground. If the tank is in the basement, any portion of the tank can be below ground level. The heat flow to or from a tank located in the basement will interact with conditional air, the ground, the tank walls and basement walls. If the tank is outside, the heat flow will interact with the ambient, the tank walls and the ground. If the tank is entirely below ground, the heat flow will interact with the tank walls and the ground. The user-specified time dependent ground temperature is used as a boundary condition for the temperature profile of the ground surrounding the tank.

2.3.7 Thermostat

The function of the thermostat is to maintain the room temperature at a desired level by controlling the amount of heat delivered to or removed from the space. The mathematical model of this element can be derived by performing an energy balance on each constituent element. Thus, the sensor temperature may be written as:

$$mc_p \frac{dT_{sen}}{d\theta} = ha\,[(1-K)T_a + K\,T_w - T_{sen}] \tag{2-52}$$

or

$$\frac{dT_{sen}}{d\theta} = \frac{1}{\tau_{sen}}\,[(1 - K)\,T_a + K\,T_w] \tag{2-53}$$

where

$\tau_{sen} = mc_p/ha_{sen}$ = sensor time constant

T_{sen} = sensor temperature

K = coupling constant

T_w = wall temperature

T_a = air temperature

and the anticipator temperature is given by:

$$mc_p \frac{dT_{ant}}{d\theta} = Q_{ELE} - ha\,(T_{ant} - T_{air}) \tag{2-54}$$

or

$$\frac{dT_{ant}}{d\theta} + \frac{1}{\tau}\,T_{out} = \frac{Q_{ELE}}{mc_p} \quad \frac{1}{\tau}\,T_{air} \tag{2-55}$$

where:

T_{ant} = mc_p/ha_{out} = anticipator time constant

T_{out} = anticipator temperature

Q_{ELE} = heat of anticipator

Given the appropriate values of K, τ_{san}, τ_{ant} and Q_{ELE} the control of the conditioning plant can be approximated by expressing the instantaneous heating/cooling requirement as a function of the deviation from set point and the steady state droop.

Hence:

$$Q_i = \frac{Q_{H_{CAP}} \quad \delta T_{SP_H}}{D_{r_H}} - \frac{Q_{C_{CAP}} \quad \delta T_{SP_C}}{D_{r_C}} \qquad (2\text{-}56)$$

where:

$Q_{H_{CAP}}$ = output heating capacity

$Q_{C_{CAP}}$ = output cooling capacity

δT_{SP_H} = deviation from heating set point

δT_{SP_C} = deviation from cooling set point

D_{r_H} = steady state droop heating

D_{r_C} = steady state droop cooling

2.3.8 Heat Delivery

The delivery system is assumed to be instantaneous and completely within the conditioned space. The amount of heat delivered by any system is a function of the net thermal load on the space and the amount of parasitics that run in conjunction with the system. Thus, the heat required is:

$$Q_{need} = Q_{net} \pm T_F \sum Q_{para_i} x_i \tag{2-57}$$

and

$$T_F = \frac{Q_{need}}{Q_{out} + \sum Q_{para\ x}} \tag{2-58}$$

or

$$Q_{need} = \frac{Q_{net}}{1 \pm \frac{\Sigma Q_{para_i} x_i}{Q_{out} + \Sigma Q_{para_i} x_i}} \tag{2-59}$$

where:

Q_{need} = heat required

Q_{net} = net thermal load

T_F = time fraction

Q_{para} = parasitic power

x = fraction of parasitic to space

Q_{out} = output of conditioning systems

2.3.9 Furnace System

The response of the furnace has been modeled based on the method presented in the energy flow section of this report. If each control volume is assumed to be at a uniform temperature, then the temperature response of the control volume is given by:

$$Q_{in} - hf\ (\Delta T,\nu)\ A\ (T_{cv_1} - T_{cv_2}) - AF_{1-2}\ \sigma(T^4_{cv_1} - T^4_{cv_2}) - (\sum_{o=o}^{1} m_o h_o - \sum_{i=o}^{n} m_i h_i) = \frac{d[c_p \rho f(T_{cv_1})\ V\ T_{cv_2}]}{d\theta} \tag{2-60}$$

with

$$\frac{\partial \rho}{\partial \theta} = -(\nabla \cdot \rho\ V)$$

which gives

$$\sum_{i=o}^{n} m_i = \sum_{o=o}^{h} + \frac{d[\rho\ f\ (T_{cv_i})\ V]}{d\theta} \tag{2-61}$$

By expanding this approach for each of the furnace control volumes, an expression for response of the furnace subsystem can be written in the form of equation (2-18). Once the functional relationships between the elements of the A matrix (pertaining to the furnace) and the system state variables have been established, the furnace response can be evaluated. The nonlinear elements of the A matrix represent the dynamic flows through the furnace and can be evaluated by solving a series of algebraic and differential equations. For the case of an adiabatic stack, the desired relation for natural furnace flows are given in equations (2-62) through (2-65).

$$\frac{\dot{m}_{ft}\,(1 - e^{-2Dr_{ss}})\,(1 + D_r)}{\dot{m}_{s_{ss}} - e^{-2Dr_{ss}}\,\dot{m}_{ft}\,(1 = D_r)} =$$

$$\left[\frac{T_\infty^{-1} - [(T_F + T_H D_r)/(1 + D_r)]^{-1}}{T_\infty^{-1} - T_{s_{ss}}^{-1}}\right]^{.555} \cdot \left[\frac{T_{s_{ss}}\,(1 + D_r)}{T_F + T_H D_r}\right]^{.644} \qquad (2\text{-}62)$$

where

$$m_{ft} = \left[\frac{T_i^{-1} - T_F^{-1}}{T_{i_{ss}} - T_{F_{ss}}}\right]^{.555} \cdot \left[\frac{T_{F_{ss}}}{T_F}\right]^{.644} \cdot m_{F_{ss}}$$

and

$$\dot{m}_F = \frac{\dot{m}_{ft}\,(1 - e^{-2D_{r_{ss}}})}{1 - \dfrac{1^{-2D_r\,ss}\,(1 + D_r)\,\dot{m}_{ft}}{m_{s_{ss}}}} \qquad (2\text{-}63)$$

$$\dot{m}_h = D_r\,\dot{m}_F \qquad (2\text{-}64)$$

$$m_s = \dot{m}_F + \dot{m}_h \qquad (2\text{-}65)$$

The natural (draft) furnace flows can be determined by solving equations (2-62) for D_r and then solving equations (2-63), (2-64) and (2-65) for the flue, draft hood, and stack flows respectively. Evaluation of the actual furnace flows requires a simultaneous solution of equations (2-62) through (2-65) and the differential equation that describes heat transfer from the stack.*

The furnace equations have been presolved, and the results of system efficiency (which includes added infiltration due to stack flow) versus percent ON time are shown in Figure 2-16. The power burner and natural draft furnaces have steady state system efficiencies of 71 percent and 63 percent, respectively.

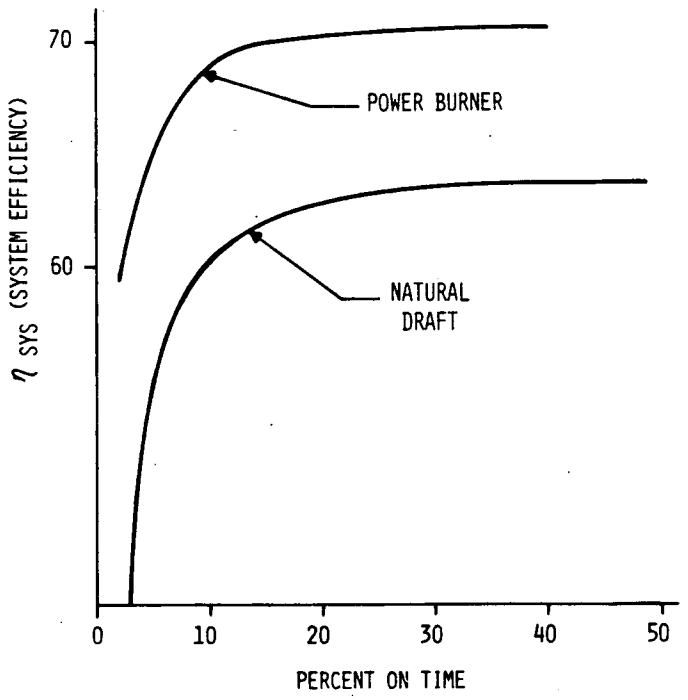


Figure 2-16. System Efficiency versus Percent ON Time (Gas and Oil Systems)

2.3.10 System Solution

All finite difference equations are solved by standard Euler marching techniques using a fixed time step size while all disturbances on the system (such as weather and internal loads) are updated every hour. The surface and midplane temperature

*For the oil furnace, equations (2-62) through (2-65) are rewritten to describe the induced draft systems.

response of the north, east, south and west walls, as well as the furnace response, for the seventh day of January 1957 (Minneapolis) can be seen in Figure 2-17. Figures 2-18 through 2-20 show the comparison between a fully dynamic system, a steady state system and a dynamic system that ignores the effect of thermostatic droop for a winter day, a summer day, and the yearly cycle.

2.3.11 Numerical Solution of Design Algorithms

2.3.11.1 Tank--The size of the tank is determined by a Newton-Ralphson algorithm. The constraints are:

$$\begin{array}{l} \text{Given:} \quad y = f\ (x) \\ \text{Find:} \quad x \text{ such that: } y = c \end{array} \Bigg| \ \theta = t \qquad (2\text{-}66)$$

The iteration procedure determines successive values of x by the expression:

$$x_{n+1} = x_n - \frac{f(x_n)}{\frac{\partial y}{\partial x}} \qquad (2\text{-}67)$$

From equation (2-8) and (2-67)

$$\frac{\partial y}{\partial x} = \frac{f(x_{n+1}) - f(x_n)}{x_{n+1} - x_n}$$

or

$$x_{n+1} = x_n - \frac{f(x_n)\ (x_{n+1} - x_n)}{f(x_{n+1}) - f(x_n)}$$

thus

$$x_{n+1} = x_n + \frac{\delta f(x_n)}{f(x_n + \delta) - f(x_n)} \qquad (2\text{-}68)$$

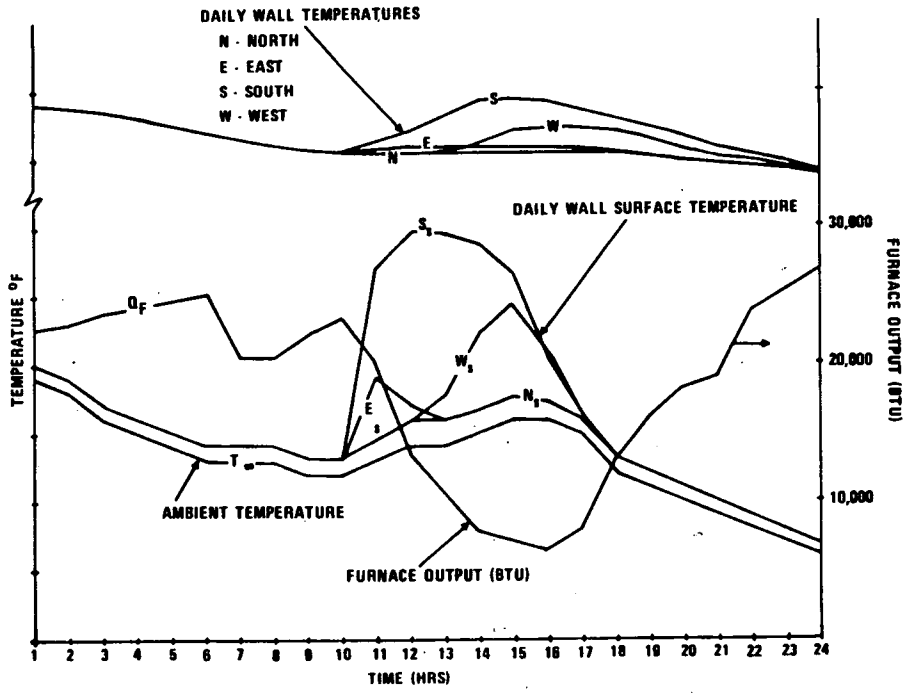


Figure 2-17. Minneapolis Single-Family System Response (Load) for January 7, 1957

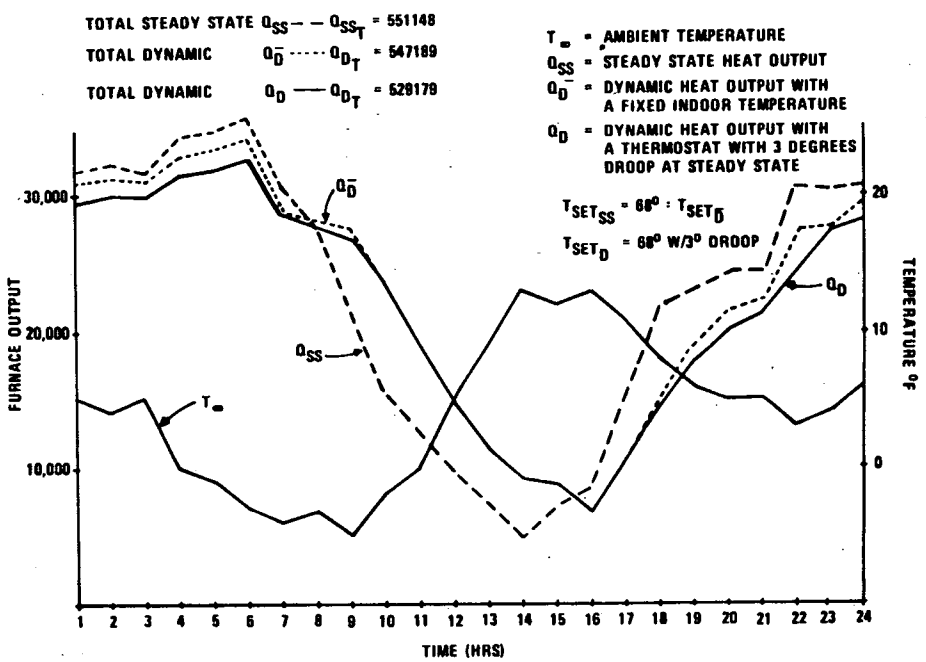


Figure 2-18. Comparison of Temperature Response for the Steady State, Quasi-Dynamic, and Fully Dynamic System for the Single-Family Residences in Minneapolis (Heating)

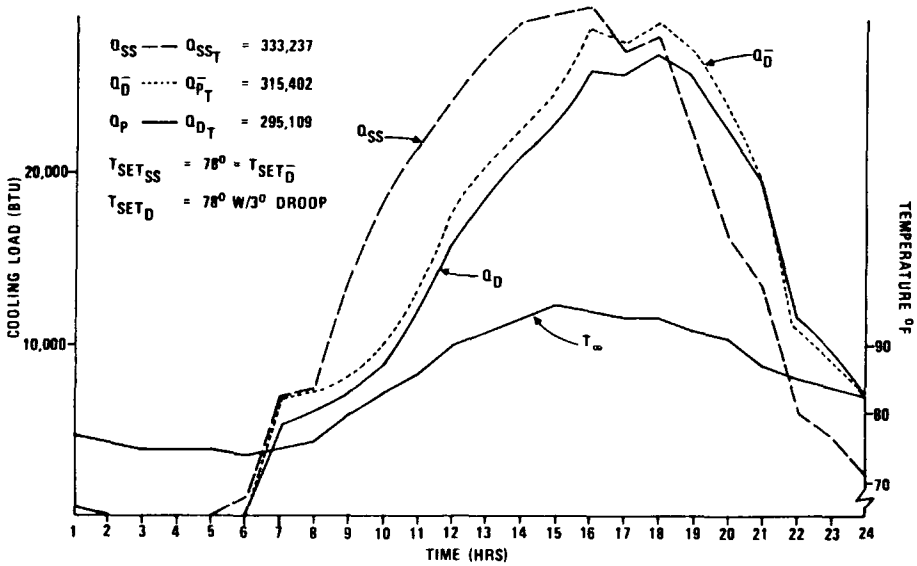


Figure 2-19. Comparison of Temperature Response for the Steady State, Quasi Dynamics, and Fully Dynamic System for the Single-Family Residences in Minneapolis (Cooling)

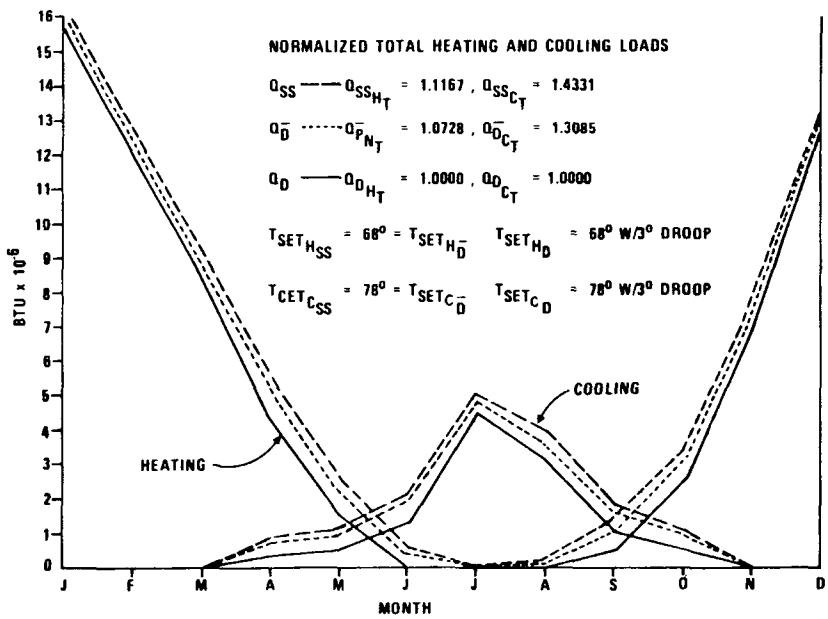


Figure 2-20. Annual Heating and Cooling Profile for the Minneapolis Single-Family Residence as Predicted by the Steady State Model, Quasi State Mode and the Dynamic Mode

where:

c = desired ice content at end of project

x = dimension of tank

f(x) = complicated function that relates the size of the tank to the energy balance between the system and the environment

δ = incremental change in variable x

The program determines tank size by making an initial guess on the tank dimension and then iterating equation (2-67) over the required time period until the system constraints have been fulfilled. This process is outlined in Figure 2-21.

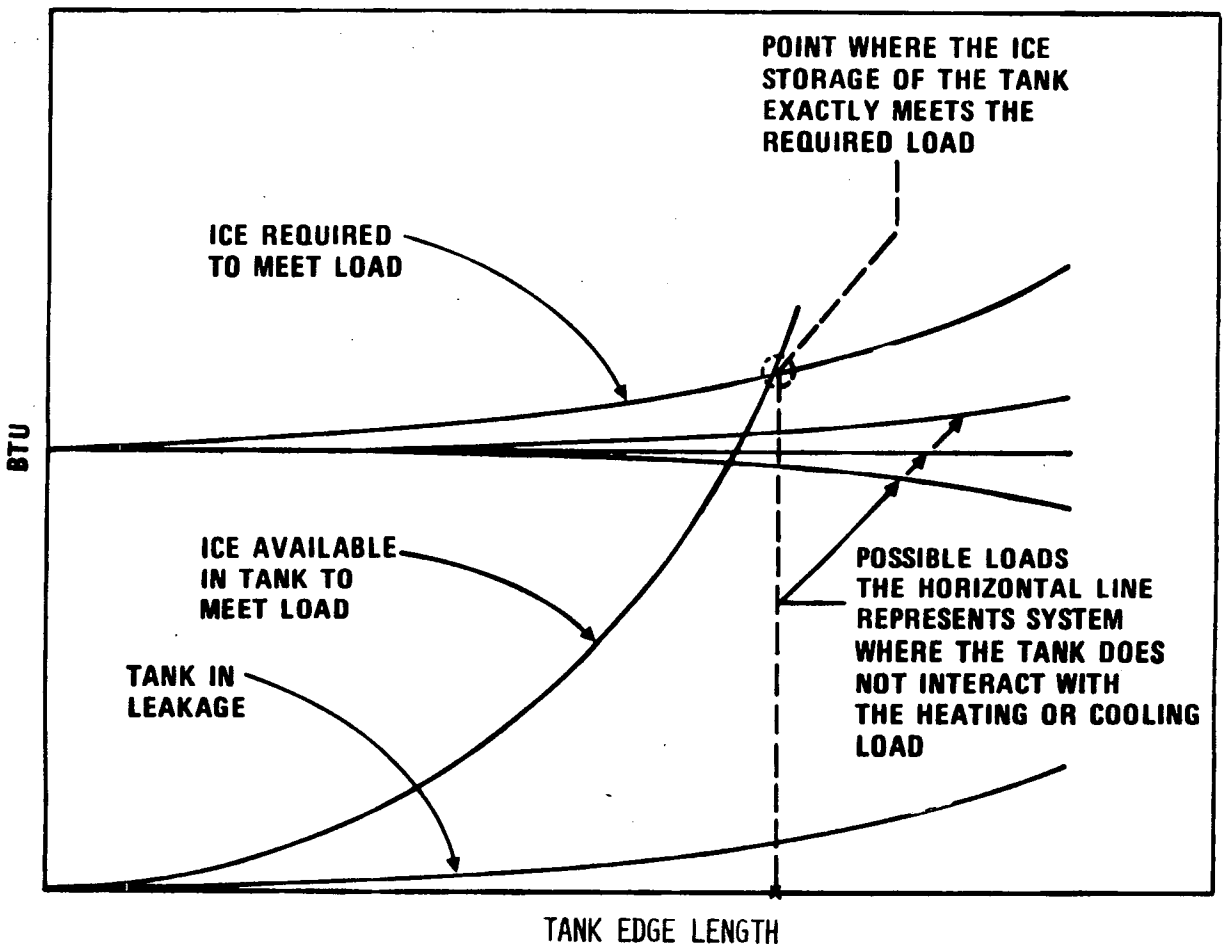


Figure 2-21. Tank Sizing Algorithm

2.3.11.2 Collectors--The size of the collectors is determined by a combination of bisection and secant algorithms. The algorithms seek to find the minimum collector area that will satisfy the constraint:

$$\vartheta = PF \tag{2-69}$$

when

$$\frac{\partial \vartheta}{\partial \theta} = 0 \tag{2-70}$$

where

ϑ = current volumetric packing fraction

PF = peak packing fraction

Figure 2-22 shows the sequence of iterations required to determine the collector area. Initially, constraint (2-70) is dropped and (2-69) is assumed to occur at the end of the heating season. A bisection method is used with the constraints:

Given: $= f(z)$

Find z such that $= PF$ @ θ = end of heating

The iteration procedure determines successive values of z on the interval [a b] if $f(z_a)\, f(z_b) \leq 0$ by:

$$XM \quad (Z_{a_n} + Z_{b_n})/Z$$

$$\begin{aligned} Z_{a_{n+1}} &= Z_{a_n} \\ Z_{b_{n+1}} &= XM \end{aligned} \qquad f(z_a)\, f(XM) \leq 0 \tag{2-71a}$$

$$\begin{aligned} Z_{a_{n+1}} &= XM \\ Z_{b_{n+1}} &= Z_n \end{aligned} \qquad f(z_a)\, f(XM) > 0 \tag{2-71b}$$

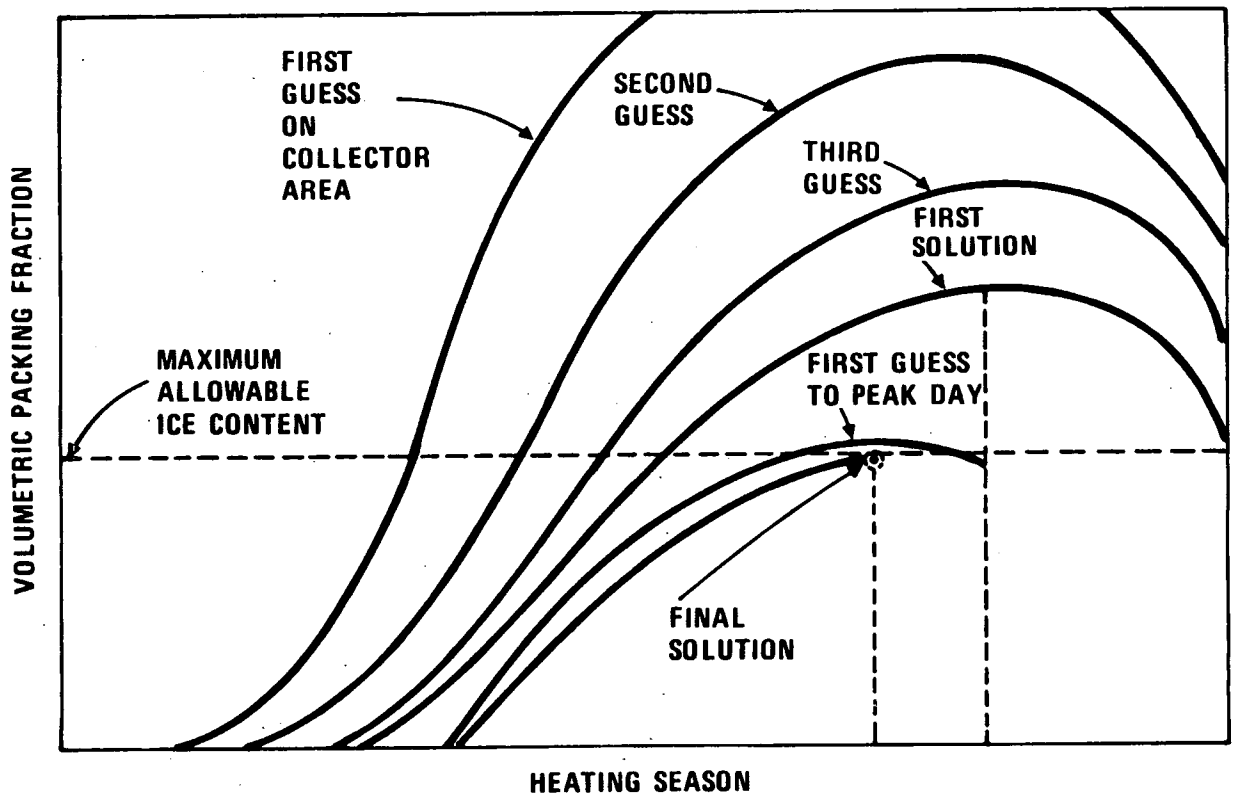


Figure 2-22. Collector Sizing Algorithm

$$Z = XM \tag{2-72}$$

Once this method has converged, the program then checks to see if constraints (2-69) and (2-70) are satisfied concurrently. If not, a second method is used subject to the constraints:*

$$\text{Given} \quad \vartheta = f(z) \tag{2-73}$$

$$\text{Find } z \text{ such that} \quad = PF \ @ \quad \frac{\partial \vartheta}{\partial \theta} = 0$$

The iteration procedure determines successive values of z by:

$$Z_{n+1} = Z_n - f(Z_n) \frac{Z_n - Z_{n-1}}{f(Z_n) - f(Z_{n-1})} \tag{2-74}$$

* The need for more than one algorithm is caused by numerical problems associated with a poor first guess on the collector area and the fact that the term $\frac{\partial \vartheta}{\partial \theta}$ is a function of the collector area.

where the last value of XM is used as an initial guess on Z_{n-1}. Once all the system constraints have been fulfilled, the minimum required collector area will be known.

2.4 COMPUTER PROGRAM

Listings of the residential versions of ACESIM are given in the Appendix. The overall subroutine flow diagrams of these programs are shown in Figures 2-23 through 2-27. Tables 2-2 through 2-5 give a brief description of the major function of the various subroutines.

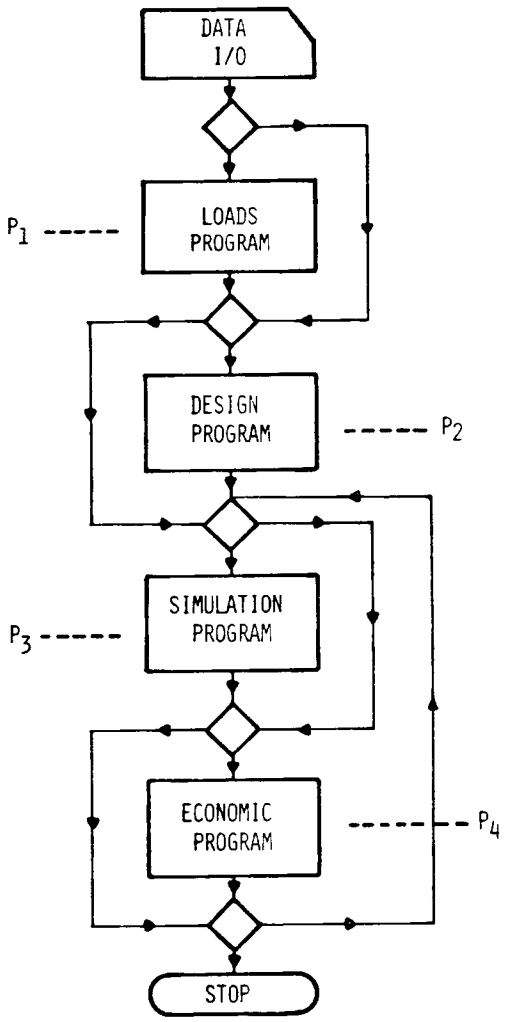


Figure 2-23. ACESIM Executive P-Level Routine

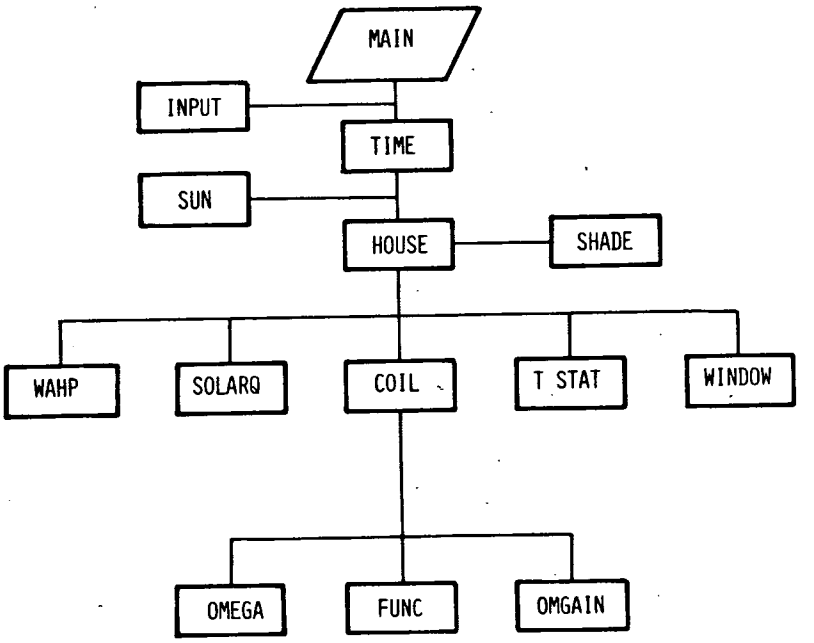


Figure 2-24. LODMOD P_1 Subroutine Flow

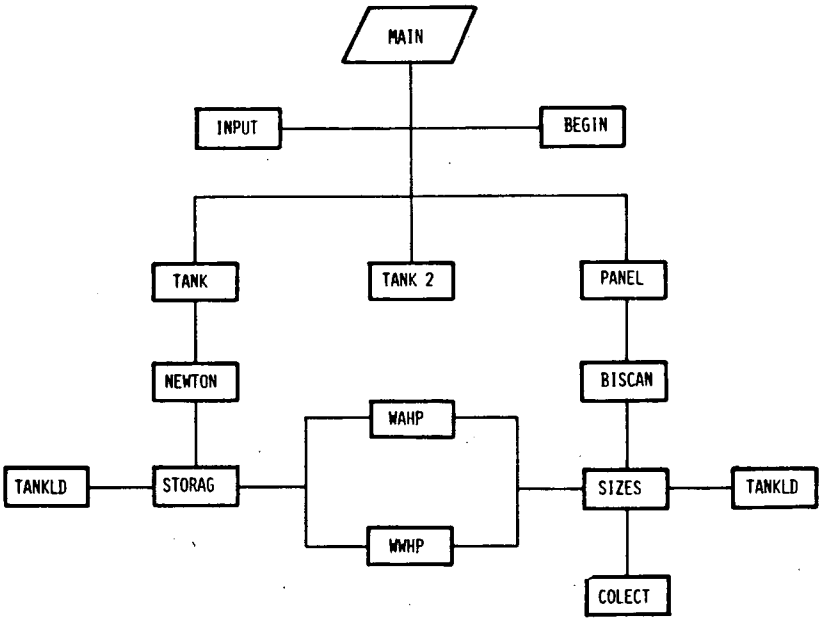


Figure 2-25. DESMOD P_2 Subroutine Flow

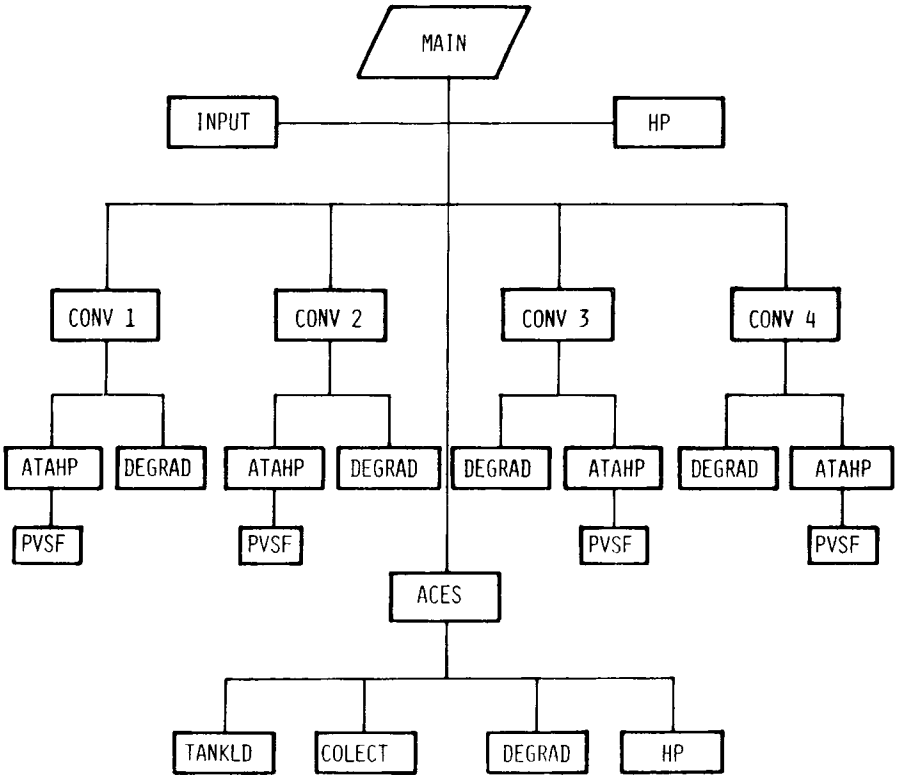


Figure 2-26. SIMMOD P_3 Subroutine Flow

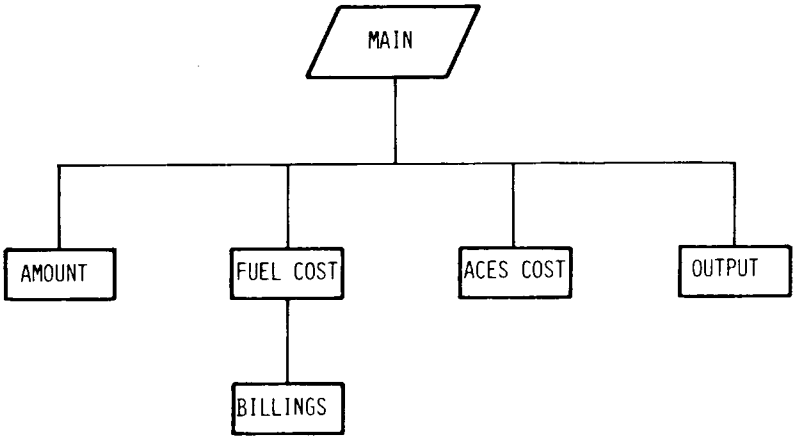


Figure 2-27. ECOMOD P_4 Subroutine Flow

Table 2-2. Functions of Subroutines Contained in LODMOD

MAIN	MAIN ROUTINE THAT CALLS OTHER ROUTINES, INITIALIZES ALL PARAMETERS, AND SETS UP OUTPUT DATA
INPUT	ROUTINE THAT READS ALL USER SPECIFIED INPUT DATA
TIME	ROUTINE THAT SEQUENCES TIME, READS WEATHER, SETS UP SOLAR CALCULATIONS
SUN	ROUTINE THAT CALCULATES APPROXIMATE SUN SURFACE ANGLES AND DETERMINES RADIATION INTENSITIES
HOUSE	ROUTINE THAT DETERMINES ENERGY FLOW WITHIN THE USER SPECIFIED STRUCTURE AND EVALUATES THE TIME DEPENDENT HEATING, COOLING AND HOT WATER LOAD
SHADE	ROUTINE FOR DETERMINING SHADE FACTORS IF NECESSARY
WAHP	ROUTINE THAT MODELS A WATER-TO-AIR HEAT PUMP
SOLARQ	ROUTINE THAT DETERMINES THE TOTAL RADIATION INCIDENT ON A SPECIFIED SURFACE
TSTAT	ROUTINE THAT APPROXIMATES THE OPERATION OF A THERMOSTAT
WINDOW	ROUTINE THAT DETERMINES THE TRANSMITTED SOLAR RADIATION
COIL	ROUTINES THAT EVALUATE THE COOLING COIL
OMEGA	
FUNC	
OMGAIN	

Table 2-3. Functions of Subroutines Contained in DESMOD

MAIN	MAIN ROUTINE THAT CALLS OTHER ROUTINES, INITIALIZES ALL PARAMETERS, AND SETS UP OUTPUT DATA
INPUT	ROUTINE THAT READS ALL USER SPECIFIED INPUT DATA
BEGIN	ROUTINE THAT DETERMINES THE EFFECTIVE BEGINNING OF THE HEATING AND COOLING SEASON
TANK	CALLING ROUTINES FOR SIZING THAT TANK FOR THE FULL ACES
NEWTON	ITERATION ROUTINE FOR DETERMINING CORRECT TANK SIZE
STORAG	GLOBAL ENERGY BALANCE ROUTINE
TANKLD	DYNAMIC TANK LOSS ROUTINE
WAHP	WATER-TO-AIR HEAT PUMP ROUTINE
WWHP	WATER-TO-WATER HEAT PUMP ROUTINE
TANK 2	ROUTINE FOR SIZING THE TANK FOR THE MINIMUM ACES
PANEL	CALLING ROUTINE FOR SIZING THE COLLECTOR ARRAY
BISCAN	ITERATION ROUTINE FOR DETERMINING THE MINIMUM COLLECTOR AREA FOR A GIVEN TANK SIZE
SIZES	GLOBAL ENERGY BALANCE ROUTINE
COLECT	ROUTINE TO MODEL SOLAR COLLECTORS

Table 2-4. Functions of Subroutines Contained in SIMMOD

MAIN	MAIN ROUTINE THAT CALLS OTHER ROUTINES, INITIALIZES ALL PARAMETERS, AND SETS UP OUTPUT DATA
INPUT	ROUTINE THAT READS ALL USER SPECIFIED INPUT DATA
HP	REPRESENTS COMBINATION OF ROUTINES THAT MODEL THE WATER-TO-WATER, WATER-TO-AIR AND AIR-TO-AIR HEAT PUMP SYSTEMS
CONV1	ROUTINE THAT DETERMINES THE ENERGY USE PROFILE OF AN ALL ELECTRIC SYSTEM
CONV2	ROUTINE THAT DETERMINES THE ENERGY USE PROFILE OF A CONVENTIONAL HEAT PUMP SYSTEM (HIGH EFFICIENCY)
CONV3	ROUTINE THAT DETERMINES THE ENERGY USE PROFILE OF AN OIL HEATING AND ELECTRIC COOLING AND HOT WATER SYSTEM
CONV4	ROUTINE THAT DETERMINES THE ENERGY USE PROFILE OF A GAS HEATING AND HOT WATER SYSTEM AND AN ELECTRIC COOLING SYSTEM
ACES	ROUTINE THAT DETERMINES THE ENERGY USE PROFILE OF AN ACES SYSTEM
ATAHP	ROUTINES THAT MODEL AN AIR-TO-AIR HEAT PUMP SYSTEM
DEGRAD	
PVSF	
TANKLD	DYNAMIC TANK LOSS ROUTINE
COLECT	SOLAR COLLECTOR ROUTINE

Table 2-5. Function of Subroutines Contained in ECOMOD

MAIN	ROUTINE THAT READS IN ECONOMIC INPUT DATA ARRAY, CALLS OTHER ROUTINES NEEDED TO DEVELOP INPUTS TO ANNUAL COST ANALYSIS, CALCULATES ANNUAL LIFE CYCLE COSTS OF OWNING AND OPERATING ACES AND CONVENTIONAL SYSTEMS, AND CALCULATES PAYBACK AND TOTAL NET BENEFITS OF ACES COMPARED TO EACH CONVENTIONAL SYSTEM
AMOUNT	ROUTINE THAT READS THE HOUR-BY-HOUR ENERGY USE FILES GENERATED BY SIMMOD AND SLOTS THE ELECTRICAL ENERGY USAGE INTO THE APPROPRIATE RATE PERIOD
FUELCOST	ROUTINE THAT CALCULATES THE ANNUAL COST (USING TODAY'S PRICES) OF FUEL REQUIRED BY THE HVAC SYSTEMS UNDER ANALYSIS, CALLS ROUTINE BILLING IF STANDARD RATE STRUCTURE IS USED
BILLINGS	ROUTINE THAT CALCULATES ELECTRICAL COSTS FOR STANDARD RATES
ACESCOST	ROUTINE THAT CALCULATES THE TOTAL PRICE OF THE ACES SIMULATED IN SIMMOD
OUTPUT	ROUTINE THAT PRINTS OUT THE ECONOMIC ANALYSIS TABLES

SECTION 3.0
COMPUTER SIMULATION PROGRAM FOR COMMERCIAL BUILDINGS

3.1 GENERAL DESCRIPTION

The computer simulation program for a commercial ACES building (CACESS) is a two-in-one comprehensive package. The program is capable of either sizing all the required components for a commercial ACES or simulating a general eleven-zone (60,000 ft^2) building with up to one-hour time steps provided that the load profile of the system is given.

Hourly weather data for a typical year in each selected geographical area was obtained from the U. S. Weather Bureau. This data, together with the structural and operational characteristics of the building, was used by NBSLD to predict the hourly zone building loads.

The NBSLD computer program was used to predict the eleven-zone load profile of the building. Although CACESS is designed for an eleven-zone building, with slight modifications the program can be used to simulate any number of zones.

3.2 SYSTEM DESCRIPTION

A typical commercial ACES system consists of a multi-zone building, a heat pump capable of transferring a design day load, a cold side storage bin capable of supplying a heating load for a specified number of consecutive worst weather days in the heating season or capable of maximizing inter-seasonal energy transfer throughout the annual cycle, and a hot side storage tank capable of supplying a design day heating load.

A collector field may be added to the system in cold regions to substitute for any additional auxiliary heating requirements. Also, an air convector panel may be connected to the hot side of the heat pump to purge any excess heat.

A variable air volume control system and an enthalpy controlled economizer are installed to minimize heating, ventilating and air conditioning energy usage. The economizer is set for minimum ventilation in winter to minimize the heat loss of the system. Heating is furnished by a baseboard radiator system.

A schematic diagram of the commercial ACES is shown in Figure 3-1. The heart of the commercial variable air volume system is the central supply unit. The required energy use of the commercial building depends intimately on how this unit is controlled. For the purpose of this study we have assumed a standard discharge air controller. The discharge air is nominally controlled to $55^{o}F$. This air is supplied to the various building zones based on demand. The return air from all zones is mixed together at the economizer terminal, where the economizer senses the mixed air enthalpy and the ambient air enthalpy to determine the percentage of ambient air to be brought in and mixed with the return air. This mixed air will then be conditioned by cold storage to meet the $55^{o}F$ requirement for supply air. When a heating load occurs, the baseboard heating system is turned on to transfer heat from the hot storage to the demanding zone.

The sensors at the hot and cold storage tanks will decide when to run the heat pump. The limit temperatures for these tanks are $32^{o}F \leq T_{sc} < 50^{o}F$ and $90^{o}F \leq T_{sh} < 130^{o}F$, where T_{sc} is the cold storage temperature and T_{sh} is the hot storage temperature. Any excess heat generated by the heat pump (mostly in southern regions) will be dissipated through an outdoor purge coil. In the northern regions where the heating demand in winter is much larger than the cooling demand (internal heat generation such as people, office equip-

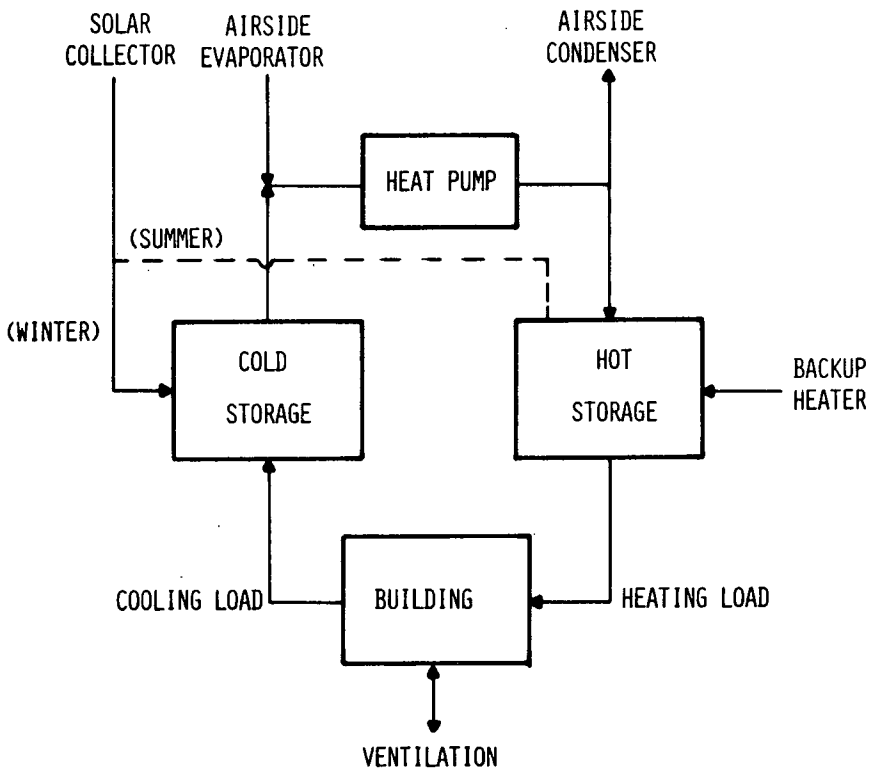


Figure 3-1. Schematic Representation of the Commercial ACES

ment, etc.), an auxiliary heat source, such as a solar collector field, will be used to melt the ice built up in the cold storage due to excessive use of the heat pump. The collector area is sized such that the packing factor in the storage tank, defined as the ratio of ice to storage volume, is not greater than 0.4.

3.3 MATHEMATICAL MODEL

The two state variables in this system are the hot storage tank temperature, T_{sh}, and the cold storage tank temperature, T_{sc}. Other components are assumed to be operated at quasi-steady state.

The temperature variation in each zone of the building affects the state of each zone thermostat which, in turn, affects the heating and cooling control of the whole system. The effect of this variation, however, is negligible for our purpose. It is assumed that the heating and cooling controllers are the function of the zonal loads only; therefore, the zonal load can be computed independently. This will save computing time since the loads will be computed only once and stored for subsequent use. The National Bureau of Standards NBSLD program is found to be best suited for hourly multi-zonal load calculation at the present time. No ventilation is allowed in this load calculation because the ventilation is controlled by the enthalpy economizer in CACESS.

The input requirements for the load calculation program include: zone temperature, zone relative humidity, sensible load, latent load, lighting load, equipment load, and people load.

The mathematical models of the equipment and the computing procedure of the system are briefly described in the following paragraphs.

3.3.1. Economizer

The economizer, shown schematically in Figure 3-2, is modeled as a variable ventilation system. Return air from the building is mixed with ambient air in a manner that minimum heating and cooling are required.

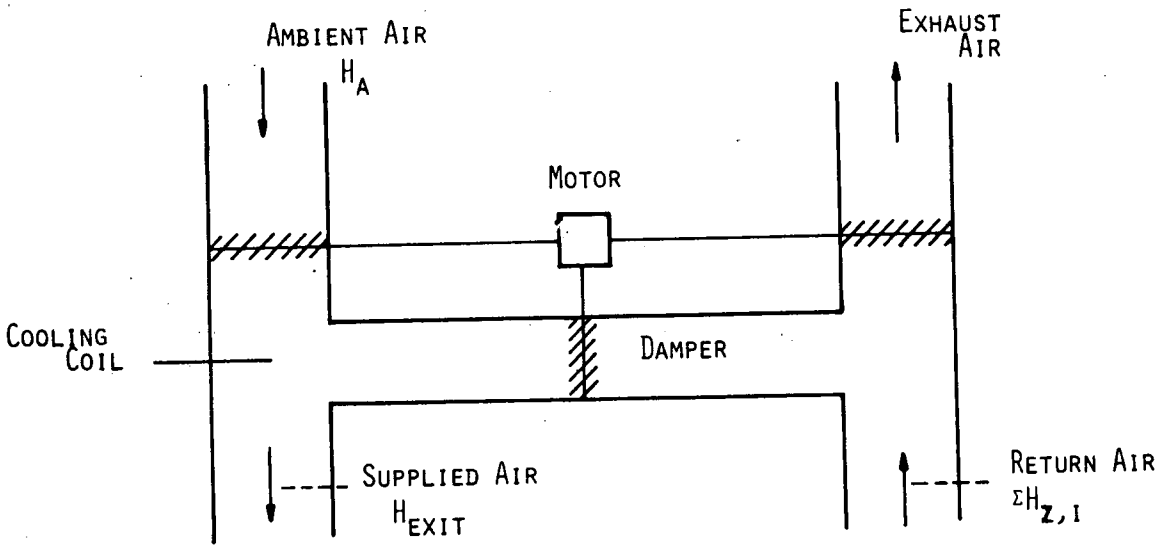


Figure 3-2. Schematic Diagram of the Economizer

For a given supplied air condition, the air flow rate at each zone is computed as follows:

$$Flow_i = - Q_{z,i}/(H_{z,i} - H_{exit}) \tag{3-1}$$

if,

$$Flow_i < F_{min}$$

$$Flow_i = F_{min}$$

where:

$Flow_i$ = Flow rate of supplied air at i^{th} zone

$Q_{z,i}$ = Zonal load (+: heating; - : cooling)

$H_{z,i}$ = Zonal enthalpy

H_{exit} = Supplied air enthalpy (@ 55°F and 95% humidity)

F_{min} = Minimum ventilation requirement

The total air circulation rate is:

$$Ftot = \sum_{i=1}^{Nzone} Flow_i \tag{3-2}$$

The fan power is calculated from the equation:

$$F_{power} = F_{pmax} \left(.8 \frac{Ftot}{Fmax} - .3\right) \tag{3-3}$$

Where:

F_{pmax} = Power consumption of fan at design condition

F_{max} = Flow rate of fan at design condition

The total load of the building is:

$$Q_{tot} = \Sigma Q_i + F_{power} + Q_{amb} \tag{3-4}$$

where:

$$Q_{amb} = \Sigma Flow_i \left(H_{z,i} - H_a\right) \tag{3-5}$$

In a commercial ACES, proper ventilation control can reduce the energy loss by up to 25 percent.

3.3.2 Heat Pump

The heat pump system is a combination of COPELAND 20KW (4RA3-2000-TSK) and 35KW (6RH1-3500-TSK) compressors. Any combination which meets the total annual cooling load of the simulated year is adequate. The performance curves for these compressors are shown in Figures 3-3 through 3-6.

The main function of the heat pump is to transport heat from the cold storage tank to the hot storage tank. A thermostat relay controls the heat pump operation. The range of operation of heat pump is:

$$32^oF \leq T_{sc} < 50^o \text{ for cold storage}$$

and

$$90^oF \leq F_{sh} < T_{seth} \text{ for hot storage}$$

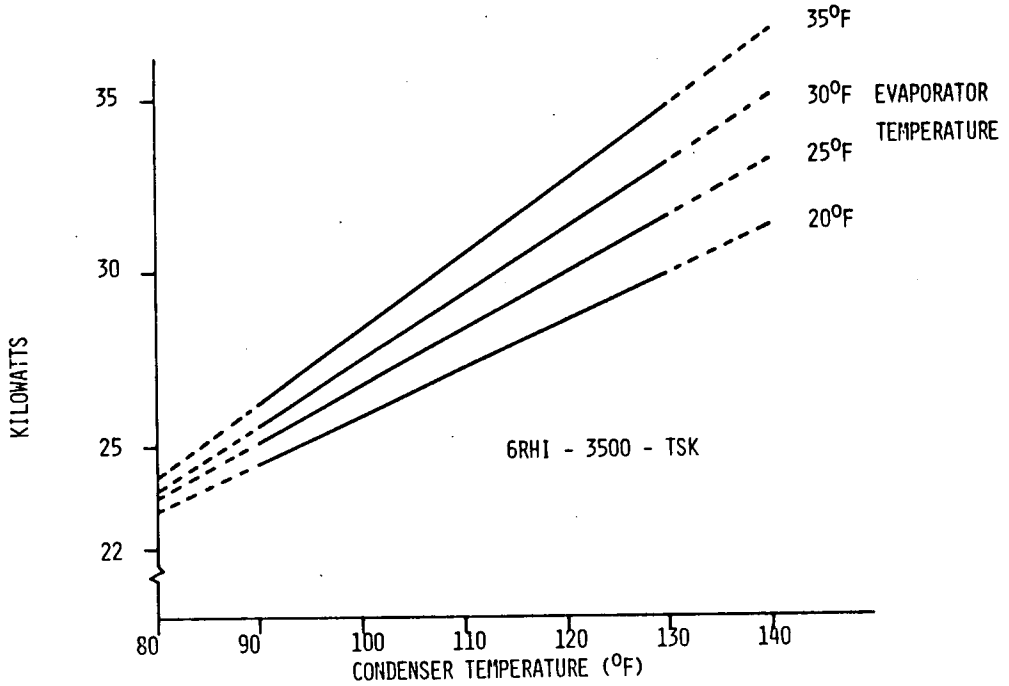


Figure 3-3. Compressor Input As A Function of Evaporator and Condenser Temperature

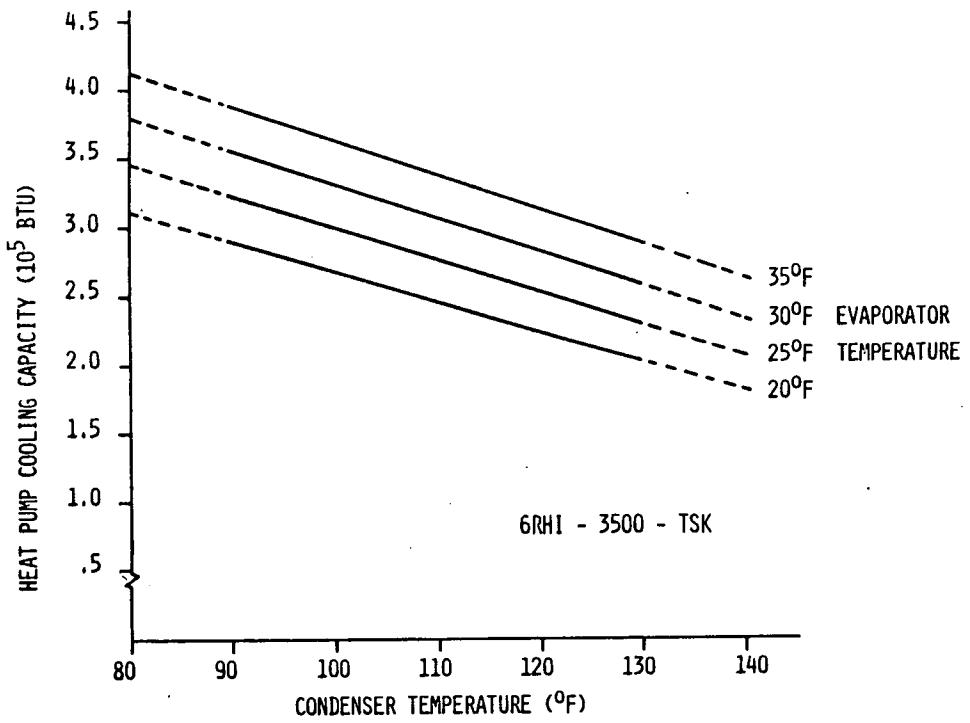


Figure 3-4. Cooling Capacity As A Function of Evaporator and Condenser Temperature

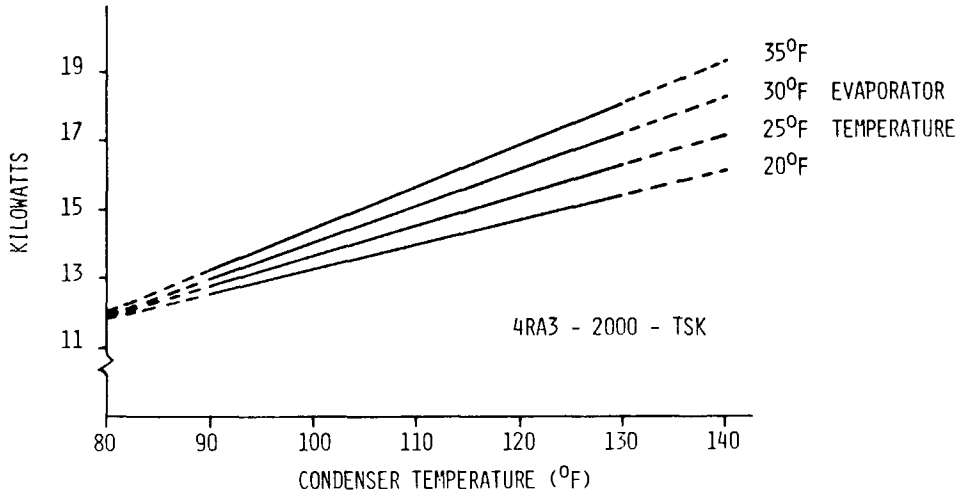


Figure 3-5. Compressor Input As A Function of Evaporator and Condenser Temperature

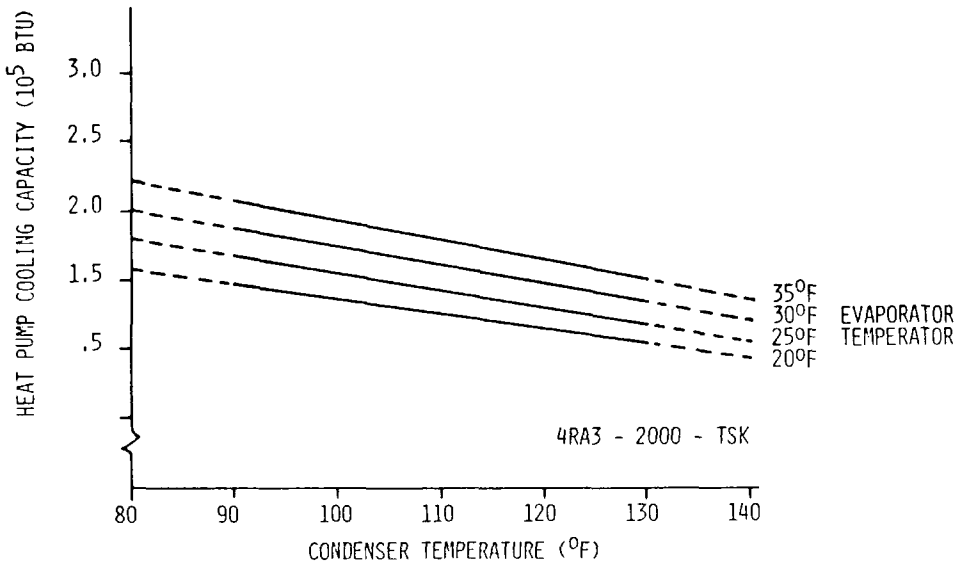


Figure 3-6. Cooling Capacity As A Function of Evaporator and Condenser Temperature

where T_{seth} is the upper limit of the hot storage temperature. This temperature is the reset thermostat set point, which is computed as follows:

$$T_{seth} = 130^{o}F - D*T_{amb} \tag{3-6}$$

$$D = \frac{(130 - 90)^{o}F}{T_{exit}} = \frac{40}{55} = .73 \tag{3-7}$$

The relay function for hot storage is defined in Figure 3-7 as follows:

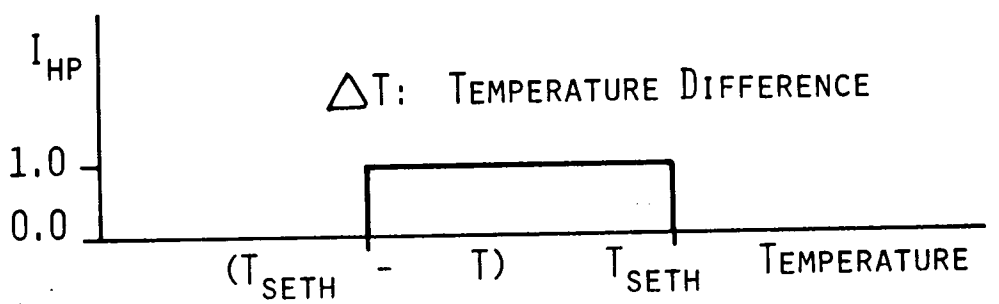


Figure 3-7. Relay Function I_{hp}

In the cold storage, the relay function depends on the season. In the heating season, it follows the hot storage relay function and equals unity when the cold storage is greater than 50°F. In the cooling season, it is unity at night if the packing factor is less than .4 in the cold storage tank, and zero otherwise.

The heat pump is ON when either (or both) relay functions are unity and OFF otherwise.

3.3.3 Storage Tank

The hot storage tank is sized so that it will hold enough heat for a user-specified number of hours on the coldest day of a year.

$$(mc_p)_h = \frac{Q_{hmax} \times time}{24} \times \frac{1}{(130 - 90)^oF} \tag{3-8}$$

where $(mc_p)_h$ is the thermal capacity of the hot tank and Q_{hmax} is the total heating load of the coldest day in the year. The heat loss of the tank is insignificant in the sizing of the hot tank.

The cold storage bin is sized according to the definitions given for the Full and Minimum ACES in Section 1.0. Since the bin is large, the heat transfer rate from the bin to the ground can be significant and is considered in the sizing calculation.

For a defined heat pump heating COP, a worst heating load Q_{hcons}, and the cooling load Q_{ccons} during the same period, the cold storage has to supply the following amount of heat:

$$Q_{nmax} = Q_{hcons} \left(1 - \frac{1}{COP}\right) - Q_{ccons} \tag{3-9}$$

This must equal the summation of the fusion heat in the cold storage tank and the heat transfer due to the temperature differences between the storage and the ground.

$$\begin{aligned} Q_{nmax} &= Q_{fuse} - UA\,(T_{sc} - T_{ground}) \\ &= \rho V h_{fuse} - UA\,(T_{sc} - T_{ground}) \end{aligned} \tag{3-10}$$

where ρ is the average density of water, V is the volume of the bin, h_{fuse} is the heat of fusion per pound of water, U is the average heat transfer coefficient at the tank surface, and A is the surface area of the bin.

The two unknowns, V and A, are related to each other from geometrical relationships. Thus A or V can be solved by an iteration method (Newton iteration method is used in this program).

The tanks transfer heat to each other through the heat pump. They also transfer heat for heating and cooling purposes to the building zones. The energy balance in each storage tank is:

$$(mc_p)_{sc} \frac{\partial T_{sc}}{\partial t} = Q_{col} - Q_{hpc} - Q_{lsc} - Q_{cb} \tag{3-11a}$$

for cold storage, and

$$(mc_p)_{sh} \frac{\partial T_{sh}}{\partial t} = Q_{hph} - Q_{lsh} - Q_{hb} \tag{3-11b}$$

for hot storage,

where:

$(mc_p)_{sc}$	= Thermal capacity of the cold storage
$(mc_p)_{sh}$	= Thermal capacity of the hot storage
Q_{col}	= Collected solar energy
Q_{hpc}	= Heat extracted by the heat pump
Q_{lsc}	= Loss from cold storage to ambient
Q_{cb}	= Cooling load from the building

$Q_{hph} = Q_{hpc} (1 + 1/COP)$ = Heat gain from the heat pump

Q_{lsh} = Loss from hot storage to ambient

Q_{hb} = Heating load from the building

These derivatives can be integrated forward in time to obtain the storage temperatures. When the cold storage temperature drops down to 32°F or less, we assume that the heat transfer is reversible, uniform, and linear, so that we can adjust the storage temperature to account for the heat of fusion of water.

$$Q_{fuse} = (mc)_{sc} (32.0 - T_{sc})$$

and reset $\partial T_{sc} / \partial t$ or T_{sc},

$$\frac{\partial T_{sc}}{\partial t} = 0 \quad \text{or} \quad T_{sc} = 32^{o} \tag{3-12}$$

3.3.4. Collectors

The solar collectors can be single, double, or unglazed. Collector efficiency is defined in the form:

$$\eta_c = FR\ (\tau\ a) - UA(V) \frac{T_{in} - T_{amb}}{I} \tag{3-13}$$

where:

$FR\ (\tau\ a)$ = Function relating the collector characteristics

UA = Collector heat loss coefficient

T_{in} = Collector inlet temperature

(V) = Denotes a function of the wind velocity

T_{amb} = Ambient temperature

I = Solar insolation rate

The collectors are arranged in the configuration depicted in Figure 3-8. Normally, the series arrangement is not desired because it would degrade the overall collector efficiency due to higher outlet temperatures. Nevertheless, due to the field area (usually the building roof) limitation, the collectors may have to be arranged in series.

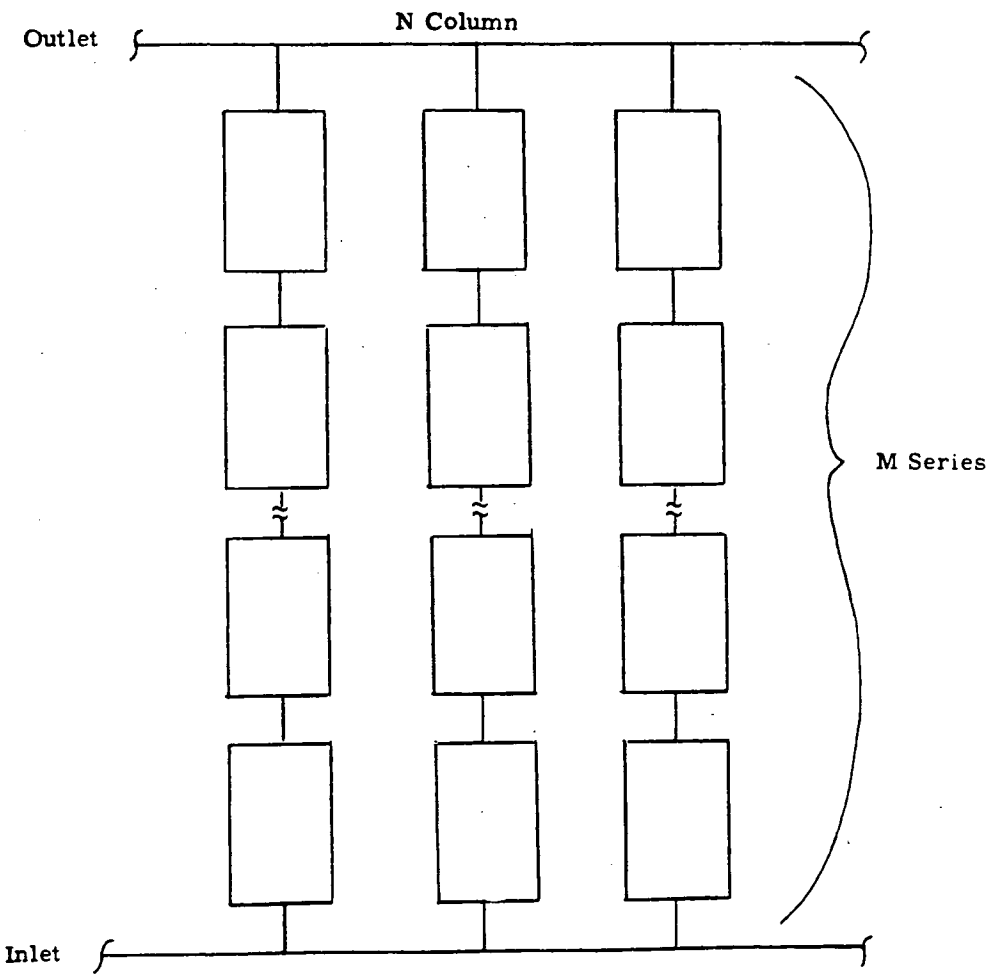


Figure 3-8. Collector Arrangement

The collectors are sized to satisfy the packing factor constraint in the cold storage, which states that the packing factor (the ratio of ice to the total water volume) cannot exceed a certain percentage throughout the year.

An iteration method, such as described below, will give an exact solution for the collector area when a packing factor is given.

The first step is to approximate the collector area such that the packing factor is within 10 percent of the desired result by using the following algorithm (Newton method):

Given a collector area A_{col}, compute packing factor F_p, compare F_p with the desired packing factor F_{po}

$\left| F_p - F_{po} \right| \leq 0.1$?

- Yes, done
- No, guess a new collector area

$$A'_{col} = A_{col} - \frac{F_{po} - F_p}{\partial F_p / \partial A_{col}}$$

Repeat the procedure until the constraint is met.

Once this constraint has been met, the algorithm then uses the bisection iteration method to find a solution that is within an acceptable error bound set by the user. Assume that we wish to obtain the solution within 1 percent error.

Compute the packing factor with a given area, A_{col}, compare it with the desired packing factor: $\left| F_p - F_{po} \right| \leq .05$?

- Yes, done
- No, guess a new collector area, A_{col1} such that the new packing factor F_{p1} will give $(F_{p1} - F_{po})$ with a sign opposite

$(F_{pl} - F_{po})$. Find the new guess: $A'_{col} = 1/2\ (A_{col} + A_{coll})$

Repeat the procedure until the constraint is met.

The reason why the second procedure is needed is the uncertainty caused by the large integration time step coupled with the relay function of the heat pump.

The collector loop is assumed to operate at quasi-steady state condition so that the energy transfer from collectors to the storage tank (through the heat exchanger) can be solved by an iteration method.

The heat exchanger is modeled as a simple counterflow one with the effectiveness defined as follows:

$$\varepsilon = \frac{1 - \exp\left[-\left(1 - \frac{C_{min}}{C_{max}}\right)\frac{UA}{C_{min}}\right]}{1 - \frac{C_{min}}{C_{max}} \exp\left[-\left(1 - \frac{C_{min}}{C_{max}}\right)\frac{UA}{C_{min}}\right]} \tag{3-14}$$

where:

UA = Total coefficient of heat transfer

C_{min} = The minimum fluid flow rate

C_{max} = The maximum fluid flow rate

3.4 COMPUTER PROGRAM

A listing of the commercial versions of ACESIM is given in the Appendix. The overall subroutine flow diagrams of this program are shown in Figures 3-9 through 3-13. Table 3-1 gives a brief description of the major functions of the various routines.

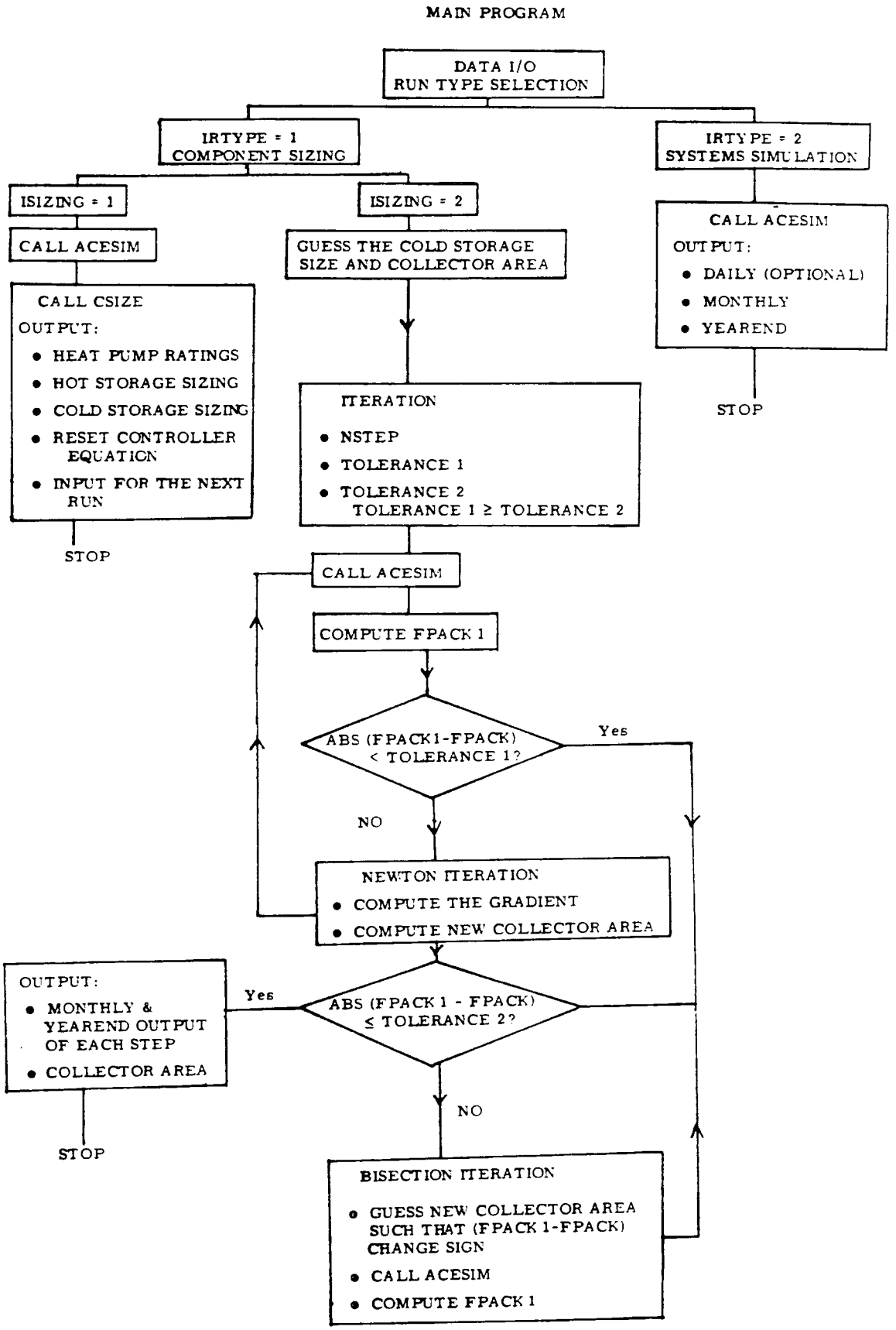


Figure 3-9. Flow Chart for the Commercial Version of ACESIM

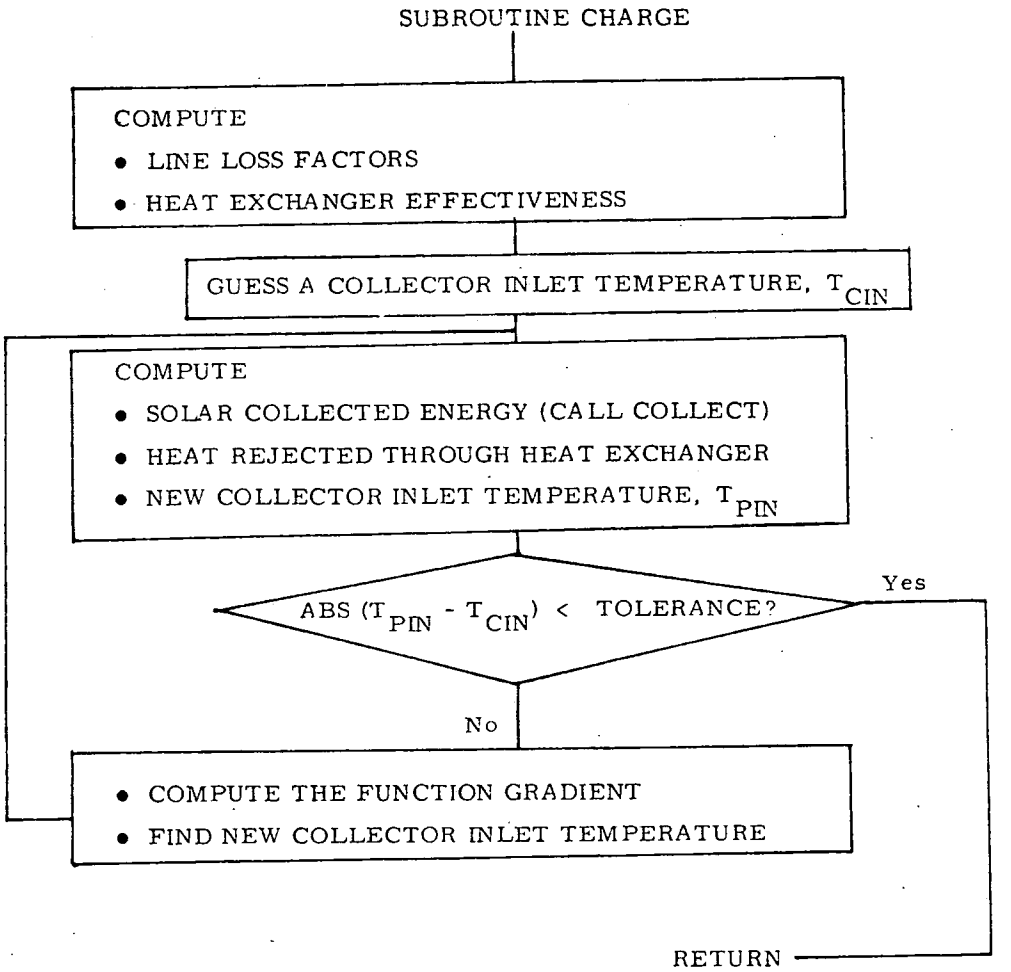


Figure 3-10. Flow Chart for the Tank Charging Process for the Commercial Building

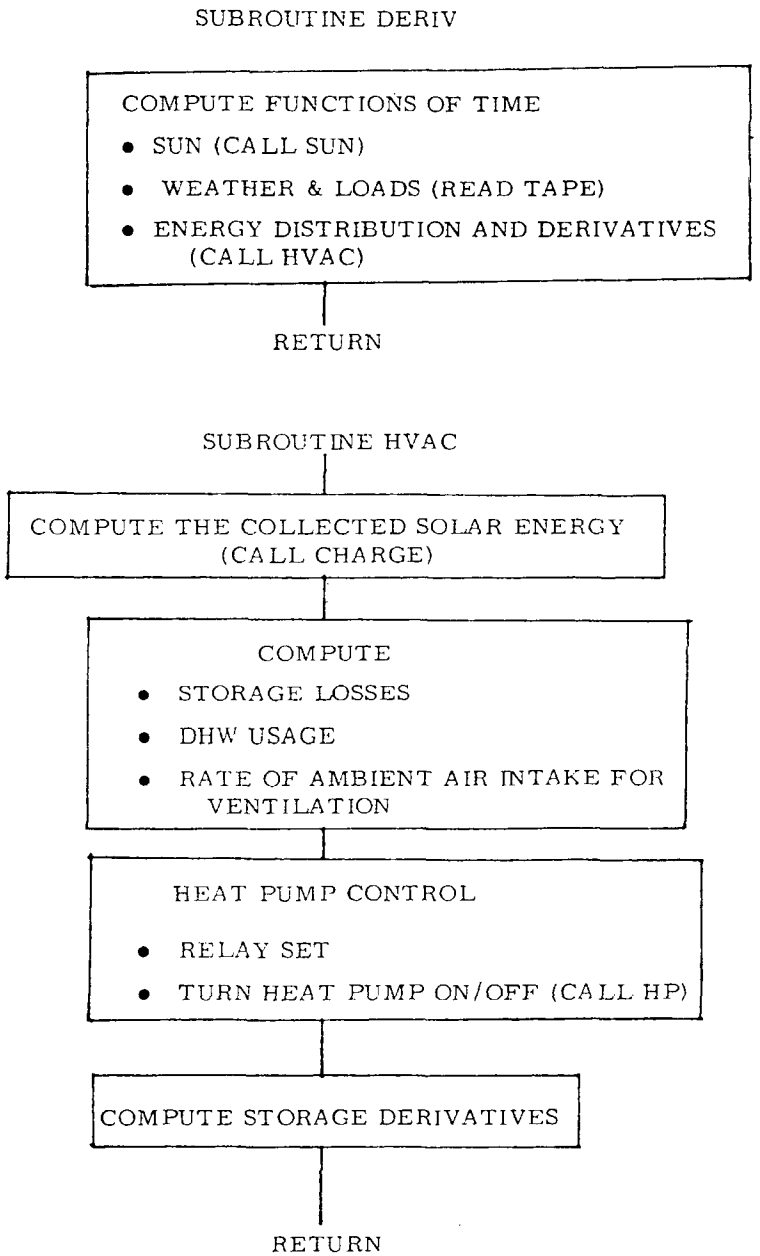
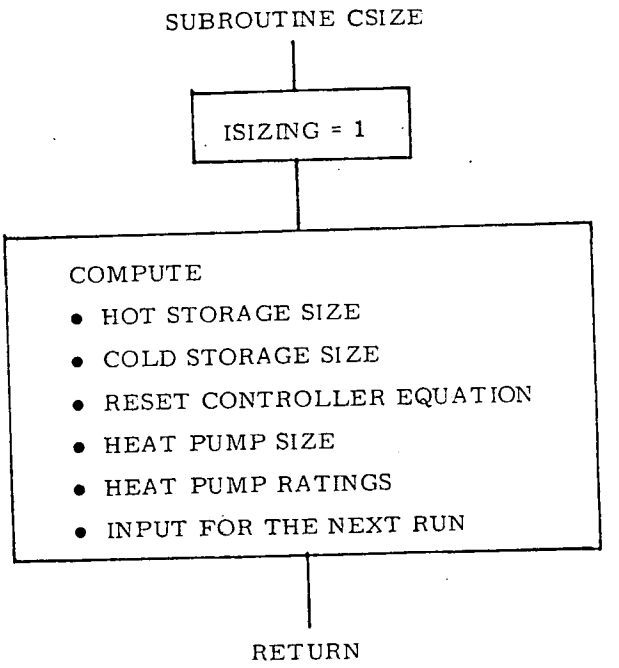


Figure 3-11. Flow Chart for the System Operation of the Commercial Building



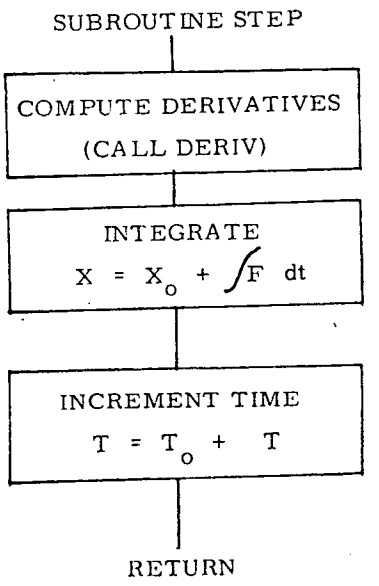


Figure 3-12. Flow Chart for the Sizing Program for the Commercial Building

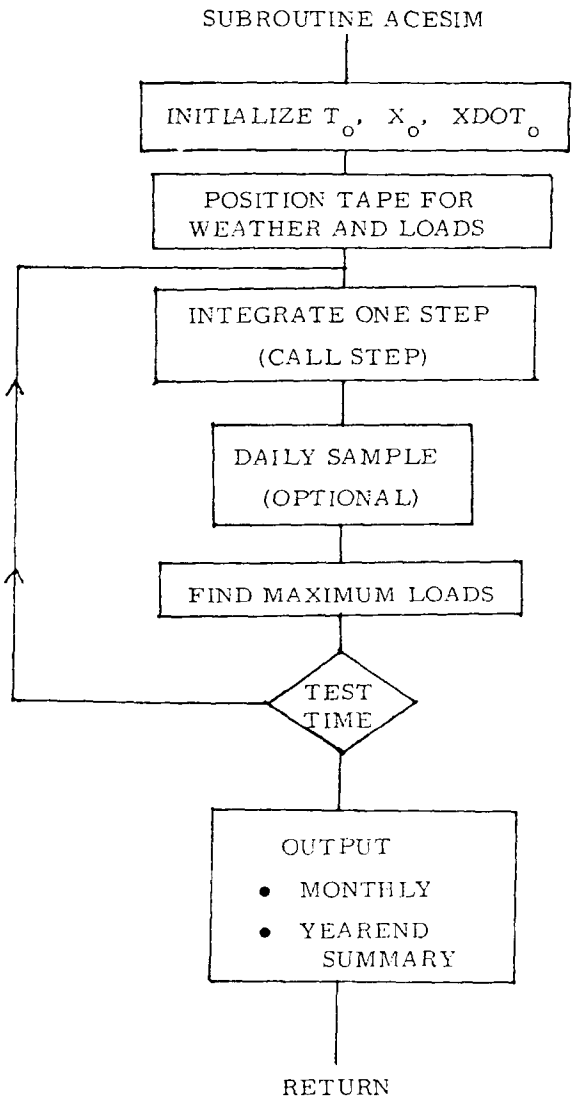


Figure 3-13. Flow Chart for the Simulation Program for the Commercial Building

Table 3-1. Functions of Subroutines Contained in the Commercial Version of ACESIM

MAIN	MAIN ROUTINE THAT CALLS OTHER ROUTINES, INITIALIZES ALL PARAMETERS AND SETS UP OUTPUT DATA
CSIZE	ROUTINE THAT SIZES THE HEAT PUMP, HOT AND COLD STORAGE TANK AND RESET CONTROLLER EQUATION
CHARGE	ROUTINE THAT DETERMINES THE AMOUNT OF ENERGY AVAILABLE FROM THE COLLECTOR FOR CHARGING THE STORAGE TANK
DERIV	ROUTINE THAT SEQUENCES THE SYSTEM STATE VARIABLES (COMPUTER DIVERTER) AND CALLS RADIATIVE AND WEATHER ROUTINES
HVAC	ROUTINE THAT PERFORMS ENERGY BALANCE ON THE COMMERCIAL OFFICE BUILDING
STEP	ROUTINE THAT INTEGRATES SYSTEM STATE VARIABLES AND SEQUENCES THE INDEPENDENT LINE VARIABLE
ACESIM	ROUTINE THAT SIMULATES THE OPERATION AND ENERGY USAGE OF THE COMMERCIAL OFFICE BUILDING

SECTION 4.0
COMPUTER SIMULATION PROGRAM FOR THE ECONOMIC EVALUATION OF THE ACES

4.1 INTRODUCTION

The economic analysis program developed for this study utilizes a comprehensive life cycle cost model. The model outputs two measures of economic effectiveness for each of the heating, ventilating and air conditioning (HVAC) systems being evaluated. These economic measures are annualized cost and payback of incremental costs. These measures are used to determine the economic effectiveness of the Full and Minimum ACES and to define the equipment configuration that will result in the Cost Optimized ACES.

4.2 LIFE CYCLE COST MODEL

The life cycle cost model used in this analysis is based on the approach described in the National Bureau of Standards report, Solar Heating and Cooling in Buildings: Methods of Economic Evaluation (NBSIR-75-712). The analytical basis of the model is illustrated in Figure 4-1. It determines the present value of the following cost items: investment, maintenance and repair, insurance, energy purchases, property tax, income tax deductions from interest payments on borrowed money, energy conservation tax credits, and, where applicable, income tax deductions for operating expenses. Of these eight cost items, the least straightforward to calculate is the investment cost.

If the HVAC system is assumed to be purchased entirely from capital funds, then the equation for computing I, the present value of the total investment cost, is simply:

$$I = IC - SV/(1+d)^N \qquad (4\text{-}1)$$

where d and N have the same definition as in Figure 4-1 and:

$$\text{Present Value} = \underbrace{I}_{\text{Investment}} + \underbrace{M_o \sum_{j=1}^{N} \left(\frac{1}{(1+d)^j}\right)}_{\text{Maintenance and Repair}} + \underbrace{\text{Ins.} \sum_{j=1}^{N} \left(\frac{1}{(1+d)^j}\right)}_{\text{Insurance}}$$

$$+ \underbrace{\sum_{K=\text{fuel types}} \sum_{j=1}^{N} \left(\frac{F_{kj}(1+e_{kj})}{(1+d)^j}\right)}_{\text{Energy Purchases}} + \underbrace{(1-\bar{t}) \sum_{j=1}^{N} \left(\frac{t \cdot G_j}{(1+d)^j}\right)}_{\text{Property Tax}} - \underbrace{\sum_{j=1}^{N} \left(\frac{\bar{t}(L_j \cdot m)}{(1+d)^j}\right)}_{\text{Interest Tax Deduction}}$$

$$- \underbrace{\sum_{j=1}^{N} \left(\frac{\bar{t} \cdot E_j}{(1+d)^j}\right)}_{\text{Operating Expense Deduction}} - \underbrace{\text{ETC}}_{\text{Energy Tax Credit}} \qquad (4\text{-}2)$$

where:

I = total investment cost

M = annual maintenance cost

Ins = annual insurance cost

d = real discount rate

F_{ji} = price of fuel type j at beginning of year i

e_{ji} = real escalation rate for fuel type j during year i

$\bar{t}$ = income tax rate

t = property tax rate

G_j = assessed value of HVAC system in year j, in present dollars

L_j = additional loan principal outstanding in period j which is associated with the HVAC system

m = market rate of interest on the borrowed money

E_j = deductions for depreciation and operating expenses in year j at present prices (only applicable to a commercial situation)

ETC = energy tax credit for installing solar or other non-conventional HVAC system

N = length of the analysis period

Figure 4-1. Life Cycle Cost Model

IC = installed cost of HVAC equipment

SV = salvage value of HVAC equipment at the end of the analysis period

However, if the HVAC system is assumed to be purchased with borrowed funds (the normal situation), then the equation for computing I becomes somewhat more complex. Suppose that the borrowed funds are to be repaid in q equal-sized installments per year over an N-year period. The magnitude of these equal-sized periodic payments (PP) is a function of the length of the repayment period (N), the market interest rate (m), the number of payments per year (q), and the initial amount borrowed (AB).

$$PP = \frac{AB \cdot (m/q)\ (1 + m/q)^{qN}}{(1+m/q)^{qN} - 1} \tag{4-3}$$

The amount borrowed is related to the installed equipment cost by the fractional down payment (FD) as follows:

$$AB = (1 - FD) \cdot IC \tag{4-4}$$

The magnitude of the period payments remains constant but the value of the dollars used to make the payments decreases with time at the rate of inflation (k). Therefore, it is necessary to account for this deflation in value by discounting the payments with inflation. When this discount factor is figured in, the equation for I using a real discount rate becomes:

$$I = PP \cdot q \cdot \sum_{i=1}^{N} \left[\frac{1}{(1 + d)\ (1 + k)}\right]^{i} + FD \cdot IC - SV/(1 + d)^{N} \tag{4-5}$$

Fuel costs are calculated from the hour-by-hour energy use files developed by the simulation module. The hour-by-hour natural gas and oil usages are summed to give a total yearly usage for each of these fuels. This number is

then multiplied by the appropriate 1978 per unit fuel cost to get the first year fuel cost. The total fuel cost is then escalated each year by the appropriate rates and then discounted back to present value.

The hour-by-hour electrical data, total energy and maximum demand, is binned into on-peak and off-peak components on a monthly basis. The appropriate 1978 unit prices are then applied to calculate the monthly demand and energy charges. The monthly charges are summed over the year to calculate the first year electricity cost which is then escalated each year and discounted back to present value.

Maintenance and repair costs are assumed to escalate at the rate of inflation. Another way of stating this is to say that their value, in constant dollars, is assumed to remain the same from year to year. Maintenance and repair costs are assumed to be one percent of the initial installed HVAC system component cost per year. The one percent per year number was drawn from Honeywell's experience with solar system demonstration projects. Since the ACES has a similar level of complexity, the one percent per year figure was adopted for use in this study. However, this rate is only applied to "degradable" components such as solar collectors and heat pumps.

Insurance costs are also assumed to remain fixed in constant dollar terms. Annual insurance costs are calculated as 0.4 percent of the initial installed HVAC system cost. Since the yearly payments are already in constant dollars, all that is necessary to calculate the present value of this cash flow is to discount each annual payment using the real-discount factor.

The yearly property tax payments associated with the HVAC system are based on the assessed value of the system in each year. For the single-family residence, where depreciation deductions are not allowed, the assessed value is set equal to the initial system cost. For the commercial case, where depreciation is allowed, the assessed value in any year is set equal

to the initial system cost minus the total depreciation that had occurred through that year. The property tax effect on total life cycle costs is calculated as the net of property tax payments minus their value as an income tax deduction.

Other income tax deductions that are calculated are those due to:

- Interest payments during each year (residential and commercial)

- Operating expenses (commercial and multi-family residences)
  - maintenance and repair
  - depreciation
  - energy purchases

A one-time 10 percent of purchase price energy conservation tax credit is also calculated for any solar system components used in the ACES. The ice storage tank is not considered to be part of the solar system as it would be present in an ACES that did not have any solar collectors.

ECOMOD, the economic computer program used in this study, calculates the total present value of the cash flows associated with each of the components of total system cost: investment, maintenance and repair, insurance, fuel, income tax deductions for mortgage interest payments, property tax payments, operating expense tax deductions (including depreciation), and energy conservation tax credits.

This information is used to calculate total annual cost, total present value and payback of incremental costs. The Total Annualized Cost is formed by summing the annualized cost contributions from each of the components. Each present value of cash flow is converted to an annualized cost by multiplying it by the capital recovery factor (CRF):

$$CRF = \frac{(1 + d)^{N} \cdot d}{(1 + d)^{N} - 1} \qquad (4\text{-}6)$$

The Total Present Value of Net Benefits is determined by subtracting the total present value of the lifetime costs of owning and operating the study system from the total present value of the lifetime costs of owning and operating the reference system. Finally, the payback of the incremental costs associated with the study system is calculated.

Figure 4-2 portrays a typical net cumulative cash flow history for an investment in energy conserving equipment. At first (Region D), the yearly incremental non-fuel costs exceed the yearly fuel cost savings because of the magnitude of the down payment and the low cost of energy. Then a point is reached (E)* when the fuel cost savings for the year equal the incremental system costs for that year. Eventually, a point is reached when the cumulative fuel cost savings equals the cumulative incremental non-fuel related cash outlays. This point (A)** is the break-even point; from here on the energy conserving system is also generating net savings. Finally, Point B is reached. The cumulative net dollar savings equals the cumulative incremental owning and operating costs remaining to be paid through the end of the analysis period. At Point B*** the cumulative fuel cost savings to that time have exceeded the cumulative value of the total life cycle incremental costs that will have to be paid because the energy conserving equipment, rather than a conventional system, was purchased.

The number of years required to reach Point B of Figure 4-2 was used in this study to define the payback period. The total Present Value of Net Benefits at the end of the analysis period is given by C, the distance from the zero axis to the Total Net Cumulative Savings curve.

*Point E is sometimes called the point of positive savings, where positive savings are defined as the year in which the annual fuel bill of the ACES plus the incremental annual cost of the ACES equals the annual conventional fuel bill.

**The break-even point (Point A) is the year when the net cumulative savings of the ACES are no longer negative. In some analyses, this point is termed the payback.

***Point B is the year in which the cumulative net savings equals the remaining incremental non-fuel costs.

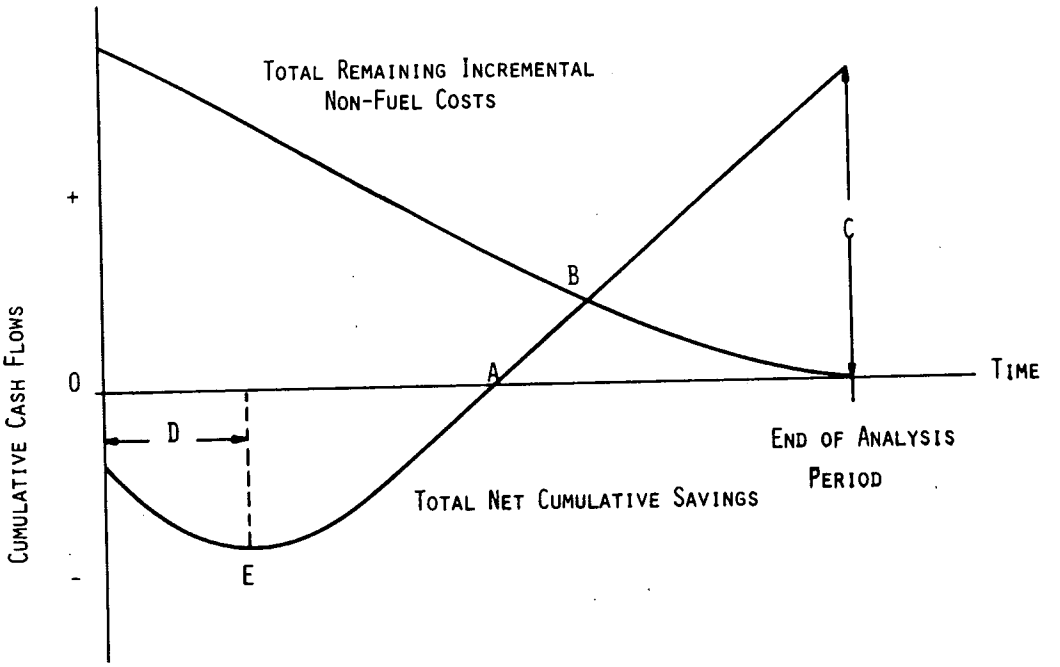


Figure 4-2. Typical Cash Flow History for Energy Conserving Equipment Investment

4.3 ECONOMIC ANALYSIS PROGRAM MODULE

The heart of the Economic Analysis Program Module (ECOMOD) is the Life Cycle Cost Model (LCCM). There are basically five major groupings of user-modifiable inputs:

- Variables needed to develop the multipliers in the LCCM
- Variables describing the price escalation scenarios for gas, oil and electricity
- Variables describing the appropriate electric rate structure
- Component cost information
- Learning curve to be applied to ACES equipment costs

The first group of variables named on the previous page contains the standard information needed to perform a life cycle cost analysis. Thus, it is within the user's power to change the length of the analysis period, the discount rate, property and income tax rates, loan interest rates, type of depreciation schedule and so forth. In addition, it is possible to examine the effects of tax credits and of exempting the solar system portion of the ACES from property tax calculations. Table 4-1 shows an example of the values used for some of these variables in the analysis of the single-family dwelling.

Table 4-1. General Economic Analysis Variables for the Single-Family Residence in Minneapolis

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	30%
INTEREST RATE ON BORROWED MONEY	10.5%
PERCENT DOWN PAYMENT	10%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDIT	10%
PROPERTY TAX RATE	3.71%
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%
DEPRECIATION SCHEDULE	NONE

The second group of variables, the price escalation scenario descriptors, is very important to the analysis as it directly affects the magnitude of the energy cost savings. Provisions have been made which allow the user to input up to four escalation rates and their associated duration of applicability for each of the fuel types being studied. Consequently, more realistic price futures can be described than when the total analysis period must be characterized by a single escalation rate. The importance of this aspect is easily illustrated.

Suppose that a fuel is expected to escalate at a rate of 5 percent per year for the first 10 years and 2 percent per year for the next 10 years. These two rates are equivalent to an average rate of 3.49 percent per year over the 20 year period. However, with a real discount rate of 2.5 percent per year, use of the average fuel price escalation rate would underestimate the total present value of fuel costs by 7.3 percent. Since the incremental cost of any energy conservation measure is to be repaid with energy cost savings, the average escalation rate would result in a longer payback period.

Table 4-2 displays an example of how this time-varying fuel escalation rate feature was used in the Philadelphia single-family dwelling analysis. In all cases, period one starts in 1978. The extra two escalation periods for natural gas prices are explained in Section 4.4.

The third group of variables, the electric rate structure descriptors, is necessary if it is desired to do an equitable analysis of the economic effectiveness of an ACES. Due to its ability to store energy for use later, the ACES has potential for utilizing electricity during times when the utility is experiencing a slump in its daily load curve. In a situation where this off-peak electricity can be purchased at lower costs than during on-peak time, the proper operation of an ACES will result in a much lower electric bill.

Table 4-2. Residential Fuel Price Escalation Rates for Philadelphia

FUEL TYPE	PERIOD 1		PERIOD 2		PERIOD 3		PERIOD 4	
	RATE*	DURATION+	RATE	DURATION	RATE	DURATION	RATE	DURATION
NATURAL GAS	4.361	7	2.444	8	1.647	2	0.768	7
HEATING OIL	3.022	7	0.768	17				
ELECTRICITY	1.136	7	0.152	17				

*ESCALATION RATE IN PERCENT PER YEAR

+NUMBER OF YEARS THAT RATE APPLIES

SOURCE: HISTORIC AND FORECASTED ENERGY PRICES BY DOE REGION AND FUEL TYPE FOR THREE MACROECONOMIC SCENARIOS AND ONE IMPORTED OIL PRICE ESCALATION SCENARIO (AM/EU/79-06) PUBLISHED BY THE ENERGY INFORMATION ADMINISTRATION, NOVEMBER 1978.

Tables 4-3 and 4-4 give examples of the amount of information that goes into entering a time-of-day rate structure in the economic analysis module.

The fourth group of variables, component installed costs, is used to calculate the incremental costs of an ACES over each of the four conventional systems. Table 4-5 presents the equipment costs used for the single-family dwelling analysis in Philadelphia.

The last group of variables is used to describe the 'learning curve' to be applied to the costs of the ACES. That is, at the present, ACES are expensive to install as they are new. Consequently, part of the reason for their high installation costs now is that people are, in essence, being paid to learn how to put the components together, both at the factory and at the site. With time, the best (most cost-effective) procedures for assembling the components will be discovered and the assembly costs will then fall. Additionally, economic theory says that it should cost less per unit when manufacturing

Table 4-3. Time-of-Day Rate Structure Used in Analysis of Residential ACES in Philadelphia

HOURS AND SEASONS (PER KWH)	SMALL COMMERCIAL	RESIDENTIAL ELECTRIC HEATING
OFF-PEAK ALL YEAR	2.8¢	2.8¢
ON-PEAK WINTER PERIOD	6.9¢	5.5¢
ON-PEAK SUMMER PERIOD	15.9¢	13.4¢
PLUS CUSTOMER CHARGE (PER MONTH)	$13.00	$14.00

Table 4-4. Time-of-Day Rate Structure Used in Analysis of Residential ACES in Philadelphia

HOURS AND SEASONS (PER KWH)	SMALL COMMERCIAL	RESIDENTIAL ELECTRIC HEATING
OFF-PEAK (WEEKLY)	10PM - 8 AM MON. - FRI. AND ALL DAY SAT. & SUN.	10PM - 8AM MON. - FRI. AND ALL DAY SAT. & SUN.
ON-PEAK (WEEKLY)	8AM - 10PM MON. - FRI.	8AM - 10PM MON. - FRI.
WINTER	OCT. 16 - MAY 14	OCT. 16 - MAY 14
SUMMER	MAY 15 - OCT. 15	MAY 15 - OCT. 15

Table 4-5. Equipment Costs for Philadelphia Single-Family Dwelling

CONVENTIONAL HEATING PLANT (G, O, E)	$660 $850 $690
CONVENTIONAL COOLING PLANT	$850
HIGH-EFFICIENCY AIR-TO-AIR HEAT PUMP	$2606
CONVENTIONAL DOMESTIC HOT WATER PLANT (G, E)	$182 $170
ICE-MAKER HEAT PUMP	$4472
SOLAR SYSTEM	$2143 + $19.65/FT^2 or ($ 9.50/FT^2)
ICE TANK	$0.24/FT^2 FOR LINER
	$4.18/FT^2 FOR WALLS
	$2.22/FT^2 FOR SLAB
	$3.41/FT^2 FOR COVER
	$5.63/FT FOR FTG. WHERE NEEDED
	$0.54/FT^2 FOR INSULATION

1000 units per year than when manufacturing 10 units per year of a product. The 'learning curve' allows the user to input his estimate of how fast and by how much these factors will tend to decrease the cost of installing an ACES.

For our analysis, the residential learning curve, based on historical and projected data from the HVAC industry, assumed that by 1980 an identically sized ACES would cost only 68 percent as much as one installed in 1978 (in constant dollars) and that one installed in 1982 would cost only 48 percent as much as its 1978 equal.

The commercial learning curve was considerably slower: a 5 percent decrease by 1980 and a 15 percent decrease by 1982. The reason for this slower curve is that commercial buildings are, by their nature, a custom construction job. Therefore, many of the advantages of equipment standardization are lost.

4.4 SOURCES OF DATA

In order to make use of ECOMOD, it is necessary to supply the input data discussed in the preceding section. To recapitulate, the user must supply six sets of data.

The values of the first set of variables were set based on common sense, past experience, current market rates on borrowed money, the Revenue Act of 1978, and the Energy Tax Act of 1978. The personal income tax bracket was set at 30 percent.

The principal determinant of the economic effectiveness of the additional costs incurred in the purchase and operation of an energy efficient system as opposed to a conventional system is the magnitude of the yearly energy cost savings relative to the magnitude of the incremental costs. Over the analysis period, the magnitude of energy cost savings are determined by the energy price escalation rates employed. For this study, the escalation rates (and the current prices for natural gas and oil) were taken from a November, 1978, analysis memorandum titled, "Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06)" published by the Energy Information Administration (EIA). This study used the mid-range geology results from the TRENDLONG model. Prices were escalated to 1978 dollars using the appropriate Consumer Price Index deflators reported in the memorandum. The EIA analysis only extends out to 1990. Since the ACES evaluation requires price data out to 2002, the 1985-1990 escalation

rates developed by the EIA were simply assumed to continue into the future. The one exception to this statement concerns the price future of natural gas. Natural gas has high expected escalation rates between now and 1990. However, it is unrealistic to assume that these high rates will continue through the year 2000. In fact, the usual assumption is that natural gas will escalate in price up to a level where oil becomes an economically attractive substitute and that, from then on, its price will track changes in the price of oil. Therefore, the escalation rate for natural gas after 1990 was set equal to the corresponding escalation rate for oil in the 1985-2000 period.

Although ECOMOD has provisions for both standard rate structures and Time-of-Day (TOD) rate structures, the study concentrated on analysis utilizing TOD rates. This attitude made it possible to realize the economic benefits of the ACES ability to load shift. The following utility rate structures were used as typical for the area in which the study city was located.

- For Minneapolis, the TOD rate structures developed by the Northern States Power Company (NSP) for the Minnesota Public Service Commission.

- For Atlanta, the current rate structures of Georgia Power and Light modified to develop a TOD rate using the same relationship between current and TOD prices as exists in the NSP rates.

- For Philadelphia, the experimental TOD rates now available from Consolidated Edison Electric Company.

The installed costs for the various equipment items were developed from a variety of sources. Air-to-air heat pump costs were based on actual manufacturer prices. Ice-maker heat pump costs were developed from data supplied by Harry C. Fischer. Tank costs were taken from the Irwin G. Cantor report and

unit costs were based on the Type-5 concrete tank (a mid-price construction). Unit tank costs used in this study were:

$0.25 per square foot for the tank liner
$4.29 per square foot for wall construction
$2.33 per square foot for the slab
$3.59 per square foot for the cover
$5.95 per foot for ftg. where needed
$0.54 per square foot for additional insulation to R40

The fixed costs associated with solar systems were based on Honeywell's experience from such programs as the Solar Heating and Cooling Systems Design and Development contract to NASA's Marshall Space Flight Center. Installed costs for flat plate collectors were derived from the same sources; those for the absorbers were developed from Fafco data. The non-solar energy transport module fixed costs were based on analysis performed in the course of Honeywell's Solar Assisted Heat Pump study for Lennox Industries. The installed costs of conventional system components were developed using Mean's Building Construction Cost Data.

The learning curve was used to investigate the effects of decreases in the installed cost of the ACES. The percentage decreases were developed from historical data supplied by various HVAC component manufacturers. The hour-by-hour energy use data was generated by the appropriate simulation package.

SECTION 5.0
RESULTS

Figure 5-1 shows the annual response of the tank temperature and ice inventory of the Full ACES for the single-family residence in Minneapolis. At the beginning of the heating season (Day 260) there is a large net flux of heat into the storage tank; thus, the storage temperature rises rapidly. Once the tank temperature reaches 42°F (the highest temperature at which the cooling coil can deliver maximum hourly fall cooling load) the collectors are set to the modulating mode. This mode of operation continues until there is no longer a fall cooling load (Day 280-300). After this period the collectors operate whenever there is available energy. During the end of fall (Day 300-320) sensible energy is stored within the tank, thereby raising its temperature (also increasing heat pump COP).

At approximately Day 330 there has been a cummulative net decrease of energy to the tank and therefore a net increase in the ice inventory. At Day 57 the ice within the tank reaches the maximum allowable volume. At this time it is possible to again have a net positive heat flux to the tank (if the collectors are left ON). This trend is shown by the decrease in ice inventory (from Day 57-81). If the collectors were left ON for the remainder of the heating season all the ice would be depleted from the tank. Therefore, after Day 81 the collectors are run in the modulating mode to charge the tank for summer operation. Finally the collectors are turned OFF at Day 150 (beginning of the cooling season).

At the onset of the cooling season the tank contains the maximum possible amount of ice. This ice is used throughout the cooling season to meet instantaneous loads. At the end of the cooling season the ice is completely depleted and has exactly satisfied the required cooling load, thereby maximizing interseasonal energy transfer.

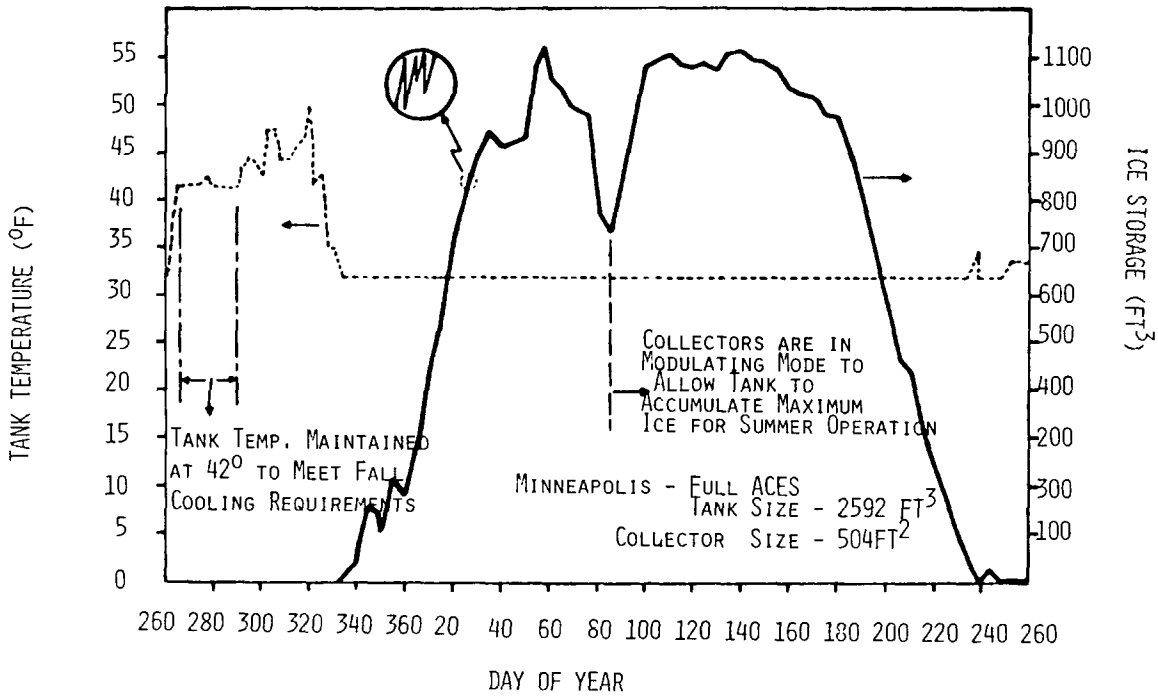


Figure 5-1. Annual Response of Tank Temperature and Ice Inventory for the Minneapolis Single-Family Full ACES

Figure 5-2 shows the annual response of the tank temperature and ice inventory of the Minimum ACES for the single-family residence in Minneapolis. The results depicted in this figure are similar to those in Figure 5-1 except that the gradients (rate of change of ice and tank temperature) for the Minimum ACES are considerably steeper; for example, the ice inventory is completely depleted from the Minimum ACES tank during the middle of the heating season. One distinct difference between the two systems is that the heat pump must run at night during the summer in the Minimum ACES since there is not enough stored ice in the tank to meet summer cooling requirements.

These two figures show basically how the Full and Minimum ACES respond to a given set of external stimuli. To determine the most cost-effective system this study not only investigates these two systems (which may or may not be the most cost-effective) but also performs a systematic evaluation of other

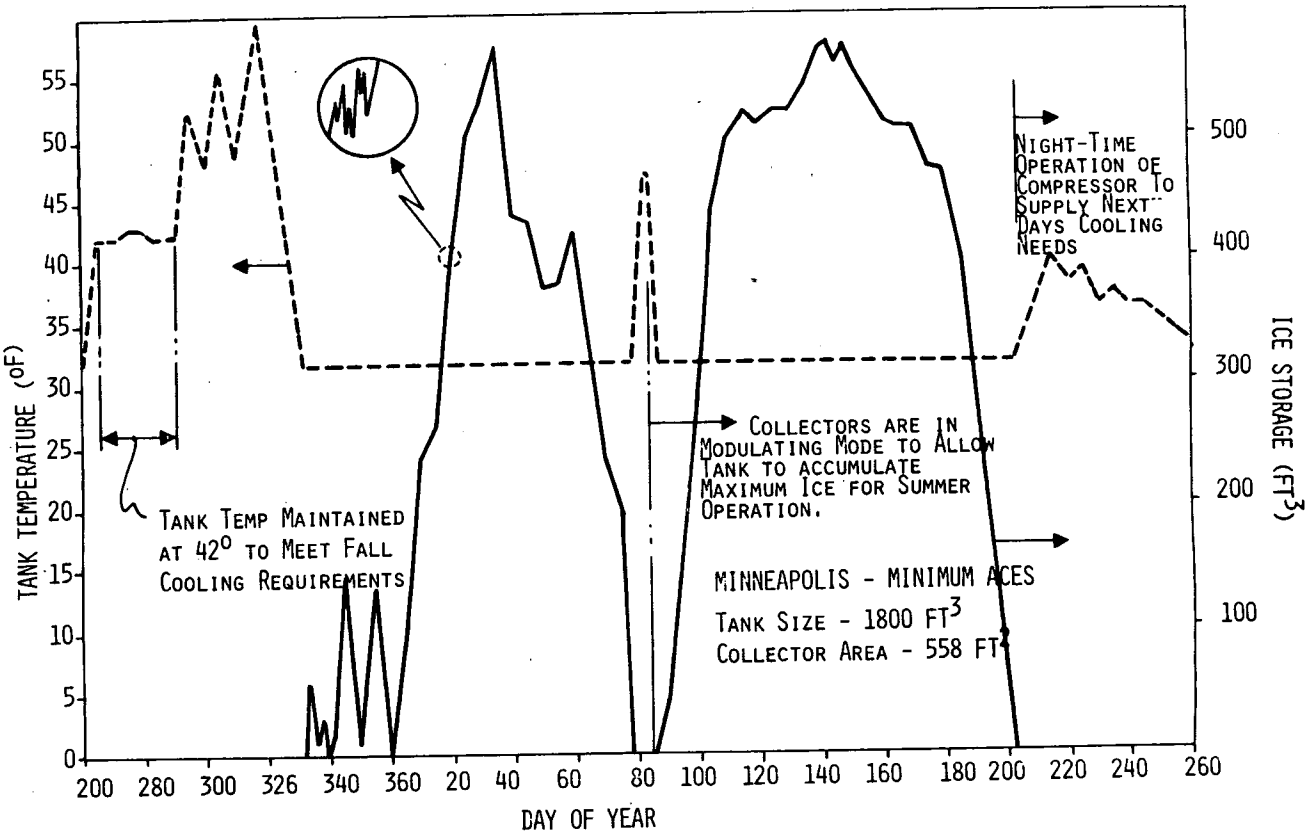


Figure 5-2. Annual response of Tank Temperature and Ice Inventory for the Minneapolis Single-Family Minimum ACES

possible ACES configurations. The systematic evaluation is performed by parametrically varying each independent variable (component), including:

- Tank Volume.
- Collector Area.
- Supplemental Heat (Supplemental heat or backup energy is defined as the electric resistance energy needed to supplement the operation of the ACES).
- Changeover Temperature (Temperatures above which the airside evaporator will be used instead of the evaporator ice plates).

- Tank Insulation (The residential tank is assumed to be located outside and any surface exposed to the air is insulated to R40. The portion of the tank below ground level will be assigned an insulation value of R40, or the tank will be considered uninsulated. The commercial tanks are assumed to be located beneath the building and all sides of the tanks are either insulated to R10 or to R40).
- Start Date (Time when system is installed--capital costs are assumed to be a function of the installation time and the number of systems installed as explained in Section 4.0).

Although these parameters are defined as independent, Figure 5-3 shows that, in fact, collector area is a dependent variable for the Full ACES (it is also a dependent variable for the Minimum ACES). That is, for a given tank size there exists one and only one collector area capable of optimally satisfying the Full and Minimum ACES constraints given in Section 1.0.

By adding extra degrees of freedom (for example, adding the capability of using backup energy or an airside evaporator), the tank volume and collector area become independent variables. This addition of extra variables allows the Cost Optimized ACES to be more flexible in meeting the building loads than the Full or Minimum ACES.

One other variable that can have a profound effect on the economic feasibility of the system is the collector type (absorber, single-glazed, or double-glazed collector). Although this parameter is not treated as an independent variable (it is not necessarily system-specific, but rather site-specific), it has been varied for each location to determine the minimum cost collector system.

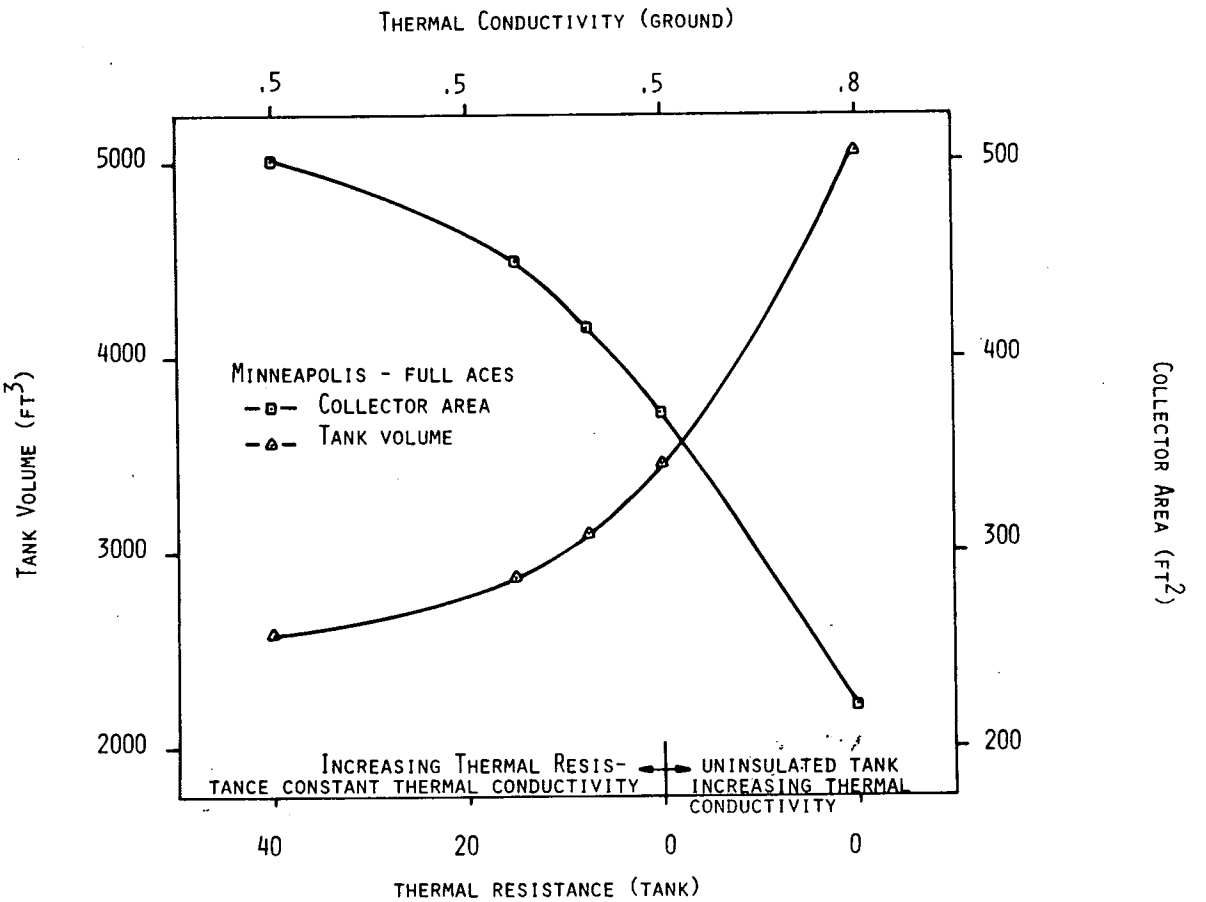


Figure 5-3. Required Tank Volume and Collector Area As A Function of Tank Insulation and Ground Conductivity for the Single-Family Full ACES in Minneapolis

5.1 MINNEAPOLIS RESULTS

Figure 5-4 shows the effect of the type of collector system on the required collector area and on the collector purchase price for the single-family Full ACES in Minneapolis. Increasing the number of covers from zero (absorber) to one (single-glazed system) decreases the required collector area as well as the total purchase price. Increasing the number of covers from one to two (double-glazed system) actually increases the required collector area slightly and increases the total purchase price of the collector system. Therefore, a single-glazed system is best suited for the operation of the ACES in Minneapolis.

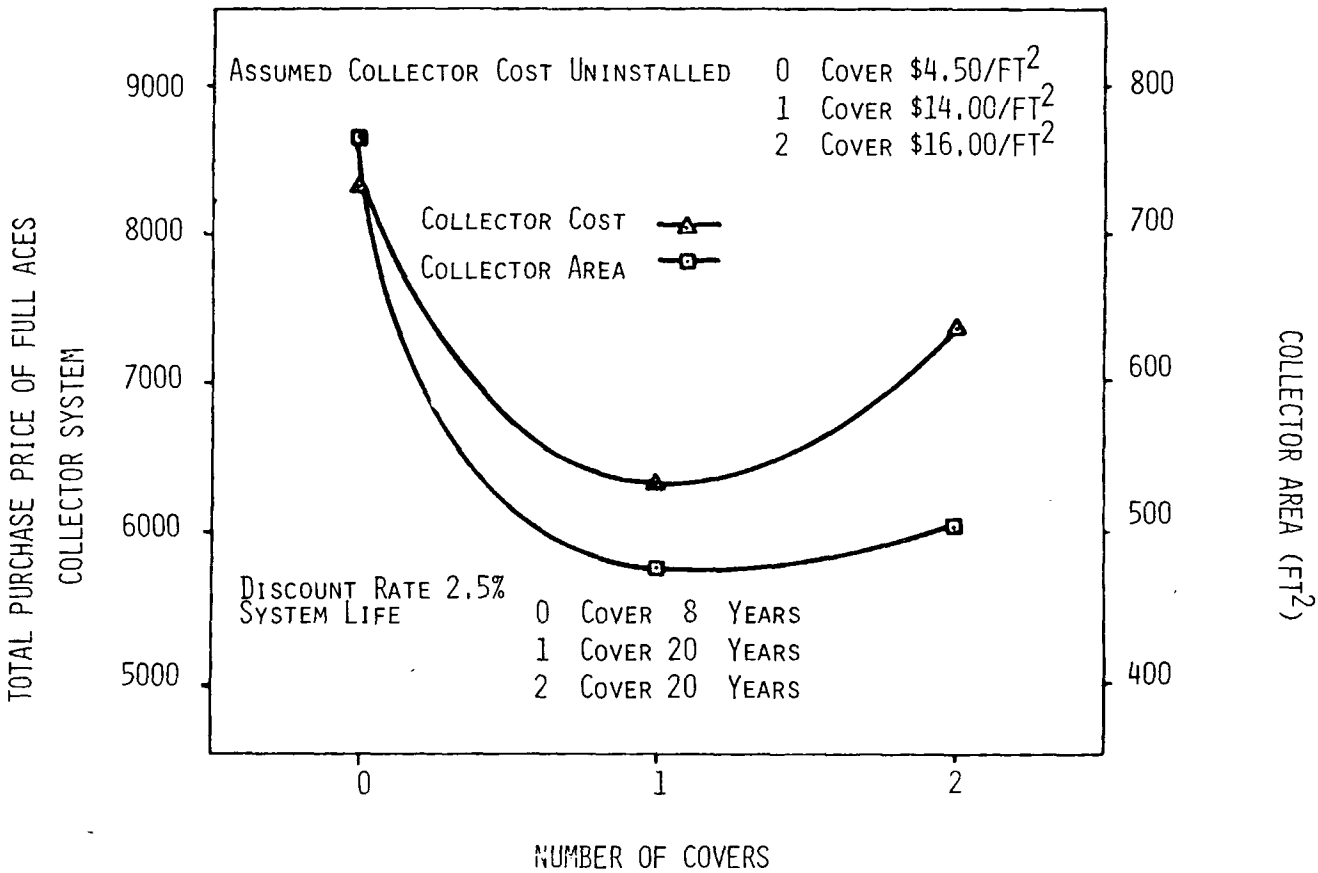


Figure 5-4. Area and Purchase Price of the Installed Collector System as a Function of the Number of Collector Covers for the Single-Family Full ACES in Minneapolis

5.1.1 Results For The Single-Family Residence

Table 5-1 gives the heating, cooling and hot water load profiles for the single-family residence located in Minneapolis (this residence is fully described in Appendix E). The annual heating, cooling and hot water loads are 82, 12, and 22 million BTUs/year respectively. Also shown in this table are the peak heating and cooling loads, the design load, and the steady state performance data of the ACES component train. As was described in Section 2.0, all load information generated in LODMOD (computer load module) is input

Table 5-1. Load and ACES Equipment Performance Data for the Minneapolis Single-Family Residence

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	196	0	19
FEBRUARY	137	0	18
MARCH	105	0	20
APRIL	52	5	20
MAY	23	6	20
JUNE	1	15	18
JULY	0	49	18
AUGUST	1	33	17
SEPTEMBER	10	8	16
OCTOBER	45	3	17
NOVEMBER	107	0	17
DECEMBER	138	0	18

PEAK HEATING LOAD 41,572 BTU/HR

PEAK COOLING LOAD 28,739 BTU/HR

DESIGN HEATING LOAD 50,000 BTU/HR

CAPACITY OF WATER-TO-AIR HEAT PUMP 50,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE; 70/67° ENTERING AIR TEMPERATURE

INFILTRATION (CFM/FT^2) .12 HOUSE .1 ATTIC

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.55

HEAT PUMP BLOWER POWER 1079 WATTS

COOLING COIL BLOWER POWER 300 WATTS

EVAPORATOR PUMP POWER 100 WATTS

COOLING LOOP PUMP POWER 100 WATTS

HOT WATER LOOP PUMP POWER 100 WATTS

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

into DESMOD (computer sizing module for the Full Minimum ACES). The information generated in these first two modules is then input into SIMMOD (computer simulation module) from which the hourly energy use profile of the HVAC system to be analyzed is generated. All of this information is then input into the final computer program ECOMOD for the economic evaluation of the ACES. Also input into ECOMOD are the economic variables described in Section 4.0. Table 5-2 gives the values of the general economic variables for the single-family residence in Minneapolis.

Table 5-2. General Economic Analysis Variables for the Minneapolis Single-Family Residence

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	30%
INTEREST RATE ON BORROWED MONEY	10.5%
PERCENT DOWN PAYMENT	10%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDIT	10%
PROPERTY TAX RATE	3.71%
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%
DEPRECIATION SCHEDULE	NONE

Table 5-3 shows the unit component costs for the ACES and the four conventional HVAC systems. The final economic information needed to evaluate the ACES and conventional systems can be found in Tables 5-4 and 5-5. These tables contain the residential time-of-day rate structures, base prices and escalation rates applied to oil, gas and electricity.

With all input defined it is then possible to perform a parametric evaluation of the ACES. The parameters used in this evaluation and the ranges over which they are to be varied are shown below for the Minneapolis single-family residence.

Tank	1800 ft^3, 2048 ft^3, 2592 ft^3, 2888 ft^2
Collector	558 ft^2, 504 ft^2, 350 ft^2, 223 ft^2, 0 ft^2
Changeover Temperature	71^0F, 40^0F, 30^0F, 25^0F, 20^0F, 15^0F, 10^0F, -10^0F
Tank Insulation	top: R40
	below ground: uninsulated, R40
Supplemental Heat	when needed
Costs	1978 prices, 1982 prices

Figure 5-5 illustrates the annual energy consumption of the single-family residence located in Minneapolis. The results are depicted as a function of tank size, tank insulation, collector area, and changeover temperature. For clarity, only three changeover temperature scenarios have been plotted. This figure shows that for the case of no airside evaporator (changeover temperature = 71^0F) and a given tank size, a decrease in the collector area results in an increase in energy consumption. This increase in energy consumption is due to the use of supplemental heat.

For the same case (no airside evaporator), increasing the tank size for a given collector area results in a decrease in the required energy input. This decrease is due to a reduction in the use of supplemental heat, an increase in the intraseasonal energy transfer, and an increase in the ground heat flux to the tank.

Table 5-3. Economic Input Data: HVAC Installed Costs for the Minneapolis Single-Family Residence

COMPONENT	COST (DOLLARS)
HEATING PLANT	
NATURAL GAS	820
OIL	730
ELECTRIC	840
CENTRAL AIR CONDITIONER	1120
HOT WATER HEATER	
NATURAL GAS	182
ELECTRIC	170
HIGH EFFICIENCY AIR-AIR HEAT PUMP	3436
ICE-MAKER HEAT PUMP	
DUAL SOURCE	4804
WATER SOURCE	4200
SOLAR COSTS - FIXED + VARIABLE	1580 + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	563
ACES CONTROLS	1150
SUPPLEMENTAL ELECTRIC HEAT	300
ICE TANK	.26/FT^2 FOR LINER, 4.31/FT^2 FOR WALLS 2.35/FT^2 FOR SLAB, 3.66/FT^2 FOR COVER 5.95/FT FOR FTG. WHERE NEEDED .54/FT^2 FOR INSULATION

*Price for absorber with a 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5-4. Time-of-Day Service Rates for the Minneapolis Single-Family Residence

	OCT.-MAY	JUNE-SEPT.
CUSTOMER CHARGE PER MONTH	$5.50	$5.50
ON-PEAK ENERGY CHARGE ($/KWH)	0.0604	0.0743
OFF-PEAK ENERGY CHARGE ($/KWH)	0.0214	0.0214
ON-PEAK HOURS	9:00 AM - 9:00 PM Monday - Friday	

NOTE: Rates are taken from the Northern States Power (NSP) Company's proposed TOD Rate design presented to the Minnesota Public Service Commission in August, 1978.

Energy consumption for the 20^{0}F changeover temperature scenario can be separated into two general catagories. The first category is for all tank sizes at collector areas greater than 400 ft^2. Here the energy consumption is greater for the 20^{0}F changeover temperature than it was for the 71^{0}F changeover temperature. The reason for this increase in energy consumption is that the system utilizing a 20^{0}F changeover temperature suffers from the use of an outside air evaporator. The second category is for all tank sizes at collector areas under 400 ft^2. In this case, the energy consumption is much less for the 20^{0}F changeover temperature than it was for the 71^{0}F changeover temperature. The reason for this reduction in energy consumption is that it is more efficient to supply energy with an airside evaporator than to supply energy with supplemental heat. In this same category a reduction in collector area results in an increase in energy consumption for all tank sizes. This increase in energy consumption is due to the use of supplemental heat.

Table 5-5. Fuel Price and Escalation Rates for the Minneapolis Single-Family Residence

	GAS	OIL	ELECTRICITY
BASE PRICE	$2.12/MCF	46.3¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	5.610	2.28	.600
PERIOD 2			
DURATION	5	17	17
RATE	5.000	.900	.500
PERIOD 3			
DURATION	2	-	-
RATE	2.170	-	-
PERIOD 4			
DURATION	10	-	-
RATE	.910	-	-

+ Escalation rate in percent per year
* Number of years that rate applies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

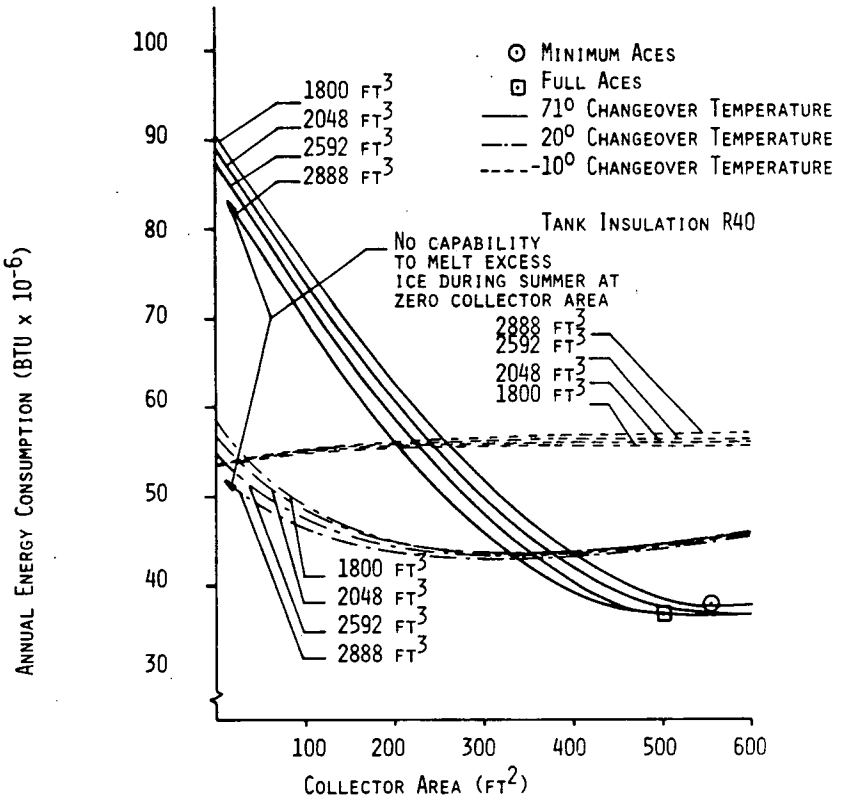


Figure 5-5. Annual Energy Consumption as a Function of Tank Size, Collector Area and Changeover Temperature for the Minneapolis Single-Family Residence.

The final set of curves illustrates the energy requirements of the ACES as a function of tank size and collector area for the -10°F changeover temperature. The system that utilizes the -10°F changeover temperature consumes significantly more energy than the system that utilizes the 20°F changeover temperature. This increase in energy consumption is caused by decreased system efficiency encountered at reduced evaporator temperatures. An increase in collector area results in a slight increase in energy consumption for all tank sizes. The reason for this increase in energy consumption is that the system utilizing a -10°F changeover temperature essentially operates as an air-to-air heat pump during the winter. Thus, any energy collected by the panels must be thermodynamically degraded for the ACES to operate in the summer cooling mode.

Each point represented in Figure 5-5 has a number of annualized costs and paybacks associated with it. For example, Figure 5-6 shows the annualized cost of the Minneapolis single-family residence for a 1978 installation. The annualized costs are a function of tank size and collector area. These costs represent a system that utilizes a 71°F changeover temperature and a well-insulated tank (R40). The general trend illustrates that annualized costs decrease as tank size and collector area decrease. Thus, it is more cost-effective to have a less energy-efficient system and to defray initial capital cost than to have a more expensive system and save energy. This result is simply a tradeoff between all energy and capital costs evaluated

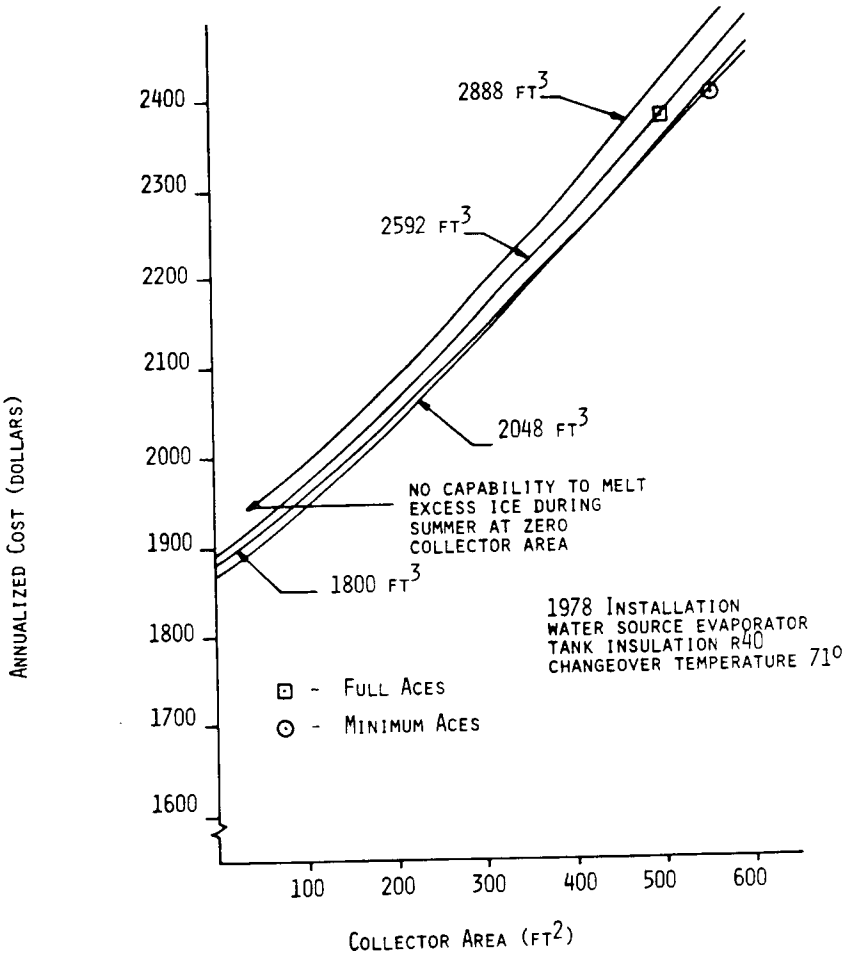


Figure 5-6. Annualized Cost as a Function of Collector Area and Tank Size for the Minneapolis Single-Family Residence (R40 Tank Insulation, 71°F Changeover Temperature, 1978 Installation)

over the system lifetime. Figure 5-7 shows that these trends are not true for the 1982 installation (assuming initial capital costs are reduced as discussed in Section 4.0). In this figure the annualized costs are also a function of collector area and tank size. For a given tank size a decrease in collector area initially results in a decrease in annualized cost; however, there is a point beyond which a further reduction in collector area results in an increase in annualized cost. A nonzero optimum collector area thus exists for a specified tank. If the locus of minimums for each tank size were plotted in a similar fashion, a tank size could be found for

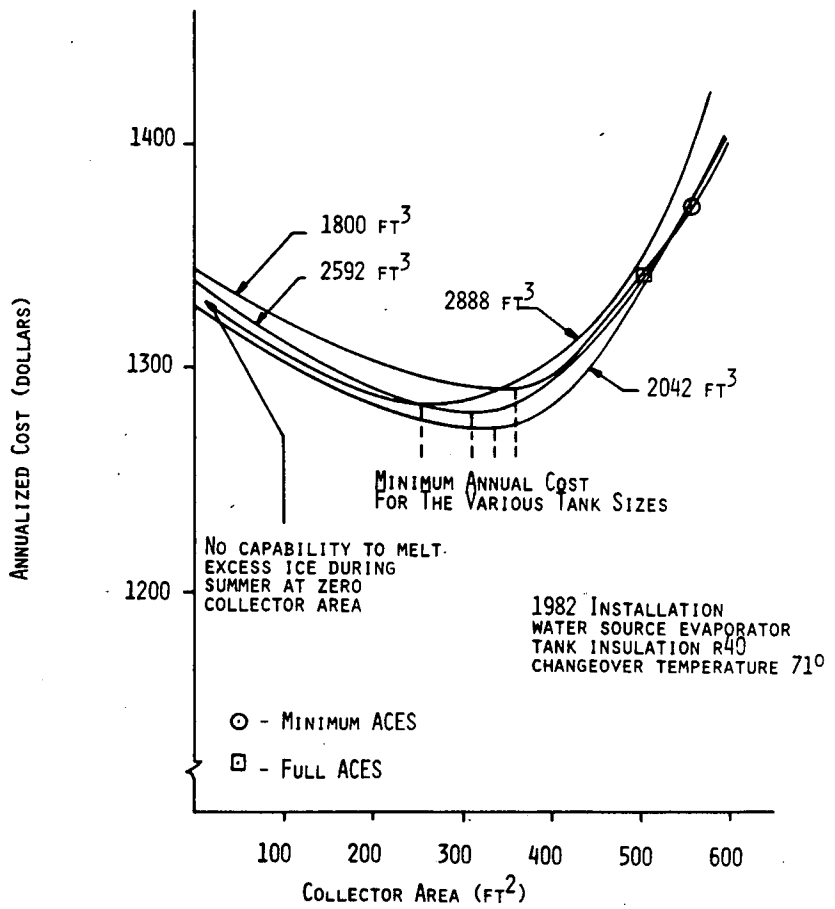


Figure 5-7. Annualized Cost as a Function of Collector Area and Tank Size for the Minneapolis Single-Family Residence (R40 Tank Insulation, 71°F Changeover Temperature, 1982 Installation)

which the annualized cost is minimized. Therefore a set of components (tank size and collector area) can be found for which the annualized costs are minimized. Unlike the 1978 installation, it now becomes worthwhile to invest in a more expensive system and to offset capital expenditures with the savings obtained by using the more energy efficient system.

Figure 5-8 shows the annualized cost of the Minneapolis single-family residence for a 1978 installation. The annualized costs are a function of tank size and collector area. Here the costs represent a system that utilizes 20°F changeover temperature and a well-insulated tank (R40). The general

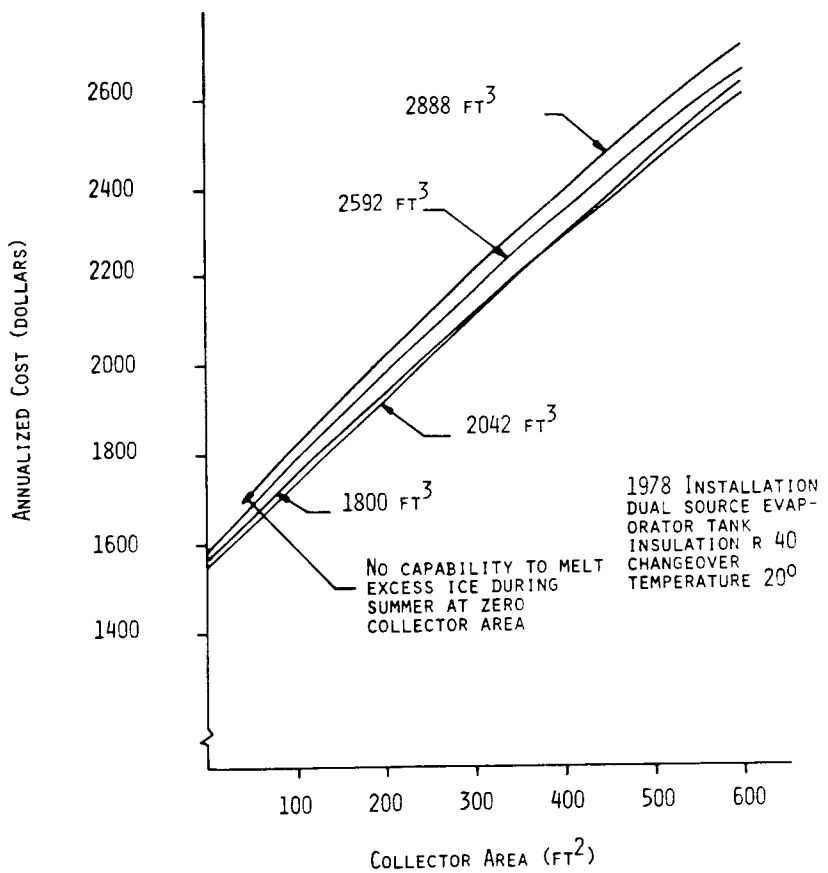


Figure 5-8. Annualized Cost as a Function of Collector Area and Tank Size for the Minneapolis Single-Family Residence (R40 Tank Insulation, 20°F Changeover Temperature, 1978 Installation)

trends are the same as they were in Figure 5-5; however, the magnitude of the annualized cost for the respective cost minimized systems has been greatly reduced. Figure 5-9 shows annualized cost as a function of tank size and collector area but assumes a 1982 installation. In this case the general trend shows a reduction in annualized cost as the collector area is reduced. For the case of zero collector area (minimum annualized cost for each tank size) a tank size between 1800 ft^3 and 2880 ft^3 exhibits a minimum annualized cost.

Results similar to those shown in Figures 5-6 through 5-9 could be presented for each possible changeover temperature. However, a more illustrative

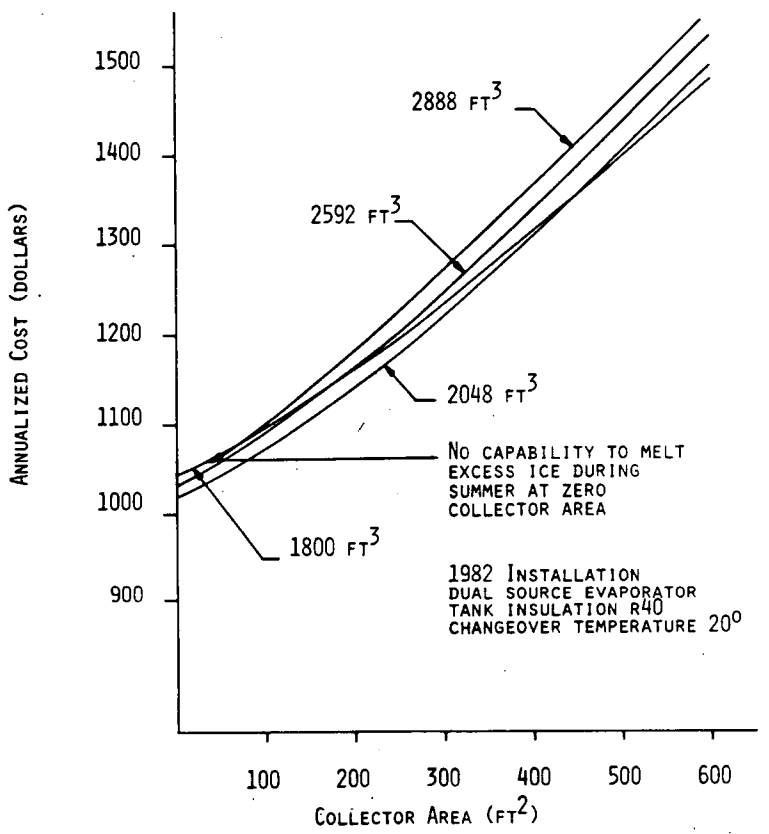


Figure 5-9. Annualized Cost as a Function of Collector Area and Tank Size for the Minneapolis Single-Family Residence (R40 Tank Insulation, 20°F Changeover Temperature, 1982 Installation

method for presenting the data is to plot the locus of minimums of the annualized costs as a function of the changeover temperature. Figure 5-10 is such a plot for the Minneapolis single-family residence for a 1978 installation. This figure shows that besides having a possible minimum in the annualized cost for tank size and collector area there can also be a minimum in the annualized cost (or cost minimum) for the changeover temperature. The minimum cost system for the single-family residence in Minneapolis with an insulated tank consists of a 2048 ft^3 storage tank, a dual source evaporator (with a 13°F changeover temperature), no solar collectors and the capability of supplying supplemental heat. Figure 5-11 shows a similar set of curves using data for a 1982 installation. The trends are similar to those in Figure 5-9 except that annualized costs are greatly reduced.

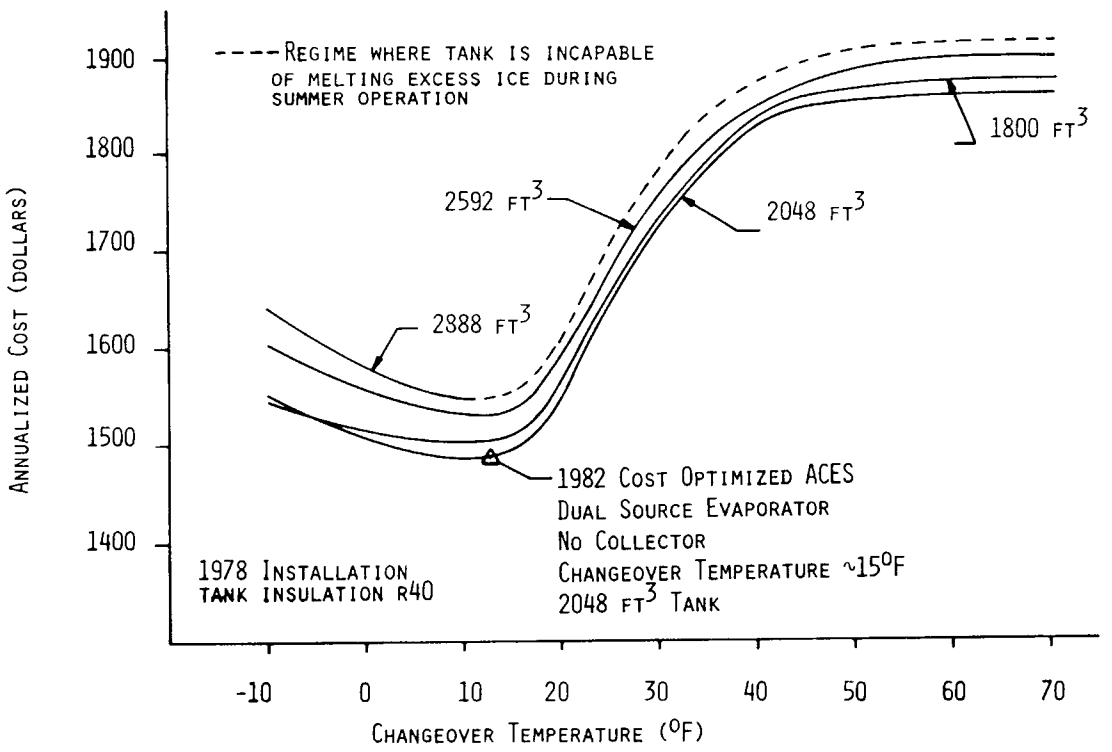


Figure 5-10. Annualized Cost as a Function of Changeover Temperature and Tank Size for the Single-Family Residence in Minneapolis (R40 Tank Insulation, 1978 Installation)

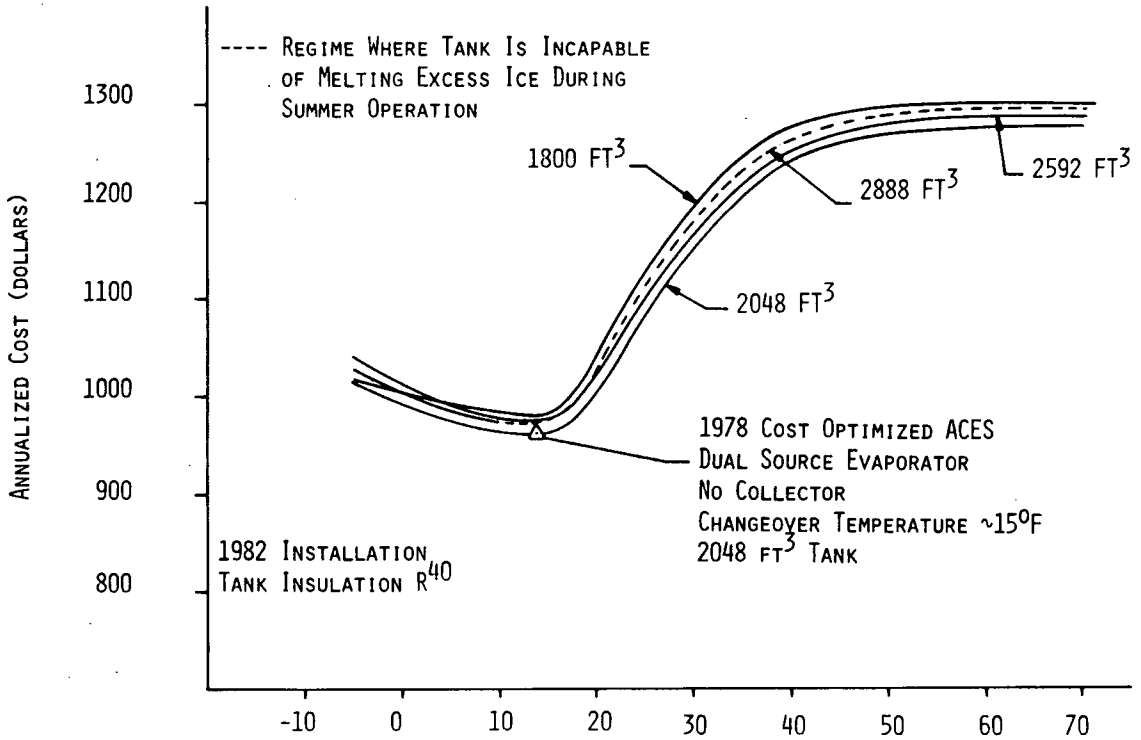


Figure 5-11. Annualized Cost as a Function of Changeover Temperature and Tank Size for the Single-Family Residence in Minneapolis (R40 Tank Insulation, 1982 Installation)

Corresponding to each annualized cost is a system payback. To avoid presenting the payback information in a similarly lengthy fashion, the data has been condensed by plotting the locus of minimums of each payback as a function of the changeover temperature. This data is shown in Figure 5-12 for a well-insulated tank (R40) assuming a 1982 installation. Here the payback is given in years and is calculated with respect to the four conventional systems. There is a distinct minimum in all the payback curves at a changeover temperature of approximately 15°F. It can be seen that the ACES is cost-competitive with the electric, oil and heat pump conventional systems. The same information presented for the 1978 installation would show that the ACES is not cost-competitive with any of the conventional systems.

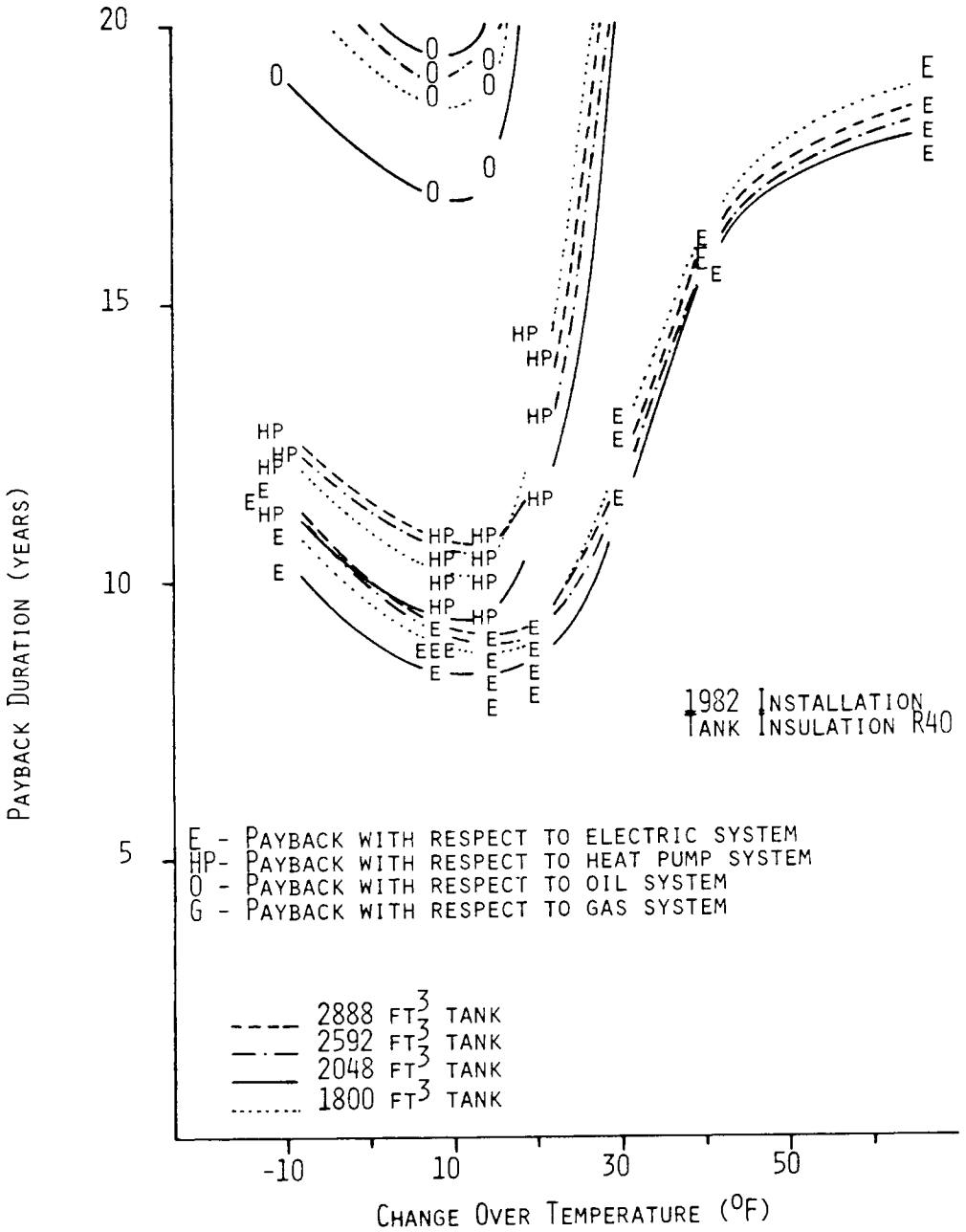


Figure 5-12. Payback as a Function of Changeover Temperature and Tank Size for the Minneapolis Single-Family Residence (R40 Tank Insulation, 1982 Installation)

The figures presented so far have only given information on the payback and annualized cost of the single-family residence in Minneapolis for the well-insulated tank (R40). Thus the cost optimized system and its respective payback have still not been determined. To find the cost optimized system the same amount of information needs to be generated for each parameter variation in tank insulation. The next step is to continue the parametric evaluation with an uninsulated tank.

Since there is a great deal of data generated for each parameter variation and since the basic trends and interactions between system components and system costs have been briefly illustrated, the remainder of the data generated in this study is condensed and only highlights are presented. For the case of the uninsulated tank in Minneapolis the energy trends are basically the same as for the insulated tank, the difference being that in general the system with the uninsulated tank uses less energy than the system with the insulated tank. The annual costs reflect this change in energy consumption as well as the reduction in tank costs because it was not insulated.

A comparison between the tank size and collector area for the Full and Minimum ACES (shown in Table 5-6) illustrates that as the heating load becomes large and the cooling load becomes small (as in the case for the Minneapolis single-family residence), the size of the components of the Full and Minimum ACES approach one another. There is also a corresponding approach in the energy effectiveness of the Full and Minimum ACES. These trends imply that the overall performance of the ACES decreases with an increasing severity in the climate.

Table 5-7 lists the installed cost of the Full, Minimum and Cost Optimized ACES as well as the installed costs of the four conventional systems. These costs have been catagorized by system component. Clearly the solar collectors are the most capital-intensive component while the tank and

Table 5-6. Alternative ACES Configuration for the Minneapolis Single-Family Residence

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	
			1978	1982
TANK INSULATION	R40	R40	UNINSULATED	UNINSULATED
TANK SIZE (FT^3)	2592	1800	2048	2048
COLLECTOR AREA (FT^2)	504	558	NONE	NONE
COLLECTOR TYPE	SINGLE GLAZE	SINGLE GLAZE	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	DUAL SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	15°	15°
ELECTRIC BACKUP	NONE	NONE	YES	YES

Table 5-7. Comparison of Initial Capital Costs (1978 Dollars) of the ACES and Conventional Systems for the Minneapolis Single-Family Residence

SYSTEM / COMPONENT	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	GAS	OIL	ELECTRIC	HEAT PUMP
HEATING PLANT[1]	4200	4200	4802	820	730	840	3436
COOLING PLANT[1]	-	-	-	1120	1120	1120	-
HOT WATER HEATER	-	-	-	182	170	170	170
SOLAR COSTS	12035	13095	-	-	-	-	-
ACES CONTROLS	1150	1150	1150	-	-	-	-
TANK	5135	4355	4145*	-	-	-	-
TOTAL	22520	22800	10097	2122	2020	2130	3606

1 Conventional system costs include associated controls
* Cost includes auxiliary heaters

heat pump are the second most capital-intensive items. All ACES components are significantly more expensive than conventional system components (with the exception of the air-to-air heat pump).

Table 5-8 lists all costs associated with the Minneapolis single-family ACES as well as all costs associated with conventional systems. The conventional systems cost between $2,000 and $3,600 whereas Full and Minimum ACES cost between $22,000 and $23,000 for the 1978 installation. It is this amortized dollar difference that must be offset by the savings obtained by using the ACES. The 1978 annualized energy and total costs show that even though energy savings are large, the ACES are not cost-competitive with conventional systems. The corresponding 1982 prices show that although the Full and Minimum ACES are only marginally competitive with conventional systems, the Cost Optimized ACES is superior to all systems except gas. The COP* column in this table shows that the Full ACES is the most energy efficient system (maximizes interseasonal energy transfer), while the minimum ACES is the second most energy efficient system (only slightly less than the Full ACES due to the load relationship in Minneapolis). Comparing the COP of the ACES with conventional systems shows that ACES are far superior to all conventional systems.

*COP is defined here as the total energy delivered for heating, cooling and hot water divided by the total energy consumed (including parasitics) by the HVAC/DHW system.

Table 5-8. Economic Characteristics of HVAC/DHW Systems for the Minneapolis Single-Family Residence

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS)		ANNUALIZED COST (DOLLARS)				YEARLY ENERGY USE (BTU X 10^{-6})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC[1]	2122	2122	907	887	720	718	7.81	156.65	0.70
OIL/ELECTRIC[2]	2020	2020	1084	1011	906	847	29.69	112.19	0.81
AIR-TO-AIR HEAT PUMP[2]	3606	3606	1233	1134	915	847	75.19	0.0	1.53
ELECTRIC/ELECTRIC[2]	2130	2130	1453	1341	1266	1171	109.12	0.0	1.05
FULL ACES[3]	22520	10813	2372	1330	462	427	36.00	0.0	3.20
MINIMUM ACES[3]	22800	10944	2408	1353	466	431	36.78	0.0	3.13
COST OPTIMIZED ACES[3]	10097	4846	1416	923	566	524	47.39	0.0	2.43

NOTES
1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

5.1.2 Results For The Multi-Family Residence

Table 5-9 gives the heating, cooling and hot water load profiles for the multi-family residence located in Minneapolis. (This residence is fully described in Appendix E.) The annual heating, cooling and hot water loads are 352, 80, and 154 million BTUs/year, respectively. Also shown in this table are the peak heating and cooling loads, the design load, and the steady state performance data of the ACES component train. This information, together with the information generated by the computer programs DESMOD and SIMMOD, is input into ECOMOD for the economic evaluation of the ACES. Also input into ECOMOD are the economic variables described in Section 4.0. Table 5-10 shows the values of the general economic variables for the multi-family residence in Minneapolis. Table 5-11 lists the unit component costs for the ACES and the four conventional HVAC systems. The final economic information needed to evaluate the ACES and conventional systems can be found in Tables 5-12 and 5-13. These tables contain the multi-family residence time-of-day rate structures, base prices and escalation rates applied to gas, oil and electricity.

The parameters used in the evaluation of the ACES and the ranges over which they are varied are shown below for the Minneapolis multi-family residence:

Tank	8192 ft^3, 12800 ft^3, 16928 ft^3
Collector	2718 ft^2, 2124 ft^2, 1000 ft^2, 0 ft^2
Changeover Temperature	71°F, 40°F, 30°F, 20°F, 15°F, 10°F, -10°F
Tank Insulation	top: R40; below ground: uninsulated, R40
Supplemental Heat	When needed
Costs	1978 prices, 1982 prices

Table 5-9. Load and ACES Equipment Performance Data for the Minneapolis Multi-Family Residence

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	896	0	136
FEBRUARY	615	0	128
MARCH	464	0	145
APRIL	211	28	142
MAY	68	41	143
JUNE	1	115	131
JULY	0	338	125
AUGUST	0	216	118
SEPTEMBER	24	47	111
OCTOBER	167	10	115
NOVEMBER	458	0	115
DECEMBER	611	0	127

PEAK HEATING LOAD 191,417

PEAK COOLING LOAD 175,584

DESIGN HEATING LOAD 210,000

CAPACITY OF WATER-TO-AIR HEAT PUMP 210,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE; 70/67° ENTERING AIR TEMPERATURE

INFILTRATION (CFM/FT^2) .1

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.55

HEAT PUMP BLOWER POWER 4198 WATTS

COOLING COIL BLOWER POWER 1799 WATTS

EVAPORATOR PUMP POWER 690 WATTS

COOLING LOOP PUMP POWER 690 WATTS

HOT WATER LOOP PUMP POWER 560 WATTS

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

Table 5-10. General Economic Analysis Variables for the Minneapolis Multi-Family Residence

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	35%
INTEREST RATE ON BORROWED MONEY	14.5%
PERCENT DOWN PAYMENT	10%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDIT	10%
PROPERTY TAX RATE	5.0%*
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%
DEPRECIATION SCHEDULE	STRAIGHT LINE

*NON-HOMESTEAD RATE

The trends of the multi-family residence in Minneapolis are basically the same as they were for the single-family residence; however, the magnitude of the numbers has increased due to the physical requirements of the structure. The relationship or ratio between heating, hot water and cooling loads have shifted when compared to those of the single-family residence. The ratio of the cooling and hot water loads to the total load is greater for the multi-family residence than for the single-family residence. This shifting of loads in colder climates allows the ACES to be more energy efficient in the multi-family residence than it would be in the single-family residence. The effect of tank insulation on system performance and costs is basically the same for each type of residence.

Table 5-11. Economic Input Data: HVAC Installed Costs for the Minneapolis Multi-Family Residence

COMPONENT	COST (DOLLARS)
HEATING PLANT	
NATURAL GAS	3316
OIL	3237
ELECTRIC	2828
CENTRAL AIR CONDITIONER	4985
HOT WATER HEATER	
NATURAL GAS	1980
ELECTRIC	1840
HIGH EFFICIENCY AIR-AIR HEAT PUMP	12047
ICE-MAKER HEAT PUMP	
DUAL SOURCE	18514
WATER SOURCE	17193
SOLAR COSTS - FIXED + VARIABLE	1900 + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	5000
ACES CONTROLS	4500
SUPPLEMENTAL ELECTRIC HEAT	785
ICE TANK	.26/FT^2 FOR LINER, 4.31/FT^2 FOR WALLS 2.35/FT^2 FOR SLAB, 3.60/FT^2 FOR COVER 5.95/FT FOR FTG. WHERE NEEDED .54/FT^2 FOR INSULATION

* Price for absorber with a 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5-12. Time-of-Day Service Rates for the Minneapolis Multi-Family Residence

	OCT.-MAY	JUNE-SEPT.
CUSTOMER CHARGE PER MONTH	$10.30	$10.30
ON-PEAK DEMAND CHARGE ($/KW)	3.75	4.75
OFF-PEAK DEMAND CHARGE ($/KW)	1.65	1.65
ON-PEAK ENERGY CHARGE ($/KWH)	0.0253	0.0253
OFF-PEAK ENERGY CHARGE ($/KWH)	0.0198	0.0198
ON-PEAK HOURS	9:00 AM - 9:00 PM Monday - Friday	

NOTE: Rates are taken from the Northern States Power (NSP) Company's proposed TOD Rate Design presented to the Minnesota Public Service Commission in August, 1978.

The results for the Minneapolis multi-family residence are summarized in Tables 5-14 through 5-16. Table 5-14 gives the system characteristics for the Full, Minimum and Cost Optimized ACES for the Minneapolis multi-family residence. The Full ACES consists of an insulated (R40) 16,928 ft^3 tank with a single-glazed 2124 ft^2 solar collector. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating systems. The Minimum ACES consists of an insulated (R40) 8192 ft^3 tank with a single-glazed 2718 ft^2 solar collector. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Cost Optimized ACES incorporates an uninsulated tank, no collector systems, a dual source evaporator with a 15°F changeover temperature and utilizes some auxiliary heat. These results show that when capital costs are large (1978) energy savings should be sacrificed by installing a smaller less cost-intensive system. However, as costs drop (1982) it becomes more cost-effective to install a larger more energy efficient system.

Table 5-13. Fuel Price and Escalation Rates for the Minneapolis Multi-Family Residence

	GAS	OIL	ELECTRICITY
BASE PRICE	$1.84/MCF	33.0¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	6.23	3.66	.7
PERIOD 2			
DURATION	5	17	17
RATE	5.54	1.11	.62
PERIOD 3			
DURATION	12	-	-
RATE	1.11	-	-
PERIOD 4			
DURATION	-	-	-
RATE	-	-	-

+ Escalation rate in percent per year
* Number of years that rate applies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

Table 5-14. Alternative ACES Configurations for the Minneapolis Multi-Family Residence

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	
			1978	1982
TANK INSULATION	R40	R40	UNINSULATED	UNINSULATED
TANK SIZE (FT^3)	16928	8192	8192	12800
COLLECTOR AREA (FT^2)	2124	2718	0	0
COLLECTOR TYPE	SINGLE GLAZE	SINGLE GLAZE	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	DUAL SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	15°	15°
ELECTRIC BACKUP	NONE	NONE	YES	YES

Table 5-15. Comparison of Initial Capital Costs (1978 Dollars) of the ACES and Conventional Systems for the Minneapolis Multi-Family Residence

COMPONENT \ SYSTEM	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	GAS	OIL	ELECTRIC	HEAT PUMP
HEATING PLANT[1]	17193	17193	18514	3316	3237	2828	12047
COOLING PLANT[1]	-	-	-	4985	4985	4985	-
HOT WATER HEATER	-	-	-	1980	1840	1840	1840
SOLAR COSTS	48635	60300	-	-	-	-	-
ACES CONTROLS	4500	4500	4500	-	-	-	-
TANK	23467	13064	12194*	-	-	-	-
TOTAL	93795	95057	35208	10281	10062	9653	13887

1 Conventional system costs include associated controls

* Cost includes auxiliary heaters

Table 5-16. Economic Characteristics of HVAC/DHW Systems for the Minneapolis Multi-Family Residence

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS)		ANNUALIZED COSTS (DOLLARS)				YEARLY ENERGY USE (BTU X 10^{-6})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC[1]	10281	10281	3550	3431	3956	3915	46.18	755.81	.73
OIL/ELECTRIC[2]	10062	10062	4256	3972	5074	4775	200.44	482.48	.85
AIR-TO-AIR HEAT PUMP[2]	13887	13887	4922	4544	5539	5148	383.98	0.0	1.52
ELECTRIC/ELECTRIC[2]	9653	9653	5423	5019	6930	6440	542.29	0.0	1.08
FULL ACES[3]	93795	45024	9895	5078	2109	1961	169.77	0.0	3.45
MINIMUM ACES[3]	95057	45633	10386	5478	2633	2447	181.79	0.0	3.22
COST OPTIMIZED ACES[3]	35208	19220	6158	4034	4551	3723	270./241.	0.0	2.16/2.44

NOTES
1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

A comparison of the tank sizes and collector areas for the Full and Minimum ACES shows that unlike the single-family residence ACES, the components of the multi-family Full ACES are considerably larger or smaller than components of the multi-family Minimum ACES. This is due to the load shifting noted previously. There is also a corresponding difference in energy efficiency between the multi-family residence Full and Minimum ACES. In general, as the difference between the heating and cooling load decreases (given that the heating load is greater than the cooling load) the overall effectiveness of the ACES increases.

Table 5-15 shows the installed cost of the Full, Minimum and Cost Optimized ACES, as well as the cost of the four conventional systems. These costs have been catagorized by system component. Clearly the solar collectors are the most capital-intensive components, while the tank and heat pump are the second most capital-intensive items. All ACES components are significantly more expensive than the conventional system components (with the exception of the air-to-air heat pump system).

Table 5-16 furnishes all costs associated with the Minneapolis multi-family ACES as well as all costs associated with the conventional systems. Note that the conventional systems cost between $9000 and $14,000 while the Full and Minimum ACES cost between $93,000 and $96,000 for the 1978 installation. It is this amortized dollar difference that must be offset by the savings obtained from using the ACES. The energy costs for the multi-family residence can be greater than the total costs. The reason for this occurrence is that the energy costs represent the total cost of all energy consumed and the total cost takes into account the tax deduction allowable for operating expenses. Thus, a significant portion of the money paid for the operating expenses (energy costs) will be refunded to the building owner thereby lowering his effective total costs. The 1978 annualized energy and total cost show that even though energy savings are substantial (the electric ACES is, however, penalized for demand charges where the fossil systems are not), the ACES are not cost-competitive with conventional systems. The corresponding 1982 prices show that

although the Full and Minimum ACES are not quite cost-competitive with conventional systems the Cost Optimized ACES is superior to all electric source system. The COP column in this table shows that the Full ACES is the most energy efficient system (maximizes interseasonal energy transfer). The seasonal COP is higher for the multi-family residence than it was for the single-family residence. This occurs because the heating load to cooling load ratio decreased; hence, the interseasonal energy transfer has increased. The yearly energy use column and the COP column show two inputs for the Cost Optimized ACES. The reason for this occurence is that there are two different Cost Optimized ACES--one based on 1978 prices and one based on 1982 prices. A comparison of the COP of the Full and Cost Optimized ACES indicates that it is better to sacrifice energy savings rather than increase capital costs. Therefore, the interseasonal energy savings of the Full ACES do not justify the required capital costs. As capital costs drop (1978-1982), the Cost Optimized ACES changes configuration causing an increase in energy effectiveness (COP goes from 2.16 to 2.44).

5.1.3. Results for the Commercial Building

The commercial building system has certain unique aspects not found in the single and multi-family systems. One of these aspects is the system's multi-zone loads. With multi-zone loads both heating and cooling loads may occur simultaneously; thus, bootstrapping (delivering internal heat to external zones) may be feasible. Another aspect is the capability to utilize both hot and cold storage tanks for off-peak charging. This high side storage capability has not been included in the single- and multi-family ACES applications.

The general trends of system behavior for the commercial systems are similar to those of the single and multi-family systems with respect to energy consumption. The major difference between the residential and commercial systems is that in a commercial building the inner zones are virtually insulated from the ambient by the outer zones. The heat generated by people, lights, and office equipment constitutes a significant internal cooling load throughout the year. In a building that does not utilize the ACES concept

this cooling load will be negated in winter by ventilation. In the commercial ACES, however, the heat pumps are available to bootstrap a portion of this energy to the outer zones via the hot and cold storage tanks; when appropriate, the ventilation is minimized to assure maximum bootstrapping. Figures 5-13 and 5-14 show the heating and cooling loads of the commercial building in Minneapolis. The loads without ventilation are generated from the National Bureau of Standard's NBSLD program. When modified with the Variable Air Volume (VAV) system and a minimum ventilation of 0.1 air change per hour, the heating load increases by 9 percent and cooling load decreases by 73 percent. With the standard VAV system control there is no cooling load in winter. However, with the ACES control the internal cooling load in winter is maximized by decreasing the ventilation. This allows for maximum bootstrapping and increases system efficiency.

With an efficient bootstrapping system, the commercial ACES needs auxiliary heat only from January to March. This is shown by the collected solar energy profile in Figure 5-15. The peak between Weeks 14 and 26 shows that the solar collectors are useful only during this period, roughly three months. During the rest of the year, most of the available solar energy is wasted as it is not needed by the system. This situation becomes more severe in southern climates.

Table 5-17 gives the heating, cooling and hot water load profiles for the commercial office building located in Minneapolis. (This building is described in Appendix E.) The annual heating, cooling and hot water loads are 707, 595, 53 million BTUs/year, respectively. Also shown in this table are the peak heating and cooling loads, the design load and the steady state performance data of the ACES component train. This information is input into CSIZE (computer sizing module for the Full and Minimum ACES). The load information, together with data generated in CSIZE, is then input into ACESIM (computer simulation module) from which the hourly energy use profile of the HVAC system to be analyzed is generated. All of this information is then

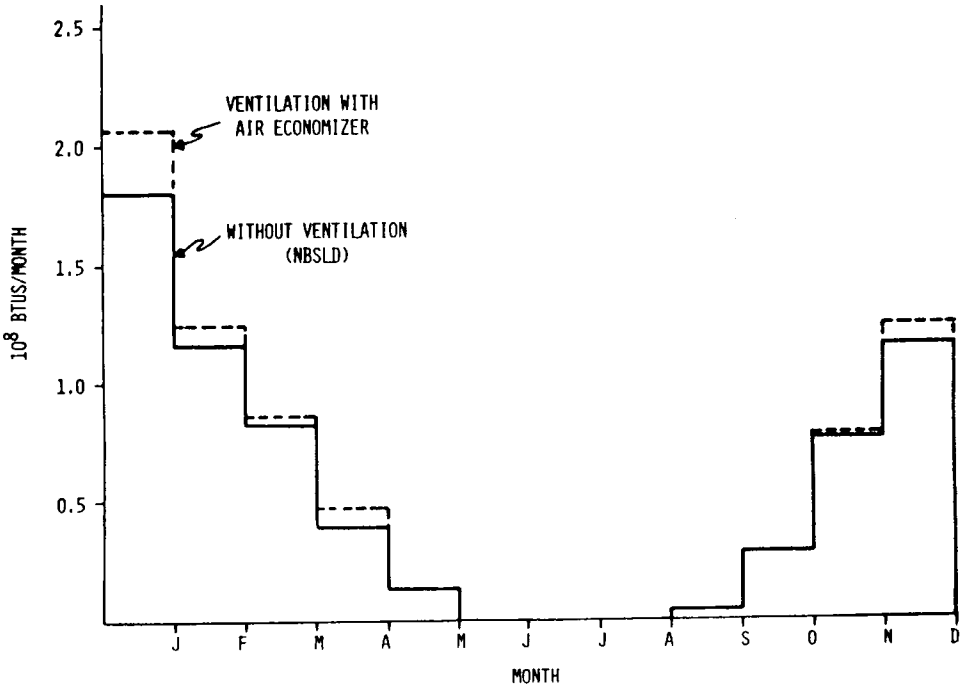


Figure 5-13. Heating Load Profile for the Minneapolis Commercial Building

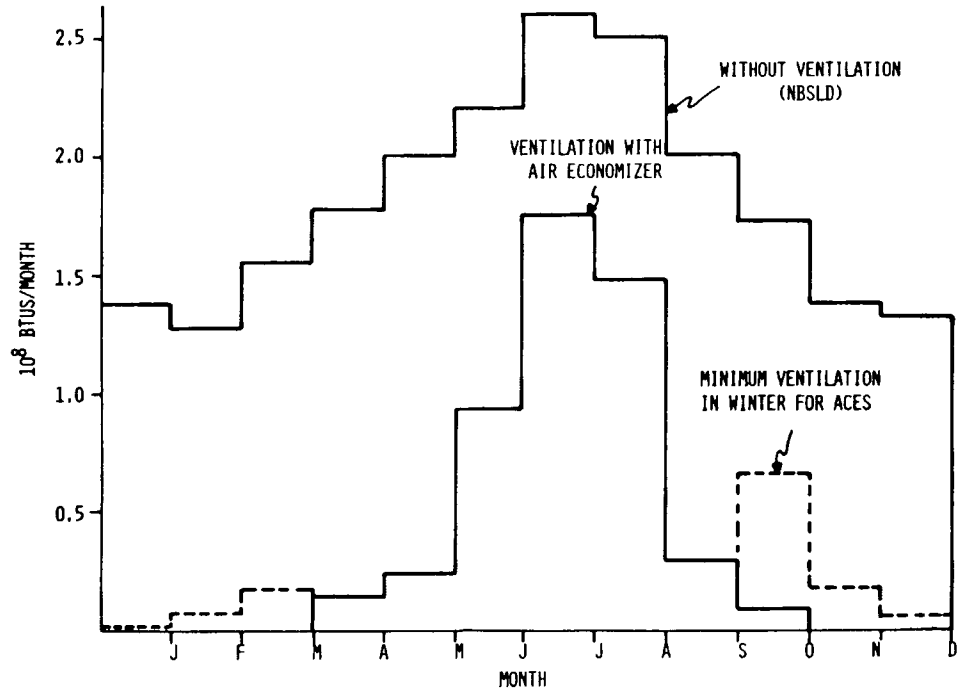


Figure 5-14. Cooling Load Profile for the Minneapolis Commercial Building

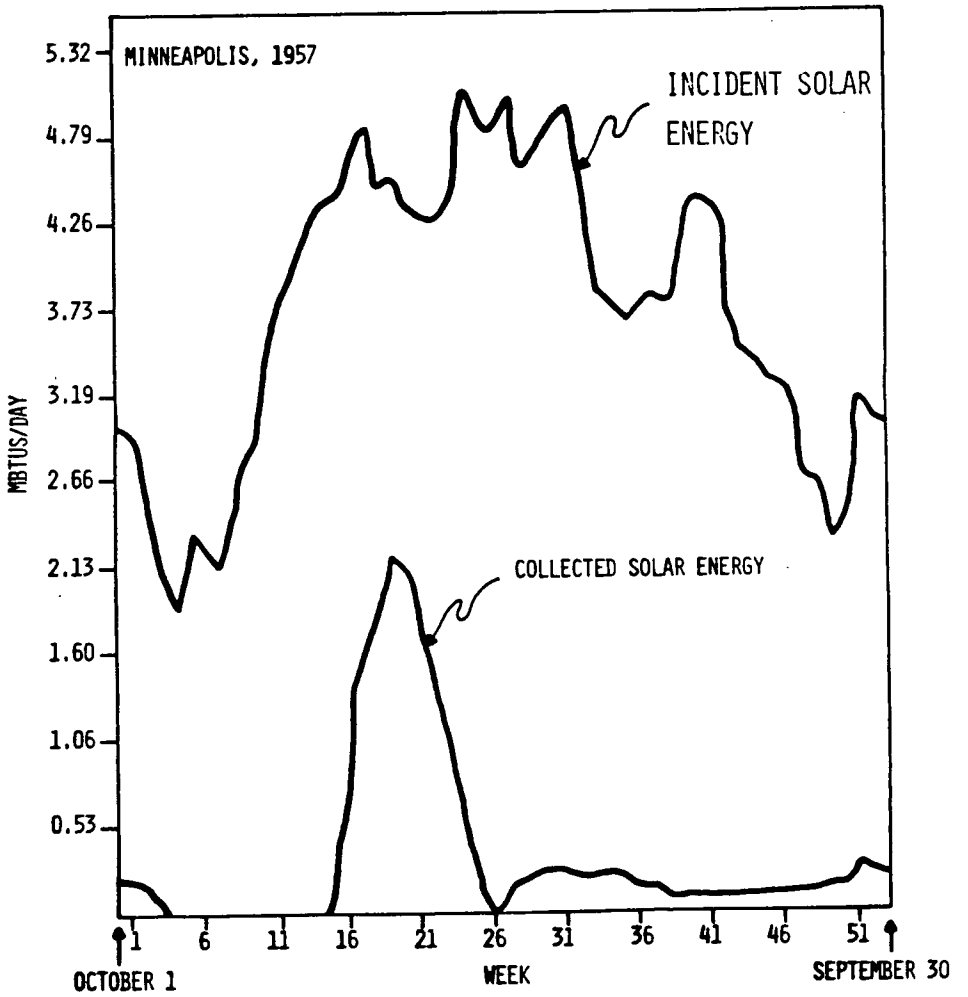


Figure 5-15. Incident and Collected Solar Energy Profile for the Minneapolis Commercial Building

Table 5-17. Load and ACES Equipment Performance Data for the Minneapolis Commercial Building

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	2073	11	47
FEBRUARY	1242	67	42
MARCH	871	179	49
APRIL	402	137	48
MAY	128	230	49
JUNE	3	934	45
JULY	0	1752	42
AUGUST	1	1473	44
SEPTEMBER	44	284	37
OCTOBER	278	657	41
NOVEMBER	788	171	40
DECEMBER	1233	53	40

PEAK HEATING LOAD 739,000 BTU/HR

PEAK COOLING LOAD 1,606,000 BTU/HR

CAPACITY OF WATER-TO-WATER HEAT PUMP 1,126,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE COLD SIDE; 120° ENTERING WATER TEMPERATURE HOT SIDE

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.20

AIR DISTRIBUTION SYSTEM POWER 1677 X 10^5 BTU/YEAR

EVAPORATOR PUMP POWER } 10KW
COOLING LOOP PUMP POWER }

HOT WATER LOOP PUMP POWER NONE

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

input into the final computer program ECOMOD for the economic evaluation of the ACES. Also input into ECOMOD are the economic variables described in Section 4.0. Table 5-18 gives the values of the general economic variables for the commercial building in Minneapolis. Table 5-19 lists unit component costs for the ACES and the three conventional HVAC systems. Tables 5-20 and 5-21 contain the commercial time-of-day rate structures, base prices, and escalation rates applied to gas, oil and electricity.

Table 5-18. General Economic Analysis Variables for the Minneapolis Commercial Building

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	46%
INTEREST RATE ON BORROWED MONEY	14.5%
PERCENT DOWN PAYMENT	10%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDIT	10%
PROPERTY TAX RATE	5.5%
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%
DEPRECIATION SCHEDULE	STRAIGHT LINE

Table 5-19. Economic Input Data: Hvac Installed Costs for the Minneapolis Commercial Building

COMPONENT	COST (DOLLARS)
HEATING PLANT	
NATURAL GAS	9260
OIL	8510
ELECTRIC	6970
CHILLER AND COOLING TOWER	58120
HOT WATER HEATER	
NATURAL GAS	1980
ELECTRIC	1840
ICE-MAKER HEAT PUMP	
20KW	7100
35KW	11900
SOLAR COSTS - FIXED + VARIABLE	40888 + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	0
ACES CONTROLS	10000
SUPPLEMENTAL ELECTRIC HEAT	2850
ICE TANK	.26/FT^2 FOR LINER, 4.31/FT^2 FOR WALLS 2.35/FT^2 FOR SLAB, 3.60/FT^2 FOR COVER 5.95/FT FOR FTG. WHERE NEEDED .54/FT^2 FOR INSULATION

* Price for absorber with a 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5-20. Time-of-Day Service Rates for the Minneapolis Commercial Building

	OCT.-MAY	JUNE-SEPT.
CUSTOMER CHARGE PER MONTH	$10.30	$10.30
ON-PEAK DEMAND CHARGE ($/KW)	3.75	4.75
OFF-PEAK DEMAND CHARGE ($/KW)	1.65	1.65
ON-PEAK ENERGY CHARGE ($/KWH)	0.0253	0.0253
OFF-PEAK ENERGY CHARGE ($/KWH)	0.0198	0.0198
ON-PEAK HOURS	9:00 AM - 9:00 PM Monday - Friday	

NOTE: Rates are taken from the Northern States Power (NSP) Company's proposed TOD Rate Design presented to the Minnesota Public Service Commission in August, 1978.

The parameters used in the evaluation of the ACES and the ranges over which they are varied are shown below for the Minneapolis commercial office building:

Tank	16251 ft^3, 47490 ft^3, 78732 ft^3
Collector	3078 ft^2, 1800 ft^2, 0 ft^2
Changeover Temperature	71°F, 20°F, -10°F
Tank Insulation	R40 for Full ACES; others R10
Supplemental Heat	When needed
Costs	1978 prices, 1982 prices

The energy consumption trends of the commercial ACES follow basically the same pattern as of energy trends of the residential ACES. One major difference between the two types of systems is that lighting energy needs to be included as an electrical energy input for the commercial system. The reason for this is that the utility rate paid by the commercial sector is a strong function of the amount of the energy consumed. Another major difference between the two types of systems is that the commercial system consumes large amounts of energy for lighting and air distribution purposes.

Table 5-21. Fuel Price and Escalation Rates for the Minneapolis Commercial Building

	GAS	OIL	ELECTRICITY
BASE PRICE	$1.84/MCF	33¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	6.230	3.660	0.700
PERIOD 2			
DURATION	5	17	17
RATE	5.540	1.110	0.620
PERIOD 3			
DURATION	12	-	-
RATE	1.110	-	-
PERIOD 4			
DURATION	-	-	-
RATE	-	-	-

+ Escalation rate in percent per year
* Number of years that rate applies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

The results for the Minneapolis commercial office building are summarized in Tables 5-22 through 5-24. Table 5-22 presents the system characteristics of the Full and Minimum ACES as well as the Cost Optimized system. The full ACES consists of an insulated (R40) 78732 ft^3 cold tank, with no collectors. This system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Minimum ACES consists of an insulated (R10) 16251 ft^3 cold tank with a row of single-glazed 3078 ft^2 solar collectors. This system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Cost Optimized system has a 16251 ft^3 cold tank, no collector and a dual source evaporator with a -10^oF changeover temperature. All of the systems have a 1327 ft^2 hot storage tank.

A comparison of tank sizes and collector areas for the commercial Full and Minimum ACES shows a dramatic difference in component sizes. This would be expected with the load profiles given in Table 5-17. The component configuration of the Cost Optimized system shows that this system no longer has the capability of interseasonal energy transfer. This lack of interseasonal energy transfer occurs because the system utilizes a -10^oF changeover temperature and essentially operates as an air-to-air heat pump during the winter. The commercial Cost Optimized system is really a load management system. This is not, however, a diurnal load management system but rather an extremely efficient load management system operating on a minimum seven day cycle.

Table 5-23 gives the installed costs of the Full, Minimum and Cost Optimized systems, as well as the cost of the three conventional systems. These costs have been catagorized by system component. Clearly the solar collector cost is the dominant cost of the Minimum ACES, whereas the tank cost is the dominant cost of the Full ACES. The difference in cost between the ACES and conventional systems is not as great for the commercial systems as for the residential systems.

Table 5-22. Alternative ACES Configurations for the Minneapolis Commercial Building

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED SYSTEM*	
			1978	1982
TANK INSULATION	R40	R10	R10	R10
TANK SIZE (FT^3)	78732	16251	16251	16251
COLLECTOR AREA (FT^2)	0	3078	0	0
COLLECTOR TYPE	NONE	SINGLE GLAZE	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	DUAL SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	-10°	-10°
ELECTRIC BACKUP	NONE	NONE	YES	YES

*COST OPTIMIZED CONFIGURATION DOES NOT CLASSIFY AS AN ACES.

Table 5-23. Comparison of Initial Capital Costs (1978 Dollars) of the ACES and Conventional Systems for the Minneapolis Commercial Building

SYSTEM / COMPONENT	FULL ACES	MINIMUM ACES	COST OPTIMIZED SYSTEM	GAS	OIL	ELECTRIC
HEATING PLANT[1]	38000	38000	38000	9620	8510	6970
COOLING PLANT[1]	-	-	-	58120	58120	58120
HOT WATER HEATER	-	-	-	1980	1840	1840
SOLAR COSTS	-	102292	-	-	-	-
ACES CONTROLS	10000	10000	10000	-	-	-
TANK (HOT & COLD)	100358	32084	32084	-	-	-
TOTAL	148358	182376	80084	69720	68470	66930

1 Conventional system costs include associated controls

Table 5-24. Economic Characteristics of HVAC/DHW Systems for the Minneapolis Commercial Building

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS)		ANNUALIZED COSTS (DOLLARS)				YEARLY ENERGY* USE (BTU X 10^{-9})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC[1]	69720	69720	16315	15272	23707	22389	.165/.379	1.35/1.35	.894/.780
OIL/ELECTRIC[2]	68470	68470	16293	15123	23751	22188	.224/.440	1.01/1.01	1.09/.932
ELECTRIC/ELECTRIC[2]	66930	66930	17702	16372	26504	24632	.980/1.20	0.0	1.38/1.13
FULL ACES[3]	148358	145085	17384	15847	19078	17731	.408/.575	0.0	3.32/2.35
MINIMUM ACES[3]	182376	149548	20417	17573	22073	20514	.559/.726	0.0	2.42/1.86
COST OPTIMIZED SYSTEM[3]	80084	78317	14486	13388	24001	22307	.743/.931	0.0	1.82/1.45

NOTES
1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

*First figure denotes consumption without distribution energy; second figure includes distribution energy.

Table 5-24 presents all costs associated with the Minneapolis commercial building systems. The conventional aystems cost between $66,000 and $70,000 while the Full and Minimum ACES cost between $148,000 and $183,000 for the 1978 installation. It is this amortized dollar difference that must be offset by the savings obtained by using the ACES.

The energy costs for the commercial systems account for the total energy consumed. This includes all lighting energy. Generally this type of energy consumption is not included in a comparative analysis, but it is imperative to do so in the commercial sector since the total energy consumed has a significant effect on the energy rate charged. The energy costs for the commercial systems are greater than the total costs. The reason for this occurance is that the energy costs represent the total cost of all energy consumed and the total cost takes into account the tax deduction allowable for operating expenses. Thus a significant portion of the money paid for the operating expenses, (energy costs) will be refunded to the building owner thereby lowering his effective total costs. The COPs shown in this table have the same definition as those given for the residential buildings; therefore, only the energy supplied to the HVAC plants are given in the energy use column of Table 5-24. Two values of energy use and COP are given for all commercial applications. The first value of energy consumption and corresponding COP do not include air distribution consumption. The second value does. These two-valued columns are included to illustrate the massive amounts of energy consumed by the air distribution system in the commercial building, and the effect that this distribution energy consumption has on the apparent energy effectiveness of the ACES. Clearly, large amounts of electrical energy (COP=1) supplied to a system with a low COP will not severely affect the resultant annual COP. This statement is true for all the conventional systems. However, supplying large amounts of electrical energy to a system with a high COP (such as the ACES) severely degrades the resultant annual COP. The commercial ACES are very efficient in meeting the building loads (high COP excluding the air distribution energy); however, when the distribution energy is included this energy effectiveness of the commercial ACES becomes overshadowed.

Table 5-24 shows that the initial capital cost of the Cost Optimized commercial system is significantly less than the cost of the Full or Minimum ACES and substantially more than the costs of the conventional systems. The Full ACES is cost-competitive with the electric conventional system. Finally, the Cost Optimized system is economically superior to all conventional systems as well as to the Full and Minimum ACES.

5.1.4 Minneapolis Summary

Tables 5-25 and 5-26 present the break-even and payback duration of the ACES and conventional systems for the single-family residence, multi-family residence and commercial office building in Minneapolis. The results are given for the 1982 installation of the single- and multi-family ACES and the 1978 installation of the commercial ACES. These installation dates are used since the costs associated with these respective dates are felt to be representative costs of the mature systems.

These tables demonstrate that the single-family Minimum ACES is not competitive with any of the conventional systems. The single-family Full ACES is marginally competitive with only the conventional electric systems. The single-family Cost Optimized ACES is economically superior to all conventional systems except gas. The multi-family Full and Minimum ACES are not cost-competitive with any of the conventional systems. However, the multi-family Cost Optimized ACES is superior to both of the elctric conventional systems. Although the commercial Cost Optimized system is not truly an ACES, it is economically superior to all conventional systems as well as to the Full and Minimum ACES.

Table 5-25. Life Cycle Cost Analysis Break-Even Duration (Years) for the Mature (1978 Commercial/1982 Residential) ACES with Respect to the Conventional Systems for Minneapolis

ACES \ CONVENTIONAL SYSTEM	ELECTRIC	OIL	GAS	HEAT PUMP
SINGLE-FAMILY RESIDENCE				
FULL ACES	19	-	-	-
MINIMUM ACES	-	-	-	-
COST OPTIMIZED ACES	1	8	-	1
MULTI-FAMILY RESIDENCE				
FULL ACES	-	-	-	-
MINIMUM ACES	-	-	-	-
COST OPTIMIZED ACES	11	-	-	11
COMMERCIAL BUILDING				
FULL ACES	19	-	-	N/A
MINIMUM ACES	-	-	-	N/A
COST OPTIMIZED SYSTEM	5	6	6	N/A

Duration greater than 20 years

N/A Not applicable

Table 5-26. Life Cycle Cost Analysis Payback Duration (Years) for the Mature (1978 Commercial/1982 Residential) ACES with Respect to the Conventional Systems for Minneapolis

ACES \ CONVENTIONAL SYSTEM	ELECTRIC	OIL	GAS	HEAT PUMP
SINGLE FAMILY RESIDENCE				
FULL ACES	20	-	-	-
MINIMUM ACES	-	-	-	-
COST OPTIMIZED ACES	7	14	-	7
MULTI-FAMILY RESIDENCE				
FULL ACES	-	-	-	-
MINIMUM ACES	-	-	-	-
COST OPTIMIZED ACES	12	-	-	12
COMMERCIAL BUILDING				
FULL ACES	19	-	-	N/A
MINIMUM ACES	-	-	-	N/A
COST OPTIMIZED SYSTEM	11	12	12	N/A

\- Duration greater than 20 years

N/A Not applicable

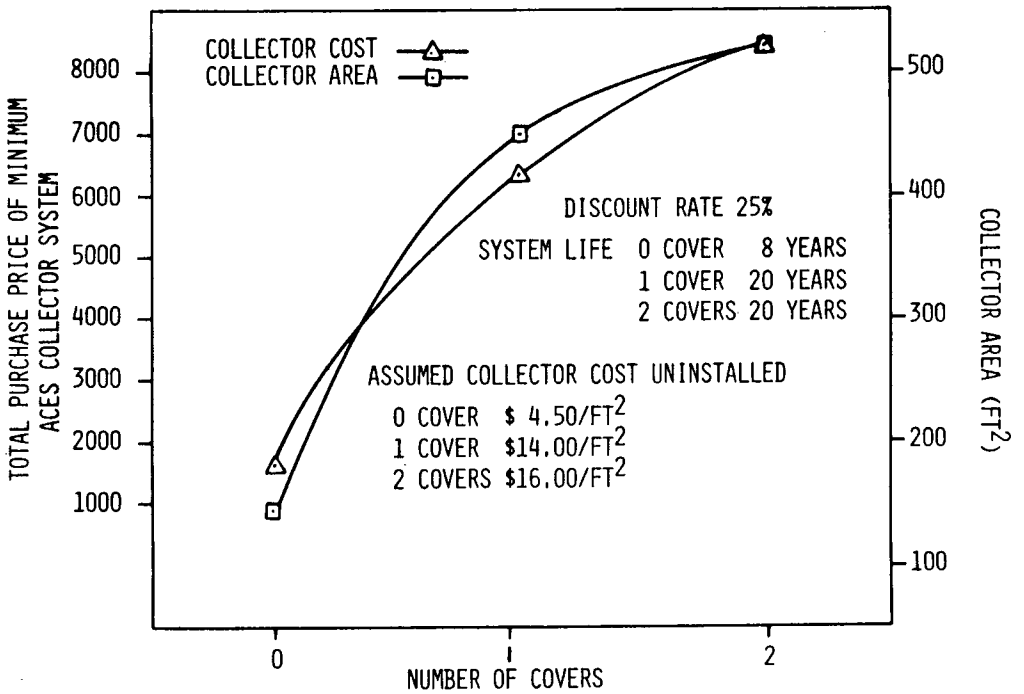


Figure 5-16. Area and Purchase Price of the Uninstalled Collector System as a Function of the Number of Collector Covers for the Multi-Family Minimum ACES in Atlanta

5.2 ATLANTA RESULTS

Figure 5-16 shows the effect of the type of collector system on the required collector area and on the collector purchase price for the multi-family Minimum ACES in Atlanta. Increasing the number of covers from zero to one increases the required collector area as well as the total purchase price of the collector system. Increasing the number of covers from one to two results in a further increase in the required collector area and also the total purchase price of the collector system. Thus, the unglazed (absorber) system is best suited for the operation of the ACES in Atlanta.

5.2.1 Results For The Single-Family Residence

Table 5-27 gives the heating, cooling and hot water load profiles for the single-family residence located in Atlanta. (This residence is fully described in Appendix E.) The annual heating, cooling and hot water loads are 24, 26, and 17 million BTUs/year, respectively. Also shown in this table are the peak heating and cooling loads, the design load and the steady state performance data of the ACES component train. This information, together with the information generated by the computer programs DESMOD and SIMMOD, is input into ECOMOD for the economic evaluation of the ACES. Also input into ECOMOD are the economic variables described in Section 4.0. Table 5-28 gives the values of the general economic variables for the single-family residence located in Atlanta. Table 5-29 lists the unit components costs for the ACES and four conventional HVAC systems. The final economic information needed to evaluate the ACES and conventional systems can be found in Tables 5-30 and 5-31. These tables contain the single-family residence time-of-day rate structures, base prices and escalation rates applied to gas, oil and electricity.

The parameters used in the evaluation of the ACES and the ranges over which they are varied are shown below for the Atlanta single-family residence:

Tank	648 ft^3, 2312 ft^3, 5000 ft^3
Collector	54 ft^2, 27 ft^2, 0 ft^2
Changeover Temperature	71°F, 40°F, 30°F, 25°F, 20°F, 15°F
Tank Insulation	top: R40; below ground: uninsulated, R40
Supplemental Heat	When needed
Costs	1978 prices, 1982 prices

The results for the Atlanta single-family residence show a large reduction in energy compared to its Minneapolis counterpart. The reason for this

Table 5-27. Load and ACES Equipment Performance Data for the Atlanta Single-Family Residence

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	48	0	17
FEBRUARY	46	0	16
MARCH	33	0	16
APRIL	11	5	15
MAY	0	23	16
JUNE	0	76	12
JULY	0	77	13
AUGUST	0	49	13
SEPTEMBER	0	24	13
OCTOBER	10	5	13
NOVEMBER	25	0	13
DECEMBER	67	0	17

PEAK HEATING LOAD 23,054 BTU/HR

PEAK COOLING LOAD 29,612 BTU/HR

DESIGN HEATING LOAD 36,000 BTU/HR

CAPACITY OF WATER-TO-AIR HEAT PUMP 36,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE; 70/67° ENTERING AIR TEMPERATURE

INFILTRATION (CFM/FT^2) .12 HOUSE .10 ATTIC

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.55

HEAT PUMP BLOWER POWER 750 WATTS

COOLING COIL BLOWER POWER 300 WATTS

EVAPORATOR PUMP POWER 100 WATTS

COOLING LOOP PUMP POWER 100 WATTS

HOT WATER LOOP PUMP POWER 100 WATTS

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

Table 5-28. General Economic Analysis Variables for the Atlanta Single-Family Residence

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	30.0%
INTEREST RATE ON BORROWED MONEY	10.5%
PERCENT DOWN PAYMENT	10.0%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDITS	10.0%
PROPERTY TAX RATE	2.51%
DEPRECIATION SCHEDULE	NONE
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%

reduction is mainly the change in the external conditions (weather). Thus, the heating load is greatly reduced, the hot water load is reduced and the cooling load is increased. The cooling load, however, has not increased proportionally to the increase that would be expected by the warmer Atlanta climate. This is due to the fact that the Atlanta single-family dwelling has been specifically designed to passively reduce transmitted solar radiation. This reduction in total load has a two-fold effect on the results of the Atlanta single-family residence ACES. The first is that the resultant seasonal COP is increased due to the closer balance between the heating and cooling loads. The second is that the overall energy requirements of the structure are decreased; hence, the potential for economic superiority of the ACES can be decreased. The reason for this potential decrease is that the economic effectiveness of the ACES is solely a function of the incremental fuel savings and the incremental costs expended in purchasing the

Table 5-29. Economic Input Data: HVAC Installed Costs for the Atlanta Single-Family Residence

COMPONENT	COST (DOLLARS)
HEATING PLANT	
NATURAL GAS	660
OIL	850
ELECTRIC	690
CENTRAL AIR CONDITIONER	850
HOT WATER HEATER	
NATURAL GAS	182
ELECTRIC	170
HIGH EFFICIENCY AIR-AIR HEAT PUMP	2690
ICE-MAKER HEAT PUMP	
DUAL SOURCE	3920
WATER SOURCE	3540
SOLAR COSTS - FIXED + VARIABLE	1580 + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	563
ACES CONTROLS	1150
SUPPLEMENTAL ELECTRIC HEAT	300
ICE TANK	.24/FT^2 FOR LINER, 4.08/FT^2 FOR WALLS 2.22/FT^2 FOR SLAB, 3.41/FT^2 FOR COVER 5.63/FT FOR FTG. WHERE NEEDED .54/FT^2 FOR INSULATION

* Price for absorber with a 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5-30. Time-of-Day Service Rates for the Atlanta Single-Family Residence

	JUNE-SEPT.	OCT.-MAY
CUSTOMER CHARGE PER MONTH	$5.75	$5.75
ON-PEAK ENERGY CHARGE ($/KWH)	0.0758	0.0494
OFF-PEAK ENERGY CHARGE ($/KWH)	0.0269	0.0269
ON-PEAK HOURS	7:00 AM - 10:00 PM Monday - Friday	

energy conserving system. Whether the cost-effectiveness of the ACES actually decreases or increases depends on the capital and operating costs evaluated over the life of the system.

Cost and energy trends for the Atlanta single-family residence are similar to those obtained for the Minneapolis single-family residence. These trends are, however, distorted due to the load relationship encountered in Atlanta. imperative to use a well-insulated tank. The results indicate that the COP of the ACES is severely degraded if the tank is uninsulated.

Table 5-32 shows the system characteristics for the Full, Minimum and Cost Optimized ACES for the Atlanta single-family residence. The Full ACES consists of an insulated (R40) 5000 ft^3 tank with no solar collector. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The minimum ACES consists of an insulated (R40) 648 ft^3 tank with an unglazed 54 ft^2 solar collector. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Cost Optimized system shows that it is more effective to have no solar collector and a dual source evaporator with either a 30°F or 40°F changeover temperature. It also shows that when capital costs are large (1978) energy savings should be sacrificed by installing a smaller, less cost-intensive system. However, as costs drop (1982) it becomes more cost-effective to install a larger, more energy efficient system and thus increase interseasonal energy transfer.

Table 5-31. Fuel Price and Escalation Rates for the Atlanta Single-Family Residence

	GAS	OIL	ELECTRICITY
BASE PRICE	$2.85/MCF	47.2¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	6.150	3.338	1.127
PERIOD 2			
DURATION	9	17	17
RATE	3.818	.824	.170
PERIOD 3			
DURATION	2	-	-
RATE	2.48	-	-
PERIOD 4			
DURATION	6	-	-
RATE	.824	-	-

+ Escalation rate in percent per year
* Number of years that rate applies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

Table 5-32. Alternative ACES Configurations for the Atlanta Single-Family Residence

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES 1978	COST OPTIMIZED ACES 1982
TANK INSULATION	R40	R40	R40	R40
TANK SIZE (FT^3)	5000	648	648	2312
COLLECTOR AREA (FT^2)	0	54	0	0
COLLECTOR TYPE	NONE	ABSORBER	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	DUAL SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	30°	40°
ELECTRIC BACKUP	NONE	NONE	NONE	NONE

Table 5-33 gives the installed costs of the Full, Minimum, and Cost Optimized ACES, as well as the costs of the four conventional systems. These costs have been catagorized by system component. The most expensive item shown is the cost of the tank for the Full ACES, followed by the ACES ice-maker heat pumps and the solar collector and minimum storage bin. Clearly, the ACES components are much more expensive than the components of conventional systems.

The costs associated with the Atlanta single-family ACES and the costs associated with the conventional systems are found in Table 5-34. Note that the conventional systems cost between $1,600 and $2,900 whereas the Full and Minimum ACES systems cost between $10,000 and $14,000 for 1978 installations. It is this amortized dollar difference that must be offset by the savings obtained from using the ACES. The 1978 annualized costs illustrate that even though energy savings are significant, the ACES are not cost-competitive with the conventional systems. The corresponding 1982 prices show that the Full and Minimum ACES are cost competitive with all conventional systems except gas. The Cost Optimized ACES is superior to all systems including the Full and Minimum ACES. The COP column shows that the Full ACES is by far the most energy efficient system. Decreasing the interseasonal energy transfer results in a degradation of seasonal COP, as can be seen by comparing the Full and Minimum ACES energy requirements (COP goes from 3.50 to 2.44). Comparing the COP of the Full and Cost Optimized ACES shows that it is better to sacrifice energy savings than to increase capital costs. Hence, the interseasonal energy savings of the Full ACES do not economically justify the required capital costs. As capital costs drop (1978 to 1982), the energy effectiveness of the Cost Optimized ACES increases (COP goes from 2.34 to 2.62). Finally, if the relative cost-effectiveness of the Atlanta single-family ACES is compared with the cost-effectiveness of the Minneapolis single-family ACES, it can be said that the residential ACES concept is more viable in Atlanta.

Table 5-33. Comparison of Initial Capital Costs (1978 Dollars) of the ACES and Conventional Systems for the Atlanta Single-Family Residence

SYSTEM / COMPONENT	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	GAS	OIL	ELECTRIC	HEAT PUMP
HEATING PLANT[1]	3540	3540	3920	660	850	690	2690
COOLING PLANT[1]	-	-	-	850	850	850	-
HOT WATER HEATER	-	-	-	182	170	170	170
SOLAR COSTS	-	3051	-	-	-	-	-
ACES CONTROLS	1150	1150	1150	-	-	-	-
TANK	8510	2616	2616	-	-	-	-
TOTAL	13200	10357	7686	1692	1870	1710	2860

1 Conventional system costs include associated controls

Table 5-34. Economic Characteristics of HVAC/DHW Systems for the Atlanta Single-Family Residence

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS)		ANNUALIZED COSTS (DOLLARS)				YEARLY ENERGY USE (BTU X 10^{-6})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC[1]	1692	1692	732	705	597	583	13.53	60.30	.91
OIL/ELECTRIC[2]	1870	1870	844	780	695	646	30.95	32.23	1.06
AIR-TO-AIR HEAT PUMP[2]	2860	2860	882	809	654	602	39.90	0.0	1.69
ELECTRIC/ELECTRIC[2]	1710	1710	937	861	800	737	53.72	0.0	1.25
FULL ACES[3]	13200	6346	1262	727	310	285	19.27	0.0	3.50
MINIMUM ACES[3]	10357	4969	1183	738	406	374	27.57	0.0	2.44
COST OPTIMIZED ACES[3]	7686	4754	1000	698	415	355	28.77/25.75	0.0	2.34/2.62

NOTES
1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

5.2.2 Results For The Multi-Family Residence

Table 5-35 gives the heating, cooling and hot water load profiles for the multi-family residence located in Atlanta. (This residence is fully described in Appendix E.) The annual heating, cooling and hot water loads are 78, 214, and 117 million BTUs/year, respectively. Also shown in this table are the peak heating and cooling loads, the design load, and the steady state performance data of the ACES component train. This information, together with the information generated by the computer programs DESMOD and SIMMOD, is input into ECOMOD for the economic evaluation of the ACES. Also input into ECOMOD are the economic variables described in Section 4.0. Table 5-36 lists the values of the general economic variables for the multi-family residence in Atlanta. Table 5-37 furnishes the unit component cost for the ACES and four conventional systems. The final economic information needed to evaluate the ACES can be found in Tables 5-38 and 5-39. These tables contain the multi-family residence time-of-day rate structures, base prices, and escalation rates applied to gas, oil and electricity.

The parameters used in the evaluation of the ACES and the ranges over which they are varied are shown below for the Atlanta multi-family residence:

Tank	2592 ft^3, 12168 ft^3, 21632 ft^3
Collector	180 ft^2, 90 ft^2, 0 ft^2
Changeover Temperature	71°F, 40°F, 30°F, 25°F, 20°F, 15°F
Tank Insulation	Top: R40; below ground: uninsulated, R40
Supplemental Heat	When needed
Costs	1978 prices, 1982 prices

Table 5-35. Load and ACES Equipment Performance Data for the Atlanta Multi-Family Residence

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	164	2	119
FEBRUARY	158	6	111
MARCH	101	13	110
APRIL	32	88	106
MAY	0	224	110
JUNE	0	546	80
JULY	0	542	83
AUGUST	0	385	83
SEPTEMBER	0	232	84
OCTOBER	25	87	87
NOVEMBER	75	20	84
DECEMBER	224	0	119

PEAK HEATING LOAD 109,386 BTU/HR

PEAK COOLING LOAD 185,659 BTU/HR

DESIGN HEATING LOAD 125,000 BTU/HR

CAPACITY (COOLING) OF WATER-TO-AIR HEAT PUMP 191,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE; 70/67° ENTERING AIR TEMPERATURE

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.55

HEAT PUMP BLOWER POWER 5437 WATTS

COOLING COIL BLOWER POWER 1919 WATTS

EVAPORATOR PUMP POWER 844 WATTS

COOLING LOOP PUMP POWER 844 WATTS

HOT WATER LOOP PUMP POWER 560 WATTS

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

Table 5-36. General Economic Analysis Variables for the Atlanta Multi-Family Residence

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	35%
INTEREST RATE ON BORROWED MONEY	14.5%
PERCENT DOWN PAYMENT	10%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDIT	10%
PROPERTY TAX RATE	2.51%
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%
DEPRECIATION SCHEDULE	STRAIGHT LINE

The trends of the multi-family residence located in Atlanta are basically the same as those of the single-family residence; however, energy consumption and costs have increased due to the physical requirements of the structure. The multi- and single-family residential loads ratio in Atlanta is similar to the ratio of loads found in the Minneapolis residences. Due to the large cooling load in Atlanta the multi-family Full ACES has a COP that is lower than the Atlanta single-family Full ACES and the Minneapolis multi-family Full ACES.

Table 5-40 gives the system characteristics for the Full, Minimum, and Cost Optimized ACES for the Atlanta multi-family residence. The Full ACES consists of an insulated (R40) 21,632 ft^3 tank with no collectors. The system utilizes a water source heat pump with no airside evaporator and no auxiliary

Table 5-37. Economic Input Data: HVAC Installed Costs for the Atlanta Multi-Family Residence

COMPONENT	COST (DOLLARS)
HEATING PLANT	
NATURAL GAS	1860
OIL	2004
ELECTRIC	2006
CENTRAL AIR CONDITIONER	6952
HOT WATER HEATER	
NATURAL GAS	1980
ELECTRIC	1840
HIGH EFFICIENCY AIR-AIR HEAT PUMP	14256
ICE-MAKER HEAT PUMP	
DUAL SOURCE	21923
WATER SOURCE	20358
SOLAR COSTS - FIXED + VARIABLE	1900* + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	5000
ACES CONTROLS	4500
SUPPLEMENTAL ELECTRIC HEAT	785
ICE TANK	.24/FT^2 FOR LINER, 4.08/FT^2 FOR WALLS 2.22/FT^2 FOR SLAB, 3.41/FT^2 FOR COVER 5.63/FT FOR FTG. WHERE NEEDED .54/FT^2 FOR INSULATION

* Price for absorber with a 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5-38. Time-of-Day Service Rates for the Atlanta Multi-Family Residence

	JAN.-DEC.
CUSTOMER CHARGE	$18
ON-PEAK DEMAND CHARGE ($/KW)	3.50
OFF-PEAK DEMAND CHARGE ($/KW)	1.54
ON-PEAK ENERGY CHARGE ($/KWH)	0.0722
OFF-PEAK ENERGY CHARGE ($/KWH)	0.0396
ON-PEAK HOURS	7:00 AM - 9:00 PM Monday - Friday

NOTE: Demand and energy charges were developed by modifying current Georgia Power and Light rates by ratios calculated from analysis of NSP rates. See Commercial Philadelphia rate note for further amplification.

heating system. The Minimum ACES consists of an insulated (R40) 2592 ft^3 tank with an unglazed 180 ft^2 collector system. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Cost Optimized system in 1978 utilizes a dual source evaporator with a 30° changeover temperature, no solar collector, no auxiliary heat, and a well-insulated (R40) 2592 ft^3 tank. The Full ACES is the Cost Optimized ACES for the 1982 price scenario. When capital costs are large (1978) energy savings should be sacrificed by installing a smaller, less cost-intensive system. However, as cost drop (1982) it becomes more cost-effective to install a larger, more energy efficient system.

Table 5-39. Fuel Price and Escalation Rates for the Atlanta Multi-Family Residence

	GAS	OIL	ELECTRICITY
BASE PRICE	$2.35/MCF	30¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	6.142	3.597	1.258
PERIOD 2			
DURATION	5	17	17
RATE	4.484	1.192	.524
PERIOD 3			
DURATION	12	-	-
RATE	1.192	-	-
PERIOD 4			
DURATION	-	-	-
RATE	-	-	-

+ Escalation rate in percent per year
* Number of years that rate applies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

Table 5-40. Alternative ACES Configurations for the Atlanta Multi-Family Residence

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	
			1978	1982
TANK INSULATION	R40	R40	R40	R40
TANK SIZE (FT^3)	21632	2592	2592	21632
COLLECTOR AREA (FT^2)	0	180	0	0
COLLECTOR TYPE	NONE	ABSORBER	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	WATER SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	30°	NONE
ELECTRIC BACKUP	NONE	NONE	NONE	NONE

Table 5-41 lists the installed costs of the Full, Minimum and Cost Optimized ACES, as well as the costs of the four conventional systems. These costs have been catagorized by system component. This table illustrates the large costs of the ACES component and the difference in costs between the ACES and conventional components.

Table 5-42 shows the costs associated with the Atlanta multi-family ACES as well as the costs associated with conventional systems. Note that the conventional systems cost between $10,000 and $17,000 while the Full and Minimum ACES cost between $38,000 and $53,000 for the 1978 installation. It is this amortized dollar difference that must be offset by the savings obtained by using the ACES. The energy costs for the multi-family residence can be greater than the total costs. The reason for this occurrence is that the total cost takes into account the tax deduction allowable for operating expenses. Thus, a significant portion of the money paid for the operating expenses (energy costs) will be refunded to the building owner, thereby lowering his effective total costs. The 1978 annualized energy and total costs show that even though energy savings are substantial (the electric ACES are, however, penalized for demand charges where the fossil systems are not), the ACES are not cost-competitive with conventional systems. The corresponding 1982 prices show that the Cost Optimized ACES and therefore the Full ACES is superior to all systems except gas. The COP column in this table shows that the Full ACES (Cost Optimized ACES in 1982) is the most energy efficient system. The interseasonal energy savings of the Full Aces do justify the required capital cost.

5.2.3 Results For The Commercial Building

Table 5-43 gives the heating, cooling and hot water load profiles for the commercial office building located in Atlanta. (This building is described in Appendix E.) The annual heating, cooling and hot water loads are 498, 1308, and 42 million BTUs/year, respectively. Also shown in this table are

Table 5-41. Comparison of Initial Capital Costs (1978 Dollars) of the ACES and Conventional Systems for the Atlanta Multi-Family Residence

SYSTEM / COMPONENT	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	GAS	OIL	ELECTRIC	HEAT PUMP
HEATING PLANT[1]	20358	20358	21923	1860	2004	2006	14256
COOLING PLANT[1]	-	-	-	6952	6952	6952	-
HOT WATER HEATER	-	-	-	1980	1840	1840	1840
SOLAR COSTS	-	9939	-	-	-	-	-
ACES CONTROLS	4500	4500	4500	-	-	-	-
TANK	27459	5091	5091	-	-	-	-
TOTAL	52317	39888	31514	10792	10796	10798	16096

1 Conventional system costs include associated controls

Table 5-42. Economic Characteristics of HVAC/DHW Systems for the Atlanta Multi-Family Residence

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS)		ANNUALIZED COSTS (DOLLARS)				YEARLY ENERGY USE (BTU X 10^{-6})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC[1]	10792	10792	3764	3550	4480	4273	102.26	279.57	1.07
OIL/ELECTRIC[2]	10796	10796	5313	4939	6861	6409	219.83	104.82	1.26
AIR-TO-AIR HEAT PUMP[2]	16096	16096	5570	5160	6613	6167	239.10	0.0	1.71
ELECTRIC/ELECTRIC[2]	10798	10798	6056	5625	8005	7465	293.93	0.0	1.39
FULL ACES[3]	52317	25111	6207	3895	3472	3238	132.08	0.0	3.10
MINIMUM ACES[3]	39888	19145	6052	4188	4551	4244	168.88	0.0	2.42
COST OPTIMIZED ACES[3]	31514	25111	5564	3895	4750	3238	177.78/132.08	0.0	2.30/3.10

NOTES
1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

Table 5-43. Load and ACES Equipment Performance Data for the Atlanta Commercial Building

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	969	341	40
FEBRUARY	909	438	36
MARCH	493	726	42
APRIL	276	325	36
MAY	73	1022	40
JUNE	0	1958	33
JULY	0	2012	29
AUGUST	0	2418	34
SEPTEMBER	6	1690	30
OCTOBER	90	1283	32
NOVEMBER	874	727	33
DECEMBER	1294	135	39

PEAK HEATING LOAD 822,000 BTU/HR

PEAK COOLING LOAD 1,515,000 BTU/HR

CAPACITY OF WATER-TO-WATER HEAT PUMP 1,331,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE COLD SIDE; 120° ENTERING WATER TEMPERATURE HOT SIDE

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.20

AIR DISTRIBUTION POWER 1857 X 10^5 BTU/YEAR

EVAPORATOR PUMP POWER
COOLING LOOP PUMP POWER } 10KW

HOT WATER LOOP PUMP POWER NONE

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

the peak heating and cooling loads, the design load and the steady state performance data of the ACES component train. This information, together with the information generated in CSIZE and ACESIM, is input into ECOMOD for the economic evaluation of the ACES. Also input into ECOMOD are the economic variables described in Section 4.0. Table 5-44 gives the values of the general economic variables for this commercial office building located in Atlanta. Table 5-45 provides the unit component costs for the ACES and the three conventional HVAC cystems. The final information needed to evaluate the ACES and conventional systems can be found in Tables 5-46 and 5-47. These tables contain the commercial time-of-day rate structures, base prices, and escalation rates applied to oil, gas and electricity.

The parameters used in the evaluation of the ACES and the ranges over which they are varied are shown below for the Atlanta commercial office building:

Tank	24876 ft^3, 17588 ft^3, 10302 ft^3
Collector	465 ft^2, 0 ft^2
Changeover Temperature	71°F, 20°F
Tank Insulation	R40 for Full ACES, others R10
Supplemental Heat	When needed
Costs	1978 prices, 1982 prices

The trends exhibited by the Atlanta commercial office building are similar to those exhibited by its Minneapolis counterpart. The one major difference is that in Atlanta there is very little need for auxiliary heat during winter operation due to efficient energy bootstrapping. The results for the Atlanta commercial office building are summarized in Tables 5-48 through 5-50. Table 5-48 presents the characteristics of the Full and Minimum ACES as well as the Cost Optimized system. The Full ACES consists of a well-insulated (R40) 24876 ft^3 cold tank, with no collectors. This system utilizes a water source heat pump with no airside evaporator and no auxilary heating system. The Minimum ACES consists of an insulated (R10) 10302 ft^3 cold tank with a row

Table 5-44. General Economic Analysis Variables for the Commercial Building in Atlanta

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	46%
INTEREST RATE ON BORROWED MONEY	14.5%
PERCENT DOWN PAYMENT	10%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDIT	10%
PROPERTY TAX RATE	2.51%
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%
DEPRECIATION SCHEDULE	STRAIGHT LINE

of unglazed 465 ft^2 solar collectors. This system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Cost Optimized system has an insulated (R10) 10302 ft^3 cold tank, no collector and a dual source evaporator with a 20°F changeover temperature. All of the systems have a 1672 ft^3 hot storage tank.

The component configuration of the Cost Optimized system does not have the capability of interseasonal energy transfer. Thus, the Atlanta Cost Optimized commercial system is essentially an effective load management system operating on a minimum seven-day cycle.

Table 5-45. Economic Input Data: HAVC Installed Costs for the Atlanta Commercial Building

COMPONENT	COST(DOLLARS)
HEATING PLANT	
NATURAL GAS	10230
OIL	9730
ELECTRIC	8770
CHILLER AND COOLING TOWER	54120
HOT WATER HEATER	
NATURAL GAS	1980
ELECTRIC	1840
ICE-MAKER HEAT PUMP	
20KW	7100
35KW	11900
SOLAR COSTS - FIXED + VARIABLE	40888 + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	0
ACES CONTROLS	1000
SUPPLEMENTAL ELECTRIC HEAT	2850
ICE TANK	.24/FT^2 FOR LINER, 4.08/FT^2 FOR WALLS 2.22/FT^2 FOR SLAB, 3.41/ FT^2 FOR COVER 5.63/FT FOR FTG. WHERE NEEDED .54/FT^2 FOR INSULATION

* Price for absorber with a 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5-46. Time-of-Day Service Rates for the Atlanta Commercial Building

	JAN.-DEC.
CUSTOMER CHARGE	$18
ON-PEAK DEMAND CHARGE ($/KW)	3.50
OFF-PEAK DEMAND CHARGE ($/KW)	1.54
ON-PEAK ENERGY CHARGE ($/KWH)	0.0722
OFF-PEAK ENERGY CHARGE ($/KWH)	0.0396
ON-PEAK HOURS	7:00 AM - 9:00 PM Monday - Friday

NOTE: Demand and energy charges were developed by modifying current Georgia Power and Light rates by ratios calculated from analysis of NSP rates. See Commercial Philadelphia rate note for further amplification.

Table 5-49 gives the installed costs of the Full, Minimum and Cost Optimized system as well as the costs of the three conventional systems. These costs have been catagorized by system component. The solar collector cost is the dominant cost of the Minimum ACES, while the tank cost is the dominant cost of the Full ACES. The difference in cost between the ACES and conventional systems is not as great for the commercial system as it was for the residential system.

Table 5-50 presents all costs associated with the Atlanta commercial systems. The conventional systems cost between $64,000 and $67,000 whereas the Full and Minimum ACES cost between $106,000 and $130,000 for the 1978 installation. It is this amortized dollar difference that must be offset by the savings obtained from using the ACES. The energy costs for the commercial systems account for the total energy consumed. This includes all lighting energy.

Table 5-47. Fuel Price and Escalation Rates for the Atlanta Commercial Building

	GAS	OIL	ELECTRICITY
BASE PRICE	$2.35/MCF	30.0¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	6.142	3.597	1.258
PERIOD 2			
DURATION	5	17	17
RATE	4.484	1.192	.542
PERIOD 3			
DURATION	12	-	-
RATE	1.192	-	-
PERIOD 4			
DURATION	-	-	-
RATE	-	-	-

+ Escalation rate in percent per year
* Number of years that rate applies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

Table 5-48. Alternative ACES Configurations for the Atlanta Commercial Building

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED SYSTEM *	
			1978	1982
TANK INSULATION	R40	R10	R10	R10
TANK SIZE (FT^3)	24876	10302	10302	10302
COLLECTOR AREA (FT^2)	NONE	465	NONE	NONE
COLLECTOR TYPE	NONE	ABSORBER	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	DUAL SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	20°	20°
ELECTRIC BACKUP	NONE	NONE	YES	YES

*Cost optimized configuration does not classify as an ACES.

Table 5-49. Comparison of Initial Capital Costs (1978 Dollars) of the ACES and Conventional Systems for the Atlanta Commercial Building

SYSTEM / COMPONENT	FULL ACES	MINIMUM ACES	COST OPTIMIZED SYSTEM	GAS	OIL	ELECTRIC
HEATING PLANT[1]	45100	45100	45100	10230	9730	8770
COOLING PLANT[1]	-	-	-	54120	54120	54120
HOT WATER HEATER	-	-	-	1980	1840	1840
SOLAR COSTS	-	48104	-	-	-	-
ACES CONTROLS	10000	10000	10000	-	-	-
TANK (HOT & COLD)	51383	26549	26549	-	-	-
TOTAL	106483	129753	81649	66330	65690	64730

1 Conventional system costs include associated controls

Table 5-50. Economic Characteristics of HVAC/DHW Systems for the Atlanta Commercial Building

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS)		ANNUALIZED COSTS (DOLLARS)				YEARLY ENERGY* USE (BTU X 10^{-9})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC[1]	66330	66330	28749	26890	44297	41587	.400/.646	1.0/1.0	1.32/1.12
OIL/ELECTRIC[2]	65690	65690	28629	26657	44141	41233	.447/.692	.741/.741	1.55/1.29
ELECTRIC/ELECTRIC[2]	64730	64730	32350	30095	50635	47217	1.00/1.24	0.0	1.84/1.48
FULL ACES[3]	106483	104292	23905	21771	36755	34278	.577/.762	0.0	3.20/2.42
MINIMUM ACES[3]	129753	107589	25073	22420	38025	35425	.666/.850	0.0	2.77/2.17
COST OPTIMIZED SYSTEM[3]	81649	78654	23694	21459	38115	35508	.644/.830	0.0	2.87/2.23

NOTES
1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

*First figure denotes consumption without distribution energy; second figure includes distribution energy.

Generally, this type of energy consumption is not included in a comparative analysis, but it is imperative to do so in the commercial sector since the total energy consumed has a significant effect on the energy rate charged. The energy costs for the commercial systems are greater than the total costs. The reason for this occurrence is that the energy costs represent the total cost of all energy consumed and the total cost takes into account the tax deduction allowable for operating expenses. Thus a significant portion of the money paid for the operating expenses, (energy costs) will be refunded to the building owner thereby lowering his effective total costs. The COPs shown in this table have the same definition as those given for the residential buildings; therefore, only the energy supplied to the HVAC plants are given in the energy use column of Table 5-50. Two values of energy use and COP are given for all commercial applications. The first value of energy consumption and corresponding COP do not include air distribution consumption. The second value does. These two-valued columns are included to illustrate the massive amounts of energy consumed by the air distribution system in the commercial building, and the effect that this distribution energy consumption has on the apparent energy effectiveness of the ACES. Clearly, large amounts of electrical energy (COP=1) supplied to a system with a low COP will not severely affect the resultant annual COP. This statement is true for all the conventional systems. However, supplying large amounts of electrical energy to a system with a high COP (such as the ACES) severely degrades the resultant annual COP. The commercial ACES are very efficient in meeting the building loads (high COP excluding the air distribution energy); however, when the distribution energy is included the energy effectiveness of the commercial ACES becomes overshadowed.

Table 5-50 shows that the initial capital cost of the Cost Optimized commercial system is significantly less than the cost of the Full or Minimum ACES and substantially more than the cost of conventional systems. The Minimum ACES is cost-competitive with all of the conventional systems. The Full ACES is far superior to all of the conventional systems. Finally, the Cost-Optimized system is economically superior to all conventional systems as well as the Full and Minimum ACES.

5.2.4 Atlanta Summary

Tables 5-51 and 5-52 present the break-even and payback duration of the ACES and conventional systems for the single-family residence, multi-family residence and commercial office building in Atlanta. The results are given for the 1982 installation of the single-family and multi-family ACES and the 1978 installation of the commercial ACES. These installation dates are used since the costs associated with the respective dates are felt to be representative costs of the mature systems.

These tables demonstrate that the single-family Full and Minimum ACES are economically superior to all conventional systems except gas. The Minimum ACES is also slightly superior to the Full ACES. The single-family Cost Optimized ACES is superior to all conventional systems as well as to the Full and Minimum ACES. These tables also indicate that the multi-family Full, Minimum and Cost Optimized ACES are superior to all conventional systems except gas. Finally, the commercial Full and Minimum ACES are superior to all conventional systems and the Full ACES is superior to the Minimum ACES (due to the high cost of installed collectors). The Cost Optimized commercial system is also superior to all conventional systems and is more cost-effective than the Full ACES.

5.3 PHILADELPHIA RESULTS

Figure 5-17 shows the effect of the type of collector system on the required collector area and on the collector purchase price for the single-family Minimum ACES in Philadelphia. Increasing the number of covers from zero to one increases the required collector area as well as the total purchase price. Increasing the number of covers from one to two results in a further

Table 5-51. Life Cycle Cost Analysis Break-Even Duration (Years) for the Mature (1978 Commercial/1982 Residential) ACES with Respect to the Conventional Systems for Atlanta

ACES \ CONVENTIONAL SYSTEM	ELECTRIC	OIL	GAS	HEAT PUMP
SINGLE-FAMILY RESIDENCE				
FULL ACES	8	15	-	8
MINIMUM ACES	6	10	-	6
COST OPTIMIZED ACES	4	3	20	5
MULTI-FAMILY RESIDENCE				
FULL ACES	9	11	-	9
MINIMUM ACES	9	11	-	8
COST OPTIMIZED ACES	9	11	-	9
COMMERCIAL BUILDING				
FULL ACES	7	7	7	N/A
MINIMUM ACES	10	12	12	N/A
COST OPTIMIZED SYSTEM	6	7	7	N/A

\- Duration greater than 20 years

N/A Not applicable

Table 5-52. Life Cycle Cost Analysis Payback Duration (Years) for the Mature (1978 Commercial, 1982 Residential) ACES with Respect to the Conventional Systems for Atlanta

ACES \ CONVENTIONAL SYSTEM	ELECTRIC	OIL	GAS	HEAT PUMP
SINGLE-FAMILY RESIDENCE				
FULL ACES	15	18	-	18
MINIMUM ACES	13	13	-	17
COST OPTIMIZED ACES	10	10	20	14
MULTI-FAMILY RESIDENCE				
FULL ACES	11	12	-	10
MINIMUM ACES	10	12	-	9
COST OPTIMIZED ACES	11	12	-	10
COMMERCIAL BUILDING				
FULL ACES	10	11	11	N/A
MINIMUM ACES	13	16	16	N/A
COST OPTIMIZED SYSTEM	10	11	11	N/A

- Duration greater than 20 years

N/A Not applicable

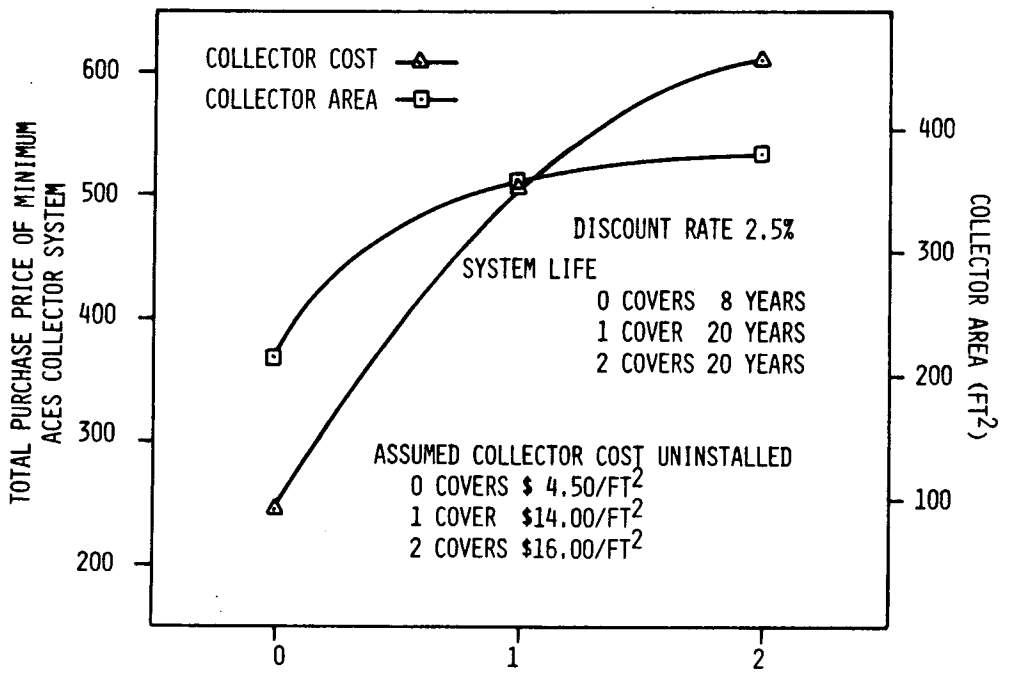


Figure 5-17. Area and Purchase Price of the Uninstalled Collector System as a Function of the Number of Collector Covers for the Philadelphia Single-Family Minimum ACES

increase in the required collector area and in the total purchase price. Therefore, the unglazed system is best suited for the operation of the ACES in Philadelphia.

5.3.1 Results For The Single-Family Residence

Table 5-53 gives the heating, cooling and hot water load profiles for the single-family residence located in Philadelphia. (This residence is fully described in Appendix E.) The annual heating, cooling and hot water loads are 50, 13, and 21 million BTUs/year, respectively. Also shown in this table are the peak heating and cooling loads, the design load, and the steady state performance data of the ACES component train. This information, together with the information generated by the computer programs

Table 5-53. Load and ACES Equipment Performance Data for the Philadelphia Single-Family Residence

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	128	0	20
FEBRUARY	88	0	18
MARCH	76	0	20
APRIL	37	1	19
MAY	9	6	18
JUNE	1	33	16
JULY	0	46	14
AUGUST	0	29	15
SEPTEMBER	3	16	15
OCTOBER	25	0	16
NOVEMBER	50	0	17
DECEMBER	85	0	19

PEAK HEATING LOAD 30,566 BTU/HR

PEAK COOLING LOAD 25,492 BTU/HR

DESIGN HEATING LOAD 38,000 BTU/HR

CAPACITY OF WATER-TO-AIR HEAT PUMP 38,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE; 70/67° ENTERING AIR TEMPERATURE

INFILTRATION (CFM/FT^2) .12 HOUSE .10 ATTIC

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.55

HEAT PUMP BLOWER POWER 750 WATTS

COOLING COIL BLOWER POWER 300 WATTS

EVAPORATOR PUMP POWER 100 WATTS

COOLING LOOP PUMP POWER 100 WATTS

HOT WATER LOOP PUMP POWER 100 WATTS

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

DESMOD and SIMMOD, is input into the computer program ECOMOD for the economic variables described in Section 4.0. Table 5-54 gives the values of the general economic variables for the single-family residence in Philadelphia. Table 5-55 lists the unit component costs for the ACES and four conventional HVAC systems. The final economic information needed to evaluate the ACES and conventional systems can be found in Tables 5-56 and 5-57. These tables contain the single-family residence time-of-day rate structure, base prices, and escalation rates applied to gas, oil and electricity.

Table 5-54. General Economic Analysis Variables for the Philadelphia Single-Family Residence

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	30.0%
INTEREST RATE ON BORROWED MONEY	10.5%
PERCENT DOWN PAYMENT	10.0%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDITS	10.0%
PROPERTY TAX RATE	2.47%
DEPRECIATION SCHEDULE	NONE
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%

Table 5-55. Economic Input Data: HVAC Installed Costs for the Philadelphia Single-Family Residence

COMPONENT	COST (DOLLARS)
HEATING PLANT	
NATURAL GAS	660
OIL	850
ELECTRIC	690
CENTRAL AIR CONDITIONER	850
HOT WATER HEATER	
NATURAL GAS	182
ELECTRIC	170
HIGH EFFICIENCY AIR-AIR HEAT PUMP	2606
ICE-MAKER HEAT PUMP	
DUAL SOURCE	4472
WATER SOURCE	4047
SOLAR COSTS - FIXED + VARIABLE	1580 + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	563
ACES CONTROLS	1150
SUPPLEMENTAL ELECTRIC HEAT	300
ICE TANK	.24/FT^2 FOR LINER, 4.08/FT^2 FOR WALLS 2.22/FT^2 FOR SLAB, 3.41/FT^2 FOR COVER 5.63/FT FOR FTG. WHERE NEEDED .54/FT^2 FOR INSULATION

* Price for absorber with 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5-56. Time-of-Day Service Rates for the Philadelphia Single-Family Residence

	OCT. - MAY	JUNE - SEPT.
CUSTOMER CHARGE PER MONTH	$14.00	$14.00
ON-PEAK ENERGY CHARGE ($/KWH)	0.055	0.134
OFF-PEAK ENERGY CHARGE ($/KWH)	0.028	0.028
ON-PEAK HOURS	8:00 AM - 10:00 PM Monday - Friday	8:00 AM - 10:00 PM Monday - Friday

*These are the experimental rates now available from Consolidated Edison Electric Company.

The parameters used in the evaluation of the ACES and the ranges over which they are varied are shown below for the Philadelphia single-family residence:

Tank	1152 ft^3, 2312 ft^3, 3872 ft^3
Collector	252 ft^2, 162 ft^2, 90 ft^2
Changeover Temperature	71°F, 40°F, 30°F, 25°F, 20°F, 15°F
Tank Insulation	Top: R40; below ground: uninsulated, R40
Costs	1978 prices, 1982 prices

The climate in Philadelphia is not as severe as the climate in Minneapolis nor is it as mild as the climate in Atlanta, so the energy consumption of the Philadelphia single-family ACES can be expected to be less than the Minneapolis ACES but more than the Atlanta ACES. The energy consumption and cost trends of the Philadelphia single-family residence are fairly similar to the trends of the Minneapolis single-family residence. Table 5-58 shows the Philadelphia single-family ACES configurations. The Full ACES consists of an

Table 5-57. Fuel Price and Escalation Rates for the Philadelphia Single-Family Residence

	GAS	OIL	ELECTRICITY
BASE PRICE	$3.22/MCF	48.1¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	4.361	3.022	1.136
PERIOD 2			
DURATION	8	17	17
RATE	2.444	.768	.152
PERIOD 3			
DURATION	2	-	-
RATE	1.647	-	-
PERIOD 4			
DURATION	7	-	-
RATE	.768	-	-

\+ Escalation rate in percent per year
* Number of years that rate applies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

Table 5-58. Alternative ACES Configurations for the Philadelphia Single-Family Residence

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	
			1978*	1982
TANK INSULATION	R40	R40	R40	R40
TANK SIZE (FT^3)	3872	1152	1152	2312
COLLECTOR AREA (FT^2)	90	252	0	0
COLLECTOR TYPE	ABSORBER	ABSORBER	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	DUAL SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	20°	30°
ELECTRIC BACKUP	NONE	NONE	NONE	NONE

*COST OPTIMIZED CONFIGURATION IN 1978 DOES NOT CLASSIFY AS AN ACES.

insulated (R40) 3872 ft^3 tank with an unglazed 90 ft^2 solar collector. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Minimum ACES consists of an insulated (R40) 1152 ft^3 tank with an unglazed 252 ft^2 solar collector. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating.

The 1978 Cost Optimized components show that for the first time the Cost Optimized residential configuration no longer has the capability of interseasonal energy transfer. The reason for this occurence is that with 1978 costs, the Philadelphia rate structures (Consolidated Edison) ensure the use of an effective load management system. Thus, the 1978 Cost Optimized system cannot be classified as an ACES. This system utilizes a dual source evaporator with a 20°F changeover temperature, no solar collector, no auxiliary heat and a well-insulated (R40) 1152 ft^3tank. As costs drop (1982 installation), however, it becomes cost-effective to install a larger more energy efficient system. The 1982 Cost Optimized configuration does have the capability for significant interseasonal energy transfer and therefore can be classified as an ACES. The 1982 Cost Optimized ACES utilizes a dual source evaporator with a 30°F changeover temperature, no solar collector, no auxiliary heat and well-insulated (R40) 2312 ft^3 tank.

Table 5-59 lists the installed costs of the Full and Minimum ACES, the Cost Optimized system and the four conventional systems. These costs have been catagorized by system component. This table illustrates the large cost of the ACES component and the difference in costs between the ACES and conventional components.

Table 5-60 shows the costs associated with the Philadelphia single-family ACES as well as the costs associated with the conventional system. Note that the conventional systems cost between $1600 and $2800 while the Full and Minimum ACES cost between $14,000 and $16,000 for 1978 installations.

Table 5-59. Comparison of Initial Capital Costs (1978) of the ACES and Conventional Systems for the Philadelphia Single-Family Residence

COMPONENT \ SYSTEM	FULL ACES	MINIMUM ACES	COST OPTIMIZED SYSTEM	GAS	OIL	ELECTRIC	HEAT PUMP
HEATING PLANT[1]	4047	4047	4472	660	850	690	2606
COOLING PLANT[1]	-	-	-	850	850	850	-
HOT WATER HEATER	-	-	-	182	170	170	170
SOLAR COSTS	3666	6401	-	-	-	-	-
ACES CONTROLS	1150	1150	1150	-	-	-	-
TANK	7036	3024	3024	-	-	-	-
TOTAL	15899	14622	8646	1692	1870	1710	2776

1 Conventional system costs include associated controls

Table 5-60. Economic Characteristics of HVAC/DHN Systems for the Philadelphia Single-Family Residence

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS)		ANNUALIZED COSTS (DOLLARS)				YEARLY ENERGY USE (BTU X 10^{-6})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC [1]	1692	1692	1017	970	883	848	8.25	105.48	.74
OIL/ELECTRIC [2]	1870	1870	1226	1136	1077	1002	29.10	65.13	.87
AIR-TO-AIR HEAT PUMP [2]	2776	2776	1253	1156	1633	951	50.18	0.0	1.68
ELECTRIC/ELECTRIC [2]	1710	1710	1485	1366	1356	1243	77.34	0.0	1.09
FULL ACES [3]	15899	7631	1700	1029	530	488	24.55	0.0	3.43
MINIMUM ACES [3]	14622	7018	1668	1042	560	516	27.17	0.0	3.10
COST OPTIMIZED ACES [3]*	8646	5032	1250	895	623	533	32.29/28.72	0.0	2.61/2.93

NOTES
1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

*Cost optimized configuration in 1978 does not classify as an ACES.

It is this amortized dollar difference that must be offset by the savings obtained by using the ACES. The 1978 annualized energy and total costs show that the Cost Optimized system is cost-competitive with both electric source conventional systems. The corresponding 1982 energy and total costs show that the Full and Minimum ACES are superior to all conventional systems except gas. The Cost Optimized ACES is superior to all systems including the Full and Minimum ACES. The COP column shows that the Full ACES is by far the most energy efficient system. Decreasing the interseasonal energy transfer (decreasing the size of the storage tank; i.e., Minimum ACES) results in a significant degradation of the seasonal COP. Comparing the annualized costs of the Full and Cost Optimized ACES shows that it is better to sacrifice energy savings than to increase capital costs. Hence, the maximum interseasonal energy transfer savings of the Full ACES do not economically justify the required capital costs. As energy costs drop (1978 to 1982), the energy effectiveness of the Cost Optimized system/ACES increases (COP goes from 2.61 to 2.93).

5.3.2 Results For The Multi-Family Residence

Table 5-61 gives the heating, cooling and hot water load profiles for the multi-family residence located in Philadelphia. (This residence is fully described in Appendix E.) The annual heating, cooling and hot water loads are 189, 120, and 146 million BTUs/year, respectively. Also shown in this table are the peak heating and cooling loads, the design load, and the steady state performance data of the ACES component train. This information, together with the information generated by the computer programs DESMOD and SIMMOD, is input into ECOMOD for the economic evaluation of the ACES. Also input into ECOMOD are the economic variables described in Section 4.0. Table 5-62 gives the values of the general economic variables for the single-family residence in Philadelphia. Table 5-63 lists the unit component costs for the ACES and four conventional HVAC systems. The final economic information

Table 5-61. Load and ACES Equipment Performance Data for the Philadelphia Multi-Family Residence

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	543	0	146
FEBRUARY	349	0	130
MARCH	287	1	140
APRIL	114	30	139
MAY	24	97	127
JUNE	2	274	111
JULY	0	359	99
AUGUST	0	256	102
SEPTEMBER	7	171	104
OCTOBER	67	10	112
NOVEMBER	165	0	119
DECEMBER	334	0	133

PEAK HEATING LOAD 142,731 BTU/HR

PEAK COOLING LOAD 165,354 BTU/HR

DESIGN HEATING LOAD 180,000 BTU/HR

CAPACITY (COOLING) OF WATER-TO-AIR HEAT PUMP 168,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE; 70/67° ENTERING AIR TEMPERATURE

INFILTRATION (CFM/FT^2) .1

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.55

HEAT PUMP BLOWER POWER 4520 WATTS

COOLING COIL BLOWER POWER 1600 WATTS

EVAPORATOR PUMP POWER 702 WATTS

COOLING LOOP PUMP POWER 702 WATTS

HOT WATER LOOP PUMP POWER 560 WATTS

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

Table 5-62. General Economic Analysis Variable for the Philadelphia Multi-Family Residence

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	35%
INTEREST RATE ON BORROWED MONEY	14.5%
PERCENT DOWN PAYMENT	10%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDIT	10%
PROPERTY TAX RATE	2.47%
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%
DEPRECIATION SCHEDULE	STRAIGHT LINE

needed to evaluate the ACES and conventional system can be found in Tables 5-64 and 5-65. These tables contain the multi-family residence time-of-day rate structures, base prices, and escalation rates applied to gas, oil and electricity.

The parameters used in the evaluation of the ACES and the ranges over which they are varied are shown below for the Philadelphia multi-family residence:

Tank	5000 ft^3, 10368 ft^3, 19208 ft^3, 33800 ft^3
Collector	1188 ft^2, 648 ft^2, 108 ft^2, 0 ft^2
Changeover Temperature	71°F, 40°F, 30°F, 25°F, 20°F, 15°F
Tank Insulation	Top: R40; below ground: uninsulated, R40
Costs	1978 prices, 1982 prices

Table 5-63. Economic Input Data: HVAC Installed Costs for the Philadelphia Multi-Family Residence

COMPONENT	COST (DOLLARS)
HEATING PLANT	
NATURAL GAS	2231
OIL	2389
ELECTRIC	2562
CENTRAL AIR CONDITIONER	6740
HOT WATER HEATER	
NATURAL	1980
ELECTRIC	1840
HIGH EFFICIENCY AIR-AIR HEAT PUMP	12726
ICE-MAKER HEAT PUMP	
DUAL SOURCE	18508
WATER SOURCE	17186
SOLAR COSTS - FIXED + VARIABLE	1900 + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	5000
ACES CONTROLS	4500
SUPPLEMENTAL ELECTRIC HEAT	785
ICE TANK	.24/FT^2 FOR LINER, 4.08/FT^2 FOR WALLS
	2.22/FT^2 FOR SLAB, 3.41/FT^2 FOR COVER
	5.63/FT FOR FTG. WHERE NEEDED
	.54/FT^2 FOR INSULATION

* Price for absorber with a 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5-64. Time-of-Day Service Rates for the Philadelphia Multi-Family Residence

	OCT. - MAY	JUNE - SEPT.
CUSTOMER CHARGE PER MONTH	$13.00	$13.00
ON-PEAK ENERGY CHARGE ($/KWH)	0.069	0.159
OFF-PEAK ENERGY CHARGE ($/KWH)	0.028	0.028
ON-PEAK HOURS	8:00 AM - 10:00 PM Monday - Friday	8:00 AM - 10:00 PM Monday - Friday

The trends of the multi-family residence located in Philadelphia are basically the same as those of the single-family residence; however, energy consumption and costs have increased due to the physical requirements of the structure. The multi-family to single-family loads ratio has shifted; hence, the COP of the multi-family Full ACES is greater than the COP of the single-family Full ACES. Table 5-66 shows the Philadelphia multi-family ACES configurations. The Full ACES consists of an insulated (R40) 33,800 ft^3 tank with an unglazed 108 ft^2 solar collector. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Minimum ACES consists of an insulated (R40) 5,000 ft^3 tank with an unglazed 1188 ft^2 solar collector. The system utilizes a water source heat pump with no airside evaporator and no auxiliary heating.

The 1978 Cost Optimized components show that contrary to the cases in Minneapolis and Atlanta the Cost Optimized multi-family residence configuration no longer has the capability of interseasonal energy transfer. The reason for this occurrence is that with 1978 costs, the Philadelphia rate structures (Consolidated Edison) ensure the use of an effective load management system. Thus the 1978 Cost Optimized system cannot be classified as an ACES. This system utilizes a dual source evaporator with a 20°F changeover temperature, no solar collector, no auxiliary heat and a well-insulated (R40) 5000 ft^3 tank. As costs drop (1982 installation), however, it becomes cost-effective to install a larger, more energy efficient system. The 1982 Cost Optimized

Table 5-65. Fuel Price and Escalation Rates for the Philadelphia Multi-Family Residence

	GAS	OIL	ELECTRICITY
BASE PRICE	$2.85/MCF	32¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	4.217	4.339	.938
PERIOD 2			
DURATION	5	17	17
RATE	2.795	1.061	.176
PERIOD 3			
DURATION	12	-	-
RATE	1.061	-	-
PERIOD 4			
DURATION	-	-	-
RATE	-	-	-

\+ Escalation rate in percent per year
* Number of years that rate applies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

Table 5-66. Alternative ACES Configurations for the Philadelphia Multi-Family Residence

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	
			1978*	1982
TANK INSULATION	R40	R40	R40	R40
TANK SIZE (FT^3)	33800	5000	5000	10368
COLLECTOR AREA (FT^2)	108	1188	0	0
COLLECTOR TYPE	ABSORBER	ABSORBER	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	DUAL SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	20°	30°
ELECTRIC BACKUP	NONE	NONE	NONE	NONE

*COST OPTIMIZED CONFIGURATION IN 1978 DOES NOT CLASSIFY AS AN ACES.

configuration does have the capability for significant interseasonal energy transfer and therefore can be classified as an ACES. The 1982 Cost Optimized ACES utilizes a dual source evaporator with a 30°F changeover temperature, no solar collector, no auxiliary heat and a well-insulated (R40) 10,368 ft^3 tank.

Table 5-67 lists the installed costs of the Full and Minimum ACES, the Cost Optimized system and the four conventional systems. These costs have been categorized by system component. This table illustrates the large costs of the ACES components and the difference in costs between the ACES and conventional components.

Table 5-68 shows the costs associated with the Philadelphia multi-family ACES as well as the costs associated with the conventional systems. Note that conventional systems cost between $10,000 and $15,000 while one Full and Minimum ACES cost between $57,000 and $71,000 for 1978 installations. It is this amortized dollar difference that must be offset by the savings obtained by using the ACES. The energy costs for the multi-family residence can be greater than the total costs. The reason for this occurrence is that the energy costs represent the total cost of all energy consumed and the total cost takes into account the tax deduction allowable for operating expenses. Thus, a significant portion of the money paid for the operating expenses (energy costs) will be refunded to the building owner, thereby lowering his effective total costs. The 1978 annualized energy and total costs show that the Cost Optimized system is cost-competitive with all conventional systems except gas. The corresponding 1982 energy and total cost show that the Full and Minimum ACES are superior to all systems including the Full and Minimum ACES. The COP column shows that the Full ACES is by far the most energy efficient system. Decreasing the interseasonal energy transfer (decreasing the size of the storage tank; i.e., Minimum ACES) results in a significant degradation of the seasonal COP. Comparing the COP of the Full and Cost Optimized ACES shows that it is better to sacrifice energy savings rather than increase capital costs. Hence, the maximum interseasonal energy transfer savings of the Full ACES do not economically justify the required costs. As energy costs drop (1978 to 1982)

Table 5-67. Comparison of Initial Capital Costs (1978 Dollars) of the ACES and Conventional Systems for the Philadelphia Multi-Family Residence

COMPONENT \ SYSTEM	FULL ACES	MINIMUM ACES	COST OPTIMIZED SYSTEM	GAS	OIL	ELECTRIC	HEAT PUMP
HEATING PLANT[1]	17186	17186	18508	2231	2389	2562	12726
COOLING PLANT[1]	-	-	-	6740	6740	6740	-
HOT WATER HEATER	-	-	-	1980	1840	1840	1840
SOLAR COSTS	8952	26994	-	-	-	-	-
ACES CONTROLS	4500	4500	4500	-	-	-	-
TANK	40182	8510	8510	-	-	-	-
TOTAL	70820	57190	31518	10951	10969	11142	14566

1 Conventional system costs include associated controls

Table 5-68. Economic Characteristics of HVAC/DHW Systems for the Philadelphia Multi-Family Residence

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS)		ANNUALIZED COSTS (DOLLARS)				YEARLY ENERGY USE (BTU X 10^{-6})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC [1]	10951	10951	3492	3287	4046	3855	61.4	490.2	.82
OIL/ELECTRIC[2]	10969	10969	4837	4464	6113	5664	207.1	257.2	.98
AIR-TO-AIR HEAT PUMP[2]	14566	14566	5139	4710	6142	5648	276.9	0.0	1.64
ELECTRIC/ELECTRIC[2]	11142	11142	5857	5374	7660	7045	389.2	0.0	1.16
FULL ACES[3]	70820	33993	6786	3817	2392	2200	123.3	0.0	3.69
MINIMUM ACES[3]	57190	27451	6162	3709	2687	2471	151.7	0.0	2.99
COST OPTIMIZED ACES[3]*	31518	18278	4475	3039	2989	2546	174.9/157.2	0.0	2.6/2.89

NOTES

1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

*Cost Optimized configuration in 1978 does not classify as an ACES.

the energy effectiveness of the Cost Optimized system/ACES increases (COP goes from 2.61 to 2.89).

5.3.3 Commercial Results

Table 5-69 gives the heating, cooling and hot water load profiles for the commercial office building located in Philadelphia. (This building is described in Appendix E.) The annual heating, cooling and hot water loads are 784, 849, and 50 million BTUs/year, respectively. Also shown in this table are the peak heating and cooling loads, the design load and the steady state performance data of the ACES component train. This information, together with the information generated in CSIZE and ACESIM, is input into ECOMOD for this economic evaluation of the ACES. Also input into ECOMOD are the economic variables described in Section 4.0. Table 5-70 gives the values of the general economic variables for the commercial office building located in Philadelphia. Table 5-71 provides the unit component costs for the ACES and the three conventional HVAC systems. The final information needed to evaluate the ACES and conventional systems can be found in Tables 5-72 and 5-73. These tables contain the commercial building time-of-day rate structures, base prices, and escalation rates applied to oil, gas and electricity.

The parameters used in the evaluation of the ACES and the ranges over which they are varied are shown below for the Philadelphia commercial office building:

Tank	78732 ft^3, 48166 ft^3, 17603 ft^3
Collector	3628 ft^2, 0 ft^2
Changeover Temperature	71^oF, 20^oF

Table 5-69. Load and ACES Equipment Performance Data for the Philadelphia Commercial Building

MONTHLY LOADS (BTU X 10^{-5})

	HEATING	COOLING	HOT WATER
JANUARY	2218	26	49
FEBRUARY	1308	88	42
MARCH	1147	142	48
APRIL	521	172	45
MAY	173	646	45
JUNE	22	1696	39
JULY	3	1547	35
AUGUST	10	1471	39
SEPTEMBER	80	1152	35
OCTOBER	281	859	40
NOVEMBER	865	506	41
DECEMBER	1215	179	42

PEAK HEATING LOAD 1,046,200 BTU/HR

PEAK COOLING LOAD 1,558,900 BTU/HR

CAPACITY OF WATER-TO-WATER HEAT PUMP 1,126,000 BTU/HR AT 32° ENTERING WATER TEMPERATURE COLD SIDE; 120° ENTERING WATER TEMPERATURE HOT SIDE

ACES STEADY STATE PERFORMANCE DATA

COP OF WATER-TO-AIR HEAT PUMP (COMPRESSOR ONLY AT DESIGN CONDITIONS) 3.20

AIR DISTRIBUTION SYSTEM POWER 2100 X 10^5 BTU/YEAR

CONDENSER PUMP POWER } 10 KW
COOLING LOOP PUMP POWER }

HOT WATER LOOP PUMP POWER NONE

PANEL PUMP POWER .75 WATTS/FT^2 OF COLLECTOR

Table 5-70. General Economic Analysis Variables for the Philadelphia Commercial Building

VARIABLE	VALUE
LENGTH OF ANALYSIS PERIOD	20 YEARS
REAL DISCOUNT RATE	2.5%
INFLATION RATE	6.0%
INCOME TAX RATE	46%
INTEREST RATE ON BORROWED MONEY	14.5%
PERCENT DOWN PAYMENT	10%
NUMBER OF LOAN REPAYMENTS PER YEAR	12
ENERGY CONSERVATION TAX CREDIT	10%
PROPERTY TAX RATE	2.47%
MAINTENANCE AND REPAIR	1.0%
INSURANCE	0.4%
DEPRECIATION SCHEDULE	STRAIGHT LINE

Tank Insulation	R40 for Full ACES, others R10
Supplemental Heat	When needed
Costs	1978 prices, 1982 prices

The trends exhibited by the Philadelphia commercial office building are similar to those exhibited by its Minneapolis counterpart. The results for the Philadelphia commercial office building are summarized in Tables 5-74 through 5-76. Table 5-74 presents the characteristics of the Full and Minimum ACES as well as the Cost Optimized system. The Full ACES consists

Table 5-71. Economic Input Data: HVAC Installed Costs for the Philadelphia Commercial Building

COMPONENT	COST (DOLLARS)
HEATING PLANT	
NATURAL GAS	10765
OIL	9980
ELECTRIC	10403
CHILLER AND COOLING TOWER	44385
HOT WATER HEATER	
NATURAL GAS	1980
ELECTRIC	1840
ICE-MAKER HEAT PUMP	
20KW	7100
35KW	11900
SOLAR COSTS - FIXED + VARIABLE	40888 + 19.65/FT^2 (9.50/FT^2)*
NONSOLAR FIXED COSTS	0
ACES CONTROLS	10000
SUPPLEMENTAL ELECTRIC HEAT	2850
ICE TANK	.24/FT^2 FOR LINER, 4.08/FT^2 FOR WALLS 2.22/FT^2 FOR SLAB, 3.41/FT^2 FOR COVER 5.63/FT FOR FTG. WHERE NEEDED .54/FT^2 FOR INSULATION

* Price for absorber with a 10-year life

NOTE: Ducting common to ACES and conventional system has not been included in costs.

Table 5.72. Time-of-Day Rates for the Philadelphia Commercial Building

	JUNE-SEPT.	OCT.-MAY
CUSTOMER CHARGE	$10.76	$10.76
ON-PEAK DEMAND CHARGE ($/KW)	6.52	5.24
OFF-PEAK DEMAND CHARGE ($/KW)	1.95	1.95
ON-PEAK ENERGY CHARGE ($/KWH)	0.0221	0.0221
OFF-PEAK ENERGY CHARGE ($/KWH)	0.0134	0.0134
ON-PEAK HOURS	7:00 AM - 7:00 PM Monday - Friday	7:00 AM - 7:00 PM Monday - Friday

NOTES: Rates were developed from analysis of commercial bills resulting from standard rate structure adjusted for the Primary Power and Light on-peak to off-peak demand charge differential. Comparison of NSP customer charges under standard rates and proposed TOD rates was used to define additional monthly metering charges. Ratio of proposed on-peak energy charges to highest price standard rate energy charge at NSP was used to define on-peak charges at Philadelphia. The same approach was used for off-peak rates with the exception that the standard per kWh price was now taken to be the tail block unit price.

of a well-insulated (R40) 78,732 ft^3 cold tank, with no collectors. This system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Minimum ACES consists of an insulated (R10) 17,603 ft^3 cold tank with a row of unglazed 3628 ft^2 solar collectors. This system utilizes a water source heat pump with no airside evaporator and no auxiliary heating system. The Cost Optimized system has an insulated (R10) 17,603 ft^3 cold tank, no collector and a dual source evaporator with a 20°F changeover temperature. All of the systems have a 1651 ft^3 hot storage tank.

Table 5-73. Fuel Price and Escalation Rates for the Philadelphia Commercial Building

	GAS	OIL	ELECTRICITY
BASE PRICE	$2.85/MCF	32¢/GAL	SEE TOD** RATES
PERIOD 1			
DURATION*	7	7	7
RATE+	4.217	4.339	.938
PERIOD 2			
DURATION	5	17	17
RATE	2.795	1.061	.176
PERIOD 3			
DURATION	12	-	-
RATE	1.061	-	-
PERIOD 4			
DURATION	-	-	-
RATE	-	-	-

+ Escalation rate in percent per year
* Number of years that rate a-plies
** Time-Of-Day

SOURCE: Historic and Forecasted Energy Prices by DOE Region and Fuel Type for Three Macroeconomic Scenarios and One Imported Oil Price Escalation Scenario (AM/EU/79-06) published by the Energy Information Administration, November 1978.

Table 5-74. Alternative ACES Configurations for the Philadelphia Commercial Building

SYSTEM CHARACTERISTICS	FULL ACES	MINIMUM ACES	COST OPTIMIZED SYSTEM *	
			1978	1982
TANK INSULATION	R40	R10	R10	R10
TANK SIZE (FT^3)	78732	17603	17603	17603
COLLECTOR AREA (FT^2)	0	3628	0	0
COLLECTOR TYPE	NONE	ABSORBER	NONE	NONE
HEAT PUMP TYPE	WATER SOURCE	WATER SOURCE	DUAL SOURCE	DUAL SOURCE
CHANGE OVER TEMPERATURE (^oF)	NONE	NONE	20°	20°
ELECTRIC BACKUP	NONE	NONE	YES	YES

*Cost Optimized configuration does not classify as an ACES.

The component configuration of the Cost Optimized system does not have the capability of interseasonal energy transfer. Thus, the Philadelphia Cost Optimized commercial system is essentially an effective load management system operating on a minimum seven-day cycle. Table 5-75 gives the installed costs of the Full, Minimum and Cost Optimized system as well as the costs of the three conventional systems. These costs have been catagorized by system component. The solar collector cost is the dominant cost of the Minimum ACES, whereas the tank cost is the dominant cost of the Full ACES. The difference in cost between the ACES and conventional systems is not as great for the commercial system as it was for the residential system.

Table 5-76 presents all costs associated with the Philadelphia commercial systems. The conventional system costs between $56,000 and $57,000, while the Full and Minimum ACES costs between $147,000 and $182,000 for the 1978 installation. It is this amortized dollar difference that must be offset by the savings obtained from using the ACES. The energy costs for the commercial systems account for the total energy consumed. This includes all lighting energy. Generally, this type of energy consumption is not included in a comparative analysis, but it is imperative to do so in the commercial sector since the total energy consumed has a significant effect on the energy rate charged. The energy costs for the commercial systems are greater than the total costs. The reason for this occurrence is that the energy costs represent the total cost of all energy consumed and the total cost takes into account the tax deduction allowable for operating expenses. Thus, a significant portion of the money paid for the operating expenses (energy costs) will be refunded to the building owner, thereby lowering his effective total costs. The COPs shown in Table 5-76 have the same definition as those given for the residential building; therefore, only the energy supplied to the HVAC plants are given in the energy use column of Table 5-76. Two values of energy use and COP are given for all commercial application. The first value of energy consumption and corresponding COP does not include air distribution consumption. The second value does. These two-valued columns

Table 5-75. Comparison of Initial Captial Costs (1978 Dollars) of the ACES and Conventional Systems for the Philadelphia Commercial Building

COMPONENT \ SYSTEM	FULL ACES	MINIMUM ACES	COST OPTIMIZED ACES	GAS	OIL	ELECTRIC
HEATING PLANT[1]	38000	38000	38000	10765	9980	10403
COOLING PLANT[1]	-	-	-	44385	44385	44385
HOT WATER HEATER	-	-	-	1980	1840	1840
SOLAR COSTS	-	96198	-	-	-	-
ACES CONTROLS	10000	10000	10000	-	-	-
TANK (HOT & COLD)	99465	37866	37866	-	-	-
TOTAL	147465	182064	85866	57130	56205	56628

1 Conventional system costs include associated controls

Table 5-76. Economic Characteristics of HVAC/DHW Systems for the Philadelphia Commercial Building

HVAC SYSTEM (HEATING PLANT/ COOLING PLANT)	INITIAL EQUIPMENT COST (DOLLARS		ANNUALIZED COSTS (DOLLARS)				YEARLY ENERGY* USE (BTU X 10^{-9})		COP
			TOTAL		ENERGY				
	1978	1982	1978	1982	1978	1982	ELECTRIC	FOSSIL	
NATURAL GAS/ELECTRIC[1]	57130	57310	18877	17516	30708	28387	.251/.497	1.45/1.45	.980/.860
OIL/ELECTRIC[2]	56205	56205	18288	16886	29686	27471	.304/.550	1.11/1.11	1.19/1.01
ELECTRIC/ELECTRIC[2]	56628	56628	19697	18082	32264	29761	1.14/1.40	0.0 0.0	1.47/1.20
FULL ACES[3]	147465	144465	17659	16073	22798	20966	.487/.697	0.0	3.45/2.41
MINIMUM ACES[3]	182064	138801	19343	16576	24005	22074	.657/.867	0.0	2.55/1.94
COST OPTIMIZED ACES[3]	85867	82867	16576	15085	23267	21396	.619/.829	0.0	2.72/2.03

NOTES
1. Natural gas fired domestic hot water heater
2. Electric domestic hot water heater
3. Desuperheater domestic hot water heater

*First figure denotes consumption without distribution energy; second figure includes distribution energy.

are included to illustrate the massive amounts of energy consumed by the air distribution system in the commercial building, and the effect that this distribution energy consumption has on the apparent energy effectiveness of the ACES. Clearly, large amounts of electrical energy (COP=1) supplied to a system with a low COP will not severely affect the resultant annual COP. This statement is true for the conventional systems. However, supplying large amounts of electrical energy to a system with a high COP (such as the ACES) severely degrades the resultant annual COP. The commercial ACES are very efficient in meeting building loads (high COP excluding the air distribution energy); however, when the distribution energy is included, the energy effectiveness of the commercial ACES becomes overshadowed.

Table 5-76 shows that the initial capital cost of the Cost Optimized commercial system is significantly less than the cost of the Full or Minimum ACES and substantially more than the cost of the conventional systems. It can be seen that the Minimum ACES is cost-competitive with the electric source system. The Full ACES is far superior to all the conventional systems. Finally, the Cost Optimized system is economically superior to all conventional systems as well as to the Full and Minimum ACES.

5.3.4 Philadelphia Summary

Tables 5-77 and 5-78 present the break-even and payback duration of the ACES and conventional systems for the single-family residence, multi-family residence and commercial office building in Philadelphia. The results are given for the 1982 installation of the single-family and multi-family ACES and the 1978 installation of the commercial ACES. These installation dates are used since the costs associated with the respective dates are felt to be representative costs of mature systems.

Table 5-77. Life Cycle Cost Analysis: Break-Even Duration (years) for the Mature (1978 Commercial/1982 Residential) ACES with Respect to the Conventional Systems for Philadelphia

ACES \ CONVENTIONAL SYSTEM	ELECTRIC	OIL	GAS	HEAT PUMP
SINGLE-FAMILY RESIDENCE				
FULL ACES	3	11	-	8
MINIMUM ACES	3	12	-	9
COST OPTIMIZED ACES	1	2	12	1
MULTI-FAMILY RESIDENCE				
FULL ACES	12	16	-	14
MINIMUM ACES	9	12	-	10
COST OPTIMIZED ACES	7	10	17	9
COMMERCIAL BUILDING				
FULL ACES	7	13	12	N/A
MINIMUM ACES	18	-	-	N/A
COST OPTIMIZED ACES	3	3	3	N/A

\- Duration greater than 20 years

N/A Not applicable

Table 5-78. Life Cycle Cost Analysis: Payback Duration (years) for the Mature (1978 Commercial/ 1982 Residential) ACES with Respect to the Conventional Systems for Philadelphia

ACES \ CONVENTIONAL SYSTEMS	ELECTRIC	OIL	GAS	HEAT PUMP
SINGLE-FAMILY RESIDENCE				
FULL ACES	10	16	-	14
MINIMUM ACES	10	16	-	15
COST OPTIMIZED ACES	6	9	16	7
MULTI-FAMILY RESIDENCE				
FULL ACES	15	17	-	16
MINIMUM ACES	11	13	-	12
COST OPTIMIZED ACES	10	11	18	11
COMMERCIAL BUILDING				
FULL ACES	13	15	15	N/A
MINIMUM ACES	19	-	-	N/A
COST OPTIMIZED SYSTEM	10	10	10	N/A

- Duration greater than 20 years

N/A Not applicable

These tables demonstrate that the single-family Full and Minimum ACES are economically superior to all conventional systems except gas. The Full ACES is also slightly superior to the Minimum ACES. The single-family Cost Optimized ACES is superior to all conventional systems as well as to the Full and Minimum ACES. These tables also demonstrate that the multi-family Full and Minimum ACES are superior to all conventional systems except gas, while the Cost Optimized ACES is superior to all systems including the Full and Minimum ACES. Finally, although the commercial Minimum ACES is only competitive with the electric source system, the Full ACES is superior to all conventional systems. The Cost Optimized residential system is far superior to all conventional systems as well as to the Full and Minimum ACES.

5.4 SENSITIVITY ANALYSIS

The preceding analysis has been based on a single set of economic assumptions. Although these assumptions are intended to be reasonable it is inconceivable that the future will exactly match these estimations. Changes in the assumptions will have a direct effect on the annual cost; this will, in turn, affect the payback and ultimately the economic viability of the ACES.

To give an indication of the effect that changes in these assumptions have on the predicted viability of the ACES, a number of perturbations on the economic inputs have been made. Figures 5-18 through 5-21 show the potential changes possible in payback and annual cost of the reference system (the Philadelphia single-family Full ACES) with respect to the conventional electric, heat pump, oil and gas systems. The following cases were investigated:

- Real Discount Rate (DR) 1% - 4%
- Mortgage Rate (MR) 9% - 12%
- Tax Bracket (TB) 25% - 35%
- Maintenance Cost (M) .1% - 3%
- Property Tax (PT) 1% - 4%
- On Period/Off Period energy charge for the primary and secondary seasons (on/off)

.104 - .033	.160 - .022
.044 - .033	.060 - .022

- Electric escalation rate in two blocks (EER)

.568%	4.94%
.076%	2.34%

- Oil escalation rate in two blocks (OER)

1.511%	9.066%
0.364%	2.304%

- Gas escalation rate in four blocks (GER)

2.180%	13.08%
1.222%	7.32%
0.873%	4.94%
0.768%	2.30%

In general, points moving toward the first quadrant (up to the right) in Figures 5-18 throuhg 5-21 are the worst possible situations. These disturbances show an increase in annualized cost and an increase in payback. Points moving toward the third quadrant (down to the left) are the best possible situations. These disturbances represent a decrease in annualized cost and a decrease in payback. Points moving toward the second quadrant (up to the left) represent situations where the payback is decreased but the annualized cost is increased. Points moving toward the fourth quadrant (down to the right) represent situations where the payback is increased but the annualized cost is decreased. Since the Philadelphia single-family Full ACES does not pay back with respect to the conventional gas system, the effects of the perturbed variables are not illustrated, unless the disturbance results in a system payback.

5.4.1 Discount Rate

All figures illustrate that increasing only the discount rate (the discount rate seen by the investor) decreases the annualized cost of the ACES and can decrease the payback duration with respect to conventional systems. Decreasing the discount rate increases the annualized cost and increases the payback period. The range of values used for this parameter shows a change in annualized cost of $70 (-20 to +50) and effects a change in the payback duration of approximately one year for all the conventional systems. Thus the figures indicate that the economic results are fairly insensitive to the assumed value of the discount rate.

5.4.2 Mortgage Rate

All figures illustrate that increasing the mortgage rate increases the annualized cost of the ACES and increases the payback period with respect to the conventional systems. Decreasing the mortgage rate decreases the annualized cost and decreases the payback duration. There is a relatively small change in the annualized cost and payback of the ACES with respect to conventional systems

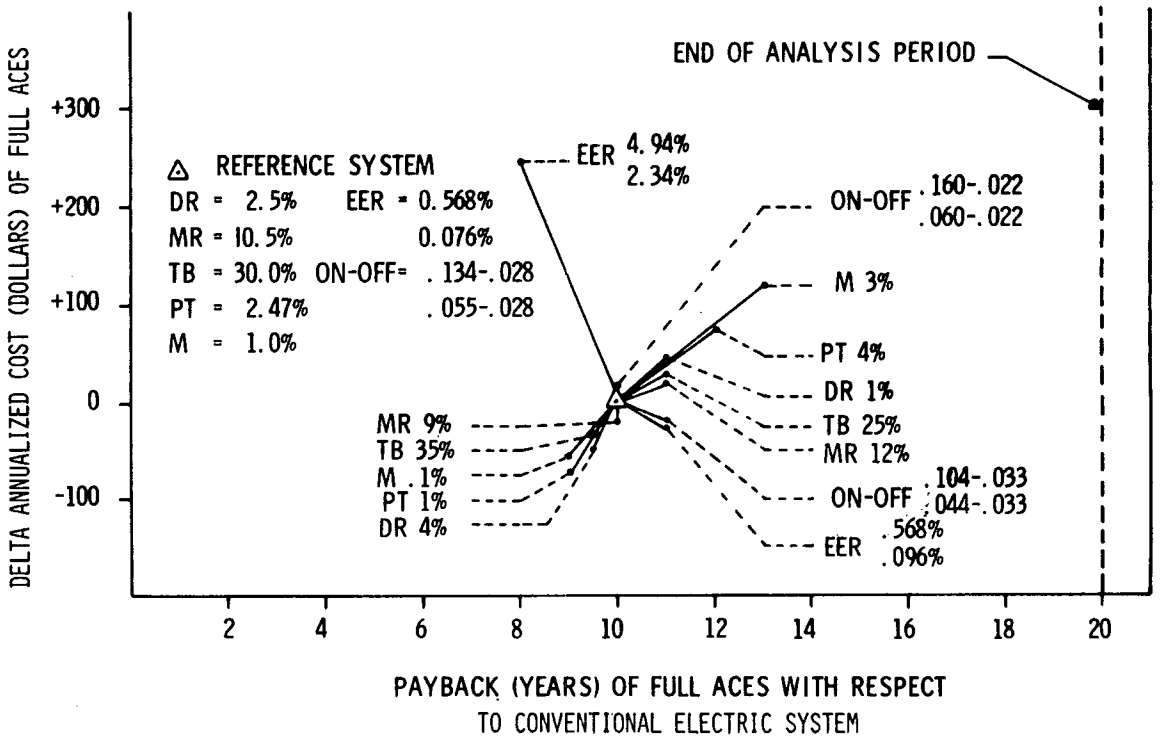


Figure 5-18. Delta Annualized Costs of the Philadelphia Single-Family Residence Full ACES and Payback Duration with Respect to the Conventional Electric System

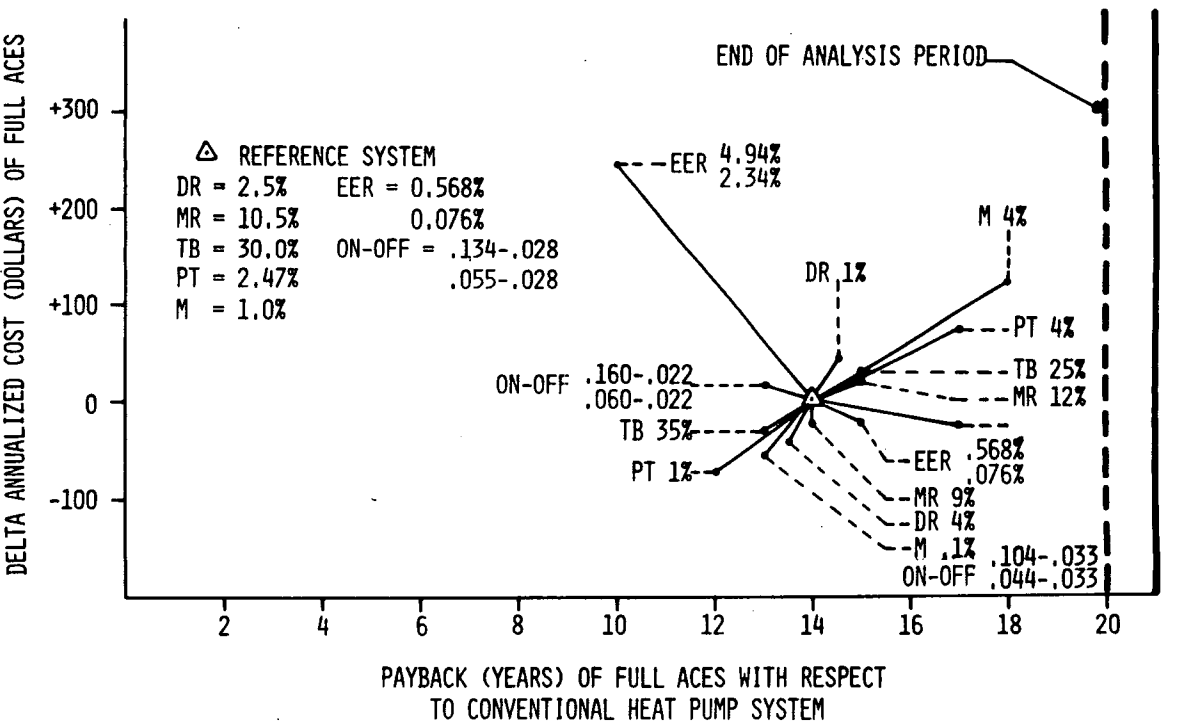


Figure 5-19. Delta Annualized Costs of the Philadelphia Single-Family Residence Full ACES and Payback Duration with Respect to the Conventional Heat Pump System

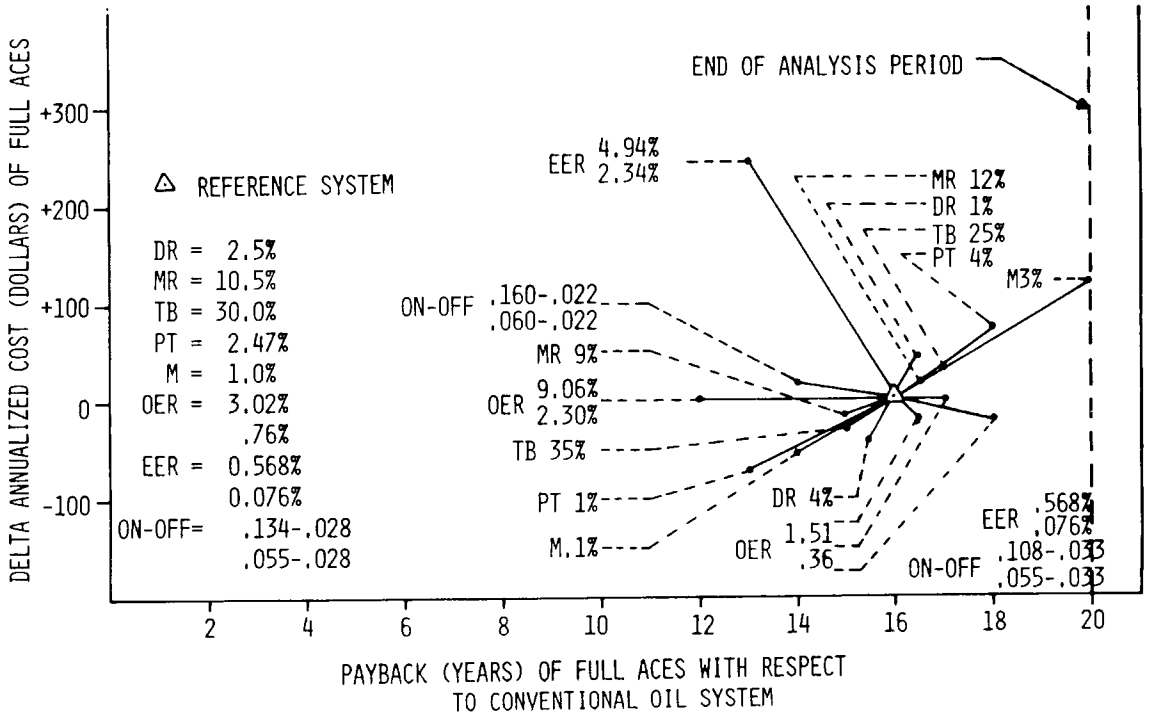


Figure 5-20. Delta Annualized Costs of the Philadelphia Single-Family Residence Full ACES and Payback Duration with Respect to the Conventional Oil System

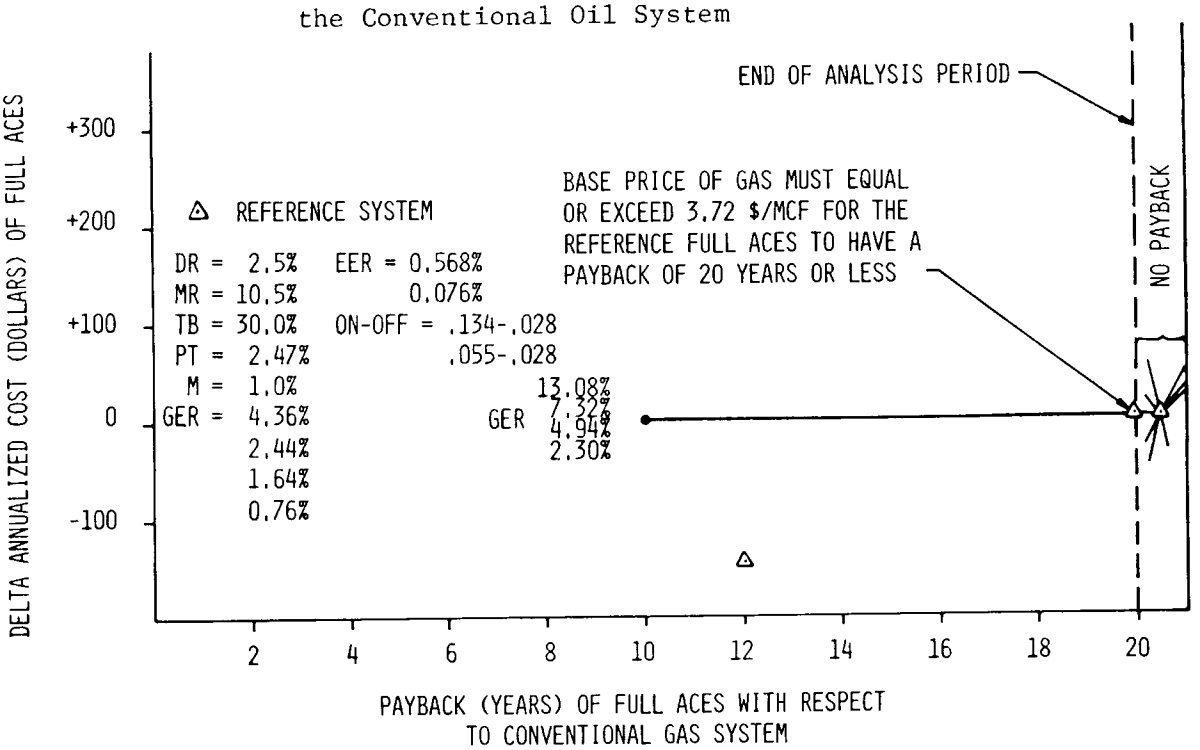


Figure 5-21. Delta Annualized Costs of the Philadelphia Single-Family Residence Full ACES and Payback Duration with Respect to the Conventional Gas System

for the range of values used for this parameter. Thus, the economic results are relatively insensitive to the assumed value of the mortgage rate.

5.4.3 Tax Bracket

All figures show that increasing the tax bracket decreases the annualized cost of the ACES and decreases the payback period with respect to conventional systems. Decreasing the tax bracket increases the annualized cost and increases the payback duration. Again, the changes in the annualized cost and payback period are relatively small over the range of values used for this parameter. The economic results are relatively insensitive to the assumed value of the tax bracket.

5.4.4 Maintenance Cost

Increasing the maintenance cost increases the annualized cost of the ACES and increases the payback duration with respect to conventional systems. Decreasing the maintenance costs decreases the annualized cost as well as the payback period. The range of values used for this parameter shows a change in annualized cost of $175 (-50 to +125) and effects a change in the payback of between four years (9 to 13) for the all-electric system and seven years (13 to 20) for the oil system. This shows that the economic results are extremely sensitive to the assumed value of the maintenance costs.

5.4.5 Property Tax

Increasing the property tax increases annualized cost of the Full ACES and increases the payback period with respect to the conventional systems. Decreasing the property tax decreases the annualized cost and decreases the payback duration. The range of values used for this parameter shows a change in annualized costs of $150 (-75 to +75) and effects a change in the payback period of between three years (9 to 12) for the conventional electric system

and five years (13 to 18) for the conventional oil system. Thus the economic results are very sensitive to the assumed value of the property tax.

5.4.6 On Period/Off Period Energy Charge

The figures show that increasing the difference between on period and off period energy (electric) costs increases the annualized cost of the ACES and decreases the payback with respect to conventional systems. Decreasing the difference between on period and off period energy charges decreases the annualized cost and increases the payback period. The range of values used for this parameter shows a change in the annualized cost of $50 (+25 to -25) and effects a change in the payback of between one year (10 to 11) for the conventional electric system and four years for the oil system. The economic results can be sensitive to the assumed values of the on period and off period energy charge, but the effect of the charge depends upon the conventional system to which it is compared.

5.4.7 Electricity Escalation Rate

Increasing the electricity escalation rate increases the annualized cost of the Full ACES and decreases the payback period with respect to conventional systems. Decreasing the electricity rate decreases the annualized cost of the ACES and increases the payback duration. The range of values used for this parameter shows a change in the annualized cost of $275 (+250 to -25) and effects a change in the payback of between three years (8 to 11) for the conventional electric system and five years (10 to 15) for the conventional heat pump system. The results are therefore very sensitive to the assumed value of the electricity escalation rate.

5.4.8 Oil Escalation Rate

This parameter only affects the economic viability between the ACES and the conventional oil system (Figure 5-20). Increasing the oil escalation rate decreases the payback period of the ACES with respect to the oil system and decreasing the escalation rate increases the payback period. As would be expected, the ACES annualized cost is not a function of the escalation rate. For the range of rates cited, the potential payback changes by five years (+12 to +17). This shows that the economic results are very sensitive to the assumed value of the oil escalation rate.

5.4.9 Gas Escalation Rate

This parameter only affects the economic relationship between the ACES and the conventional gas system (Figure 5-21). Increasing the gas escalation rate drastically decreases the payback period of the ACES with respect to the gas system. Conversely, a decrease in the gas escalation rate would increase the payback period. As would be expected, the ACES annualized cost is not a function of the escalation rate. For the rates cited, the potential payback changes enough to make the Full ACES far superior for the conventional gas system. The reason for this extreme sensitivity is that with the referenced escalation rate, the gas system is superior to the Full ACES. The reference payback period is beyond the life of the expected system and is therefore extremely sensitive to economic perturbations.

5.4.10 Energy and Equipment Cost Variations

Although these parameters have not been specifically changed for the Philadelphia Full ACES, their ramifications can be seen in the results presented for the various systems in the three climatic regions for the 1978 and 1982 installations.

SECTION 6.0
PROJECT SUMMARY

The overall objective of the project was to assess the energy effectiveness and the economic viability of the ACES concept on a regional level. The economic viability was to be assessed by determining the cost competitiveness of the single-family, multi-family, and commercial ACES with other more conventional heating and cooling systems. This section provides the highlights of the results discussed in Section 5.0:

6.1 CONCLUSIONS

The following conclusions can be made with respect to the ACES system presented in this study:

- The Full ACES are the most energy efficient systems studied due to their capability to maximize interseasonal energy transfer.
- The ACES can be three to four times more energy efficient than conventional systems.
- When compared to the Cost Optimized ACES, interseasonal energy savings of the Full ACES do not economically justify the required storage capital costs.
- In general, the residential ACES are not economically competitive with conventional systems at today's prices.
- Using a reduced price scenario, the residential ACES are economically superior to conventional systems (oil, heat pump and electric; sometimes gas); however, the ACES must be highly commercialized before these reduced prices can be realized.
- Reduction in the ACES investment cost results in a more energy efficient cost optimized system.
- The energy savings capability of the ACES, with seasonal COP as high as 3.69, is extremely favorable.

- In areas offering time-of-day rates, cost savings in the residential sector could be greatly enhanced by utilizing a high side storage system (hot tank). This configuration would reduce significantly on-peak utilization during the heating mode.
- Generally, the commercial ACES are economically superior to conventional systems at today's prices.
- The cost optimized commercial systems are economically superior to all conventional systems as well as to the Full and Minimum ACES at today's prices.

6.2 RECOMMENDATIONS

- Parasitic power losses significantly degrade seasonal COPs. Additional effort is needed on system-level integration to reduce these losses.
- Reduced cost techniques, such as open cycle excess ice disposal, should be investigated.
- Off-peak cold storage yields considerable summer dollar savings which are diminished by on-peak usage during the heating season. High side storage (hot tank) should therefore be investigated.
- The commercial ACES/cost optimized system concept should be commercialized and marketed.
- Marketing of the residential ACES should be initiated:
  - Penetrate the marketplace with viable ACES concepts.
  - Initiate government funded R & D for the first few ACES (1-20).
  - Utilize ACES in demonstration projects such as Federal building programs. This should include 50 to 500 ACES installations.
  - Sell the next 5,000 to 10,000 ACES as load shed devices. This would allow the concept to be initiated into the private sector.

6.3 ATTRACTIVE ACES OPTIONS NOT COVERED TO DATE

- High side storage.
- Open system ice disposal.
- Coupling of ground storage as a heat source.
- Heat driven/or engine driven ice-maker heat pump.

6.4 STUDY LIMITATIONS

It is the nature of scientific investigation to raise more questions than it answers. As a consequence, any answer arising from such investigation will usually be qualified by the identification of several other lines of inquiry. This economic evaluation was no exception. (Several areas outside the scope and resources of this study were identified as having a potentially large positive effect on the economic attractiveness of the ACES). These were:

- Increased Volumetric Packing Fraction--This study used a packing factor of 0.4 (the maximum currently achievable with an ice plate system). However, an in-bin type of ice-producing system would make it possible to increase the packing factor to 0.8 which would decrease the required tank size by a factor of two. Since the tank is a very costly component of the ACES, an increased packing factor will result in a dramatically decreased system cost.

- Level of Tank Insulation--Only two values of tank insulation were used in this study. They were sufficient to indicate whether a tank should have insulation or not but were not sufficient to determine the economically optimal level of insulation.

- Ice Tank Cost--This study used the costs associated with a reinforced concrete tank. The tank material was chosen because of its structural integrity over a wide range of tank sizes. Residential applications of ACES would be more attractive if a lower cost tank construction could be employed.
- Base Price of Fuels--As was shown in the sensitivity analysis, the economic viability of the ACES is extremely sensitive to the price of fuel. Only one scenario was used for this study. In the event of another oil embargo or the unavialability of natural gas, the resultant increase in price could make the ACES "instantly" cost-competitive with conventional systems.

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